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PART - A

2 Marks Questions

1.
 - (i) Specify function of element
Ex. :- Bearing, Keys
 - (ii) Determine forces acting on element
Ex :- Inertia, Friction, Torque
 - (iii) Select suitable material for element
 - (iv) Determine failure modes
 - (v) Determine geometric dimensions
 - (vi) Modify dimensions for assembly
 - (vii) Prepare working drawing of element
2.

Sol. Standardization is defined as obligatory norms to which various characteristics of a product should conform. The characteristics include materials, dimensions, shapes of the component, method of the product. The following standards are used in mechanical engineering design

 - (i) Standards for materials their chemical compositions, mechanical properties and heat treatment
 - (ii) Standards for shapes and dimensions of commonly used mechanical elements
 - (iii) Standard for fits, tolerance and surface finish of component
 - (iv) Standard of testing of products
 - (v) Standard for engineering drawing of components
3.

Sol. **Following standards used in designing of machine component-**

 - (i) Company standards
 - (ii) National standards
 - (iii) International standards
4.

Sol.

 - (i) Availability
 - (ii) Cost
 - (iii) Mechanical properties
 - (iv) Manufacturing considerations
5.

Sol.

 - (i) Failure by elastic deflection
 - (ii) Failure by general yielding
 - (iii) Failure by fracture
6.

Sol.

A static load is defined as a force, which is gradually applied to a mechanical component and which does not change its magnitude or direction with respect to time
7.

Sol.

While designing a component it is necessary to provide sufficient reserve strength in case of an accident.
This is achieved by taking a suitable factor of safety

$$N \text{ or } f_s = \frac{\text{failure stress}}{\text{allowable stress}} = \frac{\text{failure load}}{\text{working load}}$$



8.

Sol.

The allowable stress is the stress value, which is used in design to determine the dimensions of the component

9.

Sol.

There are number of factors which are difficult to evaluate accurately in design analysis

- (i) uncertainty in the magnitude of external force acting on component
- (ii) variations in the properties of materials like yield strength or ultimate strength
- (iii) Variation in the dimensions of the component due to imperfect
- (iv) There are lots of assumptions made while design analysis in order to simplify the calculations may not be exactly valid in working conditions

10.

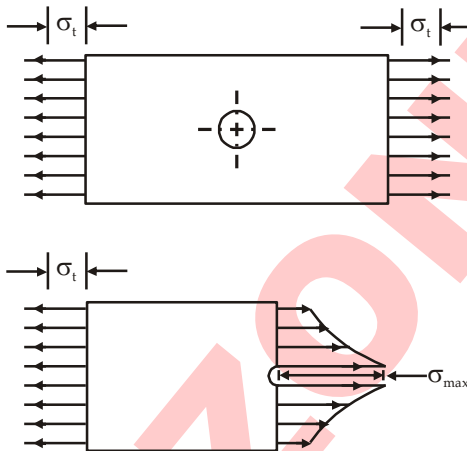
Sol. **Magnitude of FOS depends upon following factors:**

- (i) Effect of failure
- (ii) Type of load
- (iii) Degree of accuracy in force analysis
- (iv) Material of component
- (v) Reliability of component
- (vi) Cost of component
- (vii) Testing of machine element
- (viii) Service conditions
- (ix) Quality of manufacture

11.

Sol.

It is defined as the localization of high stress due to irregularities present in the component and abrupt changes of the cross-section



Highest value of actual stress
near discontinuity

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

12.

Sol.

- (i) variation in properties of materials
- (ii) load application
- (iii) abrupt change in section
- (iv) discontinuities in the component
- (v) machining scratches



13.

Sol.

There is no effect of stress concentration on ductile material under static load because of local plastic deformation but under fluctuating load the stress near the discontinuities may exceed the endurance limit.

Endurance limit of material greatly reduce by stress concentration

14.

Sol.

Effect of stress concentration is more severe in case of brittle material, due to their inability of plastic deformation

15.

Sol. Ductile material

There is no effect of stress concentration on ductile material under static load because of local plastic deformation but under fluctuating load the stress near the discontinuities may exceed the endurance limit.

Endurance limit of material greatly reduce by stress concentration

16.

Sol.

- (i) Additional notches and holes
- (ii) Fillet radius, undercutting and notch for member in bending
- (iii) Drilling additional holes for shafts
- (iv) Reducing shank diameter is threaded bolt

17.

Sol.

It is defined as time delayed fracture under cyclic loading.

Ex. Transmission shafts, connecting rods, gears, vehicle suspension system and ball bearings

It begins with crack at some point in the material

18.

Sol.

- (i) Number of cycle
- (ii) Stress concentration
- (iii) Mean stress
- (iv) Residual stresses
- (v) Stress amplitude
- (vi) Corrosion
- (vii) Creep

19.

Sol.

The fatigue or endurance limit of a material is defined as the maximum amplitude of completely reversed stress that a standard specimen can sustain for an infinite number of cycles without fatigue failure.

20.

Sol.

The fatigue life is defined as the number of cycles that a standard specimen can complete during the test before appearance of the first fatigue crack.

21.

Sol.

It is the graphical representation of stress amplitude (σ_a) versus the number of stress cycles (N) before the fatigue failure on a log-log graph paper

22.

Sol.

Any fatigue failure when the number of stress cycles are less than 1000 is called low cycle fatigue. Any fatigue failure when the number of stress cycles are more than 1000 is called high cycle fatigue

23.

Sol.

It is defined as the susceptibility of a material to succumb to the damaging effects of stress raising notches in fatigue loading

$$q = \frac{\text{Increase in actual stress over nominal stress}}{\text{Increase in theoretical stress over nominal stress}}$$

24.

Sol.

It means that effect of presence of notch highly affect the endurance strength of the components.

25.

Sol.

Soderberg Criterion

$$\text{Equation: } \frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_y} = 1$$

Basis: Uses yield strength (σ_y) for the mean stress limit.

Conservatism: Highly conservative, assumes failure occurs if the combination of stresses exceeds the yield strength, making it ideal for ductile materials where yielding defines failure.

26.

Sol.

Goodman Criterion

$$\text{Equation: } \frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_u} = 1$$

Basis: Uses ultimate tensile strength (σ_u) for the mean stress limit.

Conservatism: Less conservative than Soderberg but safer than the Gerber parabola; widely used for general applications as it fits test data well and is simple to calculate.

27.

Sol.

A parabollic curve joining S_e on the ordinate and S_{ut} on abscissa is called gerber parabolla

28.

Interchangeability can refer to:

- Interchangeable parts, the ability to select components for assembly at random and fit them together within proper tolerances.
- Interchangeability (computer science), the ability that an object can be replaced by another object without affecting code using the object.

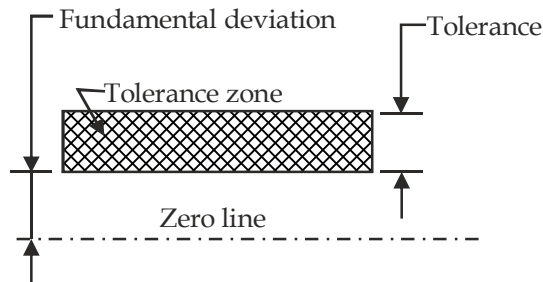
29.

- Interchangeability is a characteristic of components and subassemblies that go into modern manufactured products.
- The basic principle is that every unit made should be exactly like every other unit produced.
- All units made must conform to design specifications within a narrowly identified allowable range of variation.

30.

Sol.

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



31.

Sol. Maximum Clearance = LMC (hole) - LMC (shaft)
 $= 30.2 - 29.8 = 0.4 \text{ mm}$
 Allowance = MMC (hole) - MMC (shaft)
 $= 29.6 - 30.2 = -0.6$ (-ve value indicates interference)

This is a type of transition fit because interference and clearance may occur during assemble.

32.

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

Symbol	Surface roughness(Ra) in microns
∇	8 to 25
$\nabla \nabla$	1.6 to 8
$\nabla \nabla \nabla$	0.025 to 1.6
$\nabla \nabla \nabla \nabla$	Less than 0.025

33.

Sol. Tolerance grade for the hole is 6 and for the shaft is 5 with a nominal size of 100 mm.

34.

Sol. **Significance of '6':** This means deep groove ball bearing (the number used for roller bearing is 3)
Significance of '1': This signifies the series of the bearing. Here '1' indicates extra light series ('2'- light series, '3'- medium series, '4'- heavy series).
Significance of '08': The multiplication of 08 with 5 gives the bore diameter of the bearing in mm i.e. $5 \times 08 = 40 \text{ mm}$.

35.

Sol.

- (i) It is safe from design consideration
- (ii) Equation of straight line is simple as compared to parabollic equation
- (iii) Rough sketch is enough to construct fatigue diagram

36.

Sol.

A power screw is a mechanical device used for converting rotary motion into linear motion and transmitting power

37.

Sol.

When $\phi < \alpha$ (α = helix angle, ϕ = friction angle) torque require is negative. The load itself will begin to turn the screw and decend down, unless a restraining torque is applied. This condition is called overhauling of the screw.

38.

Sol.

When $\phi \geq \alpha$ a positive torque is required to lower the load. Under this condition, the load will not turn the screw and will not descend on its own unless an effort P is applied in this case screw is called self locking

39.

Sol.

Threaded joint is defined as a separable joint of two or machine parts that are held together by means of threaded fastening such as bolt

40.

Sol.

- (i) Through bolts
- (ii) Tap bolt
- (iii) Studs

41.

Sol.

Bolts subjected to shock and impact loads in certain applications. In these cases bolt of uniform strength is an important design consideration to prevent breakage at the threads

42.

Sol.

- (i) Make the shank diameter equal to core diameter of thread
- (ii) Drill a hole in shank part

43.

Sol.

A brake is defined as a mechanical device, which is used to absorb the energy possessed by a moving system or mechanism by means of friction

44.

Sol.

- (i) Mechanical brakes
- (ii) Electrical brakes
- (iii) Hydraulic brakes

45.

Sol.

The primary purpose of brake is to slow down or completely stop the motion of a moving system such as rotating drum, vehicle or machine.

46.

Sol.

In the form of heat energy absorbed by brakes can be dissipated

47.

Sol.

Bearing is a mechanical element that permits relative motion between two parts such as shaft and the housing with minimum friction

48.

Sol.

It is also called sleeve bearing. In this case the surface of the shaft slides over the surfaces of the bush resulting in friction and wear. There is a lubricating oil to separate bush and shaft.

49.

Sol.

- (i) Crankshaft bearing in steam engine (locomotive)
- (ii) Centrifugal pumps
- (iii) Large size electric motors
- (iv) Steam and gas turbines
- (v) concrete mixers, marine installations

50.

Sol.

For starting conditions and at moderate speeds, the frictional losses in rolling contact bearing are lower than that of equivalent hydrodynamic journal bearing that is why they are called as antifriction bearings

51.

Sol.

- (a) Inner race
- (b) Rolling element (ball or roller)
- (c) Outer race
- (d) Cage

52.

Sol.

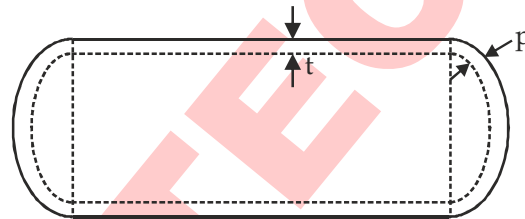
It is used to holds the rolling element together and spaces them evenly around the periphery of the shaft.

53.

Sol.

$$\text{Hoop stress in cylinder} = \frac{pd}{2t}$$

$$\text{Hoop stress in hemispherical portion} = \frac{pd}{4t'}$$



$$\text{For maximum stress to be equal, } \frac{pd}{2t} = \frac{pd}{4t'}$$

$$\therefore \frac{t'}{t} = \frac{1}{2}$$

or

$$\frac{t_{\text{spherical}}}{t_{\text{cylindrical}}} = \frac{1}{2}$$

54.

Sol.

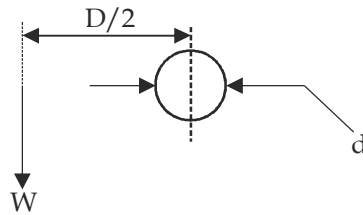
- leverage refers to the mechanical advantage gained by using a simple machine like a lever to amplify force or motion.
- It allows a smaller input force to move a larger load or achieve a greater output distance, effectively multiplying the force or distance applied.
- This is achieved by strategically positioning the fulcrum (pivot point) and applying force and load at specific points on the lever.

55.

Shaft	Axle
Shafts are rotating members which are subjected to bending moments, twisting moments and axial loads.	Axles are rotating or non rotating members which are subjected to only bending moments and axial load due to members supported by it.
It does transmit torque or power.	It does not transmit torque or power.
It does bend and twist.	It does not twist, it only bends.

