Squeeze Film Dampers
Operation, models and issues of interest

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Most common problems in rotordynamics

1. **Excessive steady state synchronous vibration levels:**
   - Improve balancing.
   - Modify rotor-bearing systems: tune system critical speeds out of RPM operating range.
   - Introduce damping to limit peak amplitudes at critical speeds that must be traversed.

2. **Subharmonic rotor instabilities**
   - Eliminate instability mechanism, i.e. change bearing design if oil whip is present.
   - Rise natural frequency of rotor system as much as possible.
   - Introduce damping to raise onset rotor speed above the operating speed range.
In a SFD, the journal **whirls but does not spin**. The lubricant film is squeezed due to rotor motions, and fluid film (damping) forces are generated as a function of the journal velocity.
SFD dynamic forced performance

depends on

a) **Geometry** \((L, D, c)\)
b) **Lubricant** (density, viscosity)
c) **Supply pressure and through flow conditions** (grooves)
d) **Sealing devices**
e) **Operating speed** (frequency) & journal kinematics

- **Flow regimes**: (laminar, superlaminar, turbulent)
- **Type of lubricant cavitation**: gaseous or vapor
  - air ingestion & entrapment
Brief history of SFDs

Parsons (1889)
Discloses first use of a SFD as a part of the first modern-day steam turbine.

Cooper (1963)
Rolls Royce engineer investigates experimentally the performance of rotating machinery with a SFD.

In 1970s, SFDs become essential components in aircraft engines and multistage high pressure centrifugal compressors.
Brief history of SFD (Turbomachinery Symposium)

Zeidan et al. (1996)

History SFDs since 1960’s and discuss major technical issues for their integration into turbomachinery, including oil cavitation vs. air ingestion and fluid inertia effects.

Kuzdal and Hustak (1996)

Tested various damper configurations (open and sealed ends) → optimized SFD reduces rotor synchronous motion amplitudes and raises stability threshold of a rotor bearing system.
Other relevant past work

• Della Pietra and Adilleta (2002): Comprehensive review of research conducted on SFDs over last 40 years.

Parameter identification in SFDs:

• Tiwari et al. (2004): Comprehensive review of parameter identification in fluid film bearings.


(2012-2021) San Andrés and students (sealed ends SFDs for aircraft)

Jet engines with rolling element bearings:
a) To reduce synchronous peak amplitudes,
b) Limit peak amplitudes at critical speeds,
c) To isolate structural components (lower transmissibility), and
d) To provide a margin of safety for blade loss.

Light hydrocarbon compressors with instability problems
a) To stabilize unit by introducing damping and reducing cross-coupled effect of seals, hydrodynamic bearings, etc.
b) To enhance limited damping available from tilting pad bearings.

Other benefits of SFDs on rotordynamic performance are:
• Tolerance to larger rotor motions
• Reduced balancing requirements
  * Simpler alignment
  * Less mount fatigue
Intershaft dampers whirl motions (precessional and spinning) result from the combined imbalance responses of both LP and HP rotors.

In multi-spool jet engines, intershaft dampers locate at the interfaces between rotating shafts.

Schematic view of an intershaft SFD
Types of end seals for SFDs

Reduce thru flow and increase damping. Most seal types cannot prevent air ingestion

Industry uses O-rings, while jet engines use piston rings.

**O-ring issues:**
- Low weight (replace squirrel cage),
- Special groove machining,
- Material compatibility

**Piston ring issues:**
- Cocking and locking
- Splits – leak too much

*Design is highly empirical*, except for end plate seals
SFD feed groove and exit grooves

Feed holes with small diameter (high flow) resistance or with check valves prevent back flow and distortions in dynamic film pressures.

Feed & discharge grooves

Interact with film flow, develop large dynamic film pressures, Induce inertia force coefficients even in small clearance (c) SFDs.

"Too shallow grooves": increase damping ($d_g/c < 10$)

"Too deep grooves": increase added mass ($d_g/c > 10$)
The amount of damping (needed) is the critical design consideration.

