Fluid Film bearings

Instructional Objectives:

At the end of this lesson, the students should be able to understand:

- Types of bearings
- Comparison of bearing friction characteristics
- Basics of hydrodynamic theory of lubrication
- Design methods for journal bearings

14.1.1 Brief overview of bearings

Bearings are broadly categorized into two types, fluid film and rolling contact type.

Fluid Film bearings

In fluid film bearing the entire load of the shaft is carried by a thin film of fluid present between the rotating and non-rotating elements. The types of fluid film bearings are as follows,

Sliding contact type
Journal bearing
Thrust bearing
Slider bearing

Rolling contact bearings

In rolling contact bearings, the rotating shaft load is carried by a series of balls or rollers placed between rotating and non-rotating elements. The rolling contact type bearings are of two types, namely,

Ball bearing
Roller bearing

14.1.2 Comparison of bearing frictions

The Fig. 14.1.1 shows a plot of Friction vs. Shaft speed for three bearings. It is observed that for the lower shaft speeds the journal bearing have more friction than roller and ball bearing and ball bearing friction being the lowest. For this reason, the ball bearings and roller bearings are also called as anti friction bearings. However, with the increase of shaft speed the friction in the ball and roller bearing phenomenally increases but the journal bearing friction is relatively lower than both of them. Hence, it is advantageous to use ball bearing and roller bearing at low speeds. Journal bearings are mostly suited for high speeds and high loads.

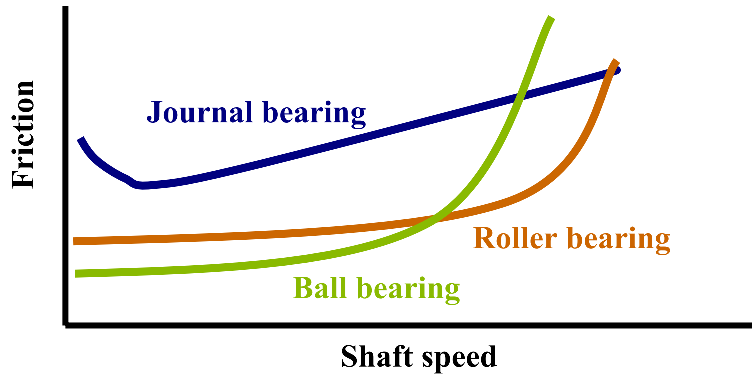


Fig. 14.1.1 Comparison of friction for different bearings

The ball and roller bearings require less axial space but more diametrical space during installation and low maintenance cost compared to journal bearings. Ball bearings and roller bearing are relatively costly compared to a journal bearing. The reliability of journal bearing is more compared to that of ball and roller bearings.

Here, we will discuss only about journal, ball and roller bearings, being most commonly used in design.

