MEASUREMENTS TO QUANTIFY THE EFFECT OF A REDUCED FLOW RATE ON TILTING PAD JOURNAL BEARING PERFORMANCE – STATIC AND DYNAMIC

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TRC interest

Reduction in flow rate $\rightarrow$ reduction in drag power loss and more efficiency, though with increased pad temperatures & drop in damping. Savings in pumping and lubricant storage make the case for low flow…

how low is a low flow rate enough to maintain reliability (and energy efficient) TPJB operation?
Objective and tasks

• To quantify the effects of lubricant flowrate on tilting pad bearing performance:
  • Drag power & load capacity
  • Pad metal temperatures
  • Force coefficients (K, C, M)

Prior art


Test Rig

Legacy of D. Childs & students
Test Rig Features

**Test-Rig Capabilities**

<table>
<thead>
<tr>
<th>Feature</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. rotor speed</td>
<td>16 kRPM</td>
</tr>
<tr>
<td>Max. applied static load</td>
<td>20 kN</td>
</tr>
<tr>
<td>Max. measurable torque</td>
<td>100 Nm</td>
</tr>
<tr>
<td>Max. supply oil flow rate</td>
<td>~20 GPM</td>
</tr>
<tr>
<td>Available shaft OD sizes</td>
<td>3.5”, 4”, 4.5”</td>
</tr>
<tr>
<td>Max. bearing length</td>
<td>3.5”</td>
</tr>
</tbody>
</table>

Strain gage torquemeter & coupling directly measures drag torque. Floating bearing on rigid rotor.
Test Rig Load Devices

- Pneumatic cylinder applies static load.
- Pair of hydraulic actuators deliver dynamic loads via stingers.
Test Bearing

TL bearing tested by Coghlan (D. Childs team) in 2015-ff
Test bearing – load between pads

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft diameter</td>
<td>4.0 in (101 mm)</td>
</tr>
<tr>
<td>Length</td>
<td>2.4 in (61 mm)</td>
</tr>
<tr>
<td>B radial cold clearance</td>
<td>4.52 mil (0.115 mm)</td>
</tr>
<tr>
<td>hot clearance (6 &amp; 12 krpm)</td>
<td>4.20 mil (0.106 mm)</td>
</tr>
<tr>
<td>Design pad preload</td>
<td>0.3</td>
</tr>
<tr>
<td>Spherical Pivot Offset</td>
<td>0.5</td>
</tr>
<tr>
<td>Pad Arc Length (°)</td>
<td>72°</td>
</tr>
<tr>
<td>AISI 1018 Pad Thickness</td>
<td>0.75 in</td>
</tr>
<tr>
<td>Babbitted pad surface</td>
<td></td>
</tr>
<tr>
<td>Lubrication condition</td>
<td>Single Orifice b/w pads, Flooded (with end seals)</td>
</tr>
<tr>
<td>Applied Load, $W$</td>
<td>2,135, 6,405, 12,810 N</td>
</tr>
<tr>
<td>Specific Load, $W/(LD)$</td>
<td>345, 1,034, 2,068 kPa</td>
</tr>
<tr>
<td>ISO VG46 oil at 60°C</td>
<td>16.4 cPoise &amp; 837 kg/m³</td>
</tr>
<tr>
<td>Thermocouples in pad and in oil supply outer annulus</td>
<td></td>
</tr>
</tbody>
</table>

303 psi
Oil supplied flow rate - theory

Flow rate ~ shaft speed

\[ Q = N_p \frac{1}{2} \left( \frac{1}{2} \Omega D \right) L C_r (1 - \lambda) \]

- \( N_p \): number of pads
- \( \Omega \): shaft speed (rad/s)
- \( D \): shaft diameter (m)
- \( L \): bearing axial length (m)
- \( C_r \): bearing radial clearance (m)
- \( \lambda \): hot-oil carry over coefficient.

Tests at two shaft speeds
1. 6 krpm (32 m/s) \( \rightarrow \) ~14.4 LPM
2. 12 krpm (64 m/s surface speed) \( \rightarrow \) ~28.8 4 LPM

VARY Flow from 150% \( \rightarrow \) 100% (nominal) \( \rightarrow \) 20% or less (if safe)
# Test Results

Load between pads (LBP)

2 shaft speeds x 3 static loads

<table>
<thead>
<tr>
<th>Shaft speed</th>
<th>6 and 12 krpm</th>
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<td>Specific Load, $W/(LD)$</td>
<td>345, 1,034, 2,068 kPa</td>
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</table>
Eccentricity is nearly parallel to load direction and increases with load and is much smaller as shaft speed doubles.

