MODULE 2

II	Rolling contact bearing- Design of bearings, Types, Selection of a bearing type, bearing life, static and dynamic load capacity, axial and radial loads, selection of bearings, dynamic equivalent load	4	15%
	Sliding contact bearing- lubrication, lubricants, viscosity, Journal bearings, hydrodynamic theory, Sommerfield number, design considerations, heat balance, bearing housing and mountings	4	

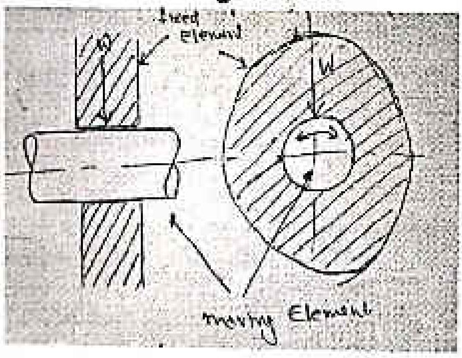
BEARING

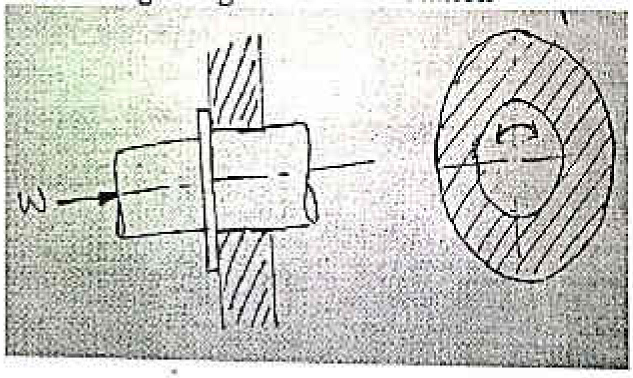
Bearing is a machine element which supports another moving machine element (known as journal). It permits relative motion between the contact surfaces of the member while carrying the load.

BEARING CLASSIFICATION

1. ***Depending upon direction of load to be supported***

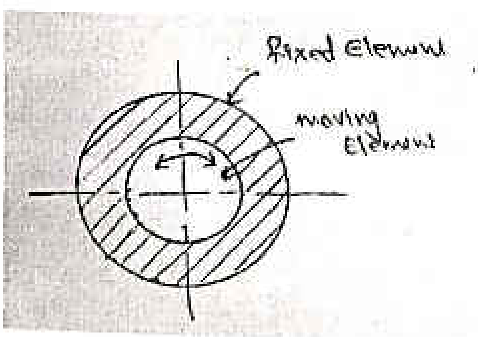
(a) Radial bearing	(b) Thrust or axial bearing
Load acting perpendicular to the direction of motion of moving element	Load acting along the axis of rotation





2. **Depending upon the nature of contact**

(a) Sliding contact bearing	(b) Rolling contact bearing
Sliding take place along the surface of the contact between moving element and fixed element	Steel balls or rollers are interposed between moving and fixed element



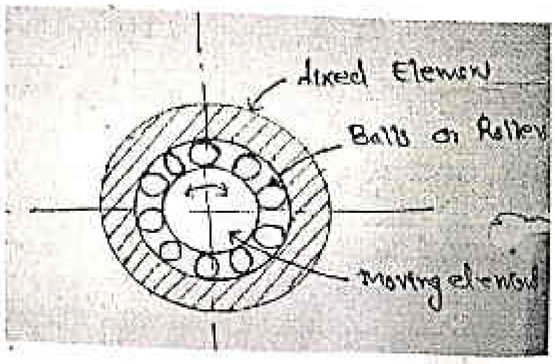
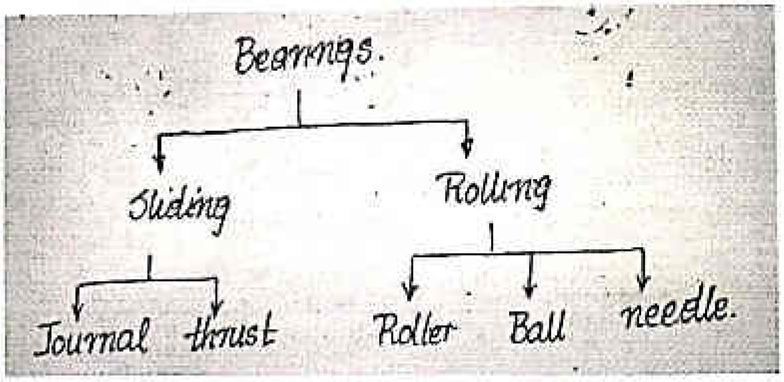


Table 1.1 Classifications of bearings

Sliding contact bearings	Rolling contact bearings
1. Zero film bearings (Bearings without lubricant) 2. Thin film bearings (Boundary lubricated bearings)	1. Ball bearings a) Deep groove ball bearings b) Self-aligning ball bearings c) Angular contact ball bearings i) Single row angular contact ii) Double row angular contact d) Thrust bearings i) Single thrust ii) Double thrust
3. Thick film bearings Hydro - dynamic bearings) a) Full journal bearings b) Partial journal bearing	2) Roller bearings a) Cylindrical roller bearings b) Spherical roller bearings c) Taper roller bearings d) Needle roller bearings e) Thrust roller bearings
4) Externally pressurised bearings (Hydro-statio bearings) a) Oil journal bearing b) Air journal bearings	
5) Pivot bearings	
6) Collar bearings	

SLIDING CONTACT MATERIALS

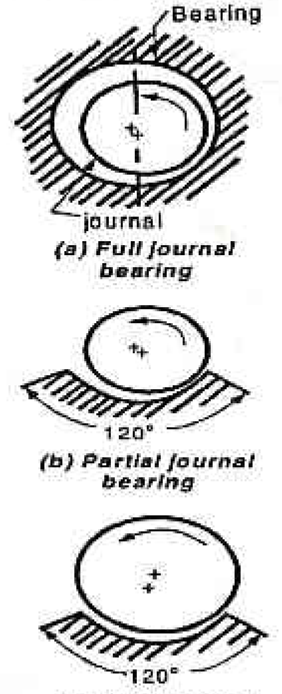
Aluminum alloy, copper alloy, Babbitt alloys, silver, cast-iron, steel



