Gas lubricated mechanical face seal are ubiquitous in many high performance applications such as compressors and gas turbines. The literature contains various analyses of seals having orderly face patterns (radial taper, waves, spiral grooves, etc.). These are useful for design purposes and for performance predictions. However, seals returning from service (or from testing) inevitably contain wear tracks and warped faces that depart from the aforementioned orderly patterns. Questions then arise as to the heat generated at the interface, leakage rates, axial displacement and tilts, minimum film thickness, contact forces, etc. This work describes an analysis of seals that may inherit any (i.e., random) face pattern. A comprehensive computer code is developed, based upon the Newton-Raphson method, which solves for the equilibrium of the axial force and tilting moments that are generated by asperity contact and fluid film effects. A contact mechanics model is incorporated along with a finite volume method that solves the compressible Reynolds equation. Results are presented for a production seal that has sustained a testing cycle.
Modeling Challenges

Q. Are models useful, and useful for what?
A. Typically used for design, predicting trends, etc.

Q. How “Complete” or “Robust” are they?
A. Limited by assumptions (how valid are they?), and capabilities (math models & complexity, numerical implementation, and CPU time)

Q. Can models be used for postmortem analysis?
- Faces maybe flat upon installation – highly unlikely that they remain as such.
- Cracked faces/shafts (they happen, but are these modeled?).
- Worn faces (”wear models” are empirical; first-law & robust “wear models” are yet to be developed).

Q. How robust are existing models?
(I) First Generation (classify, “contacting,” “non-contacting,” etc.)
(II) Next Generation (no classification needed, including multi-effects)
Flexibly Mounted Rotor
Face Seal Test Rig

- Shaft
- Rotor
- Stator
- Proximity probe
- Sealing dam
- Carbon ring
- Rotor chamber
- Contacting seal
- Spindle
- Pressurized air
- Pressurized water
- Coning
- Spacer
- Bolt
- Stator holder

Part I
Part II
Part III
FMR Mechanical Face Seal Test Rig
(Photograph)
Prong I: Real-Time Diagnostics

Three indicators:

- Time domain – probe signals
- Frequency domain -- Power spectral density functions (FFT)
- Angular orbit plots – seal absolute and/or relative misalignment, $\gamma_x$ vs. $\gamma_y$

All calculations are performed and plotted in real-time (using a PC with a dSpace DAQ board).
Noncontacting Operation (in Real-Time)

Intermittent Contacting Operation (in Real-Time)
Prong II: Seal Control — Contact Reduction/Elimination

- Clearance control of a mechanical face seal is achieved using cascade dual PI controllers with anti-windup acting on the variance of probe signals.

- System identification: experimentally (phenomenologically) determined seal model - theoretical model is not required.

- Using eddy current proximity probes to directly measure seal clearance and tilts as opposed to indirect methods (such as using thermocouples that measure face temperature).

- The controlled seal can follow seal clearance set-point changes with minimum control effort, while not being affected by disturbances in shaft speed and/or sealed water pressure.
Set Point Clearance Control

### Table

<table>
<thead>
<tr>
<th>Time</th>
<th>Clearance</th>
<th>Rotation Speed</th>
<th>Sealed water Pressure</th>
</tr>
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<tr>
<td>0</td>
<td>4(*)</td>
<td>15(*)</td>
<td>207(*)</td>
</tr>
<tr>
<td>1</td>
<td>20(+5)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>3(-1)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>5(+2)</td>
<td></td>
<td>241(+37)</td>
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<tr>
<td>4</td>
<td>15(-5)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>15(-5)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>207(-37)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>4(-1)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Uncontrolled/Controlled Seal

Controlled Seal:
- Rotor better tracks stator misalignment
- Virtual elimination of higher harmonic oscillations
- Closer to circular orbits, i.e., noncontacting operation
Prong III: Crack Detection in Seal/Rotor Driving Shaft (Seal Absent)
Modeling

- Cracked Rotors
- Crack Modeling
- Crack Indicators
- Methods of Detection

Analytical Work - Part I
(Green and Casey, 2005)

Experimental Work - Part II
Supercritical 2X Component

Experimental Orbits at various crack depths
Robust Modeling Objectives

Robust modeling should address these issues:
- Dynamics (high speeds, large masses -> inertia effects)
  -- stability, transients response, steady-state:
    misalignments, secondary seals and anti-rotation pins (Green (1985, 2006))
  -- coupled rotordynamics? (systems approach)
- Asperity Contact
  -- mechanical (“dry”) friction in sliding
  -- mechanical load support and deformation (EP)
- Mechanical Deformations (Pressure)
- Thermal Deformations (viscous and dry friction, TEI)
- Wear
- Face patterns (lift-off seals, typically for compressible seals)
Objectives (cont.)