If damping is too large the SFD acts as a rigid constraint to the rotor-bearing system with large forces transmitted to the supporting structure.

If damping is too low, the damper is ineffective and likely to permit large amplitude vibratory motion at synchronous and sub harmonic frequencies.
What is the effect of **viscous damping** on the dynamic response of a mechanical system?
Simple spring-damper-mass system

**EOM:**

\[ M \ddot{X} + C \dot{X} + KX = F(t) \]

\[ F = Mu\omega^2 \]

Response amplitude \( |X/u| \)

Transmissibility (to ground)

Damping helps only when rotor traverses a critical speed (natural frequency \( f_n \)) but increases force transmissibility for operation above \( 1.44 f_n \)

System response defined by natural frequency \( (f_n) \) & damping ratio \( (\zeta) \)

\[ f_n = 2\pi \sqrt{\frac{K}{M}} ; \quad \zeta = \frac{C}{2\sqrt{KM}} \]
More complex K-C-M system: rotor on flexible supports

\[ F = M \omega^2 \]

Response amplitude \( |X/u| \)

Transmissibility (to ground)

Damping ratio \( \zeta \) increases

More complicated response. Damping helps only when traversing a critical speed (natural frequency=\( f_{n1} \) and \( f_{n2} \)) but increases force transmissibility.

Excessive damping LOCKS supports and increases system response.
**SFDs – the bottom line**

Too little damping may not be enough to reduce vibrations. **Too much damping** may lock damper & will degrade system performance.

**SFDs must be designed with consideration of the whole rotor-bearing system.**

Physical damping magnitude is not as important as the system damping ratio!

\[ \zeta = \frac{C}{C_{\text{crit}}} = \frac{C}{2\sqrt{K M}} \]
SFD models for forced response

Damping is needed for safe passage through critical speeds and to provide or increase system stability. Thus, models for SFD forced response are:

**Imbalance response analysis:**
SFD forces for circular centered whirl orbits.

**Rotordynamic eigenvalue & stability analysis:**
SFD force coefficients for dynamic journal motions about a static (equilibrium) position.

**Numerical nonlinear formulations** for transient response analysis of rotor-bearing response. Abused in academic studies; nowadays **too common** with fast PCs.
Journal bearing model: steady state

Pressure field is invariant with time and increases as film thickness decreases to a minimum.
SFD model: journal motions off-centered

SFD reaction force for small amplitude motions about a static off centered journal position

\[
\begin{align*}
\begin{bmatrix}
F_x \\
F_y
\end{bmatrix} &=
\begin{bmatrix}
C_{XX} & C_{XY} \\
C_{YX} & C_{YY}
\end{bmatrix}
\begin{bmatrix}
\ddot{x} \\
\ddot{y}
\end{bmatrix} +
\begin{bmatrix}
M_{XX} & M_{XY} \\
M_{XY} & M_{YY}
\end{bmatrix}
\begin{bmatrix}
\ddot{x} \\
\ddot{y}
\end{bmatrix}
\end{align*}
\]

\textbf{C:} damping, \textbf{M:} inertia force coefficients

\textbf{SFD force coefficients}

\textbf{NL functions of static journal eccentricity} \(e_s\)
SFD model: pure radial squeeze (plunging motion)

Pressure field changes with time and increases as film decreases. Note pressure reversals.

\[
0 ; \quad v_r = f(t) \\
\dot{a}_r = 0; \quad a_t=0
\]
Kinetics of whirl (circular) orbits

Journal center velocity with radial & tangential \((v_r, v_t)\) components, and also acceleration \((a_r, a_t)\)

For circular centered orbits, amplitude \(e\) is constant and whirl frequency = \(\omega\)

Circular centered orbit
\[
v_t = e \omega; \quad v_r = 0
\]
\[
a_r = -e \omega^2; \quad a_t = 0
\]

SFD reaction forces:
\[
F_r = - (C_{rt} v_t + M_{rr} a_r)
\]
\[
F_t = - (C_{tt} v_t + M_{tr} a_r)
\]

\(C\): damping
\(M\): inertia coefficients
**SFD model: circular centered orbits**