SLIDING CONTACT BEARING

(a) Journal or sleeve bearing

- When sliding action of moving element along the circumference of a circle, and carrying the radial load then the bearing known as **journal bearing**
- **Full journal bearing:** When angle of contact with bearing with the journal is 360 degree, then the bearing known as **full journal bearing.**
- **Partial journal bearing:** When angle of contact with bearing with the journal is 120 degree, then the bearing known as **partial journal bearing.** Less friction than full journal bearing.
- **Clearance bearing-** diameter of journal less than the bearing
- When partial journal bearing has no clearance is called **fitted journals bearing.**



(a) Full journal bearing
(b) Partial journal bearing
(c) Fitted journal bearing

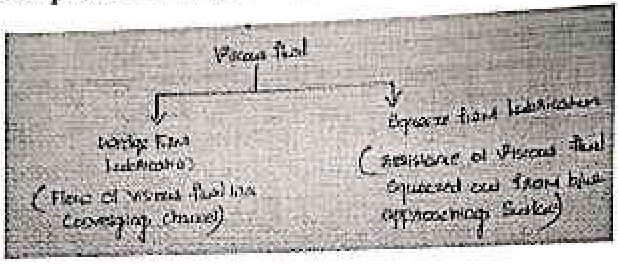
Fig. 2.13. Sliding and rolling contact bearings.

According to thickness of layer of lubricant

1. **Thick film bearing:** Working surface complete separated from each other by lubricant. It divided in to two.
   (a) **Hydrostatic bearing:** Support steady load without any relative motion between journal and bearing. This is achieved by forcing pressurized lubricant between members
   (b) **Hydro dynamic bearing:** Atleat one surface tends to move relative to each other and relative motion between the journal and bearing used to pressurized the lubricant in wedge between the faces
2. **Thin film bearing:** Lubricant is present in working surface partially contact each other at least part of time.
3. **Zero film bearing:** Operate without lubricant

Hydrodynamic bearing

- Thick film lubricant between journal and the bearing
- Fluid pressure developed by sufficient lubricant. In clearance space journal rotating about an axis.
- Load supported by this fluid pressure without any actual contact between journal and bearing



Assumption

1. Lubricant obey Newton's law of viscous flow
2. Lubricant assumed to be in compressible
3. Pressure assumed to be constant though out the film thickness
4. Viscosity assumed to be constant though out film

Important factors for formation of thick film in hydrodynamic bearing

1. Continues supply of oil
2. Relative motion between two surfaces in direction approximately tangential to surface.

Ability of one of surface to take up a small inclination to other surface in direction of relative motion

Lubrication

- In order to reduce friction wear and heating of bearing and journal that move relative to each other lubrication are provided between them.

Types of lubrication (lubrication theory)

1. Full Film Lubrication
(a) Hydrostatic Lubrication (b) Hydrodynamic Lubrication

2. Elasto-Hydrodynamic Lubrication

1. Full Film Lubrication
(a) Hydrostatic Lubrication

- Hydrostatic lubrication is defined as a system of lubrication in which the load supporting fluid film separates the two surfaces by using an external source, like a pump, supplying sufficient fluid under pressure.
- Since the Lubricant is supplied under pressure; this type of bearing is called externally pressurized bearing. Initially the shaft rests on bearing surface.
- As the pump starts, the high pressure fluid is admitted into the clearance space, forcing the surfaces of bearing and journal to separate out. These bearings are used in vertical turbo generators.

(b) HYDRODYNAMIC THEORY OF LUBRICATION | Page 361 of design of data Book

WEDGE FILM LUBRICATION

- Journal bearing work on base theory of converging film.
- When load is supported on film of lubricant
- fig (a) at rest and fig (b) start position there is contact between journal and bearing bush
- The load act downward and when journal is at rest , it will touch bottom of bearing and as journal starts rotating in clock wise
- It deflects to the right side of bearing and thin film of lubricant is found between the contact surfaces.
- As journal keeps rotating, it will draw more oil between the surface, resulting a thick film of lubricant

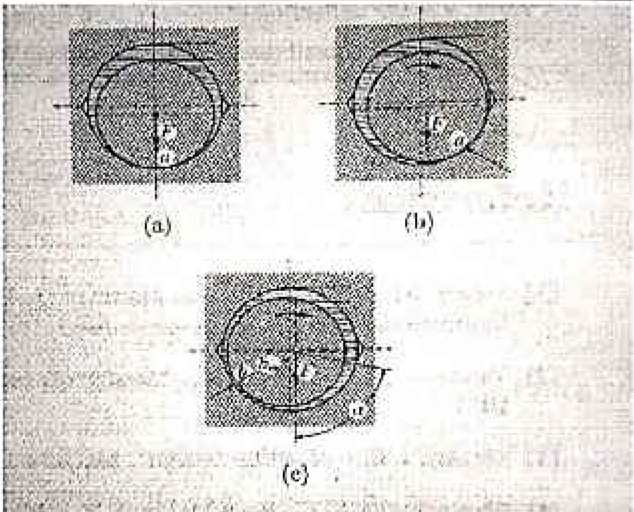


Fig. 15.18: *Oil film at various phases of rotation of a journal*

- The oil drawn under two journal build up an internal pressure, which lift the journal upwards and to the rigid and an equilibrium state achieved.
- Due to leakage of lubricant from end of the bearing, the pressure in the middle is axial direction is maximum and towards zero.

(c) Elasto-Hydrodynamic Lubrication

- Elasto hydrodynamic Lubrication is a type of hydrodynamic lubrication in which significant elastic deformation of the surface takes place due to high fluid film pressure and the surfaces to be separated are not sufficiently rigid.
- In this type, the lubricant used gets dragged into the interface and builds pressure.
- This build-up pressure increases the viscosity of the lubricant and enhances the hydrodynamic film formation by elastic deformation of the surface.
- Eg: rubber seals, gear teeth.

