Bearing Ⅰ - Sliding Bearing

R&D Center

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1. Bearing

1-1. Definition

A bearing is a machine element that constrains relative motion to only the desired motion and reduces friction between moving parts. A shaft attached to the bearing is called a journal. Heat due to friction arises in the bearing part and power loss occurs.

1-2. Classification (By the way of contact)

(1) Sliding bearing : There arises sliding friction and an oil film is formed by the lubricant between the journal and bearing.
(2) Rolling bearing : There arises rolling friction in the moving bearing.

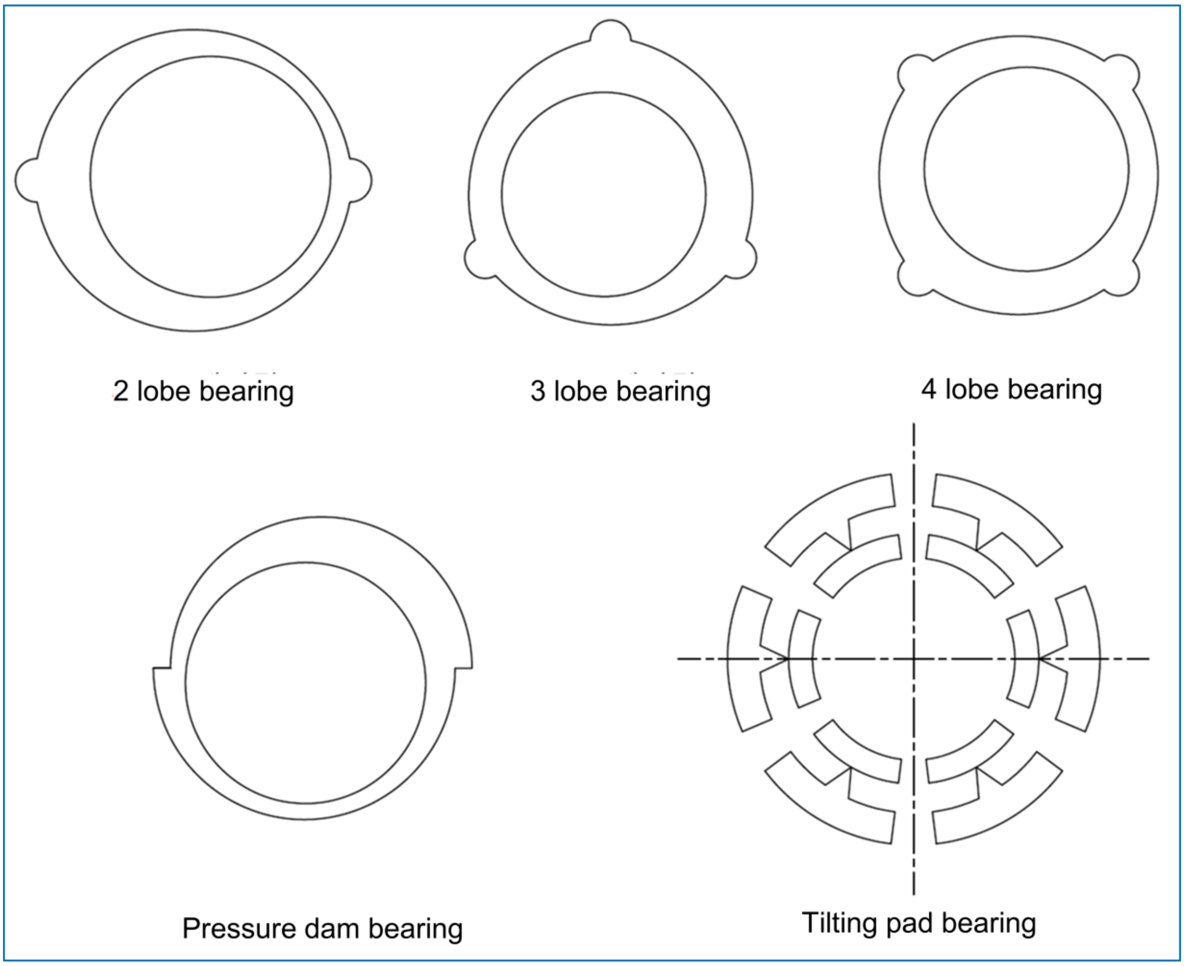
1-3. Sliding Bearing and Rolling Bearing

	Sliding bearing	Rolling bearing
Starting torque	High when lubricant formation is late.	Starting torque is low.
Shock absorption	Damping force by an oil film is excellent.	Shock absorption is low because of low damping force.
Convenience	Expertise needed to install it.	Easy to install.
Stiffness	Low	High
Driving velocity	Can be driven it over the resonance velocity.	Must be driven within resonance velocity.
High temperature	Lubricant viscosity decreases.	Cooler needed due to thermal expansion of moving bearing.
Standardization	There are a lot of cases of self-production.	A standard product and is interconvertible.

