Chapter 6

“No amount of experimentation can ever prove me right; a single experiment can prove me wrong”

Albert Einstein

Development of Journal Bearing Test Rig and Experimental Studies using TiO$_2$ Nanolubricants

Details of the work involved in developing a Journal Bearing Test Rig used in the experimental analysis of TiO$_2$ nanolubricants is explained. The test rig was developed to measure the hydrodynamic pressures, friction force, and side leakage in the test bearing lubricated with TiO$_2$ nanolubricants at different additive concentrations. The performance of the test rig is validated for plain engine oil in comparison with the published results, as well as, the static performance characteristics obtained from the code for plain engine oil. Comparative results of the static performance characteristics of TiO$_2$ nanolubricants at varying concentrations is presented and analysed.

6.1 Introduction

Studies described in Chapter 3 enables the production of stable TiO$_2$ nanoparticle dispersions in lubricants without causing physical or chemical degradation of the base oil. Theoretical studies described in Chapter 4 and 5 reveals significant improvement in static and dynamic characteristics of journal bearings operating on TiO$_2$ nanolubricants. In line with the above observations, a Journal Bearing Test Rig (JBTR) is developed to demonstrate the feasibility of using TiO$_2$ nanolubricants in a conventional journal bearing.
As described in section 2.4 of the literature review, Free-Sleeve design of JBTR is prescribed as efficient in studying the static characteristics of journal bearings [307, 308, 309]. Hence, the new test rig developed as part of this work used Free-Sleeve design approach.

Initially, an existing journal bearing test rig in the Department of Mechanical and Manufacturing Engineering at MIT – Manipal developed by Dr K. Jagannath was used as the basis for this work. However, as the work progressed, difficulties experienced with the sensors and vibrational issues coupled with newer pressure measurement ideas led to the development of a new test rig. However, the author would like to acknowledge and state on record his gratitude for the freedom offered by the Department and Dr K. Jagannath for permitting the use of the existing test rig and allowing the use of components considered useful for the new test rig. The test rig allows the measurement of hydrodynamic pressure distribution, friction force, and side leakage of the journal bearing system.

### 6.2 Objectives

The primary objective of this section of the work is to experimental demonstrate the feasibility of using TiO$_2$ nanolubricants in journal bearings and obtain a comparative analysis of bearing performance with plain engine oil. More specifically, the objectives are listed below.

I. Develop a Journal Bearing Test Rig to measure hydrodynamic pressures, friction force, and side leakage journal bearing system.

II. Explore the possibility of measuring pressure distribution using sensor-in-shaft approach.

III. Validate the measured bearing performance characteristics with published results and computational code for plain lubricants.

IV. Obtain bearing performance characteristics of test bearing operating TiO$_2$ nanolubricants at different particle concentration.

### 6.3 Description of the Journal Bearing Test Rig

The JBTR consists of the following main units.

a. Drive system
b. Test bearing assembly
c. Closed circuit oil supply unit
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d. Bearing loading system
e. Pressure measurement system
f. Friction force measurement system
g. Data acquisition system

Fig. 103 illustrates a schematic presentation of the JBTR rig highlighting the components involved. Fig. 104 is a photograph of the assembled JBTR. Detailed description of the sub-assemblies and components of the test rig is presented below.

![Schematic of the Journal Bearing Test Rig](image_url)

Figure 103: Schematic of the Journal Bearing Test Rig.

### 6.3.1 Drive system

The drive unit of the JBTR consists of a 5 HP AC Motor with speed control unit coupled to a hollow shaft using belt drive. The drive unit allows for a maximum speed of 1500 rpm. The hollow mild steel machined shaft is mounted on two plummer blocks.
Dimensional details of the drive shaft presented in Fig. 105 is followed by a photograph of the machined drive shaft in Fig. 106.

Figure 104: Photograph of the Journal Bearing Test Rig

Figure 105: Dimensional details of the drive shaft (All dimensions are in mm)
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The location of test bearing within the shaft is machined using cylindrical grinding to obtain the desired bearing clearance. Fig. 107 shows the design of sensor – cap arrangement to hold the dynamic pressure sensor within the shaft at the mid-plane of the bearing area. The sensor – cap arrangement is also highlighted in Fig. 106(b). The design of the shaft is carried out with the following considerations:

- The shaft diameter was designed such as to obtain a bearing clearance of 150 microns between the bearing bore and the shaft diameter. The clearance value was decided in accordance with the primary nanoparticle size and also the machining capabilities of neighboring machining centers.
- After cylindrical grinding, the shaft diameter was reduced to 49.73 mm to obtain the required clearance.
- A hollow mild steel shaft was chosen to permit the positioning of a dynamic pressure sensor in the bearing mid plane.
- Appropriate support bearings were purchased to provide for a rigid-mount of the shaft.

Figure 106: Photograph of drive shaft. (a) full length shaft with one end mounted b) details of cap arrangement to mount pressure sensor on the shaft surface in the bearing mid-plane.
6.3.2 Test bearings

In accordance with the existing test rig in the Department of Mechanical and Manufacturing Engineering at MIT - Manipal, test bearing is designed to be of finite bearing class with 50 mm bore diameter and length. The bearing grooves in the test bearing were designed in accordance with ESDU data [42] and includes two axial grooves.