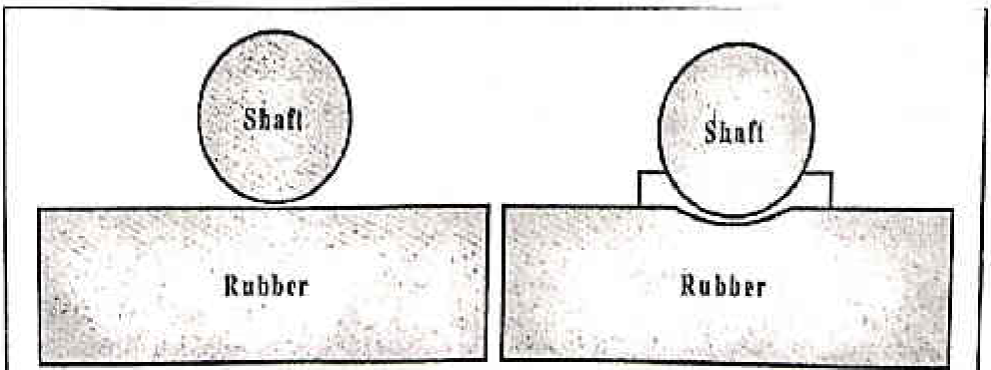


Fig: 2.15 Elastic Deformation of Rubber

Lubricants

- A lubricant is any substance when applied between relative moving bodies will reduce friction and wear and also carry away heat generated.

Types of lubricant

- Solid lubricant
  Eg: Graphite, Mica, Soap, wax
- Semi-liquid
  Eg: Grease
- Liquid Lubricant
  Eg: Vegetable Oil, Mineral oils, Synthetic oils

Property of lubricants

1. Oiliness

It is the property of lubricant and bearing surface in contact. It is the measure of ability of maintains unbroken oil film between along surface. It is used to denote to degree of variation of viscosity with change in temperature.

2. Viscosity

- It is the measure of internal resistance of a fluid. The absolute viscosity also called dynamic viscosity.
- In CGS measured poise and Pas in SI unit.

Z= absolute viscosity in Pas

1 centipoises ≡ 10^{-3} Ns/m^2 (Pascal seconds)

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Viscosity: Newton's law of viscous flow states that the shearing stress in the fluid is proportional to the rate of change of velocity (Fig. 15.1)	$\tau = \frac{F}{A} = Z\frac{dv}{dy}$	15.1(a)
If the rate of shear is constant then the shearing stress, MN/m^2 (kgf/mm^2)	$\tau = \left(Z\frac{v}{h}\right) \times 10^{-3}$	15.1(b)

where $\frac{dv}{dy}$ = rate of change of velocity with distance and may be called the rate of shear or the velocity gradient

Z is the absolute or dynamic viscosity of oil, Ns/m^2 ($P_{a\text{-}s}$)

v = velocity, m/s; h = thickness of lubricant, mm

The kinematic viscosity based upon seconds saybolt (S') or Saybolt Universal Viscosity (SUV) in seconds

(i) in centistoke (cm^2/s) $Z'_k = \left\{0.22S' - \frac{180}{S'}\right\}$ 15.1(c)

(ii) in m^2/s $Z_k = \left\{0.22S' - \frac{180}{S'}\right\} \times 10^{-6}$ 15.1(d)

where S' is the number of seconds saybolt, sec

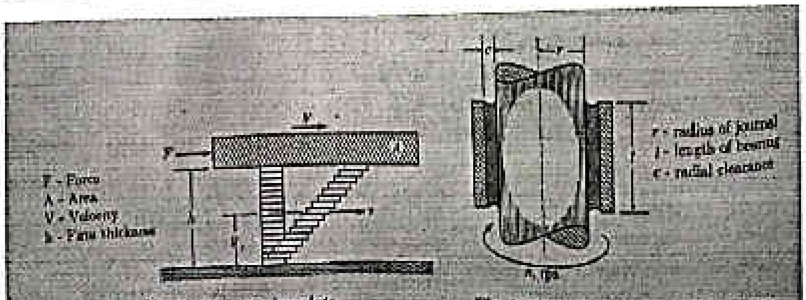

Fig. 15.1: *Load on moving plate (Newton's viscous effect)*

Fig. 15.2(a): *Loaded journal in bearing*

3. Dencity

- The property has no lubricating value but useful in changing kinematic viscosity to absolute viscosity
- Absolute viscosity(CENTI POISE) = $\rho \times$ kinematic viscosity (m/s^2)

ρ = density of lubricating oil

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The absolute or dynamic viscosity of oil in SI units (Ns/m^2) or pascal-seconds ($P_{a\text{-}s}$).	$Z = \rho Z_k = \rho\left\{0.22S' - \frac{180}{S'}\right\} \times 10^{-6}$	15.1(e)
Specific gravity of oil at any temperature, $t°C$	$\rho_t = \rho_{15} - 0.00063(t° - 15)$ or $= \rho_{15} - 0.000657(t°)$	15.1(f)
where ρ_{15} is the specific gravity of oil at 15°C (Table 15.1)		

4. Viscosity index: it used to denote the degree of variation of viscosity with temperature.

5. Flash point: lowest temperature at which an oil give off sufficient vapour to support a momentarily flash without actually setting fire to the oil when flame is brought within 6mm at the surface of oil

6. Fire point: temperature at which oil give off sufficient vapour to burn it consciously when ignited

7. Pour point of freezing point: it temperature at which an oil will cease to flow when cooled.

Properties of sliding contact material

1. **Compressive strength:** Enough compressive strength to withstand maximum pressure developed by radial load
2. **Fatigue strength:** good fatigue strength to overcome failure due to change of load in magnitude and direction
3. Conformability
4. Corrosion resistance
5. Thermal conductivity
6. High thermal conductivity.
7. Easily available and cheaper

- ***Journal bearing material (page 364 , table 15.3)***
- ***Loaded in journal bearing(page 351, fig 15.2(b))***

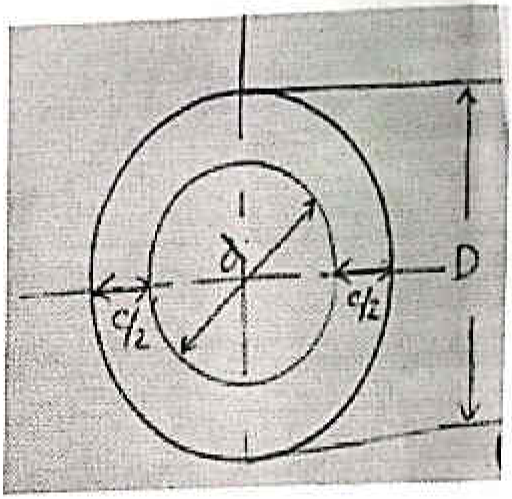
IMPORTANT TERMS IN HYDRODYNAMIC JOURNAL BEARING

1. **Diametrical clearance (c_d)**

$$c_d = D - d$$

Where D = diameter of bearing
d = diameter of journal

$$Radial\ clarenace, c = \frac{c_d}{2}$$



2. **Diametrical clearance ratio (φ)**

$$\varphi = \frac{c_d}{d}$$

3. **Eccentricity ratio** $\in = \frac{e}{c} = cos\emptyset = 1 - \frac{h_0}{c}$......................(page 357, $eq\ 15.7(b)$