2. Sliding Bearing

2-1. Forms of sliding bearing

(1) Classification by the pressure retention method between the shaft and the bearing
  1) Hydrostatic bearing : relatively excellent rotational accuracy
  2) Hydrodynamic bearing
  3) Tilting pad bearing : combination of pad to control the pressure between the shaft and the bearing
(2) Classification by the fluid between the shaft and the bearing
  1) Oil bearing : It is divided into the cylindrical bearing, 2-lobe bearing, 3-lobe bearing, 4-lobe bearing, and the pressure dam bearing by the form of the internal diameter. (It is known that the pressure dam bearing is the best form of bearing.)
  2) Air bearing



[Figure 1] Forms of journal bearing

2-2. Type of friction

(1) Classification by the relative motion

1) Sliding friction

① Friction force at the moment of relative motion = Coefficient of static friction X Normal force

② Friction force after relative motion begins = Coefficient of dynamic friction X Normal force

2) Rolling friction

① Rolling without sliding → Doesn't exceed the static frictional force

② Rolling with sliding → Coefficient of dynamic friction X Normal force

(2) Classification by friction surface

1) Solid friction

① Dry friction without lubricant between friction surfaces → High caloric value due to high friction resistance.

② Friction coefficient is connected with the friction surface and relative velocity, and normally 0.1~1.

2) Fluid friction

① An oil surface is formed between the friction surfaces and there is relative motion in the state of the totally split two surfaces.

② Shear force occurs because fluid is attached to both sides of the surfaces and there is the relative motion of the middle fluid. → Sum of shear force = Frictional force

③ It is in connection with the viscosity of oil. Coefficient of friction is normally 0.001~0.01

3) Boundary friction

① A mean between solid friction and fluid friction, the friction at the ultra-thin oil surface

② Coefficient of friction is 0.01 ~ 0.1.

2-3. Newton's law of the viscous fluid

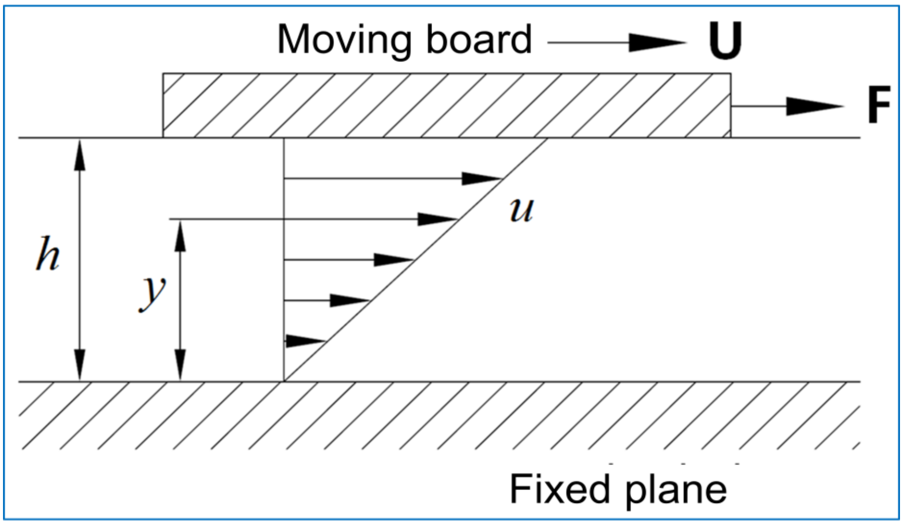
(1) Shear force of fluid

$$\tau = \eta \frac{du}{dy}$$

τ : Shear force

η : Coefficient of viscosity

$\frac{du}{dy}$: Change rate of velocity



[Figure 2] Shear force and change rate of velocity

As shown in figure, when the area of the moving board at the force F is A, Shear force τ is F/A. The distance between the fixed plane and the moving board at the velocity U is h, and the velocity is changing linearly, gradient of velocity is U/h. The former equation is expressed as follows:

$$\tau = \frac{F}{A} = \eta \frac{du}{dy} = \eta \frac{U}{h}$$

(2) Viscosity and measurement

1) Viscosity

Fluid viscosity is expressed as absolute viscosity η.

