CHAPTER 1
INTRODUCTION

Hydrodynamic journal bearings are considered to be a vital component of all the rotating machinery. These are used to support radial loads under high speed operating conditions. In a hydrodynamic journal bearing, pressure of hydrodynamic lift generates thin film of lubricating oil which separates the shaft and bearing thus preventing metal-to-metal contact. Journal bearing configurations are susceptible to large amplitude, lateral vibrations due to ‘self-exited instability’ known as oil whirl. Oil whirl is independent of shaft unbalance or misalignment. Forces generated in lubricating oil film due to hydrodynamic action cause a self-exited instability. During oil whirl shaft orbits in its bearing at a frequency approximately half the angular speed of shaft. If not controlled this non synchronous, self-excited orbiting motion will grow without bound and may lead to catastrophic failure. The stiffness of the shaft itself combined with the stiffness of bearing that support the journal determines several forms of natural frequencies of vibration called critical speed or threshold of whirl instability or stability of journal bearings.

If the shaft is considered to be rigid mass in connections with the fluid film spring there will be natural frequency of vibration. There is also disturbing force coming from residual unbalance due to variation in load and thus speed in the system. Therefore the resonant vibration will be at shaft rotation speed called synchronous whirl and has been observed as a precession or orbiting of the center of shaft about the center of the bearing. Thus for rotor dynamic system synchronous oil whirl is a more serious issue than critical speed resonance. From different experimental work it has been observed that lubricant viscosity plays a major role in oil whirl instability. Under normal operating conditions the lubricant undergoes a significant change in viscosity and other bearing performance parameters such as minimum oil film thickness and load carrying capacity. Viscosity variations due to changes in temperature and oil film thickness variations will affect the stability of journal bearing system.

1.1 Research Problem
Cylindrical journal bearings are widely applied in different rotating machineries. These bearings allow for transmission of large loads at mean speed of rotation. In some cases the rotating machine equipped in these bearings can operate at high speed.
When a bearing is operating at high speeds there is possibility of whirl instability. This limits the operating speed of journal. Therefore it is important to know the speed above which the bearing system will become unstable i.e. stability of journal bearing. The minimum oil film thickness plays significant role in the design of journal bearings. As \( h_{\text{min}} \) decreases the load carrying capacity increases, metal to metal contact occurs which leads to failure of journal bearings. In process industries incorporating journal bearings in different rotating machineries it is necessary to avoid catastrophic failure of journal bearing which can further avoid loss of economy and time.

1.2 Aim of Research
The main aim of the study was to do the stability analysis of hydrodynamic journal bearing theoretically using stiffness coefficients and experimental verification of same on journal bearing test rig. In addition experimental oil film thickness of hydrodynamic journal bearing is to be determined for different conditions of speed and load. Further to develop a technique to predict oil film pressure distribution and oil film thickness at different operating conditions which cannot be simulated experimentally on journal bearing system. Further the aim is to find transfer function of hydrodynamic journal bearing system using experimental values of oil film pressure and oil film thickness. Lastly to develop a control system for stability of oil film thickness in which oil film thickness is considered as controlled parameter and speed of journal as controlling parameter.

1.3 Research methods
The operation of hydrodynamic journal bearing was studied theoretically to determine synchronous whirl (stability) using stiffness coefficients and same stability speed was verified experimentally on journal bearing test rig. To determine oil film thickness experimentally in journal bearing system, inductive transducer is attached. Artificial Neural Network technique is used to predict pressure distribution in journal bearings at different operating conditions by using experimental data of pressure distribution. Similarly oil film thickness can be predicted at different operating conditions by using Artificial Neural Network technique. Simulation and determination of oil film pressure and oil film thickness is not possible for practical bearings due to different operating constraints, wherein Artificial Neural Network technique is a handy tool for prediction.
System identification technique is used to determine transfer function of hydrodynamic journal bearing system using oil film pressure and oil film thickness measurement. Transfer function determined by system identification is further used for development of simulation model of feedback control system in Matlab. Simulation results are further used for selection and design of PID controller which is required for development of actual feedback control system for stability of oil film thickness in journal bearing system. Oil film thickness is most direct parameter for monitoring journal bearing system as it is predictive in nature while other parameters rely on some amount of damage which has already taken place. On line condition monitoring of oil film thickness in hydrodynamic journal bearing to predict the sudden failure is developed by feedback control system which provides, control and remedy which is essential in real industrial situations. Attachment of inductive transducer to a journal bearing system will continuously record the oil film thickness. This is used as a controlled parameter and journal speed as controlling parameter. Control system receives the signal of oil film thickness from inductive transducer as a feedback signal, based on this feedback signal controller controls the speed of journal and thus maintains the oil film thickness. Thus a feedback control system that predicts and provides a remedy for failure of plain journal bearings is developed.