56.

$$\text{Sol. Direct shear stress on the cross-section of spring wire} = \frac{W}{\frac{\pi}{4}d^2} \text{ (downward)}$$



In inner radius of coil $\tau = \frac{16 \left(W \frac{D}{2} \right)}{\pi d^3}$ (downward)

In outer radius of coil $\tau = \frac{16 \left(W \frac{D}{2} \right)}{\pi d^3}$ (upward)

Hence at inner coils radius direct stress adds up to shear stress and at outer radius it is subtracted. So stress at inner coil radius is more.

57.

Sol. Given data: $N_A = 9$; $N_B = 8$; $\delta_A = 8\text{mm}$; $\delta_B = 3\text{mm}$; $D_A = 80\text{mm}$; $D_B = ?$

$$\delta_A = \frac{8PD_A^3 N_A}{Gd^4} \quad \text{and} \quad \delta_B = \frac{8PD_B^3 N_B}{Gd^4}$$

$$\frac{\delta_B}{\delta_A} = \left(\frac{D_B}{D_A} \right)^3 \frac{N_B}{N_A}$$

or $\frac{3}{8} \times \frac{9}{8} = \left(\frac{D_B}{D_A} \right)^3$

or $\left(\frac{D_B}{D_A} \right)^3 = \frac{3^3}{4^3}$

or $\frac{D_B}{D_A} = \frac{3}{4}$

$\therefore D_B = 0.75 \times 80 = 60\text{mm}$

58.

Sol. Comparison between involute and cycloidal gears:

S. No.	Cycloidal Teeth	Involute Teeth
1	Pressure angle varies from maximum at the beginning of engagement, reduces to zero at the pitch point and again increases to maximum at the end of engagement resulting in less smooth running of the gears.	Pressure angle is constant throughout the engagement of teeth. This results in smooth running of the gears.
2	It involves double curve for the teeth, epicycloid and hypocycloid. This complicates the manufacture.	It involves single curve for the teeth resulting in simplicity of manufacturing and of tools.
3	Owing to difficulty of manufacture, these are costlier.	These are simple to manufacture and thus are cheaper.
4	Exact centre-distance is required to transmit a constant velocity ratio.	A little variation in the centre distance does not affect the velocity ratio.
5	Phenomenon of interference does not occur at all.	Interference can occur if the condition of minimum number of teeth on a gear is not followed.
6	The teeth have spreading flanks and thus are stronger.	The teeth have radial flanks and thus are weaker as compared to the cycloidal form for the same pitch.
7	In this, a convex flank always has contact with a concave face resulting in less wear.	Two convex surfaces are in contact and thus there is more wear.

PART - B

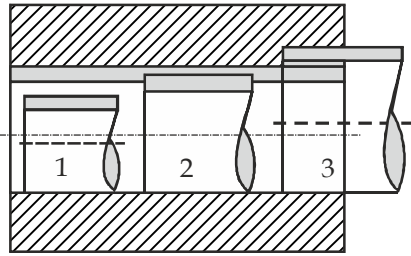
5 Marks Questions

59.

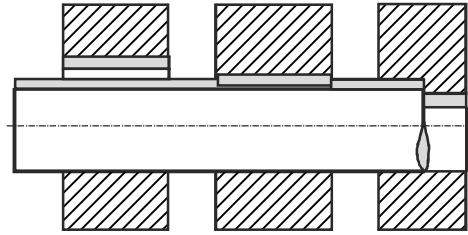
The following are two bases of limit system :

- Hole basis system:** When the hole is kept as a constant member and different fits are obtained by varying the shaft size, then the limit system is said to be on a hole basis.
- Shaft basis system:** When the shaft is kept as a constant member and different fits are obtained by varying the hole size, then the limit system is said to be on a shaft basis.





Hole Basis System



Shaft Basis System

Out of these two bases of limits systems, hole basis system is preferred over shaft basis system due to from manufacturing perspective it is easy to adjust the shaft dimension by using grinding or turning operation whereas holes are produced with drilling operations and it becomes complicated to do any further turning or grinding operations in holes.

60.

Aspect	Limit	Tolerance
Definition	A limit specifies the extreme boundaries or constraints for a specific dimension or parameter. It includes both the lower limit (LL) and upper limit (UL).	Tolerance defines the acceptable range of variation or deviation from the nominal dimension or parameter.
Purpose	Limits establish the absolute extremes that a dimension or parameter must not exceed. They denote the physical boundaries.	Tolerance ensures that even if the manufacturing process is not perfectly precise, the component will still function properly and fit within the intended assembly.
Range	It defines a specific range, encompassing the LL and UL values.	It delineates the allowable deviation from the nominal dimension, specifying how much a dimension can vary above or below the nominal value.
Example	A dimension with a limit might be specified as 10.0 mm to 10.2 mm, indicating that the dimension must fall within this exact range.	A dimension with tolerance might be specified as 10.0 mm $\pm$ 0.1 mm, indicating that the dimension can range from 9.9 mm to 10.1 mm.
Application	Limits are typically used when exact values or specific boundaries are critical for the functionality of a component.	Tolerances are commonly applied in various engineering and manufacturing contexts to accommodate variations during production and ensure parts fit and perform as intended.

61.

Sol. Tolerance:

- It is the difference between the upper limit and lower limit of a dimension.
- In other words, it is the maximum permissible variation in a dimension.
- The tolerance may be unilateral or bilateral.
- When all the tolerance is allowed on one side of the nominal size, e.g. $20^{+0.004}_{-0.000}$, then it is said to be unilateral system of tolerance.
- The unilateral system is mostly used in industries as it permits changing the tolerance value while still retaining the same allowance or type of fit.
- When the tolerance is allowed on both sides of the nominal size, e.g. $20^{+0.002}_{-0.002}$, then it is said to be bilateral system of tolerance.
- In this case +0.002 is the upper limit and -0.002 is the lower limit.

62.

Hole: 25.00 mm Shaft: 24.97 mm
25.02 mm 24.95 mm

Find the hole tolerance, shaft tolerance and allowance?

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

Hole tolerance:

We know that hole tolerance
 = Upper limit of hole - Lower limit of hole
 = 25.02 - 25 = 0.02 mm

Shaft tolerance:

We know that shaft tolerance
 = Upper limit of shaft - Lower limit of shaft
 = 24.97 - 24.95 = 0.02 mm

Allowance:

We know that allowance
 = Lower limit of hole - Upper limit of shaft
 = 25.00 - 24.97 = 0.03 mm

63.

Sol. Fit is a relationship that exists between two mating parts a hole and a shaft with respect to their dimensional difference before assembly.

There are three types of fits.

(a) Clearance fit:

- In this fit the size of the Hole is always greater than the size of the shaft. Clearance is the difference between the size of the hole and the size of the shaft which is always positive. Here the tolerance zone of the hole will be above the tolerance zone of the shaft.
- Examples: Slide fit, easy sliding fit, running fit, slack running fit and loose running fit.

(b) Interference fit:

- In this fit the size of the Hole is always less than the size of the shaft. Interference is the difference between the size of the hole and the size of the shaft which is always negative i.e. shaft is always larger than the hole size. Here the tolerance zone of the hole will be below the tolerance zone of the shaft.
- Examples: Shrink fit, heavy drive fit and light drive fit.

(c) Transition fit:

- It may sometimes provide clearance and sometimes interference. Here the tolerance zones of the hole and shaft will overlap each other.
- Examples: Tight fit, pushfit, wringing fit and press fit.



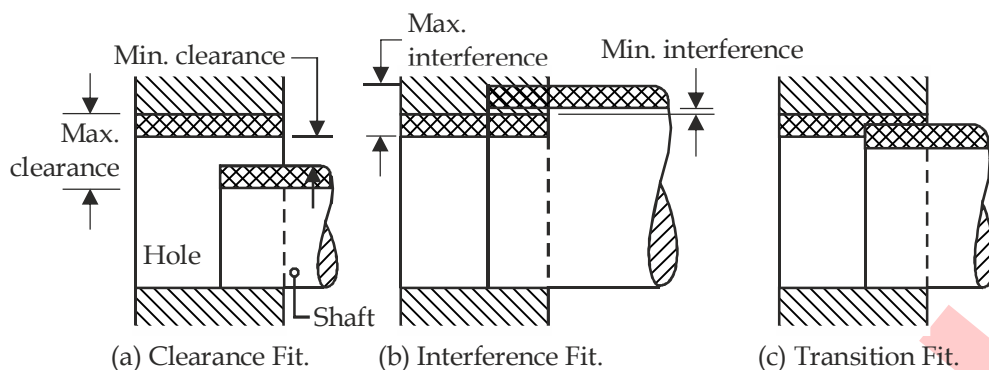


Fig.:- Types of Fits.

64.

Sol. Preferred Numbers:

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

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

- These ratios are approximately equal to 1.58, 1.26, 1.12 and 1.06.
- The series of preferred numbers are designated as R5, R10, R20 and R40 respectively. These four series are called basic series.
- The other series called derived series may be obtained by simply multiplying or dividing the basic sizes by 10, 100, etc.

65.

Sol. According to Indian standards, the system of limits and fits comprise 18 grades of fundamental tolerances which are designated as IT 01, IT 0, and IT 1 to IT 16. These are called standard tolerances. The standard tolerances for grades IT5 to IT7 are determined in terms of standard tolerance unit (i) in microns, where

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

The values of standard tolerances corresponding to grades IT01, IT0 and IT1 are as given below:

$$\text{For IT01, } i(\text{microns}) = 0.3 + 0.008 D,$$

$$\text{For IT0, } i(\text{microns}) = 0.5 + 0.012 D, \text{ and}$$

$$\text{and IT1, } i(\text{microns}) = 0.8 + 0.020 D,$$

where D is the size or diameter in mm.

66.

Sol. Given:

$$P = 3.5 \text{ MN} = 3.5 \times 10^6 \text{ N};$$

$$\sigma_t = 85 \text{ MPa} = 85 \text{ N/mm}^2;$$

$$E = 210 \text{ kN/mm}^2 = 210 \times 10^3 \text{ N/mm}^2;$$

$$l = 2.5 \text{ m} = 2.5 \times 10^3 \text{ mm}$$

1. Diameter of the rods

Let

$$d = \text{Diameter of the rods in mm.}$$

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

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

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

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

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

$$d^2 = 1.75 \times 10^6 / 66.76 = 26213$$

$$d = 162 \text{ mm}$$

2. Extension in each rod

Let δl = Extension in each rod.

We know that Young's modulus (E),

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

$$\therefore \delta l = 212.5 \times 10^3 / (210 \times 10^3) = 1.012 \text{ mm}$$

67.

Sol. Factors Affecting Factor of Safety (FoS):

Component design codes and standards typically prescribe a minimum factor of safety for the given component, but selecting the precise safety factor depends on various parameters, including:

- Material Type: Ductile vs. Brittle
- Loading Type: Static vs. Dynamic
- Presence of Cyclical Loads
- Intensity of Stress Concentration
- Unforeseen Misuse of the Component
- Accuracy and Complexity of Stress Calculations
- Environment and Design Temperature/Pressure
- Impact of Failure
- Cost of Component or Material
- Corrosion Rate
- Maintenance Frequency

68.

Sol. Working Stress:

When designing machine parts, it is desirable to keep the stress lower than the maximum or ultimate stress at which failure of the material takes place.

This stress is known as the working stress or design stress. It is also known as safe or allowable stress.

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

Factor of Safety:

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

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

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

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

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

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

This relation may also be used for ductile materials.

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



69.

Sol. Given : $P = 100 \text{ kW} = 100 \times 10^3 \text{ W}$;
 $N = 1600 \text{ r.p.m}$;
 $T_{\max} = 1.25T_{\text{mean}}$;
 $\tau = 70 \text{ MPa}$

Let T_{mean} = Mean torque transmitted by the shaft in N-m, and
 d = Diameter of the shaft in mm.

We know that the power transmitted (P),

$$100 \times 10^3 = \frac{2\pi N \cdot T_{\text{mean}}}{60} = \frac{2\pi \times 1600 \times T_{\text{mean}}}{60} = 16.76 T_{\text{mean}}$$

$$\therefore T_{\text{mean}} = 100 \times 10^3 / 16.76 = 5966.6 \text{ N-m}$$

and maximum torque transmitted,

$$T_{\max} = 1.25 \times 5966.6 = 7458 \text{ N-m} = 7458 \times 10^3 \text{ N-mm}$$

We know that maximum torque ($T_{\max}$),

$$7458 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 70 \times d^3 = 13.75 d^3$$

$$\therefore d^3 = 7458 \times 10^3 / 13.75 = 542.4 \times 10^3 \text{ or } d = 81.5 \text{ mm}$$

70.

