CBCS SCHEME

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18ME822

Eighth Semester B.E. Degree Examination, June/July 2023
Tribology

Time: 3 hrs. Max. Marks: 100

Note: Answer any FIVE full questions, choosing ONE full question from each module.

Important Note : 1. On completing your answers, compulsorily draw diagonal cross lines on the remaining blank pages.
2. Any revealing of identification, appeal to evaluator and /or equations written eg, 42+8 = 50, will be treated as malpractice.

Module-1

1 a. Define the term Tribology. Discuss industrial importance of Tribology. **(10 Marks)**

b. Briefly explain the mechanism of lubrication with the help of Stribeck curve. **(10 Marks)**

OR

2 a. Define viscosity. State and explain Newton's law of viscosity, with a neat sketch. **(10 Marks)**

b. With a neat sketch, explain any two types viscometer. **(10 Marks)**

Module-2

3 a. Explain Bowden and Tabor's adhesion theory of friction. **(10 Marks)**

b. What are the theories of friction? Explain any two theories and test measurement. **(10 Marks)**

OR

4 a. Define Wear. Briefly explain different types of Wear. **(10 Marks)**

b. How do you classify mechanism of wear and explain any one measurement of test method? **(10 Marks)**

Module-3

5 a. State the assumptions of Petroff's equation. Derive Petroff's equation for coefficient of bearing of friction in a lightly loaded bearing. **(10 Marks)**

b. A full journal bearing having the following:
Shaft diameter 45mm, bearing length 65mm, radial clearance ratio is 0.001, Speed 2000 rpm, radial load 800 N, Viscosity of the lubricant at effective temperature of oil 1.2×10^{-6} Reyn. Considering the bearing as lightly loaded, determine
i) Friction torque at the shaft ii) Coefficient of friction iii) Power loss. **(10 Marks)**

OR

6 State clearly the assumptions made in the derivation of Reynold's equation in 2D. Derive the equation. **(20 Marks)**

Module-4

7 a. A rectangular slider bearing with fixed shoe has the following specification:
Bearing length = 0.0762m ; Shoe width = 0.065, Slider velocity = 2.54 m/s, load on bearing = 5383 N, Minimum of oil film thickness = 1.27×10^{-5}m, Mean viscosity of oil = 0.06803 N-s/m^3, find the inclination of the surface in radians, degree and coefficient of friction. **(10 Marks)**

b. Derive an expression for the load carrying capacity of a plane slider bearing with fixed shoe. **(10 Marks)**

OR

8 a. Derive an expression for the load carrying capacity and rate of flow of oil through a hydrostatic step bearing. (10 Marks)

b. A hydrostatic circular thrust bearing has the following data:
Shaft diameter = 300mm, diameter of packet = 200mm, shaft speed = 100 rpm, Pressure at the pocket = 500 kN/m^2, Film thickness = 0.07mm, Viscosity of lubricant = 0.5 Pas.
Determine (i) Load carrying capacity (ii) Oil flow rate (iii) Power loss due to friction. (10 Marks)

Module-5

9 a. Discuss any ten desirable properties of a good bearing material. (10 Marks)

b. Briefly discuss the common bearing materials that are used in practice. (10 Marks)

OR

10 a. What is Surface Engineering? Explain the scope of surface engineering. (10 Marks)

b. Briefly explain different techniques to achieve surface modifications. (10 Marks)

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Sub: Tribology 18ME822
June / July 2023
Module - 1

1.) a) The word tribology is derived from the Greek word "tribos" meaning rubbing, so the literal translation would be "the science of rubbing". It is also defined as the "science and technology of interacting surfaces in relative motion and of related subjects and practices".

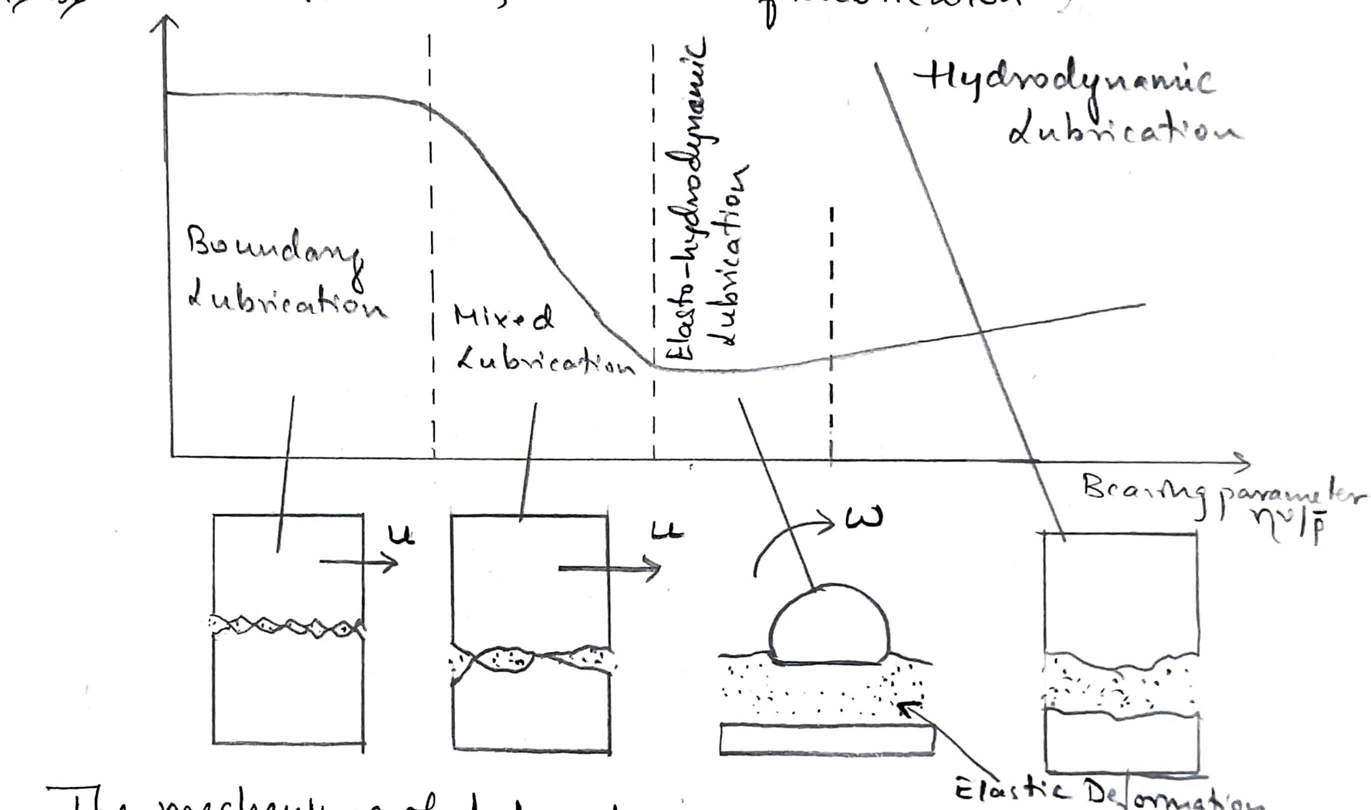
Industrial importance of Tribology: Tribology is crucial to modern machinery which uses sliding and rolling surfaces. Examples of productive friction are brakes, clutches, driving wheels on trains and automobiles, bolts and nuts. Examples of productive wear are writing with a pencil, machining, polishing and shaving. Examples of unproductive friction and wear are internal combustion and aircraft engines gears, cams, bearings etc.

According to some estimates, losses resulting from ignorance of tribology amount in the United States to about 4% of its gross national product and approximately one-third of the world's energy resources in present use appear as friction in one form or another. Thus, the importance of friction reduction and wear control cannot be overemphasized for economic reasons and long term reliability. The purpose of research in tribology is understandable in terms of minimization and elimination of losses resulting from friction and wear at all levels of technology where the rubbing of surfaces is involved. Research in tribology leads to greater plant efficiency better performance, fewer breakdowns & significant savings.

Since 1800's tribology has been important in numerous industrial applications requiring relative motion for example railroads, automobiles, aircraft and machine components.

Some of the tribological machine components used in these applications include bearings, seals, gears and metal cutting. Since the 1980s other applications have included magnetic storage devices and micro/nano electrical mechanical systems (MEMS/NEMS) as well as bio medical and beauty care products.

1) b) Stribeck Curve & Mechanism of Lubrication



The mechanisms of lubrication are illustrated in fig above. The lubrication is provided by either liquid (e.g natural oil, organic oil and inorganic oil) or solid (e.g Grease, graphite, graphene etc.

The liquid lubrication can be studied by a plot known as the "Stribeck curve". This curve was developed in the first half of the twentieth century to categorize the friction properties between two liquid lubricated surfaces. The curve is plotted with friction co-efficient as a function of a parameter given by hv/p: where 'h' is the viscosity of the lubricant, 'v' is the relative velocity and 'P' is the contact pressure. The Stribeck curve contains three main regimes known as Boundary lubrication, Mixed lubrication and Hydrodynamic lubrication (HL) with Elasto-hydrodynamic lubrication (EHL) regimes in between mixed and HL regimes.

