A FE Model for Static Load in Tilting Pad Journal Bearing with Pad Flexibility

TRC-B&C-03-2013

Luis San Andrés
Mast-Childs Professor

Yingkun Li
Graduate Research Assistant

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TRC Project 32513/15196B

Computational Model for Tilting Pad Journal Bearings
Past TRC work 2010-12

Pivot flexibility reduces the force coefficients in heavily loaded tilting pad journal bearings (TPJBs).

TRC-B&C-01-2013

Developed XLTPJB®, benchmarked by test data, to predict K-C-M coefficients of TPJBs. Code accounts for thermal energy transport and the (nonlinear) effects of pivot flexibility.
Objective: Enhance TPJB code to accurately predict pad surface deformations

Hydrodynamic pressure $P$

Pad surface elastic & thermal deformations change bearing & pad clearances

$$K\delta = P + C\Delta T$$

$K$, Pad stiffness matrix
$P$, Fluid film pressure vector
$C$, Mechanical-thermal stiffness matrix
$\delta$, Pad displacement vector
Proposed work 2012-2013

- Build a 3-D FE ANSYS® model to obtain pad stiffness matrix. Reduce model with active DOFs.
- Implement oil feed arrangements (LEG, spray bar blockers etc.) in the FE model
- Construct new Excel GUI and Fortran code for XLTRC²
- Digest test data and continue to update predictions using enhanced code.

$40,984
TRC funded 2012-13

Under moderate to heavy loads, pivot and pad flexibility affect the static and dynamic forced response of TPJBS.

XLTPJB© includes pivot flexibility and delivers improved predictions but does not account for pad flexibility.

Research objective:
Extend TPJB© code with effects of pad flexibility.

$40,984
Tasks completed

• Built FE structural models and extracted reduced stiffness matrices from ANSYS®

• Included pad flexibility into XLTPJB© code for static force case

• Compared predictions from the TPJB code against test data

• Composed a guide for pad FE analysis with ANSYS®
FE pad model with ANSYS®

**8-noded, isoparametric element**

Degrees of freedom (DOFs) of each node: $u_r, u_\theta, u_z$

NO ROTATIONS

$$K^e u^e = Q^e + S^e$$

- $K$, stiffness matrix
- $u$, Displacement load vector
- $Q$, Load vector
- $S$, Vector of surface tractions

FE Model built in ANSYS®

Cylindrical coordinate ($r, \theta, z$)

Assemble over whole domain

$$K^G u^G = Q^G + S^G$$
Pad reduced stiffness matrix

Boundary conditions: Desbordes’s model [1]

- Constraints along two lines:
  \[ u_r = 0 \]

- Constraints at a point (pivot):
  \[ u_r = u_\theta = u_z = 0 \]

Loads: pressure field acting on pad upper surface

\[ K u_p = f \]

- \( K \), Reduced stiffness matrix
- \( u_p \), Displacements load vector
- \( f \), Loads vector

K is symmetric positive definite, then **Cholesky decomposition** is applicable:

\[ K = LL^T \]

\[ Ku_p = L \left( L^T u_p \right) = Lu_{p1} = f \]

- **System of linear equations**
- **Back substitution**

\[ Lu_{p1} = f \quad \rightarrow \quad u_{p1} \quad \rightarrow \quad L^T u_p = u_{p1} \quad \rightarrow \quad u_p \]

**L** is calculated prior to running **XLTPJB®. Savings in processing time!**
Fluid film thickness in a pad

**Symbols**
- $C_p$: Pad radial clearance
- $C_B = C_p - r_p$: Bearing assembled clearance
- $R_d = R_p + t$: Pad radius and thickness
- $r_p$: Pad dimensional preload
- $\delta_p$: Pad tilt angle
- $\xi_{piv}$, $\eta_{piv}$: Pivot radial and transverse deflections
- $u_p$: Pad upper surface deformation

**Equation**

\[
h^k = C_p + e_X \cos \theta + e_Y \sin \theta \\
+ \left( \xi_{piv}^k - r_p \right) \cos \left( \theta - \theta_p^k \right) \\
+ \left( \eta_{piv}^k - R_d \delta_p^k \right) \sin \left( \theta - \theta_p^k \right) + u_p^k
\]
Predictions for a four-pad TPJB

(Tschoepe) Four pad, Rocker-back tilting pad bearing (LOP)

Specific load, $\frac{W}{LD}$: 0 MPa - 2.9 MPa (421 psi)
Journal speed, $\Omega$: 6.8krpm-13.2krpm

