Chapter 12

Lubrication and Journal Bearings

Chapter Outline

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Introduction

- The object of lubrication is to reduce friction, wear, and heating of machine parts that move relative to each other.

- A lubricant is any substance that, when inserted between the moving surfaces, accomplishes these purposes.

- In the study of lubrication and journal bearings, additional fundamental studies, such as chemistry, fluid mechanics, thermodynamics, and heat transfer, must be utilized in developing the material.
Types of Lubrication

Five distinct forms of lubrication may be identified:

1. Hydrodynamic
2. Hydrostatic
3. Elastohydrodynamic
4. Boundary
5. Solid film

*Hydrodynamic lubrication* means that:

- the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent metal-to-metal contact,
- the stability thus obtained can be explained by the laws of fluid mechanics.

Hydrodynamic lubrication is also called *full-film*, or *fluid, lubrication*.
Types of Lubrication

*Hydrostatic lubrication* is obtained by introducing the lubricant, which is sometimes air or water, into the load-bearing area at a pressure high enough to separate the surfaces with a relatively thick film of lubricant. So, unlike hydrodynamic lubrication, this kind of lubrication does not require motion of one surface relative to another.
Elastohydrodynamic lubrication is the phenomenon that occurs when a lubricant is introduced between surfaces that are in rolling contact, such as mating gears or rolling bearings.
**Types of Lubrication**

*Boundary lubrication* happens when the highest asperities may be separated by lubricant films only several molecular dimensions in thickness. The reasons of this can be one of the following:

a- Insufficient surface area.

b- A drop in the velocity of the moving surface.

c- A lessening in the quantity of lubricant delivered to a bearing

d- An increase in the bearing load, or an increase in lubricant temperature resulting in a decrease in viscosity.
Solid film lubrication when bearings must be operated at extreme temperatures, a solid-film lubricant such as graphite or molybdenum disulfide must be used because the ordinary mineral oils are not satisfactory.
Viscosity

- In Fig. 12–1 let plate A be moving with a velocity $U$ on a film of lubricant of thickness $h$. We imagine the film as composed of a series of horizontal layers and the force $F$ causing these layers to deform or slide on one another just like a deck of cards.

- The layers in contact with the moving plate are assumed to have a velocity $U$; those in contact with the stationary surface are assumed to have a zero velocity.

- Intermediate layers have velocities that depend upon their distances $y$ from the stationary surface.

![Figure 12-1](image-url)
Newton’s viscous effect states that the shear stress in the fluid is proportional to the rate of change of velocity with respect to \( y \). Thus

\[
\tau = \frac{F}{A} = \mu \frac{du}{dy}
\]  
(12–1)

where \( \mu \) is the constant of proportionality and defines absolute viscosity, also called dynamic viscosity. The derivative \( du/dy \) is the rate of change of velocity with distance and may be called the rate of shear, or the velocity gradient. The viscosity \( \mu \) is thus a measure of the internal frictional resistance of the fluid. For most lubricating fluids, the rate of shear is constant, and \( du/dy = U/h \). Thus, from Eq. (12–1),

\[
\tau = \frac{F}{A} = \mu \frac{U}{h}
\]  
(12–2)
Viscosity

- Fluids exhibiting this characteristic are said to be *Newtonian fluids*.

- The absolute viscosity is measured by the pascal-second (Pa s) in SI; this is the same as a Newton-second per square meter.

*Figure 12-2*

A comparison of the viscosities of various fluids.
Petroff’s Equation

- The phenomenon of bearing friction was first explained by Petroff on the assumption that the shaft is concentric.

- Let us now consider a vertical shaft rotating in a guide bearing. It is assumed that the bearing carries a very small load, that the clearance space is completely filled with oil, and that leakage is negligible (Fig.12–3). We denote the radius of the shaft by \( r \),

![Figure 12-3](image-url)

**Figure 12-3**

Petroff’s lightly loaded journal bearing consisting of a shaft journal and a bushing with an axial-groove internal lubricant reservoir. The linear velocity gradient is shown in the end view. The clearance \( c \) is several thousandths of an inch and is grossly exaggerated for presentation purposes.
Petroff’s Equation

- the radial clearance by \( c \), and the length of the bearing by \( l \), all dimensions being in inches. If the shaft rotates at \( N \) rev/s, then its surface velocity is \( U = 2\pi r N \) in/s. Since the shearing stress in the lubricant is equal to the velocity gradient times the viscosity, from Eq. (12–2) we have

\[
\tau = \mu \frac{U}{h} = \frac{2\pi r \mu N}{c} \quad (a)
\]