1 poise → the unit of viscosity when force 1 dyne is needed where the 1 cm^2 area board is moving at the velocity of 1 cm/s by the 1 cm oil surface thickness

$$\frac{1[dyne]}{[cm^2]} = 1[poise]\frac{1[cm/s]}{1[cm]}$$

$$1[poise] = 1[dyne \cdot s / cm^2] = 0.1[Pa \cdot s]$$

※ 1 dyne = 0.00001 N

[cp](centi-poise) is used as a practical unit.

$$1[cp] = \frac{1}{100}[p]$$

Dynamic viscosity ν is expressed as follows where the density is ρ and the absolute coefficient of viscosity is η.

$$\nu = \frac{\eta}{\rho}$$

[St] is used as unit of dynamic viscosity

$$1[St](stroke) = 1[cm^2 / s]$$

$$1[cSt](centi-stroke) = \frac{1}{100}[St]$$

2) Measurement of viscosity

A Saybolt viscometer is commonly used to measure viscosity. Viscosity is calculated as follows:

$$\eta[cp] = \rho\left(0.22 \cdot SUV - \frac{180}{SUV}\right)$$

ρ : Density $[g / cm^3]$

SUV : the time required for 60 cc oil passing through the capillary tube at a certain temperature

As shown in the above equation, the viscosity decreases as the density decreases.

3) Relation between oil temperature and density

Relation between oil temperature and density is as follows:

$$\rho_t = \rho_0 - 0.000657(t - t_0)$$

t : temperature [℃]

Density ρ_0 at temperature t_0, density ρ_t at temperature t ρ_t.

As shown in the above equation, the viscosity of oil decreases as the temperature increases.

2-4. Petroff's Equation of Bearing

(1) Rotation axis supported by oil bearing rotates on the eccentric condition due to the weight and rotation of the shaft.

(2) When the shaft is light and rotates at a high speed, the center of the shaft changes close to the center of bearing.

(3) The friction state of the sliding bearing in the case that the center of shaft and the center of bearing coincide is as follows:

The concentric bearing is defined when the shaft center and bearing center coincide. We assume that velocity distribution is linearly changing from the bearing to the radius direction. If the gap between the shaft and the bearing is smaller than the bearing radius, we can interpret the relative motion between the shaft and the bearing as the motion of fluid between two plates.

r : Radius of a shaft

δ : Thickness between the shaft and bearing

l : The width of a journal

N[rpm] : Rotation speed of a shaft

Shear stress of oil surface by relative motion of the shaft and the bearing is as follows:

$$\tau = \eta\frac{du}{dy} = \eta\frac{1}{\delta}\left(r\frac{2\pi N}{60}\right)$$

Product shear stress and shear area is shear force at oil surface and product shear force and the shaft radius is torque loss by shear stress.

$$T = \tau Ar = \eta\frac{2\pi r}{\delta}\frac{N}{60} \cdot 2\pi rl \cdot r$$

Mean pressure (p) at bearing surface is the normal force applied to the friction surface (P) divided by the projected area (2rl).

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

Frictional force by the relative motion between the shaft and the bearing is μP (Product of friction coefficient and the weight of shaft), and the torque by the frictional force is as follows.