4. **Bearing characteristic number**

The coefficient of friction indesin bearing is great importance. Because itafford a means for determing the loss of powerdue to bering friction. Itshown by experiment that coefficient of friction for full lubricating journal bearing a function of three variable.

l Bearings 353

Particular	Equation	Eqn. No.
Petroff's relation to determine the coefficient of friction for lightly loaded bearing	$f = (2\pi^2 \times 10^{-6})\left(\frac{Zn}{p}\right)\left(\frac{r}{c}\right)$	15.4(a)

Where f = coefficient offriction
Z= Absolute vicocity of lubricant in kg/ms
N= speed of jounal (rev/sec)
P= bering pressure (N/mm^2)
r= radius
c= radial clearance

In above reltion ship , the quantity $\frac{Zn}{p}$ is termeed as bearing characteristic number and it is dimensionless number

5.Beaing modulus

The value of coefficient of friction varies with variation of bearingcharacteristic number $\frac{Zn}{p}$.

The value $\frac{Zn}{p}$ for which coefficient of friction is minimum identified as bearingmodulus

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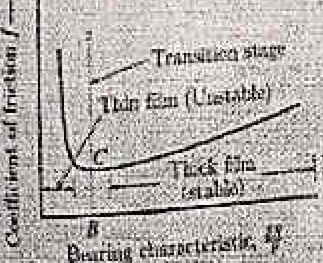

Fig. 15.5(a): Variation of the coefficient of friction with Bearing

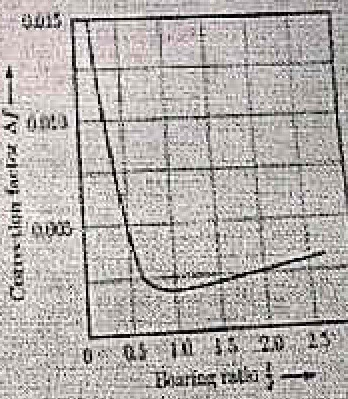

Fig. 15.5(b): Correction factor for Eq. 15.4(b)

6. coefficient of friction

- The coefficient of friction can be expressed by either (a)Petroffs equation or (b) makees equation

Particular	Equation	Eqn. No.
Petroff's relation to determine the coefficient of friction for lightly loaded bearing	$f = (2\pi^2 \times 10^{-9})\left(\frac{Zn}{p}\right)\left(\frac{r}{c}\right)$	15.4(a)
where the minimum value of Zn/p is called bearing modulus or bearing characteristic (Fig. 15.5(a))		
The empirical equation developed by Mc Kee, S.A., and Mc Kee, T.R., that is used for estimating values of coefficient of friction for well lubricated bearings	$f = K_a\left(\frac{Zn}{p}\right)\left(\frac{r}{c}\right) \times 10^{-10} + (\Delta f)$	15.4(b)
where $K_a = 541.33\,\beta$, where β is the circumferential length of the bearing in deg $= 0.195 \times 10^6$ for a full bearing i.e., $\beta = 360°$ $(\Delta f) = 0.002$ for bearing having $l/d = 0.75$ to 2.8 (Fig. 15.5(b))		
When the bearings are partially lubricated, the coefficient of friction according to Louis Illmer	$f = \frac{C_1 C_2}{269.44}\sqrt[4]{p_a/v}$ SI Units	15.4(c)
	$= \frac{C_1 C_2}{152.26}\sqrt[4]{p_a/v}$ Metric units	15.4(d)
average p_a is the pressure on projected area, but must never be assumed to be less than one-half of the maximum pressure imposed during a complete revolution, MN/m² (kgf/mm²) v = rubbing velocity of the journal, m/s C_1 = constant (Table 15.4); C_2 = constant (Table 15.5)		

7. Sommerfeld number (S)

- Sommerfield number is a dimensionless parameter used extensively in the design of journal bearing...............Mathematically

The Sommerfeld number (s/min)		
(a) without side leakage	$S = \left(\frac{r}{c}\right)^2 \left(\frac{ZN}{p}\right) \times 10^{-6}$	15.6(a)
(b) with side leakage	$S_o = \left(\frac{r}{c}\right)^2 \left(\frac{ZN}{p}\right) K_w \times 10^{-6}$	15.6(b)

where K_w is the load correction factor for side leakage (Fig. 15.6)
N is the speed, rev/min; p = bearing pressure, MN/m^2 (kgf/mm^2)

8 .CRITICAL PRESSURE OF JOURNAL BEARING (p)

Pressure at which oil film break down metal to metal contact begins

354 Journal Bearings

Particular	Equation	Eqn. No.
Bearing Pressure:		
General electric company's formula for bearing pressure in the design of motor and generator bearing	$p = 0.622\sqrt[3]{v}$ SI Units $= 0.0634\sqrt{v}$ Metric units	15.5(a)
The safe operating bearing pressure due to Tatarinoff, MN/m^2 (kgf/mm^2)	$p = \frac{13.30}{10^6} Zn \left(\frac{d}{c}\right)^2 \left(\frac{l}{d+l}\right)$	15.5(b)
Moore equation for unit pressure	$p = 0.726\sqrt{v}$ SI units $= 0.074\sqrt{v}$ Metric units	15.5(c)

9. Heat generated (H_g)

- The power lost to friction in the bearing is converted into heat and must be radiated from the housing without producing excessive temperatures.
- If the temperature of the bearing increases, the viscosity of the oil decreases as a result the oil squeezes out and the bearing seizes. '

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Particular	Equation	Eqn. No.
Power loss or heat generated, J/s (kgf m/s)	$H_g = fWv$ where $v = \frac{\pi dn}{1000}$, rubbing velocity, m/s	15.6(j)

Where f = coefficient of friction

W= load in Bering = PLd

P= Bearing pressure

L= length of bearing d= diameter of journal

10. Heat dissipated (H_D)

- The amount of heat dissipated or radiated by the bearing depends on the temperature difference, radiating surface, mass of the bearing and the amount of oil flowing around the bearing.