**SFDs DO NOT have a stiffness**

Misnomer: \( K_{rr} = \omega C_{rt} \)

**Circular centered orbit**

\[ v_t = e \omega ; \quad v_r = 0 \]
\[ a_r = -e \omega^2 ; \quad a_t = 0 \]

**SFD reaction forces:**

\[ F_r = -(C_{rt} v_t + M_{rr} a_r) \]
\[ F_t = -(C_{tt} v_t + M_{tr} a_r) \]

Pressure is invariant in rotating frame. \( P \) follows \(-dh/dt\) rather than \( h \) (film)
SFD model: small amplitudes centered orbit

Damping \( (C) \) & inertia \( (M) \) force coefficients by Reinhart & Lund (1975)

\[
C_{XX} = C_{YY} = C_{rr} = 12\pi \frac{\mu R^3 L}{c^3} \left[ 1 - \frac{\tanh\left(\frac{L}{D}\right)}{\left(\frac{L}{D}\right)} \right]
\]

\[
M_{XX} = M_{YY} = M_{rr} = \pi \frac{\rho R^3 L}{c} \left[ 1 - \frac{\tanh\left(\frac{L}{D}\right)}{\left(\frac{L}{D}\right)} \right]
\]

Damping \( \sim (R/c)^3 \), Inertia \( \sim R^3/c \)
SFD sealed vs open

Open ends

\[
C_{xx} = C_{yy} = C_{tt} = \frac{1}{2} \pi \frac{\mu DL^3}{c^3}
\]

\[
M_{xx} = M_{yy} = M_{rr} = \pi \frac{\rho D L^3}{24} \left( \frac{L^3}{c} \right)
\]

(fully) Sealed ends

\[
C_{xx} = C_{yy} = C_{tt} = \pi \frac{12}{8} \frac{\mu D^3 L}{c^3}
\]

\[
M_{xx} = M_{yy} = M_{rr} = \pi \frac{\rho D^3 L}{8c}
\]

\[
\frac{C_{tt} \text{ sealed}}{C_{tt} \text{ open}} = \frac{M_{rr} \text{ sealed}}{M_{rr} \text{ open}} = 3 \left( \frac{D}{L} \right)^2
\]

Increase in damping (and inertia) is large!
For \((L/D)=0.2=1/5\), increase is **25x3 fold**
SFD model: circular centered orbits

\[ F_t = -(C_{tt} v_t + M_{tr} a_r) \]

\[ F_r = -(C_{rt} v_t + M_{rr} a_r) \]

\[ v_t = e \omega ; \quad v_r = 0 \]

\[ a_r = -e \omega^2 ; \quad a_t = 0 \]

\[ \pi \text{ FILM MODEL} \]

Damping & inertia force coefficients

\[ C_{tt} = \frac{\pi \mu D}{4(1-\epsilon^2)^{3/2}} \left( \frac{L}{c} \right)^3 \]

\[ C_{rt} = \frac{\mu \epsilon D}{(1-\epsilon^2)^2} \left( \frac{L}{c} \right)^3 \]

\[ M_{rr} = \frac{\pi \rho D}{24} \left( \frac{L^3}{c} \right) \left[ 1 - 2(1-\epsilon^2)^{1/2} \right] \left\{ \frac{(1-\epsilon^2)^{1/2} - 1}{\epsilon^2 (1-\epsilon^2)^{1/2}} \right\} \]

\[ M_{tr} = -\frac{27}{140 \epsilon} \rho D \left( \frac{L^3}{c} \right) \left[ 2 + \frac{1}{\epsilon} \ln \left( \frac{1-\epsilon}{1+\epsilon} \right) \right] \]

Damping \sim (L/c)^3, Inertia \sim L^3/c
SFD model: circular centered orbits

\[ F_r = -(C_{rt} v_t + M_{rr} a_r) \]
\[ F_t = -(C_{tt} v_t + M_{tr} a_r) \]

