Rolling-Contact Bearings

- Rolling-contact bearing, antifriction bearing, and rolling bearing are all used to describe that class of bearing in which the main load is transferred through elements in rolling contact.
- The four essential parts of a bearing are the outer ring, the inner ring, the balls or rolling elements, and the seperator. (Figure 11-1)

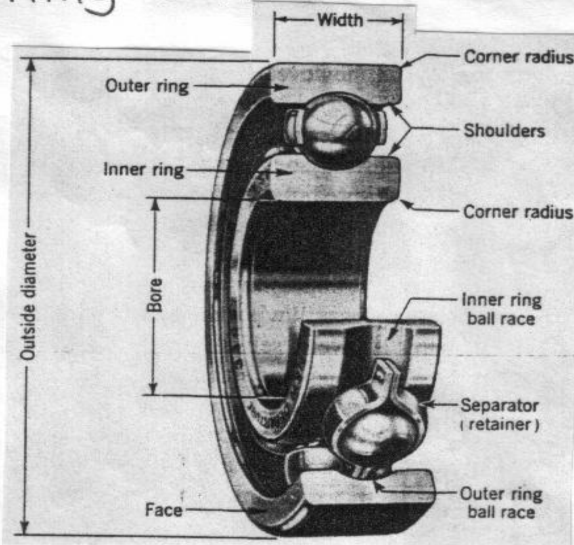

Figure 11-1

Nomenclature of a ball bearing. *(Courtesy of New Departure–Hyatt Division, General Motors Corporation.)*

- Types of bearings (Figure 11-2 and 11-3)
- Bearings are manufactured to take pure radial loads, pure thrust loads, or a combination of the two kinds of loads.

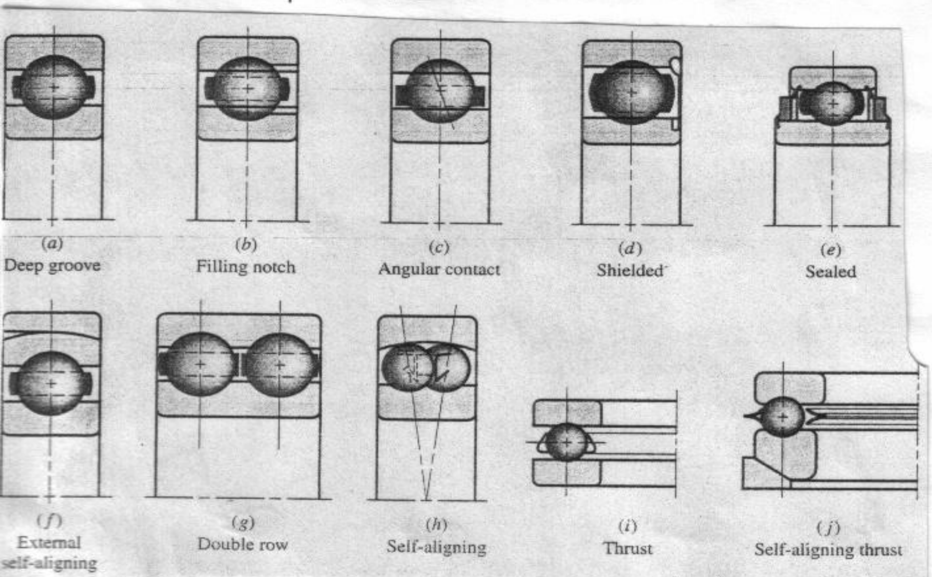

Figure 11-2

Various types of ball bearings.

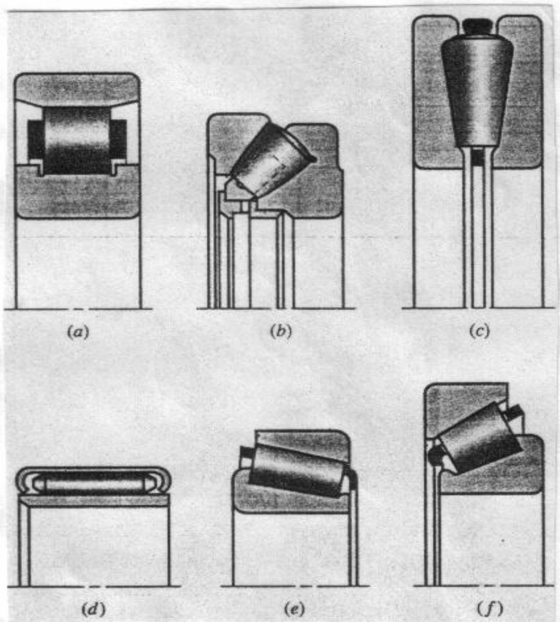

Figure 11-3

Types of roller bearings: (*a*) straight roller; (*b*) spherical roller, thrust; (*c*) tapered roller, thrust; (*d*) needle; (*e*) tapered roller; (*f*) steep-angle tapered roller. *(Courtesy of The Timken Company.)*

Bearing life

$F_e \cdot L^{1/a} = C \Rightarrow L = \left(\frac{C}{F_e}\right)^a$ (million cycles)

