ROBUSTNESS OF MODELING OF OUT-OF-SERVICE GAS MECHANICAL FACE SEAL

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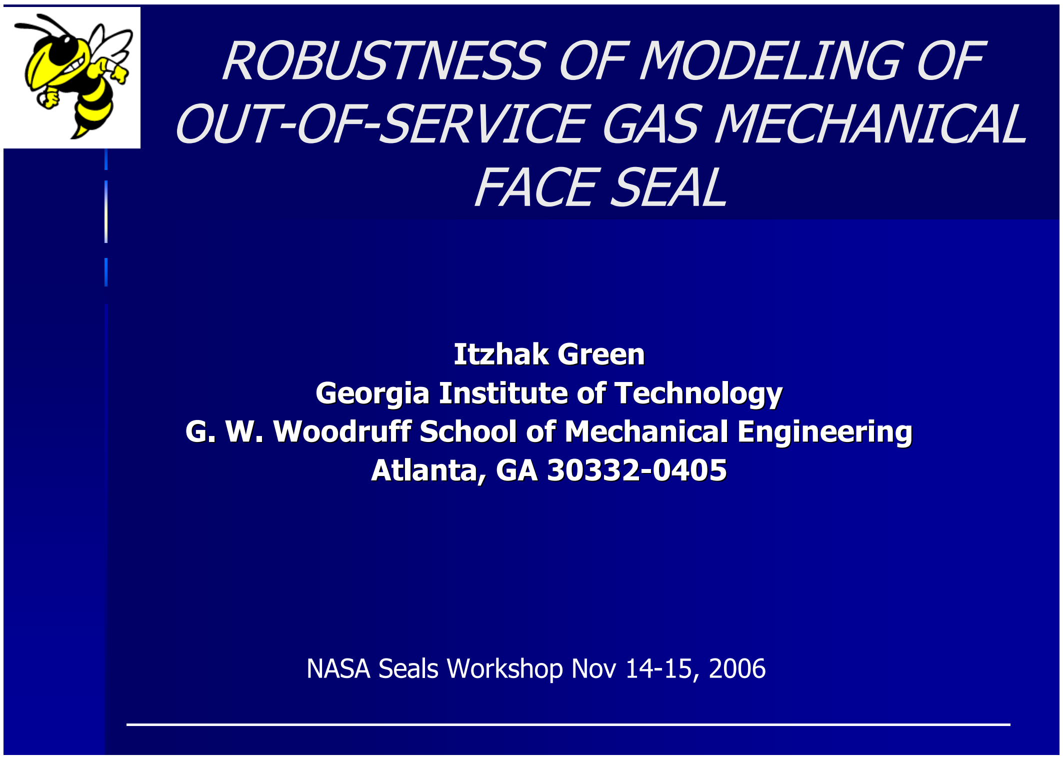

Gas lubricated mechanical face seal are ubiquitous in many high performance applications such as compressors and gas turbines. The literature contains various analyses of seals having orderly face patterns (radial taper, waves, spiral grooves, etc.). These are useful for design purposes and for performance 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)

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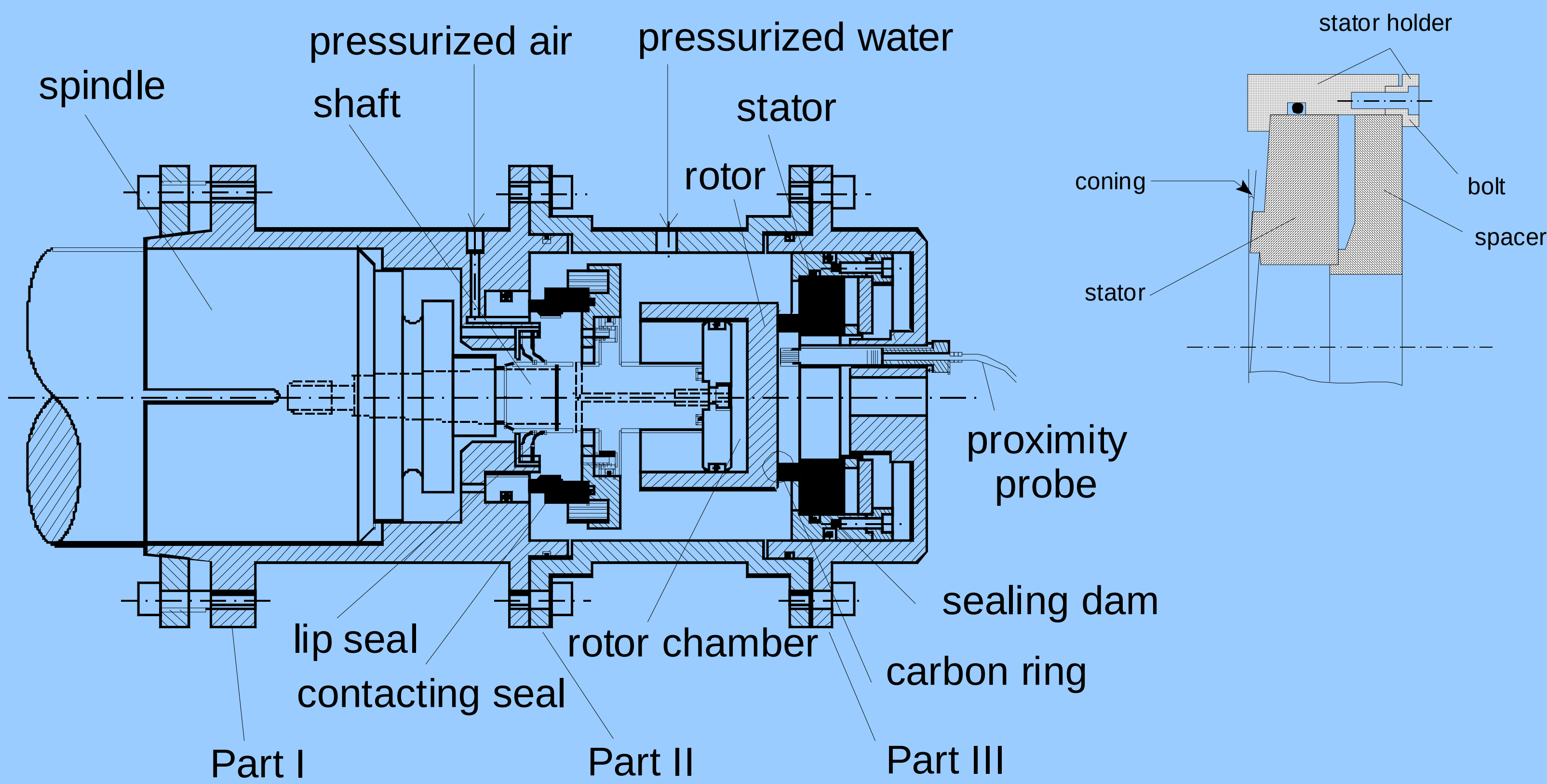
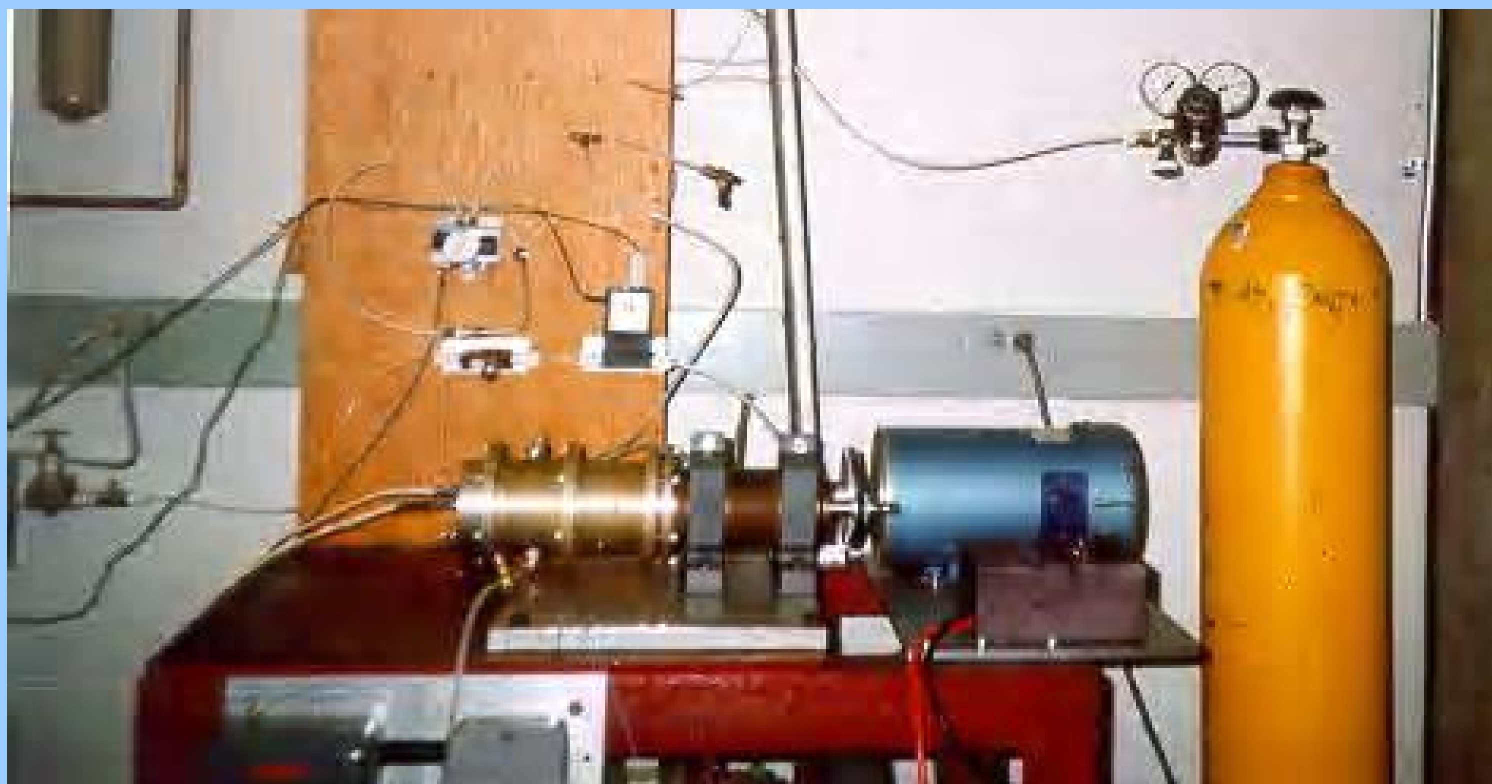
Flexibly Mounted Rotor Face Seal Test Rig



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, γ_x vs. γ_y.