Heat Dissipation of Bearings:

Heat generated in the bearing J/s (kgf m/s)	$H_g = fWv$	15.11(a)

where f is the coefficient of friction; W is the bearing load, N; $v = \pi dn/1000$ rubbing velocity, m/s

General expression for the heat dissipating capacity of a bearing, J/s (kgf m/s)	$H_D = K_h A'(t_B - t_A)$	15.11(b)

where $K_h = Mv^{0.89}$, the film coefficient, $J/sm^2\ °C$

$= 11.4\ J/sm^2\ °C$ for still air

$= 15.3\ J/sm^2\ °C$ for average design conditions

$= 33.5\ J/sm^2\ °C$ for air moving at 2.5 m/s

$M = 14.82$, a constant for the selected units of K_h

$v =$ the air velocity, m/s

≈ 0.75 m/s for normal conditions without ventilation

≈ 2.50 m/s for a well-ventilated bearing

$A' = (20dl) \times 10^{-6}$, exposed area of bearing housing, m^2

d is the diameter of the bearing, mm

l is the length of the bearing, mm

t_B is the temperature of the exposed surface, °C

t_A is the temperature of the surrounding air, °C

$(t_B - t_A)$ is the temperature rise of bearing wall, °C (Fig. 15.16)

t_0 is the temperature of the oil, °C

On the basis of projected area of journal, the heat dissipating capacity, J/s	$H_D = K_P A_p = K_P ld \times 10^{-6}$	15.11(c)

11. Power loss due to friction (P)

Power loss, kW	$P = \left(\frac{2\pi}{10^6}\right) Tn$	SI Units	15.6(h)
Power loss, (mhp)	$P = \left(\frac{2\pi}{75 \times 10^3}\right) Tn$	Metric Units	15.6(i)

12. Flow variable (Q)

Particular	Equation	Eqn. No.
Flow:		
The flow variable (Fig. 15.13)	$(FVL) = \frac{Q}{rcNl}$	15.8(a)
(a) flow without side leakage, mm^3/min	$Q = (FLV)rcNl$	15.8(b)
(b) flow with side leakage, mm^3/min	$Q = (FLV)rcNlK_Q$	15.8(c)

where N is the spped, rpm; K_Q is the flow correction factor (Fig. 15.14)

Note: If a central groove is used, then l is one-half the length of the bearing; in this case, the total flow should be found for two half bearings

13. Minimum oil film thickness (MFT)

- The criteria to determine whether the bearing operates satisfactorily or not is a function of minimum oil thickness.
- It is defined as the minimum distance between the bearing and the journal under the condition of complete lubrication, so as to prevent metal-to-metal contact.
- Its value depends upon the surface finish of the parts, cleanliness of lubricant, deflection of parts, etc. The surface determines the required oil film thickness. The rougher the surface, the thicker would be the film required.

$$MTF = \frac{h_0}{c} \text{ (Page 356, 15.7(a))}$$

14. Length to diameter ratio

$\frac{l}{d} < 1$, bearing short bearing, $\frac{l}{d} > 1$ bearing long bearing $\frac{l}{d} = 1$, bearing square bearing

2.10 COMPARISON OF HYDRODYNAMIC AND HYDROSTATIC BEARINGS

S.No.	Hydrodynamic Bearing	Hydrostatic Bearing
1.	The pressure required by the lubricant in order to avoid metal to metal contact is produced by relative motion of the parts.	An external pressure supply is used to pressurize lubricant in order to avoid metal to metal contact.
2.	Load carrying capacity of the bearing is less.	Load carrying capacity of the bearing is high.
3.	Load carrying capacity increases linearly with speed.	Load carrying capacity is independent of speed.
4.	Simple in construction and easy to maintain.	Complex in construction and difficult to maintain.
5.	Lower initial and maintenance cost.	Higher initial and maintenance cost.
6.	Starting friction is present.	No starting friction is present.
7.	Power loss is high during starting with hydrodynamic bearings.	No power loss takes place during starting with hydrostatic bearings.
8.	No additional space is required.	Additional space is required for lubrication system like pump.

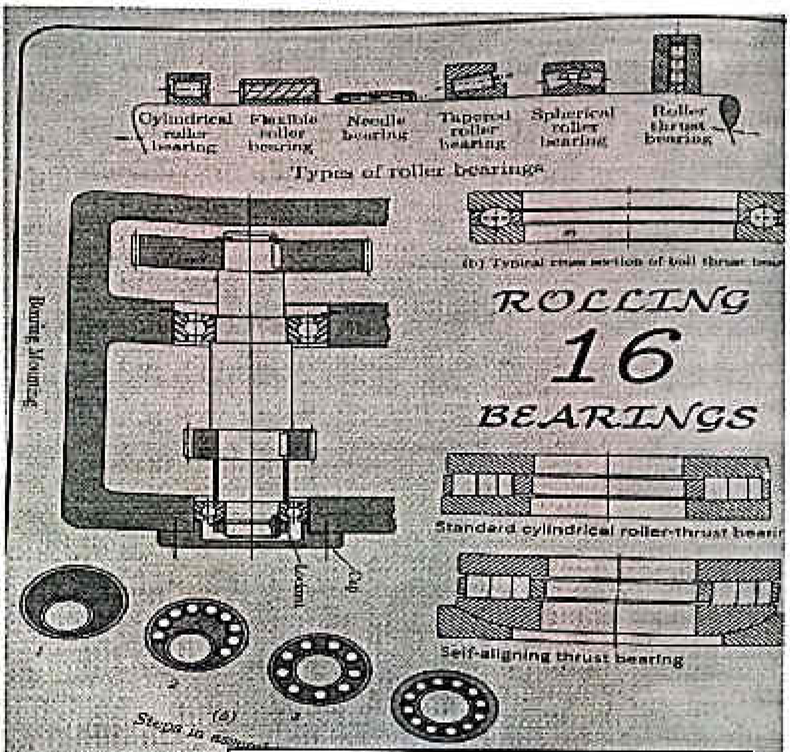
oil

BEARING HOUSING AND MOUNTING

A bearing is a mechanical component that carries a bush and supports the shaft. Generally solid housing is used for up to 50 mm journal diameters. Sometimes, for ease of assembly and replacement, split type housing with split bushes is. Preferred. When housing is of the split-type, the two pieces called the cap and the housing are joined by nut-bolt fasteners.