The boundary lubrication regime is characterized by solid-solid interactions even though there is presence of liquid lubricant. This condition exists when the contact pressure is high and/or the relative speed is low.

The hydrodynamic lubrication regime is characterized by whole separation of the two mating surfaces by a thin layer of lubricant. This thin lubricant film is maintained by the fluid as it enters the minimum gap between the two mating parts. The smallest film thickness must be greater that the surface roughness. The mixed lubrication regime is categorized by partial solid-solid interaction at asperity level with partial fluid film separation.

The Elasto hydrodynamic lubrication (EHL) is a distinct case of HL where there is elastic deformation of one or both solid surfaces in contact and thus facilitating the formation of a fluid film in between.

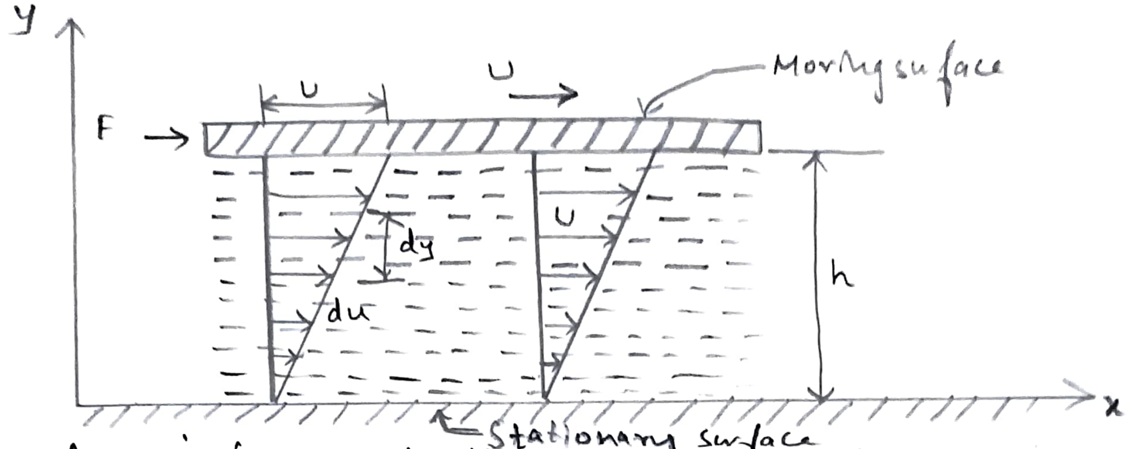
The coefficient of friction is high when there is solid-solid interaction at asperity level as in the boundary lubrication regime. It can be extremely low in the range of 0.001 in the hydrodynamic film regime.

2) a) Viscosity : Viscosity of a liquid is its ability to resist a change in shape. The resistance is due to internal friction and is a molecular phenomenon.

Newton's Law of Viscous Flow

Consider two plane, parallel surfaces separated by a film of fluid having the thickness 'h' as shown in the figure. ~~below~~

(P.T.O)



A force 'F' is applied to the upper surface so that it moves with a constant velocity 'U' with respect to the lower stationary plane.

Assume that there is no vertical load applied to the surface, that the oil is incompressible. The particles of the lubricant adhere strongly to the moving and stationary surfaces, so that the oil adhering to the stationary surface remains at rest while the oil adhering to the upper surface is moving with the same velocity 'U'. Under this condition the velocity of oil at any point in the fluid film is proportional to the distance of the point from stationary surface.

Newton's laws of viscous flow states that at any point in a fluid the shear stress 'T' is directly proportional to the rate of shear.

i.e. $T \propto \frac{du}{dy}$

or $T = \eta \cdot \frac{du}{dy}$

where

η = constant of proportionality & is called as absolute viscosity

$\frac{du}{dy}$ = rate of shear

In the above case, the rate of shear $\frac{du}{dy}$ is a constant value for any point in the fluid film

$\therefore \frac{du}{dy} = \frac{u}{y} = \frac{U}{h}$

$\therefore T = \eta \cdot \frac{U}{h}$ or $\eta = T \cdot \frac{h}{U}$ — (1)

If 'A' is the area of the plane surface in contact with the lubricant then shear stress is given by.

$$\tau = \frac{F}{A} \quad \text{— (2)}$$

Substituting equ (2) in equ(1) we get

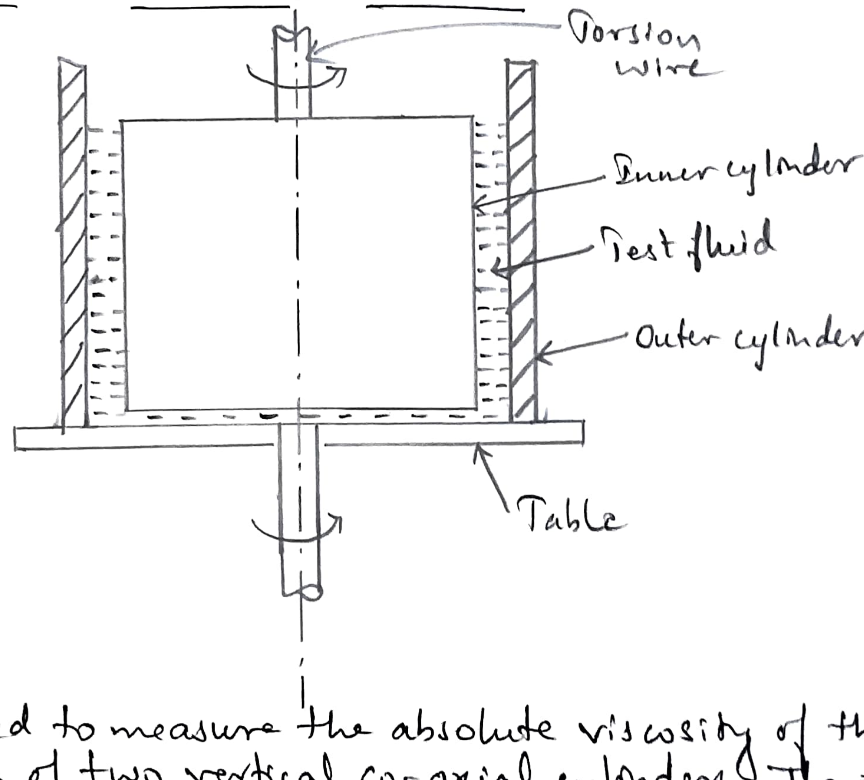
$$\eta = \frac{F.h}{UA} \quad \text{— (3)}$$

Then absolute viscosity is defined as a force required to move a plane surface having a unit area over another plane surface at unit velocity, when the two surfaces are separated by a layer of fluid of unit thickness

unit $\eta = \frac{Fh}{UA} = \frac{N.m}{m/sec \times m^2} = N/m^2 . sec = Pa.sec$

2) b) The two types of Viscometer are

1) Couette - Hatschek Viscometer

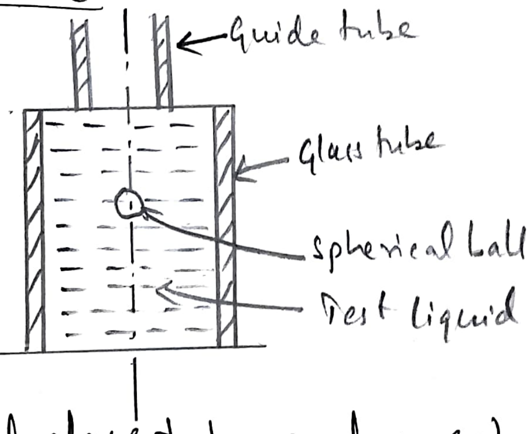


It is used to measure the absolute viscosity of the fluid. It consists of two vertical co-axial cylinders. The first outer cylinder is fixed to the table which can be made to rotate about its vertical axis at a constant speed.

while the inner cylinder is suspended by a torsion wire. The liquid whose viscosity is to be measured is filled in the space between the two cylinders.

The outer cylinder is made to rotate at constant speed by rotating the table and the resulting torque on the inner cylinder is measured by the twist of the wire when the outer cylinder rotates the motion of the fluid between the two cylinders is similar to the flow of fluid between the two parallel plates. In this apparatus the outer surface of the inner cylinder acts as stationary plate while the inner surface of the outer cylinder acts as moving plate. This viscometer is also known as rotational viscometer and is usually used to determine viscosities of liquids having high viscosity.

2) Falling sphere Viscometer



It consists of vertical glass tube and a spherical ball. The diameter of the ball is smaller than the inner diameter of the glass tube. The liquid whose viscosity is to be measured is filled in the tube and the spherical ball is released from the top of the tube along the axis by means of a guide tube. The time taken for the falling body to cover a measured distance is recorded by an electronic clock. By knowing the velocity of the falling sphere & by using stokes formula the velocity can be computed

According to stokes equation, viscosity

$$\eta = \frac{2R^2(\rho - \rho')g}{9v}$$

Module - 2

3) a) Bowden & Tabor's adhesion theory of friction

Bowden and Tabor's explained the adhesion theory of friction when metal surfaces are loaded against each other, they make contact only at the tips of the asperites. Because the real contact area is small the pressure over the contacting asperities is assumed high enough to cause them to deform plastically. This plastic flow of the contacts causes an increase in the area of contact until the real area of contact is just sufficient to support the load. Under these conditions for on ideal elastic-plastic material.

$$W = A \cdot P_o$$

where 'A' is the real area of contact
'P_o' is the yield pressure of the metal
'W' is the normal load

When the metals are in the contact, cold welding takes place due to the adhesion. So a force per unit area of contact necessary to shear the junction

$$F = A.S + P_e$$

where 'P_e' is the force required to plough hard asperities through a softer surface

For most situations involving unlubricated metals P_e is small compared to A.S and may be neglected

Therefore, $F = A.S$

$$F = (W/P_o).S = F/W = S/P_o$$

Therefore, $\mu = F/W = S/P_o$

Thus this theory explains two laws of friction
- The friction is independent of the apparent area of contact
- Friction force is proportional to the load.

