Measurement of leakage and estimation of force coefficients in a short length seal supplied with a liquid/gas mixture (stationary journal)

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Bubbly & Foamy 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


Announce / assess the effects of two phase flow in wet gas compression: drop in efficiency and reliability, excessive vibration and rotordynamic stability.
Literature: Two Phase Flow in Annular Seals

Experimental – Seals (two phase)


Computational – Seals (two phase)


Arghir, M., Zerarka, M., Pineau, G., 2009


NEED EXPERIMENTAL validation.
Experimental – Seals (two phase)


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
Past TRC work 2013-14 – Year I

• Prepared test rig for measurements and improved lubrication system.
• Made oil-gas mixtures with liquid volume fraction (LVF) at inlet plane: gas to liquid.
• Measured flow rates with pure oil, all air, and with mixture.
• Designed (manufactured) new test rig.
Completed work 2014-15 Year II

• Revamped vertical test rig and troubleshoot.
• Generated mixtures and measured flow rate.
  – LVF = 0.02% (air) to 100% (liquid)
  – Supply pressure to 50 psig (3.5 bar (abs))
  – Rotor speed: stationary
• Conducted dynamic load tests and estimated force coefficients.
  – Installed shakers and performed single frequency dynamic load measurements
  – Identification of force coefficients
• Visual inspection of bubbly mixture.
Wet Seal Test Rig

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

127mm (5in)
Seal cartridge & fluid properties

<table>
<thead>
<tr>
<th>Seal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter 127 mm (5 in)</td>
</tr>
<tr>
<td>Length 46 mm (1.8 in)</td>
</tr>
<tr>
<td>clearance 0.127 mm (5 mil)</td>
</tr>
</tbody>
</table>

Supply pressure 1.0~3.5 bar (abs)

**Oil ISO VG 10**
- density 830 kg/m³
- viscosity 18 cP at 20°C

**Air**
- density 1.2 kg/m³ at 1bar
- viscosity 0.02 cP at 20°C, 1bar

$L/D=0.36$
Flows & Mixture

- Oil Inlet (ISO VG 10)
- Air Inlet
- Valve
- Sparger (mixing) element
- Test seal section

$P_s/P_a = 1.5$, inlet LVF = 2%
Flow Rate Measurement

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

\[
\text{LVF} = \frac{Q_l}{Q_l + Q_g \frac{P_a}{P_s}}
\]

Mass fraction (\(\lambda\)) is constant as there is no mass transfer between oil and air.
Mass Flow Rate in \textit{Wet Seal} \quad P_{s}/P_{a}=3.5

\textbf{Liquid mass fraction (\(\lambda\))} is large even for small LVF because of large density ratio (oil/air)\(\sim\)200. Flow ranges from turbulent (pure air) to laminar (pure oil). Predicted flow \(\sim\) 6\% lower than measured.

Supply \(P_{s}=3.5\) bar(abs), ambient \(P_{a}=1\) bar(abs), (non-rotating) journal.

\(L/D=0.36\)
\(c=0.127\) mm
Flow visualization in Wet Seal

ambient pressure $P_a=1 \text{ bar(abs)}$, inlet temperature $20^\circ\text{C}$.
Non-rotating journal.

Inlet LVF=0.45

Smaller bubbles with higher supply pressure

$L/D=0.36$
$c=0.127 \text{ mm}$

$P_s=1.5 \text{ bar(abs)}$
$\beta=0.45$

$P_s=3.0 \text{ bar(abs)}$
$\beta=0.45$

Seal land length: 46mm
Seal reaction force is a function of the fluid properties, flow regime, operating conditions and geometry.

For small amplitudes of rotor motion: the force is linearized with stiffness, damping and inertia force coefficients:

\[
\begin{align*}
\{F_X\} &= -\begin{bmatrix} K_{XX} & K_{XY} \\ K_{YX} & K_{YY} \end{bmatrix} \{x\} - \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix} \{\dot{x}\} - \begin{bmatrix} M_{XX} & M_{XY} \\ M_{YX} & M_{YY} \end{bmatrix} \{\ddot{x}\}, \\
\{F_Y\} &= -\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \{y\} - \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \{\dot{y}\} - \begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix} \{\ddot{y}\}.
\end{align*}
\]
**Parameter Identification**

