EXPERIMENTAL ANALYSIS OF THE DYNAMIC CHARACTERISTICS OF A FOIL THRUST BEARING

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   3. Foil structure and air film

IV. Summary and conclusion
The aerodynamic Foil Thrust Bearing

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner Diameter</td>
<td>5.08 cm</td>
</tr>
<tr>
<td>Outer Diameter</td>
<td>10.16 cm</td>
</tr>
<tr>
<td>Pad Angular Extent</td>
<td>45°</td>
</tr>
<tr>
<td>Unit Foil Area</td>
<td>7.6 cm²</td>
</tr>
<tr>
<td>Number of Pads</td>
<td>6</td>
</tr>
<tr>
<td>Total Pad Area</td>
<td>45.6 cm²</td>
</tr>
<tr>
<td>Top Foil Coating</td>
<td>PTFE</td>
</tr>
<tr>
<td>Top Foil Thickness</td>
<td>152 pm</td>
</tr>
<tr>
<td>Bump Foil Thickness</td>
<td>102 pm</td>
</tr>
<tr>
<td>Number of Bump Foil Strips</td>
<td>5</td>
</tr>
<tr>
<td>Bump Pitch (L.D. to O.D.)</td>
<td>5.36, 5.19, 4.65, 3.77, and 5.00 mm</td>
</tr>
</tbody>
</table>

The Foil Thrust Bearing used in the analysis [1]


Experimental analysis of start-up characteristics

Experimental analysis of dynamic characteristics
Test rig layout

- Shaker
- Piezo. force sensor
- Springs
- Shaft
- Aerodynamic bearing
- Load cell
- Foil thrust bearing
- Ball bearings

The test rig
Test rig schematic and explanations

The test rig schematic

- Aerostatic bearing
- Weight relief spring
- Static loading spring
- Load cell
- Thrust bearing holding bell
- Runner
- Bearings
- Upper shaft
- Shaft (non rotating)
- Lower shaft
- Runner (rotating)
Dynamic analysis of the test rig

The 2 DOF dynamic model of the upper shaft

\[
\begin{align*}
(m_1 \ddot{x}_1 + k_r x_1 + k_c (x_1 - x_2)) &= f \cos(\omega t) \\
(m_2 \ddot{x}_2 + k_c (x_2 - x_1)) &= 0
\end{align*}
\]

Identified stiffness of the static force transducer

\[
\begin{align*}
(m_1 A_1 + k_r X_1 + k_c (X_1 - X_2)) &= F \\
m_2 A_2 + k_c (X_2 - X_1) &= 0
\end{align*}
\]

\[
(1) \quad k_c = \text{Re}\left(\frac{m_2 A_2 \omega^2}{A_2 - A_1}\right)
\]

\[
(2) \quad k_c = \text{Re}\left(\frac{(F - m_1 A_1) \omega^2 + k_r A_1}{A_2 - A_1}\right)
\]
Dynamic model of the test rig

The test configuration for the foil structure only (Spinner replaced by a bolted plate)

The test configuration for the foil structure and the air film
The impedance of the foil structure

The e.o.m. of the 2d.o.f. model

\[
\begin{align*}
&m_1 \ddot{x}_1 + (k_r + k_c)x_1 - k_c x_2 = f \cos(\omega t) \\
&m_2 \ddot{x}_2 + k_c(x_2 - x_1) + k_b x_2 + c_b \dot{x}_2 = 0
\end{align*}
\]

\[
\begin{align*}
&m_1 A_1 + (k_r + k_c)X_1 - k_c X_2 = F \\
&m_2 A_2 + k_c (X_2 - X_1) + Z_b X_2 = 0
\end{align*}
\]

\[
\begin{align*}
&\begin{align*}
-m_1 \omega^2 X_1 + (k_r + k_c)X_1 - k_c X_2 = F \\
-m_1 \omega^2 X_2 + (X_2 + X_1)k_c + Z_b X_2 = 0
\end{align*}
\end{align*}
\]

\[
Z_b = m_2 \omega^2 - \frac{k_c (A_2 - A_1)}{\omega^2 A_2}
\]
Dynamic stiffness of the foil structure