1.4 Scope of the Research
In present work stability analysis is carried out on journal bearing of specified dimensions (L/D = 1), speed range of 200 to 2000 rpm and load range of 150 to 600 N on a journal bearing test rig. Feedback control system for stability of oil film thickness is developed for hydrodynamic journal bearing system available in journal bearing test apparatus.

1.5 Introduction to journal bearings
Hydrodynamic journal bearing is a bearing operating with hydrodynamic lubrication, in which the bearing surface is separated from the journal surface by the lubricant film generated by the journal rotation. Most of engine bearings are hydrodynamic journal bearings. In a hydrodynamic journal bearing if journal rotates in a clockwise direction, Journal rotation causes pumping of the lubricant (oil) flowing around the bearing in the rotation direction. If there is no force applied to the journal its position will remain
concentric to the bearing position. However a loaded journal displaces from the
concentric position and forms a converging gap between the bearing and journal surfaces.
The pumping action of the journal forces the oil to squeeze through the wedge shaped
gap generating a pressure. The pressure falls to the cavitation pressure (close to the
atmospheric pressure) in the diverging gap zone where cavitation forms.
The oil pressure creates a supporting force separating the journal from the bearing
surface. The force of oil pressure and the hydrodynamic friction force counterbalance the
external load $F$. The final position of the journal is determined by the equilibrium
between the three forces. In the hydrodynamic regime the journal “climbs” in the rotation
direction (left side of the bearing). [1]

![Striback Curve](image)

**Fig.1.1 Striback Curve**

Journal bearing can operate in any of three lubrication regimes, thick-film lubrication,
thin-film lubrication, or boundary lubrication. Generally, thick-film operation is
preferred. Fig 1.1 is a diagram of the three lubrication regimes. Table 1.1 provides
characteristics of lubrication regimes. Journal bearing may be classified according to the
fluid mechanism that establishes the film load capacity. Hydrodynamic journal bearing,
also called self-acting bearings, depend entirely on the relative motion of the journal and
the bearing to produce film pressure for load support. Hydrostatic journal bearings, also
called externally pressurized bearings, achieve load support by the supply of the fluid
from an external high-pressure source and require no relative motion between journal and
bearing surfaces. Hybrid journal bearings are designed to use both hydrodynamic and hydrostatic principles to achieve load support between moving surfaces.

### Table 1.1 Characteristics of Lubrication Regimes

<table>
<thead>
<tr>
<th>Lubrication Regime</th>
<th>Bearing Surfaces</th>
<th>Range of film</th>
<th>Coefficient of friction</th>
<th>Degree of wear</th>
<th>Comments</th>
</tr>
</thead>
</table>
| Thick film         | Only during startup or stopping | $10^{-3}$ to $10^{-4}$ | 0.01-0.005 | None | 1. Light-loading high-speed regime  
2. Friction coefficient Proportional to $\mu N/(W/LD)$ |
| Thin film          | Intermittent dependent on surface roughness | $10^{-4}$ to $0.5 \times 10^{-4}$ | 0.005-0.05 | Mild | 1. High operating temperatures |
| Boundary           | Surface to surface | $0.5\times10^{-4}$ to molecular thickness | 0.05-0.15 | Large | 1. Heavy-loading  
2. Heat generating and friction not dependent on lubricant viscosity |

### 1.6 Types of Lubrication

Five distinct forms of lubrication are [1]

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

#### 1.6.1 Hydrodynamic Lubrication

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. Hydrodynamic lubrication does not depend upon the introduction of the lubricant under pressure, though that may occur, but it does require the existence of an adequate supply at all times. The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing. Hydrodynamic lubrication is also called full-film, or fluid lubrication.

Formation of a lubricant film in a journal bearing is shown in Fig. 1.2. A journal which is just beginning to rotate in a clockwise direction is shown. Under starting conditions, the
bearing will be dry, or partly dry, and hence the journal will climb or roll up the right side of the bearing as shown in Fig.1.2 Under the conditions of a dry bearing, equilibrium will be obtained when the friction force is balanced by the tangential component of the bearing load. When a lubricant is introduced into the top of the bearing as shown in Fig.1.2 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.1.2. This is explained by the fact that a film pressures in the converging half of the film reaches a maximum somewhere to the left of the bearing center. Fig. 1.2 shows how to decide whether the journal, under hydrodynamic lubrication, is eccentrically located on the right or on the left side of the bearing.