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

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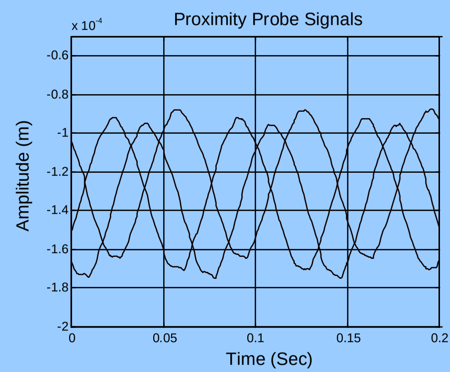
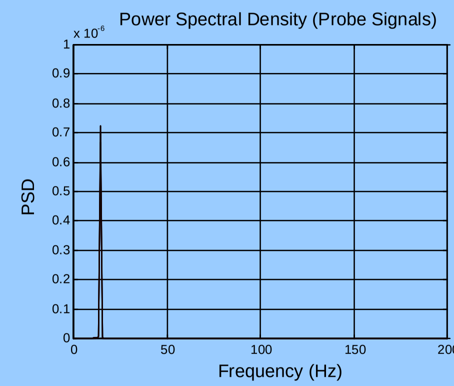
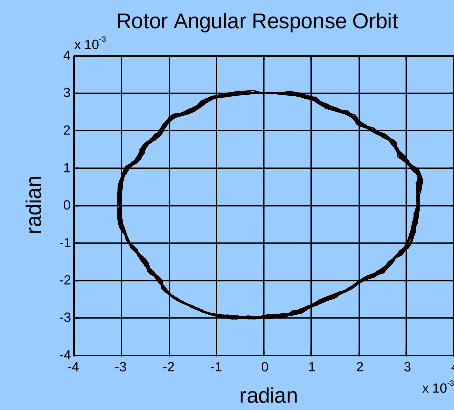
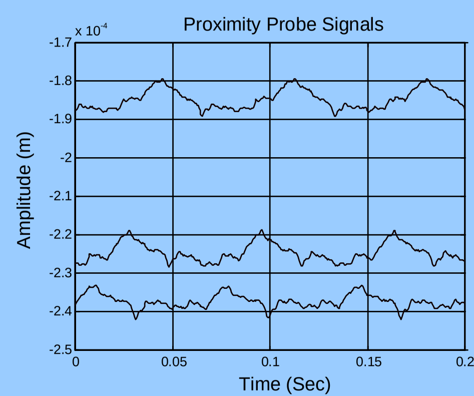
Noncontacting Operation (in Real-Time)

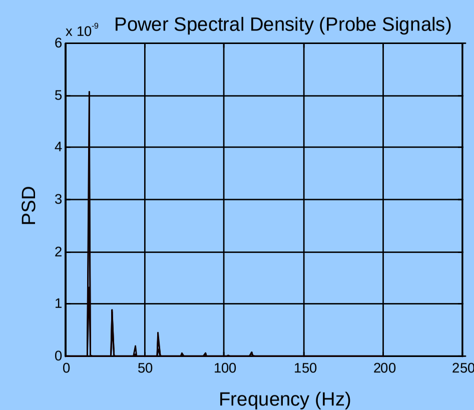


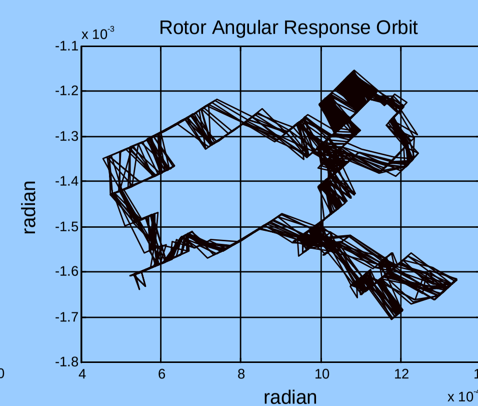




Intermittent Contacting Operation (in Real-Time)







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.

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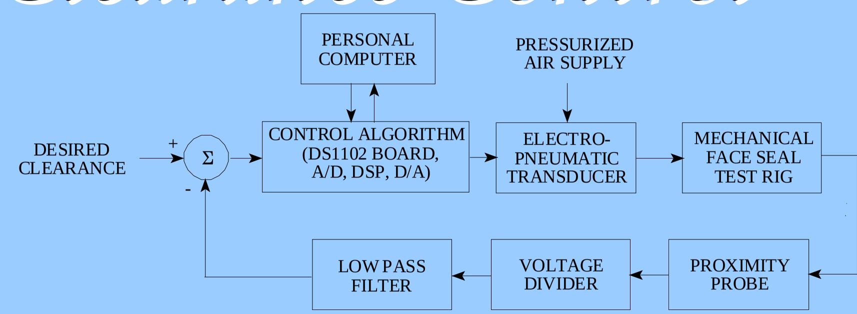
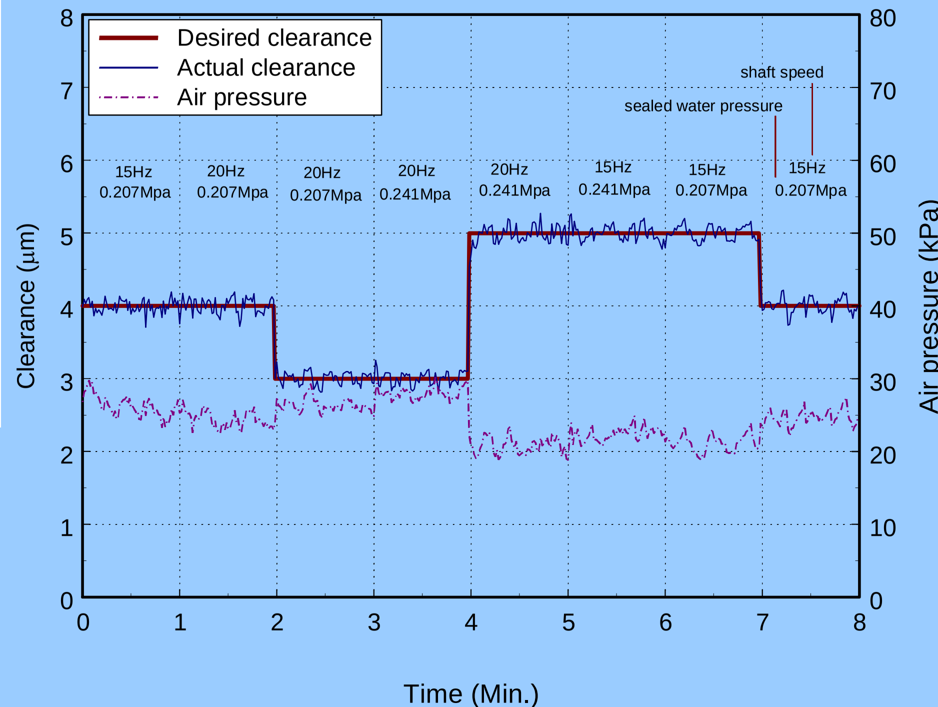
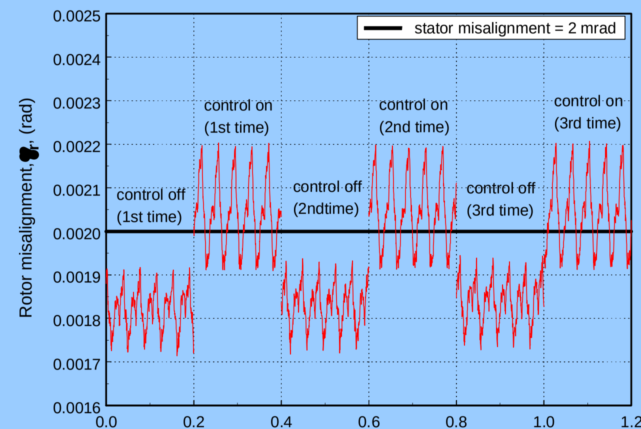
Set Point Clearance Control

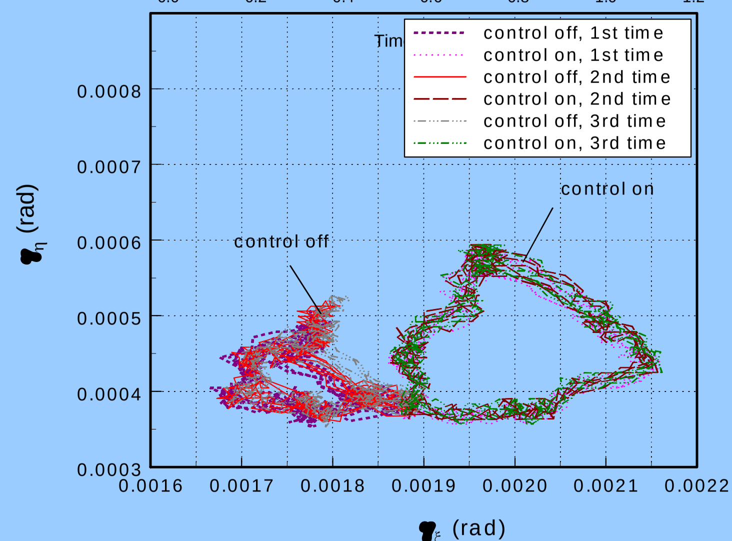
Time Min	Clearance µm	Rotation Speed Hz	Sealed water Pressure KPa
0	4(*)	15(*)	207(*)
1		20(+5)	
2	3(-1)		
3			241(+37)
4	5(+2)		
5		15(-5)	
6			207(-37)
7	4(-1)		