For most metals

$$S = P_o/5$$

So adhesion theory predicts that $\mu = 0.2$, when the material pairs are similar. It should be true for any combination of same material. But it is not true usually because of junction growth and work hardening.

The above theory is true for static contacts. But when tangential force is applied, yielding takes place as a result of the combined normal and shear stresses. So Bowden and Tabor re-examine some of the assumptions and present a modified description of friction as

$$A^2 = \left(W/P_o\right)^2 + \alpha.\left(F/P_o\right)^2$$

where W/P_o is the area of contact derived from the above theory in which only the effect of normal load is considered and the additional term $\alpha.(F/P_o)^2$ represents the increase caused by the shear or friction force

3) b) Theories of Friction

The following are the theories of friction

- Bowden and Tabor's Simple Adhesion Theory
- Modified Adhesion Theory: Junction Growth
- Deformation Theory: Ploughing

Measurement Methods

1) Inclined Plane Rig
2) Pin on disk Rig

1.) Inclined Plane Rig

The simplest arrangement is the inclined plane test shown in fig. A specimen is placed on a flat plane whose inclination with the horizontal is gradually increased until the specimen on it starts to slide. If the inclination at this moment be q, then $m_s = \tan q$.

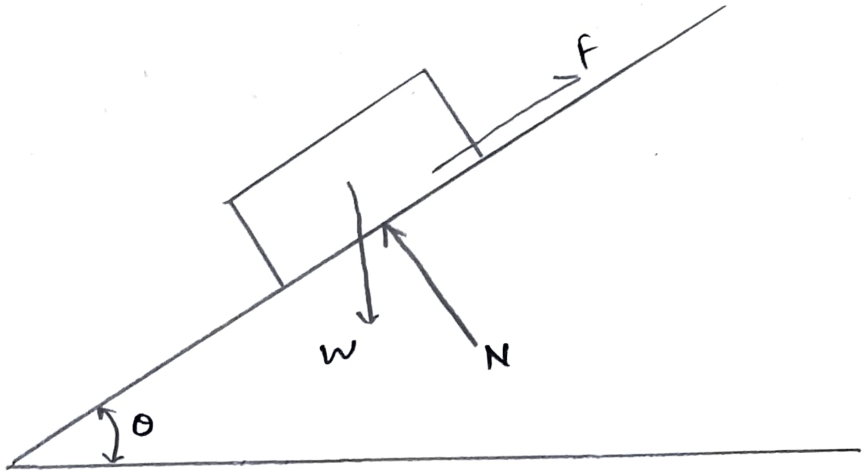
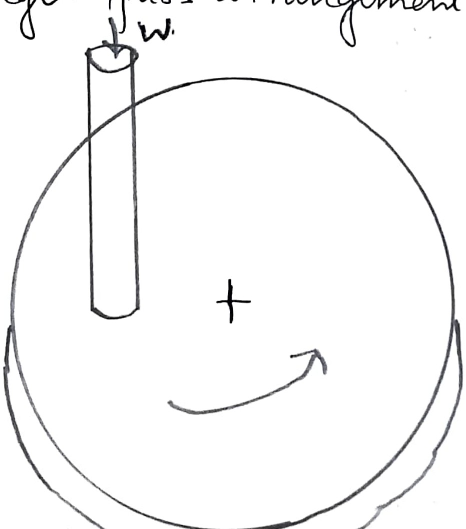


Fig. Measuring friction by an inclined plane test rig

Obviously, this method is incapable of evaluating friction in continuous sliding.

2) Pin-on-disk Rig.

In continuous sliding cases, the rig based on pin-on-disk configuration is used. The pin is held stationary under a normal load ~~while~~ while the disk is made to rotate. The loading can be provided by simple dead weight or by spring loading or hydraulic or pneumatic pressure. The friction force is measured with the help of the calibrated tangential movement of a capacitive or inductive transducer mounted on the stationary specimen. for a multiple-pass arrangement, the pin is held at a constant radial distance from the centre of the disk, but in a single-pass arrangement it is moved radially during the experiment.



4) a) Definition of wear: The removal of material from one or both of the two solid surfaces in relative motion (sliding, rolling, impact) is termed as 'Wear'

Surface damage due to material displacement with no net change in volume or weight is also called 'wear'

Types of wear

a) Adhesive wear

b) Abrasive wear

c) Corrosive wear

d) Fatigue wear

a) Adhesive wear

- Adhesive wear takes place when two nominally flat solid surfaces are in sliding contact.
- At the interface asperities, contact and bonding take place due to adhesion. These contacts get sheared during sliding, resulting in detachment of a fragment from one surface and its attachment to the other surface or formation of loose wear particles.
- Some get fractured by a fatigue process during repeated loading and unloading process resulting in the formation of loose wear particles
- Several mechanisms exist for the formation of wear particles. According to Archard's theory of sliding wear, shearing of the asperity junctions can occur in one of the two bodies depending on the relative magnitude of interfacial adhesion strength and the breaking (shearing) strength of surrounding local regions.
- Wear fragments results from such shearing whereas no wear occurs when shearing takes place along the interface
- In another mechanism, plastic shearing of successive layers of an asperity contact results in detachment of a wear fragment.

2) Abrasive Wear

- The term 'abrasive wear' includes two types of wear situation. Known as two-body abrasion and three-body abrasion respectively.
- In both the cases, a soft surface is ploughed by a relatively hard material.
- In two body abrasion a rough hard surface slides against a relatively soft mating surface.
- In three-body abrasion rough hard particles trapped between the two sliding surfaces cause one or both of them to undergo abrasive wear.
- Examples of two-body abrasion are grinding, cutting and machining whereas those of three-body abrasion are free abrasive lapping and polishing.
- In many cases, the wear mechanism at the start is adhesive, which generates wear particles that get sandwiched at the interface, resulting in three body abrasive wear.
- In most abrasive wear situations, scratching is seen as a series of grooves parallel to the direction of sliding (ploughing). Depending on the degree of severity, abrasive wear is also termed as scratching, scoring or goughing.
- In abrasive wear, material may be removed from a surface by several plastic deformation modes.
- These include ploughing, wedge formation and cutting.
- In the ploughing mode, the material is displaced from a groove to the sides and ridges form the sides of the ploughed grooves. These ridges get flattened and finally fracture after repeated loading and unloading cycles.

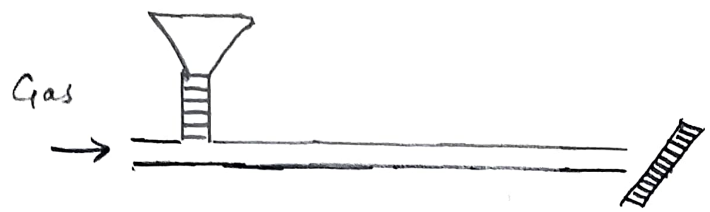
4) b) wear processes may be classified as

- Sliding wear
- Rolling wear
- oscillation wear
- Impact wear
- Erosive wear

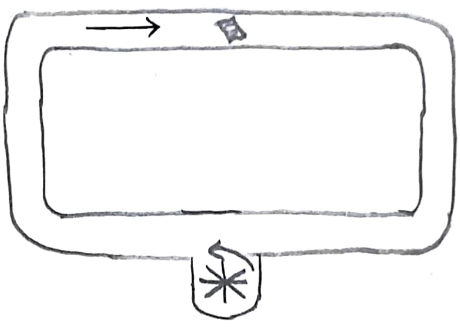
Many different experimental arrangements are used to study sliding wear. These are usually carried out either to examine the process by which wear takes place or to simulate practical situations to generate design data on wear rates and co-efficients of friction.
Close control and monitoring of all the variables which may influence wear are essential if the results of a test are to be useful for wider scientific purposes.
The figure shows the geometrical arrangements in several common types of wear testing apparatus.
For adhesive wear between identical materials, the two surfaces are made of the same material. For abrasive wear testing, one of the surfaces is made of abrasive material, generally the larger one is made of abrasive material. Changes in geometries and arrangements are done for testing of different mechanisms of wear.
For two-body abrasive wear, commercial-bonded abrasive paper or cloth is usually used for the counter-face carrying evenly distributed grit particles of narrow size distribution, bonded to the substrate by a strong resin. In simulating three-body abrasion, silica (quartz) particles of a narrow size distribution and, from a specified source are fed at a constant rate into the contact region.
The figure further shows schematic diagrams of four types of testing methods for erosive wear.