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of pads, $N_{pad}$</td>
<td>4</td>
</tr>
<tr>
<td>Configuration</td>
<td>LOP</td>
</tr>
<tr>
<td>Rotor diameter, $D$</td>
<td>101.75 mm (4 inch)</td>
</tr>
<tr>
<td>Pad axial length, $L$</td>
<td>60.33 mm (2.4 inch)</td>
</tr>
<tr>
<td>Pad arc angle, $\Theta_P$</td>
<td>72°</td>
</tr>
<tr>
<td>Pivot offset</td>
<td>57%</td>
</tr>
<tr>
<td>Preload, $\bar{r}_P$</td>
<td>0.589(pad1) 0.457(pad2)</td>
</tr>
<tr>
<td>Pad clearance, $C_P$</td>
<td>112 µm (4.4mil)</td>
</tr>
<tr>
<td>Pad inertia, $I_P$</td>
<td>1.81kg.cm² (0.618lb.in²)</td>
</tr>
<tr>
<td>Oil inlet temperature</td>
<td>~43.3 °C (110 °F)</td>
</tr>
<tr>
<td>Lubricant type</td>
<td>ISO VG32</td>
</tr>
<tr>
<td>Supply viscosity, $\mu_0$</td>
<td>0.023 Pa.s</td>
</tr>
</tbody>
</table>

Light load  

Rotor speed $\Omega = 6.8$ krpm  
Specific load $W/LD = 726\text{kPa}$

Film thickness and pad deformation:

Deformation of loaded pad (#1) is very small compared to film thickness.
Large load

Rotor speed $\Omega = 6.8$ krpm
Specific load $W/LD = 2.9$ MPa

Film thickness and pad deformation:

Deformation of loaded pad (#1) is 30% of the minimum film thickness

Journal eccentricity vs. static load

Rotor speed $\Omega = 6.8$ krpm

Max. 421 psi

Journal eccentricity agrees well with test data. Pad flexibility affects little the journal eccentricity.

Predictions agree well with temperature in loaded pad.

Large load

Rotor speed $\Omega = 6.8$ krpm
Specific load $W/LD = 2.9$MPa

Pad temperatures

Predictions underestimate the temperature

Rotor speed $\Omega = 6.8$ krpm
Specific load $W/LD = 2.9$MPa

Predictions for a five-pad TPJB

Five pad, Rocker-back tilting pad bearing (LBP)

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific load, $W/ LD$</td>
<td>1MPa-2.5MPa (363 psi)</td>
</tr>
<tr>
<td>Journal speed, $\Omega$</td>
<td>500rpm-3krpm</td>
</tr>
<tr>
<td>Number of pads, $N_{pad}$</td>
<td>5</td>
</tr>
<tr>
<td>Configuration</td>
<td>LBP</td>
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<tr>
<td>Rotor diameter, $D$</td>
<td>500mm (19.7 inch)</td>
</tr>
<tr>
<td>Pad axial length, $L$</td>
<td>350mm (13.7 inch)</td>
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<tr>
<td>Pad arc angle, $\Theta_P$</td>
<td>56°</td>
</tr>
<tr>
<td>Pivot offset</td>
<td>60%</td>
</tr>
<tr>
<td>Preload, $\bar{r}_p$</td>
<td>0.23</td>
</tr>
<tr>
<td>Bearing clearance, $C_B$</td>
<td>300µm (11.81mil)</td>
</tr>
<tr>
<td>Pad inertia, $I_P$</td>
<td>0.438kg. m² (1496.7lb.in²)</td>
</tr>
<tr>
<td>Oil inlet temperature</td>
<td>~50°C (122 °F)</td>
</tr>
<tr>
<td>Lubricant type</td>
<td>ISO VG32</td>
</tr>
<tr>
<td>Oil supply viscosity, $\mu_0$</td>
<td>0.0201 Pa.s</td>
</tr>
</tbody>
</table>

Predicted film thickness correlates poorly with measurements

Rotor speed $\Omega = 3\text{krpm}$
Specific load $W/LD = 2.5\text{MPa}$
Film pressures

Rotor speed $\Omega = 3\text{krpm}
Specific load $W/LD = 2.5\text{MPa}$

Predicted film pressure < test data. ~20bar difference on loaded pads (#1 & #2).

Notes: flow in bearing is turbulent & test data shows much larger pad deformations (mechanical and thermal)

Conclusions

Selected examples for comparison do not show pad flexibility affects TPJB static load performance.