Uncontrolled/Controlled Seal

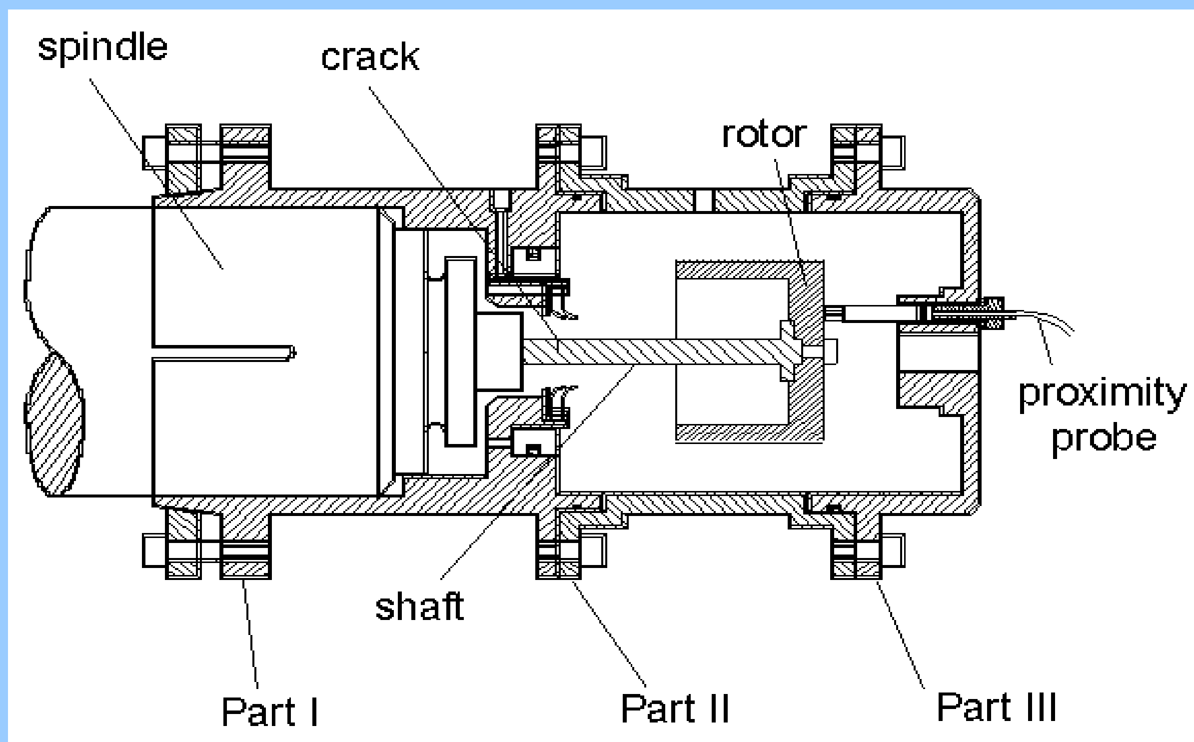




Controlled Seal:

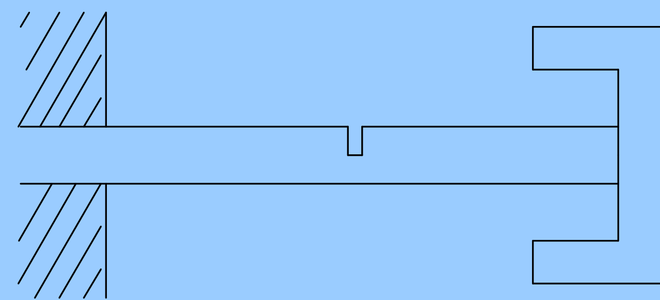
- Rotor better tracks stator misalignment
- Virtual elimination of higher harmonic oscillations
- Closer to circular orbits, i.e., noncontacting operation

Prong III: Crack Detection in Seal/Rotor Driving Shaft (Seal Absent)



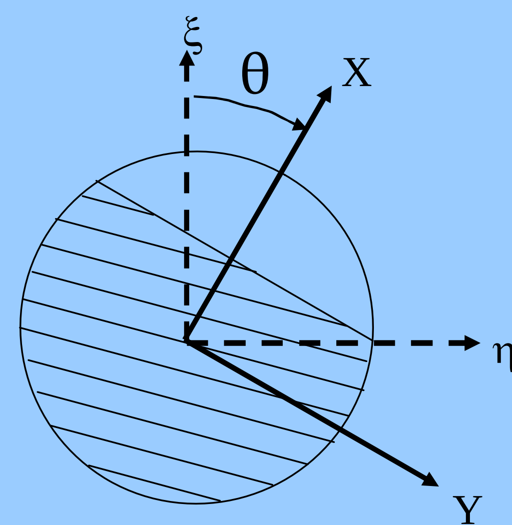
Modeling

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

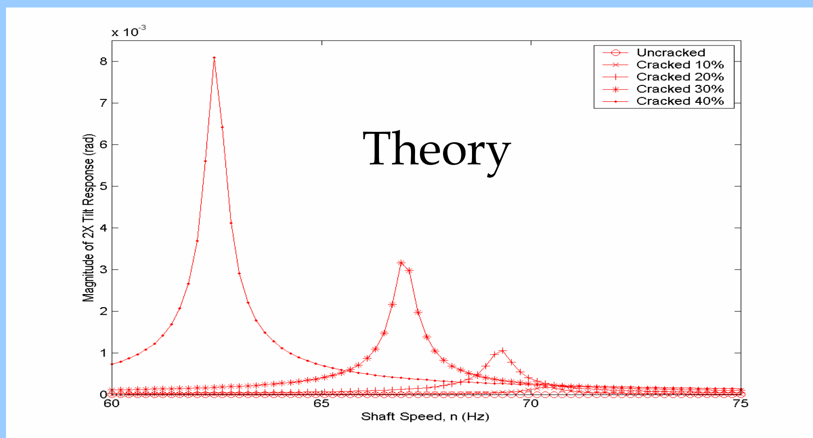


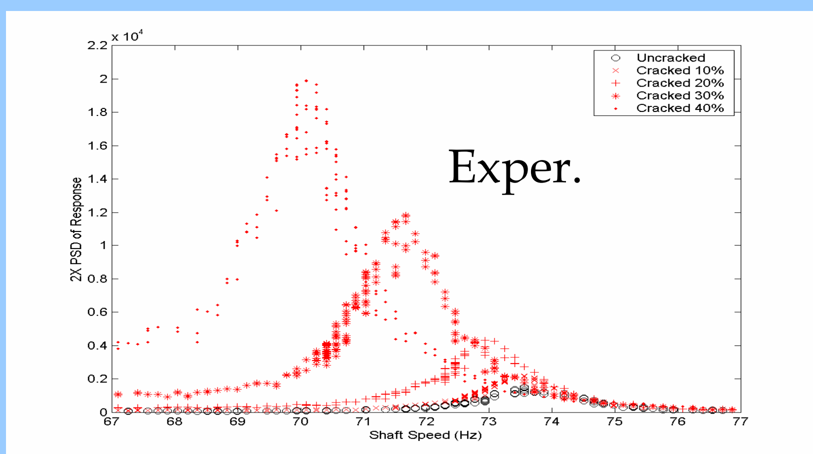
- Analytical Work - Part I (Green and Casey, 2005)

- Experimental Work-Part II

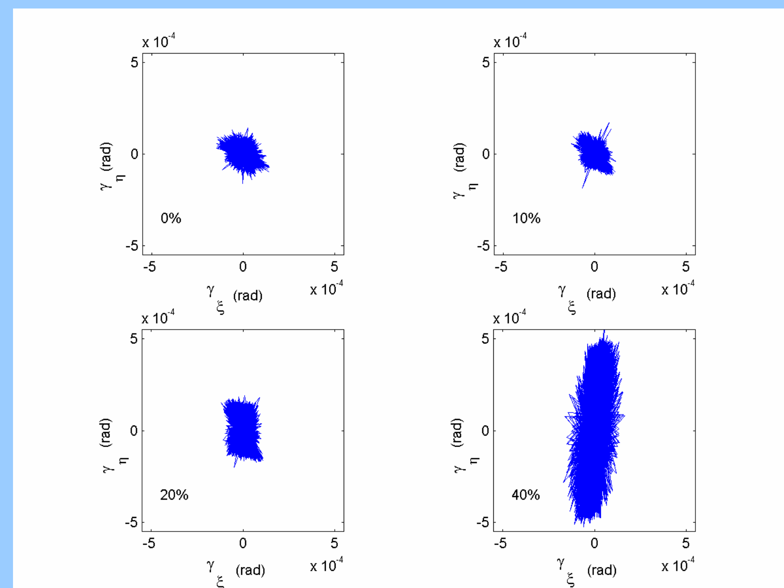


Supercritical 2X Component





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)

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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.