Nonlinear force coefficients, Large damping, Large inertia for \( \text{Re}_s = \rho \omega^2 c/\mu > 10 \)
Identification of SFD force coefficients for two SFDs: open ends & sealed ends
SFD test rig

Static loader

Shaker assembly (Y direction)

Shaker assembly (X direction)

SFD test bearing
Test rig photograph

- shaker Y
- Static loader
- shaker X
- SFD
- base
- support rods

Static loader
X
Y
shaker X
shaker Y
Static loader
SFD
base
support rods
Static loader
X
Y
Lubricant flow path

Fluid properties ISO VG2

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply temperature ($T_{in}$)</td>
<td>25 °C</td>
</tr>
<tr>
<td>Viscosity</td>
<td>2.96 c-Poise</td>
</tr>
<tr>
<td>Density</td>
<td>785 kg/m³</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>Long journal (A)</th>
<th>Short journal (B)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Land length ($L$)</td>
<td>25.4 mm</td>
<td>12.7 mm</td>
</tr>
<tr>
<td>Land clearance ($c$)</td>
<td>0.14 mm</td>
<td>0.13 mm</td>
</tr>
</tbody>
</table>

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Journal diameter ($D$)</td>
<td>12.7 cm</td>
</tr>
<tr>
<td>Groove length ($L_G$)</td>
<td>12.7 mm</td>
</tr>
<tr>
<td>Groove depth ($d_G$)</td>
<td>9.52 mm</td>
</tr>
</tbody>
</table>
Multiple-year test program (2008-2018)

Objective:
Optimize SFD influence on rotor dynamics.

- **Damper A** ($c_A=140 \, \mu m (5.5 \, mil)$)
  - Central feed groove
  - Piston ring seals
  - $L/D=0.4, 2 \times 25.4 \, mm$ lands

- **Damper B** ($c_B=137 \, \mu m (5.4 \, mil)$)
  - Central feed groove
  - Piston ring seals
  - $L/D=0.2, 2 \times 12.7 \, mm$ lands

- **Damper C** ($c_C=130 \, \mu m (5.1 \, mil)$)
  - Piston ring seals
  - $L/D=0.2, 25.4 \, mm$ land

- **Damper D** ($c_D=254 \, \mu m (10.0 \, mil)$)

- **Damper E** ($c_E=122 \, \mu m (4.8 \, mil)$)
  - No end grooves

- **Damper F** ($c_F=267 \, \mu m (10.5 \, mil)$)
  - $L/D=0.2, 25.4 \, mm$ land
Identification of SFD force coefficients
Test procedure

Evaluate SFD force coefficients from whirl orbits: amplitude \((r)\) grows with offset or static eccentricity \((e_s)\) – 45° away.
(1) Apply loads $\rightarrow$ record BC motions

Shakers apply forces

$F^1 = \text{Re} \left( \begin{bmatrix} F_x^1 \\ iF_y^1 \end{bmatrix} e^{i\omega t} \right)$

$F^2 = \text{Re} \left( \begin{bmatrix} F_x^2 \\ -iF_y^2 \end{bmatrix} e^{i\omega t} \right)$

Record BC displacements and accelerations

$z^1 = \begin{bmatrix} x^1(t) \\ y^1(t) \end{bmatrix} = \begin{bmatrix} X^1 \\ Y^1 \end{bmatrix} e^{i\omega t}$

$z^2 = \begin{bmatrix} x^2(t) \\ y^2(t) \end{bmatrix} = \begin{bmatrix} X^2 \\ Y^2 \end{bmatrix} e^{i\omega t}$

$z = \begin{bmatrix} z^1 \\ z^2 \end{bmatrix}$

$z = \begin{bmatrix} X \\ Y \end{bmatrix} e^{i\omega t}$

Load $F(t)$, displacement $z(t)$ and acceleration $a(t)$ recorded at each frequency