$$T = \mu P \cdot r = \mu(p2rl)r = 2\mu r^2 lp$$

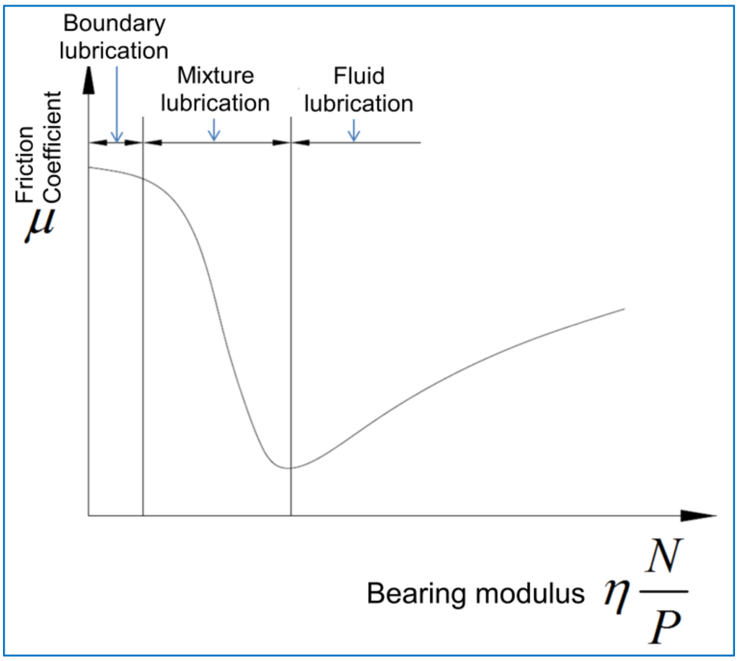
Friction coefficient between the shaft and the bearing is calculated as follows:

$$\mu = \frac{\pi^2}{30} \cdot \eta \frac{N}{p} \cdot \frac{r}{\delta}$$

This equation is Petroff's Equation and it is the friction coefficient of the sliding bearing when the eccentricity is low.

$\eta \frac{N}{p}$ is bearing modulus, $\frac{\delta}{r}$ is gap ratio, expressed as φ. These two factors are important factors that determine the performance of the bearing. A gap ratio of 1/1000 is standard.

As shown in the above equation, μ is theoretically proportional to $\eta \frac{N}{p}$. This is for the case that an shaft is light and rotates at high speed and the shaft center is close to the bearing center. The tendency is aligned in the fluid friction section. Real bearing, in which the shaft rotates at low speed, features an axial center eccentric to the bearing center. This is different from the supposition in Petroff's Equation.



[Figure 3] Stribeck Curve

Relation μ and $\eta \frac{N}{p}$ in real bearing is like the Stribeck Curve.

The lubrication section is divided into the complete lubrication section (Fluid lubrication) and the Incomplete lubrication section (Boundary, mixture lubrication).

μ is proportional to $\eta \frac{N}{p}$ in the complete lubrication section, and this section is the most suited to Petroff's Equation.

The incomplete lubrication section is divided into the surface lubrication section and the mixture lubrication section. In the surface lubrication section, the friction coefficient is high and constant because of solid friction between the shaft and the bearing. In the mixture lubrication section, μ is inversely proportional to $\eta \frac{N}{p}$, and changes rapidly.

2-5. Principle of pressure formation on oil surface

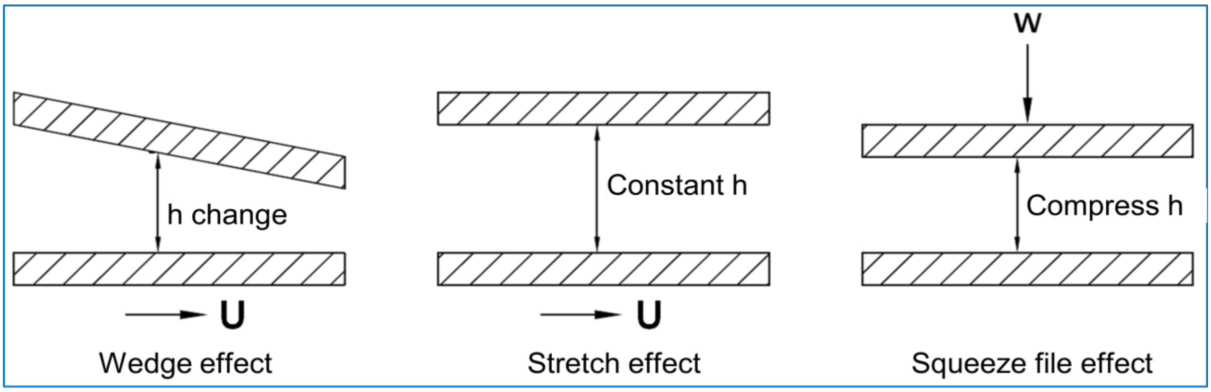
(1) The shaft is supported when pressure is formed in a bearing oil surface.

(2) The principle of pressure formation comes from the relative motion between the shaft and the bearing. It is divided into 3 effects as follows:

1) Wedge effect : The oil surface gets narrower and pressure increases because many molecules squeeze between the narrow surfaces.

2) Stretch effect : Fluid velocity gets slower as it get closer to the flow direction of the fluid. Pressure increases because of more fluid inflow than fluid outflow.