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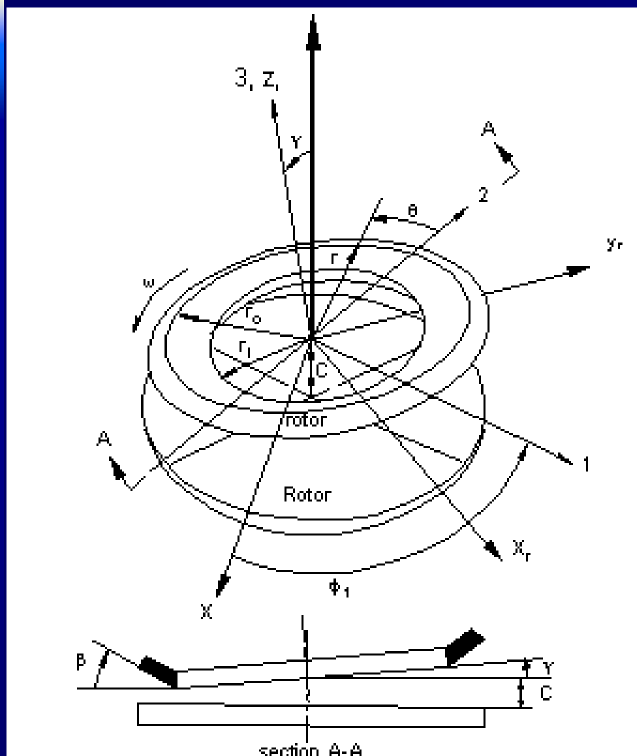
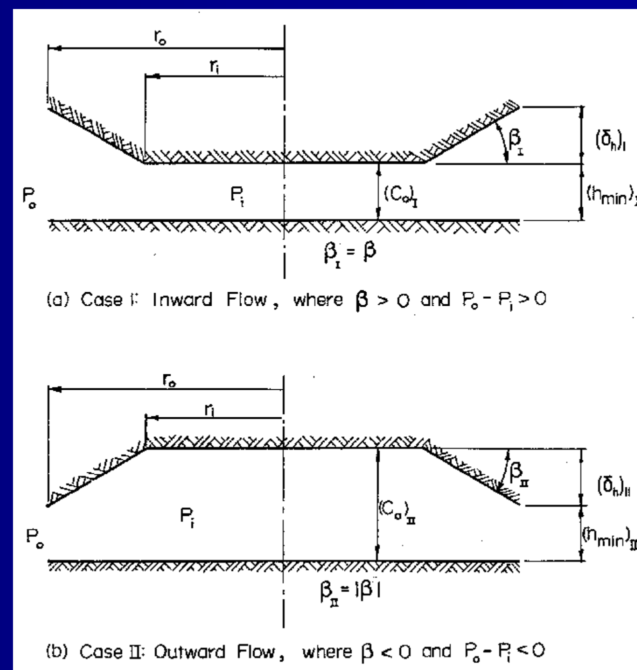
Flat/coned face



- Inward/Outward Flow



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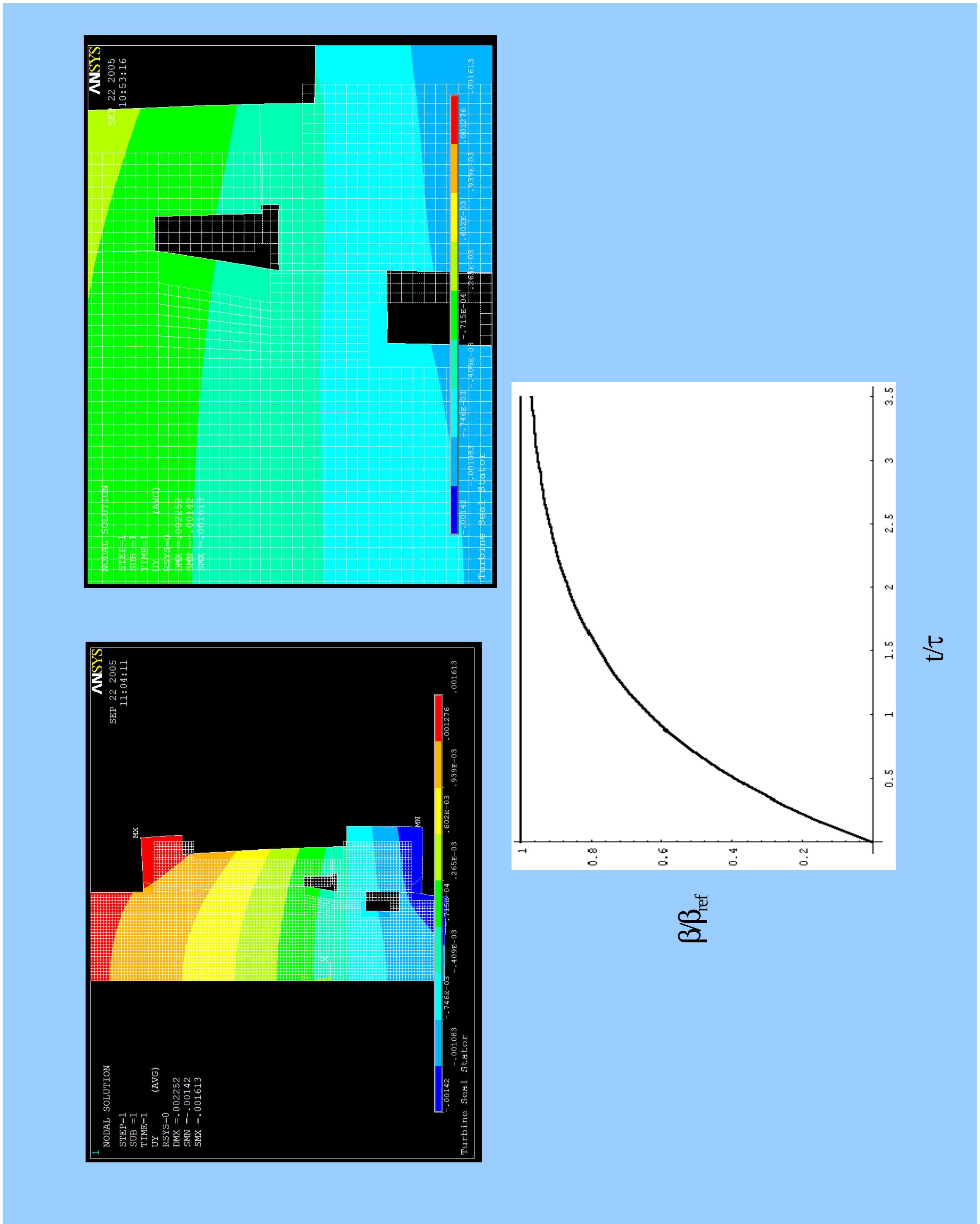
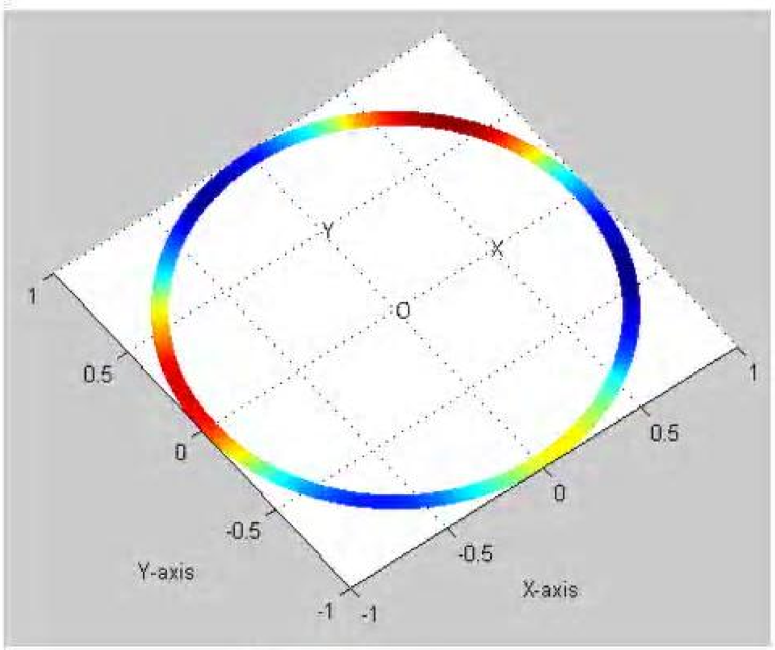


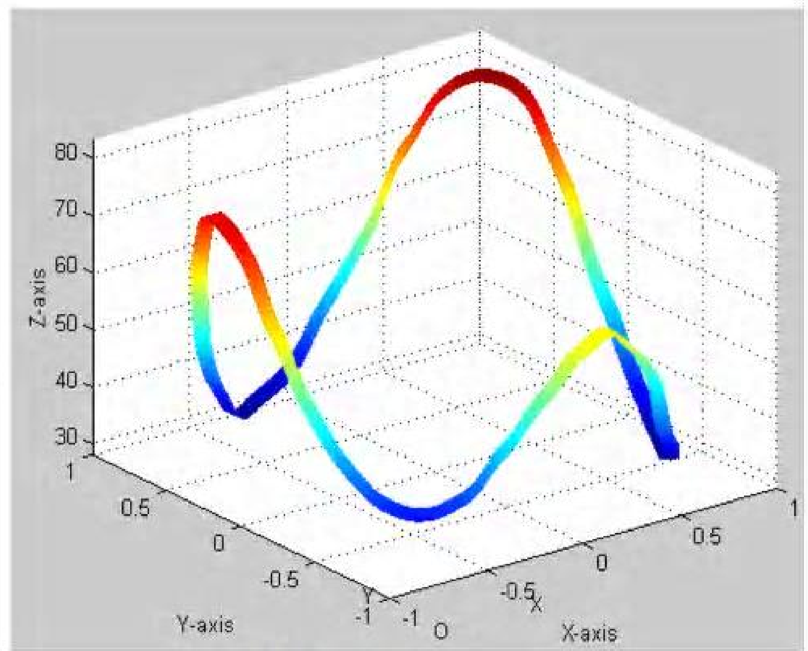
ANSYS
SEP 22 2005
11:04:11
NODAL SOLUTION
STEP=1
SUB =1
TIME=1
UY (AVG)
RSYS=0
DMX =.002252
SMN =-.00142
SMX =.001613
MX
MN
-.00142
-.001083
-.746E-03
-.409E-03
-.715E-04
.265E-03
.602E-03
.939E-03
.001276
.001613
Turbine Seal Stator
ANSYS
SEP 22 2005
10:53:16
t/τ
β/β_ref

Out-of-Service faces*

- wear?
- thermal/mechanical warping?