Sol. Given, $r_2 = \frac{45}{2} = 22.5 \text{ cm}$

$$r_1 = \frac{20}{2} = 10 \text{ cm}$$

$$W = 39240 \text{ N}$$

As per question, we have to assume uniform wear theory,

$$\begin{aligned} T_f &= \frac{1}{2} \mu W (r_1 + r_2) \\ &= \frac{1}{2} \times 0.25 \times 39240 (22.5 + 10) \times 10^{-2} \end{aligned}$$

$$T_f = 1594 \text{ N.m}$$

Power absorbed in friction

$$\begin{aligned} &= \frac{2\pi N}{60} \times \frac{T_f}{1000} \text{ kW} \\ &= \frac{2 \times 3.14 \times 150}{60} \times \frac{1594}{1000} = 25 \text{ kW} \end{aligned}$$

71.

Sol. Given,
 Shaft diameter,

$$d = 40 \text{ mm} = 0.04 \text{ m}$$

$$\Rightarrow r = 20 \text{ mm} = 0.02 \text{ m}$$

$$\text{Length } L = 40 \text{ mm} = 0.04 \text{ m}$$

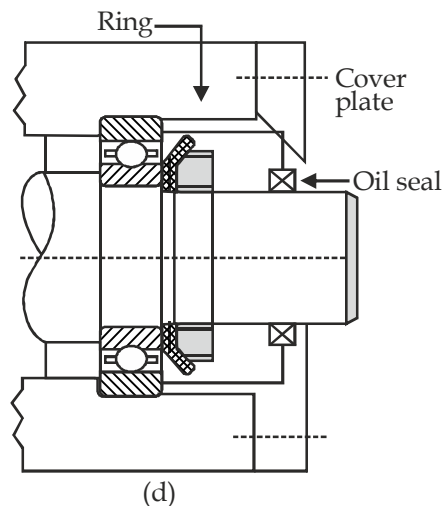
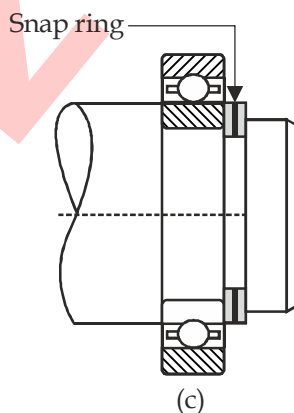
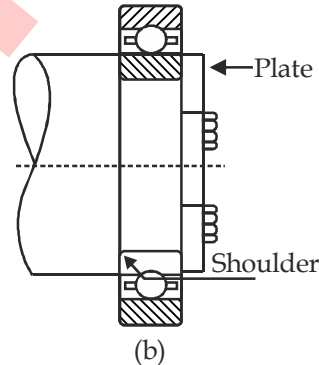
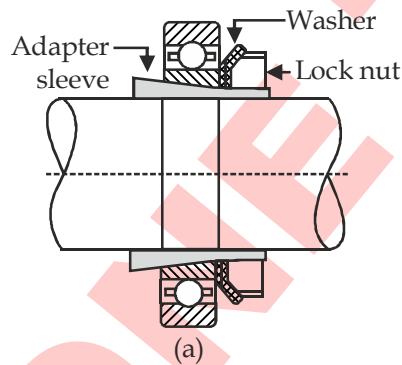
Rotational speed of shaft,

$$\begin{aligned}\omega &= 20 \text{ rad/s} \\ \text{Viscosity, } \mu &= 20 \text{ mPa.s} \\ \text{Clearance, } C &= 0.020 \text{ mm} \\ \text{Velocity at surface of bearing, } V &= r \times \omega = 20 \times 20 \times 10^{-3} \text{ m/s} = 0.04 \text{ m/s} \\ \tau &= \mu \frac{V}{C} = \frac{20 \times 10^{-3} \times 0.4}{0.02 \times 10^{-3}} = 400 \text{ N/m}^2 \\ \text{Force, } F &= \tau \times A = \tau \times \pi d l \\ \text{Torque} &= F \times r \\ &= 400 \times 3.14 \times 0.04 \times 0.04 \times 0.02 = 0.040 \text{ Nm}\end{aligned}$$

72.

Sol.

1. It is necessary to position inner and outer races axially by positive means. There are several methods such as providing shoulders for the shaft or the housing, locknut, snap ring or cover plates, as shown in the figure (a). The basic principle is to restrict the displacement of inner as well as outer race in axial direction by positive means.
2. Two methods of restricting the displacement of the inner race are illustrated in figure (b). In both the cases, the shaft is provided with a shoulder to restrict the displacement of the inner race to the left side
3. In the figure (c), a snap-ring is used in place of the plate.
4. In figure (d), the housing is provided with a shoulder to restrict the displacement of the outer race to the left side. A circular ring of the cover plate restricts the displacement to the right side.



73.

- Sol.** (a) Self aligning bearings are commonly employed where accurate alignment is not possible or not feasible. They permit minor angular misalignment of the shaft relative to the bearings.
- (b) In deep groove ball bearing, the radius of the ball is slightly less than the radius of curvature of the grooves in the races. Kinematically this gives a point of contact between the balls and the races. Therefore they may roll freely without any sliding and due to relatively large size of the balls, deep groove ball bearing has high load carrying capacity and suitability for running at high speeds even. Hence from the above reasons, we opt deep groove ball bearing to take a heavy axial load.
- (c) The thrust ball bearings are used for carrying thrust loads exclusively but at speeds below 2000 rpm. At high speeds, centrifugal force causes the balls to be forced out of the races.

74.

- Sol.** Fluid film lubrication naturally divides into two categories.

Thin-film lubrication: Thin-film lubrication is usually met with in counter-formal contacts, principally in rolling bearings and in gears. The thickness of the film in these contacts is of order of $1\text{ }\mu\text{m}$ or less, and the conditions are such that the pressure dependence of viscosity and the elastic deformation of the bounding surfaces must both be taken into account.

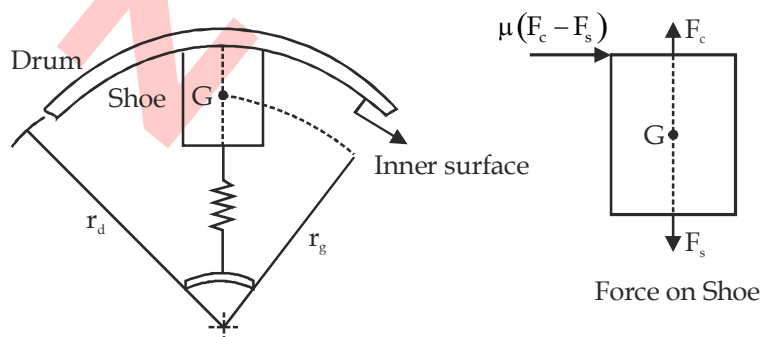
Thick-film lubrication: Thick-film lubrication is encountered in externally pressurized bearings, also called hydrostatic bearings, and in self-acting bearings, called hydrodynamic bearings. Mainly, there are two kinds: journal bearings and thrust bearings. The film thickness in these conformal-contact bearings is at least an order of magnitude larger than in counter-formal bearings. In consequence, the prevailing pressures are orders of magnitude smaller, so that neither the pressure dependence of viscosity nor the elastic deformation of the surfaces plays an important roles

75.

- Sol.** **Basic Static Load Rating:** The basic static radial load rating (C_0) applies to bearings which rotate at very slow speeds, subjected to very slow oscillations, or stationary under load. It should be considered when heavy shock loading occurs to a rotating bearing. The basic static radial load rating is defined as the static radial load which corresponds to a calculated contact stress at the center of the most heavily loaded ball/ raceway contact of 609000 PSI for all radial and angular contact ball bearings. For this contact stress, a total permanent deformation of rolling element and raceway occurs which is approximately 0.0001 of the rolling element diameter. Basic Static Load Rating for stainless steel is (75-80%) of that for standard bearing steel.

Basic Dynamic Load Rating: The basic dynamic load rating (C_r) of a bearing with rotating inner ring and stationary outer ring is that load of constant direction and magnitude which a sufficiently large sample group of apparently identical bearings can endure for a basic rating life of one million revolutions.

76.



Sol.

The centrifugal clutch works on the principle of centrifugal force. As the angular speed of input shaft increases, the centrifugal force acting on the sliding shoes increases, causing the shoes to move in a radially outward direction. The shoes continue to move with increasing speed until they contact

inner surface of drum. Power is transmitted due to frictional force between shoe lining and the inner surface of drum.

Application: Lawn movers, golf cart, cranes, cement mills, three wheelers.

77.

Sol. Assumptions made in Lewis equation:

1. Effect of radial load (i.e. axial compressive stresses) are neglected.
2. Tangential load is assumed to be uniformly distributed over the face width of gear tooth.
3. Contact ratio is assumed as 1.
4. Effect of stress concentration at the root of gear tooth is neglected.
5. Manufacturing errors (i.e. errors in tooth spacing & inaccuracies in tooth profiles) are neglected.
6. Each gear tooth is considered as cantilever beam.

78.

Sol. Factors to be Considered while Designing Machine Parts to Avoid Fatigue Failure

The following factors should be considered while designing machine parts to avoid fatigue failure:

1. The variation in the size of the component should be as gradual as possible.
2. The holes, notches and other stress raisers should be avoided.
3. The proper stress de-concentrators such as fillets and notches should be provided wherever necessary.
4. The parts should be protected from corrosive atmosphere.
5. A smooth finish of outer surface of the component increases the fatigue life.
6. The material with high fatigue strength should be selected.
7. The residual compressive stresses over the parts surface increases its fatigue strength.

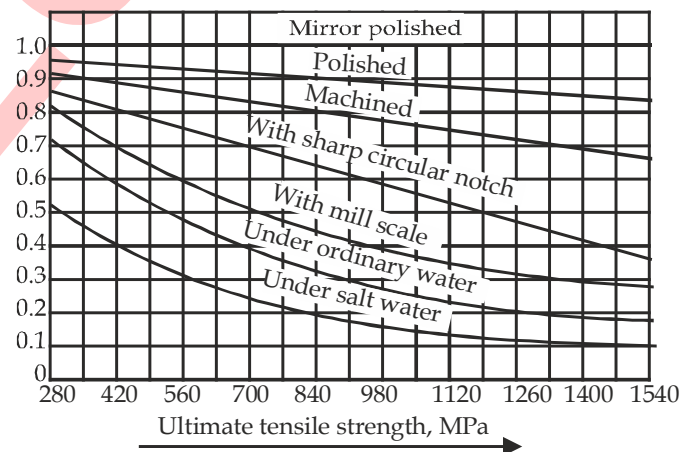
79.

- Sol.**
1. The stresses which vary from a minimum value to a maximum value of the same nature, (i.e. tensile or compressive) are called **fluctuating stresses**.
 2. The stresses which vary from zero to a certain maximum value are called **repeated stresses**.
 3. The stresses which vary from a minimum value to a maximum value of the opposite nature (i.e. from a certain minimum compressive to a certain maximum tensile or from a minimum tensile to a maximum compressive) are called **alternating stresses**.

80.

Sol. Effect of Surface Finish on Endurance Limit- (Surface Finish Factor):

When a machine member is subjected to variable loads, the endurance limit of the material for that member depends upon the surface conditions. Fig. shows the values of surface finish factor for the various surface conditions and ultimate tensile strength.



When the surface finish factor is known, then the endurance limit for the material of the machine member may be obtained by multiplying the endurance limit and the surface finish factor.

We see that for a mirror polished material, the surface finish factor is unity. In other words, the endurance limit for mirror polished material is maximum and it goes on reducing due to surface condition.

Let K_{sur} = Surface finish factor.

∴ Endurance limit,

$$\begin{aligned}\sigma_{el} &= \sigma_{eb} \cdot K_{sur} = \sigma_e \cdot K_b \cdot K_{sur} = \sigma_e \cdot K_{sur} && \dots (\because K_b = 1) \\ &&& \dots \text{ (For reversed bending load)} \\ &= \sigma_{ea} \cdot K_{sur} = \sigma_e \cdot K_a \cdot K_{sur} && \dots \text{ (For reversed axial load)} \\ &= \tau_e \cdot K_{sur} = \sigma_e \cdot K_s \cdot K_{sur} && \dots \text{ (For reversed torsional or shear load)}\end{aligned}$$

Note: The surface finish factor for non-ferrous metals may be taken as unity.

81.

Sol. The 3 Stages of Creep Failure

Creep failure is the slow buckling and deformation of a material under constant load.

Creep is a time-dependent deformation and one that can be hard to spot in the early stages. The best way to protect against creep failure is to know the key stages and implement the correct testing processes to identify each one in every new material application.

How does creep failure occur?

Creep failure results in the permanent deformation of the material and runs the risk of compromising the integrity of the product. It affects a wide range of materials, including metals and polymers, and tends to happen most frequently at high temperatures.

Understanding the 3 stages of creep failure

Here are the three distinct phases of creep failure and their scientific definition.