3) Squeeze file effect : Parallel plates get closer to each other. Fluid between the two surfaces escapes and the resistance to escape causes pressure to increase.



[Figure 4] Physical meaning of formation of oil surface

2-6. Factors for design of journal bearing

(1) Length to diameter (l/d)

1) Bearing diameter is defined by a shaft diameter.

2) Length to diameter (l/d) is bearing width divided by the diameter.

3) Mean pressure in bearing can be calculated by the load divided by a projected area of bearing ($d \times l$).

4) It is a base value to adjust mean pressure in bearing to a proper value and it is normally 0.25~2.

5) If l/d is low, the ability to support grows lower by high bearing pressure.

6) If l/d is high, the ability to support grows higher and the amount of oil leak grows lower by low bearing pressure. On the other hand, there is a possibility of releasing of friction heat and touching the shaft at the edge of bearing because the shaft is bent.

(2) Mean pressure of bearing (p_m value)

1) Mean pressure is radial load (P) divided by a subjected area (dl), expressed as p.

$$p_m = \frac{P}{dl}$$

(3) pV value

1) pV value is a product of mean pressure of bearing p and circumferential speed V and Expressed as pV or pv.

2) It is proportional to the temperature of the bearing.

3) If circumferential speed V is given, mean pressure p is limited.

4) Fluid viscosity decreases as bearing temperature increases by increase in the number of rotations.

5) Viscosity change affects the bearing property. Material, lubricant, and the way of lubricating are decided according to pV value
6) If pV value is high, the width of the bearing can be long. If pV value is low, the width of bearing should be short.
7) If the width of the bearing is too long, the bearing can be transformed or the shaft could make contact with the bearing due to a tilted shaft.

(4) Bearing modulus ($\eta \frac{N}{p}$)

1) An high-speed bearing is designed by the pV value but a common bearing is designed by $\eta \frac{N}{p}$.

2) As one can know in the Pedroff's Bearing Equation, $\eta \frac{N}{p}$ is the bearing modulus and it is related with the friction coefficient of bearing. Increasing friction coefficient is the cause of power loss and heating.

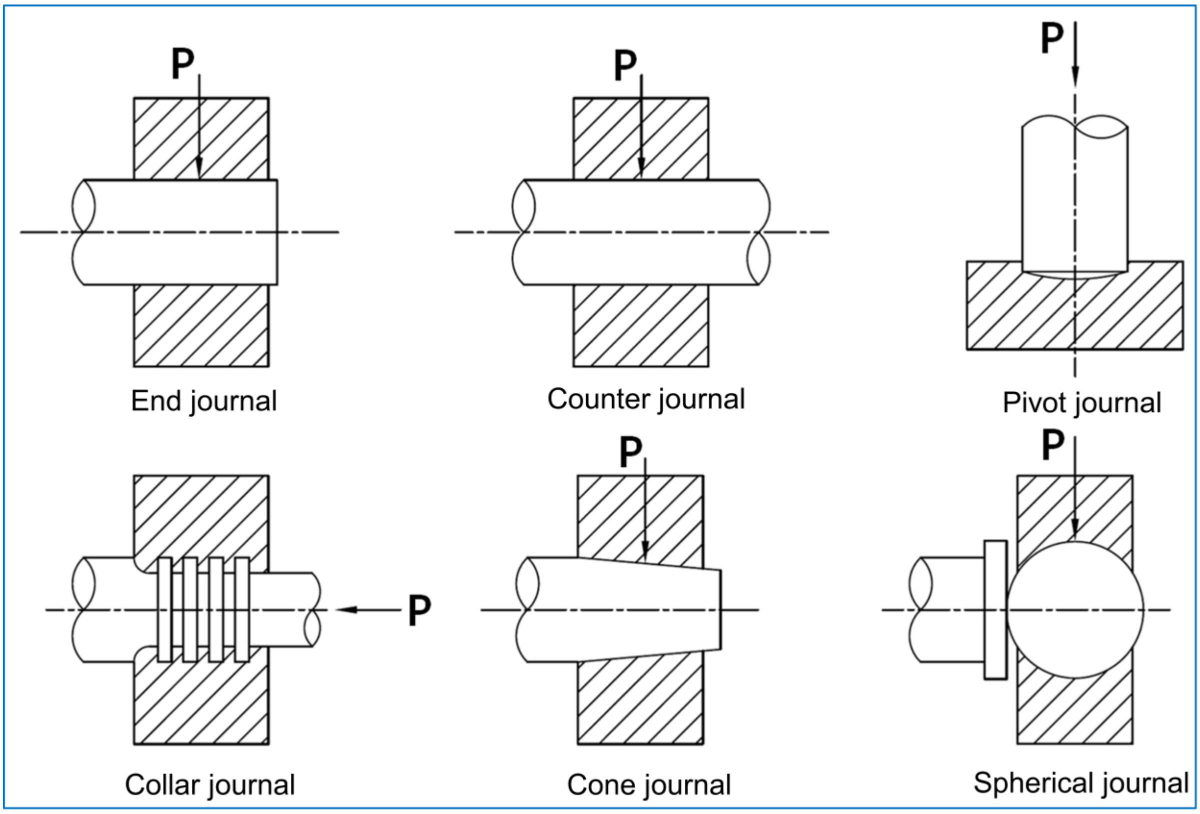
(5) Sommerfeld Number
1) It is also called the dimensionless load supporting force. The load that a bearing can support is expressed as a dimensionless type.
2) The Sommerfeld Number can be universally applied to design bearings in spite of differences in size and pressure. If the Sommerfeld Number of two bearings is same, we treat them as the same bearing.