Two steel test bearings for the study is sponsored by Michell India Pvt. Ltd., Bangalore. The test bearings are provided with a 1 mm white metal (Babbitt) lining to protect the hardened bearing surface from wear. In addition to the steel bearings, two Nylon bearings are also fabricated. The Nylon bearings is also expected to assist in the initial run-in of the test rig without damage to the shaft surface. The axial groove dimensions in the test bearings were calculated in accordance with the ESDU data [42] as follows:

- The ratio of axial groove length to the bearing length $\frac{a}{b} = 0.8$, therefore, the length of the axial groove $a = b \times 0.8 = 50 \times 0.8 = 40 \text{ mm}$.
- The circumferential width of the bearing, $w = 0.25 \times d$, (angular extent of about 30°) where $d$ is the diameter of the bearing. Therefore, width $w = 0.25 \times 50 = 12.5 \text{ mm}$.
- The depth of the groove should not be less than 20 times the diametric clearance $C_d$. Therefore, $h \geq 20 \times 0.2 = 4 \text{ mm}$. Considering ease of machining, a groove depth of 5 mm is chosen.

Fig. 108 presents the dimensional details of the machined Nylon and Michell test bearings.
Fig. 109 provides the positions of the pressure tap holes drilled along the bearing circumference at the mid plane to lead pressurized oil to the pressure sensor. With reference to the literature review it was decided to have 12 pressure tap points of 1.5 mm diameter [319, 320]. As presented in Fig. 108, the oil supply drills and pressure tap lines are machined through the side surface of the bearing.

The positioning of the pressure taps is carried out with consideration to the direction of shaft rotation and the direction of the net load on the bearing. As shown in Fig. 110, the drill taps are so located that irrespective of the direction of shaft rotation and net load, three points will be present in the convergent region of fluid film. Photographs of the machined test bearings are shown in Fig. 111.

![Figure 109: Location of pressure tab holes along the bearing circumference.](image)

![Figure 110: Location of pressure tab holes in relation to load direction](image)
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Figure 108: Dimensional details of test bearings
6.3.3 Bearing Housing

Bearing housing is designed with the purpose of not only housing the test bearing but also to perform the following functions:

- Loading unit applying the net load on the bearing is mounted on the bearing housing.
- Drains are built into the bearing housing to collect the side leakage of lubricant.
- The bearing housing also supports the needle bearing mounted on the test bearing sleeve. The needle bearing with double race was selected such that the inside race will form a tight fit with the test bearing OD. This arrangement allows for the free rotation of the test bearing.

The bearing housing is machined on a Nylon block as this makes the bearing assembly lighter and easier to handle. Figs. 112 (a to c) provides the schematic of the bearing housing and the test bearing assembly.

![Photographs of the machined test bearings](image)

*Figure 111: Photographs of the machined test bearings*
As shown in Figs. 112 (a to c), the bearing housing is machined in split part design facilitating easier insertion of test bearing and needle bearing assembly. Fig. 113 presents photographs of the bearing housing components and assembly.

![Figure 112(a): LHS of the bearing housing](image)

![Figure 112(b): RHS of the bearing housing](image)

### 6.3.4 Loading system

The loading system in the JBTR is designed to apply a net load on the test bearing. The loading system uses a dead weight assembly load the test bearing in the downward direction. The loading unit also involves a tension mechanism to apply a corresponding load in the upward direction. Therefore, a net load acts on the bearing.
This arrangement permits us to apply a resultant load acting upwards and then decrease the load in steps to reach a zero net load and then increase the load in steps in the downward direction. Fig. 114 presents a schematic representation of the loading unit.
6.3.5 Closed circuit oil supply unit

Closed circuit oil supply unit includes a gear oil pump with filter, pressure regulation valves, side leakage collection unit, and transfer tubes. Gear oil pump used is a Willy Vogel make with 3-liter storage capacity. The pump possesses a built-in filter to screen the system debris. Oil supply pressure is regulated using two precision restrictor valves and monitored by a Bourdon pressure gauge, located at the delivery side of the pump. The pumped oil enters the bearing mid-plane through two 5 mm circular supply holes provided on the right-side face of the bearing leading to the axial-grooves as shown in Fig. 108. A rectangular oil collector was placed under the bearing to collect the end leakage of oil from the bearing and it was connected to oil sump through supply pipes. This ensured the continuous circulation of oil. The rig thus allowed the regulation of rotational speed, applied load and oil supply pressure.

6.3.6 Hydrodynamic pressure measurement systems

The developed journal bearing test rig attempts the measurement of hydrodynamic pressure distribution using two methods. In the first method, a single pressure measurement sensor is set up in the shaft itself, such that the pressure sensing face of the sensor is open
to the oil film converging on the shaft surface at the bearing mid-plane. The pressure sensor is held in its place with the sensor-cap arrangement described in section 6.3.1. During operation, the pressurized oil bears on the face of the sensor and the hydrodynamic pressure experienced by the sensor during complete orbit of the journal is measured. The lead wires from the flush sensor are taken out through the hollow shaft to the data acquisition system. The sensor-in-shaft pressure measurement approach is illustrated in Fig. 115.