In jet impingement method: (a) particles are accelerated in a fluid stream along a nozzle to strike the target material, which is held some way from the end of the nozzle at a fixed angle.

In recirculating loop test, (b) a two-phase flow of particles and fluid are driven around a loop of pipe work where the specimen is kept completely immersed in the flow.



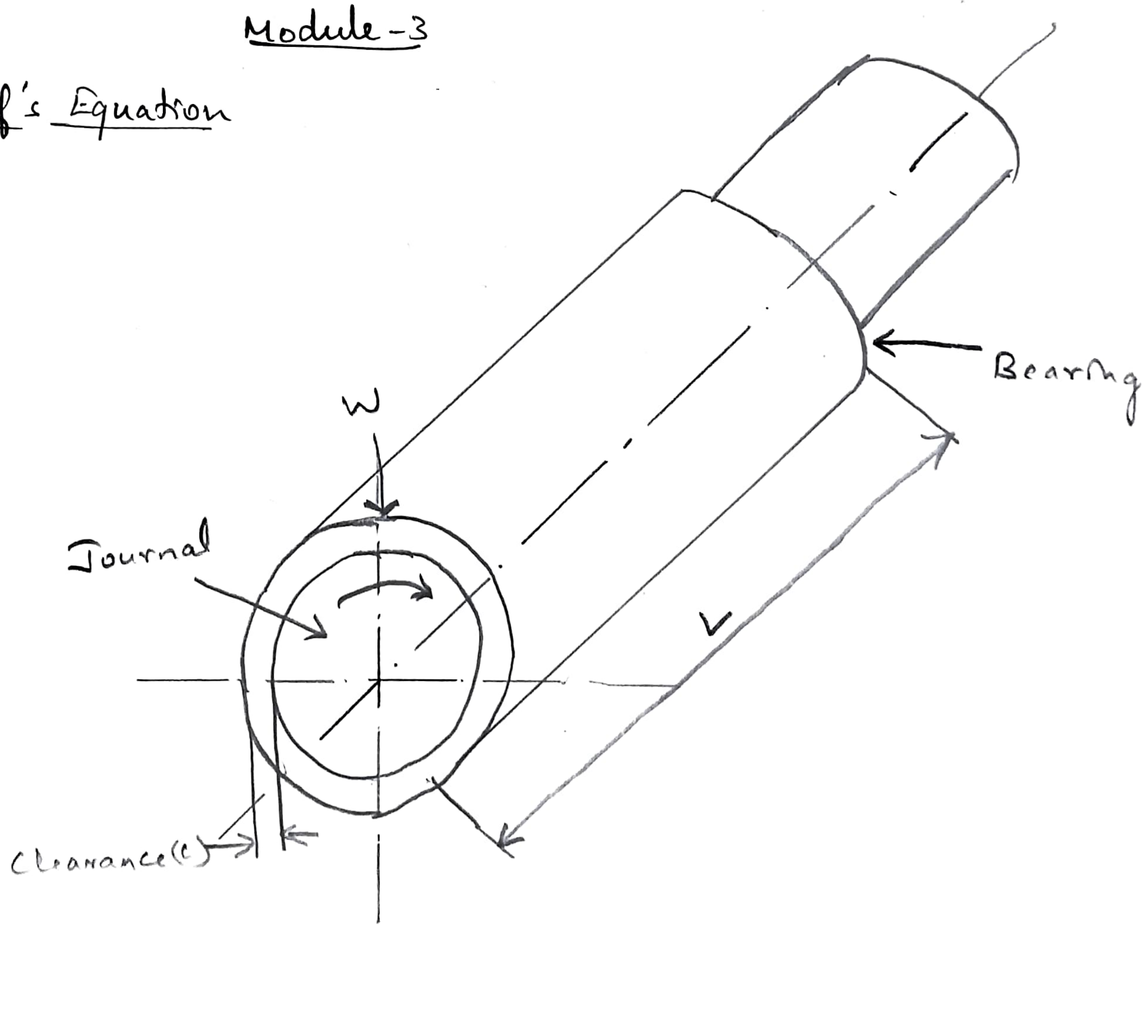
(a)



(b)

Module-3

5) a) Petroff's Equation



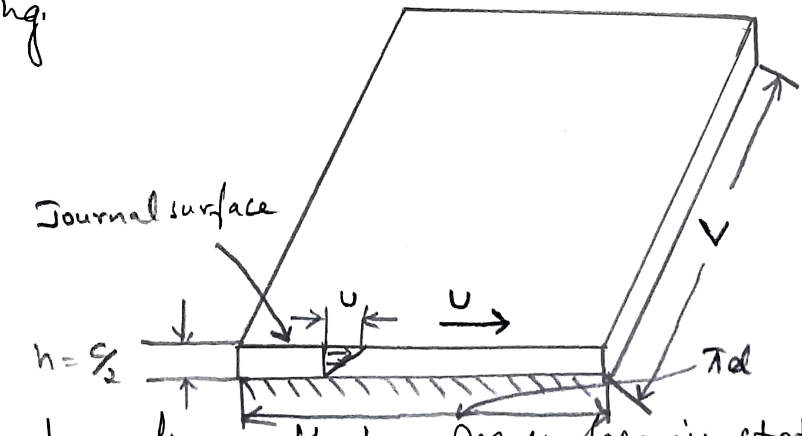
Consider a full journal bearing with a journal running concentrically with the bearing. In actual practice the journal will not run concentrically with the bearing. It is possible only when

1) The radial load acting on the journal is equal to zero
2) The viscosity of the lubricant is equal to infinity
3) The speed of the journal is infinity.

The bearing which ~~satisfaction~~ satisfies the above condition is called as lightly loaded bearing. But above condition is not possible in practice.

Assumptions

1) Viscosity is constant throughout the oil film
2) Fluid obeys newtons law of viscosity.
3) Flow is steady and laminar
4) Fluid is incompressible
5) The oil film is always thin relative to the radius of the bearing. Therefore the curvature of the bearing surface may be ignored and the film may be considered as an unwrapped straight body having a thickness equal to πd and 'a' width 'l' equal to the length of the bearing.



In the above figure the bearing surface is stationary and the journal surface is moving with a constant velocity 'U' relative to the bearing surface.

The shear force or frictional force 'F' resisting the relative motion of the surface is

$$F = \tau \cdot A \quad \text{— (1)}$$

where τ = shear stress in the fluid film
A = Area of the journal surface

According to newtons law of viscous flow

$$\tau = \eta \frac{U}{h} \quad \text{— (2)}$$

where, η = Absolute viscosity
h = Radial clearance $= \frac{c}{2}$

Substituting equ(2) in (1)

$$F = \eta \frac{U}{h} \cdot A \quad \text{— (3)}$$

velocity $U = \frac{\pi d n}{60} = \pi d n' \quad \left(\because n' = \frac{n}{60}\right)$

Radial clearance, $h = \frac{c}{2}$

Surface area of journal $A = \pi d L$

Substituting these values in equ (3)

$$F = \eta \times \frac{\pi d n'}{c/2} \times \pi d L$$

$$\therefore F = \frac{2\pi^2 \eta n' L d}{c/d}$$

$$\boxed{F = \frac{2\pi^2 \eta n' L d}{\psi}} \quad \left(\because \psi = c/d\right) \quad \text{— (4)}$$

Above equ is known as petroff's equ for frictional force.

Frictional torque is given by

$$M_t = F \cdot r \quad \text{— (5)}$$

where, F = frictional force
r = radius of journal = $d/2$

Substituting equ (4) in (5), we get

$$M_t = \frac{\pi^2 \eta n' L . d^2}{\psi} \quad \text{—— (6)}$$

Above equ is known as Petroff's equation for frictional torque

Power loss due to frictional force is

$$N = F . U$$

$$= \frac{2\pi^2 \eta . n' L d}{\psi} \times \pi d n'$$

$$\therefore N = \frac{2\pi^3 \eta (n')^2 L . d^2}{\psi} \quad \text{—— (7)}$$

Above equation is known as Petroff's equation for power loss.

Co-efficient of friction is given by

$$\mu = F/W \quad \text{—— (8)}$$

where F = Frictional force
W = Radial load

Bearing pressure,

$$P = \frac{\text{Radial Load}}{\text{Projected area of the journal}}$$

$$P = \frac{W}{L . d}$$

$$\therefore W = P . L . d$$

Substituting the value of 'W' & 'F' in equ (8)

$$\mu = \frac{2\pi^2 \eta . n' L . d}{\psi \times P L . d} = \boxed{\mu = 2\pi^2 \left(\frac{\eta n'}{P}\right) \frac{1}{\psi}} \rightarrow$$

Petroff's equ for co-efficient of friction.

