Computational Model for Tilting Pad Journal Bearings

THERMAL EFFECTS ON PAD DEFORMATIONS AND TILTING-PAD JOURNAL BEARING PERFORMANCE

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Previous TRC Work 2010 - 2014

- Develop XLTPJB® to predict \([K, C, M]\) coefficients of TPJBs. Code accounts for thermal energy transport and the (nonlinear) effects of pivot flexibility.
- Include pressure induced pad deformation into XLTPJB®.
- Extend flow model to include laminar, transition regime, and fully developed turbulent flow conditions.
- Validate XLTPJB® predictions vs. TAMU test data & predictions in literature.


Film Temperature and Heat Fluxes

- Thermal deformation depends on the boundary conditions of temperature and heat flow ($\Phi$) surrounding a pad.
- The heat fluxes in a pad depend on the heat transfer coefficients.
- Researchers assumed values for heat transfer coefficients which varied greatly.
Film Temperature and Heat Fluxes

- **Lumped Parameter Thermal Model**
  \[ \Phi = h_p(T - T_{s\text{ump}}) + h_s(T - T_s) \]
  \[ \Phi_{\text{pad}} \quad \Phi_{\text{shaft}} \]

- **Circumferentially Averaged Film Temperature**:
  \[ \bar{T}_{\text{film}} = \frac{1}{\theta_{pad}} \int T(\theta) \, d\theta \]

- **Radial Temperature Distribution in a pad**:
  \[ \bar{T}(r) = a \ln \left( \frac{r}{R_{in}} \right) + b \]

- Heat transfer coefficients come from empirical correlations.
Thermally Induced Pad Deformation

- **Circumferential:**
  \[ u_{(r, \hat{\theta})} = -\alpha \left[ a R_{\text{back}} \ln \left( \frac{R_{\text{back}}}{R_{\text{in}}} \right) \cos \hat{\theta} - a r \ln \left( \frac{r}{R_{\text{in}}} \right) + (r - R_{\text{back}} \cos \hat{\theta})(a - b') \right] \]

- **Axial:**
  \[ u'_{(z)} = \alpha \frac{\bar{T}_{\text{in}} - \bar{T}_{\text{back}}}{2t} z^2 \]

- **Inner Surface Deformation:**
  \[ \delta(\hat{\theta}, z) = u_{(R_{\text{in}}, \hat{\theta})} + u'_{(z)} \]
Thermally Induced Pad Deformation

- **Pad Inner Surface**

- **Shaft expands outward**

- **Housing expands outward or contracts (depending on installation)**

\[
C_{p,\text{hot}} = C_p - \Delta R_s + \Delta R_h \\
C_{b,\text{hot}} = C_b - \Delta R_s + \Delta R_h
\]
Modification to Film Thickness

- **Film Thickness Equation**

\[
H(\theta, z) = \delta(\theta, z) + C_{p, hot} + e_x \cos \theta + e_y \sin \theta \\
+ [\xi_{pivot} - C_{p, hot} + C_{b, hot}] \cos (\theta - \theta_p) \\
+ [\eta_{pivot} - R_{back} \delta_p] \sin (\theta - \theta_p)
\]

- **Parameters**
  - \(C_p\): Pad radial clearance
  - \(C_b\): Bearing assembled clearance
  - \(R_{back}\): Back of the pad radius
  - \(\delta_p\): Pad tilt angle
  - \(\xi_{pivot}, \eta_{pivot}\): Pivot radial and transverse deflections
  - \(\delta(\theta, z)\): Pad inner surface pressure and thermally induced deformation

# Predictions for a large size TPJB

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft rotational speed $\Omega$ [RPM]</td>
<td>500–3000</td>
</tr>
<tr>
<td>Shaft surface speed $\Omega R$ [m/s]</td>
<td>13–79</td>
</tr>
<tr>
<td>Specific Load $W/(LD)$ [MPa]</td>
<td>1–2.5</td>
</tr>
<tr>
<td>Load orientation</td>
<td>LBP</td>
</tr>
<tr>
<td>Number of pads</td>
<td>5</td>
</tr>
<tr>
<td>Shaft diameter [mm]</td>
<td>500</td>
</tr>
<tr>
<td>Pad thickness [mm]</td>
<td>72.5</td>
</tr>
<tr>
<td>Bearing axial length [mm]</td>
<td>350</td>
</tr>
<tr>
<td>Pad arc length</td>
<td>56°</td>
</tr>
<tr>
<td>Pivot offset</td>
<td>0.6</td>
</tr>
<tr>
<td>Pad clearance [(\mu)m]</td>
<td>300</td>
</tr>
<tr>
<td>Lubricant</td>
<td>ISO VG32</td>
</tr>
</tbody>
</table>