One of the most common type of split housing for journal bearings is the plumber block. It is available in two series light and medium. The dimensions of plumber block should conform to 18:4773 1979.

In a bearing the inner race is fitted on the shaft by means of an interference fit, which prevents the relative rotation and the corresponding wear between the inner race and the shaft. Care must be taken to select the fit in such a way that it provides sufficient tightness to give a firm mounting and at the same time, it should not be very tight to cause deformation of the inner race and destroying clearance between the rolling element and the races.



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The outer race of a bearing is also mounted in the housing with interference fit, but to a lesser degree of tightness than that of the inner race. Insufficient tightness of outer race in the housing seat may cause 'creep'. The slow rotation of the outer race relative to its sealing is called 'creep'. It is caused by radial loads that rotate and change the direction. When on a same shaft two bearings are mounted, the outer race of one of them should be permitted to shift axially to take care of axial deflection of the shaft caused either by thrust load or by temperature variation.

It is important to use correct method of mounting and to observe cleanliness if the bearing is to function with satisfaction and to achieve the required life. Some of the precautions to be taken during mounting operation are as follows:

1. The shaft and the housing bore should be inspected before assembly, Any kind of burrs on the shaft and the shoulders should be removed.
2. Mounting operation should be carried out in (113/ and dust free environment.
3. Small bearings are mounted on the shaft with the help of a small piece of tube (or) ring, on which the blows are applied by the hammer. Application of direct blows to bearing should be avoided, otherwise the race or the cage may get damaged.
4. Medium sized bearings are mounted on the shaft by pressing the tube (or) the metallic ring by means of hydraulic (or) mechanical press.
5. Large sized bearings are mounted by heating them to 80 to 90°C above the ambient temperature by induction heating and then shrinking them on the shaft.

6. A bearing should not be taken out from its package until it is assembled. The inner and outer surface of the bearing should be cleaned with white spirit and wiped with clean cloth.

Design consideration of bearing

SELECTION OF JOURNAL -OR-ANTIFRICTION BEARING

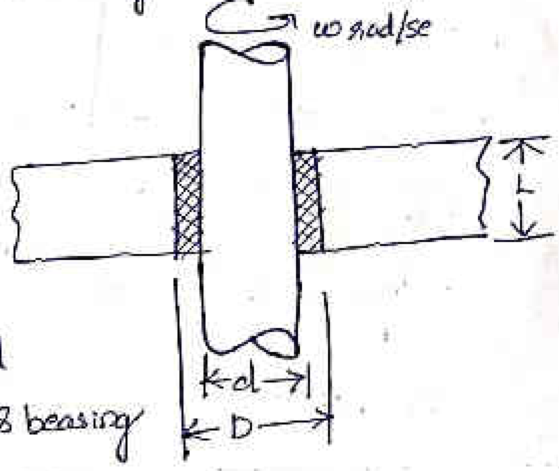
1. When space is a constraint, rolling bearing is preferred.
2. Roller bearings produce noise prior to failure, while that of journal bearing is sudden. .
3. Ball bearings are preferred for small and medium radial loads, while roller bearings are preferred for heavy loads.
4. Roller bearings can take a combination of radial and thrust loads. For high overloads rolling bearings are preferred.
5. Rolling bearings are noisy compared to journal bearings.

(a) PETROFF'S EQUATION (Derivation)

An equation for Coefficient of friction, assuming journal is concentric with bearing

$$\boxed{f = (2\pi^2 \times 10^{-6})\left(\frac{Zn}{P}\right)\left(\frac{r}{c}\right)}$$ page 353, eq 15.4(a)

Consider a vertical shaft rotating in a Guide bearing as shown in figure



Assumption

1. Bearing is lightly loaded
2. The clearance is completely filled with oil
3. There is no end leakage
4. Viscosity of oil is high
5. Journal rotates at a very high speed
6. There is no eccentricity b/w journal & bearing

Shearing stress in the lubricant $\tau = Z \cdot \frac{V}{C} = \frac{2\pi r Z n}{C}$ —— 15.3(a) [page 351]

Viscous shearing force on shaft

$F_s = \tau \times Area = \tau \times 2\pi r l = \frac{4\pi^2 r^2 l Z n}{C}$ 15.3(b) [page 351]

Shearing Torque $\Rightarrow T = F_s \times r = \frac{4\pi^2 r^3 l Z n}{C}$ —— ① 15.3(c)

Frictional Torque $T_f = f W r = f(2 r l p)(r) = 2r^2 f l p$ —— ②

Equating equation ① & ②

$$2\,\frac{\cancel{4}\pi^2 \cancel{r^3} \cancel{l} Z n}{C} = \cancel{2} \cancel{r^2} f \cancel{l} p$$

$$\boxed{f = 2\pi^2 \left(\frac{Zn}{P}\right)\left(\frac{r}{c}\right)}$$ page 353, eq 15.4(a)

Journal Bearing problems

① The operating condition of a journal bearing for a centrifugal pump is loaded on journal = 11.5 kN, speed of journal 1440 rpm, diameter of journal = 75 mm. Operating temperature = 70°C, Ambient Temperature = 40°C. Design the bearing.

Solution :- G.D $W = 11.5 \times 10^3 N$, $d = 75mm$, $r = \frac{d}{2} = 37.5mm = 37.5 \times 10^{-3} m$

Step 1 **Recommended Design data**

page 366, Table 15.7 Max pressure $P = 0.7 - 1.4 \approx 1 Mpa = 1 \times 10^6 N/m^2$

Clearance ratio $\frac{C}{r} = 0.0013$

Ratio of $\frac{l}{d} = 1 - 2 \approx 1.5$ (Avg Value)

Viscocity $Z = 0.025 \; Ns/m^2 = 25cp$, $\frac{ZN}{P} = 29.01$

Step 2 **Length of Bearing (l)**

$P = \frac{W}{2rl}$ (page 351, below)

$1 \times 10^6 = \frac{11.5 \times 10^3}{2 \times 37.5 \times 10^{-3} \times l} \Rightarrow l = 0.153m \approx 0.150m$