Primary creep

Also sometimes referred to as transient creep, primary creep describes the initial stage. The material experiences an increase in strain hardening and creep resistance – where it responds to pressure while attempting to hold its shape.

Secondary creep

In the secondary stage, the material experiences a repeating process of strain hardening and recovery. The creep rate is constant and this stage, also known as steady-state creep, can last for a significant portion of time.

Tertiary creep

The third and final stage of creep takes place at an accelerated pace as the material rapidly moves towards deformation and failure.

82.

Sol. Creep is the time-dependent deformation of a material under a constant applied stress, occurring at elevated temperatures.

Factors that increase creep rate include:

1. Increased Temperature:

Higher temperatures generally lead to a higher creep rate as atomic mobility increases.

2. Increased Stress:

A higher applied stress will typically result in a faster creep rate.

3. Time:

Creep is inherently a time-dependent phenomenon, with deformation accumulating over time under sustained stress.

4. Material Properties:

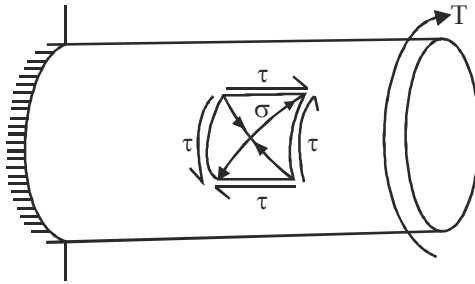
The specific material's composition, microstructure, and grain size significantly influence its resistance to creep.

5. Environment:

Corrosive environments can sometimes accelerate creep, especially at high temperatures.

83.

- Sol.** Only a single strain gauge is mounted on the surface. This means it will be along maximum normal stress because strain gauges are sensitive to only normal strain and not shear strain.



$$\therefore \text{Normal strain at the surface, } \epsilon = 3.98 \times 10^{-4}$$

$$E = 105 \text{ GN/m}^2$$

$$\epsilon_1 = \frac{\sigma_1}{E} - \frac{\nu \sigma_2}{E}$$

For pure shear case $\sigma_1 = \tau$, $\sigma_2 = -\tau$

$$3.98 \times 10^{-4} = \frac{\tau(1+0.3)}{105 \times 10^9}$$

$$\tau = 32.14 \text{ MPa}$$

From torsion equation we have

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

$$\frac{T}{\frac{\pi}{32} \times (60)^4 \times 10^{-12}} = \frac{32.14 \times 10^6}{30 \times 10^{-3}}$$

$$T = 1363.10 \text{ N-m}$$

$$P = \frac{2\pi NT}{60} = \frac{2\pi \times 800 \times 1363.10}{60} = 114.19 \text{ kW}$$

84.

- Sol.** Assuming diameter of solid shaft is d_s and external diameter of hollow shaft is D_o . Where subscript 1 is used for solid shaft and subscript 2 is used for hollow shaft.

Given, Torque, $T_1 = T_2$; Length, $L_1 = L_2$; Maximum shear stress, $\tau_1 = \tau_2$

$$\frac{W_s}{W_H} = ?$$

where W_s is weight of solid shaft and W_H is weight of hollow shaft.

We know that,
$$\tau = \frac{T \times r}{J}$$

where J is polar moment of inertia.

r is distance from centre.

Given,

$$\tau_1 = \tau_2$$

$$\frac{T_1 \times d_s}{2 \times \frac{\pi}{32} d_s^4} = \frac{T_2 \times D_0}{2 \times \frac{\pi}{32} \left[D_0^4 - \left(\frac{2}{3} D_0 \right)^4 \right]}$$

Given,

$$T_1 = T_2$$

So,

$$\frac{16}{\pi d_s^3} = \frac{16}{\pi D_0^3 \left[1 - \left(\frac{2}{3} \right)^4 \right]}$$

$$D_0^3 \left[1 - \frac{16}{81} \right] = d_s^3$$

$$\left(\frac{65}{81} \right) D_0^3 = d_s^3$$

$$D_0 = \left(\frac{81}{65} \right)^{1/3} d_s = 1.07611 d_s$$

$$d_s = 0.929272 D_0$$

$$\text{Weight of solid shaft} = \frac{\pi}{4} d_s^2 \times L_1 \times \rho_1$$

$$\text{Weight of hollow shaft} = \frac{\pi}{4} \left[D_0^2 - \left(\frac{2}{3} D_0 \right)^2 \right] \times L_2 \times \rho_2$$

Given, Length of shaft is same and shafts are made of same material.

$$\frac{W_s}{W_H} = \frac{\frac{\pi}{4} d_s^2 \times L_1 \times \rho_1}{\frac{\pi}{4} D_0^2 \left[1 - \frac{4}{9} \right] \times L_2 \times \rho_2} \quad [\because L_1 = L_2, \rho_1 = \rho_2]$$

$$= \frac{9 \times d_s^2}{5 D_0^2} = \frac{1.8 \times (0.929272)^2 D_0^2}{D_0^2}$$

$$\frac{W_s}{W_H} = 1.5544$$

85.

Sol.

Given :

$$D = 50 \text{ mm};$$

$$d = 5 \text{ mm};$$

$$n = 20;$$

$$W = 500 \text{ N}$$

We know that the spring index,

$$C = \frac{D}{d} = \frac{50}{5} = 10$$

$\therefore$ Shear stress factor,

$$K_s = 1 + \frac{1}{2C} = 1 + \frac{1}{2 \times 10} = 1.05$$

and maximum shear stress (neglecting the effect of wire curvature),

$$\begin{aligned}\tau &= K_s \times \frac{8W.D}{\pi d^3} = 1.05 \times \frac{8 \times 500 \times 50}{\pi \times 5^3} = 534.7 \text{ N/mm}^2 \\ &= 534.7 \text{ MPa}\end{aligned}$$

86.

Sol. Given: $n = 12$; $n_1 = 5$; $n_2 = 7$

We know that the deflection of the spring,

$$\delta = \frac{8W.D^3.n}{G.d^4} \text{ or } \frac{W}{\delta} = \frac{G.d^4}{8D^3.n}$$

Since G , D and d are constant, therefore substituting

$$\frac{G.d^4}{8D^3} = X, \text{ a constant, we have } \frac{W}{\delta} = k = \frac{X}{n}$$

or $X = k.n = 12k$

The spring is cut into two springs with $n_1 = 5$ and $n_2 = 7$.

Let k_1 = Stiffness of spring having 5 turns, and
 k_2 = Stiffness of spring having 7 turns.

$$\therefore k_1 = \frac{X}{n_1} = \frac{12k}{5} = 2.4k$$

$$\text{and } k_2 = \frac{X}{n_2} = \frac{12k}{7} = 1.7k$$

PART - C

20 Marks Questions

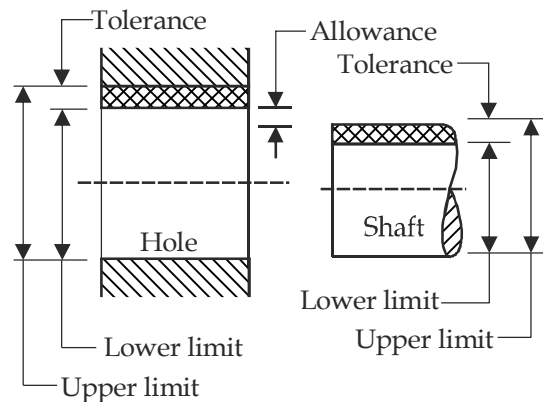
87.

Sol. In order to control the size of finished part with due allowance for error for interchangeable parts is called limit system. The various important terms used in limit system (or interchangeable system) are as follows:

- 1. Nominal size-** It is the size of a part specified in the drawing as a matter of convenience.
- 2. Basic size-** It is the size of a part to which all limits of variation (i.e. tolerances) are applied to arrive at final dimensioning of the mating parts. The nominal or basic size of a part is often the same.
- 3. Actual size-** It is the actual measured dimension of the part.
- 4. Limits of sizes-** There are two extreme permissible sizes for a dimension as shown in Fig. The largest permissible size for a dimension is called upper or high limit whereas the smallest size is known as lower limit.
- 5. Allowance-** It is the difference between the basic dimensions of the mating parts.
- 6. Tolerance-** It is the difference between the upper limit and lower limit of a dimension. In other words, it is the maximum permissible variation in a dimension. The tolerance may be unilateral or bilateral. When all the tolerance is allowed on one side of the nominal size, e.g. $20^{+0.004}_{-0.000}$, then it is said to be unilateral system of tolerance. The unilateral system is mostly used in industries as it

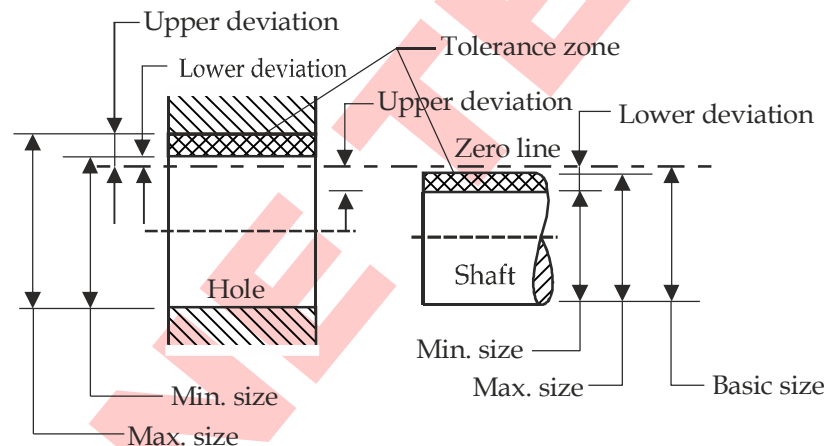


permits changing the tolerance value while still retaining the same allowance or type of fit.



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

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



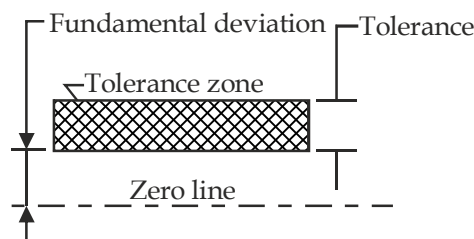
8. **Zero line-** It is a straight line corresponding to the basic size. The deviations are measured from this line. The positive and negative deviations are shown above and below the zero line respectively.

9. **Upper deviation-** It is the algebraic difference between the maximum limit and the basic size.

10. **Lower deviation-** It is the algebraic difference between the minimum limit and the basic size.

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

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



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

88.

Sol. Given :

$$d = 600 \text{ mm} = 0.6 \text{ m}$$

$$N = 200 \text{ r.p.m.}$$

$$\mu = 0.25$$

$$\theta = 160^\circ = 160 \times \frac{\pi}{180} = 2.793 \text{ rad}$$

$$T_1 = 2500 \text{ N}$$

We know that velocity of the belt,

$$v = \frac{\pi d N}{60} = \frac{\pi \times 0.6 \times 200}{60} = 6.284 \text{ m/s}$$

Let

T_2 = Tension in the slack side of the belt.

We know that $2.3 \log \left(\frac{T_1}{T_2} \right) = \mu \cdot \theta = 0.25 \times 2.793 = 0.6982$

$$\log \left(\frac{T_1}{T_2} \right) = \frac{0.6982}{2.3} = 0.3036$$

$$\therefore \frac{T_1}{T_2} = 2.01 \quad \dots (\text{Taking antilog of } 0.3036)$$

and $T_2 = \frac{T_1}{2.01} = \frac{2500}{2.01} = 1244 \text{ N}$

We know that power transmitted by the belt,

$$P = (T_1 - T_2)v$$

$$P = (2500 - 1244)6.284 = 7890 \text{ W}$$

$$P = 7.89 \text{ kW}$$

89.

Sol. **Design Considerations of disc clutch :**

1. The contact surface should develop a frictional force that may pick-up and hold the load with reasonable low pressure between the contact surface.
2. The heat of friction should be rapidly dissipated and tendency to grab should be minimum.
3. the surface should be backed by a material stiff enough to ensure a reasonably uniform distribution of pressure.