Pressure sensor used in the test rig is a flush type diaphragm type pressure sensor of Honeywell make with maximum pressure limit of 20 bar (300 psi). An inline amplifier was connected between the sensor and the data acquisition board to amplify the output signal from the sensor.

The second method of pressure measurement is the conventional mode of hydrodynamic bearing pressure measurement process involving delivering of pressurized oil from the bearing zone to the sensor surface using oil tap lines machined in the bearing mid-plane. The arrangement is shown in Fig. 116. A developed view of the bearing area showing the location of the pressure tap holes is illustrated in Fig. 109.

As explained in section 6.3.2, 12 pressure taps are provided on the bearing area such that maximum holes are present on the positive pressure zone. The flush type diaphragm
sensor is used at each pressure tap alternatively to obtain the hydrodynamic pressure distribution.

![Figure 116: Pressure measurement process with the sensor placed outside the bearing](image)

The leads from the pressure sensor are connected to the amplifier through a Mercotac electrical connector. A schematic representation of the measurement process is illustrated in Fig. 117.

![Figure 117: Hydrodynamic pressure measurement process](image)

### 6.3.7 Friction measurement system

In theoretical analysis of fluid film lubrication, friction force developed within the oil film and affecting the journal rotation power requirement is evaluated by computing the
shear stresses within the layers of oil film between the tribo-surfaces. Experimental computation of friction force however, requires the free-sleeve mounting of the bearing causing it to float on the journal as explained in section 6.3.3. During operation of the test rig, journal rotation induces friction force within the oil film causing the free-sleeve bearing to also rotate with it. The accompanying tangential force acting on the bearing proportional to the bearing friction is sensed using a weighing balance. A schematic of the arrangement is shown in Fig. 118. Force measurement is carried out in two perpendicular directions and the average force is used in the analysis.

6.3.8 Data acquisition system

The data acquisition process is illustrated in Fig. 117 and is the final stage of pressure measurement process. The data acquisition system consists of a Quanser Q8 data acquisition board and related QUARC data acquisition software. The necessary sensor circuits are built using the available in-built components of the QUARC software. MATLAB interface is developed to connect with the QUARC software and the output is continuously monitored during measurement; the results are saved as a data file for further processing.

![Figure 118: Hydrodynamic friction measurement process](image)
6.3.9 Side leakage measurement system

The side leakage from the bearing, which is routed through the collector tray to the sump, as part of the closed-circuit oil supply unit is intercepted with a graduated beaker and flow rate is measured. Every measurement is done after a minimum of 15 min of operation to ensure formation of stable oil film.

### Highlights of development of Journal Bearing Test Rig

A Journal Bearing Test Rig is designed and fabricated to experimentally study the bearing performance characteristics of journal bearing systems running on TiO₂ nanolubricants. Key points related to the development of JBTR are listed below.

- The journal bearing test rig is designed in accordance with the free-sleeve approach, in which the test bearing is freely mounted on the journal.
- The JBTR allows for the measurement of hydrodynamic pressure distribution, friction force, and side leakage.
- The test bearing is a Babbitt lined finite journal bearing of 50 mm diameter of Michell make. Two axial grooved are machined in the test bearing in accordance with ESDU data [42].
- In addition to the conventional pressure tap approach of measuring hydrodynamic pressure distribution, an alternate approach of sensor-in-shaft method is attempted to obtain the pressure distribution.
- The loading system is designed to apply load in both upward and downward directions, such that a net resultant load acts on the test bearing.
- A 5 HP AC motor drives the journal with speed control of up to 1500 RPM. A Willy Vogel oil pump of 3-liter capacity provides for a closed-circuit oil supply.
- A flush type diaphragm type pressure sensor of Honeywell make, with maximum pressure limit of 20 bar (300 psi) is used to obtain the pressure distribution.

6.4 Procedure for Experimentation

The experimentation performed in this study dealt with obtaining the pressure distribution, friction force, and side leakage of journal bearing operating on TiO₂ nanolubricants. The concentration of TiO₂ nanoparticles is varied and the corresponding
variation in measured bearing properties is analyzed. The procedure followed for experimentation is listed below.

- Stable TiO$_2$ nanolubricant samples of 1-liter quantity is formulated using the two step nanolubricant formulation process described in Chapter 3.
- The formulated TiO$_2$ nanolubricant is transferred to the oil pump.
- The pump is started and the oil circulation is carried out without starting the journal drive. Complete closed circuit oil flow is enabled in about 5 minutes.
- With zero load on the bearing the motor is switched on and the speed is increased slowly to attain the desired speed.
- Continue running the rig at the constant speed for 5 minutes.
- The desired load is applied on the bearing in steps.
- The loaded set up is run for 15 minutes to ensure the formation of stable oil film.
- The pressure distribution is measured by connecting the flush diaphragm pressure sensor; alternating between all 12 pressure taps.
- The reading on the spring balance corresponding to the friction force is noted down.
- Side leakage is interrupted with a graduated beaker and the flow rate is noted down.
- The bearing is then unloaded in steps.
- Without switching off the oil pump, the motor drive speed is adjusted to zero.
- Motor is switched off followed by switching off the oil pump.