$C_p/R = 0.0012$

$L/D = 0.7$

Hagemann et al., 2013, ASME Turbo Expo 2013, pp. V07BT30A019
The peak pressure in TEHD prediction is 8\% larger than the measured magnitude, whereas, the THD prediction is 22\% smaller.
**Film Thickness**

- $N = 3000$ RPM
- $W/(LD) = 2.5$ MPa
- Pad Clearance = 300 μm

![Diagram of film thickness](image)

- Improve minimum film thickness prediction.
- Substantially improve prediction at the pad leading edge.
Film Temperature and Pad Deformation

\( N = 3000 \text{ RPM} \)
\( \frac{W}{(LD)} = 2.5 \text{ MPa} \)
\( \lambda = 0.8 \)

Pad Inner Surface

Predicted Flow = 658 LPM
Test Flow = 420 LPM

Increase in Preload
Stiffness and Damping Coefficients

- \([K, C]\) Model
- \(K_{yy} > K_{xx}\)
- Stiffness Orthotropy
- Better agreement in load direction
- Better agreement for \(N = 1500\) RPM
- \(C_{yy} > C_{xx}\)
# Predictions for a Four-Pad TPJB

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft rotational speed $\Omega$ [RPM]</td>
<td>7000-16000</td>
</tr>
<tr>
<td>Shaft surface speed $\Omega R$ [m/s]</td>
<td>38-85</td>
</tr>
<tr>
<td>Specific Load $W/(LD)$ [MPa]</td>
<td>0.7-2.9</td>
</tr>
<tr>
<td>Load orientation</td>
<td>LBP</td>
</tr>
<tr>
<td>Number of pads</td>
<td>4</td>
</tr>
<tr>
<td>Shaft diameter [mm]</td>
<td>101.59</td>
</tr>
<tr>
<td>Pad thickness [mm]</td>
<td>190</td>
</tr>
<tr>
<td>Bearing axial length [mm]</td>
<td>61</td>
</tr>
<tr>
<td>Pad arc length</td>
<td>72°</td>
</tr>
<tr>
<td>Pivot offset</td>
<td>0.5</td>
</tr>
<tr>
<td>Pad clearance [µm]</td>
<td>134</td>
</tr>
<tr>
<td>Lubricant</td>
<td>ISO VG46</td>
</tr>
</tbody>
</table>

$C_p/R = 0.0013$

$L/D = 0.6$

Journal Eccentricity

- Journal Eccentricity in load direction ($e_y$)
- $e_y$ increases for a higher load, but decreases for a higher speed.
- TEHD predictions correlate better with test data.

<table>
<thead>
<tr>
<th>$W/(LD)$</th>
<th>0.7 MPa</th>
<th>2.1 MPa</th>
<th>2.9 MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft Speed (RPM)</td>
<td>THD</td>
<td>TEHD</td>
<td>Test data</td>
</tr>
<tr>
<td>7000</td>
<td>57.3</td>
<td>45.0</td>
<td>40.3</td>
</tr>
<tr>
<td>10000</td>
<td>54.2</td>
<td>39.9</td>
<td>29.3</td>
</tr>
<tr>
<td>13000</td>
<td>54.9</td>
<td>38.6</td>
<td>20.9</td>
</tr>
<tr>
<td>16000</td>
<td>55.6</td>
<td>38.0</td>
<td>18.4</td>
</tr>
</tbody>
</table>
**Film Temperature**

- Over-predicted film temperature for loaded pads
- Under-predicted film temperature for unloaded pads
- Consistently over-predict max temperature.

\[ \lambda = 0.8 \]

- ↑ Load ⇒ Pad 1 and 2 Temperature ↑
- ↑ Load ⇒ Pad 3 and 4 Temperature ↑
Flow Rate

- Predicted flow is lower than the flow rate in the test.
- For \( W/(LD) = 2.1 \) and \( 2.9 \) MPa, inlet flow rate for pad 1 becomes negative and must be set to zero.
- Over-flooded
- Power loss is inversely proportional to flow rate.

