Predictions vs. Test Results for Leakage and Force Coefficients of a Fully Partitioned Pocket Damper Seal and a Labyrinth Seal – Limitations of the Current Computational Model

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TRC Project 32514/15196PD Year I

Analyses of Pocket Damper Seals and Combined Labyrinth-Brush Seals
Trends in High Performance Turbomachinery
• Higher speeds & more compact units
• Extreme operating temperatures and pressures
• More efficient & reliable

Issues of Importance
• Reduce secondary flows (parasitic leakage)
• Reduce specific fuel consumption
• Increase power delivery
• Eliminate potential for rotordynamic instability
Labyrinth seals (LS) in a straight-through compressor

Labyrinth seals reduce leakage

Leakage model between sharp blade and rotor treated as an orifice.
Disadvantages of labyrinth seals

- Direct damping coefficient is usually small, even negative.
- Large cross coupled stiffness drives rotor-bearing system instability.

\[ C_{\text{eff}} = C - \frac{k}{\omega} \]

LSs provide limited effective damping and could even destabilize a whole rotor-bearing system.
Labyrinth Seals (LS)  
Pocket Damper Seals (PDS)

- PDS leaks more than LS.
- PDS provides ++ more effective damping and reduces rotor vibration amplitudes more effectively than a LS.

Vance, J. M., and Li, J., 1996
Neumann leakage model

\[ \dot{m}_i = \frac{(C_k C_f H)_i}{R_g T} \sqrt{P_{i-1}^2 - P_i^2} \]

Main flow equation

\[ \frac{1}{R_g T} \left[ \frac{\partial (P A)_i}{\partial t} + \frac{\partial (P A U)_i}{R_a \partial \Theta} \right] + \zeta_r (\dot{m}_{i+1} - \dot{m}_i) = 0 \]

Circumferential momentum equation

\[ \frac{1}{R_g T} \left[ \frac{\partial (P A U)_i}{\partial t} + \frac{\partial (P A U^2)_i}{R_a \partial \Theta} \right] + \zeta_r (\dot{m}_{i+1} U_i - \dot{m}_i U_{i-1}) = -\frac{A_i}{R_a} \frac{\partial P_i}{\partial \Theta} + \Delta \tau_{xi} \]

Li, J., San Andrés, L., and Vance, J., 1999
• PDSeal over predicts leakage (4-10%) compared to test results.

• PDSeal predicts direct damping coefficients in agreement with test data.

• Direct stiffness & damping coefficients and leakage are weak functions of rotor speed. Cross-stiffnesses are typically small.

Li, J., San Andrés, L., Vance, J., Ransom, D., and Aguilar, R.
Progress in 2013

**XLPDS© GUI created to interface with PDSEAL©**

GUI linked to XLTRC² suite to predict performance of pocket damper seals (sharp blades)

(a) Leakage
(b) Stiffness and damping coefficients

vs. pressure difference, rotor speed and excitation frequency.

Contact me for a demonstration on the use of the GUI.
Commercial PDS & FPDS

Commercial PDS and FPDS have **thick walls**

Original PDS had **sharp blades**

Pocket damper seal (PDS)

Fully partitioned pocket damper seal (FPDS)

**Ertas, B.H., Vance, J.M., 2007**
Examples – seals geometry


<table>
<thead>
<tr>
<th>14 bladed LS</th>
<th>8 bladed, 8 pocket FPDS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blades properties</td>
<td>All active</td>
</tr>
<tr>
<td>Cavity depth</td>
<td>4 mm</td>
</tr>
<tr>
<td>Cavity axial length</td>
<td>5 mm</td>
</tr>
<tr>
<td>Blade thickness (tip)</td>
<td>~ 0</td>
</tr>
<tr>
<td>Radial clearance</td>
<td>0.3 mm</td>
</tr>
<tr>
<td>Seal overall length</td>
<td>65 mm</td>
</tr>
<tr>
<td>Rotor diameter</td>
<td>170 mm</td>
</tr>
</tbody>
</table>
Examples: operating conditions

<p>| | | | | |</p>
<table>
<thead>
<tr>
<th></th>
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<th></th>
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</thead>
<tbody>
<tr>
<td>Inlet pressure</td>
<td>6.9 bar</td>
<td>(Absolute pressure)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Back pressure (Atmosphere)</td>
<td>1 bar</td>
<td>(Absolute pressure)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Excitation frequency</td>
<td>0 - 250 Hz</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>286 K (13°C)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rotor speed</td>
<td>7 krpm</td>
<td>15 krpm</td>
<td>7 krpm</td>
<td>15 krpm</td>
</tr>
<tr>
<td>Rotor surface velocity</td>
<td>62 m/s</td>
<td>133 m/s</td>
<td>62 m/s</td>
<td>133 m/s</td>
</tr>
<tr>
<td>Inlet preswirl velocity</td>
<td>0</td>
<td>0</td>
<td>60 m/s</td>
<td>60 m/s</td>
</tr>
<tr>
<td>Preswirl ratio</td>
<td>0</td>
<td>0</td>
<td>0.96</td>
<td>0.45</td>
</tr>
</tbody>
</table>

