Measurements Of Leakage and Force Coefficients in a Short Length Annular Seal Operating with a Gas in Oil Mixture

Luis San Andrés
Mast-Childs Chair Professor
Fellow STLE

Q. Liu & X. Lu
Graduate Research Assistants

Supported by TAMU Turbomachinery Research Consortium

2015 STLE Annual Meeting & Exhibition, May 17-21, 2015, Dallas, TX
Bubbly Mixture Annular Pressure Seals

Justification

Seals operate with either liquids or gases, but not both……

• As oil fields deplete compressors work off-design with liquid in gas mixtures, mostly inhomogeneous.

• Similarly, oil compression station pumps operate with gas in liquid mixtures.

• The flow condition affects compressor or pump overall efficiency and reliability.

• Little is known about seals operating under 2-phase conditions, except that the mixture affects seal leakage, power loss and rotordynamic force coefficients; perhaps inducing random vibrations that are transmitted to the whole rotor-bearing system.
### Subsea Application: Wet Gas Compression

<table>
<thead>
<tr>
<th>Author(s)</th>
<th>Title</th>
<th>Year</th>
<th>Conference/Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brenne, L., Bjørge, T., Gilarranz, J., Koch, J., and Miller, H.</td>
<td>“Performance Evaluation of A Centrifugal Compressor Operating under Wet Gas Conditions”</td>
<td>2005</td>
<td>Proc. 34th Turbomachinery Symposium, Houston, TX, pp. 111~120</td>
</tr>
</tbody>
</table>

**Announce / assess the effects of two phase flow in wet gas compression:** drop in efficiency and reliability, excessive vibration and rotordynamic stability.
Squeeze film dampers

Background literature

Experimental & Physical Modeling


Sponsored by National Science Foundation and TAMU Turbomachinery Research Consortium, June 1998- May 2002
Literature: Two Phase Flow in Annular Seals

Experimental – Seals (two phase)


Computational – Seals (two phase)


NEED EXPERIMENTAL validation.

NO description of water lubricated seal (L, D, c) or gas type.....

Tests at shaft speed 1.5-3.5 krpm and supply pressure=1.2 - 4.7 bar. Gas/liquid volume fraction \( \beta \gamma = 0, 0.25, 0.45, 0.70 \)

Stiffness, damping and mass coefficients decrease as gas volume fraction increases.
Motivation

- **Application**
  - Subsea compression and pumping
    (wet compressors must operate with LVF up to 5%)

- **Effect of two component flow on seal**
  - Leakage rate
  - Dynamic force coefficients
  - Stability

- **Status of current research:**
  - Bulk-flow model available
  - Lack of experimental validation
Overview of this work

- Design and construct test rig.
- Make oil-gas mixtures with liquid volume fraction (LVF) at inlet plane from 0.0~1.0.
- Measure flow rate thru seal for range of LVF & pressure supply/discharge = 1 to 4.
- Measure test system periodic forced response and perform parameter identification.

room temperature 20°C.

Stationary (non-rotating) journal.
Wet Seal Test Rig

- Top plate
- Support pipe
- Main frame
- Bearing cartridge
- Dynamic pressure sensor
- Rotor
- Thermocouple
- Accelerometer
- Load cell
- Eddy current sensor
- Air-Oil Mixture
- Shaker
- Stinger
- Rotor
- Bearing
- Seal clearance

127mm (5 in)
Flows & Mixture

Air Inlet

Oil Inlet (ISO VG 10)

Valve

Sparger (mixing) element

Test seal section

\[ \frac{P_s}{P_a} = 1.5, \text{ inlet LVF} = 2\% \]
Seal cartridge

<table>
<thead>
<tr>
<th>Seal</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter</td>
<td>127 mm (5in)</td>
</tr>
<tr>
<td>Length</td>
<td>46 mm (1in)</td>
</tr>
<tr>
<td>clearance</td>
<td>0.127 mm (5mil)</td>
</tr>
<tr>
<td>Supply pressure</td>
<td>1~3.5 bar</td>
</tr>
<tr>
<td><strong>Oil ISO VG 10</strong></td>
<td></td>
</tr>
<tr>
<td>density</td>
<td>846 kg/m³</td>
</tr>
<tr>
<td>viscosity</td>
<td>18 cP at 20°C</td>
</tr>
<tr>
<td><strong>Air</strong></td>
<td></td>
</tr>
<tr>
<td>density</td>
<td>1.2 kg/m³ at 1bar</td>
</tr>
<tr>
<td>viscosity</td>
<td>0.02 cP at 20°C, 1bar</td>
</tr>
</tbody>
</table>