### 6.5 Results and Discussions

The developed test rig was initially run on plain engine oil and the obtained performance characteristics were compared with published results and with results from developed theoretical codes. Following the validation of the test rig performance, the test rig was used to analyse the performance characteristics of TiO$_2$ nanolubricants. The operating parameters used in the experimentation is listed in Table 20.

#### 6.5.1 Validation of JBTR performance

The experimentation initially attempted to obtain the hydrodynamic pressure distribution using the sensor-in-the-shaft approach, previously explained in section 6.3.6 and illustrated in Fig. 115. The pressure distribution that is obtained from this technique is plotted as shown in Fig. 119.
Table 20: Operating Parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft diameter D</td>
<td>49.75 mm</td>
</tr>
<tr>
<td>Bearing length L</td>
<td>50 mm</td>
</tr>
<tr>
<td>Radial clearance C</td>
<td>75 microns</td>
</tr>
<tr>
<td>Load range W</td>
<td>450, 850, and 1250 N</td>
</tr>
<tr>
<td>Shaft speed N</td>
<td>18 rps</td>
</tr>
</tbody>
</table>

Lubricant details

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil type</td>
<td>SAE30 Engine oil</td>
</tr>
<tr>
<td>Lubricant viscosity $\mu$</td>
<td>$0.1078 Pa.s @ 40^\circ C$</td>
</tr>
<tr>
<td></td>
<td>$0.0137 Pa.s @ 80^\circ C$</td>
</tr>
<tr>
<td>TiO$_2$ Nanolubricant concentrations in volume fractions</td>
<td>$\phi = 0.001, 0.005, 0.01, \text{ and } 0.02$</td>
</tr>
</tbody>
</table>

Pressure sensor details

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Make</td>
<td>Honeywell Model G CP</td>
</tr>
<tr>
<td>Range</td>
<td>0 – 300 psi</td>
</tr>
<tr>
<td>Accuracy</td>
<td>1% full scale</td>
</tr>
</tbody>
</table>

It is observed in the pressure plot that, two pressure peaks of magnitude 0.31 MPa appears in the distribution. Further, it is observed that the peaks are entirely responsive to the oil supply to the two axial grooves. The pressure peak is observed to reduce when the oil supply to the corresponding groove is reduced. This reveals that the pressure sensor is only able to read the oil pressures on both grooves and is not able to sense the hydrodynamic pressures within the land region. A possible reason for this behavior could be the inefficiency of the sensor in reading dynamic pressures. Considering a journal speed of 18 rps, the sensor will have to measure hydrodynamic pressures in the circular oil film in $1/18$ of a second. The flush type sensor employed has a rated capability of recording 500 readings per second. This would mean that 27 readings could be recorded in the one revolution. As shown in Fig. 119, these 27 readings exhibit two peaks which are found to vary with oil supply to the grooves. At very low speeds it was visually evident that pressure spikes were seen only when the sensor approaches the grooves. This leads us to believe that the sensor was not able to read the pressure variation in the $1/18$th of a second, time duration. A more sensitive sensor might resolve the issue. The approach of having the sensor in the shaft
makes the measurement process simple, however requirement of sensor with high dynamic sensitivity might make the process economically challenging. The technique needs to be developed further.

![Figure 119: Pressure distribution plot using sensor-in-shaft approach](image)

Following this, the hydrodynamic pressures are measured using the second technique, which is the conventional pressure tab method explained in section 6.3.6. In this method, the sensor is placed outside the bearing assembly as illustrated in Fig. 116. Pressurized oil from the bearing mid-plane is drawn to the sensor surface through short length tubing. The journal speed is kept constant at 18 rotations per second and the bearing is subjected to three loads of 450 N, 850 N and 1250 N. The obtained pressures are validated by comparing them with theoretical pressures obtained using Raimondi and Boyd charts [30]. Theoretical pressures corresponding to Sommerfeld numbers for the operating bearing parameters listed in Table 20 are read from the Raimondi and Boyd charts [30]. Sommerfeld numbers for the test conditions are computed using the standard equation given below.

\[
S = \left( \frac{R}{C} \right)^2 \frac{\mu N}{P} \tag{6.1}
\]

where, \( P \) is the unit pressure obtained as load per unit projected area.

\[
P = \frac{W}{LD} \tag{6.2}
\]
For the computed Sommerfeld numbers, the corresponding ratio of unit pressure $P$ to maximum theoretical pressure $P_{\text{max}}$ is read from Raimondi and Boyd chart [30]. The readings are presented in Table 21 along with the ratio of unit pressure $P$ to experimental maximum pressures $P_{\text{max}}$. The theoretical maximum pressures are also computed and compared with experimental maximum pressures. It is observed from Table 21 that, the experimental pressures are lower than the simulated theoretical maximum pressures. A maximum difference of ~20% is observed. This difference in pressures is most likely due to losses in the loading system. The loading system employed currently dead weights applied through wires wound around the bearing housing. A more sophisticated hydraulic loading unit with load cell measurements might provide a closer pressure reading.