![Graph showing total inlet flow rate vs. rotor speed for different test conditions and predictions.](image)

![Graph showing pad inlet flow rate vs. pad number for different pressures.](image)
Bearing Clearance and Preload

- Bearing Clearance reduces about 25%.
- Preload for the loaded pad becomes more than double, for unloaded pads it increases about 30%.

\[ m = \frac{C_p - C_b}{C_b} \]

- Includes both pressure and thermally induced deformations.

Graph showing:
- Cold Bearing Clearance
- Hot Bearing Clearance
- Estimates of hot bearing clearance from test data
- Prediction

Graphs comparing:
- Loaded Pads (1, 2)
- Unloaded Pads (3, 4)
- Cold Preload

Rotor speed N (RPM) vs. Hot Bearing Clearance (µm)
Stiffness and Damping Coefficients

- \([K, C, M]\) Model
- \(K_{yy} > K_{xx}\)
- Stiffness Isotropy (predictions)
- \(C_{yy} > C_{xx}\)
Conclusion

• The TEHD model reduces the discrepancy between THD prediction and test data for peak pressure by 14%, maximum temperature by 11%, and minimum film thickness by 56% (Haggeman et al. test bearing).

• *Hot* bearing clearance is about 25% smaller than *cold* bearing clearance.

• The preload of the loaded pads increases about 100% during operation.

• Including thermally induced deformation of the pad and expansion of the shaft and bearing housing increases the stiffness when thermal deformations are dominant (low loads, 0.7 MPa). At a high load (2.9 MPa), pressure induced deformation offsets this increase.

• The predicted damping coefficients by TEHD analysis are slightly smaller than those from THD analysis.

• For accurate temperature prediction, the flow rate must be similar to the flow rate set or monitored in the test.
Current Mixing Model

\( \lambda \) is an empirical coefficient

\[
Q_{\text{supply}} = Q_{i+1} - \lambda Q_i
\]

\[
T_{i+1} = \frac{Q_{\text{supply}}T_{\text{supply}} + \lambda Q_i T_i}{Q_{i+1}}
\]

Does not account for:
- Inner groove mixing with sump oil
- Heat convection from the pads
- Reverse leading edge flow
Reverse Leading Edge Flow

Low to Moderate Pressure Gradient (Specific Load < 1.7 MPa)

Pure Shear (Couette) Flow + Pure Pressure (Poiseuille) Flow = Shear & Pressure Flow

Steep Pressure Gradient (Specific Load > 1.7 MPa)

Pure Shear (Couette) Flow + Pure Pressure (Poiseuille) Flow = Shear & Pressure Flow
2016-2017 Proposal to TRC

Impose the actual inlet flow:
- Starved bearing
- Over-flooded bearing

Thermal Mixing model that includes:
- Heat flow due to inlet and outlet lubricant flows in the groove boundary.
- Heat convection from the pads and bearing housing.
- Accounts for the groove geometry.
### TRC Budget

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Support for graduate student (20 hr/wk) $2,400 x 12 months</td>
<td>$28,800</td>
</tr>
<tr>
<td>Fringe benefits (2.5%) and medical insurance ($360/month)</td>
<td>$4,995</td>
</tr>
<tr>
<td>Travel to (US) technical conference</td>
<td>$1,200</td>
</tr>
<tr>
<td>Tuition &amp; Fees three semesters ($363/credit hr)</td>
<td>$9,090</td>
</tr>
</tbody>
</table>

**Total Cost:** $44,085

XLTPJB® continues to improve and has a large database of comparisons. The model and GUI reduce the burden on the unseasoned user by calculating actual operating (hot) clearances, minimizing the specification of empirical parameters, and considering sound boundary conditions for a proper analysis with thermal effects.
Acknowledgements

Turbomachinery Research Consortium

Turbomachinery Laboratory Staff and Students

Questions?

Learn more at http://rotorlab.tamu.edu