$$S = \left(\frac{r}{\delta}\right)^2 \frac{\eta N}{p}$$

δ : Gap between the shaft and the bearing to the radius direction
r : Radius of a shaft
N : Rotational angular speed
p : Pressure

(6) Minimum thickness of oil surface ($h_{\min}$)
1) This means the minimum distance between the shaft and the bearing. It is a design factor to prevent sticking.
2) Sum of surface roughness of an shaft and a bearing should not exceed minimum thickness of oil surface.

2-7. Design of sliding bearing
(1) Classification of sliding bearing
1) Radial journal bearing : Supports load to the radius direction.
① There are end journals that support the edge of a shaft and counter journals that support the counter of a shaft.
2) Thrust journal bearing : Supports load to the shaft direction.
① There are pivot journals that support the load at the edge of an shaft and collar journals that support the load at the counter of a shaft.



[Figure 5] Classification of sliding bearing

2-7-1. Design of radial journal bearing

(1) Bearing pressure (p)

1) Bearing pressure to the shaft direction is the highest at the bearing center, and it is lowest at the bearing edge.

2) Pressure to the radius direction by the shaft rotation differs in the parts of the bearing. If we suppose pressure distribution is constant, mean pressure in the bearing (p) is force to the bearing(P) divided by a subjected area(dl).

(2) Design journal for shaft strength

Suppose mean pressure between an shaft and a bearing (p) is uniformly distributed throughout this shaft. Pressure is substituted for concentrated load to the shaft, the center of the bearing (P). Then, calculate shaft diameter that can bear the load. Design bearing diameter for shaft diameter because gap between the bearing and the shaft is much shorter than shaft diameter.

1) End journal

Substitute mean pressure in bearing (p) for concentrated load to the center of bearing (P), and treat the shaft like a cantilever. The width of the bearing is l.

* Diameter

Maximum bending moment is $M_b = \frac{1}{2} Pl$ and applied to the bending stress equation.

$$\sigma = \frac{M_b y_{\max}}{I_{yy}} = \frac{16Pl}{\pi d^3}$$

Solving for shaft diameter,

$$d = \sqrt[3]{\frac{16Pl}{\pi \sigma_a}}$$

σ_a : Allowable bending stress of a shaft

* Width
Load to bearing (P) is product of mean pressure in the bearing (p) and a subjected area (dl).

$$P = pdl$$

Eliminating P of the above equation, the equation for width l design is derived.

$$\frac{l}{d} = \sqrt{\frac{\pi}{16}\frac{\sigma_a}{p}}$$

σ_a : Allowable bending stress of a shaft
p : Mean pressure in bearing

2) Counter journal
* Diameter
Max bending moment is generated in the shaft, the center of the bearing.