Torque capacity : Consider there is no slipping takes place-

Area of contact of friction surface = $2\pi r dr$

Normal or axial force on the ring $\delta w = P \times (2\pi r) dr$

Friction force $F_r = \mu \delta w = \mu P (2\pi r) dr$

Friction torque $T_r = F_r \times r = \mu P (2\pi r^2) dr$

We shall now consider two cases

1. Uniform pressure
2. Uniform wear
1. **Uniform pressure**



$$\text{Intensity of pressure } P = \frac{W}{\pi(r_1^2 - r_2^2)}$$

$$\begin{cases} P = \text{Pressure} \\ r = \text{Radius} \\ \mu = \text{Coefficient of friction} \\ W = \text{Load} \end{cases}$$

Frictional torque on elementary spring

$$T_r = 2\pi\mu Pr^2 dr$$

Total friction torque

$$T = \int_1^2 2\pi\mu Pr^2 dr = 2\pi\mu P \left[\frac{r_1^3 - r_2^3}{3} \right]$$

$$T = 2\pi\mu \frac{W}{(r_1^2 - r_2^2)\pi} \times \left(\frac{r_1^3 - r_2^3}{3} \right)$$

$$T = \frac{2}{3} \mu W \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right] = \mu WR$$

R = Mean radius of friction surface

$$= \frac{2}{3} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right]$$

2. **Uniform wear** : If P is the intensity of pressure and r is the distance from the axis of clutch, the P varies inversely with the distance.

$$P \propto \frac{1}{r}$$

$$P = \frac{c}{r} \Rightarrow Pr = c \quad \dots(i)$$

Normal force on ring

$$\begin{aligned} \delta W &= P \cdot 2\pi r dr = \frac{c}{r} 2\pi r dr \\ &= 2\pi c dr \end{aligned}$$

Total force on friction surface

$$W = \int_1^2 2\pi c dr = 2\pi c(r_1 - r_2)$$

$$c = \frac{W}{2\pi(r_1 - r_2)} \quad \dots(ii)$$

Frictional torque acting on ring

$$\begin{aligned} T_r &= 2\pi\mu Pr^2 dr = 2\pi\mu \frac{c}{r} r^2 dr \\ &= 2\pi\mu c r dr \end{aligned}$$

Total frictional torque

$$\begin{aligned} T &= \int_1^2 2\pi\mu c r dr \\ &= 2\pi\mu c \left[\frac{r_1^2 - r_2^2}{2} \right] = \pi\mu c (r_1^2 - r_2^2) \end{aligned}$$

$$T = \pi\mu \frac{W}{2\pi(r_1 - r_2)}(r_1^2 - r_2^2)$$

$$T = \frac{1}{2} \mu W(r_1 + r_2) = \mu WR$$

R = Mean radius of friction surface

$$= \frac{r_1 + r_2}{2}$$

90.

Sol. Given :

$$P = 300 \text{ kW} = 300 \times 10^3 \text{ W}$$

$$N = 90 \text{ r.p.m}$$

$$C_E = 0.1$$

$$k = 2 \text{ m}$$

We know that the mean angular speed,

$$\omega = \frac{2\pi N}{60} = 2\pi \times \frac{90}{60} = 9.426 \text{ rad/s}$$

Let ω_1 and ω_2 = Maximum and minimum speeds respectively

Since the fluctuation of speed is $\pm 0.5\%$ of mean speed, therefore total fluctuation of speed,

$$\omega_1 - \omega_2 = 1\% \omega = 0.01 \omega$$

and coefficient of fluctuation of speed,

$$C_s = \frac{\omega_1 - \omega_2}{\omega} = 0.01$$

We know that work done per cycle

$$= P \times \frac{60}{N} = 300 \times 10^3 \times \frac{60}{90} = 200 \times 10^3 \text{ N-m}$$

$\therefore$ Maximum fluctuation of energy,

$$\begin{aligned} \Delta E &= \text{Work done per cycle} \times C_E \\ &= 200 \times 10^3 \times 0.1 \\ &= 20 \times 10^3 \text{ N-m} \end{aligned}$$

Let m = Mass of the flywheel

We know that maximum fluctuation of energy (ΔE),

$$\begin{aligned} 20 \times 10^3 &= m \cdot k^2 \cdot \omega^2 C_s \\ &= m \times 2^2 \times (9.426)^2 \times 0.01 \\ &= 3.554 m \end{aligned}$$

$$\therefore m = 20 \times \frac{10^3}{3.554} = 5630 \text{ kg}$$

91.

Sol. Given :

$$P = 45 \text{ kW} = 45 \times 10^3 \text{ W}$$

$$N = 1000 \text{ rpm}$$

$$\alpha = 12.5^\circ$$

$$D = 500 \text{ mm}$$

$$R = 250 \text{ mm}$$

$$\mu = 0.2$$

or

$$p_n = 0.1 \text{ N/mm}^2$$

(i) The face width required

Let b = Face width of the clutch in mm
We know that torque developed by the clutch

$$T = \frac{P \times 60}{2\pi N} = \frac{45 \times 10^3 \times 60}{2\pi \times 1000}$$

$$= 430 \text{ N-m} = 430 \times 10^3 \text{ N-mm}$$

We also know that torque developed by the clutch (T),

$$430 \times 10^3 = 2\pi \cdot \mu \cdot p_n \cdot R^2 \cdot b = 2\pi \times 0.2 \times 0.1(250)^2 b = 7855 b$$

$$\therefore b = \frac{430 \times 10^3}{7855} = 54.7 \approx 55 \text{ mm}$$

(ii) The axial spring force necessary to engage the clutch

We know that the normal force acting on the contact surfaces.

$$W_n = p_n \times 2\pi R \cdot b = 0.1 \times 2\pi \times 250 \times 55 = 8640 \text{ N}$$

$\therefore$ Axial spring force necessary to engage the clutch,

$$W_e = W_n (\sin \alpha + 0.25 \mu \cos \alpha)$$

$$= 8640 (\sin 12.5^\circ + 0.25 \times 0.2 \cos 12.5^\circ) = 2290 \text{ N}$$

92.

Sol. Given :

$$D = 150 \text{ mm or } R = 75 \text{ mm}$$

$$d = 50 \text{ mm or } r = 25 \text{ mm}$$

$$p = 0.8 \text{ N/mm}^2$$

$$N = 100 \text{ r.p.m}$$

$$\mu = 0.015$$

(i) The load to be supported

Let W = Load to be supported

Assuming that the pressure is uniformly distributed over the bearing surface, therefore bearing pressure (p),

$$0.8 = \frac{W}{\pi(R^2 - r^2)} = \frac{W}{\pi[(75)^2 - (25)^2]} = \frac{W}{15710}$$

$$\therefore W = 0.8 \times 15710 = 12568 \text{ N}$$

(ii) The power lost in friction

We know that total frictional torque,

$$T = \frac{2}{3} \mu \cdot W \left(\frac{R^3 - r^3}{R^2 - r^2} \right)$$

$$T = \frac{2}{3} \times 0.015 \times 12568 \left[\frac{(75)^3 - (25)^3}{(75)^2 - (25)^2} \right] \text{ N-mm}$$

$$T = 125.68 \times 81.25 = 10.212 \text{ N-m}$$

$$\therefore T = 10.212 \text{ N-m}$$

(iii) The heat generated at the bearing

We know that heat generated at the bearing

$$= \text{Power lost in friction} = 0.107 \text{ kW or kJ/s}$$

$$= 0.107 \times 60 = 6.42 \text{ kJ/min}$$

93.

Sol. Given :

$$W = 10 \text{ kN} = 10 \times 10^3 \text{ N}$$

$$p = 12 \text{ mm; } d = 50 \text{ mm; } P_1 = 100 \text{ N}$$

$$\mu = \tan \phi = 0.15$$

Length of the lever

Let

l = Length of the lever

We know that $\tan \alpha = \frac{p}{\pi d} = \frac{12}{\pi \times 50} = 0.0764$

$\therefore$ Effort required at the circumference of the screw to raise the load,

$$P = W \tan(\alpha + \phi) = W \left[\frac{\tan \alpha + \tan \phi}{1 - \tan \alpha \cdot \tan \phi} \right]$$

$$P = 10 \times 10^3 \left[\frac{0.0764 + 0.15}{1 - 0.0764 \times 0.15} \right] = 2290 \text{ N}$$

and torque required to overcome friction,

$$T = P \times \frac{d}{2} = 2290 \times \frac{50}{2} = 57.250 \text{ N-mm} \quad \dots(i)$$

We know that torque applied at the end of the lever,

$$T = P_1 \times l = 100 \times l \text{ N-mm}$$

Equating equation (i) and (ii)

$$l = 57 \frac{250}{100} = 572.5 \text{ mm}$$

Mechanical advantage

We know that mechanical advantage,

$$\text{M.A.} = \frac{W}{P_1} = \frac{10 \times 10^3}{100} = 100$$

Self locking of the screw

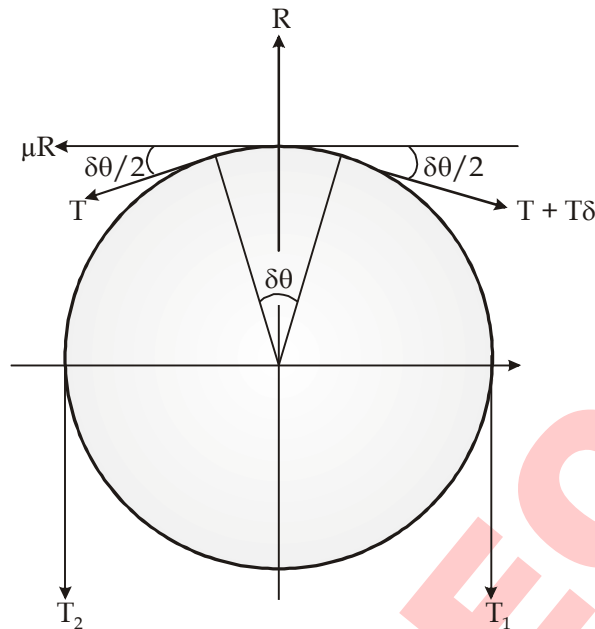
We know that efficiency of the screw jack,

$$\begin{aligned} \eta &= \frac{\tan \alpha}{\tan(\alpha + \phi)} = \frac{\tan \alpha (1 - \tan \alpha \cdot \tan \phi)}{\tan \alpha + \tan \phi} \\ &= \frac{0.0764(1 - 0.0764 \times 0.15)}{0.0764 + 0.15} = \frac{0.0755}{0.2264} = 0.3335 \text{ or } 33.35\% \end{aligned}$$

Since the efficiency of the screw jack is less than 50%, therefore the screw is a self locking screw.

94.

Sol. Flat Belt

Let T_1 = tension on tight side T_2 = tension on slack side θ = angle of lap or contact of the belt over the pulley μ = coefficient of friction between the belt and the pulleyConsider a short length of belt subtending an angle $\delta\theta$ at the centre of the pulley (Fig. 9.14)Let R = normal (radial) reaction between the element length of belt and the pulley T = tension on slack side of the element δT = increase in tension on tight side than that on slack side $T + \delta T$ = tension on tight side of the element

Tensions T and $(T + \delta T)$ act in directions perpendicular to the radii drawn at the ends of the elements. The friction force will act tangentially to the pulley rim resisting the slipping of the elementary belt on the pulley.

Resolving the forces in the tangential direction,

$$\mu R + T \cos \frac{\delta\theta}{2} - (T + \delta T) \cos \frac{\delta\theta}{2} = 0$$

As $\delta\theta$ is small,

$$\cos \frac{\delta\theta}{2} = 1$$

$$\therefore \mu R + T - T - \delta T = 0 \quad \text{or} \quad \delta T = \mu R$$

....(i)

Resolving the forces in the radial direction,

$$R - T \sin \frac{\delta\theta}{2} - (T + \delta T) \sin \frac{\delta\theta}{2} = 0$$

As $\delta\theta$ is small,

$$\sin \frac{\delta\theta}{2} = \frac{\delta\theta}{2}$$

Thus,

$$R - T \frac{\delta\theta}{2} - T \frac{\delta\theta}{2} - \frac{\delta T \delta\theta}{2} = 0$$

Neglecting product of two small quantities,

$$R = T \delta\theta$$

....(ii)

Inserting this value of R in (i),

$$\delta T = \mu \cdot T \delta \theta \quad \text{or} \quad \frac{\delta T}{T} = \mu \delta \theta$$

Integrating between proper limits,

$$\int_{T_2}^{T_1} \frac{dT}{T} = \int_0^\theta \mu d\theta$$

or $\log_e \frac{T_1}{T_2} = \mu \theta$

or $\frac{T_1}{T_2} = e^{\mu \theta} \quad (9.13)$

It is to be noted that the above relation is valid only when the belt is on the point of slipping on the pulleys.

95.

Sol. Velocity ratio of Belt Drive

It is the ratio between the velocities of the driver and the follower or driven. It may be expressed, mathematically, as discussed below :

Let d_1 = Diameter of the drive
 d_2 = Diameter of the follower
 N_1 = Speed of the driver in rpm
 and N_2 = Speed of the follower in rpm.

$\therefore$ Length of the belt that passes over the driver, in one minute
 $= \pi d_1 \cdot N_1$

Similarly, length of the belt that passes over the follower, in one minute
 $= \pi d_2 \cdot N_2$

Since the length of belt that passes over the driver in one minute is equal to the length of belt that passes over the follower in one minute, therefore

$$\pi d_1 \cdot N_1 = \pi d_2 \cdot N_2$$

$\therefore$ Velocity ratio, $\frac{N_2}{N_1} = \frac{d_1}{d_2}$

When the thickness of the belt (t) is considered, then velocity ratio,

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}$$

Note : The velocity ratio of a belt drive may also be obtained as discussed below.