$a=3$ for ball bearings

$a=10/3$ for roller bearings (cylindrical and tapered roller)

C: dynamic load factor

$F_e = X \cdot V \cdot F_r + Y \cdot F_a$

F_e: equivalent load; F_r: radial load; F_a: axial load

$F_e \leq C_o$ → static load factor

$\frac{F_a}{V \cdot F_r} \Rightarrow X, Y$

$V=1$ (inner ring rotates)

$V=1.2$ (outer ring rotates)

$C_o = S_o \cdot F_e$ — Table 11-5

$L_h = \frac{L \cdot 10^6}{60 \cdot n}$ → operating hours

Lubrication

- Use grease when
  - The temperature is not over 200°F. (93°C)
  - The speed is low.
  - Unusual protection is required from the entrance of foreign matter.
  - Simple bearing enclosures are desired.
  - Operation for long periods without attention is desired.
- Use oil when
  - Speeds are high
  - Temperatures are high
  - Oiltight seals are readily employed.
  - Bearing type is not suitable for grease lubrication.
  - The bearing is lubricated from a central supply which is also used for other machine parts.

Mounting

- One bearing at each end of a shaft.
  - The inner rings are backed up against the shaft shoulders and are held in position by round nuts threaded onto the shaft.

Bearing Load-Life Trade-Off at Constant Reliability

$$C_{10} \cdot L_{10}^{1/a} = F \cdot L^{1/a}$$

$$\underbrace{C_{10}}_{\text{catalog rating}} \cdot \Big(\underbrace{L_R}_{\text{rating lif in hours}} \cdot \underbrace{n_R}_{\text{rating speed, rpm}} \cdot 60 \Big)^{1/a} = \underbrace{F_D}_{\text{desired radial load}} \cdot \Big(\underbrace{L_D}_{\text{desired life, hours}} \cdot \underbrace{n_D}_{\text{desired speed, rpm}} \cdot 60 \Big)^{1/a}$$

Load-Life-Reliability Trade-Off

$$C_{10} = F_D \cdot \left[\frac{x_D}{x_0 + (\theta - x_0)\left(\ln 1/R_D\right)^{1/b}} \right]^{1/a}$$

θ: characteristic parameter corresponding to the 63.2121 percentile value of the variate

Variable Loading

$$F_{eq} = \left[\frac{\sum n_i t_i F_{e_i}^{a}}{\sum n_i t_i} \right]^{1/a}$$

n_i: speed

t_i: duration of speed (i)

Adequacy Assessment for Selected Rolling-Contact Bearings

$$R = \exp\left(- \left\{ \frac{\left[x_D - x_0 \left(\frac{C_{10}'}{f_A \cdot F_D} \right)^{a} \right]}{(\theta - x_0)\left(\frac{C_{10}'}{f_A \cdot F_D} \right)^{a}} \right\}^{b} \right)$$

C_{10}': basic load rating

- The inner races are backed up against the shaft shoulders as before but no retaining devices are required.

- Two or more bearings at one end of shaft
  - face-to-face
  - back-to-back
  - tandem arrangement

- The inner ring usually takes $\underbrace{\text{circumferential load}}_{\text{the direction of load changes over time}}$

- The outer ring usually takes $\underbrace{\text{point load}}_{\text{the direction of load doesn't change over time}}$

The mounting rule

- The ring taking circumferential load is mounted tightly.

- The ring taking point load is mounted less tightly.

Typical sealing methods

- Felt seals may be used with grease lubrication when the speeds are low.

- The comercial seal is an assembly consisting of the rubbing element, spring backing, which are retained in a sheet-metal jacket.

- The labyrinth seal is especially effective for high-speed installations and maybe used with either oil or grease.

Lubrication and Journal Bearings

Types of Lubrication

- Hydrodynamic: Relative motion of surfaces
- Hydrostatic: Requires a pressure
- Elastohydrodynamic: Lubricant introduced in rolling contact.
- Boundary: Thin film
- Solid-film: Extreme temperatures. Graphite, molybdenum, composite bearing materials.

Viscosity

$$\tau = \frac{F}{A} = \mu \frac{du}{dy}$$

μ → absolute viscosity (dynamic viscosity)

$\frac{du}{dy}$ → rate of shear (velocity gradient)

If $\frac{du}{dy}$ is constant, then $\frac{du}{dy} = \frac{U}{h} \Rightarrow \tau = \frac{F}{A} = \mu \frac{U}{h}$

Unit of viscosity Pa.s (pascal-second)