**EOM: Frequency Domain**

$[K_L + i\omega C_L - \omega^2 M_L] \bar{z} = \bar{F} - M_{BC} \bar{a}$

$\rightarrow H_L z$

**Unknown Parameters:**

$K_L, C_L, M_L$
Identification of parameters

Step 2: Transform to frequency domain and curve fit $H_L$'s

\[ \text{Re} \left( \left[ \bar{F} - M_{BC} \bar{a} \right] z^{-1} \right) \rightarrow K_L - \omega^2 M_L \]

\[ \text{Im} \left( \left[ \bar{F} - M_{BC} \bar{a} \right] z^{-1} \right) \rightarrow C_L \omega \]

Complex dynamic stiffness

Physical model $\text{Re}(H_{xx}) \rightarrow K - \omega^2 M$ and $\text{Im}(H_{xx}) \rightarrow C \omega$ agree well with experimental data. Damping $C$ is constant over frequency range.

$\frac{r}{c_A} = 0.2$

SFD coefficients

$$(K, C, M)_{SFD} = (K, C, M)_L - (K, C, M)_S$$

SFD

Test system (lubricated)

Dry structure
Force coefficients for two open ends SFDs

1 inch land ($L/D=0.20$)
Clearance:
$c=$ small (5 mil) vs. large (10 mil)
SFD test bearing and film geometry

### Geometry of SFD

<table>
<thead>
<tr>
<th></th>
<th>A</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Film land length, (L) (mm)</strong></td>
<td>25.4</td>
<td></td>
</tr>
<tr>
<td><strong>Radial clearance, (c_A), (c_B) (mm)</strong></td>
<td>0.129</td>
<td>0.254</td>
</tr>
<tr>
<td><strong>End grooves: depth × width (mm)</strong></td>
<td>3.8 × 2.5</td>
<td></td>
</tr>
<tr>
<td><strong>Total wetted length, (L_{tot}) (mm)</strong></td>
<td>36.8</td>
<td></td>
</tr>
</tbody>
</table>
Normalization of force coefficients

Force coefficients normalized to magnitudes from classical formulas (prior slide):

\[
\frac{C}{C^*} \quad \text{and} \quad \frac{M}{M^*}
\]

Damper A
\[c_A = 0.129 \text{ mm} \]
\[C^*_A = 6.01 \text{ kN.s/m}, \quad M^*_A = 2.69 \text{ kg}\]

Damper B
\[c_B = 0.254 \text{ mm} \]
\[C^*_B = 0.95 \text{ kN.s/m}, \quad M^*_B = 1.37 \text{ kg}\]

\[\bar{C} \sim 1 \quad \text{and} \quad \bar{M} \sim 1\] denote one to one agreement with predictive formulas.
Findings: Damping coefficients increase with increasing orbit amplitude and static eccentricity. At $r/c_A \leq 0.2$, $\bar{C}_{A-XX} \sim 0.85$. 

*Damper A ($c_A = 129 \, \mu\text{m}$) – damping coeffs.*
Findings: Added mass coefficients increase with increasing static eccentricity; but decrease with increasing orbit amplitude. Theory under predicts inertia coefficient, even for small amplitude motions.
Compare **damping coefficients** of two dampers

\[
\hat{C} = \frac{C}{C_B(r/c_B = 0.15)}
\]

**Recall**

\[
C \sim \mu \left( \frac{1}{c} \right)^3
\]

\[
\left( \frac{c_B}{c_A} \right)^3 \left( \frac{\mu_A}{\mu_B} \right) = \left( \frac{0.25}{0.13} \right)^3 \left( \frac{2.5}{2.7} \right) = 6.9
\]

**Damping coefficients** for small film clearance \( (c_A) \) damper are \(~4\) times larger than the coefficients obtained with larger clearance \( (c_B) \) SFD.