To develop a (numerical) procedure where the solution includes multi-coupled phenomena (e.g., the Reynolds and energy Eqs., the EOM, contact mechanics, wear models).

The solution must be simultaneous in all the degrees of freedom.
Flat/coned face

- Inward/Outward Flow

(a) Case I: Inward Flow, where $\beta > 0$ and $P_0 - P_1 > 0$

(b) Case II: Outward Flow, where $\beta < 0$ and $P_0 - P_1 < 0$
Out-of-Service faces*

- wear?
- thermal/mechanical warping?

* Green and Artiles (2006), manuscript in preparation

NASA Seals Workshop
November 14-15, 2006

Dr. Itzhak Green, GA Tech, Mechanical Engineering, Atlanta GA 30332
green@gatech.edu
Schematic of Noncontacting Mechanical Face Seal with Flexibly Mounted Stator

Face pattern (waves, grooves, wear)

secondary seal

spring

stator

housing

rotor

shaft

$\Omega_2$
Finite Element Discretization (FEM)

Multiplying RE by a weight factor $W^T$ and integration by parts gives the weak form:

$$\int_\Omega \left\{-\nabla W^T \left[ \Phi p\, \frac{\partial p}{\partial t} - 6\mu \frac{\partial u_p}{\partial t} \right] - W^T 12\mu \frac{\partial(p\mu)}{\partial t} \right\} \, d\Omega = 0$$

Discretize domain into small finite elements:

Cartesian coordinate discretization

Polar coordinate discretization

$$p(\xi, \eta) = \sum_{i=1}^{9} N_i(\xi, \eta) \, p_i$$

$$\frac{\partial p(\xi, \eta)}{\partial \xi} = \sum_{i=1}^{9} N_i(\xi, \eta) \frac{\partial p_i}{\partial \xi}$$

$$\frac{\partial p(\xi, \eta)}{\partial \eta} = \sum_{i=1}^{9} N_i(\xi, \eta) \frac{\partial p_i}{\partial \eta}$$
Finite Volume Discretization (FVM)

Apply Green’s theorem to RE - represents mass conservation over the domain

\[ \int_{\Gamma} \left[ \Phi p \frac{\partial h}{\partial t} + 6\mu \frac{\partial}{\partial t} (h) \right] \cdot \vec{n} \, d\Gamma = \int_{\Omega} \left\{ 12\mu p \frac{\partial h}{\partial t} + 12\mu h \frac{\partial p}{\partial t} \right\} \, d\Omega \]

Discretize the domain into small finite volumes:

Cartesian coordinate finite volume discretization

Polar coordinate finite volume discretization
Incompressible Flow

Issues:
- Flow factors (Patir & Chang)
- Cavitation (Elrod, JFO)
- Starvation (?) (can be an issue in low pressure seals)
Compressible Flow – Herringbone wavy seal (w/o and w/ face def.)
EP Contact Load Support
(Jackson & Green, 2005)

\[ 0 \leq \omega^* \leq \omega_t^* = 1.9 \]

\[ P_F^* = \left( \omega^* \right)^{3/2} \]

\[ \omega_t^* \leq \omega^* \]

\[ P_F^* = \exp \left( -\frac{1}{4} \left( \omega^* \right)^{5/12} \right) \left( \omega^* \right)^{3/2} + \frac{4H_G}{CS_y} \left[ 1 - \exp \left( -\frac{1}{25} \left( \omega^* \right)^{5/9} \right) \right] \omega^* \]

- Statistically this formulation differs from the FEM data for all five materials by an average error of 0.94% and a maximum of 3.5%.
- Found to be valid not only for steels, but also for cupper, aluminum, and other metallic materials (Quicksall, Jackson and Green, 2004).
Rough Surfaces -- Statistical Model

Greenwood and Williamson (1966) formulated the statistical model using Hertz contact.

- The integrals are evaluated using Gauss-Legendre quadrature.