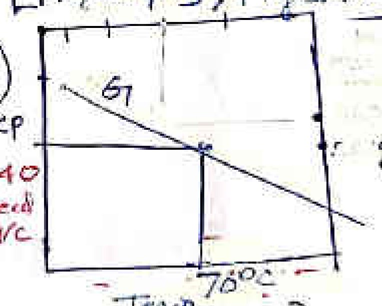
Actual pressure $P = \frac{W}{2rl} = \frac{11.5 \times 10^3}{2 \times 37.5 \times 10^{-3} \times 0.15} = 1.022 \times 10^6 pa = 1.022 Mpa$

Step 3 **Selection of oil for lubrication**

Lingaiah databook

Corresponding of viscocity $Z = 25cp$ and temperature = 70°C - [Page 24.5, Fig 24-2]

Select oil G1 (near higher level oil)



Actual viscocity for selected oil at 70°C (Absolute Viscosity)

$Z = 31cp = 31 \times 10^{-3} pa$

page No: 363, T(15.1) G1 = Automobile oil SAE40 Medium speed M/C

Step 4 Check for Suitability of oil

Bearing Modulus (Bearing characteristic) $= \frac{ZN}{P}$ (page 353, Below fig 15.5(a))

$= \frac{31 \times 10^{-3} \times 1440}{1.022} = 43.679$

(Thick film expected) 43.679 > (Recommended 29.01)

PAGE 363 ⇒ G1 = AutoMobile oil, SAE 40, Medium Speed M/C

Step 5 Coefficient of Friction

page 353, eq 15.4(b)

$$f = k_a\left(\frac{Zn}{P}\right)\left(\frac{r}{c}\right)\times 10^{-10} + \Delta f$$

$$f = 0.195\times10^{6}\left(\frac{31\times10^{-3}\times24}{1.022}\right)\left(\frac{1}{0.0013}\right)10^{-10} + 0.002$$

$$= \underline{\underline{0.01290}}$$

$k_a = 0.195\times10^{6}$ for $\beta = 360$

$Z = 31\times10^{-3}$ pas

$n = rev/s = \frac{1440}{60} = 24\, rps$

$P = 1.022$ Mpa

$\frac{r}{c} = \frac{1}{0.0013}$ (given

$\Delta f = 0.002$ for $\frac{l}{d} = 0.75-2.8$

Step 6 Heat Generated

page 356, eq 15.6(i) $\Rightarrow$ $H_g = fWv$

$$H_g = f(2rlp)v$$

$$= 0.01290\times\left(2\times\frac{75}{2}\times150\times1.022\right)\times5.652$$

$$= \underline{\underline{838.3\,J/s}}$$

$$v = \frac{\pi d n}{1000} = \frac{\pi\times175\times24}{1000} = \underline{\underline{5.652\, m/s}}$$

Step 7 Heat dissipated

General expression for heat dissipating capacity of bearing $H_D = k_h A'(t_B - t_A)$ [page 360 eq 15.11(b)]

where $k_h = 11.4\, J/s\,m^2\,°c$ for still air

$A = (20dl)\times10^{-6}$, exposed area of bearing housing

$$A = 20\times75\times150\times10^{-6} = \underline{0.225\, m^2}$$

$$t_B - t_A = \frac{t_o - t_A}{2}\ (\text{page } 361) = \frac{70-40}{2} = \underline{\underline{15°c}}$$

$$H_D = 11.4\times0.225\times15 = \underline{38.475}\ J/s$$

Here $H_d < H_g$, Bearing required artificial cooling.

Amount of Heat removed $= H_g - H_D = 838.3 - 38.475 = \underline{\underline{799.825}}$ J/s

Step 8 power loss (Heat generated)

$$P = \left(\frac{2\pi}{10^6}\right)Tn \quad (\text{page } 355)$$

eq: 15.6(h)

$$P = \left(\frac{2\pi}{10^6}\right)\times5563.125\times24$$

$$= \underline{\underline{0.838\, kW}}$$

$T = fWr$ (15.6(g))

$= 0.01290\times11.5\times10^{3}\times\frac{75}{2}$

$= 5563.125$ Nmm

Step 9 Minimum Oil film Thickness

page 356 ⇒ fig 15.9

Min. film thickness ⇒ $\frac{h_o}{c}$

where c = radial clearance = $0.0013 \times \frac{0.075}{2}$ $= 4.875 \times 10^{-5}\,m$

$\frac{r}{c} = \frac{1}{0.0013}$ (given)

$r = \frac{0.075}{2}$

Bearing characteristic Number

page 358 fig 15.13 x-axis

$$S = \left(\frac{r}{c}\right)^2\left(\frac{ZN}{P}\right)\times 10^{-6}$$

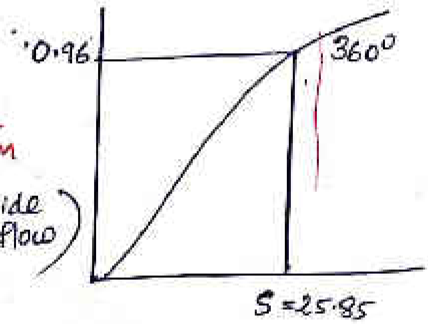
$$= \left(\frac{1}{0.0013}\right)^2 \times \left(\frac{31\times10^{-3}\times1440}{1.022}\right)\times10^{-6} = 25.85$$

From page 856 ⇒ fig 15.9

$\frac{h_o}{c} = 0.96$

$h_o = 0.96 \times 4.875\times10^{-5}\,m$

$= 4.68\times10^{-5}\,m$ (No side flow)



Step 10 : Flow Variable & Flow rate of Oil in Bearing

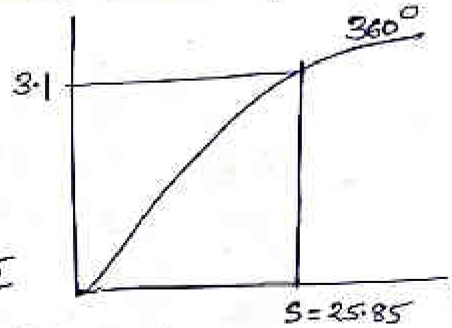
page 858, fig 15.13

$3.1 = \frac{Q}{rcNl}$

$Q = 3.1 \times rcNl$

$= 3.1 \times \frac{0.075}{2} \times 0.0013 \times \frac{0.075}{2} \times 1440 \times 0.150$

$= 1.224\times10^{-3}\,m^3/\text{minute}$



$\frac{r}{c} = \frac{1}{0.0013}$

$c = 0.0013 \times \frac{0.075}{2}$

- A load of 3kN supported by a journal bearing of dia 75mm length. Diametrical clearance = 0.05mm at Viscocity = 21CP at operating temperature. Determine Max: speed of rotation of bearing is capable of dissipating 80W by heat transfer.