14.1.3 Journal Bearing

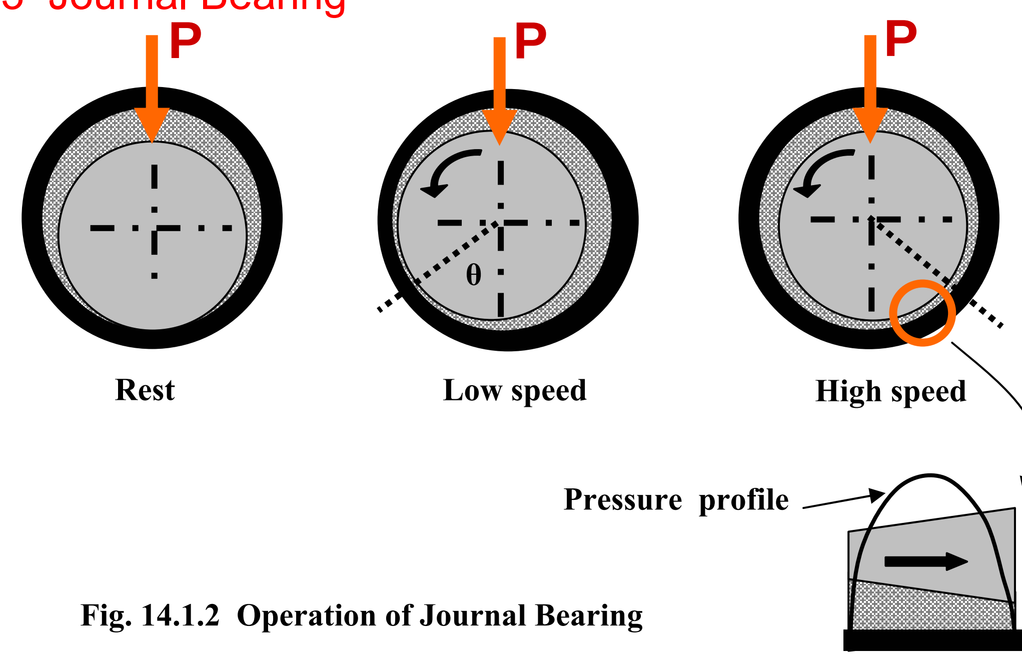


Fig. 14.1.2 Operation of Journal Bearing

Fig. 14.1.2 describes the operation of a journal bearing. The black annulus represents the bush and grey circle represents the shaft placed within an oil film shown by the shaded region. The shaft, called journal, carries a load P on it. The journal being smaller in diameter than the bush, it will always rotate with an eccentricity.

When the journal is at rest, it is seen from the figure that due to bearing load P, the journal is in contact with the bush at the lower most position and there is no oil film between the bush and the journal. Now when the journal starts rotating, then at low speed condition, with the load P acting, it has a tendency to shift to its sides as shown in the figure. At this equilibrium position, the frictional force will balance the component of bearing load. In order to achieve the equilibrium, the journal orients itself with respect to the bush as shown in figure. The angle θ, shown for low speed condition, is the angle of friction. Normally at this condition either a metal to metal contact or an almost negligible oil film thickness will prevail. At the higher speed, the equilibrium position shifts and a continuous oil film will be created as indicated in the third figure above. This continuous fluid film has a converging zone, which is shown in the magnified view. It has been established that due to presence of the converging zone or wedge, the fluid film is capable of carrying huge load. If a wedge is taken in isolation, the pressure profile generated due to wedge action will be as shown in the magnified view.

Hence, to build-up a positive pressure in a continuous fluid film, to support a load, a converging zone is necessary. Moreover, simultaneous presence of the converging and diverging zones ensures a fluid film continuity and flow of fluid. The journal bearings operate as per the above stated principle.

The background of hydrodynamic theory of lubrication

Petroff (1883) carried out extensive experimental investigation and showed the dependence of friction on viscosity of lubricant, load and dimensions of the journal bearing. Tower (1883 and later) also conducted experimental investigation on bearing friction and bearing film pressure.

The experimental investigations by Petroff and Tower form the background of the hydrodynamic theory. Later on Osborne Reynolds conducted experiments and published the findings in the form of present day hydrodynamic theory of lubrication and the corresponding mathematical equation is known as Reynolds' equation.

14.1.4 The Reynolds' equation (simplified form)

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right)=\frac{U}{2}\frac{\partial h}{\partial x}$$

(14.1.1)

where,

U : surface speed of the wedge, in x-direction
p : pressure at any point(x,z) in the film
μ : Absolute viscosity of the lubricant
h : film thickness, measured in y-direction

The left hand side of the equation represents flow under the pressure gradient. The corresponding right hand side represents a pressure generation mechanism. In this equation it has been assumed that the lubricant is incompressible and Newtonian. The wedge shape, that was discussed earlier, is assumed to be a straight profile as shown in Fig.14.1.3. The bearing is very long in the Z direction and the variation of pressure is in the X and Z direction.

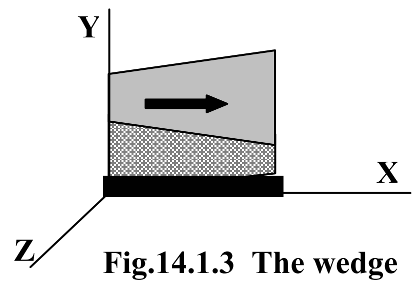
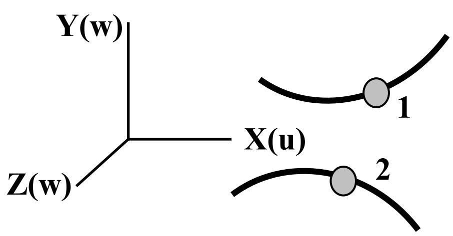


Fig.14.1.3 The wedge

Let us have a look at the right hand term in details.



