On the Effect of Journal Kinematics on The Force Coefficients of a Test Squeeze Film Damper Supplied With an Air in Oil Mixture

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Whirl motion from the journal squeezes the lubricant film and generates a dynamic pressure field.

SFDs 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.
Squeeze film dampers

Literature in the 2000s

Experimental & Physical Modeling


Funded by National Science Foundation and TAMU Turbomachinery Research Consortium, June 1998- May 2002
To explore novel SFD designs & benchmark SFD empirical data.
Develop & validate SFD force performance model.
Optimize SFD influence on rotor dynamics.

Multiple papers – several awards at ASME Gas Turbo-Expo and IFToMM Rotordynamics Conferences
Operation with a controlled gas volume content supplied to film.

Tangential force reduces,

\[ \text{GVF} = 0 \rightarrow 1 \]

Radial forces increases,

\[ \text{GVF} = 0 \rightarrow 0.85 \]
**SFD with bubbly flow**

**Diaz and San Andres (2001)**

Mixture feeds from one end, and exits to ambient ($D = 50.8$ mm, $L/D = 0.5$, $c = 0.29$ mm)

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**Distinct journal motions produce different damping coefficients.**
Motivation

• 2020’s application: subsea industry
  – High performance compressors and pumps operating under wet gas/bubbly liquid conditions.

• Effect of two component flow and journal kinematics on SFD forced performance
  – Dynamic force coefficients
  – Rotor low whirl frequency motions
Bubbly seals in wet gas compression

Experimental & Physical Modeling


Funded by TAMU Turbomachinery Research Consortium, 2014- 2018
SFD/Seal Test Rig for Air/Oil Mixture

San Andrés, L., and Lu, X., 2018
SFD/Seal test rig with air/oil mixture

Oil Inlet (ISO VG 10)

Air Inlet Valve

Sparger (mixing) element

Test SFD/seal section

Valve

Air

Oil

D = 26 mm

Ps/Pa=2.5, inlet GVF= 50%, 0 rpm

L = 46 mm
### SFD/Seal geometry and fluid properties

<table>
<thead>
<tr>
<th></th>
<th>SFD/Seal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter ($D$)</td>
<td>127 mm (5 in)</td>
</tr>
<tr>
<td>Length ($L$)</td>
<td>46 mm (1.8 in)</td>
</tr>
<tr>
<td>Clearance ($c$)</td>
<td>0.18 mm at 28 °C</td>
</tr>
<tr>
<td>Supply pressure ($P_s$)</td>
<td>0~4.5 bar (abs)</td>
</tr>
<tr>
<td>Oil ISO VG 10 density ($\rho_l$)</td>
<td>830 kg/m³</td>
</tr>
<tr>
<td>viscosity ($\mu_l$)</td>
<td>13.5 cP at 28 °C</td>
</tr>
<tr>
<td>Air density ($\rho_{ga}$)</td>
<td>1.2 kg/m³ at 1 bar</td>
</tr>
<tr>
<td>viscosity ($\mu_{ga}$)</td>
<td>0.02 cP at 20 °C, 1 bar (abs)</td>
</tr>
<tr>
<td>Shaft speed ($\Omega_{max}$)</td>
<td>0 – 6 krpm</td>
</tr>
<tr>
<td>Rotor surface speed $\frac{1}{2} D\Omega_{max}$</td>
<td>40 m/s</td>
</tr>
</tbody>
</table>
Air and oil circulation systems

GVF at inlet:

\[ \alpha_{in} = \frac{Q_g \left( \frac{P}{P_A} \right)}{Q_l + Q_g \left( \frac{P}{P_A} \right)} \]

\( \alpha \): Gas volume fraction

\( P_s \): pressure at inlet plane

\( P_a \): ambient pressure = 1 bar(a)

\( Q_g \): gas flow rate at \( P_s \)