$L/D = 0.36$

![Seal cartridge diagram](image-url)

Flow

Seal

Rotor & journal
Flow Rate Measurement

Flow rate of mixture vs. supply pressure
1.1 ~3.5bar (abs)
Room temperature ~20ºC
– Liquid volume fraction (at supply pressure) at inlet:

\[ \text{LVF} = \beta_{inlet} = \frac{Q_l}{Q_l + Q_g} \]

Mass fraction (\( \lambda \)) is constant as there is no mass transfer between oil and air.

Mass fraction (\( \lambda \)) is constant as there is no mass transfer between oil and air.
Mass Flow Rate in Wet Seal $P_s/P_a=3.5$

**Mass flow rate**

![Graph showing mass flow rate vs. seal inlet LVF](image)

**Liquid mass fraction**

![Graph showing liquid mass fraction vs. seal inlet LVF](image)

**Reynolds #**

![Graph showing Reynolds number vs. seal inlet LVF](image)

Supply $P_s=3.5$ bar(abs), ambient $P_a=1$ bar(abs), (non-rotating) journal.

$L/D=0.36$

c=0.127 mm

Liquid mass fraction ($\lambda$) is large even for small LVF because of large density ratio (oil/air)~200.

Flow ranges from turbulent (pure air) to laminar (pure oil).

Predicted flow ~ 10% lower than measured.
Flow visualization in **Wet Seal**

- Ambient pressure $P_a = 1$ bar (abs), inlet temperature $20^\circ$C.
- Non-rotating journal.
- Inlet LVF = 0.45

**Seal land length** = 46 mm

**PS** = 1.5 bar (abs), $\beta = 0.45$

**PS** = 3.0 bar (abs), $\beta = 0.45$

**L/D** = 0.36, $c = 0.127$ mm

Smaller bubbles with higher supply pressure.
Force Coefficients in Annular Seals

Seal reaction force is a function of the fluid properties, flow regime, operating conditions and geometry.

For small amplitudes of rotor lateral motion: forces are linearized with stiffness, damping and inertia force coefficients:

\[
\begin{align*}
\begin{pmatrix}
F_x \\
F_y
\end{pmatrix} &= - \begin{pmatrix}
K_{xx} & K_{xy} \\
K_{yx} & K_{yy}
\end{pmatrix} \begin{pmatrix}
x \\
y
\end{pmatrix} - \begin{pmatrix}
C_{xx} & C_{xy} \\
C_{yx} & C_{yy}
\end{pmatrix} \begin{pmatrix}
x' \\
y'
\end{pmatrix} - \begin{pmatrix}
M_{xx} & M_{xy} \\
M_{yx} & M_{yy}
\end{pmatrix} \begin{pmatrix}
x'' \\
y''
\end{pmatrix}
\end{align*}
\]
**Parameter Identification**

**Model system (2-DOF): structure + seal**

**EOM: Time Domain**

\[ (K_S + K_{SEAL}) \ddot{z} + (C_S + C_{SEAL}) \dot{z} + (M_{BC} + M_{SEAL}) \dot{z} = F \]

**EOM: Frequency Domain**

\[ \left[ K + i \omega C - \omega^2 M \right] \vec{z} = \vec{F} \]

**Measure:** Load \( F = F_0 \sin(\omega t) \) \( \rightarrow \) Displacement \( z \),

\[ H(\omega) \vec{z} = \vec{F} \quad \rightarrow \quad H(\omega) = K + i \omega C - \omega^2 M = \frac{\vec{F}}{\vec{z}^{-1}} \]

**Estimate Parameters:**

\[ K, C, M \]
Applied load and seal motion

Load

Displacement

FFT

Unexplained drift at low frequencies

X direction, frequency = 30 Hz
Parameter Identification

Instrumental Variable Filter Method (IVFM) (Fritzen, 1985, J.Vib, 108)