Journal eccentricity increases slightly as flow rate decreases → small impact on film thickness.

ISO VG46 inlet $T = 60^\circ C$
Eccentricity decreases with shaft speed and increases with load.

Journal eccentricity increases slightly as flow rate decreases. (semi-log scale).

Low flow does not produce a too small film thickness.
A very low flow (50% & below) does produce large increase in pad peak temperature. Load and shaft speed have minor effect.

Inflection in temperature vs flow due to uneven thermal field in supply annulus (more later).

ISO VG46 inlet $T = 60^\circ$C
A low flow (50% or less of nominal) produces a quick temperature rise. Load and shaft speed have negligible effect.

Larger than 100% flow rate produces no changes.

ISO VG46 inlet $T = 60^\circ C$
Shaft speed has an effect on exit oil temperature; more so than load.

Low flow rates produce a significant oil exit temperature rise. Alarming only for very low flows (20% nominal or less).

ISO VG46 inlet $T = 60^\circ$C
Drag torque drops quickly as flow rate decreases. Twice shaft speed $\Rightarrow \sim 2 \times$ drag torque.
Drag power drops quickly as flow rate decreases. Savings of 50% or more in drag power with low flow rate (40% or lower).

Overflow (> 100%) increases power consumption (~20%) for operation at 12 krpm (64 m/s).

ISO VG46 inlet $T = 60\degree C$
Force Coefficients

Load between pads (LBP)

2 shaft speeds x 3 static loads

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<thead>
<tr>
<th>Shaft speed</th>
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</table>
Procedure for force coefficients identification

Step 1: Apply loads and measure bearing motions

Apply forces with shakers ➔ pseudo-random frequency

\( F^1 = \text{Re} \left( \begin{bmatrix} F_X^1 \\ 0 \end{bmatrix} e^{i \omega t} \right) \)

\( F^2 = \text{Re} \left( \begin{bmatrix} 0 \\ F_Y^2 \end{bmatrix} e^{i \omega t} \right) \)

\( \omega \) is a set of frequencies \( = (1, 2, 3, \ldots, 17) \times 9.77 \text{ Hz} \).

Record bearing displacement \( z \) and acceleration \( a \)

\[
\begin{align*}
Z^1 &= \begin{bmatrix} x^1_{(t)} \\ y^1_{(t)} \end{bmatrix} = \begin{bmatrix} X^1 \\ Y^1 \end{bmatrix} e^{i \omega t} & a^1 \\
Z^2 &= \begin{bmatrix} x^2_{(t)} \\ y^2_{(t)} \end{bmatrix} = \begin{bmatrix} X^2 \\ Y^2 \end{bmatrix} e^{i \omega t} & a^2
\end{align*}
\]

EOM: Frequency domain

\[
\begin{bmatrix} K + i \omega C - \omega^2 M \end{bmatrix} \bar{z} = \bar{F} - M_{S} \bar{a}
\]

Find parameters:

\[
\rightarrow H = K - \omega^2 M + i \omega C
\]
Procedure for force coefficients identification

**Step 2: Estimate dry structure parameters**

\[ [K_S - \omega^2 M_S + i\omega C_S]z = \bar{F} \]

\[ \rightarrow H_S = K_S - \omega^2 M_S + i\omega C_S \]

**NO lubricant**

**Step 3: Bearing force coefficients = Lubricated system – Dry system**

\[ (K, C, M)_{\text{bearing}} = (K, C, M)_L - (K, C, M)_S \]

Bearing

Test system (Lubricated)

Dry structure
Direct stiffness $K_{YY}$ is a function of load more than shaft speed or even flow rate.

ISO VG46 inlet $T = 60^\circ\text{C}$
Direct damping $C_{yy}$ decreases as flow rate decreases (all pads starve). Significant drop for lowest load (345 kPa [50 psi]).
SSV Subsynchronous vibrations

6.5 kRPM, 345 kPa (50 psi) load, 0.36 LPM (4%)

Low frequency spectrum (SSV hash) recorded for operation with berry-berry 😊 low flow rates (and small load).