Table 21: Comparison of theoretical (Raimondi Boyd Charts) and experimental values

<table>
<thead>
<tr>
<th>Load (N)</th>
<th>Sommerfeld Number $S = \left( \frac{R}{C} \right)^2 \frac{\mu N}{P}$</th>
<th>$P/P_{\text{max}}$ Theoretical maximum pressure $P_{\text{max}}$ MPa</th>
<th>$P/P_{\text{max}}$ Experimental maximum pressure $P_{\text{max}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>450</td>
<td>1.19</td>
<td>0.54</td>
<td>0.64</td>
</tr>
<tr>
<td>850</td>
<td>0.63</td>
<td>0.53</td>
<td>0.60</td>
</tr>
<tr>
<td>1250</td>
<td>0.43</td>
<td>0.52</td>
<td>0.59</td>
</tr>
</tbody>
</table>

Fig. 120 illustrates the experimental pressure distribution obtained using the second technique at a journal speed of 18 rps for the three loads specified in Table 20. The obtained hydrodynamic pressures are in agreement to the theoretical pressures; thus validating the experimentation. The results of the experimentation along with details of the test rig are published in article Binu et al. [36] attached in the Appendix as paper P6.

The experimentally obtained pressure readings are also compared with simulated pressures using the computational MATLAB codes developed as explained in Chapter 4. Sommerfeld number $S = \left( \frac{R}{C} \right)^2 \frac{\mu N}{P}$ is taken as the basis for comparison. For the operating conditions employed in the journal bearing test rig, as specified in Table 20, the Sommerfeld number is computed for the three specified loads at a fixed speed of 18 rps (1080 rpm) and

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presented in Table 21. For these Sommerfeld numbers, corresponding maximum pressure is theoretically computed using the code and compared with the experimental pressures.

The non-dimensional pressure is computed as \( \bar{p} = \frac{pC^2}{\mu UR} \). Considering operational parameters: fluid viscosity \( \mu = 0.1078 \) Pa-s, linear velocity \( U = 2.81 \) m/s (corresponding to 18 rps speed), shaft radius \( R = 24.875 \) mm, and radial bearing clearance \( C = 0.07785 \) mm, the dimensional pressures are computed and compared with pressures obtained theoretically. The maximum pressures obtained theoretically and experimentally for all three loads are compared in Table 22 provided below.

For a constant speed of 18 rps, at loads of 450 N, 850 N, and 1250 the hydrodynamic pressure distributions were obtained both theoretically and experimentally. The pressure distributions obtained are illustrated in Figs. 121 and 122 respectively. The secondary peak at 255° nodal coordinate is characteristic of misalignment effect in the journal bearing test rig, as specified in Jang and Khonsari [403].

As observed from Table 22, the maximum experimental pressures are in good agreement with simulated theoretical pressures (code) with a maximum of 14% difference. The experimental pressures are thus validated. The pressure measurement system is then
used to measure the hydrodynamic pressures of journal bearings lubricated with TiO$_2$ nanolubricants.

Table 22: Comparison of theoretical (code) and experimental pressures

<table>
<thead>
<tr>
<th>Load (N)</th>
<th>Sommerfeld Number $S = \left( \frac{R}{C} \right)^2 \frac{\mu N}{P}$</th>
<th>Theoretical maximum pressure in MPa $P_{\text{t max}}$ from Code in MPa (2)</th>
<th>Experimental maximum pressure $P_{\text{max}}$ in MPa (3)</th>
<th>Percentage difference between (2) and (3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>450</td>
<td>1.19</td>
<td>0.33</td>
<td>0.313106</td>
<td>0.28</td>
</tr>
<tr>
<td>850</td>
<td>0.63</td>
<td>0.64</td>
<td>0.6037145</td>
<td>0.56</td>
</tr>
<tr>
<td>1250</td>
<td>0.43</td>
<td>0.96</td>
<td>0.9087973</td>
<td>0.84</td>
</tr>
</tbody>
</table>

Figure 121: Hydrodynamic pressure distribution obtained from code for plain oil
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Fig. 122: Hydrodynamic pressure distribution experimentally obtained for plain oil

Highlights of JBTR pressure measurement process

Two approaches, i) sensor-in-shaft and ii) conventional pressure tap method, of hydrodynamic pressure measurements are attempted. The obtained pressure measurements are validated in comparison with published results. Key points related to the validation process are listed below.

- The sensor-in-shaft approach of pressure measurement failed to sense the hydrodynamic pressure distribution. Possible reasons for the failure could be: i) inability of sensor to acquire dynamic pressures and ii) Insufficient amplification of sensor output.

- Pressure tap method of acquiring hydrodynamic bearing pressures is successful and is used as the preferred method in the study to obtain the pressure distribution.

- Experimental maximum hydrodynamic pressures are found to be lower than the maximum pressures read from Raimondi-Boyd charts [30]. A maximum difference of $\sim$20% is observed [36].