**Damper A**
- \( c_A = 0.129 \) mm

**Damper B**
- \( c_B = 0.254 \) mm
**Compare inertia coeffs. of two dampers**

Recall

\[ M = \frac{M}{M_B r/c_B = 0.15} \]

\[ e_S/c = 0 \]

<table>
<thead>
<tr>
<th>Damper A</th>
<th>Damper B</th>
</tr>
</thead>
<tbody>
<tr>
<td>( c_A = 0.129 ) mm</td>
<td>( c_B = 0.254 ) mm</td>
</tr>
</tbody>
</table>

Added mass coefficients for the small film clearance \( c_A \) damper are \(~1.8\) times higher than the coefficients obtained with larger clearance \( c_B \) SFD.

\[ \left( \frac{c_B}{c_A} \right) = \left( \frac{0.25}{0.13} \right) = 1.96 \]
From circular orbit tests

(a) For both dampers, **direct damping** coefficients do not show + sensitivity to the size of the orbit radius \( (r) \).

(b) **Inertia** coefficients for the large clearance damper B are **insensitive** to orbit amplitude \( (r) \), while small clearance damper A shows added masses **decreasing** with orbit size \( (r) \).
SFD force coefficients

Comparison between short and long open ends dampers with a central groove

Long: 2 x 25.4 mm lands

Short: 2 x 12.7 mm lands
Generation of dynamic pressure in film and groove

Does a (deep) central groove isolate a damper into two independent halves?

Conventional knowledge: A groove has constant pressure
Generation of dynamic pressure in a groove

Measured:

Open ends with central feed groove

Sealed ends with central feed groove

Film land

Groove

Does a central groove isolate a damper into two independent halves?

No! grooves produce squeeze film pressures!
compare SFD damping

Ratio of coefficients ~ \((L/c^3)\)

Long and short SFDs (circular orbits)

Long (\(L=1\) inch)

Short (\(L=0.5\) inch)
compare SFD inertia

Ratio of coefficients ~ \((L/c)\)

Long and short SFDs (circular orbits)
Closure II: Long vs short SFDs

Open ends long damper shows ~ 7 times more damping than short length damper. Inertia coefficients are twice larger.

SFD force coefficients are more a function of static eccentricity (max. 2 mil) than amplitude of whirl & changing little with ellipticity of orbit.

For all damper configurations and most test conditions: cross-coupled damping and inertia force coefficients are small.
Experimental SFD force coefficients
Comparison open ends & sealed ends long (1") SFD
compare SFD damping

Open ends vs sealed ends (circular orbits)

SFD (1 inch land lengths)

Damping coefficients (lbf-s/in) vs Eccentricity $e_s$ (mil)

- Sealed ends $C_{YY}$ (B-B)
- Sealed ends $C_{XX}$ (B-B)
- Open ends $C_{YY}$
- Open ends $C_{XX}$

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

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

Circular orbits

B-B sealed SFD
Added mass coefficients (lb)

Eccentricity $e_s$ (mil)

SFD (1 inch land lengths)

$M_{SFD}$

Sealed ends $M_{YY}(B-B)$

Sealed ends $M_{XX}(B-B)$

Open ends $M_{YY}$

Open ends $M_{XX}$

Open ends vs sealed ends (circular orbits)
Closure III: Open vs Sealed SFDS

Sealed ends long damper has ~ 3 times more damping than open ends damper. Inertia coefficients are 1.5 times larger.

SFD force coefficients are more a function of static eccentricity (max. 50 micro-m) than of the amplitude of whirl. Coefficients change little with ellipticity of orbit (up to 5:1 ratio)

Proper installation of piston rings is crucial for adequate sealing.
Oil cavitation OR air ingestion in SFDs?
Cavitation in liquid bearings

- Generation of (+) hydrodynamic pressure
- Dissolved gases liberated (1-5% in volume)
- Fluid vapor liberated
- Gaseous Cavitation
- Fluid vapor pressure
- Vapor cavitation
- Absolute zero pressure
- Negative pressure (fluid in tension)
Pressure is uniform (constant) inside cavitation “bubble” – Flow reformation at trailing edge of bubble

But… air ingestion & entrainment persist under dynamic load conditions.

Classical cavitation models do not apply to air entrainment under dynamic loading.
SFD Operation Issue

Bubbly lubricant exits from top and bottom ends of damper

Air ingestion and entrapment
Onset of air ingestion

Sealed ends SFD

Oil foamy mixture evolves from lubricant exiting through piston ring slit.
After 10 years of continued work, what did we learn?
Conclusion (1):

(a) Damping $(C)$ and inertia $(M)$ coefficients are ~ isotropic, i.e., $C_{XX} \sim C_{YY}$ and $M_{XX} \sim M_{YY}$. Cross-coupled coefficients are small for most whirl type motions.