Solution $W = 3\times10^3 N$, $d = 75mm$, $r = 37.5mm$, $l = 75mm$, $C_d = 0.05mm$

Heat dissipated, $H_D = 80W$, $Z = 21CP = 21\times10^{-3} Ns/m^2$, $n = ?$ $c = \frac{0.05}{2} = 0.025$

For safe operation $H_D = H_g$

Heat generated $\Rightarrow H_g = fWv$ (page 356, eq 15.6(d))

$$80 = (1.152\times10^{-3}n + 0.002)\,3\times10^3 \times \frac{\pi d n}{1000}$$

$$80 = (1.152\times10^{-3}n + 0.002)\,3\times10^3 \times \frac{\pi\times75\times n}{1000}$$

$$80 = 0.814n^2 + 1.413n$$

$n = 9.08$ rps

$N = 9.08\times60 = \underline{\underline{544.8}}$ rpm

where

$$\boxed{f = K_a\left(\frac{Zn}{P}\right)\left(\frac{r}{c}\right)10^{-10} + \Delta f}$$ → page 353 eq 15.4(a)

where $k_a = 0.195\times10^6$

$Z = 21\times10^{-3} Ns/m^2$

$P = \frac{W}{2rl} \Rightarrow$ (page 355, eq 15.6(c))

$= \frac{3\times10^3}{2\times37.5\times75} = 0.533\ N/mm^2$

$= \underline{0.533\ MN/M^2}$

$$f = 0.195\times10^6 \times \frac{21\times10^{-3}n}{0.533}\left(\frac{37.5}{0.025}\right)10^{-10} + 0.002$$

$$f = 1.152\times10^{-3}n + 0.002$$

- A Full Journal Bearing of 50mm diameter of 100mm long has a bearing pressure of $1.4N/mm^2$. Speed of journal is 900rpm of $\frac{d}{c} = 1000$. Absolute Viscocity at operating temperature 75°c is 0.011 kg/ms. Room temp is 35°c. Find (a) Amount of artificial cooling required (b) Mass of lubricating oil required. If difference b/w outlet & inlet temp is 10°c. Take Specific heat of oil is 1850 J/kgK

Solution

G.D $d = 50mm$, $l = 100mm$

$P = 1.4\ N/mm^2 = 1.4$ MPa, $N = 900$ rpm, $\frac{d}{C_d} = \frac{r}{c} = 1000$, $t_o = 75°c$

$Z = 0.011$ kg/ms $n = 15$ rps

(a) To find artificial cooling $\Rightarrow$ Amount of Heat required $= H_g - H_D$

where $H_g = fWv$ (page 356, eq 15.6(d))

$$H_g = 4.29\times10^{-3}\times7000\times\frac{\pi\times50\times15}{1000}$$

$\underline{= 70.72\ J/s}$

$P = \frac{W}{2rl}$ (page 355, eq 15.6(c))

$W = P\times2rl = 1.4\times2\times25\times100$

$= \underline{7000N}$

$$f = k_a\left(\frac{Zn}{P}\right)\left(\frac{r}{c}\right)10^{-10} + \Delta f$$

$$= 0.195\times10^6\left(\frac{0.011\times15}{1.4}\right)1000\times10^{-10} + 0.002$$

$= 4.29\times10^{-3}$

Heat dissipating capacity $H_D = k_h A'(t_B - t_A)$ [page 360, eq. 15.11(b)]

$= 11.4 \times 20 \times 50 \times 100 \times 20 \times 10^{-6}$

$= \underline{22.8 J/s}$

$k_h = 11.4 J/s^2$

$A' = 20 d l \times 10^{-6}$

$= 20 \times 50 \times 100 \times 10^{-6}$

$t_B - t_A = \frac{1}{2}(t_o - t_A)$

$= \frac{1}{2}(75 - 35) = \underline{20}$

Amount of artificial cooling required $= H_g - H_d = 70.72 - 22.8$

$= \underline{47.92 J/s}$

(b) Heat Taken away from oil

$H_g = m c_p \Delta T$

$70.72 = m \times 1850 \times 10$

$m = \underline{3.82 \times 10^{-3} kg/s}$

$C_p = 1850 J/kg K$

$\Delta T = 10$

- A lightly loaded journal bearing has the following specification

Bearing diameter = 80mm diametrical clearance = 0.12mm

Bearing length = 60mm Journal speed = 24000 rpm

radial load = 900N Absolute viscosity = 4cP

Determine (a) Frictional force (b) Torque (c) Coefficient of friction (d) power loss

Solution

$d = 80mm$, $l = 60mm$, $W = 900N$, $Z = 4cP = 4 \times 10^{-3}$ pas, $N = 24000 rpm$, $n = \frac{24000}{60}$

$= 400 rps$

$C_d = 0.12mm$, $C = \frac{0.12}{2} = \underline{0.06mm}$

$P = \frac{W}{2rl} = \frac{900}{2 \times 40 \times 60} = \underline{0.1875 MPa}$

(a) Frictional Force

[page 351 eq 15.3(b)] $F_s = \frac{4\pi^2 r^2 l Z n \times 10^{-6}}{c} = \frac{4\pi^2 \times 40^2 \times 60 \times 4 \times 10^{-3} \times 400 \times 10^{-6}}{0.06} = \underline{100.96N}$

(b) Torque $\rightarrow T = F_s \times r$ (page 351, eq 15.3(c))

$= 100.96 \times 40 = \underline{4038.492 \text{ Nmm}}$

(c) Coefficient of friction (using petroff's eq: lightly loaded Bearing)

$f = 2\pi^2 \times 10^{-6} \left(\frac{Zn}{P}\right)\left(\frac{r}{c}\right)$ [page 353, eq 15.4(a)]

$= 2\pi^2 \times 10^{-6} \left(\frac{4 \times 10^{-3} \times 400}{0.188}\right)\left(\frac{40}{0.06}\right) = \underline{0.1118}$

(d) power loss (page 355, eq 15.6(b))

$P = \frac{2\pi}{10^6} Tn = \frac{2\pi}{10^6} \times f W r \times n = \frac{2\pi}{10^6} \times 0.1118 \times 900 \times 40 \times 400 = \underline{10.11 kW}$