- Experimental pressures are also compared with theoretical pressures simulated using developed computational code described in Binu et al. [33]. A 14% difference in maximum pressures is observed in the experimental pressures when compared with theoretical pressures.
6.5.2 Experimentation with TiO$_2$ nanolubricants

Stable TiO$_2$ based nanolubricant samples are prepared using the optimized formulation procedure explained in Chapter 3. The samples are prepared in 1-liter volume. To study the influence of varying concentrations of TiO$_2$ nanoparticle additives on the bearing characteristics, samples were prepared in four volume fractions of $\phi = 0.001, 0.005, 0.01, \text{ and } 0.02$. The prepared samples are then tested in the journal bearing test rig. The pressure distribution, friction force and side leakage are measured using systems described in section 6.3. The obtained bearing characteristics are compared with theoretical values obtained from the developed code explained in Chapter 4.

6.5.2.1 Hydrodynamic pressure distribution

Figs. 123 to 126 presents the hydrodynamic pressure distribution at bearing mid-plane for journal bearing operating on TiO$_2$ nanolubricants at volume fractions of $\phi = 0.001, 0.005, 0.01, \text{ and } 0.02$ respectively. Fig. 127 provides a sample comparison of experimental pressure distribution at bearing mid-plane for a TiO$_2$ volume fraction of $\phi = 0.01$ with the pressure distribution for plain engine oil at a constant load of 1250 N.

![Pressure Distribution for TiO$_2$ Nano-oil, $\phi = 0.001$](image)

*Figure 123: Hydrodynamic pressure distribution experimentally obtained for TiO$_2$ nanolubricant at a volume fraction of $\phi = 0.001$*  

As observed in the Figs. 123 – 127, the experimental results reveal that the hydrodynamic pressure distribution of journal bearing operating on TiO$_2$ nanolubricants
does not show any significant change when compared to experimental pressure distributions obtained for plain engine oil at similar conditions of load and speed. This is contrary to the reported theoretical observations of hydrodynamic pressures, while accounting for the increase in viscosity of nanolubricants, had predicted a ~40% increase in maximum pressures and load carrying capacity, as described in Chapter – 4 [32 - 35]. It is therefore apparent from the experimental results that, the addition of nanoparticles in small concentrations of $\phi = 0.001$, 0.005, 0.01, and 0.02 does not play any significant role in increasing the pressure distribution of journal bearings. One main reason for this effect could be the thinning of nanolubricants at high temperatures.

Continuous operation of more than 30 min, increases the temperature of the lubricant to more than 50 °C, at which the viscosity of the oil is reduced and is almost equal to the viscosity of plain oil [33]. Therefore, a thermohydrodynamic analysis of journal bearings operating on nanolubricants might provide a better theoretical insight on the operation of nanoparticle additives used in journal bearings. An additional factor to be investigated is the size of the primary nanoparticle additive size used in the formulation of nanolubricants. The current research does not consider variation in size of nanoparticles in the experimental analysis.

Figure 124: Hydrodynamic pressure distribution experimentally obtained for TiO$_2$ nanolubricant at a volume fraction of $\phi = 0.005$
Figure 125: Hydrodynamic pressure distribution experimentally obtained for TiO$_2$ nanolubricant at a volume fraction of $\phi = 0.01$

Figure 126: Hydrodynamic pressure distribution experimentally obtained for TiO$_2$ nanolubricant at a volume fraction of $\phi = 0.02$
6.5.2.2 Friction force

The friction force developed within the oil film is experimentally measured using the set-up described in section 6.3.7. The friction force measurements are carried out after stable pressure distribution was attained as sensed by the pressure sensors. The friction force is measured as a reaction force to the rotation of bearing sensed by weighing balance in both directions. The average force is then computed. Friction force is measured for both plain oil and TiO$_2$ nano-oils at concentrations of $\phi = 0.001$, 0.005, 0.01, and 0.02. Experimentally obtained friction forces at operating parameters listed in Table 20 are presented in Table 23 and compared with theoretical friction forces obtained from the developed computational code described in Chapter 4 and published in Binu et al. [32, 33].

Fig. 128 presents the theoretical variation in friction force for the operating conditions presented in Table 20. The friction forces are computed for loads ranging from 150 N to 1250 N using the developed computational code. The theoretical computations for TiO$_2$ based nanolubricants are obtained at volume fractions of $\phi = 0.001$, 0.005, 0.01, and 0.02. Computational code takes into account, the aggregate TiO$_2$ particle size value of 777 nm corresponding to a couple stress factor of $d = 0.03108$ as obtained in the analysis on static characteristics that was described in Chapter 4, and published in Binu et al. [32]. The code also takes into account the modified Krieger-Dougherty viscosity model in computing the
viscosity of nano-oils at different volume fractions and uses them in evaluating the friction forces.