Jackson & Green (2005)

\[
\phi = (2\pi)^{-1/2} \left( \frac{\sigma}{\sigma_s} \right) \exp \left[ -0.5 \left( \frac{z}{\sigma_s} \right)^2 \right]
\]

\[
A(d) = \eta A_n \int_0^\infty A(z - d) \phi(z) dz
\]

\[
P(d) = \eta A_n \int_0^\infty P(z - d) \phi(z) dz
\]

Plasticity Index

\[
\psi = \sqrt{\frac{\sigma_s}{\omega_c}}
\]
Subsystem Coupling

Kinetic equations (including time-dependent thermal effects):

\[
\begin{bmatrix}
\dot{Z} \\
\dot{Y}
\end{bmatrix}
= 
\begin{bmatrix}
\frac{(F_{a2} + F_{a2} - F_{a2})}{m} \\
\frac{M_m + M_p}{I + \psi \dot{Y}^2} \\
\left[ \frac{(M_m + M_p)}{I - 2\psi \dot{Y}^2} \right] \dot{Y}^2 \\
\psi
\end{bmatrix}
\]

Lubrication equations:

\[
[S] \{ \ddot{p} \} = \{ R \}
\]

or

\[
\{ \ddot{p} \} = \{ R \}
\]

Coupled set of first-order ODEs:

\[
FEM: \quad [A(t, \varphi)] \{ \dot{\varphi} \} = \{ R(t, \varphi) \}
\]

\[
FVM: \quad \{ \ddot{\varphi} \} = \{ R(t, \varphi) \}
\]

1) Systematic coupling of kinetic and lubrication equations

2) Simultaneous solution using numerical ODE solver
Spiral Groove – Load Support in Compressible Flow

Tilts are small, so treated as vector tilts:

$$\vec{\gamma}_s = \gamma_X \vec{e}_X + \gamma_Y \vec{e}_Y$$

3 degrees of freedom:

$$
\begin{bmatrix}
  k_{f,Z} & k_{f,\gamma_X} & k_{f,\gamma_Y} \\
  k_{m_X,Z} & k_{m_X,\gamma_X} & k_{m_X,\gamma_Y} \\
  k_{m_Y,Z} & k_{m_Y,\gamma_X} & k_{m_Y,\gamma_Y}
\end{bmatrix}
$$

According to linearized gas film properties, the axial mode is decoupled from the tilt modes:

$$
\begin{bmatrix}
k_{f,Z} & 0 & 0 \\
0 & k_{m_X,\gamma_X} & k_{m_X,\gamma_Y} \\
0 & k_{m_Y,\gamma_X} & k_{m_Y,\gamma_Y}
\end{bmatrix}
$$
Spiral Groove (example)
Axial Force Frequency Responses for Mechanical Face Seal

![Graph showing Axial Force Frequency Responses for Mechanical Face Seal](image)

- **Storage Modulus - Real Part**
- **Loss Modulus - Imaginary Part**

Parameters:
- $\Omega = 1256.6 \text{ rad/s}$
- $\Omega = 1675.5 \text{ rad/s}$
- $\Omega = 2094.4 \text{ rad/s}$
- $\Omega = 2513.3 \text{ rad/s}$
- $\Omega = 2932.2 \text{ rad/s}$
Direct Moment Frequency Responses for Mechanical Face Seal

\[ \omega \cdot \sigma / \Omega \]

\[ G_{mX}, G_{mY}, \gamma \]

\[ \Omega = 1256.6 \text{ rad/s} \]
\[ \Omega = 1675.5 \text{ rad/s} \]
\[ \Omega = 2094.4 \text{ rad/s} \]
\[ \Omega = 2513.3 \text{ rad/s} \]
\[ \Omega = 2932.2 \text{ rad/s} \]
Cross-Coupled Moment Frequency Responses for Mechanical Face Seal

![Graph showing cross-coupled moment frequency responses](image)

- **Storage Modulus - Real Part**
- **Loss Modulus - Imaginary Part**

Frequency Responses for Different Angles:
- \( \Omega = 1256.6 \text{ rad/s} \)
- \( \Omega = 1675.5 \text{ rad/s} \)
- \( \Omega = 2094.4 \text{ rad/s} \)
- \( \Omega = 2513.3 \text{ rad/s} \)
- \( \Omega = 2932.2 \text{ rad/s} \)
Equations of motion with pseudo springs representing the gas film stiffness
(Axial mode decoupled from the tilt modes):

\[
m^* \ddot{Z} = -k_{Zg} Z - k_{sZ} \dot{Z} - d_s \ddot{Z}
\]

\[
I_t^* \ddot{\gamma}_X = -k_{m_x} \left[ \gamma_X - \gamma_r \cos(\Omega t) \right] - k_s \gamma_X - d_s \dot{\gamma}_X - k_{m_y} \left[ \gamma_Y - \gamma_r \sin(\Omega t) \right] + M_{Xi}
\]

\[
I_t^* \ddot{\gamma}_Y = -k_{m_y} \left[ \gamma_X - \gamma_r \cos(\Omega t) \right] - k_{m_y} \left[ \gamma_Y - \gamma_r \sin(\Omega t) \right] - k_s \gamma_Y - d_s \dot{\gamma}_Y
\]