$$\frac{\partial}{\partial x}\left(\rho\frac{u_1+u_2}{2}h\right)+\frac{\partial}{\partial y}\left(\rho\frac{v_1+v_2}{2}h\right)+\rho\frac{\partial h}{\partial t}+h\frac{\partial \rho}{\partial t} \qquad (14.1.2)$$

squeeze film **compression**

$$\rho\frac{U}{2}\frac{\partial h}{\partial x}+\frac{1}{2}\rho h\frac{\partial U}{\partial x}+\frac{1}{2}Uh\frac{\partial \rho}{\partial x} \qquad (14.1.3)$$

Physical wedge **stretch**

Fig. 14.1.4 Pressure generation mechanism

There are two moving surfaces 1 and 2 as indicated in Fig. 14.1.4. For 1 the velocities are u_1, v_1 and w_1 along the three coordinate axes X, Y and Z respectively. For 2, similarly the velocities are u_2, v_2 and w_2 respectively. Equation (14.1.2) represents the full form of the right hand side of Reynolds' equation. For the purpose of explanation, partial derivative of only the first term of equation (14.1.2) is written in equation (14.1.3). Here $u_1 + u_2$ have been replaced by U.

The first term of (14.1.3), $\rho \frac{U}{2} \frac{\partial h}{\partial x}$, represents a physical wedge. The second term $\frac{1}{2}(\rho h)\frac{\partial U}{\partial x}$ is known as the stretch. All the three terms of (14.1.3) contribute in pressure generation mechanism.

The term, $\rho \frac{\partial h}{\partial t}$ in equation (14.1.2) is called squeeze film; with respect to time how the film thickness is changing is given by this term.

The last term, $h \frac{\partial \rho}{\partial t}$ is the compressibility of the fluid with time and it is termed as compression.

The simplified form of the Reynolds's equation, (14.1.1), has only the physical wedge term, $\rho \frac{U}{2} \frac{\partial h}{\partial x}$.

14.1.5 Design parameters of journal bearing

The first step for journal bearing design is determination of bearing pressure for the given design parameters,

Operating conditions (temperature, speed and load)
Geometrical parameters (length and diameter)
Type of lubricant (viscosity)

The design parameters, mentioned above, are to be selected for initiation of the design. The bearing pressure is known from the given load capacity and preliminary choice of bearing dimensions. After the bearing pressure is determined, a check for proper selection of *design zone* is required. The selection of design zone is explained below.

Selection of design zone

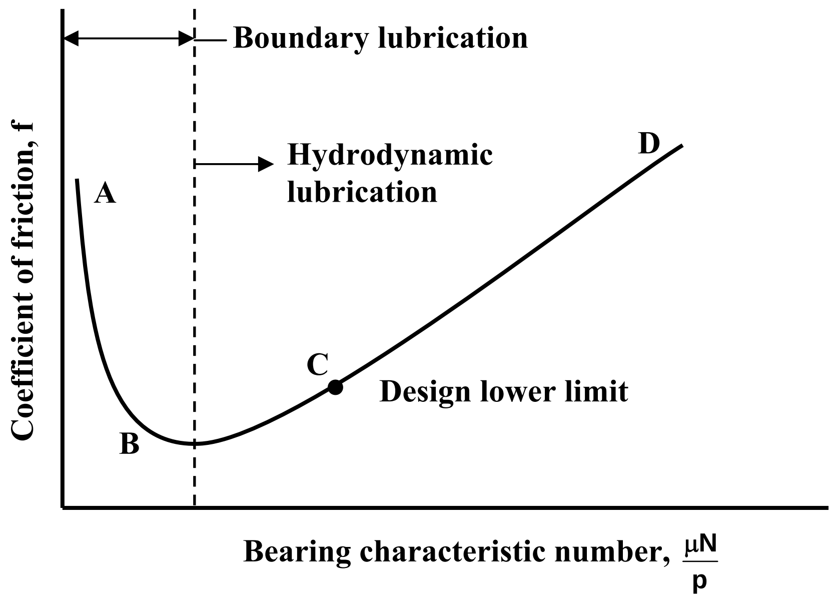


Fig. 14.1.5 Results of test of friction (McKee brothers)

The Fig. 14.1.5 shows the results of test of friction by McKee brothers. Figure shows a plot of variation of coefficient of friction with bearing characteristic number. Bearing characteristic number is defined as,

$$\text{Bearing characteristic number} = \frac{\mu N}{p}$$

It is a non-dimensional number, where μ is the viscosity, N is the speed of the bearing and p is the pressure given by $p = \frac{P}{dl}$, d and l being diameter and length of the journal respectively.