Table 23: Experimental friction forces for plain oil and TiO$_2$ nanolubricants

<table>
<thead>
<tr>
<th>Speed in rps</th>
<th>Lubricant type</th>
<th>Load N</th>
<th>Eccentricity ratio $\varepsilon$</th>
<th>Experimental Friction Force (N)</th>
<th>Theoretical Friction Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>18 rps</td>
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Chapter 6: Experimental Bearing Studies using TiO$_2$ Nanolubricants

<table>
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Fig. 128: Variation in theoretical friction force with applied loads at different TiO$_2$ volume fractions, at 18 rps speed, and a couple stress parameter of 0.03108
It is observed in Fig. 128 that theoretical friction forces increase with the use of nanoparticle additives. Friction forces are also observed to increase with addition of higher TiO$_2$ nanoparticle concentrations. This increase in friction force is expected due to the related increase in shear stresses within the oil-film owing to the presence of particulate additives. This increase in shear stress is a result of increased lubricant viscosity as modeled by modified Krieger-Dougherty model [32]. However, this variation in friction force does not take into account the physical interactions of nanoparticles among themselves and with the tribo-surfaces. The theoretical model employed also doesn’t take into account the thermal variation in viscosity of nanolubricants. Thus, the results need to be experimentally validated.

Variation in experimental friction forces, presented in Table 23 as a function of bearing load is shown in Fig. 129.

![Figure 129: Variation in experimental friction force with applied loads at different TiO$_2$ volume fractions, at 18 rps speed.](image)

It is observed in Fig. 129 that, in contradiction to theoretical friction force variation depicted in Fig. 128, experimental friction forces are found to decrease with the use of TiO$_2$ nanoparticle additives at low volume fractions ranging of $\phi = 0.001$, 0.005, 0.01, and 0.02. At an applied load of 1050 N and 18 rps journal speed, a comparison of theoretical and experimental friction forces for plain oil and TiO$_2$ nanolubricants at different concentrations
are presented in Figs. 130. Fig. 131 provides the percentage difference in theoretical and experimental friction forces of TiO\(_2\) nanolubricants at different TiO\(_2\) nanoparticle concentrations in comparison to plain engine oil. It is observed in Figs. 130 and 131 that, friction force is found to decrease due to the addition of TiO\(_2\) nanoparticles. At a load of 1050 N and 18 rps speed, a decrease of 21% is observed for TiO\(_2\) nanolubricant, at a volume fraction of $\phi = 0.01$ in comparison to plain oil.

It is also seen in Fig. 129 that; friction force decreases with increase in applied load on the bearing. A decrease of ~ 15% was observed in friction force at 1250 N load in comparison to friction force at 250 N. Possible reasons for the above behaviour of TiO\(_2\) nanolubricants are summarized below:

- The theoretical framework currently employed in analysing nanolubricants does not consider the physical interactions of nanoparticles among themselves and interactions between nanoparticles and bearing surfaces. Published results on nanofluids [363, 251, 226] reveal that these physical interactions dominate the variation in viscosity due to the addition of nanoparticles and hence, results in decreased friction force.
The theoretical framework also does not take into account the decrease in viscosity of oil film due to temperature variation and hence does not reflect the actual variation in shear stress within the formed nanolubricant film.

With increasing load, the temperature rise becomes significant enough to decrease the oil viscosity. At low film thicknesses associated with high loads, the aggregate TiO$_2$ nanoparticles might play the role of ball bearings between journal and bearing surface and hence cause a decrease in friction force.

However, performing a thermohydrodynamic analysis might provide improved simulation of friction forces for bearings operating on nanolubricants. This might provide greater clarity on the decrease in friction force in bearings due to the use of nanoparticle additives.

### 6.5.2.3 Side leakage

Side leakage from the journal bearing operating on nanolubricant is experimentally measured at different loads and compared with the theoretical values obtained using the computational code described in Chapter – 4. The end leakage from the bearing is directed...
to a graduated beaker and the time taken for the collection of 50 ml of oil is noted down. The side leakage is expressed in m³/s.

Table 24 presents the experimental side leakage from the journal bearing at different bearing loads and running on nanolubricants at different TiO₂ nanoparticle concentrations. Table 25 provides the theoretical side leakages obtained from the computational code at different volume fraction and a couple stress parameter of $\overline{a} = 0.03108$ corresponding to an aggregate particle size of 777 nm.

### Table 24: Experimental Side Leakage

<table>
<thead>
<tr>
<th>Load N</th>
<th>$\phi = 0$</th>
<th>$\phi = 0.001$</th>
<th>$\phi = 0.005$</th>
<th>$\phi = 0.01$</th>
<th>$\phi = 0.02$</th>
</tr>
</thead>
<tbody>
<tr>
<td>650</td>
<td>5.52E-06</td>
<td>5.51E-06</td>
<td>5.53E-06</td>
<td>5.36E-06</td>
<td>5.00E-06</td>
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<tr>
<td>850</td>
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<td>5.75E-06</td>
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<td>1250</td>
<td>6.28E-06</td>
<td>6.24E-06</td>
<td>6.14E-06</td>
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### Table 25: Theoretical Side Leakage

<table>
<thead>
<tr>
<th>Load N</th>
<th>$\phi = 0$</th>
<th>$\phi = 0.001$</th>
<th>$\phi = 0.005$</th>
<th>$\phi = 0.01$</th>
<th>$\phi = 0.02$</th>
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<td>6.32E-07</td>
<td>4.97E-07</td>
<td>3.54E-07</td>
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<tr>
<td>850</td>
<td>8.46E-07</td>
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<td>9.64E-07</td>
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<td>8.91E-07</td>
<td>6.4E-07</td>
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</table>