We know that peripheral velocity of the belt on the driving pulley.

$$v_1 = \frac{\pi d_1 \cdot N_1}{60} \text{ m/s}$$

and peripheral velocity of the belt on the driven or follower pulley.

$$v_2 = \frac{\pi d_2 \cdot N_2}{60} \text{ m/s}$$

When there is no slip, then $v_1 = v_2$.

$\therefore \frac{\pi d_1 \cdot N_1}{60} = \frac{\pi d_2 \cdot N_2}{60} \quad \text{or} \quad \frac{N_2}{N_1} = \frac{d_1}{d_2}$



Slip of Belt :

- In the previous articles, we have discussed the motion of belts and shafts assuming a firm frictional grip between the belts and the shafts. But sometimes, the frictional grip becomes insufficient. This may cause some forward motion of the driver without carrying the belt with it. This may also cause some forward motion of the belt without carrying the driven pulley with it. This is called slip of the belt and is generally expressed as a percentage.
- The result of the belt slipping is to reduce the velocity ratio of the system. As the slipping of the belt is a common phenomenon, thus the belt should never be used where a definite velocity ratio is of importance (as in the case of hour, minute and second arms in a watch).

Let $s_1\%$ = Slip between the driver and the belt

and $s_2\%$ = Slip between the belt and the follower.

$\therefore$ Velocity of the belt passing over the driver per second

$$v = \frac{\pi d_1 \cdot N_1}{60} - \frac{\pi d_1 \cdot N_1}{60} \times \frac{s_1}{100} = \frac{\pi d_1 \cdot N_1}{60} \left(1 - \frac{s_1}{100}\right) \quad \dots(i)$$

and velocity of the belt passing over the follower per second,

$$\frac{\pi d_2 \cdot N_2}{60} = v - v \times \frac{s_2}{100} = v \left(1 - \frac{s_2}{100}\right)$$

Substituting the value of v from equation (i)

$$\begin{aligned} \frac{\pi d_2 N_2}{60} &= \frac{\pi d_1 N_1}{60} \left(1 - \frac{s_1}{100}\right) \left(1 - \frac{s_2}{100}\right) \\ \frac{N_2}{N_1} &= \frac{d_1}{d_2} \left(1 - \frac{s_1}{100} - \frac{s_2}{100}\right) \quad \dots \left(\text{Neglecting } \frac{s_1 \times s_2}{100 \times 100}\right) \\ &= \frac{d_1}{d_2} \left(1 - \frac{s_1 + s_2}{100}\right) = \frac{d_1}{d_2} \left(1 - \frac{s}{100}\right) \\ &\quad \dots(\text{where } s = s_1 + s_2, \text{ i.e. total percentage of slip}) \end{aligned}$$

If thickness of the belt (t) is considered, then

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t} \left(1 - \frac{s}{100}\right)$$

Creep of Belt :

- When the belt passes from the slack side to the tight side, a certain portion of the belt extends and it contracts again when the belt passes from the tight side to slack side. Due to these changes of length, there is a relative motion between the belt and the pulley surfaces. This relative motion is termed as creep. The total effect of creep is to reduce slightly the speed of the driven pulley or follower. Considering creep, the velocity ratio is given by

$$\frac{N_2}{N_1} = \frac{d_1}{d_2} \times \frac{E + \sqrt{\sigma_2}}{E + \sqrt{\sigma_1}}$$

where, σ_1 and σ_2 = Stress in the belt on the tight and slack side respectively

and E = Young's modulus for the material of the belt

96.

Sol. Maximum Efficiency of a Screw Jack

We know, that the efficiency of a screw jack,

$$\eta = \frac{\tan \alpha}{\tan(\alpha + \theta)} = \frac{\frac{\sin \alpha}{\cos \alpha}}{\frac{\sin(\alpha + \phi)}{\cos(\alpha + \phi)}} = \frac{\sin \alpha \times \cos(\alpha + \phi)}{\cos \alpha \times \sin(\alpha + \phi)} \quad \dots(i)$$

$$= \frac{2 \sin \alpha \times \cos(\alpha + \phi)}{2 \cos \alpha \times \sin(\alpha + \phi)}$$

...(Multiplying the numerator and denominator by 2)

$$= \frac{\sin(2\alpha + \phi) - \sin \phi}{\sin(2\alpha + \phi) + \sin \phi} \quad \dots(ii)$$

$$\dots \left[\begin{array}{l} \because 2 \sin A \cos B = \sin(A + B) + \sin(A - B) \\ 2 \cos A \sin B = \sin(A + B) - \sin(A - B) \end{array} \right]$$

The efficiency given by equation (ii) is maximum when $\sin(2\alpha + \phi)$ is maximum. i.e., when

$$\sin(2\alpha + \phi) = 1 \text{ or when } 2\alpha + \phi = 90^\circ$$

$$2\alpha = 90^\circ - \phi$$

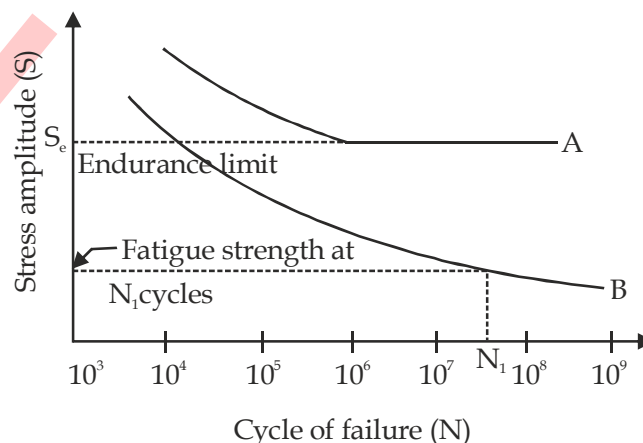
or $\alpha = 45^\circ - \frac{\phi}{2}$

Substituting the value of 2α in equation (ii), we have maximum efficiency,

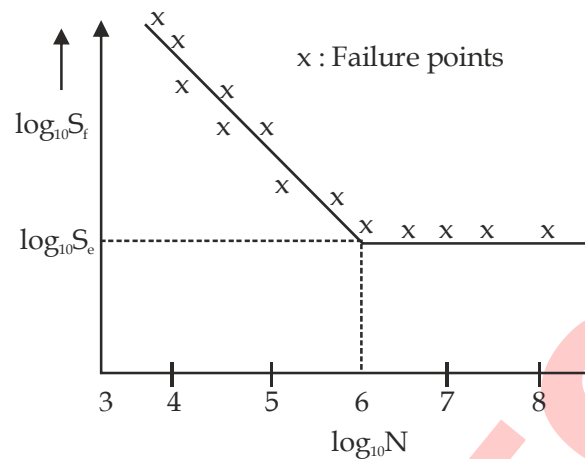
$$\eta_{\max} = \frac{\sin(90^\circ - \phi + \phi) - \sin \phi}{\sin(90^\circ - \phi + \phi) + \sin \phi} = \frac{\sin 90^\circ - \sin \phi}{\sin 90^\circ + \sin \phi} = \frac{1 - \sin \phi}{1 + \sin \phi}$$

97.

Sol. The S-N diagram is the plot between nominal stress amplitude (S_f) versus cycle to failure. There are numerous testing procedures to generate the required data for a proper S-N diagram. The S-N curve is the graphical representation of stress amplitude (S_f) versus the number of stress cycle (N) before the fatigue failure on a log-log plot. It is obtained from R-R-Moore's test. The S-N diagram is also called Wohler diagram, after August Wohler, a German engineer who published his fatigue research in 1870. Each test on the fatigue testing machine gives one failure point on the S-N diagram. In practice, the points are scattered in the figure and an average curve becomes asymptotic at 10^6 cycle, which indicates the stress amplitude corresponding to infinite number of stress cycles. The magnitude of this stress amplitude at 10^6 cycles represents the endurance limit of the material.



Endurance limit, in a true sense, is not exactly a property of material like ultimate tensile strength. It represents a stress level below which the material does not fail and can be cycled infinitely. If the applied stress level is below the endurance limit of the material, the structure is said to have an infinite life.



The concept of endurance limit is used in infinite-life or safe designs.

It is due to interstitial elements (such as carbon or nitrogen in iron) that pin dislocations, thus preventing the slip mechanism, that leads to the formation of microcracks. Care must be taken when using endurance limit in design application because it can disappear due to:

- (a) Periodic overloads (Unpin dislocation).
- (b) Corrosive environments (due to fatigue corrosion interactions)
- (c) High temperature (mobilize dislocation).

However, a test value (S'_e) obtained from polished specimens provides a baseline to which other factors can be applied. Factors that can affect the endurance limit includes surface finish, temperature, stress concentration, notch sensitivity, size of component, environment and sensitivity, size of component, environment and reliability.

Corrected endurance strength (S'_e) is the maximum amplitude stress that the actual component, used in a machine, can withstand for a minimum of 10^6 cycles without fatigue failure.

$$S_e = k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot \frac{1}{k_f} S'_e$$

$$S'_e = 0.5 \times S_{ut} \text{ [for steels]}$$

$$S'_e = 0.4 S_{ut} \text{ [for cast iron and cast-steels]}$$

k_a = Surface finish factor;

k_b = Size factor

k_c = Load factor;

k_d = Temperature factor

k_e = Reliability factor;

k_f = Fatigue stress concentration factor

98.

- Sol.** Given journal diameter, $D = 60 \text{ mm}, \Rightarrow f = 30 \text{ mm}$
 Bearing length, $L = 60 \text{ mm}, L/D = 1$
 Radial load, $W = 2.8 \text{ kN}$
 Journal speed, $N = 1020 \text{ rpm}$
 Journal speed, $n_s = \frac{1020}{60} \text{ rps}$

Radial clearance,	$c = 0.05 \text{ mm}$
Viscosity of oil,	$\mu = 80 \times 10^{-9} \text{ N-s/mm}^2$
Density of oil,	$\rho = 860 \text{ kg/m}^3$
Specific heat of oil,	$c_0 = 1.76 \text{ kJ/kg}^\circ\text{C}$

(i) Sommerfeld number, $S = \left(\frac{r}{c}\right)^2 \frac{\mu n_s}{\rho}$

Bearing pressure, $p = \frac{W}{LD} = \frac{2.8 \times 10^3 \text{ N}}{60 \times 60} = 0.78 \text{ N/mm}^2$

$\therefore S = \left(\frac{30}{0.05}\right)^2 \times \frac{(80 \times 10^{-9}) \times \left(\frac{1020}{60}\right)}{0.78} = 0.627 \approx 0.630$

From the given table,
for the value of $S = 0.630$,

$$\left(\frac{r}{C}\right)^f = 12.8$$

$$\frac{h_0}{c} = 0.8, \text{ and } \frac{Q}{rcn_s l} = 3.59$$

where coefficient of friction variable (C.F.V.) $= \left(\frac{r}{c}\right)^f$

and flow variable, $\text{F.V.} = \frac{Q}{rcn_s l}$

$$\left(\frac{r}{c}\right)^f = 3.22$$

Coefficient of friction, $f = 12.8 \times \frac{0.05}{30} = 0.0213$

(ii) Power loss in friction, $(\text{kW}) = f.W.r.\omega$

$$= \frac{2\pi n_s f W r}{10^6} = \frac{2\pi \times \left(\frac{1020}{60}\right) \times 0.0213 \times 2.8 \times 10^3 \times 30}{10^6} = 0.1911 \text{ kW}$$

(iii) We know temperature rise,

$$\Delta T = \frac{8.3 \times \rho \times (\text{CFV})}{(\text{FV})} = \frac{8.3 \times 0.78 \times 12.80}{3.59} = 23.082^\circ\text{C}$$

(iv) Minimum film thickness,

From the table for $\frac{h_0}{c} = 0.8, S = 0.630$

we get $\phi = 74.02^\circ$ i.e., angle of eccentricity (or location)

$$\frac{h_0}{c} = 0.8$$

Minimum film thickness,

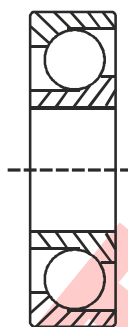
$$h_0 = 0.8 \times c = 0.80 \times 0.05 = 0.04 \text{ mm} = 40 \text{ microns}$$

99.

Sol. Angular contact bearing: In angular contact bearing, the grooves in inner and outer races are so shaped, that the line of reaction at the contact between balls and races makes an angle with the axis of the bearing. This reaction has two components-radial and axial. Therefore, angular contact bearing can take radial and thrust loads. Angular contact bearings are often used in pairs, either side by side or at the opposite ends of the shaft, in order to take the thrust load in both directions. These bearings are assembled with a specific magnitude of pre-load.