5.) b.) Given: $d = 45mm = 0.045m$

$l = 65mm = 0.065m$

$W = 800N$

$c = 0.001$

$n = s = 2800rpm = \frac{2800}{60}$

$\eta = 1.2 \times 10^{-6}$ Reynolds $= 8.274 \times 10^{-3}$ Pa-s

a) Friction Torque

$$T = \frac{4\pi^2 r^3 n L \eta}{cr} = \frac{4\pi^2 (2.25\times10^{-3})^3 \left(\frac{2800}{60}\right)(65\times10^{-3})(8.2\times10^{-3})}{0.0015 \times 22.25 \times 10^{-3}}$$

$$T = 0.3343 \text{ Nm}$$

b) Co-efficient of friction $\mu = 2\pi^2 \left(\frac{\eta . n}{P}\right)\left(\frac{r}{cr}\right)$

$$P = \frac{W}{Ld} = \frac{800}{(65\times10^{-3})(22.5\times10^{-3})} = 273504.273 \text{ N/m}^2$$

$$\therefore \mu = 2\pi^2 \left(\frac{8.274\times10^{-3} \left(2800/60\right)}{273504.273}\right)\left(\frac{1}{0.0015}\right) = 0.018566$$

c) Power loss

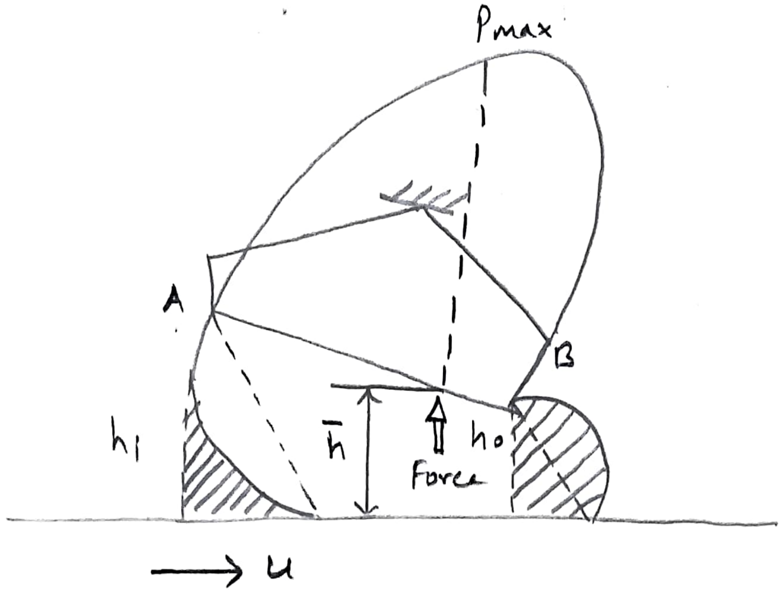
$$P_u = F_v \times U = 2\pi n T_\mu = 2\pi\left(2800/60\right) 0.3343 = 98 \text{ watts}$$

$$= 0.098 \text{ kW}$$

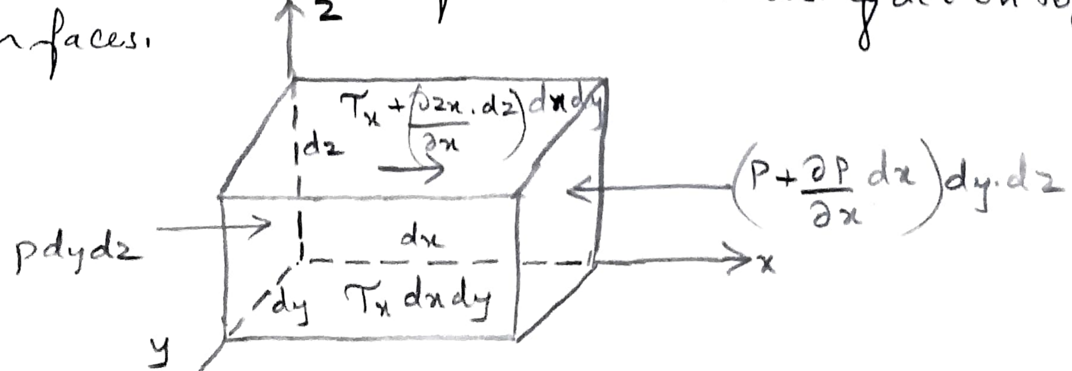
6.) Reynolds equation in 2-Dimension

~~Assumptions~~ made

1.) The inertia forces in the oil film are negligible
2.) The motion of the fluid is laminar
3.) No slip occurs between the lubricant and bearing surface
4.) Fluid is incompressible
5.) Viscosity of the oil is constant throughout the oil film
6.) Lubricant is Newtonian fluid
7.) The clearance between the bearing surfaces is so small that the change in pressure across the clearance can be neglected.
8.) There is no flow in the direction perpendicular to motion (no end leakage)



Consider the equilibrium of a fluid element of dimension dx, dy and dz As shown in fig. Normal forces due to pressure acting on the left and right sides, while shear forces due to viscosity act on top and bottom faces.



A force equilibrium results

$$p\,dy\,dz - \left(p + \frac{\partial p}{\partial x}dx\right)dy.dz + \left(T_x + \frac{\partial T_x}{\partial z}dz\right)dx.dy - T_x\,dx.dy = 0 \quad — (1)$$

By simplification,

$$\frac{\partial T_x}{\partial z} = \frac{\partial P}{\partial x} \quad — (2)$$

A similar force equilibrium in y direction results

$$\frac{\partial T_y}{\partial z} = \frac{\partial P}{\partial y} \quad — (3)$$

By Newton's law of viscous flow

$$T_x = \eta \frac{\partial u}{\partial z} \quad — (4)$$

where, T_x = Shear stress acting in the x-direction, Pa

u = Sliding velocity in x-direction, m/s

Shear stress along y direction $T_y = \eta \frac{\partial v}{\partial z}$ — (5)

T_y = Shear stress acting in y direction, Pa

v = Sliding velocity in y direction, m/s

Substituting equ (4) in (2), we get equilibrium condition for the forces as

$$\frac{\partial p}{\partial x} = \frac{\partial}{\partial z}\left(\eta \frac{\partial u}{\partial z}\right) \quad — (6)$$

Substituting equ (5) in (3)

$$\frac{\partial p}{\partial y} = \frac{\partial}{\partial z}\left(\eta \frac{\partial v}{\partial z}\right) \quad — (7)$$

Since the viscosity of the fluid is assumed to be constant rearranging eqn (6)

$$\frac{\partial^2 u}{\partial z^2} = \frac{1}{\eta}\left(\frac{\partial P}{\partial x}\right) \quad — (8)$$

Integrating twice with respect to z we get velocity distribution as

$$u = \frac{z^2}{2\eta}\frac{\partial p}{\partial x} + C_1 z + C_2 \quad -(9)$$

To determine the values of integration constants C_1 and C_2 we apply boundary conditions

$u = u_2$ at $z = 0$

$u = u_1$ at $z = h$

In the general case there are two velocities u_1 and u_2 corresponding to two surfaces. Applying boundary conditions the values of C_1 and C_2 obtained as

$$C_1 = \frac{u_1 - u_2}{h} - \frac{h}{2\eta}\frac{\partial p}{\partial x}$$

$$C_2 = u_2$$

Substitute in to equ (9) we get,

$$u = \frac{z^2}{2\eta}\frac{\partial p}{\partial x} + \left(\frac{u_1 - u_2}{h} - \frac{h}{2\eta}\frac{\partial p}{\partial x}\right)z + u_2$$

Simplifying, $u = \left(\frac{z^2 - zh}{2\eta}\right)\frac{\partial p}{\partial x} + (u_1 - u_2)\frac{z}{h} + u_2 \quad -(10)$

in the case when $u_2 = 0$, we can write equ (10) as

$$u = \left(\frac{z^2 - zh}{2\eta}\right)\frac{\partial p}{\partial x} + u\frac{z}{h} \quad -(11)$$

In a similar manner a formula for velocity distribution in y direction obtained as

$$v = \left(\frac{z^2 - zh}{2\eta}\right)\frac{\partial p}{\partial x} + (v_1 - v_2)\frac{z}{h} + v_2.$$

$$\therefore \mu = 2\pi^2\left(\frac{\eta n'}{p}\right)\left(\frac{1}{\psi}\right)$$

//

Module-4

7.) a) Given: $B = 0.0762\,m$

$L = 0.065\,m$

$U = 2.54\,m/sec$

$W = 5383.9\,N$

$h_2 = 1.27\times10^{-5}\,m$

$\eta = 0.06805\,Pa\text{-}s$

1) Inclination

$$W = \frac{6\eta UL}{\alpha^2}\left[\ln\left(\frac{a-\alpha}{a}\right) + \frac{2\alpha}{2a-\alpha}\right]$$

$$a = \frac{h_2}{B} = \frac{1.27\times10^{-5}}{0.0762} = 1.67\times10^{-4}$$

$$5383.9 = \frac{6\times0.06805\times2054\times0.065}{\alpha^2}\times\left[\ln\left(\frac{1.67\times10^{-4}-\alpha}{1.67\times10^{-4}}\right) + \frac{2\alpha}{2\times1.67\times10^{-4}-\alpha}\right]$$

$$= \frac{1}{\alpha^2}\left[\ln\left(\frac{1.67\times10^{-4}-\alpha}{1.67\times10^{-4}}\right) + \frac{2\alpha}{3.34\times10^{-4}-\alpha}\right]$$

$$(79867.6\alpha^2) = \ln\left(\frac{1.67\times10^{-4}-\alpha}{1.67\times10^{-4}}\right) + \frac{2\alpha}{3.34\times10^{-4}-\alpha}$$

