EXPERIMENTAL RESPONSE OF AN OPEN ENDS SFD AND A SEALED ENDS SFD

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Whirl motion from the journal squeezes the lubricant film and generates dynamic pressures that aid to damp the rotor vibrations.

Aid to attenuate rotor vibrations, suppress system instabilities, and provide mechanical isolation.

Too little damping may not be enough to reduce vibrations. Too much damping may lock damper & will degrade system performance.
SFD Test Rig – cut section

- Test Journal
- Bearing Cartridge
- Piston ring seal (location)
- Supply orifices (3)
- Main support rod (4)
- Flexural Rod (4, 8, 12)
- Journal Base
- Pedestal

Lubricant supply
Static loader
Shaker Y
Shaker X
Support rods
Base
SFD
Top view
Shaker X
Static loader
SFD
Shaker Y
Lubricant flow path

ISO VG 2 oil

Supply temperature, $T_{in}$: 23 °C (73 °F)
Lubricant viscosity @ $T_{in}$, $\mu$: 2.6 cP
Lubricant density, $\rho$: 800 kg/m$^3$
SFD Test Rig – cut section

Geometry (three feed holes 120° apart)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Journal Diameter, $D$</td>
<td>12.7 cm (5.0 in)</td>
</tr>
<tr>
<td>Land Length, $L$</td>
<td>2.54 cm (1.0 in)</td>
</tr>
<tr>
<td>Radial Land Clearance, $c$</td>
<td>254 μm (10.0 mil)</td>
</tr>
<tr>
<td>Feed orifice Diameter, $\phi$</td>
<td>2.54 mm (0.1 inch)</td>
</tr>
</tbody>
</table>

$L/D = 0.2$

Piston ring seals

Bearing Cartridge

Pressure sensor

Oil inlet

Top cover

Stinger attachment

Oil out

Journal

Supply orifices (3)
Funded TRC (2015-2016)

Justification

End seals amplify viscous damping while reducing the flowrate and reducing air ingestion.

Evaluate the forced performance of sealed ends SFD.

Tasks:

1. Test a short length ($L/D=0.2$), Sealed ends SFD with nominal clearance 254 μm (10 mil).
2. Conduct dynamic load tests for motions from centered and off-centered positions.
3. Evaluate SFD dynamic forced performance.
Reduce flow rate and side leakage \(\rightarrow\) raise film dynamic pressures and increase damping while also reduce air ingestion.

Piston ring design as an end seal is highly empirical.
Leakage vs lub. supply pressure

Piston ring end seals are effective in reducing the side leakage.
How do the whirl amplitude & static eccentricity affect sealed ends SFD force coefficients?
Sealed ends SFD: Damping coeff.

Direct

$C_{A-XX \ SFD}$

Cross-coupled

$C_{A-XY \ SFD}$

$C_{XX} \sim C_{YY}$

Damping grows with orbit amplitude $r$ while remain constant with static eccentricity ($e_s$)
Sealed ends SFD: Inertia coeff.

Direct

\[ M_{A-XX} \text{ SFD} \]

Cross-coupled

\[ M_{A-XY} \text{ SFD} \]

Direct added masses increase with static eccentricity, but decrease with orbit size \((r)\).
How much more damping if damper has end seals?
Sealed ends SFD gives 12X more damping and 11X more added mass than open ends SFD at small $r/c_A$. 

$C_{ave-SF}=0.56 \text{ LPM/bar}$ 

Open ends vs Sealed ends:

- Damping coefficients $e_y c_A = 0.0$
- Added mass coefficients $e_x c_A = 0.0$

Sealed ends:
- $L_{tot} = 3.68 \text{ cm}$
- $L_{off} = 2.97 \text{ cm}$
- $L = 2.54 \text{ cm}$

Open ends:
- $L_{tot} = 3.68 \text{ cm}$
- $L_{off} = 2.97 \text{ cm}$
- $L = 2.54 \text{ cm}$
How do the **seal conductance** affect sealed ends SFD force coefficients?

Two pairs of piston seals

\[ C_{\text{ave}-s_1} = 0.56 \frac{\text{LPM}}{\text{bar}} \]

Leaks less, more resistance

\[ C_{\text{ave}-s_2} = 0.89 \frac{\text{LPM}}{\text{bar}} \]

Leaks more, less resistance
Effect of flow conductance

Sealed ends 
\( (c_A = 254 \mu m) \)

At fixed \( P_{in-1} = 0.69 \text{ bar} \)

Leaks less, 
more resistance

Leaks more, 
less resistance

Two pairs of piston rings

Sealed ends SFD with smaller flow conductance gives \(~20\%\) larger damping at \( r/c_A = 0.6 \).