Angular contact bearings offer following advantages:

- Angular contact bearing can take both, radial and thrust loads.
- In angular contact bearing, one side of the groove in the outer race is cut away to permit the insertion of more number of balls than that of deep groove ball bearing. This permits the bearing to carry relatively large axial and radial loads. Therefore, the load carrying capacity of angular contact bearing is more than that of deep groove ball bearing.



Angular contact bearing

The disadvantages of angular contact bearings are as follows:

- Two bearings are required to take thrust load in both directions.
- Angular contact bearing must be mounted without axial play.
- Angular contact bearing requires initial pre-loading

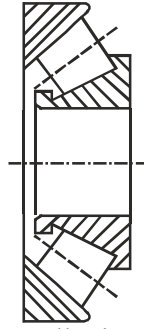
Taper roller bearing: The taper roller bearing consists of rolling elements in the form of a frustum of cone. They are arranged in such a way that the axes of individual rolling elements intersect in a common apex point on the axis of the bearing. In kinematics' analysis, this is the essential requirement for pure rolling motion between conical surfaces. In taper roller bearing, the line of resultant reaction through the rolling elements makes an angle with the axis of the bearing. Therefore, taper roller bearing can carry both radial and axial loads. In fact, the presence of either component results in the other, acting on the bearing. In other words, a taper roller bearing subjected to pure radial load induces thrust component and vice versa. Therefore, taper roller bearings are always used in pair to balance the thrust component. Taper roller bearing has separable construction. The outer ring is called 'cup' and the inner ring called 'cone'. The cup is separable from the remainder assembly of the bearing elements including the rollers, cage, and the cone.

Taper roller bearing offers following advantages:

- Taper roller bearing can take heavy radial and thrust loads.
- Taper roller bearing has more rigidity.
- Taper roller bearing can be easily assembled and disassembled due to separable parts.

The disadvantages of taper roller bearing are as follows:





Taper roller bearing

- It is necessary to use two taper roller bearings on the shaft to balance the axial force.
- It is necessary to adjust the axial position of the bearing with pre-load. This is essential to coincide the apex of the cone with the common apex of the rolling elements.
- Taper roller bearing cannot tolerate misalignment between the axes of the shaft and the housing bore.
- Taper roller bearings are costly.

100.

Sol. Given: $D = 320$ mm; $w = 0.03$ m; $P_{\max} = P_n^l = 80$ kpa, $\phi_1 = 15^\circ$, $\phi_2 = 145^\circ$, $a = 220$ mm, $c = 125$ mm, $\mu = 0.32$
For leading shoe,

$$F \times a - \int_{\phi_1}^{\phi_2} P_n^l c \sin \theta + \int_{\phi_1}^{\phi_2} \mu P_n^l (r - c \cos \theta) = 0$$

$$F \times 0.22 - \frac{0.16 \times 0.125 \times 0.03}{4} \times (80 \times 10^3) \left[2 \times \frac{\pi}{180} \times 130 - \sin 290 + \sin 30 \right] \\ + \frac{0.32 \times 0.16 \times 0.03 \times (80 \times 10^3)}{4} [4 \times 0.16 (\cos 15 - \cos 145) - 0.125 (\cos 30 - \cos 290)] = 0$$

$$F \times 0.22 - 12 \times 5.977 + 30.72 [1.0769] = 0$$

$$F = 175.643 \text{ N}$$

For the trailing shoe,

$$175.643 \times 0.22 - 8.9655 \times 10^{-4} P_n^l - 4.135 \times 10^{-4} \times P_n^l = 0$$

$$P_n^l = 29496.172 \text{ N / m}^2$$

$\therefore$ Braking torque

$$T_B = r^2 \mu w (P_n^l + P_n^t) (\cos \phi_1 \cos \phi_2) \\ = 0.16^2 \times 0.32 \times 0.03 (80000 + 29496.172) (\cos 15 - \cos 145) \\ T_N = 48.036 \text{ N}$$

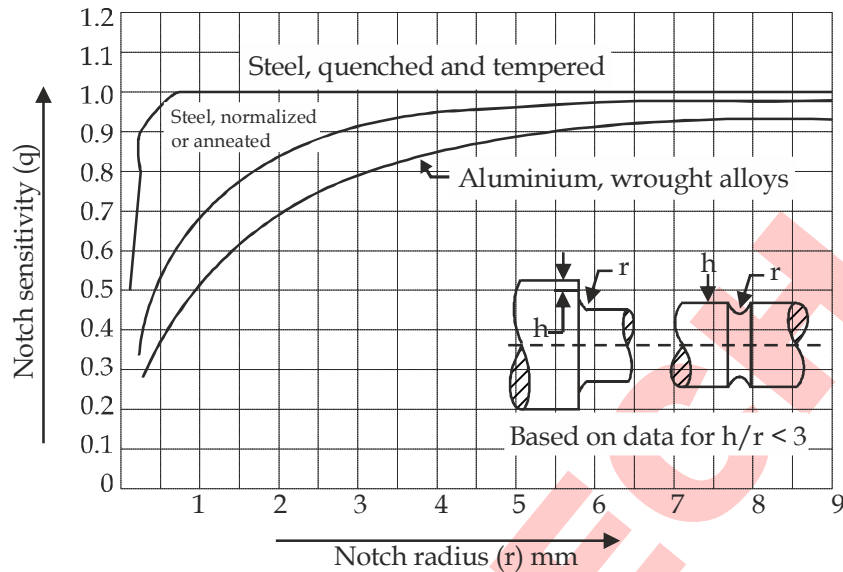
101.

Sol. Notch Sensitivity

In cyclic loading, the effect of the notch or the fillet is usually less than predicted by the use of the theoretical factors as discussed before. The difference depends upon the stress gradient in the region of the stress concentration and on the hardness of the material. The term notch sensitivity is applied to this behaviour. It may be defined as the degree to which the theoretical effect of stress concentration is actually reached. The stress gradient depends mainly on the radius of the notch, hole or fillet and on the grain size of the material. Since the extensive data for estimating the notch sensitivity factor



(q) is not available, therefore the curves, as shown in Fig., may be used for determining the values of q for two steels.



When the notch sensitivity factor q is used in cyclic loading, then fatigue stress concentration factor may be obtained from the following relations:

$$q = \frac{K_f - 1}{K_t - 1}$$

or

$$K_f = 1 + q(K_t - 1) \quad \dots[\text{For tensile or bending stress}]$$

and

$$K_{fs} = 1 + q(K_{ts} - 1) \quad \dots[\text{For shear stress}]$$

where

K_t = Theoretical stress concentration factor for axial or bending loading, and

K_{ts} = Theoretical stress concentration factor for torsional or shear loading.

Fatigue Stress Concentration Factor

When a machine member is subjected to cyclic or fatigue loading, the value of fatigue stress concentration factor shall be applied instead of theoretical stress concentration factor. Since the determination of fatigue stress concentration factor is not an easy task, therefore from experimental tests it is defined as

Fatigue stress concentration factor,

$$K_f = \frac{\text{Endurance limit without stress concentration}}{\text{Endurance limit with stress concentration}}$$

Stress Concentration

Whenever a machine component changes the shape of its cross-section, the simple stress distribution no longer holds good and the neighbourhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. It occurs for all kinds of stresses in the presence of fillets, notches, holes, keyways, splines, surface roughness or scratches etc.

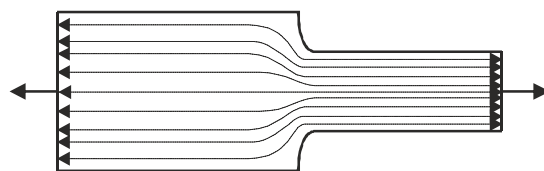


Fig.:- Stress concentration

In order to understand fully the idea of stress concentration, consider a member with different cross-section under a tensile load as shown in Fig. A little consideration will show that the nominal stress in the right and left hand sides will be uniform but in the region where the cross-section is changing, a re-distribution of the force within the member must take place. The material near the edges is stressed considerably higher than the average value. The maximum stress occurs at some point on the fillet and is directed parallel to the boundary at that point.

102.

Sol. Important Terms Used in Screw Threads

The following terms used in screw threads, as shown in Fig. are important from the subject point of view :

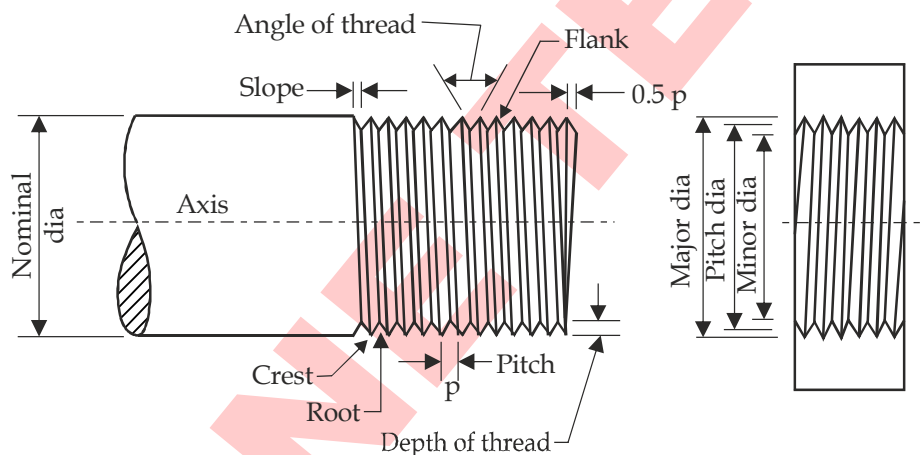


Fig.:- Terms used in screw threads.

1. **Major diameter:** It is the largest diameter of an external or internal screw thread. The screw is specified by this diameter. It is also known as outside or nominal diameter.
2. **Minor diameter:** It is the smallest diameter of an external or internal screw thread. It is also known as core or root diameter.
3. **Pitch diameter:** It is the diameter of an imaginary cylinder, on a cylindrical screw thread, the surface of which would pass through the thread at such points as to make equal the width of the thread and the width of the spaces between the threads. It is also called an effective diameter. In a nut and bolt assembly, it is the diameter at which the ridges on the bolt are in complete touch with the ridges of the corresponding nut.
4. **Pitch:** It is the distance from a point on one thread to the corresponding point on the next. This is measured in an axial direction between corresponding points in the same axial plane. Mathematically,

$$\text{Pitch} = \frac{1}{\text{No. of threads per unit of screw}}$$

5. **Lead:** It is the distance between two corresponding points on the same helix. It may also be defined as the distance which a screw thread advances axially in one rotation of the nut. Lead is



equal to the pitch in case of single start threads, it is twice the pitch in double start, thrice the pitch in triple start and so on.

6. **Crest:** It is the top surface of the thread.
7. **Root:** It is the bottom surface created by the two adjacent flanks of the thread.
8. **Depth of thread:** It is the perpendicular distance between the crest and root.
9. **Flank:** It is the surface joining the crest and root.
10. **Angle of thread:** It is the angle included by the flanks of the thread.
11. **Slope:** It is half the pitch of the thread.

103.

Sol. Deviation:

Deviation is the algebraic difference between the size and basic size of the job and can be positive, negative, or zero.

There are four types of deviations, including Upper Deviation, Lower Deviation, Actual Deviation, and Fundamental Deviation.

Upper Deviation:

Upper Deviation is the algebraic difference between the maximum diameter of the job size and the basic diameter.

Lower Deviation:

Lower Deviation is the algebraic difference between the minimum diameter of the job size and the basic diameter.

Actual Deviation:

Actual Deviation is the algebraic difference between the actual and basic size of a job.

Fundamental Deviation:

Fundamental Deviation is either the lower or upper deviation, shown in relation to the zero lines.