$M_b = \frac{P}{2}\left(\frac{l}{2}+\frac{l_1}{2}\right) - \frac{P}{2}\frac{l}{4} = \frac{1}{8}PL$ is as follows when it is applied to the bending stress equation.

$$\sigma = \frac{M_b y_{\max}}{I_{yy}} = \frac{4PL}{\pi d^3}$$

Solving for shaft diameter,

$$d = \sqrt[3]{\frac{4PL}{\pi\sigma_a}}$$

$$L = l + 2l_1$$

σ_a : Allowable bending stress of a shaft

* Width
Load to bearing (P) is the product of mean pressure (p) to the bearing and a subjected area (dl).

$$P = pdl$$

Eliminating P of the above equation, the equation for width l design is derived.

$$\frac{l}{d} = \sqrt{\frac{\pi}{4e}\frac{\sigma_a}{p}}$$

$L = l + 2l_1 = el$ (e is normally 1.5)

σ_a : Allowable bending stress of a shaft
p : Mean pressure of bearing

(3) Power loss and friction heat
* Frictional force at bearing

$$F = \mu P$$

P : Pressure to the bearing in the radius direction

* Power loss by friction

$$H'[kW] = \frac{\mu P[N] \cdot v[m/s]}{1000}$$

$$H'[kW] = \frac{\mu P[kgf] \cdot v[m/s]}{102}$$

$$H[PS] = \frac{\mu P[kgf] \cdot v[m/s]}{75}$$

$$H[PS] = \frac{\mu P[N] \cdot v[m/s]}{735.5}$$

v : Circumferential speed of a shaft

* pv value is the product of mean pressure of the bearing and circumferential speed of a shaft.

$$pv = \left(\frac{P}{dl}\right) \cdot \left(\frac{d/2}{1000} \cdot \frac{2\pi N}{60}\right) = \frac{P}{dl} \cdot \frac{\pi d N}{1000 \times 60}$$

dl : A projected area of bearing to the radius direction

2-7-2. Design of thrust journal bearing

(1) Classification of journal bearing that supports load to the shaft direction

1) Collar bearing : Installed at the middle or many bearings are installed to support load to the shaft direction

2) Pivot bearing : Supports load at the edge of the shaft.

(2) If rotational shaft inclines to the bearing or vibrational amplitude is broad, one should take notice the shaft meets the bearing.

(3) Collar bearing

1) Bearing pressure (p)

$$p = \frac{P}{\frac{\pi}{4}(d_2^2 - d_1^2)Z}$$

P : Force to the shaft direction

$\frac{\pi}{4}(d_2^2 - d_1^2)$: Area by a collar

Z : The number of collars

2) pv value

pv value is the product of mean pressure and circumferential speed at the mean radius of collar surface.

$$pv = \frac{P}{\frac{\pi}{4}(d_2^2 - d_1^2)Z} \cdot \frac{(d_2 + d_1)/4}{1000} \cdot \left(\frac{2\pi N}{60}\right)$$

The equation for an external diameter d_2 and an internal diameter d_1 can be derived as:

$$d_2 - d_1 = \frac{P \cdot N}{30000 \cdot Z \cdot p \cdot v} \quad \text{[mm]}$$

P : Force to the shaft direction

N : Rotational angular velocity

Z : The number of collars

p : Bearing pressure

v : Circumferential speed at a mean radius

(4) Pivot bearing

Designing pivot bearings is the same as designing collar bearings of which collar number Z is one. If the center of the rotational shaft meets the bearing center, rotation direction at the center changes by vibration of shaft to the radius direction and direction of frictional force changes and causes an unstable rotational shaft. In addition, there is a possibility to destroy the oil surface due to high center pressure and high lubrication oil temperature. To prevent this phenomenon, one can round off the center.

By doing this, the center of rotational shaft doesn't meet the bearing center, pressure distribution changes are reduced, and the instability of the rotation shaft can be eliminated.

1) Bearing pressure (p)

$$p = \frac{P}{\frac{\pi}{4}(d_2^2 - d_1^2)}$$

P : Force to the shaft direction

$\frac{\pi}{4}(d_2^2 - d_1^2)$: The area of the pivot bearing

2) pv value

pv value is the product of mean pressure and mean radius at collar contact surface.

$$pv = \frac{P}{\frac{\pi}{4}(d_2^2 - d_1^2)} \cdot \frac{(d_2 + d_1)/4}{1000} \cdot \left(\frac{2\pi N}{60}\right)$$

The equation about an external diameter d_2 and an internal diameter d_1 can be derived as:

$$d_2 - d_1 = \frac{P \cdot N}{30000 \cdot p \cdot v} \text{ [mm]}$$

P : Force to the shaft direction
N : Rotational angular velocity
p : Bearing pressure
v : Circumferential speed at a mean radius

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