![Fig.1.2 Journal bearing with usual notations.](image)

The nomenclature of a journal bearing is shown in Fig.1.2 The dimension $c$ is the radial clearance and is the difference in the radii of the bearing and journal. In Fig.1.2 the center of the journal is at $0_1$ and the center of the bearing at $0_2$. The distance between these centers is the eccentricity and is denoted by $e$. The minimum oil 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 $h$. Eccentricity ratio $\varepsilon$ is define as,

$$\varepsilon = e / c$$ (1)
If the radius of the bearing is same as the radius of the journal, it is known as a fitted bearing. If the bushing encloses the journal, it is called as a full bearing.

1.6.2 Hydrostatic Lubrication
Hydrostatic lubrication is obtained by introducing the lubricant, which is sometimes air, water or oil into the load-bearing area at a pressure high enough by a pump 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.

1.6.3 Elastohydrodynamic lubrication
Elastohydrodynamic lubrication is the phenomenon that occurs when a lubricant is introduced between surfaces which are in rolling contact, such as mating gears or rolling bearings. The mathematical explanation requires the Hertzian theory of contact stress and fluid mechanics.

1.6.4 Boundary Lubrication
Insufficient surface area, a drop in the velocity of the moving surface, a decreasing in the quantity of lubricant delivered to a bearing, an increase in the bearing load, or an increase in lubricant temperature resulting in a decrease in viscosity-anyone of these-may prevent the buildup of a film thick enough for full-film lubrication. When this happens, lubricant films may separate the highest asperities only several molecular dimensions in thickness. This is called boundary lubrication. The change from hydrodynamic to boundary lubrication is not at all a sudden or abrupt one. It is probable that a mixed hydrodynamic-and boundary-type lubrication occurs first, and as the surfaces move closer together, the boundary type lubrication becomes predominant. The viscosity of the lubricant is not of as much importance with boundary lubrication as is the chemical composition.

1.6.5 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. Much research is currently being carried out in an effort to find composite bearing material with low wear rates as well as small frictional coefficients.

1.7 Sommerfeld Equation
1.7.1 Full Sommerfeld Equation
A rotating shaft is supported in a bush which completely or partially surrounds it with a small clearance as in Fig.1.3 (a). If a load is applied to the journal it will be displaced from the center, thus forming, as it rotates, a convergent clearance space which is conducive to the building up of a lubricating film to support the load.

As the pressurized film is created, the journal moves round the bearing in the same sense as the rotation until it reaches an equilibrium position as in Fig.1.3 (b). The line of centers of the journal and the bearing does not coincide with the line of action of the load, but is displaced by an angle $\Theta$. The distance between the centers, the eccentricity $e$, divided by the radial clearance of the bearing $C$, is called the eccentricity ratio $\varepsilon$. Obviously $\varepsilon=0$ represents concentricity, and $\varepsilon=1$ represents contact between the two surfaces. If the bearing is unwrapped, the form of the clearance can be clearly seen as in Fig.1.3(c). The film thickness at any point may be written as,

$$h = C (1+\varepsilon \cos \theta) \quad (2)$$

Providing $C/R\ll 1$. This is almost invariably the case, typical values of $C/R$ ranging from 0.0004 to 0.004. The film thickness may be substituted into Reynolds equation, together with $U1 = U, U2 = 0$ and $x = R\theta$

$$\frac{dP}{d\theta} = \frac{6\eta UR}{C^2} \left[ \frac{1}{(1+\varepsilon \cos \theta)^2} + \frac{h^*}{C(1+\varepsilon \cos \theta)^3} \right] \quad (3)$$

The solution of this equation was first achieved by Sommerfeld by means of the Sommerfeld transformation,

$$1+\varepsilon \cos \theta = \frac{1-\varepsilon^2}{(1-n \cos \gamma)} \quad (4)$$

But the resulting equation for $P$ contains two unknown constants, which require evaluation by the substitution of two boundary conditions.
Fig.1.3 (a) Journal bearing, (b) Journal bearing notation, (c) Development of journal bearing clearance

Boundary Conditions

Unlike the slider bearings, where we usually know that the pressures at the inlet and outlet of the film are ambient, the journal bearing, being a cyclic device, has a continuous film of fluid round its circumference. In the bearing of finite length, the lubricant can flow out and in from the bearing sides, but for the infinitely long journal bearing presently being considered this is precluded. Therefore the condition which must be satisfied is that

\[ p\theta = p_0 + 2\pi \]  \hspace{1cm} (5)
In practice this datum pressure is often the pressure of the lubricant supply where it is introduced into the clearance of the bearing. This supply may or may not be close to ambient pressure and, although always located in the lower pressure area of the bearing, it need not coincide with $\theta = 0$. Sommerfeld assumed a pressure $P_0$ at $\theta = 0$ and $\theta = 2\pi$. This produces the full Sommerfeld solution, with a pressure distribution given by following equation,

$$P - P_0 = \frac{6\eta \mu e R}{C^2} \frac{\sin(2 + e \cos \theta)}{(2 + e^2)(1 + e \cos \theta)^2}$$

(6)

As there is a divergent clearance present, impossibly large negative pressures can be predicted by these boundary conditions. In a few cases it is possible that these negative pressures are eliminated by a sufficiently high $P_0$, but in general the predicted distribution has the form shown in Fig.1.4 with the negative shaded portion.