However, both results are within uncertainty range

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\[ \frac{C_{ave-S2}}{C_{ave-S1}} = 1.6 \]

\[ C_{ave-S1} = 0.56 \text{ LPM/bar} \]

\[ C_{ave-S2} = 0.89 \text{ LPM/bar} \]
How does the lubricant supply pressure affect force coefficients?
Effect of flow supply pressure

At fixed

\[ \frac{P_{in-2}}{P_{in-1}} = 4 \]

Larger supply pressure

SFD supplied with higher pressure gives \(~26\% - 50\%\) more damping. Differences increase as eccentricity \(e/c_A\) increases.
Pressure sensors in housing
Peak pressure increases with orbit amplitude. Damper with larger $P_{in-2}$ generates more dynamic pressure and reduces air ingestion. Oil vapor cavitation is at constant pressure.
Effect of feed pressure ($P_{in}$) to reduce bubbly mixture (air in lubricant).
Visual inspection

Top oil collector

\[ r/c_A = 0.45, \quad \omega = 80\text{Hz} \]
Comparison of force coefficients: test data vs predictions

Damper \((c=254 \text{ } \mu m \text{ (10 mil)})\)

L/D = 0.2, 25.4 mm (1 inch) land
Damping: predictions & tests

Sealed ends ($c_A = 254 \, \mu m$)

Good correlation: predictions vs. tests

$C_{XX}, C_{YY}$

$M_{XX}, M_{YY}$

$e_s/c_A = 0.0$

$P_{in-1} = 0.69 \, \text{bar}$

$C_{ave-s1} = 0.56 \, \text{LPM/bar}$

$C_{XX}, C_{YY}$

$M_{XX}, M_{YY}$

$e_s/c_A = 0.0$

$P_{in-1} = 0.69 \, \text{bar}$

$C_{ave-s1} = 0.56 \, \text{LPM/bar}$
Conclusion

(a) Damping coefficient grows with orbit amplitude \((r)\), but not with static eccentricity \((e_s)\). Added mass increases with amplitude \((r)\), but decreases with eccentricity \((e_s)\).

(b) Ends sealed SFD provide 12x more damping and 11x more added mass than open ends SFD.

(b) A higher lubricant inlet pressure produces larger damping and avoids air ingestion.

(c) Predictive model reproduces test data.
Engine qualification requires dampers to operate with oil delivery failure over a short period of time (~30 s), due to a malfunction or under a sudden 0 g maneuver load.

1. Machine journal with film land length and diameter \((L/D=0.2)\) and film clearance of 5 mil.
2. Characterize flow conductance of an open ends damper.
3. Perform transient - dynamic load measurements (fixed amplitude and frequency) while the supply of lubricant is suddenly cut.
4. Perform analysis to model test system, compare predictions vs. test data to validate damper flow model.

**Objective:** evaluate the performance of a SFD with film starvation due to sudden loss of oil supply.
### TRC Budget 2016-2017 Year VI

<table>
<thead>
<tr>
<th>Description</th>
<th>Cost</th>
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<tbody>
<tr>
<td>Support for graduate student (20 h/week) x $ 2,400 x 12 months</td>
<td>$ 28,800</td>
</tr>
<tr>
<td>Fringe benefits (2.7%) and medical insurance ($360/month)</td>
<td>$ 4,995</td>
</tr>
<tr>
<td>Supplies for test rig (Lubricant $ 700, Machining a new journal $ 1400)</td>
<td>$ 2,100</td>
</tr>
<tr>
<td>Tuition three semesters ($ 363 credit hour x 24 h/year)</td>
<td>$ 9,090</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>$ 44,985</strong></td>
</tr>
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</table>

*The TAMU SFD research program is the most renowned in the world. The proposed research is of interest for SFDs applied in gas turbines, hydrodynamic bearings in compressors, cutting tool and grinding machines.*
Acknowledgments

Turbomachinery Research Consortium & Pratt & Whitney Engines

Questions (?)
Predicted damping vs flow conductance:

- **C\(_{\text{seal-1}}\)**: 2.3E-04
- **C\(_{\text{seal-2}}\)**: 3.7E-04

**Notes:**
- Less leak
- More leak
- Flow rate
- Short length open ends SFD model (0.9 kN.s/m)