Static stator misalignment provides constant moment about X axis

\[
M_{Xi} = \gamma_m \cdot k_{sY}
\]
Comparison of Solutions for Transmissibility

Correspondence Principle:

\[ \left| \frac{\gamma_{rel}}{\gamma_r} \right|_{\max} = 0.135 \]

Direct Numerical Simulation:

\[ \left| \frac{\gamma_{rel}}{\gamma_r} \right|_{\max} = 0.144 \]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_o = 6.0 \text{ mm} )</td>
<td>( r_j = 5.16 \text{ mm} )</td>
</tr>
<tr>
<td>( C_0 = 6.0 \mu \text{ m} )</td>
<td>( \Delta \Omega = 2094.4 \text{ rad/s} )</td>
</tr>
<tr>
<td>( P_i = 1.0 \times 10^5 \text{ Pa} )</td>
<td>( P_o = 2.0 \times 10^5 \text{ Pa} )</td>
</tr>
<tr>
<td>( I_i = 1.8 \times 10^3 \text{ kg m}^2 )</td>
<td>( N_i = 12 )</td>
</tr>
<tr>
<td>( \beta = 0.5 )</td>
<td>( \delta = 12 \times 10^{-6} \text{ m} )</td>
</tr>
<tr>
<td>( d_{ij} = 300.0 \text{ N s/m} )</td>
<td>( k_{ij} = 900.0 \text{ N m/rad} )</td>
</tr>
<tr>
<td>( \gamma_i = 0.2 \text{ mrad} )</td>
<td>( \gamma_m = 0.5 \text{ mrad} )</td>
</tr>
</tbody>
</table>
Transient operation conditions

\[
f = 0 \quad t < 0, \; t > t_3 \\
f = V \frac{t}{t_1} \quad 0 \leq t \leq t_1 \\
f = V \quad t_1 \leq t \leq t_2 \\
f = V \left(1 - \frac{t - t_2}{t_3 - t_2}\right) \quad t_2 \leq t \leq t_3
\]

where \( V \) is a desired steady-state value

\[\{V\} = \{\psi, \alpha, p_i, \text{or } p_o\}\]
Fig. 4a: Balance ratio effects upon transient response ($\tau = 2$ s)

Fig. 4b: Balance ratio effects upon coning and flow ($\tau = 2$ s)
# Mechanical Seal Codes

<table>
<thead>
<tr>
<th></th>
<th>INCOMP</th>
<th>COMP</th>
<th>SEPARATE</th>
<th>MIXED3D</th>
<th>TAU</th>
<th>TAU-G</th>
<th>Comments</th>
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<td>yes</td>
<td>yes</td>
<td>no (1a)</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>(1a) can predict analytically separation speed, (1n) separation speed is obtained from numerical simulation</td>
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<tr>
<td>Degrees of Freedom</td>
<td>3 (non-axisymmetric)</td>
<td>3 (non-axisymmetric)</td>
<td>2 (axisymmetric)</td>
<td>3 (non-axisymmetric)</td>
<td>3 (non-axisymmetric)</td>
<td>3 (non-axisymmetric)</td>
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<td>Incompressible</td>
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<td>yes</td>
<td>no</td>
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<td>no</td>
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<tr>
<td>Compressible</td>
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<td>yes</td>
<td>yes</td>
<td>no</td>
<td>yes</td>
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<tr>
<td>Noncontacting</td>
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<td>yes</td>
<td>yes(2)</td>
<td>yes(2)</td>
<td>yes(2)</td>
<td>yes(2)</td>
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<tr>
<td>Contacting</td>
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<td>yes(3)</td>
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<td>Coning</td>
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<td>yes(4)</td>
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<td>Wavy</td>
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<td>no</td>
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<td>no</td>
<td>yes(5)</td>
<td>(3) linear coning; (4) cubic coning</td>
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<td>Spiral grooves</td>
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<td>no</td>
<td>no</td>
<td>yes(6)</td>
<td>no</td>
<td>yes(6)</td>
<td>(5) periodic; or arbitrary (read from file)</td>
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<td>no</td>
<td>no</td>
<td>yes(7)</td>
<td>no</td>
<td>yes(7)</td>
<td>(6) includes a sector solution (as an option)</td>
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<td>no</td>
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<td>yes(7)</td>
<td>(7) using time-dependent ad hoc differential equation (allows complete transient analysis)</td>
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<td>yes(1n)</td>
<td>yes(1n)</td>
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<tr>
<td>Wear Model</td>
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<td>no</td>
<td>no</td>
<td>no</td>
<td>yes(8)</td>
<td>yes</td>
<td>(8) Under development</td>
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</table>