Inlet preswirl ratio = \(\text{inlet circumferential flow speed} / \text{rotor surface velocity}\)

<table>
<thead>
<tr>
<th>Gas</th>
<th>Air</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molecular weight</td>
<td>28.97</td>
</tr>
<tr>
<td>Gas compressibility factor</td>
<td>1</td>
</tr>
<tr>
<td>Specific heat ratio</td>
<td>1.4</td>
</tr>
<tr>
<td>Viscosity</td>
<td>18 (\mu\text{Pa\cdot s at 13°C})</td>
</tr>
</tbody>
</table>

Direct Stiffness

rotor speed 15 kpm
preswirl ratios=0 & 0.45

Fully partitioned pocket damper seal

Labyrinth seal

PDSeal© predicts well LS stiffness & misses stiffness for FPDS

Direct Damping

rotor speed 15 kpm
prespwirl ratios=0 & 0.45

Fully partitioned pocket damper seal

Labyrinth seal

PDSeal© predicts well LS damping & gives too little damping for FPDS

PDSeal© predicts well cross stiffness for both seals

Effective Damping

rotor speed 15 kpm
prespwirl ratios=0 & 0.45

Fully Partitioned pocket damper seal

Labyrinth seal

\[ C_{\text{eff}} = C - \frac{k}{\omega} \]

PDSear\textsuperscript{©} does a poor job in predicting the effective damping of a FPDS

Conclusions

Predicted effective damping for FPDS is distinct from test data.

PDS\textsuperscript{eal}\textsuperscript{\copyright} needs to be improved for better prediction for FPDS with thick walls.
Why the differences?

PDS with 4 pockets and 3 cavities

FPDS with 4 pockets and 3 cavities

Original model of PDS with sharp teeth in TAMU PDS code

PDS密封© does not consider axial thickness of the partition walls
Engineering Analyses for Pocket Damper Seals and Combined Labyrinth-Brush Seals

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May 2013
Update bulk-flow flow model for PDS and FPDS. Model will include real gas properties including supercritical CO2 and steam.

Perform more code calibrations: compare predictions to test data for leakage and force coefficients.

Begin extensions of the model to include two-component mixtures (liquid and gas).
Model PDS as a grooved seal

Continuity equation

\[
\frac{L}{\xi_r} \frac{\partial (dP)_i}{\partial t} + \frac{\partial (HPV)_i}{\partial z} + \frac{\partial (dPU)_i}{R_r \partial \theta} = 0
\]

Circumferential momentum equation

\[
-\frac{1}{R_r} \tau_{\theta,i} - \frac{\partial (dLP)_i}{R_r \partial \theta} = \frac{1}{Z \cdot R \cdot g \cdot T} \left [ \frac{1}{\xi_r} \frac{\partial (dLPU)_i}{\partial t} + \frac{\partial (dLPU^2)_i}{R_r \partial \theta} + \frac{\partial (HLPUV)_i}{\partial z} \right ]
\]

Axial momentum equation

\[
-\left [ \frac{1}{R_r} \tau_{z,i} + \frac{\partial (HLP)_i}{\partial z} \right ] = \frac{1}{Z \cdot R \cdot g \cdot T} \left [ \frac{1}{\xi_r} \frac{\partial (dLPV)_i}{\partial t} + \frac{\partial (HLPV^2)_i}{\partial z} + \frac{\partial (dLPUV)_i}{R_r \partial \theta} \right ]
\]

Replaces empirical leakage equation

Considers blade thickness

Kim, C. H., Childs, D. W., 1987
Year II

Support for graduate student (20 h/week) x $ 1,950 x 12 months $23,400
Fringe benefits (0.6%) and medical insurance ($185/month) $2,360
Travel to (US) technical conference $1,200
Tuition & fees three semesters ($362/credit hour x 24) $8,686
Others (Mathcad® and portable data storage) $220

Total Cost: $35,866

Year 2: Develop computational models for predictions of leakage, drag power and force coefficients of FPDS, and combined labyrinth-bush seals for gas and steam turbines
Thank you!

More information at:

http://rotorlab.tamu.edu