\( Q_l \): liquid flow rate
Procedure for dynamic load excitations

Step 1: Apply loads and measure BC motions

Apply force with impact hammer or with shakers (single frequency)

\[ F^1 = \text{Re} \begin{bmatrix} F_x^1 \\ 0 \end{bmatrix} e^{i\omega t} \]

\[ F^2 = \text{Re} \begin{bmatrix} 0 \\ F_y^2 \end{bmatrix} e^{i\omega t} \]

Record BC displacement \( z \) and acceleration \( a \)

\[ z^1 = \begin{bmatrix} x_{(t)}^1 \\ y_{(t)}^1 \end{bmatrix} = \begin{bmatrix} X^1 \\ Y^1 \end{bmatrix} e^{i\omega t} \]

\[ a^1 \]

\[ z^2 = \begin{bmatrix} x_{(t)}^2 \\ y_{(t)}^2 \end{bmatrix} = \begin{bmatrix} X^2 \\ Y^2 \end{bmatrix} e^{i\omega t} \]

\[ a^2 \]

EOM: Frequency domain

\[ [K + i\omega C - \omega^2 M]z = \bar{F} - M_s \bar{a} \]

Find parameters:

\[ \rightarrow H = H_R + i H_I = (K - \omega^2 M) + i \omega C \]
Identify force coefficients – dry system

Step 2: Curve fit to find **dry** structure (BC) parameters

\[
[K_s - \omega^2 M_s + i\omega C_s] z = F
\]

\[\Rightarrow H_s = K_s - \omega^2 M_s + i\omega C_s \]

NO lubricant

**Re\( (H_s) \rightarrow K_s - \omega^2 M_s**

**Im\( (H_s) \rightarrow C_s \omega**

\[
\text{Re}(H_{xx}) \quad \text{Re}(H_{xx})
\]

\[
K_{xx} - \omega^2 M_{xx}, \ R^2 = 0.98
\]

\[
K_{yy} - \omega^2 M_{yy}, \ R^2 = 0.99
\]

\[
\omega C_{xx}, \ R^2 = 0.83
\]

\[
\omega C_{yy}, \ R^2 = 0.85
\]

\[K_s = 690 \text{ kN/m}, \ M_s = 7 \text{ kg}, \ C_s = 0.2 \text{ kN-s/m}\]
Lubricated system

Step 3: For lubricated system Curve fit $H_L$'s using KCM model

$P_s = 0.1 \text{ bar(g)}$

$GVF = 0$, liquid only

$\text{Re}(H_{SFD}) \rightarrow K_{SFD} - \omega^2 M_{SFD}$

$K_{SFD} \sim 0 \text{ at } \omega = 0 \text{ Hz}$

$\text{Im}(H_{SFD}) \rightarrow C_{SFD} \omega$

Test:
$C_{SFD} = 43 \text{ kNs/m}$, $M_{SFD} = 6.4 \text{ kg}$

Theory SFD
$C_{SFD} = \frac{1}{2} \pi \mu (L/c)^3$ $D = 45 \text{ kNs/m}$ $M_{SFD} = \pi \cdot \rho \cdot \left(\frac{D}{2c}\right) \cdot \left(\frac{L^3}{12}\right) = 7.5 \text{ kg}$
Direct dynamic stiffness

\[ \text{Re}(H_{SFD}) \rightarrow K_{SFD} - \omega^2 M_{SFD} \]

Symbols: test data
Line: Curve fit

Unidirectional load, motion amplitude = 0.08 \( c \)

\[ P_s = 0.1 \text{ bar(g)} \]

GVF = 0.2~0.95

\[ \text{Re}(H_{SFD}) \text{ reduces with } \omega \text{ for GVF} = 0; \text{ and hardens with } \omega \text{ for GVF} > 0. \]

Added or virtual mass “disappears” for operation with a mixture as \( \text{Re}(H_{SFD}) \) increases with frequency.
Dynamic quadrature stiffness

\[ \text{Im}(H_{SFD}) \rightarrow C_{SFD} \omega \]

Symbols: test data
Line: Curve fit

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Line: Curve fit

Unidirectional load, motion amplitude = 0.08 c

\[ P_s = 0.1 \text{ bar(g)} \]

GVF

= 0.2~0.95

Ima(\(H_{SFD}\)) proportional to excitation frequency (\(\omega\)) and reduces with GVF.
Linear curve fit produces viscous damping coefficient. 