(b) Simple theory does a modest job in producing physically accurate results for test SFDs with feed groove.

(c) SFDs generate large added mass coefficients, more so for sealed ends configurations and with (deep) feed and discharge grooves.
Conclusion (2):
(d) A sealed SFD produces significantly $(3+X)$ more damping and $\sim$twice the added mass than an open ends SFD.

(e) The amplitude and shape of whirl motion have small effect on the SFD force coefficients.

(f) Air ingestion impairs the growth of film pressures for increasing orbit amplitudes and frequency $\rightarrow$ damping coefficients decrease.

The experimental results demonstrate SFDs are mostly linear mechanical elements.
Modern SFDs

Integral SFDs

A few pointers ...
Integral Squeeze Film Damper (ISFD)

No squirrel cage
EDM manufacturing process produces separate arcuate damper film lands with S-shape flexural springs.

Advantages
- low number of parts
- short axial span
- light weight
- higher tolerance precision.
2011 Identification of Force Coefficients in a 5-pad Tilting Pad Bearing with an Integral Squeeze Film Damper (Delgado et al. at GE)

GT2012-68564 Rotordynamic Characteristics of a Flexure Pivot Pad Bearing with an Active and Locked Integral Squeeze Film Damper (Agnew and Childs)

2017 Dynamic Characterization of an Integral Squeeze Film Bearing Support Damper for a Supercritical CO₂ Expander (Ertas et al. at GE)

ISFDs: the sum of experiments

ISFD produces significant damping & inertia coefficients.
Model predicts well damping but over-estimates inertia by 30%.
End seals amplify viscous damping!

Quantify the effect of various end seal gaps on the dynamic forced performance of an ISFD.

- 2019: Conduct dynamic load tests on a dedicated test rig to obtain ISFD force coefficients for ready comparison and validation of the model.
Test ISFD \(\rightarrow\) Load between pads

- **ISFD inner ring**
- **ISFD outer ring**
- **Bearing Cartridge (BC)**
- **End plate**
- **Shim**

### Dimensions and Specifications

- **Diameter at film land,** \(D_{ISFD}\) 157 mm (6.18 in)
- **Length,** \(L\) 76 mm (3 in)
- **Film clearance,** \(c\) 0.356 mm (14 mil)
- **Arc radius,** \(\alpha\) 73° - 4 pads
- **End seals gap,** \(b_1\) 0.28 mm, 0.43 mm, 0.53 mm, open ends

### Fluid Properties

- **ISO VG46 Viscosity,** \(\mu\) 31.2 cP (at 46 °C)
- **Density,** \(\rho\) 860 kg/m³
- **Supply flow rate,** \(Q\) 9.5 L/min (set pressure)
End seals ISFD $\rightarrow$ Shimmed End Plates

$\frac{l_1}{b_1^3} > \frac{l_2}{b_2^3}$

$l_1 = 7.75 \text{ mm}, \ l_2 = 6.35 \text{ mm}, \ b_2 = 0.85 \text{ mm} \gg b_1$

$D_{ISFD} = 157 \text{ mm}, \ D_{plate} = 172.5 \text{ mm}.$

ISFD film clearance $c = 0.356 \text{ mm} \ (14 \text{ mil})$

End seals gap = shim thickness $b_1$

<table>
<thead>
<tr>
<th>$b_1$ (mm)</th>
<th>$c/b_1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.53</td>
<td>1.49</td>
</tr>
<tr>
<td>0.43</td>
<td>1.21</td>
</tr>
<tr>
<td>0.28</td>
<td>0.79</td>
</tr>
</tbody>
</table>
Test Rig for Dynamic Load Experiments