Top view



Oblique view

* Green and Artiles (2006), manuscript in preparation

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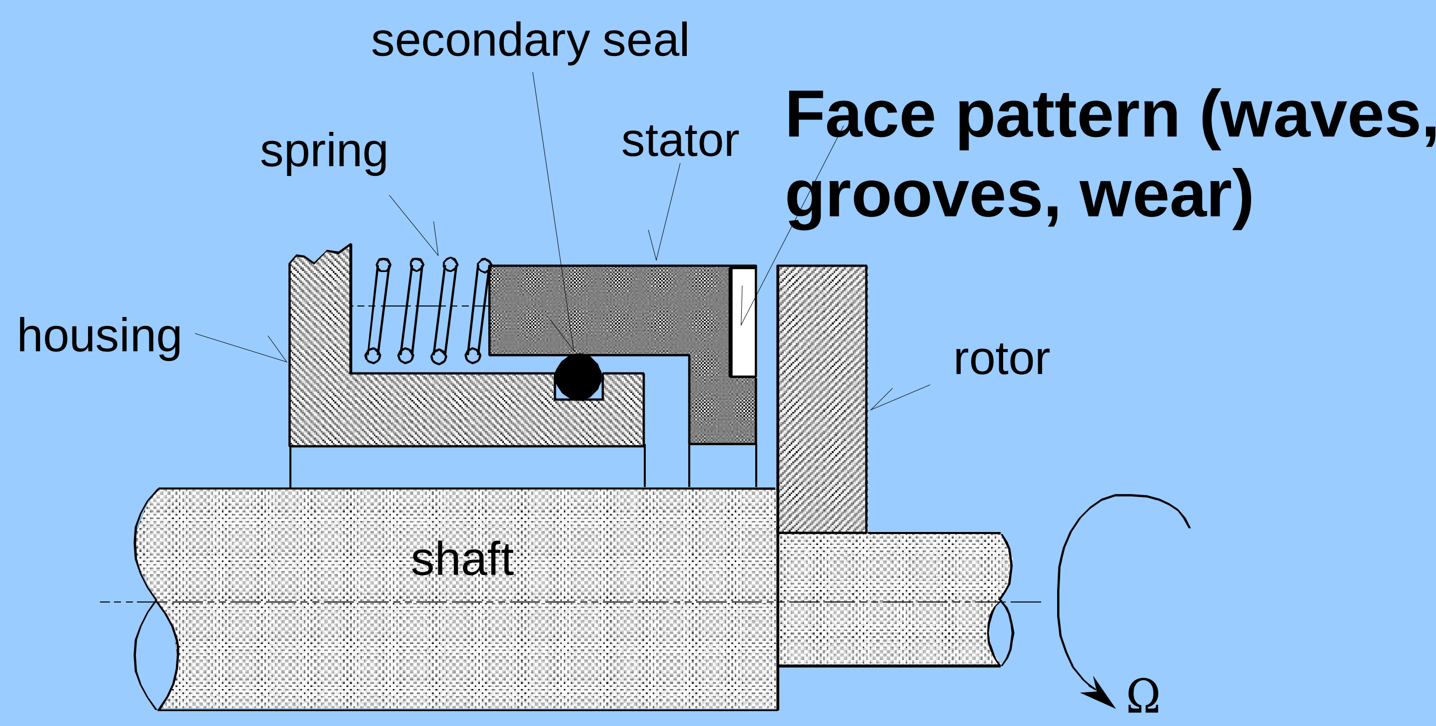


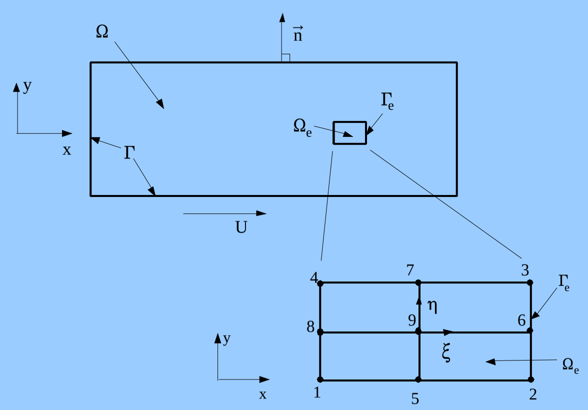
Schematic of Noncontacting Mechanical Face Seal
with Flexibly Mounted Stator

Finite Element Discretization (FEM)

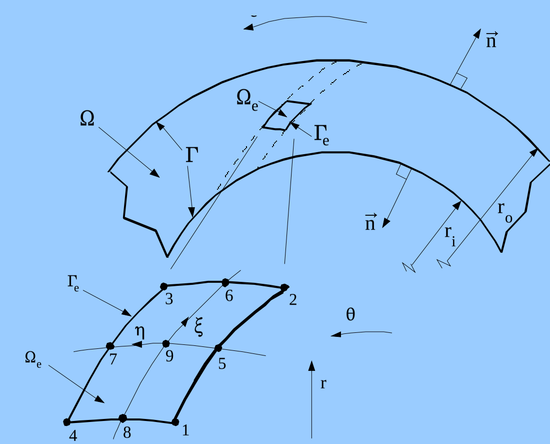
Multiplying RE by a weight factor $\mathbf{W}^T$ and integration by parts gives the weak form:

$$\int_{\Omega} \left\{ -\vec{\nabla} \boldsymbol{W}^{\boldsymbol{T}} \cdot \left[\Phi\, ph^3 \vec{\nabla} p - 6\mu\, \vec{u}\, ph \right] - \boldsymbol{W}^{\boldsymbol{T}}\, 12\mu \frac{\partial (ph)}{\partial t} \right\} d\Omega = 0$$

Discretize domain into small finite elements:



Cartesian coordinate discretization



Polar coordinate discretization

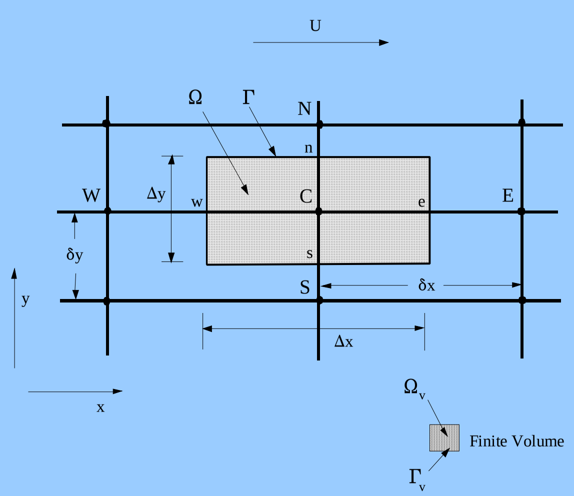
$$p(\xi,\eta) = \sum_{i=1}^{9} \boldsymbol{N_i}(\xi,\eta)\, p_i \qquad \frac{\partial p(\xi,\eta)}{\partial \xi} = \sum_{i=1}^{9} \boldsymbol{N_{i,\xi}}(\xi,\eta)\, p_i \qquad \frac{\partial p(\xi,\eta)}{\partial \eta} = \sum_{i=1}^{9} \boldsymbol{N_{i,\eta}}(\xi,\eta)\, p_i$$

Finite Volume Discretization (FVM)

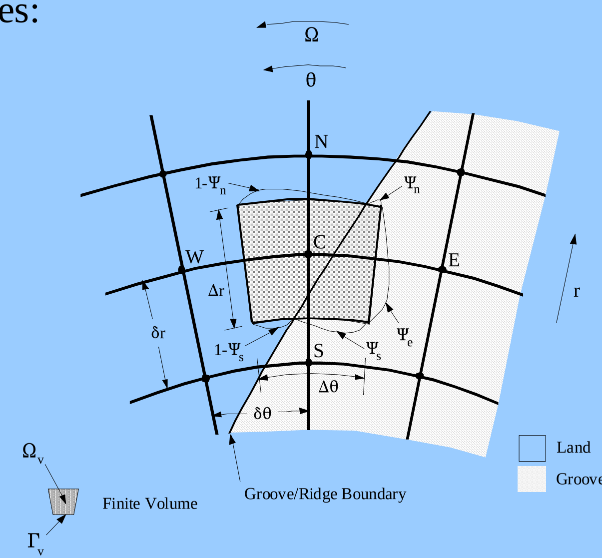
Apply Green's theorem to RE - represents mass conservation over the domain

$$\int_{\Gamma} \left[\Phi p h^3 \vec{\nabla} p - 6\mu \vec{u} p h \right] \cdot \vec{n}\, d\Gamma = \int_{\Omega} \left\{ 12\mu p \frac{\partial h}{\partial t} + 12\mu h \frac{\partial p}{\partial t} \right\} d\Omega$$

Discretize the domain into small finite volumes:

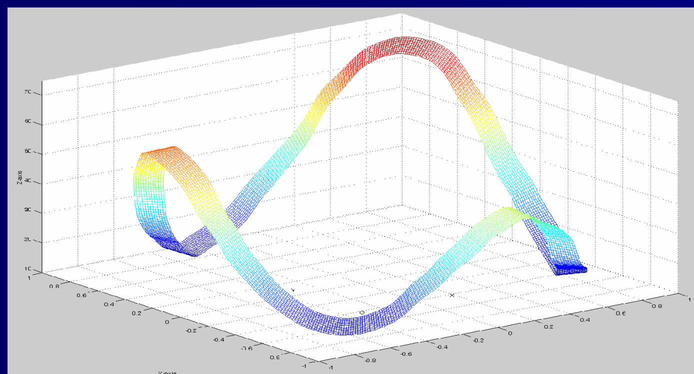


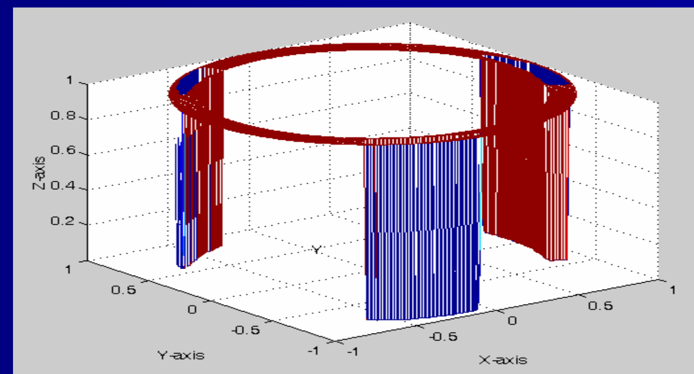
Cartesian coordinate finite volume discretization