- 2nd order inter. poly.
- 30 N static load
- Standard deviation

- 60 N static load
- Standard deviation

- 90 N static load
- Standard deviation

- 120 N static load
- Standard deviation

- 150 N static load
- Standard deviation

Static load
Equivalent viscous damping of the foil structure
Validation of the impedance of the foil structure

Remark: only the 2nd e.o.m. was used in \(Z_b\) identification

\[
\begin{align*}
-m_1 \omega^2 X_1 + (k_r + k_c)X_1 - k_c X_2 &= F \\
-m_2 \omega^2 X_2 + (X_2 + X_1)k_c + Z_b X_2 &= 0
\end{align*}
\]

From both e.o.m. of the 2d.o.f. model:

\[
\frac{A_1}{F} = \frac{\omega^2 (m_2 \omega^2 - k_c - Z_b)}{\delta}
\]

\[
\frac{A_2}{F} = \frac{-k_c \omega^2}{\delta}
\]

\[
\delta = (-m_1 \omega^2 + k_r + k_c)(-m_2 \omega^2 + k_c + Z_b) - k_c^2
\]

<table>
<thead>
<tr>
<th>Mode</th>
<th>Exp (Hz)</th>
<th>Th (Hz)</th>
<th>Rel. error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\omega_1), anti-resonance (m_1)</td>
<td>480</td>
<td>520</td>
<td>7.70</td>
</tr>
<tr>
<td>(\omega_1), anti-resonance (m_1)</td>
<td>710</td>
<td>685</td>
<td>-3.65</td>
</tr>
</tbody>
</table>
The loss factor $\eta = \frac{c}{\omega k}$ of the foil structure

Similar experimental values reported in the foil journal bearing literature:

The impedance of the foil thrust bearing with air film

The e.o.m. of the 3d.o.f. model:

\[
\begin{align*}
m_1 \ddot{x}_1 + k_r x_1 + k_c (x_1 - x_2) &= f \cos(\omega t) \\
m_2 \ddot{x}_2 + k_c (x_2 - x_1) + k_b (x_2 - x_3) + c_b (\dot{x}_2 - \dot{x}_3) &= 0 \\
m_3 \ddot{x}_3 + k_b (x_3 - x_2) + c_b (\dot{x}_3 - \dot{x}_2) + k_{rl} x_3 &= 0
\end{align*}
\]

\[
Z_b = -\frac{m_2 A_2}{X_{23}} + \frac{k_c (A_2 - A_1)}{\omega^2 X_{23}}
\]

The test configuration for the foil structure and the air film
Operating conditions:
- Static load: 30N, 60N and 90N
- Rotation speed: 0 and 35 krpm
Polynomial approximation of the equivalent viscous damping

- Structure only (without air film): Equivalent viscous damping increases with the static load.
- Conclusion on the impact of the different parameters cannot be drawn, due to the high standard deviations observed in the last slide.
Summary and conclusions

- The characteristics of the bump foil structure ($\Omega=0$) were measured for $W=30$ N and 150 N while the FTB with air film was tested at $\Omega=35$ krpm for $W=30$ N, 60 N and 90 N.

  - The measured dynamic stiffness showed low standard deviations.

  - The equivalent damping was measured with larger standard deviations especially for tests with rotation speed i.e. when the air film was present.

  - The stiffness of the bump foil structure and of the FTB (i.e. with air film) increases with the excitation frequency and with the static load.

  - The equivalent viscous damping decreases with the excitation frequency and increases with the static load.

  - The loss factor of the bump foil structure shows almost constant values for excitation frequencies larger than 300 Hz; the measured values are close to the ones given in the foil journal bearings literature.

  - For 35 krpm rotation speed, the stiffness and the equivalent viscous damping of the FTB are lower than for the bump foil structure alone thus showing the influence of the air film; this tendency increases with increasing load.
Thanks for your attention