Fig. 132 provides a comparison of experimental side leakages with theoretical side leakages obtained from the computational code. As seen in Fig. 132, experimental side leakages are considerably higher than the simulated side leakages obtained from the code. However, the trend in variation of experimental side leakage is consistent with theoretical results. The side leakage is found to increase with increasing load. The side leakage is also found to decrease with increasing volume fractions of TiO₂ nanoparticles. The significant mismatch between theoretical and experimental results could be due to the temperature variation in oil viscosity which is not considered in the analysis. As the fluid becomes thinner with increasing temperature, oil flow will increase. It is also to be noted that the influence of nanoparticle additives on side leakage is not of much significance, since the side flow is more dependent on speed and load considerations and not on the class of lubricants. Published results on couple stress fluids with polymer additives resulting in significant higher couple stress
parameters compared to nanolubricants also depict very little variation in side leakages in comparison to plain lubricants [102]

![Comparison of experimental side leakages with theoretical side leakages](image)

**Figure 132: Comparison of experimental side leakages with theoretical side leakages**

**Highlights on experimental bearing performance studies of journal bearings using TiO$_2$ nanolubricants**

The developed JBTR is used in studying the bearing performance characteristics of journal bearings running on TiO$_2$ nanolubricants. Key observations from the study is listed below.

- TiO$_2$ nanolubricant samples at volume fractions of $\phi = 0.001$, 0.005, 0.01 and 0.02 is tested in the developed in JBTR using a two-axial groove finite journal bearing.
- The bearings were subjected to loads of 450 N, 850 N, and 1250 N at a constant speed of 18 rps.
- The presence of TiO$_2$ nanolubricant additives is found to have negligible impact on the pressure distribution at the considered additive volume fractions and bearing loads.
6.6 Conclusions

A journal bearing test rig with proven capabilities to measure steady state bearing performance characteristics was designed and fabricated. Experimental studies were carried out to study the influence of TiO$_2$ nanoparticle lubricant additives on the bearing performance of two-axial groove journal bearing. Key points from the experimental work is highlighted below.

- The developed journal bearing rig test rig successfully provides hydrodynamic pressures developed within the lubricant film at loads ranging from 250 N to 1250 N at journal speed of up to 1500 rpm.
- The test rig also measures friction force within the oil film at different operating parameters.
- The experimental pressures obtained are validated in comparison with maximum pressures obtained theoretically from the code. Maximum experimental pressures obtained are found to be 14% lower than the theoretical pressures.
- It is also observed from the results that, addition of nanoparticle additives at volume fractions ranging from 0.001 to 0.02, does not affect the pressure distribution within the oil film.
- Studies also reveal that, friction force developed within the oil film is found to decrease with increasing loads. Results also reveal that, addition of nanoparticle additives, results in decreased friction forces developed within the oil film. This finding contradicts the theoretical results which predicts an increased friction force for nanolubricants. This points to the fact that, physical interactions of nanoparticles and thermal variation in viscosity of nanolubricants, which were not considered in the theoretical framework, are dominant effects which influences friction forces.
• Experimental measurements of side leakage reveal significant deviation from theoretical predictions. The measured side leakages are found to be considerably higher in comparison to predicted values. However, the increasing trend in experimental oil leakage with increasing loads is consistent with theoretical predictions.

• It is observed during experimentation that addition of nanoparticles is found to decrease side leakage from the bearing system, which is consistent with theoretical predictions.

It is concluded from the experimental studies that, the most significant influence of TiO$_2$ nanoparticle lubricant additives to SAE30 engine oil used in fluid film bearings is the decrease in friction force developed within the oil film. Nanolubricants with a couple stress parameter of $d = 0.03108$ at a concentration of 0.01 volume fraction is found to decrease the friction force by 20%.

### 6.7 Future scope of work

The functionality of the developed test rig could be enhanced to enable the study of more extensive bearing characteristics. Some of the additions and modifications required in the test rig is listed below.

I. More studies related to sensor-in-shaft approach of pressure measurements needs to be explored to enable its working. This method holds the potential to offer more accurate pressure readings in comparison to pressure tap approach.

II. Measurement of journal misalignment, and motion of the journal center using non-contact displacement transducer could enable the computation of eccentricity leading to enhanced bearing performance analysis.

Some of the additional studies that could be performed related to the performance analysis of TiO$_2$ nanolubricants are listed below.

I. The impact of TiO$_2$ nanolubricants on the surface characteristics of journal and sleeve could be studied to understand the impact of nanoparticle additives on surface wear, especially during system start-up and shut-down.

II. Particle size analysis of lubricant samples post usage in the test rig could offer insights into the effect of bearing shear on the aggregate nanoparticle size. This could have significant bearing on the dispersion stability of nanolubricants.
III. A more comprehensive study involving different nanoparticle, volume fractions, and particle size could offer deeper insights into the overall performance of nanolubricants.