Max Speed: 16 krpm
Max Static Load: 22 kN
Max Dynamic Load: 4.4 kN, 1 k Hz
**ISFD (lubricated) dynamic complex stiffness** $H_L$

$$(K, C, M)_{SFD} = (K, C, M)_L - (K, C, M)_S$$

**SFD Film**

**Test system** (Lubricated)

**Dry structure**

Centered $e=0$. Open ends and with end seals gap = 0.53 mm $\rightarrow$ 0.28 mm

$$\text{Re}(H_L) \rightarrow (K_L - \omega^2 M_L)$$

$$\text{Im}(H_L) \rightarrow (C_L \omega)$$

**Graphs:**
- **Re($H_L$) $\rightarrow$** $(K_L - \omega^2 M_L)$
  - 0.28 mm end seal
  - 0.53 mm end seal
  - 0.43 mm end seal
- **Im($H_L$) $\rightarrow$** $(C_L \omega)$
  - 0.28 mm end seal
  - 0.53 mm end seal
  - 0.43 mm end seal
  - 0.53 mm end seal
- **Frequency (Hz):**
  - 0 to 180
- **Re($H$) (MN/m):**
  - 0 to 100
- **Im($H$) (MN/m):**
  - 0 to 150

**Legend:**
- Structure
- Open ends
- 0.53 mm end seal
- 0.43 mm end seal
- 0.28 mm end seal
Stiffness vs. static eccentricity vs. gap in end seal

\[ K = \frac{1}{2} (K_{xx} + K_{yy}) \] [MN/m]

ISFD film stiffness
\( K \) is small.

S-structure stiffness
\( \sim 60 \) MN/m)

c=356 μm, \( e/c=0-0.7 \), \( \omega=9-166 \) Hz, \( Q \sim 9.5 \) L/min, \( P_s=1 \sim 2 \) barg
Damping vs. static eccentricity vs. gap in end seal

\((C_{XX}, C_{YY}) [kN.s/m]\)

- Damping increases 22 times from open ends \(\rightarrow\) ends seal with gap = 0.28mm

\[
c = 356 \mu m, \ e/c = 0\text{-}0.7, \ 9\text{-}166 \text{ Hz}, \ Q \sim 9.5 \text{ L/min}, \ P_s = 1 \sim 2 \text{ barg}
\]
Added mass vs. static eccentricity vs. gap in seal

\[(M_{XX}, M_{YY}) [\text{kg}]\]

- Open ends
- 0.53 mm end seal
- 0.43 mm end seal

Inertia coefficient (added mass) larger than bearing mass (19 kg)

\[c = 356 \, \mu\text{m}, \, e/c = 0 - 0.7, \, 9 - 166 \, \text{Hz}, \, Q \sim 9.5 \, \text{L/min}, \, P_s = 1 \sim 2 \, \text{barg}\]
(a) ISFD does not produce a film direct stiffness $K_{ISFD}$ except for the test condition with the tightest end seal.

(b) Damping $C_{ISFD}$ increases with static eccentricity (large static load) but not as pronounced as theory predicts.

(c) Added mass $M_{ISFD}$ increases as gap decreases but ISFD with tightest gap ($b_1 = 0.28 \text{ mm} < c$) produces a stiffening hardening (negative virtual mass).

(d) End seals with small gap amplify $C_{ISFD}$. Configuration with gap $b_1=0.28 \text{ mm}$ produces 22 more damping that the open ends ISFD.

(e) For static eccentricity ($e<0.4c$), model with fluid compressibility predicts well the ISFD experimental damping coefficients but not its added mass.
Acknowledgments

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TAMU Turbomachinery Research Consortium (TRC)

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

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References
Relevant past work


Parameter identification:

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<tr>
<th>Year</th>
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<th>Title</th>
<th>Journal/Conference</th>
<th>Reference</th>
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<tr>
<td>2018</td>
<td>San Andrés, L., Koo, B., and Jeung, S-H.</td>
<td>“Experimental Force Coefficients for Two Sealed Ends Squeeze Film Dampers (Piston Rings and O-rings): An Assessment of Their Similarities and Differences”</td>
<td>ASME GT2018-76224</td>
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