Considerable discrepancies exist between predictions and measurements for large size TPJBs.

FE pad model is too stiff leading to underestimation of the pad surface elastic deformation.

XLTPJB® models laminar flow bearings. Large size TPJBs operate in both the laminar & turbulent flow regions.
Enhanced Computational Model for Tilting Pad Journal Bearing with Pad Flexibility

May 2013

Year IV
• Enable model for operation with laminar/transition & turbulent flow conditions.
• Validate the constructed pad FE structural surface deformation model with comparisons to test data.
• Include pad flexibility on the prediction of frequency reduced TPJB dynamic force coefficients.
• Construct the FE model that relates pad elastic deformation to thermally induced stresses. \( K_T U_T \sim B(T) \)
• Compare predictions from code to test data.
## TRC Budget 2013-2014 Year IV

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Support for graduate student (20 h/week) x $ 2,050 x 12 months</td>
<td>$ 24,600</td>
</tr>
<tr>
<td>Fringe benefits (0.6%) and medical insurance ($185/month)</td>
<td>$ 2,368</td>
</tr>
<tr>
<td>Travel to (US) technical conference</td>
<td>$ 1,200</td>
</tr>
<tr>
<td>Tuition &amp; fees three semesters ($362/credit hour)</td>
<td>$ 8,686</td>
</tr>
<tr>
<td>Other (Mathcad® and portable data storage HD)</td>
<td>$ 220</td>
</tr>
<tr>
<td><strong>Total Cost:</strong></td>
<td><strong>$ 37,073</strong></td>
</tr>
</tbody>
</table>

XLTPJB® will continue to assist TRC members in modeling accurately TPJBs for specialized applications. The model and GUI reduce the burden on the unseasoned user by minimizing the specification of empirical parameters and guessing the correct boundary conditions for a proper analysis with thermal effects.
Questions (?)
Matrix reduction

For the whole domain

\[ \mathbf{K}^G \mathbf{q}^G = \mathbf{Q}^G \]

- **Apply boundary conditions**

\[
\begin{align*}
\mathbf{K}^G \mathbf{q}^G &= \begin{bmatrix} \mathbf{K}_{BC}^G & \mathbf{K}_S^G \\ \mathbf{K}_T^G & \mathbf{K}_A^G \end{bmatrix} \begin{bmatrix} \mathbf{q}_{BC}^G \\ \mathbf{q}_A^G \end{bmatrix} = \begin{bmatrix} \mathbf{K}_{BC}^G & \mathbf{K}_S^G \\ \mathbf{K}_S^G & \mathbf{K}_A^G \end{bmatrix} \begin{bmatrix} 0 \\ \mathbf{q}_A^G \end{bmatrix} = \mathbf{Q}^G = \begin{bmatrix} \mathbf{Q}_{BC}^G \\ \mathbf{Q}_A^G \end{bmatrix} \\
\mathbf{K}_A^G \mathbf{q}_A^G &= \mathbf{Q}_A^G
\end{align*}
\]

- **Loads on pad upper surface**

\[
\begin{align*}
\mathbf{K}_A^G \mathbf{q}_A^G &= \begin{bmatrix} \mathbf{K}_{AN}^G & \mathbf{K}_{AS}^G \\ \mathbf{K}_{AS}^G & \mathbf{K}_{AU}^G \end{bmatrix} \begin{bmatrix} \mathbf{q}_{AN}^G \\ \mathbf{q}_{AU}^G \end{bmatrix} = \mathbf{Q}_A^G = \begin{bmatrix} \mathbf{Q}_{AN}^G \\ 0 \end{bmatrix} \\
\text{Reduce matrix}
\end{align*}
\]

\[
(\mathbf{K}_{AN}^G - \mathbf{K}_{AS}^G \mathbf{K}_{AU}^{-1} \mathbf{K}_{AS}^G) \mathbf{q}_{AN}^G = \mathbf{Q}_{AN}^G
\]

Desbordes’s model [1]

\[ z=L/2 \]
\[ z=0 \]
\[ z=-L/2 \]

\[ \mathbf{K} \mathbf{u}_p = \mathbf{f} \]

\[ \mathbf{K}: \text{reduced stiffness matrix} \]

\[ \mathbf{u}_p: \text{displacement vector of pad upper surface} \]

\[ \mathbf{f}: \text{force vector converted from pressure} \]

Predictions for a four-pad TPJB

Max. 421 psi

Rotor speed $\Omega = 6.8$ krpm

Direct stiffness $K_{yy}$ is underestimated at heavy load