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

**EOM: Time Domain**

\[
(K_s + K_{\text{seal}})z + (C_s + C_{\text{seal}})\ddot{z} + (M_{\text{BC}} + M_{\text{seal}})\dddot{z} = F
\]

**EOM: Frequency Domain**

\[
\begin{bmatrix}
K + i\omega C - \omega^2 M
\end{bmatrix}
\begin{bmatrix}
\bar{z}
\end{bmatrix} = \bar{F}
\]

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

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

**Estimate Parameters:**

\( K, C, M \)

**Re \((H(\omega))\) → \( K - \omega^2 M \)

**Im \((H(\omega))\) → \( \omega C \) (??)
Applied load and seal motion

X direction, frequency = 30 Hz

Load

Displacement

FFT

Unexplained drift at low frequencies
Parameter Identification

Complex stiffness

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

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

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

\(P_s/P_a = 2\), inlet LVF 2%

Instrumental Variable Filter Method (IVFM) (Fritzen, 1985, J.Vib, 108)
Test System Dynamic Stiffness

2.0 = Supply pressure $P_s$/ambient pressure $P_a$

inlet temperature 20°C. (non-rotating) journal.

$K - \omega^2 M$ fits well the test data.

liquid volume fraction at inlet:

$\beta_{inlet} = 0.02$

$\beta_{inlet} = 0.04$

Uncertainty ±0.7MN/m
Test System Quadrature Stiffness

2.0 = Supply pressure $P_s$/ambient pressure $P_a$

inlet temperature 20°C. (non-rotating) journal.

liquid volume fraction at inlet ($\beta$)

$\beta_{inlet} = 0.0$
$\lambda = 0.0$ (all air)
$(m_m = 6 \pm 0.8 \text{ g/s})$

$\beta_{inlet} = 0.02$
$\lambda = 0.90$
$(m_m = 6 \pm 0.2 \text{ g/s})$

$\beta_{inlet} = 0.04$
$\lambda = 0.95$
$(m_m = 7 \pm 0.2 \text{ g/s})$

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

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

2.0 = 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 inhomogeneous flow
# Wet seal stiffness & mass coefficients

<table>
<thead>
<tr>
<th>LVF at seal inlet</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>0</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, $C_{XX} > C_{YY}$ due to uneven clearance and mixture not homogenous (one phase travels faster than the other).
Conclusion

• Mass flow rate through seal dictated by liquid content (density ratio $\rho_l/\rho_{ga} = 830/1.2 = 692$).

• System dynamic stiffness $K-\omega^2M$ fits well with test data.

• 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.
Proposed Work

- Redesign stingers and mount shakers rigidly to avoid low frequency drift.
- Install a bubble eliminator to remove air content from the bubbly lubricant.
- Improve DAQ process to perform ensemble averages over multiple periods of external excitation.
- Benchmark bulk-flow model predictions\(^1\).
- Extend existing predictive model\(^1\) to consider inhomogeneous flow.

## TRC Budget

### 2015-2016 Year III

<table>
<thead>
<tr>
<th>Category</th>
<th>Amount</th>
</tr>
</thead>
<tbody>
<tr>
<td>Support for GS (20 h/week) x $2,200 x 12 months</td>
<td>$26,400</td>
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<tr>
<td>Support for UGS (10 h/week) x 9 months</td>
<td>$2,250</td>
</tr>
<tr>
<td>Fringe benefits (2.7%) &amp; medical insurance ($150/month)</td>
<td>$2,574</td>
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<tr>
<td>Registration and travel to technical Conference</td>
<td>$1,250</td>
</tr>
<tr>
<td>Tuition three semesters ($363 credit hour x 24 ch/year)</td>
<td>$8,712</td>
</tr>
<tr>
<td>Supplies: Bubble eliminator $1,050+flow meter repair $790+ oil $600</td>
<td>$2,440</td>
</tr>
<tr>
<td>PC upgrade ($1,150) + Mathcad ($105) + storage ($100)</td>
<td>$1,355</td>
</tr>
<tr>
<td><strong>Total BUDGET</strong></td>
<td><strong>$44,981</strong></td>
</tr>
</tbody>
</table>
Acknowledgments

Thanks to the Turbomachinery Research Consortium for its support

Questions (?)