- where the radial clearance \( c \) has been substituted for the distance \( h \). The force required to shear the film is the stress times the area. The torque is the force times the lever arm \( r \). Thus

\[
T = (\tau A)(r) = \left( \frac{2\pi r \mu N}{c} \right) (2\pi rl)(r) = \frac{4\pi^2 r^3 l \mu N}{c} \quad (b)
\]
Petroff’s Equation

- If we now designate a small force on the bearing by \( W \), in pounds-force, then the pressure \( P \), in pounds-force per square inch of projected area, is \( P = W/2rl \). The frictional force is \( fW \), where \( f \) is the coefficient of friction, and so the frictional torque is

\[
T = fWr = (f)(2rlP)(r) = 2r^2 f l P
\]

- Substituting the value of the torque from Eq. (c) in Eq. (b) and solving for the coefficient of friction, we find

\[
f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c}
\]

- Equation (12–6) is called Petroff’s equation and was first published in 1883.
Petroff’s Equation

- The \textit{bearing characteristic number}, or the \textit{Sommerfeld number}, is defined by the equation

\[ S = \left( \frac{r}{c} \right)^2 \frac{\mu N}{P} \]  \hspace{1cm} (12-7)

- The Sommerfeld number is very important in lubrication analysis because it contains many of the parameters that are specified by the designer. Note that it is also dimensionless. The quantity \( r/c \) is called the \textit{radial clearance ratio}. If we multiply both sides of Eq. (12–6) by this ratio, we obtain the interesting relation

\[ f \frac{r}{c} = 2\pi^2 \frac{\mu N}{P} \left( \frac{r}{c} \right)^2 = 2\pi^2 S \]  \hspace{1cm} (12–8)
Stable Lubrication

- The difference between boundary and hydrodynamic lubrication can be explained by reference to Fig. 12–4.

- A design constraint to keep thick-film lubrication is to be sure that

\[
\frac{\mu N}{P} \geq 1.7 \times 10^{-6}
\]
It is also helpful to see that a small viscosity, and hence a small $\mu N/P$, means that the lubricant film is very thin and that there will be a greater possibility of some metal-to-metal contact, and hence of more friction. Thus, point C represents what is probably the beginning of metal-to-metal contact as $\mu N/P$ becomes smaller.
Let us now examine the formation of a lubricant film in a journal bearing. Figure 12–5a shows a journal that is just beginning to rotate in a clockwise direction. Under starting conditions, the bearing will be dry, or at least partly dry, and hence the journal will climb or roll up the right side of the bearing as shown in Fig. 12–5a.

Now suppose a lubricant is introduced into the top of the bearing as shown in Fig. 12–5b.
Thick-Film Lubrication

- The action of the rotating journal is to pump the lubricant around the bearing in a clockwise direction. The lubricant is pumped into a wedge shaped space and forces the journal over to the other side.

- A *minimum film thickness* $h_0$ occurs, not at the bottom of the journal, but displaced clockwise from the bottom as in Fig. 12–5b. This is explained by the fact that a film pressure in the converging half of the film reaches a maximum somewhere to the left of the bearing center.

- Figure 12–5 shows how to decide whether the journal, under hydrodynamic lubrication, is eccentrically located on the right or on the left side of the bearing. Visualize the journal beginning to rotate. Find the side of the bearing upon which the journal tends to roll. Then, if the lubrication is hydrodynamic, mentally place the journal on the opposite side.
The nomenclature of a journal bearing is shown in Fig. 12–6. The dimension $c$ is the *radial clearance* and is the difference in the radii of the bushing and journal.
Thick-Film Lubrication

- In Fig. 12–6 the center of the journal is at $O$ and the center of the bearing at $O'$. The distance between these centers is the eccentricity and is denoted by $e$. The minimum film thickness is designated by $h_0$, and it occurs at the line of centers. The film thickness at any other point is designated by $h$. We also define an eccentricity ratio $\epsilon$ as

$$\epsilon = \frac{e}{c}$$

- The bearing shown in the figure is known as a partial bearing. If the radius of the bushing is the same as the radius of the journal, it is known as a fitted bearing. If the bushing encloses the journal, as indicated by the dashed lines, it becomes a full bearing. The angle $\beta$ describes the angular length of a partial bearing. For example, a 120° partial bearing has the angle $\beta$ equal to 120°.
Hydrodynamic Theory

- Reynolds pictured the lubricant as adhering to both surfaces and being pulled by the moving surface into a narrowing, wedge-shaped space so as to create a fluid or film pressure of sufficient intensity to support the bearing load.