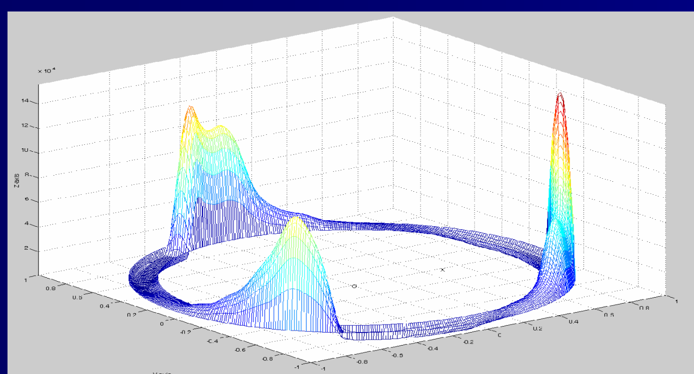


Polar coordinate finite volume discretization

Incompressible Flow







Issues:

- Flow factors (Patir & Chang)

- Cavitation (Elrod, JFO)

- Starvation (?) (can be an issue in low pressure seals)

NASA Seals Workshop
November 14-15, 2006

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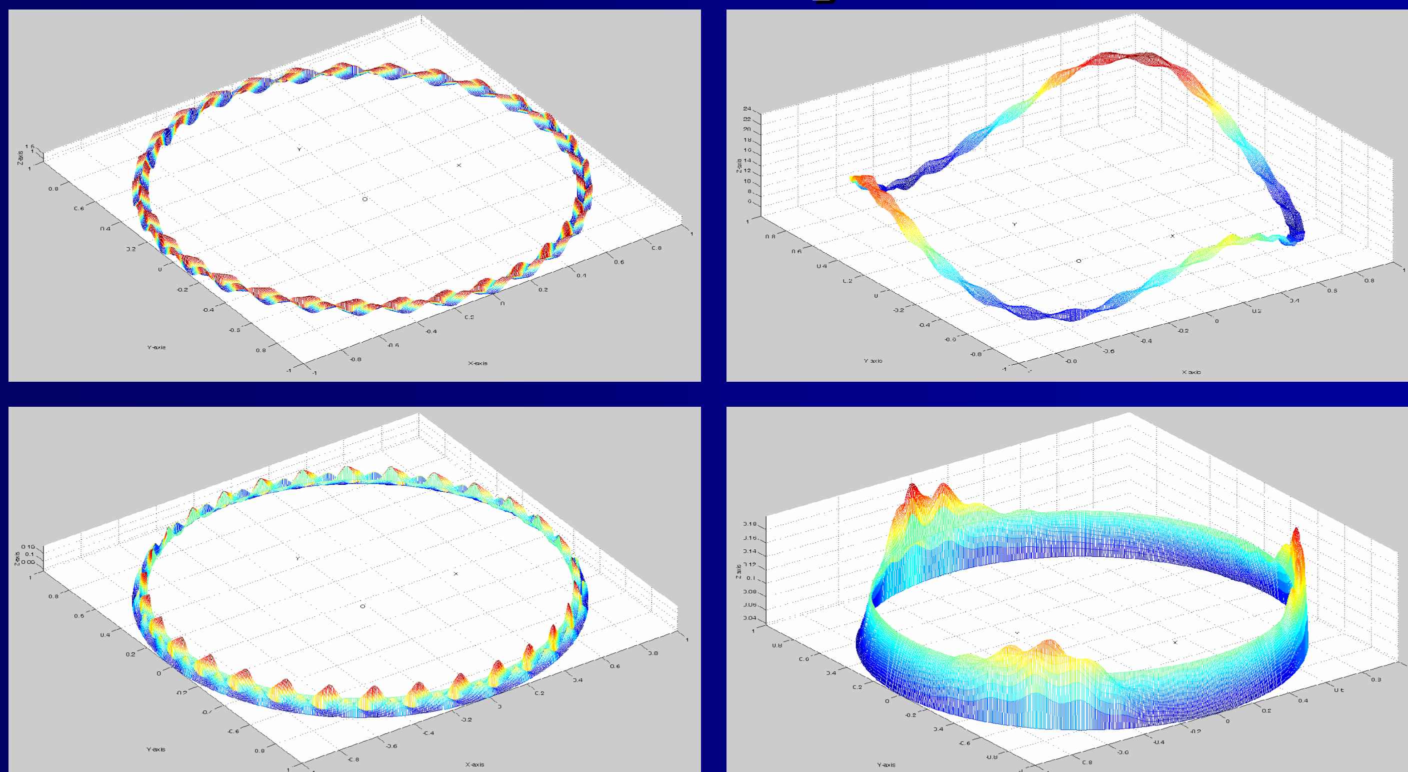
Compressible Flow – Herringbone wavy seal (w/o and w/ face def.)



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EP Contact Load Support (Jackson & Green, 2005)

$$0 \le \omega^* \le \omega_t^* = 1.9$$

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

$$\omega_t^* \le \omega^*$$

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

- 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

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

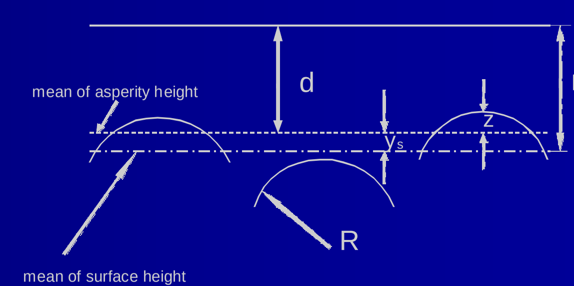
$$A(d) = \eta A_n \int_d^\infty \overline{A}(z-d)\phi(z)dz$$

$$P(d) = \eta A_n \int_d^\infty \overline{P}(z-d)\phi(z)dz$$

Plasticity Index

$$\psi = \sqrt{\frac{\sigma_s}{\omega_c}}$$

- Greenwood and Williamson (1966) formulated the statistical model using Hertz contact.
- The integrals are evaluated using Gauss-Legendre quadrature.
- **Jackson & Green (2005)**



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Subsystem Coupling

Kinetic equations (including time-dependent thermal effects):

$$\frac{\partial}{\partial t}\begin{Bmatrix} \dot{Z} \\ Z \\ \dot{\gamma}_s \\ \gamma_s \\ \dot{\psi} \\ \psi \\ \beta \end{Bmatrix} = \begin{Bmatrix} \left(F_{sZ} + F_{fZ} - F_{cls}\right)/m \\ \dot{Z} \\ \left(M_{sx} + M_{fx}\right)/I + \dot{\psi}^2\gamma_s \\ \dot{\gamma}_s \\ \left[\left(M_{sy} + M_{fy}\right)/I - 2\dot{\psi}\dot{\gamma}_s\right]/\gamma_s \\ \dot{\psi} \\ \left\{\beta_{ref}\left[\left(h_{ref}/h\right)\left(\dot{\psi}_r/\omega_{ref}\right)^2\right] - \beta\right\}/\tau \end{Bmatrix}$$

+

Lubrication equations:

$$\left[S\right]\left\{\dot{p}\right\} = \left\{R\right\}$$

or

$$\{\dot{p}\} = \{R\}$$

=

Coupled set of first-order ODEs:

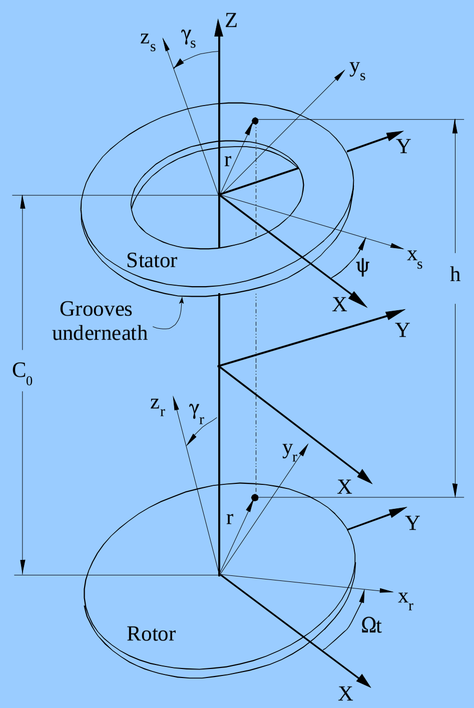
$$FEM: \quad [A(t,\varphi)]\{\dot{\varphi}\} = \{R(t,\varphi)\}$$

$$FVM: \quad \{\dot{\varphi}\} = \{R(t,\varphi)\}$$

1) Systematic coupling of kinetic and lubrication equations

2) Simultaneous solution using numerical ODE solver

Spiral Groove – Load Support in Compressible Flow

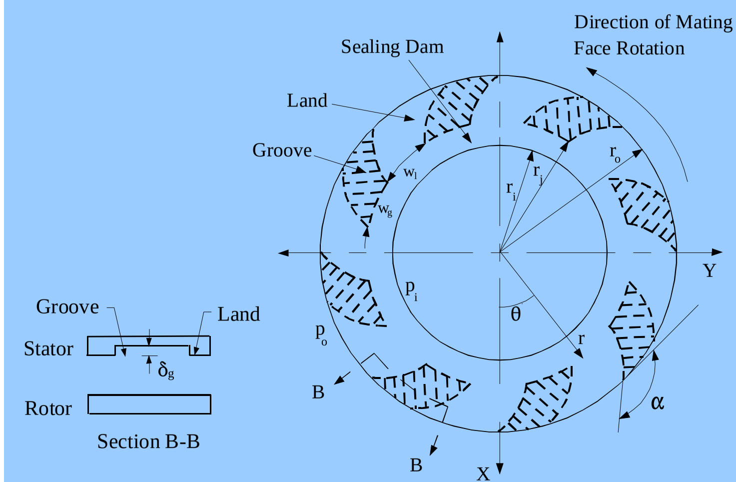


Tilts are small, so treated as vector tilts:

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

3 degrees of freedom:

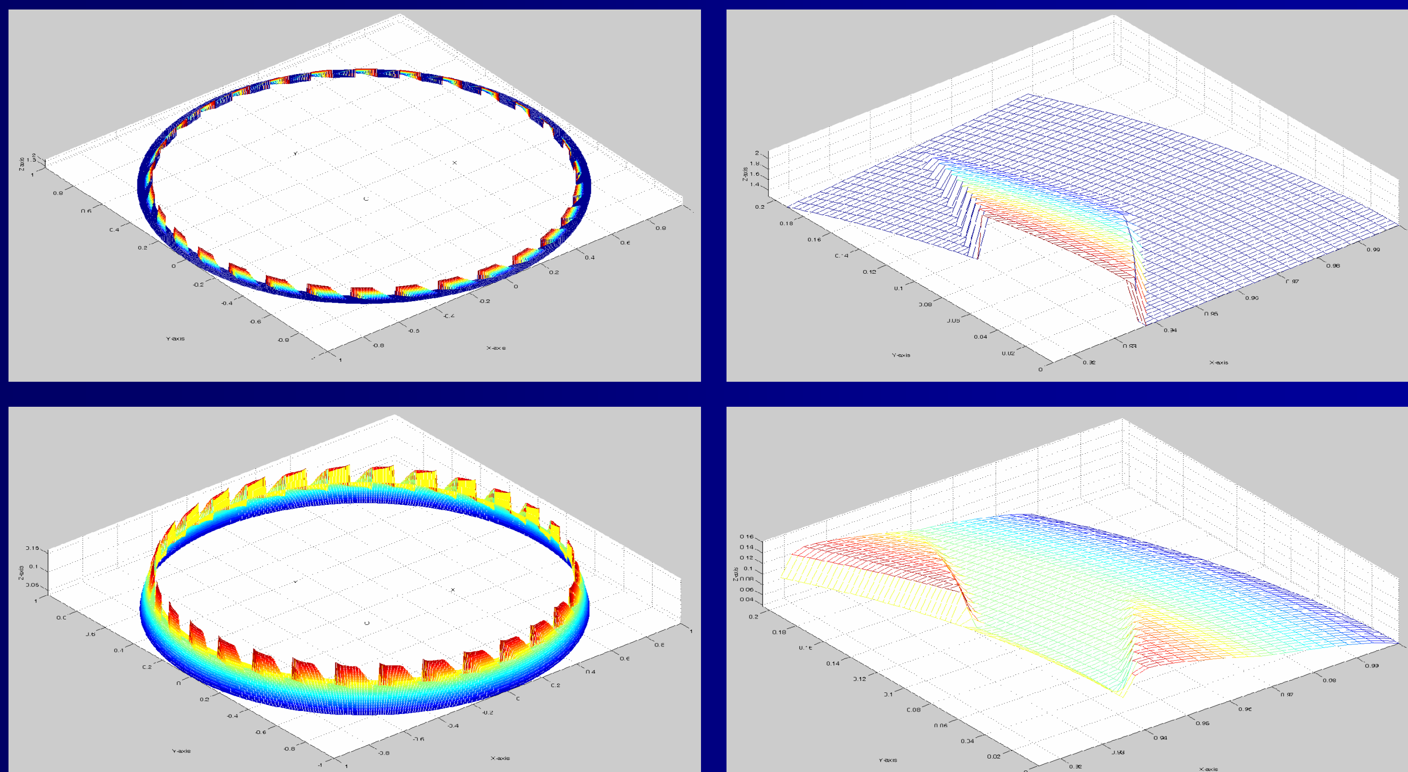
$$\begin{bmatrix} k_{f,Z} & k_{f,\gamma_X} & k_{f,\gamma_Y} \\ k_{m_X,Z} & k_{m_X,\gamma_X} & k_{m_X,\gamma_Y} \\ k_{m_Y,Z} & k_{m_Y,\gamma_X} & k_{m_Y,\gamma_Y} \end{bmatrix}$$



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)



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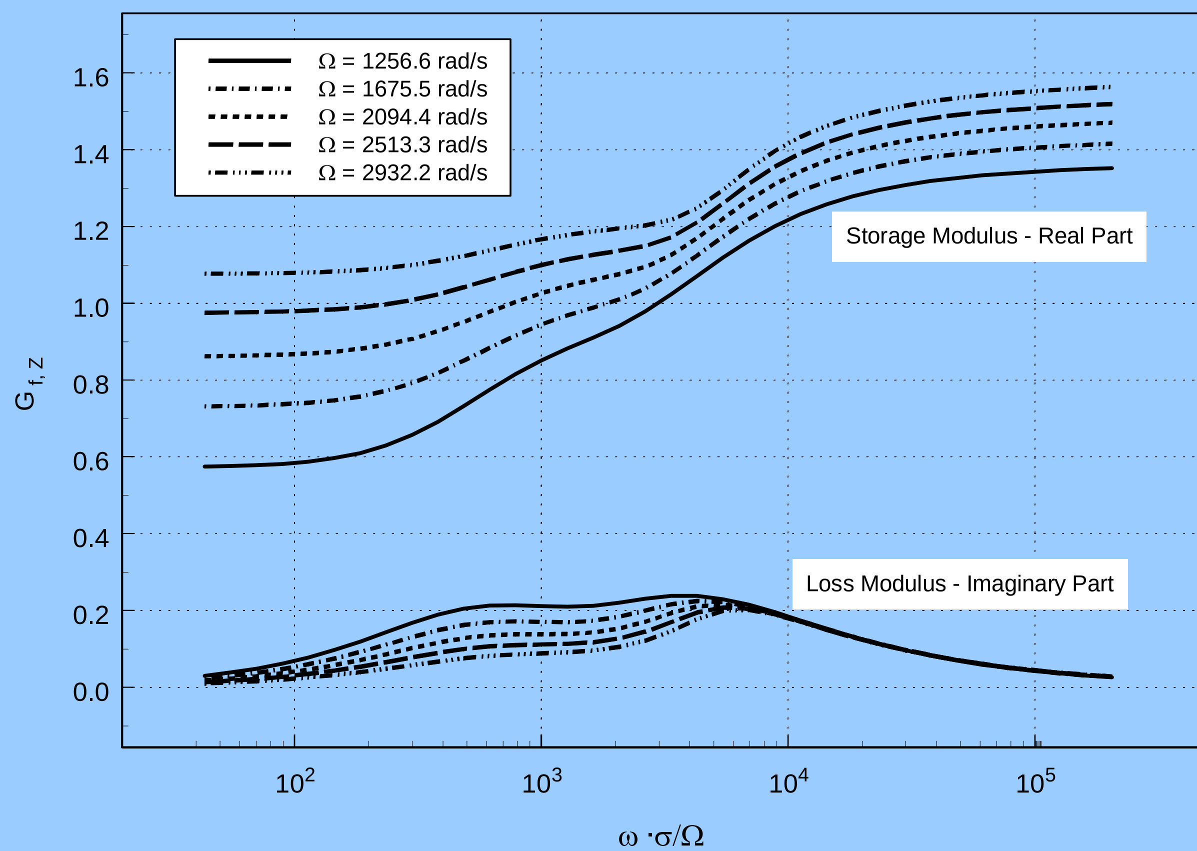
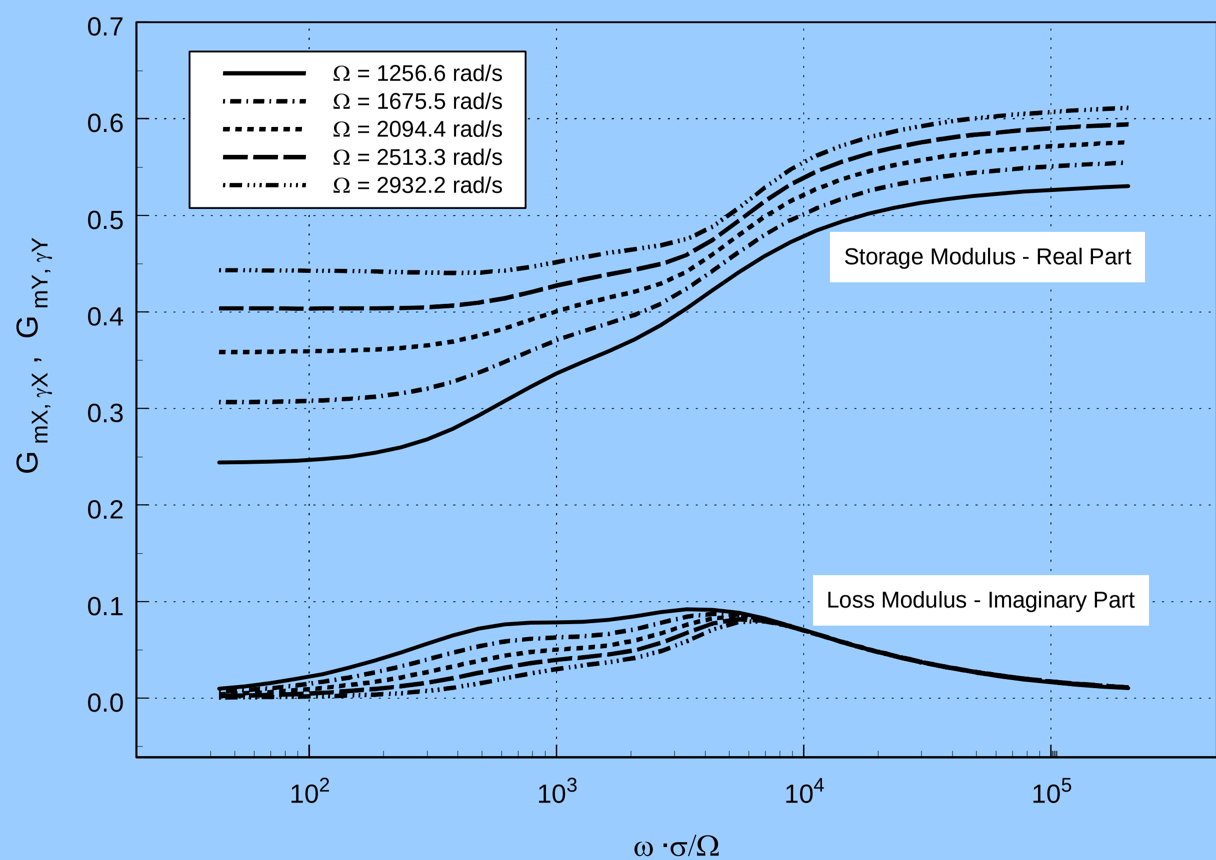
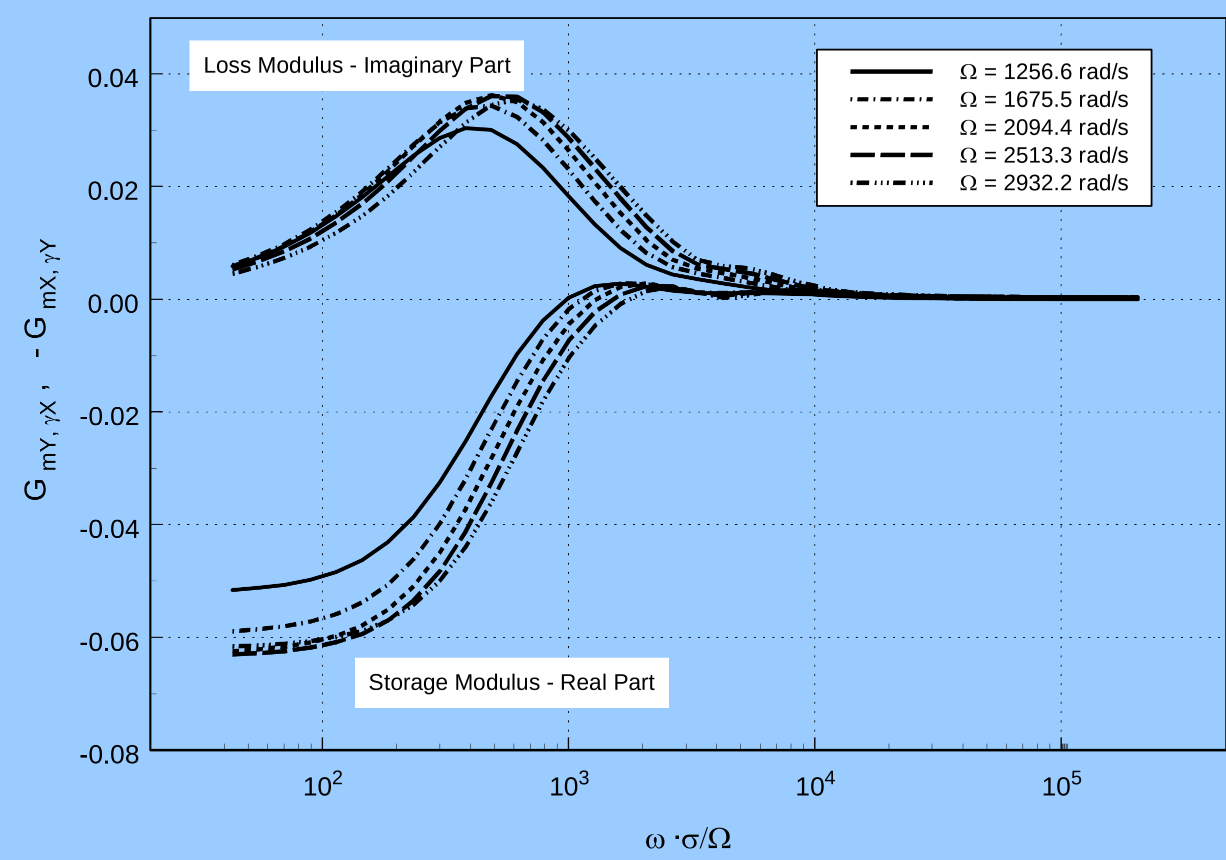
Axial Force Frequency Responses for Mechanical Face Seal