$\alpha = -04.20\times10^{-3}$

$\alpha = -0.0042$ radians $= -0.2406$ degree

2) Co-efficient of friction

$$\mu = \frac{F_{\mu m}}{W}$$

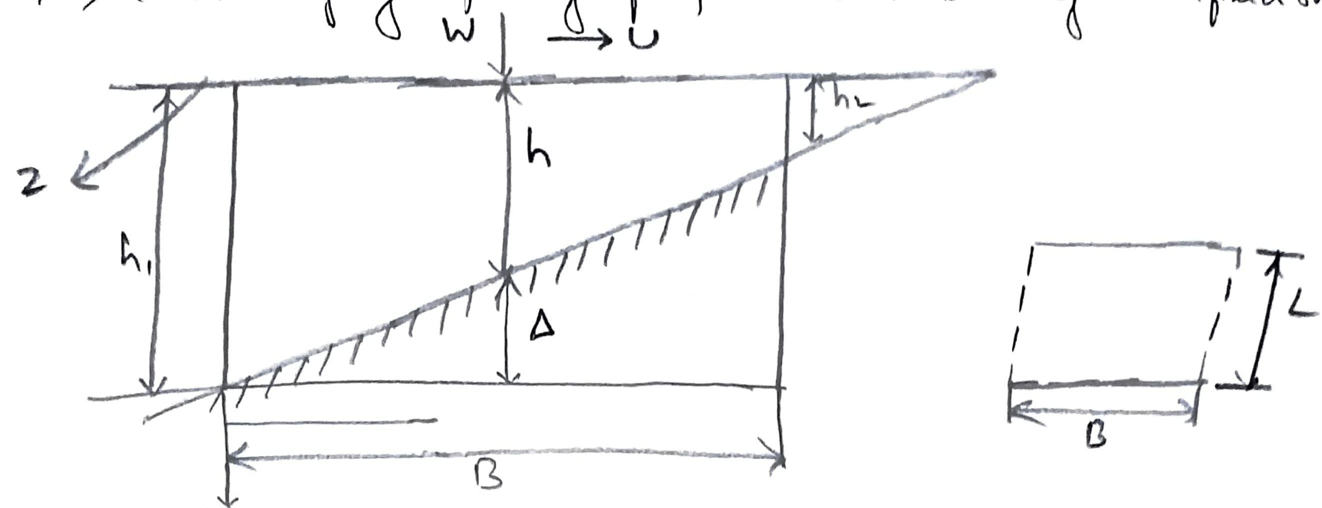
$$F_{\mu m} = \eta UL\left[-\frac{4}{\alpha}\ln\left(\frac{a-\alpha}{a}\right) - \frac{6}{2a-\alpha}\right]$$

$$F_{\mu m} = 0.06805\times2.54\times0.065\times\left[\frac{-4}{-4.2\times10^{-3}}\ln\left(\frac{1.67\times10^{-4}-0.0042}{1.67\times10^{-4}}\right) - \frac{6}{3.34\times10^{-4}+0.0042}\right]$$

$F_{\mu m} = 20.05\,N$

$\mu = \frac{F_m}{W} = \frac{20.05}{5383.9}$ $\quad \mu = 0.00372$ //

7.) b.) Load carrying capacity of a plane slider bearing with fixed shoe



An idealized plane slider bearing consists of two non parallel plane surfaces separated by an oil film. One surface is stationary & the other is moving with uniform velocity U and inclinations are such that a converging oil film is formed between the surfaces. The positive pressure built up in the oil film supports a load. The stationary plate may be fixed or pivoted in this case as shown in fig. 'B' is the length of bearing & 'L' is the width of the bearing.

Step (1) Thickness of fluid film (h)

$h = h_1 - \Delta$

From similar triangles

$$\frac{\Delta}{x} = \frac{h_1 - h_2}{B}$$

$$\Delta = \frac{x(h_1 - h_2)}{B}$$

$$h = h_1 - \frac{x(h_1 - h_2)}{B}$$

$$h = h_1 + x\left(\frac{h_2 - h_1}{B}\right)$$

$\alpha = \frac{h_2 - h_1}{B}$ $\quad h_2 - h_1 = B \cdot \alpha$

$a = \frac{h_2}{B}$ $\quad h_2 = B \cdot a$

$x_1 = \frac{x}{B}$ $\Rightarrow x = Bx_1$

$\therefore h = h_1 + x\alpha$

$h_1 = h_2 - (h_2 - h_1)$

$= Ba - B\alpha$

$= B(a - \alpha)$

$h = h_1 + x\alpha$

$h = B(a - \alpha) + x\alpha$

$h = B(a - \alpha) + Bx_1\alpha$

$h = B(a - \alpha + x_1\alpha)$

Step 2 : Reynolds equation

$$\frac{d}{dx}\left(h^3\frac{dP}{dx}\right) = 6\eta U\frac{dh}{dx}$$

$$x = Bx_1$$

$$dx = B\,dx_1$$

$$\frac{d}{B\,dx_1}\left(h^3\frac{dP}{\cancel{B}dx_1}\right) = 6\eta U\frac{dh}{\cancel{B}dx_1}$$

$$\frac{1}{B}\frac{d}{dx_1}\left(h^3\frac{dP}{dx_1}\right) = 6\eta U\frac{dh}{dx_1}$$

Integrating both sides

$$\int\frac{d}{dx}\left(h^3\frac{dP}{dx_1}\right)dx_1 = \int 6\eta U\frac{dh}{dx_1}\,dx_1$$

$$h^3\frac{dP}{dx_1} + C_1 = 6\eta UBh + C_2$$

$$h^3\frac{dP}{dx_1} = 6\eta UBh + (C_2 - C_1)$$

$$h^3\frac{dP}{dx_1} = 6\eta UB\left[h - \frac{C_1 - C_2}{6\eta UB}\right]$$

$$h^3\frac{dp}{dx_1} = 6\eta UB(h - c)$$

where $c = \dfrac{C_1 - C_2}{6\eta UB}$

$$\frac{dp}{dx_1} = 6\eta UB\left[\frac{h}{h^3} - \frac{c}{h^3}\right]$$

$$\frac{dP}{dx_1} = 6\eta UB\left[\frac{1}{h^2} - \frac{C}{h^3}\right]$$

$$dP = 6\eta UB\left[\frac{dx_1}{h^2} - C\frac{dx_1}{h^3}\right]$$

$$dP = 6\eta UB\left[\frac{dx_1}{B^2(a-\alpha+x_1\alpha)^2} - \frac{C\,dx_1}{B^3(a-\alpha+x_1\alpha)^3}\right]$$

$$dP = \frac{6\eta U}{B}\left[\frac{dx_1}{(a-\alpha+x_1\alpha)^2} - \left(\frac{C}{B}\right)\frac{dx_1}{(a-\alpha+x_1\alpha)^3}\right]$$

Integrating above equation

$$P = \frac{6\eta U}{B}\left[\int \frac{dx_1}{(a-\alpha+x_1\alpha)^3} - \left(\frac{C}{B}\right)\int \frac{dx_1}{(a-\alpha+x_1\alpha)^3}\right]$$

$$\int \frac{dx_1}{(a-\alpha+x_1\alpha)^2} = \int (a-\alpha+x_1\alpha)^{-2}\, dx_1$$

$$= \frac{(a-\alpha+x_1\alpha)^{-2+1}}{-2+1} \times \frac{1}{\alpha} + c'$$

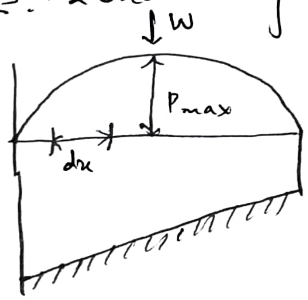
$$= -\frac{1}{\alpha}(a-\alpha+x_1\alpha) + c'$$

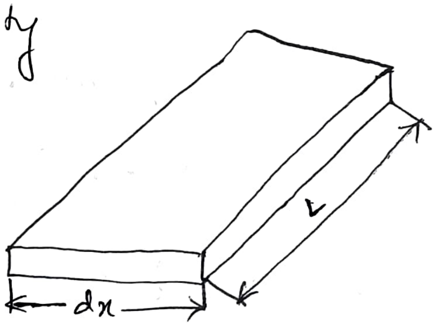
$$\int \frac{dx_1}{(a-\alpha+x_1\alpha)^3} = \int (a-\alpha+x_1\alpha)^{-3}\, dx_1$$

$$= \frac{(a-\alpha+x_1\alpha)^{-2}}{-2} \times \frac{1}{\alpha} + c''$$

$$\therefore P = \frac{6\eta U}{B}\left[\frac{-1}{\alpha(a-\alpha+x_1\alpha)} + \left(\frac{C}{B}\right)\frac{1}{2\alpha(a-\alpha+x_1\alpha)^2} + c' + c''\right]$$