SSV “breathed in” and need to be excited.
Complete Set of Force Coefficients ($X$ & $Y$)...

upcoming
Other issues
Thermocouples affixed to bearing OD facing feed annulus near inlet orifices.
Temperatures in Oil Supply Annulus

For very low oil flow rates supplied → Temperatures around annulus not uniform & unsteady (likely not wetted surface). Difficult to control flow & oil inlet temperature (set 140F).

![Diagram showing temperature readings at different points A, B, C, D with flow rates 2.6 LPM, 3.2 LPM, 2.1 LPM, 3.6 LPM and temperature readings 250°F, 200°F, 150°F, 120°F.]

- 6kRPM, 2068 kPa (300 psi) load
- Temperature readings after 17 min
- Set inlet temperature $T = 60°C = 140°F$
Low flow limit found by reducing oil flowrate at a constant rotor speed and specific load until:

- 1) Pad Temperature exceeds 121°C (250°F) or
- 2) SSV vibration appears
- 3) Inlet temperature below target 60°C and/or annulus temperatures not uniform → Cannot maintain control flowrate and/or oil inlet temperature)

### Limit of Low Oil Supply

<table>
<thead>
<tr>
<th>Load</th>
<th>Flow</th>
<th>Limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>345 kPa</td>
<td>2% (0.36 LPM)</td>
<td>3</td>
</tr>
<tr>
<td>1034 kPa</td>
<td>10% (1.4 LPM)</td>
<td>3</td>
</tr>
<tr>
<td>2068 kPa</td>
<td>5% (1 LPM)</td>
<td>1</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Load</th>
<th>Flow</th>
<th>Limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>345 kPa</td>
<td>15% (4.3 LPM)</td>
<td>SSV</td>
</tr>
<tr>
<td>1034 kPa</td>
<td>15% (4.3 LPM)</td>
<td>1</td>
</tr>
<tr>
<td>2068 kPa</td>
<td>23% (6.8 LPM)</td>
<td>1</td>
</tr>
</tbody>
</table>
Reducing flow rate reduces power consumption. Yet How low is too low?

The minimum flow is application specific but must prevent too large pad/film temperatures to avoid:

- Babbitt failure
- Varnishing of pads or (long term) degradation of oil
- Collapse of load capacity with excessive reduction in stiffness and damping coefficients

Recall nominal flow rate at 6 krpm: ~ 14.4 LPM
Continuation Proposal to TRC

**Further Measurements to Quantify the Effect of a Reduced Flow Rate on Tilting Pad Journal Bearing Performance – Static and Dynamic (LOP & Evacuated Ends)**

How low is a low flow rate enough for safe (and energy efficient) TPJB operation?

Year II
Tasks - Year II

(a) Install bearing w/o end seals $\rightarrow$ evacuated configuration

(b) Conduct measurements at shaft speed=6 krpm and 12 krpm, and three static loads (50 psi to 300 psi), and decreasing flow rate to limit determined by (a) excessive pad temperatures (above 250F), or (b) onset and persistence of SSV, (c) inability to control set flow and oil inlet temperature.  
$\rightarrow$ drag torque (power loss), journal eccentricity, oil exit temperature, and pad (sub surface) temperatures.

(c) Perform dynamic load measurements $\rightarrow$ bearing stiffness, damping and virtual mass coefficients.

(d) Produce predictions of bearing performance for (flooded & evacuated) TPJBs.

(e) Correlate measurements against predictions & write technical report (MS Thesis).
# Budget 2019-20 (Year II)

**Quantify the Effect of a Reduced Flow Rate on Tilting Pad Journal Bearing Performance – Evacuated Bearing**

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Support for graduate student (20 h/week) x $ 2,300 x 12 months</td>
<td>$ 26,400</td>
</tr>
<tr>
<td>Fringe benefits (2.4%) and medical insurance ($422) x 12 months</td>
<td>$ 5,698</td>
</tr>
<tr>
<td>Tuition &amp; fees three semesters (24 ch) * Legacy student</td>
<td>$ 13,275</td>
</tr>
<tr>
<td>Conference travel and registration</td>
<td>$ 1,800</td>
</tr>
<tr>
<td>Supplies: bearing parts and rig ancillary parts</td>
<td>$ 2,827</td>
</tr>
</tbody>
</table>

**Total BUDGET**  

$ 50,000

Questions (?)