The plot shows that from B with the increase in bearing characteristic number the friction increases and from B to A with reduction in bearing characteristic number the friction again increases. So B is the limit and the zone between A to B is known as boundary lubrication or sometimes termed as imperfect lubrication. Imperfect lubrication means that metal – metal contact is possible or some form of oiliness will be present. The portion from B to D is known as the hydrodynamic lubrication .The calculated value of bearing characteristic number should be somewhere in the zone of C to D. This zone is characterized as design zone.

For any operating point between C and D due to fluid friction certain amount of temperature generation takes place. Due to the rise in temperature the viscosity of the lubricant will decrease, thereby, the bearing characteristic number also decreases. Hence, the operating point will shift towards C, resulting in lowering of the friction and the temperature. As a consequence, the viscosity will again increase and will pull the bearing characteristic number towards the initial operating point. Thus a self control phenomenon always exists. For this reason the design zone is considered between C and D. The lower limit of design zone is roughly five times the value at B. On the contrary, if the bearing characteristic number decreases beyond B then friction goes on increasing and temperature also increases and the operation becomes unstable.

Therefore, it is observed that, bearing characteristic number controls the design of journal bearing and it is dependent of design parameters like, operating conditions (temperature, speed and load), geometrical parameters (length and diameter) and viscosity of the lubricant.

14.1.6 Methods for journal bearing design

Broadly there are two methods for journal bearing design, they are,

First Method: developed by M. D. Hersey

Second Method: developed by A. A. Raimondi and J. Boyd

Method developed by M. D. Hersey

This method is based on dimensional analysis, applied to an infinitely long bearing. Analysis incorporates a side-flow correction factor obtained from the experiment of S. A. McKee and T. R. McKee (McKee Brothers).

McKee equation for coefficient of friction, for full bearing is given by,

$$\text{Coefficient of friction, } f = K_1 \frac{\mu N}{p} \frac{d}{c} + K_2$$

(14.1.4)

Where,

- p : pressure on bearing (projected area) = $\frac{P}{Ld}$
- L : length of bearing
- d : diameter of journal
- N : speed of the journal
- μ : absolute viscosity of the lubricant
- c : difference bush and journal diameter
- K_2 : side-flow factor = 0.002 for (L/d) 0.75-2.8

The constant K_1 is dependent on the system of units. For example, $K_1 = \frac{473}{10^{10}}$, when μ is in centipoise, p is in psi , N is in rpm and d and c in inches.

The steps to be followed are,

Basic design parameters are provided by the designer from the operating conditions. These are,

Bearing load (P)
Journal diameter (d)
Journal speed (N)

Depending upon type of application, selected design parameters are obtained from a design handbook, these are,

L/d ratio
Bearing pressure(p)
c/d ratio
Proper lubricant and an operating temperature

The heat generation in the bearing is given by,

$H_g = fPv$ where, v is the rubbing velocity

The heat dissipation is given by,

$H_d = KA(t_b - t_a)$

where,

A = projected bearing area
K = heat dissipation coefficient
t_b = bearing surface temperature
t_a = temperature of the surrounding

Next steps are as follows,

Value of $\frac{\mu N}{p}$ should be within the design zone
Equation (14.1.7) is used to compute *f*
Heat generation and heat dissipation are computed to check for thermal equilibrium.
Iteration with selected parameters is required if thermal equilibrium is not established.

Provision for external cooling is required if it is difficult to achieve thermal equilibrium.

The method described here is relatively old. The second method is more popular and is described below.

Method developed by A. A. Raimondi and J. Boyd

This method is based on hydrodynamic theory. The Reynolds equation (14.1.1) does not have any general solution. Assuming no side flow, Sommerfeld (1904) proposed a solution and defined a parameter, known as Sommerfeld number, given as,

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

(14.1.5)

where,
φ = A functional relationship, for different types of bearings

$\left[\left(\frac{r}{c}\right)^2 \frac{\mu N}{p}\right]$ = Sommerfeld number, S (dimensionless)

The Sommerfeld number is helpful to the designers, because it includes design parameters; bearing dimensions r and c , friction f , viscosity μ, speed of rotation N and bearing pressure p. But it does not include the bearing arc. Therefore the functional relationship can be obtained for bearings with different arcs, say 360^o , 60^o etc.

Raimondi and Boyd (1958) gave a methodology for computer–aided solution of Reynolds equation using an iterative technique. For L/d ratios of 1, 1:2 and 1:4 and for bearing angles of 360^0 to 60^0 extensive design data are available.