Step 3:- Load carrying capacity





$$W = \int_{x=0}^{x=B} P\, dx\, L$$

$$W = \int_{x=0}^{x=B} \frac{\eta U}{B} \frac{6\alpha x_1(1-x_1)}{(\alpha-2a)(a-\alpha+x_1\alpha)^2}\, dx\, L$$

$$p = \frac{W}{A}$$

$$W = p \times A$$

$$= p \times dx \times L$$

$x = Bx_1$

$dx = B\,dx_1$

$x_1 = \dfrac{x}{B}$

At $x = 0$, $x_1 = \dfrac{x}{B} = 0$

At $x = B$, $x_1 = \dfrac{B}{B} = 1$

$$W = \frac{6\eta U\alpha L}{(\alpha - 2a)B} \int_{x_1=0}^{x_1=1} \frac{x_1(1-x_1)}{(a-\alpha_1+\alpha x_1)^2} B\,dx_1$$

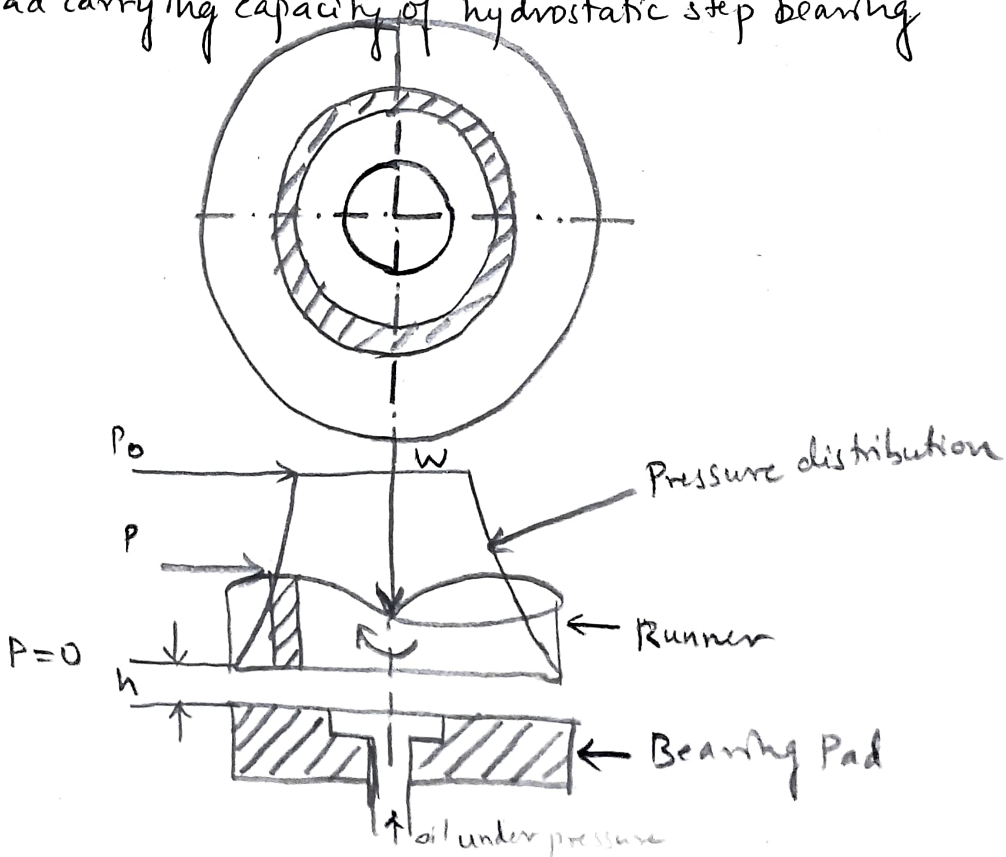
$$W = \frac{6\eta U\alpha L}{(\alpha - 2a)} \int_{x_1=0}^{x_1=1} \frac{x_1(1-x_1)}{(a-\alpha_1+\alpha x_1)^2} dx_1$$

Integrating & simplifying the above equation

$$W = \frac{6\eta UL}{\alpha^2} C_{s2}$$

where $C_{s2} = \ln \dfrac{a-\alpha}{a} + \dfrac{2\alpha}{2a-\alpha}$ //

8) a) Load carrying capacity of hydrostatic step bearing



Consider a circular pad hydrostatic bearing in which oil is pressurized by a pump. The bearing pad has a recess of radius r_1 and an annular sill area between radius r_1 to r_2 as shown in Fig.
The runner having weight W and rotational speed N, is supported on the bearing pad. The oil flows from the centre of the bearing pad, moves radially out in the annular portion and having a film thickness of 'h'. The flow of oil between the bearing pad and the runner can be regarded as the flow betn the parallel plates, for which the expression for flow ~~be~~ can be given as

$$Q = \frac{Bh^3}{12\eta} \cdot \frac{dp}{dx}$$

The same equation in cylindrical co-ordinates can be written for an elemental annulus of radius r and annular thickness dr with the following substitutions

$B = 2\pi r$ and $\frac{dp}{dx} = -\frac{dp}{dr}$ (-ve sign because the pressure oil decreases with increase in radius)

$$Q = \frac{2\pi r h^3}{12\eta}\left(-\frac{dp}{dr}\right) \quad = \quad Q = -\frac{\pi r h^3}{6\eta}\frac{dp}{dr}$$

$dp = -\frac{6\eta Q}{\pi r h^3}\,dr$ Integrating this expression w.r.t 'r'.

$p = -\frac{6\eta Q}{\pi h^3}\int \frac{dr}{r}$ $\quad p = -\frac{6\eta Q}{\pi h^3}\ln r + C$ where 'C' is constant of integration and can be determined by applying boundary condition. (i) at $r = r_1 \Rightarrow p = p_0$ & (ii) $r = r_2 \Rightarrow p = 0$

Applying the (ii) boundary condition.

$$0 = -\frac{6\eta Q}{\pi h^3}\ln r_2 + C$$

$$\therefore C = \frac{6\eta Q}{\pi h^3}\ln r_2$$

Hence $p = -\frac{6\eta Q}{\pi h^3}\ln r + \frac{6\eta Q}{\pi h^3}\ln r_2$

$$p = \frac{6\eta Q}{\pi h^3}(\ln r_2 - \ln r)$$

$$p = \frac{6\eta Q}{\pi h^3} \ln\left(\frac{r_2}{r_1}\right) \quad \text{— (1)}$$, this is the expression for pressure distribution

Applying boundary condition (i)

The pressure required in the pocket to support the load

$$p_0 = \frac{6\eta Q}{\pi h^3} \ln\left(\frac{r_2}{r_1}\right) \quad \text{— (2)}$$

The rate of flow of lubricant through the bearing

$$Q = \frac{\pi p_0 h^3}{6\eta \ln\left(\frac{r_2}{r_1}\right)} \quad \text{— (3)}$$

Substituting the value of Q from (3) in equation (1),

$$p = \frac{6\eta}{\pi h^3} \ln\left(\frac{r_2}{r_1}\right) \frac{\pi p_0 h^3}{6\eta \ln\left(\frac{r_2}{r_1}\right)}$$

$$p = p_0 \frac{\ln\left(\frac{r_2}{r}\right)}{\ln\left(\frac{r_2}{r_1}\right)}$$

Total load carried by the bearing pad = load carried by the recess (pocket) + load carried by the sill area.

$$W = p_0 \pi r_1^2 + \int p \, dA$$

$$W = p_0 \pi r_1^2 + \int_{r_1}^{r_2} p \times 2\pi r \, dr$$

$$= p_0 \pi r_1^2 + 2\pi \int_{r_1}^{r_2} p_0 \frac{\ln\left(\frac{r_2}{r}\right)}{\ln\left(\frac{r_2}{r_1}\right)} r \, dr$$

$$= p_0 \pi r_1^2 + \frac{2\pi p_0}{\ln\left(\frac{r_2}{r_1}\right)} \int_{r_1}^{r_2} \ln\left(\frac{r_2}{r}\right) r \, dr$$

$$= p_0 \pi r_1^2 + \frac{2\pi p_0}{\ln\left(\frac{r_2}{r_1}\right)} \left[\left\{ \ln\left(\frac{r_2}{r}\right) \frac{r^2}{2} \right\} - \int_{r_1}^{r_2} \frac{r^2}{2} \left(\frac{-1}{r}\right) dr \right]_{r_1}^{r_2}$$

$$= p_0 \pi r_1^2 + \frac{2\pi p_0}{\ln\left(\frac{r_2}{r_1}\right)} \left[\frac{r_2^2}{2} \ln\left(\frac{r_2}{r_2}\right) - \frac{r_1^2}{2} \ln\left(\frac{r_2}{r_1}\right) + \int_{r_1}^{r_2} \left(\frac{r}{2}\right) dr \right]$$

$$= p_0 \pi r_1^2 + \frac{2\pi p_0}{\ln\left(\frac{r_2}{r_1}\right)} \left[\frac{-r_1^2}{2} \ln\left(\frac{r_2}{r_1}\right) + \left(\frac{r^2}{4}\right)^{r_2} \right]$$

$$= p_o \pi r_1^2 + \frac{2\pi p_o}{\ln\left(\frac{r_2}{r_1}\right)}\left[\frac{-r_1^2}{2}\ln\left(\frac{r_2}{r_1}\right) + \left(\frac{r_2^2 - r_1^2}{4}\right)\right]$$

$$\therefore W = \frac{\pi p_o (r_2^2 - r_1^2)}{2\ln\left(\frac{r_2}{r_1}\right)} \quad — (4)$$

$$W = \frac{\pi p_o}{2\ln\left(\frac{d_2/2}{d_1/2}\right)}\left(\frac{d_2^2}{4} - \frac{d_1^2}{4}\right)$$