ASME cgs units dyn.s/cm² (P) (poise), cP (centipoise)

$$\mu\,(\mathrm{Pa.s}) = (10)^{-3} Z\,(\mathrm{cP})$$

kinematic viskosity $Z_k = \left(0.22t - \frac{180}{t}\right)$

Z_k → cSt (centistoke)

t → number of seconds Saybolt

$$\nu\,(m^2/s) = 10^{-6} Z_k\,(\mathrm{cSt})$$

$$\nu = \left(0.22t - \frac{180}{t}\right)(10^{-6})$$

$$\mu = \rho \cdot \nu$$

Petroff's Equation

$$\tau = \mu \frac{U}{h} = \frac{2\pi r \mu N}{c}$$

$$T = (\tau A)\cdot(r) = \left(\frac{2\pi r \mu N}{c}\right)\cdot(2\pi r l)\cdot(r) = \frac{4\pi^2 r^3 l \mu N}{c}$$

$$T = f \cdot W \cdot r = \frac{4\pi^2 r^3 l \mu N}{c} \Rightarrow f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c}$$

$$P = \frac{W}{2rl}$$

$$S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu N}{P}\right)$$

S → bearing characteristic number (Sommerfeld number)

r → journal radius (mm)

μ → absolute viscosity (Pa.s)

N → significant speed (r/s)

c → radial clearance (mm)

P → load per unit of projected bearing area (MPa)

Temperature rise

$$\dot{m} = l.c.\rho.U/2 = l.c.\rho.\pi.r.N_j$$

N_j → journal angular speed

$$\bar{T}_f = T_s + \Delta T/2$$

$\bar{T}_f$ → average film temperature

$$\dot{Q}_{loss} = U_o A_o (T_b - T_\infty) = U_o A_o (\bar{T}_f - T_\infty)/2$$

$\dot{Q}_{loss}$ → heat loss rate by convection and radiation

U_o → overall combined coefficient

A_o → bearing housing lateral surface area

T_b → bearing metal surface temperature

T_∞ → ambient temperature

Not: bushing and housing metal temperature is midway between the $\bar{T}_f$ and T_∞

$$\dot{Q}_{gen} = \dot{m}.C_p.\Delta T = l.c.\rho.\pi.r.N_j C_p \Delta T$$

$\dot{Q}_{gen}$ → heat generation rate by viscous friction

$$\dot{Q}_{gen} = 2\pi T N_j = 2\pi f W r N_j = 2\pi \frac{fr}{c}\frac{c}{r} r W N_j$$

$$= 4\pi^3 \left(\frac{r}{c}\right)^2 \frac{\mu N_j (2rL)}{W} \frac{c}{r} r W N_j$$

$$\bar{T}_f = T_\infty + 16\pi^3 \frac{\mu.N_j^2.L.r^3}{U_o.A_o.c}$$

$$\Delta T = \frac{\dot{Q}_{gen}}{\dot{m}.C_p} = \frac{U_o.A_o(\bar{T}_f - T_\infty)/2}{l.c.\rho.\pi.r.N_j.C_p}$$

ΔT → temperature rise

Stable Lubrication

$$\frac{\mu.N_j}{P} \geq 0.362(10^{-6})$$

Thick Film Lubrication and Hydrodynamic Theory

Assumptions

1) The curvature could be neglected
2) The lubricant obeys Newton's viscous effect, $\tau = \mu \frac{du}{dy}$
3) The forces due to inertia of the lubricant are neglected
4) The lubricant is assumed to be incompressible
5) The viscosity is assumed to be constant throughout the film
6) The pressure does not vary in the axial direction
7) There is not lubricant flow in the z direction
8) The film pressure constant in the y direction
9) The velocity depends only on the coordinates x and y

$$\frac{dp}{dx} = \mu \frac{\partial^2 u}{\partial y^2} \Rightarrow u = \frac{1}{2\mu}\frac{dp}{dx}(y^2 - hy) - \frac{U}{h} y$$

$$Q = \int_0^h u\,dy \qquad \frac{dQ}{dx} = 0 \Rightarrow \frac{r}{c} f = \phi\left[\left(\frac{r}{c}\right)^2 \frac{\mu N}{P}\right]$$

Q: volume of lubricant flowing in the x-direction

$\frac{dQ}{dx} = 0$: incompressible lubricant

Design Considerations

1) The viscosity μ
2) The load per unit of projected bearing area, P
3) The speed N
4) The bearing dimensions r, c, β, and L

Performance factors ⇒ 1) The coefficient of friction, f
2) The temperature rise ΔT 3) The flow of oil, Q
4) The minimum film thickness h_o

$h_o \geq 0.005 + 0.00004 \cdot D$ mm (D: journal diameter)

$T_{max} \leq 120°C$ $\quad n_d \geq 2$

$\frac{W_{st}}{LD} \leq 2$ MPa (starting load) $\quad \frac{\partial h_o}{\partial c} > 0$

The relations of the variables are given in Figures 12-11 ---12-19.

The design plane is the rc plane (Fig. 12-27).

Pressure-Fed Bearings

The lubricant is supplied to the bearing under pressure.
rc plane (Fig. 12-34)

Loads and Materials

Table 12-5 unit loads for several applications
Table 12-6 some characteristic of bearing alloys
- satisfactory compressive and fatigue strength
- soft, have a low melting point and a low modulus of elasticity
- to permit the material to wear or breakin
- resistance to wear
- resistance to corrosion, low cost

Bearing Types

Solid bushing, lined bushing, two-piece bushings, thrust bearings, boundary-lubricated bearings (some bearings are boundary lubricated at all times, some only at starting, stopping, overloading, lubricant deficiency)