- The important simplifying assumptions resulted from Reynolds’ realization that the fluid films were so thin in comparison with the bearing radius that the curvature could be neglected. This enabled him to replace the curved partial bearing with a flat bearing, called a plane slider bearing.
Hydrodynamic Theory

- Figure 12–9a shows a journal rotating in the clockwise direction supported by a film of lubricant of variable thickness $h$ on a partial bearing, which is fixed. We specify that the journal has a constant surface velocity $U$. Using Reynolds’ assumption that curvature can be neglected, we fix a right-handed $xyz$ reference system to the stationary bearing.
Hydrodynamic Theory

- Other assumptions made were:

1. The lubricant obeys Newton’s viscous effect, Eq. (12–1).
2. The forces due to the inertia of the lubricant are neglected.
3. The lubricant is assumed to be incompressible.
4. The viscosity is assumed to be constant throughout the film.
5. The pressure does not vary in the axial direction.
6. The bushing and journal extend infinitely in the $z$ direction; this means there can be no lubricant flow in the $z$ direction.
7. The film pressure is constant in the $y$ direction. Thus the pressure depends only on the coordinate $x$.
8. The velocity of any particle of lubricant in the film depends only on the coordinates $x$ and $y$. 
Hydrodynamic Theory

We now select an element of lubricant in the film (Fig. 12–9a) of dimensions $dx$, $dy$, and $dz$, and compute the forces that act on the sides of this element. As shown in Fig. 12–9b, normal forces, due to the pressure, act upon the right and left sides of the element, and shear forces, due to the viscosity and to the velocity, act upon the top and bottom sides. Summing the forces in the $x$ direction gives

$$
\sum F_x = p \, dy \, dz - \left( p + \frac{dp}{dx} \right) dy \, dz - \tau \, dx \, dz + \left( \tau + \frac{\partial \tau}{\partial y} \right) dx \, dz = 0
$$

In solving this equation and using the boundary conditions, at $y = 0$ the lubricant velocity $u = 0$ and at $y = h$ the lubricant velocity $u = U$ (see Figure 12-10), Reynolds found an approximate solution is due to Sommerfeld expressed by the form

$$
\frac{r}{c} f = \phi \left[ \left( \frac{r}{c} \right)^2 \frac{\mu N}{P} \right]
$$

(12–12)
where $\varnothing$ indicates a functional relationship. Sommerfeld found the functions for halfbearings and full bearings by using the assumption of no side leakage.
Design Considerations

- We may distinguish between two groups of variables in the design of sliding bearings. In the first group are those whose values either are given or are under the control of the designer. These are:

  1. The viscosity $\mu$
  2. The load per unit of projected bearing area, $P$
  3. The speed $N$
  4. The bearing dimensions $r$, $c$, $\beta$, and $l$

- Of these four variables, the designer usually has no control over the speed, because it is specified by the overall design of the machine. Sometimes the viscosity is specified in advance, as, for example, when the oil is stored in a sump and is used for lubricating and cooling a variety of bearings. The remaining variables, and sometimes the viscosity, may be controlled by the designer and are therefore the decisions the designer makes. In other words, when these four decisions are made, the design is complete.
In the second group are the dependent variables. The designer cannot control these except indirectly by changing one or more of the first group. These are:

1. The coefficient of friction $f$
2. The temperature rise $\Delta T$
3. The volume flow rate of oil $Q$
4. The minimum film thickness $h_0$

This group of variables tells us how well the bearing is performing, and hence we may regard them as *performance factors*. Certain limitations on their values must be imposed by the designer to ensure satisfactory performance. These limitations are specified by the characteristics of the bearing materials and of the lubricant. The fundamental problem in bearing design, therefore, is to define satisfactory limits for the second group of variables and then to decide upon values for the first group such that these limitations are not exceeded.
Significant Angular Speed

In the next section we will examine several important charts relating key variables to the Sommerfeld number. To this point we have assumed that only the journal rotates and it is the journal rotational speed that is used in the Sommerfeld number. It has been discovered that the angular speed $N$ that is significant to hydrodynamic film bearing performance is

$$N = |N_j + N_b - 2N_f|$$

(12–13)

Where $N_j = \text{journal angular speed, rev/s}$

$N_b = \text{bearing angular speed, rev/s}$

$N_f = \text{load vector angular speed, rev/s}$

When determining the Sommerfeld number for a general bearing, use Eq. (12–13) when entering $N$. Figure 12–11 shows several situations for determining $N$. 
Design Considerations

Figure 12-11

How the significant speed varies. (a) Common bearing case. (b) Load vector moves at the same speed as the journal. (c) Load vector moves at half journal speed, no load can be carried. (d) Journal and bushing move at same speed, load vector stationary, capacity halved.