Load carrying capacity, $W = \dfrac{\pi p_o (d_2^2 - d_1^2)}{8\ln\left(\frac{d_2}{d_1}\right)}$ — (5)

$\therefore$ The pressure required in the pocket to support the load

$$p_o = \frac{8W\ln\left(\frac{d_2}{d_1}\right)}{\pi(d_2^2 - d_1^2)} \quad — (6)$$

8) b) Given : $n = 100$ rpm , $n' = 1.667$ rps

$p_o = 500 \times 10^3$ N/m^2

$\eta = 0.05$ pas

$h = 0.00007$ m

1) Load carrying capacity

$$W = \frac{p_o \pi (d_2^2 - d_1^2)}{8\ln(d_2/d_1)}$$

$$= \frac{500 \times 10^3 \times \pi(0.3^2 - 0.2^2)}{8\ln(3/2)}$$

$W = 24.21$ KN //

2) Oil flow rate (Q)

$$Q = \frac{\pi p_o h^3}{6\eta \ln\left(\frac{d_2}{d_1}\right)} = \frac{\pi \times 5 \times 10^5 \times 0.00007^3}{6 \times 0.05 \times \ln\left(\frac{0.3}{0.2}\right)} = 4.429 \times 10^{-6} \text{ m}^3/\text{sec} //$$

3) Power loss

$$P_L = \frac{0.062\,\eta (n')^2}{16h} (d_2^4 - d_1^4)$$

$$= \frac{0.062 \times 1.667^2 \times 0.05}{16 \times 0.00007} (0.3^4 - 0.2^4)$$

$$= 0.05 kW$$

Module - 5

9) a) The ten desirable properties of bearing materials

1. Compatibility
2. Conformability / Embedability
3. Compressive strength
4. Fatigue strength
5. Corrosion resistance
6. Co-efficient of friction
7. Thermal Conductivity
8. Co-efficient of expansion / Thermal expansion
9. Modulus of Elasticity.
10. Wettability
11. Bondability

1) Compatibility: Some bearing materials have a tendency to weld to the shaft when there is a contact between the journal and bearing surfaces. In some materials this tendency is more in other materials it is less. The measure of anti-welding property of a bearing material, operating with a given journal material is called compatibility.

2) Conformability / Embedability: It is ability of the bearing material to accomodate shaft deflections and bearing inaccuracies by plastic deformation or (creep) without excessive wear and heating. Embedability is the ability of bearing material to accomodate or embed small particles of dust, grit etc without scoring the material of the journal.

3) Compressive strength; The maximum bearing pressure is considerably greater than the average pressure obtained by dividing the load to the projected area. Therefore the bearing material should have high compressive strength to withstand this maximum pressure so as to prevent extrusion or other permanent deformation of the bearing.

4) Fatigue strength: The bearing material should have sufficient fatigue strength so that it can withstand repeated loads without developing surface fatigue crack. It is of major importance in aircraft and automotive engines.

5.) Corrosion resistance: The bearing material should not corrode away under the action of lubricating oil. This property is of particular importance in internal combustion engines where the same oil is used to lubricate the cylinder walls and bearings. In the cylinder, the lubricating oil comes into contact with hot cylinder walls and may oxidise and collect carbon deposits from the walls.

6.) Thermal conductivity: The bearing material should be of high thermal conductivity so as to permit the rapid removal of the heat generated by friction.

7.) Thermal expansion: The bearing material should be of low co-efficient of thermal expansion.

8.) Bondability: Many high capacity bearings are made by bonding one or more thin layers of a bearing material to a high strength steel shell. Thus, the strength of the bond i.e. bondability is an important consideration in selecting bearing material.

9) Wettability :- Bearings having good wetting properties are desirable inorder to be compatible with lubricants.

10) Modulus of Elasticity :- The good bearing material should have high modulus of elasticity inorder to accomodate the load applied.

9) b) The most common bearing materials used in practice are

1.) **Babbit metal**: The tin base and lead base babbits are widely used as a bearing material, because they satisfy most requirements for general applications. The babbits are recommended where the maximum bearing pressure (on projected area) is not over 7 to 14 N/mm^2; when applied in marine bearings, automobiles the babbit is generally used as a thin layer, 0.05mm to 0.15mm thick, bonded to an insert or steel shell.

The composition of the babbit metals is as follows

Tin base babbits: Tin 90%; Copper 4.5%; Antimony 5%; Lead 0.5%

Lead base babbits: Lead 84%; Tin 60%; Antimony 9.5%; Copper 0.5%

2.) **Bronzes**: The bronzes (alloys of copper, tin and zinc) are generally used in the form of machined bushes pressed into the shell. The bush may be in one or two pieces. The bronzes commonly used for bearing material are gun metal and phosphor bronzes.

The gun metal Copper 88%; Tin 10%; Zinc 2% is used for high grade bearings subjected to high pressures and high speeds.

~~The gun metal (Copper 88%; Tin 10%; ... ; ...)~~

The phosphor bronze (Copper 80%; Tin 10%; Lead 9%; Phosphorus 1%) is used for bearing subjected to very high pressures (not more than 14 N/mm^2 of projected area) and speeds.

3.) **Cast Iron**: The cast iron bearings are usually used with steel journals. Such type of bearings are fairly successful where lubrication is adequate and the pressure is limited to 3.5 N/mm^2 and speed to 40 metres per minute.

4.) **Silver**: The silver and silver lead bearings are mostly used in aircraft engines where the fatigue strength is the most important consideration.

5.) **Non-metallic bearings**: The various non-metallic bearings are made of carbon-graphite, rubber, wood and plastics. The carbon-graphite bearings are self lubricating, dimensionally stable over a wide range of operating conditions, chemically inert and can operate at higher temperatures than other bearings.

10) a) Surface Engineering: Surface engineering refers to wide range of technologies designed to modify the surface properties of metallic and non-metallic components for decorative and/or functional purposes. Examples include improving corrosion and wear resistance to extend component life; making items more visually attractive and giving special properties such as lubricity enhancement, non-stick surfaces etc.

Surface Engineering for tribological applications aims at two basic objectives
a) to increase the wear resistance of the surface material
b) to modify the frictional behaviour.

If a component is not completely separated from its counter face by a fluid film, its tribological behaviour critically depends on the properties of the contacting surfaces and hence the choice of contacting materials becomes important. Thus, a material selected on the basis of strength and bulk properties may be unsuitable for its tribological applications.
It is well known that wear resistance of a surface increases with hardness and hence to resist wear one should go for a material of high hardness. However, high hardness is often accompanied by brittleness and low-impact strength which may make the material unsuitable for the bulk of the component.
On the other hand, materials like self lubricating solids having low frictional wear characteristics often have low mechanical strength. Also materials with good tribological characteristics are often very expensive and difficult to fabricate. Sometimes the cost of using such tribologically good materials to manufacture bulk components prohibits their use. The methods used in surface engineering may be broadly divided into two categories
a) Surface treatments
b) Surface coatings

10.) b.) The different techniques adopted for surface modifications are
a) Hard facing
b) Vapour deposition
c) Miscellaneous deposition processes.

Hard facing is used for depositing thick coatings typically more than 50μm of hard wear resistant materials.
Vapour deposition techniques are used to deposit thin and reproducible coatings with excellent adhesion and significant flexibility. Miscellaneous deposition processes are widely used for application of polymer coatings, non-metallic coatings and composite coatings for wear and corrosion resistance.

1.) Hard Facing: Hard facing is done by thermal spraying, welding or cladding. Thermal spraying techniques are chosen for applications that require hard coatings applied with minimum thermal distortion of the part and good process control.
Welding processes are restored to for applications that require dense / relatively thick lining of wrought material.
Thermal spraying is practically applicable to depositing any material but welding and cladding are mostly applicable to depositing metals and alloys.
Thermal spraying and welding require the parts to be post finished to attain proper roughness, but cladding does not require this. All these hard facing techniques are line of sight processes and hence it is difficult to apply them on complex geometry parts.

Thermal spraying is one of the most widely used methods for deposition of coating materials. Coating material is melted in a heating zone and then sprayed to the pretreated, coded base material.
The thermal energy required for melting the spraying material can be obtained from a flame created by combustion gases, detonation of combustion gas by a spark plug or an electric plug or electric plasma produced by electric discharge.

2) Vapour Deposition Processes

Many soft and hard coatings can be deposited from the vapour phase. The three types of vapour deposition techniques are as follows.

a) Physical vapour deposition (PVD)
b) Chemical vapour deposition (CVD)
c) Plasma enhanced chemical vapour deposition (PECVD)

These vapour deposition processes are versatile in terms of coating materials. Very thin coating of a few nanometer thick with high purity, high adhesion and exceptional microstructures are possible. They reproduce surface topography and require no post finishing.
The requirement of vacuum systems increases the cost of such processes.