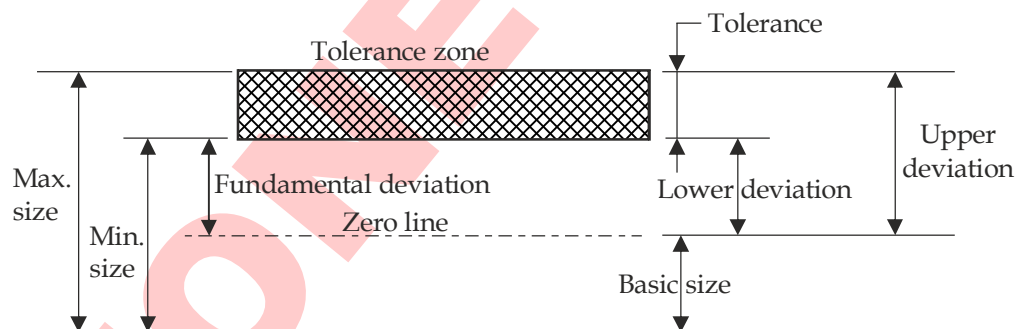
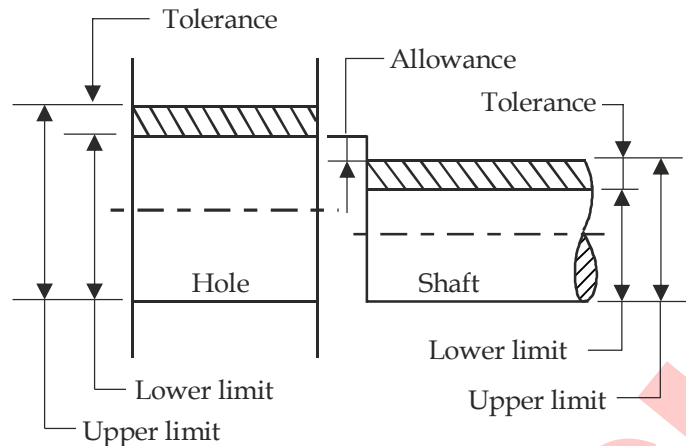


Fig.:- Fundamental deviation

Allowance:

Allowance is the difference in sizes or gaps between two mating parts and can be positive or negative based on the fitting nature.

There are two types of allowances: Maximum Allowance (the difference between the upper limit of a hole size and the lower limit of a shaft size) and Minimum Allowance (the difference between the lower limit of a hole size and the upper limit of a shaft size).



104.

Sol. Terms used in Compression Springs

The following terms used in connection with compression springs are important from the subject point of view.

1. **Solid length.** When the compression spring is compressed until the coils come in contact with each other, then the spring is said to be solid. The solid length of a spring is the product of total number of coils and the diameter of the wire. Mathematically,

Solid length of the spring,

$$L_s = n' \cdot d$$

where

n' = Total number of coils, and

d = Diameter of the wire.

2. **Free length.** The free length of a compression spring, as shown in Fig., is the length of the spring in the free or unloaded condition. It is equal to the solid length plus the maximum deflection or compression of the spring and the clearance between the adjacent coils (when fully compressed). Mathematically,

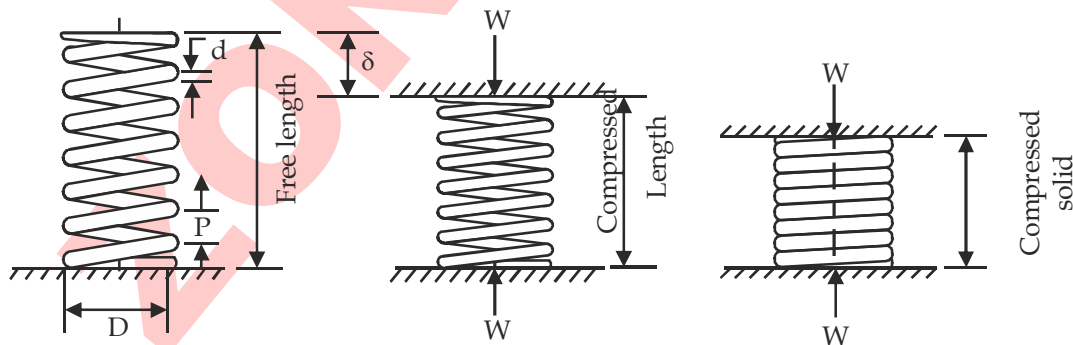


Fig.:- Compression spring nomenclature

Fig. Compression spring nomenclature.

Free length of the spring,

$$L_f = \text{Solid length} + \text{Maximum compression} + \text{*Clearance between adjacent coils (or clash allowance)}$$



$$= n' \cdot d + \delta_{\max} + 0.15\delta_{\max}$$

The following relation may also be used to find the free length of the spring, i.e.

$$LF = n' \cdot d + \delta_{\max} + (n' - 1) \times 1 \text{ mm}$$

In this expression, the clearance between the two adjacent coils is taken as 1 mm.

- 3. Spring index.** The spring index is defined as the ratio of the mean diameter of the coil to the diameter of the wire. Mathematically,

$$\text{Spring index, } C = D/d$$

where

D = Mean diameter of the coil, and

d = Diameter of the wire.

- 4. Spring rate.** The spring rate (or stiffness or spring constant) is defined as the load required per unit deflection of the spring. Mathematically,

$$\text{Spring rate, } k = W / \delta$$

where

W = Load, and

δ = Deflection of the spring.

- 5. Pitch.** The pitch of the coil is defined as the axial distance between adjacent coils in uncompressed state. Mathematically,

$$\text{Pitch of the coil, } P = \frac{\text{Free length}}{n' - 1}$$

The pitch of the coil may also be obtained by using the following relation, i.e.

$$\text{Pitch of the coil, } P = \frac{L_F - L_s}{n'} + d$$

where

L_F = Free length of the spring,

L_s = Solid length of the spring,

n' = Total number of coils, and

d = Diameter of the wire.

In choosing the pitch of the coils, the following points should be noted:

- The pitch of the coils should be such that if the spring is accidentally or carelessly compressed, the stress does not increase the yield point stress in torsion.
- The spring should not close up before the maximum service load is reached.

Note: In designing a tension spring, the minimum gap between two coils when the spring is in the free state is taken as 1 mm. Thus the free length of the spring,

$$LF = n \cdot d + (n - 1)$$

and pitch of the coil,

$$p = \frac{L_F}{n - 1}$$

Advantages:

Versatility: Used across many industries for various tasks, from valves to mattresses.

Energy Storage: Efficiently store and release energy, providing consistent pressure.

Shock Absorption: Excellent for dampening impacts and providing cushioning.

Compactness: Deliver high functionality in a small space.

Cost-Effective: Economical to produce, especially in large quantities, and generally maintenance-free.

Linear Motion: Ideal for applications needing precise linear movement and return force.

Disadvantages:

Buckling: Prone to bending or buckling if compressed too far or if they are too slender.

Fatigue: Repeated compression can lead to material fatigue and eventual failure, notes Tangshan Hengguang Mechanical Parts Co., Ltd..

Repair Difficulty: If a spring breaks, it's often difficult or impossible to repair, requiring replacement.

Sensitivity to Load: Performance depends heavily on staying within specified load and compression limits; exceeding them causes permanent issues.

Material Limits: May have limited durability in extreme high-stress environments compared to specialized springs, says Tangshan Hengguang Mechanical Parts Co., Ltd..

105.

Sol. Lever for a Lever Safety Valve

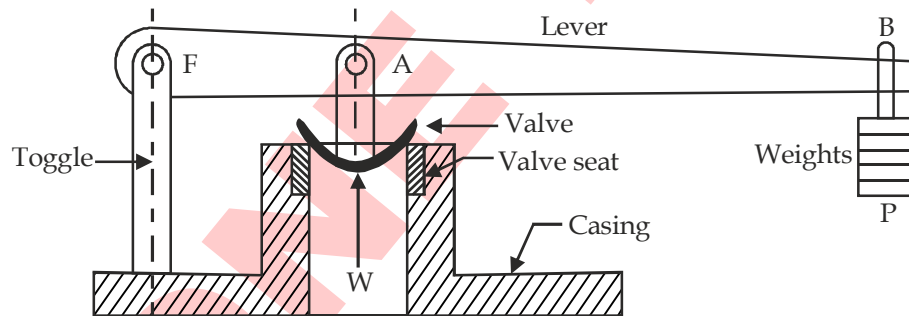
A lever safety valve is shown in Fig. . It is used to maintain a constant safe pressure inside the boiler. When the pressure inside the boiler increases the safe value, the excess steam blows off through the valve automatically. The valve rests over the gunmetal seat which is secured to a casing fixed upon the boiler. One end of the lever is pivoted at the fulcrum F by a pin to the toggle, while the other end carries the weights. The valve is held on its seat against the upward steam pressure by the force P provided by the weights at B. The weights and its distance from the fulcrum are so adjusted that when the steam pressure acting upward on the valve exceeds the normal limit, it lifts the valve and the lever with its weights. The excess steam thus escapes until the pressure falls to the required limit. The lever may be designed in the similar way as discussed earlier. The maximum steam load (W), at which the valve blows off, is given by

$$W = \frac{\pi}{4} \times D^2 \times p$$

where

D = Diameter of the valve, and

p = Steam pressure.



106.

Sol. Hydrodynamic bearing works by floating the rotating components on a thin film of fluid. In hydrodynamic bearings, separation of the rotating and stationary surfaces through a fluid film reduces friction and wear, results in long life of bearing and can achieve life of bearing equivalent to the life of machine. Hydrodynamic bearing are used in high load, high speed and high precision application, where other ball bearings are unable to achieve desired performance due to its short life, high noise and vibration. It is also termed as fluid film bearings and used to reduce cost.

Since there is no contact between the surfaces, the surface finish, have little or no influence on the performance of the bearing. The resistance to relative motion arises from the viscous resistance of the fluid. Therefore, the viscosity of the lubricant affects the performance of the bearing.



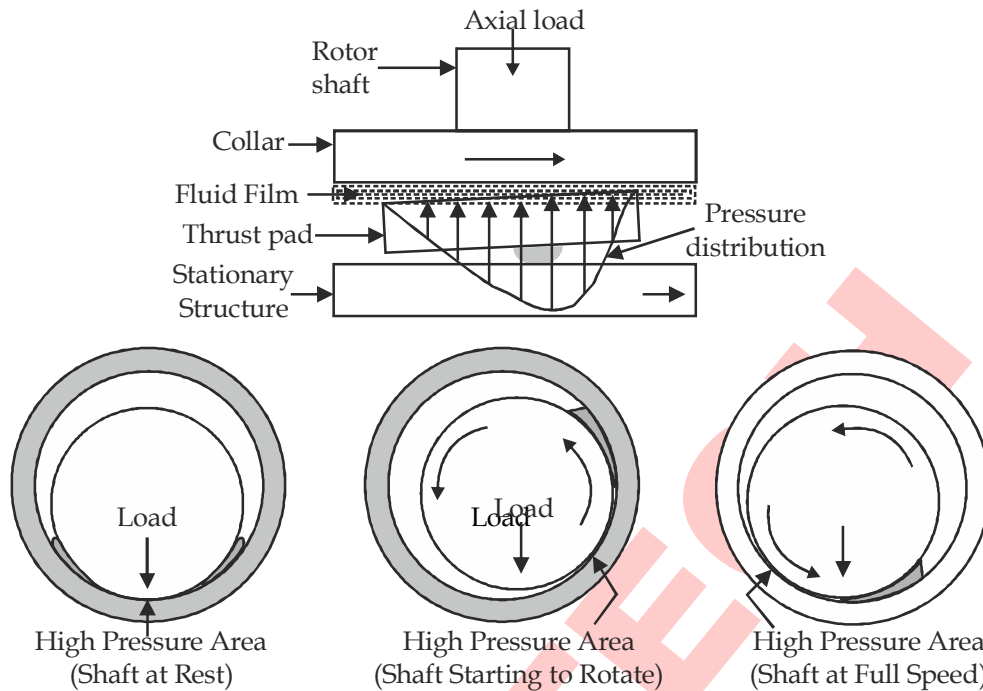


Fig.:- Hydrodynamic bearing

Hydrodynamic lubrication : It is defined as a system of lubrication in which the load supporting fluid film is created by the shape and relative motion of the sliding surfaces. This system is beneficial to reduce friction and wear of rubbing surfaces with the aid of liquid (or semi-solid) lubricant. Theoretical analysis of hydrodynamic lubrication was carried out by Osborne Reynolds (1842-1912) and the equations resulted from the analysis has served a basis for designing hydro-dynamically lubricated bearings.

The following assumptions were made by Reynolds in the analysis :

- The lubricating fluid is Newtonian.
- The flow is laminar.
- The shear stress between the flow layers is proportional to the velocity gradient in the direction perpendicular to the flow (Newton's law of viscosity),

$$\tau = \eta \frac{\partial v}{\partial y}$$

Where,

τ = dynamic viscosity of oil

v = linear velocity of laminar layer

y = the axis perpendicular to the flow direction

- The inertia forces resulted from the accelerated movement of the flowing lubricant are neglected.
- The lubricating fluid is incompressible.
- The pressure of the fluid p is constant in the direction perpendicular to the laminar flow, $dp/dy=0$ (assumption of thin lubrication film).
- The viscosity of the fluid is constant throughout the lubrication film.

Following are the variables involved in lubrication theory :

$$\text{Sommerfeld Number, } S_o = F\psi^2 / (DB\omega\eta)$$

Where,

F = Bearing load

ψ = Relative bearing clearance = $(D - D_j) / D$

D = Bearing diameter

B = Bearing length

ω = Angular speed of journal

D_j = Journal diameter

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