Charts have been prepared by Raimondi and Boyd for various design parameters, in dimensionless form, are plotted with respect to Sommerfeld number.
All these charts are for $360^0 - 60^0$ bearings.

The detailed work is given in *A solution of finite journal bearing and its application to analysis and design* Part I, II and III, Trans ASLE, Vol. 1 No.1, Lubrication Science and Technology, Pergamon, New York 1958, pp 159-202.

The design parameters which are given by Raimondi and Boyd are as follows, *Design parameters*

h_0/c : Minimum film thickness
$(r/c)f$: Coefficient of friction
$Q/(rcNL)$: Flow
Q_S/Q : Flow ratio
p/p_{max} : Maximum film pressure ratio
θ_{p_0}, deg : Terminating position of film
θ_{h_0}, deg : Minimum film thickness position

The above design parameters are defined in the Fig. 14.1.6. The pressure profile shown is only for the positive part of the bearing where the converging zone is present. Negative part has not been shown because it is not of use.

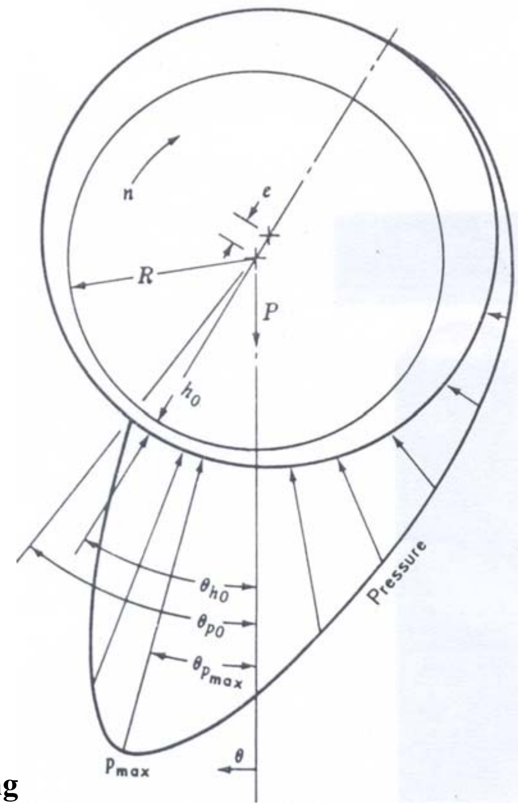


Fig.14.1.6 Nomenclature of a journal bearing

14.1.7 Materials for bearing

The common materials used for bearings are listed below.

Lead based babbits : around 85 % Lead; rest are tin, antimony and copper
(pressure rating not exceeding 14MPa)
Tin based babbits : around 90% tin; rest are copper, antimony and lead
(pressure rating not exceeding 14MPa)
Phosphor bronze : major composition copper; rest is tin, lead, phosphorus
(pressure rating not exceeding 14MPa)
Gun metal : major composition copper; rest is tin and zinc
(pressure rating not exceeding 10MPa)

Cast iron : pressure rating not exceeding 3.5 MPa

Other materials commonly used are, silver, carbon-graphite, teflon etc.

Questions and answers

Q1. Broadly, what are the types of bearings?

A1. Broadly bearings are of two types; Fluid Film bearings, where the entire load of the shaft is carried by a thin film of fluid present between the rotating and non-rotating elements and Rolling contact bearings, where the rotating shaft load is carried by a series of balls or rollers placed between rotating and non-rotating elements.

Q2. Highlight friction characteristics of bearings.

A2. For the lower shaft speeds the journal bearing have more friction than roller and ball bearing and ball bearing friction being the lowest. However, with the increase of shaft speed the friction in the ball and roller bearing phenomenally increases but the journal bearing friction is relatively lower than both of them.

Q3. Can a block moving over a constant height fluid film carry load?

A3. In this case the block can not carry any load. It can be shown mathematically that a wedge shaped fluid film can only generate pressure, thereby can withstand load.

Q4. What is Sommerfeld number? What importance it has in context of journal bearing design?

A4. Sommerfeld number is given by, $S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{p}$ with usual notations. This number includes design parameters; bearing dimensions r and c , friction f , viscosity μ, speed of rotation N and bearing pressure p. Only it does not include the bearing arc. Therefore for a given bearing arc, the Sommerfeld number indicates the operational state of a fluid film bearing.

References

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