![Fig.1.4 Pressure distribution](image)

Unlike the case of the discs, the inclusion of negative pressure, even with $P_0 = 0$, does not imply zero load capacity, since the positive and negative pressure areas occur in opposite halves of the bearing, and therefore both tend to move the journal in the same direction. Since the negative pressures would assist the bearing to support the load, the full Sommerfeld solution overestimates the load capacity.

To dismiss these large negative pressures, and to use the Reynolds cavitation boundary condition that is,

$$P = \frac{dP}{d\theta} = 0 \quad \text{at} \quad \theta = \theta_2 \pi < \theta < 2\pi$$

(7)

The inclusion of this condition increases the difficulty in solving equation, but the pressure distribution that is finally achieved is shown in Fig.1.4.
1.7.2 Half Sommerfeld Equation

Since almost half the clearance of the full journal bearing is occupied by low pressure or cavitated fluid, there are many cases where, provided that the load is in an approximately constant direction, this part of the bearing may be dispensed with altogether. Indeed, this is often a distinct advantage because, although the low pressure region contributes little to the load capacity, it does add to the viscous drag. The result is the partial journal bearing as shown in Fig.1.5

![Partial Journal Bearing](image)

**Fig.1.5 Partial Journal Bearing**

The partial journal bearing has the same form of film thickness as the full journal bearing, and may be analyzed using the Reynolds equation. The boundary conditions however, are different for partial bearings. The cyclic form is no longer present and usually the inlet boundary condition is that the pressure is ambient at the beginning of the bearing. \( P=0 \) at \( \theta=\alpha \). At the outlet it will either be sufficient to put the pressure equal to ambient at the end of the arc \( (\theta=\alpha+\beta) \), as in Fig.1.5 or the Reynolds boundary condition will have to be adopted, as in Fig.1.4 which condition is applicable will depend on the condition of operation and the length of the divergent clearance space.

1.7.3 Load Capacity of Journal Bearings

As the load on the journal is increased from zero, the centre of the journal will move into a more and more eccentric position, with decreasing minimum film thickness. If the Reynolds equation is applied using the full Sommerfeld solution, assuming no cavitation,
theory indicates that the journal moves at right angles to the load line as shown in Fig.1.6. In practice the locus of the journal centre is very different from the full Sommerfeld prediction, and the Reynolds boundary condition must be used adequately.

![Fig.1.6 Locus of shaft centre within clearance circle with increasing load](image)

The load capacity of a journal is expressed in terms of the Sommerfeld number, $S$, a dimensionless parameter which is given by,

$$ S = \frac{\eta N LD}{W} \left( \frac{R}{C} \right)^2 \quad (8) $$

and which depends upon the eccentricity of the bearing.

$$ \frac{W}{L} = \frac{\eta ND}{S} \left( \frac{R}{C} \right)^2 \quad (9) $$

This gives the load per unit width of the journal in terms of the Sommerfeld number and the properties of the fluid and the bearing. Values of the Sommerfeld number for different eccentricity ratios, assuming the Reynolds boundary condition and ambient pressure at $\theta =0$, are presented by Pinkus and Sternlicht. The angle $\varnothing$ between the line of centers and load line varies with the operating conditions, even if the bearing is designed so that the supply is located at $\theta =0$ for one particular case, it will not be so for other loads, speeds and viscosities. In practice it is seen that the performance is virtually unaffected by the position of the inlet within $+30^0$ of $\theta =0$. Indeed, there is remarkably little difference between the performance of the full journal bearing and $180^0$, or even $150^0$, partial bearing. However, if the inlet should move a considerable distance into the convergent zone, the load capacity will be reduced [2].

### 1.8 Reynolds Equation

The theory of hydrodynamic journal bearing is based on differential equation derived by Osborne Reynolds.
The pressure distribution as obtained with earlier boundary conditions can be applied here. Thus

\[ P = \frac{6\eta UR\varepsilon (2+\varepsilon\cos\theta)\sin\theta}{C^2 (2+\varepsilon^2)(1+\varepsilon\cos\theta)^2} \text{ for } 0 \leq \theta \leq \pi \]  

(10)