GVF

= 0.2~0.95
SFD with bubbly mixture

Unidirectional – single frequency load vs impact load

Distinct journal kinematics produce different damping forces.

$P_s = 0.1$ barg

<table>
<thead>
<tr>
<th>GVF</th>
<th>Damping</th>
<th>Impact</th>
<th>single frequency</th>
</tr>
</thead>
<tbody>
<tr>
<td>All liquid</td>
<td>$0 \rightarrow 0.4$</td>
<td>$\uparrow$</td>
<td>$\downarrow$</td>
</tr>
<tr>
<td>All liquid</td>
<td>$0.4 \rightarrow 0.9$</td>
<td>$\downarrow$</td>
<td>$\downarrow$</td>
</tr>
<tr>
<td>All liquid</td>
<td>$0.9 \rightarrow 1$</td>
<td>$\downarrow$</td>
<td>$\downarrow$</td>
</tr>
</tbody>
</table>
Unexpected phenomenon

Observed (and quantified) low frequency SFD/seal cartridge motions exacerbated by GVF.
Applied load and seal motion

**Load**

- **Frequency (Hz)**
  - **inlet LVF 0%**
  - **inlet LVF 4%**

- **Time (s)**
  - **Load (N)**
  - **Displacement (um)**

**Displacement**

- **Frequency (Hz)**
  - **inlet LVF 0%**
  - **inlet LVF 4%**

- **Time (s)**

**FFT**

- All gas
- Liquid 4%

- Journal motion drifts with a low frequency component

X direction, frequency = 30 Hz
Low frequency self-excited vibrations

Small amplitude SSV at \( \frac{1}{2}X \) for GVF = 0 and 0.01 (~ Liquid)

Low frequency at ~12 Hz for GVF > 0.1. Amplitude \( \rightarrow \) increases
Low frequency self-excited vibrations

As GVF increases, low frequency motions increase in magnitude.

Liquid 10% → GVF = 0.6

Liquid 40% → GVF = 0.8

Liquid 10% → GVF = 0.7

Liquid 10% → GVF = 0.9

Self-Excited broad band motion

Ω = 58.3 Hz, 3,500 rpm
ω = 80 Hz, load excitation

Amplitude [µm]

Time [s]

Frequency [Hz]
Why the low frequency motions?

Sound speed in mixture:

\[ V_s = \frac{1}{\sqrt{\rho_m \left( \frac{\alpha}{\rho_g V_{sg}^2} + \frac{1-\alpha}{\rho_l V_{sl}^2} \right)}} \]

\( V_{sg} \): sound speed in air, 351 m/s at 34 °C and 1 bara.

\( V_{sl} \): sound speed in oil (1,432 m/s).

Acoustic resonance in a circular duct with one end open:

\[ f_n = \frac{nV_s}{8\pi \cdot (L + \zeta D)} \text{ (Hz)} \]

\( n = 1, 2, 3 \ldots \)

\( n = 1 \) fundamental frequency, \( \zeta = 0.4 \) correction for tube with one end open.

Mixture low sound speed produces resonance → tests demonstrate it!
Conclusion

1. SFD damper offers no static stiffness.

2. For operation with a pure liquid, SFD generates significant added mass.

3. For operation with an air/liquid mixture, the SFD direct stiffness hardens with frequency.

4. Type of journal motion affects greatly the performance of a SFD/seal. Forced response from impact load is different than that from single frequency loads.

5. When lubricated with a mixture, low frequency self-excited vibrations relate to the acoustic resonance from sound speed in the mixture.
Acknowledgments

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Questions (?)

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