Complex stiffness

\[ \text{Re}(H) = K - \omega^2 M \]

\[ \text{Im}(H) = (?) C \omega \]

Seal coefficients

\[(K, C, M)_{\text{SEAL}} = (K, C, M) - (K, C, M_{\text{BC}})_{\text{S}}\]

\[P_s/P_a = 2, \text{ inlet LVF 2\%}\]
**Test System Dynamic Stiffness**

- **Supply pressure** $P_s$ / ambient pressure $P_a$
- **Inlet temperature** 20°C.
- **(non-rotating) journal.**

**Liquid volume fraction at inlet:**

- $\beta_{inlet} = 0.02$
- $\beta_{inlet} = 0.04$
- Uncertainty ±0.7MN/m

*Note: The graph shows real part of the stiffness $H_{yy}$ and $H_{xx}$ with fitting $K - M\omega^2$ that fits well the test data.*
Test System Quadrature Stiffness

liquid volume fraction at inlet ($\beta$)

$\beta_{inlet}=0.0$
$\lambda=0.0$ (all air)
$m_m=6\pm0.8 \text{ g/s}$

$\beta_{inlet}=0.02$
$\lambda=0.90$
$m_m=6\pm0.2 \text{ g/s}$

$\beta_{inlet}=0.04$
$\lambda=0.95$
$m_m=7\pm0.2 \text{ g/s}$

2.0 = Supply pressure $P_s$/ambient pressure $P_a$
inlet temperature 20°C.
(non-rotating) journal.

Ima($H$) shows damping is frequency dependent.
Very little damping with all gas.

Uncertainty: ±0.7MN/m
Wet Seal Direct Damping $C = \text{Im}(H)/\omega$

$2.0 = \text{Supply pressure} P_s/\text{ambient pressure} P_a$

Stationary (non-rotating) journal.

liquid volume fraction at inlet ($\beta$): 0% (all gas), 2%, 4%.

Damping is frequency dependent.

Small LVF causes damping to increase 20 fold w/r to air only.

$C_{xx}$ & $C_{yy}$ differ due to uneven clearance and inhomogenous flow.
### Seal mass and stiffness coefficients

<table>
<thead>
<tr>
<th>LVF at seal</th>
<th>Liquid mass fraction</th>
<th>$K_{XX\text{seal}}$ (MN/m)</th>
<th>$K_{YY\text{seal}}$ (MN/m)</th>
<th>$M_{XX\text{seal}}$ (kg)</th>
<th>$M_{YY\text{seal}}$ (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>inlet</td>
<td>0</td>
<td>1.4</td>
<td>1.0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2%</td>
<td>90%</td>
<td>1.5</td>
<td>1.3</td>
<td>0.3</td>
<td>0.7</td>
</tr>
<tr>
<td>4%</td>
<td>95%</td>
<td>2.5</td>
<td>1.3</td>
<td>1.2</td>
<td>0.9</td>
</tr>
</tbody>
</table>

With mixture, $C_{xx} > C_{yy}$
mixture is not evenly distributed in seal clearance.
Liquid volume is larger in $X$ direction than in $Y$ direction.
• Assembled new test rig and trouble shoot.
• **Generate mixtures and measure flow rates:**
  – LVF = 1% to 100%
  – supply pressure up to 50 psig (3.5 bar abs)
  – Rotor speed: stationary
• **Forced response**
  – Installed shakers and performed single frequency
dynamic load measurements
  – Identification of force coefficients
• **Visual inspection of bubbly mixture**
Conclusion

- Mass flow rate through seal dictated by liquid content (density ratio ~ 200).
- Damping coefficients are frequency dependent. For operating conditions with a small volume fraction of oil in gas (4%), damping can be up to twenty times larger than that obtained for the pure gas condition.
- The effect of a few droplets of liquid on affecting the test system forced response is overwhelming.
Future Work

• Extend the range of LVF at seal inlet in tests to quantify its effect on force coefficients.

• Operate test seal rig with journal rotation (6 krpm) and with a higher supply pressure (6 bar).

• Benchmark bulk-flow model predictions¹.

• Extend existing predictive model¹ considering inhomogeneous flow.

Acknowledgments

- Turbomachinery Research Consortium
- Turbomachinery Laboratory Staff and Students

Questions (?)

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