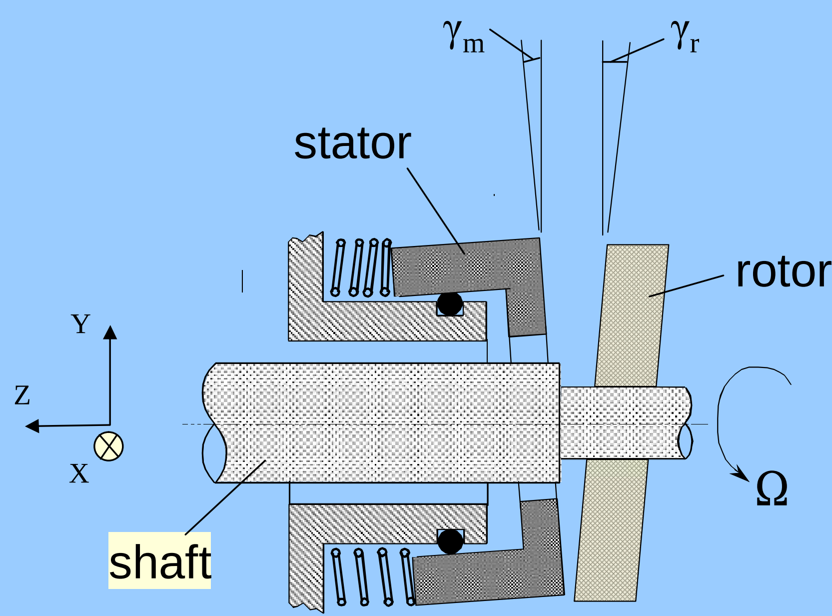
Direct Moment Frequency Responses for Mechanical Face Seal



Cross-Coupled Moment Frequency Responses for Mechanical Face Seal



Rotor runout and static stator misalignment



Static stator misalignment provides constant moment about *X* axis

$$M_{Xi} = \gamma_m \cdot k_{s\gamma}$$

Equations of motion with pseudo springs representing the gas film stiffness (Axial mode decoupled from the tilt modes:

$$m^* \ddot{Z} = -k_{f,Z,g} Z - k_{sZ} Z - d_{sZ} \dot{Z}$$

$$I_t^* \ddot{\gamma}_X = -k_{m_X,\gamma_X,g}\left[\gamma_X - \gamma_r \cos(\Omega t)\right] - k_{s\gamma}\gamma_X - d_{s\gamma}\dot{\gamma}_X - k_{m_X,\gamma_Y,g}\left[\gamma_Y - \gamma_r \sin(\Omega t)\right] + M_{Xi}$$

$$I_t^* \ddot{\gamma}_Y = -k_{m_Y,\gamma_X,g}\left[\gamma_X - \gamma_r \cos(\Omega t)\right] - k_{m_Y,\gamma_Y,g}\left[\gamma_Y - \gamma_r \sin(\Omega t)\right] - k_{s\gamma}\gamma_Y - d_{s\gamma}\dot{\gamma}_Y$$

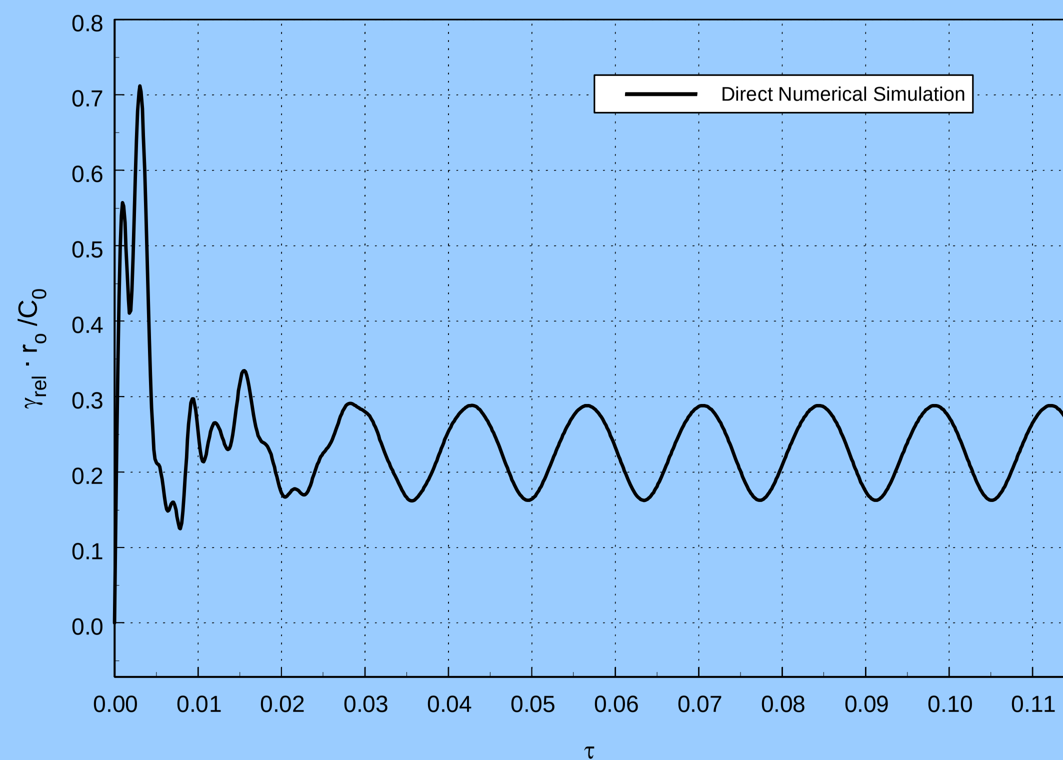
Comparison of Solutions for Transmissibility

Correspondence Principle:

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

Direct Numerical Simulation:

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



r_o=6.0 mm	r_j=5.16 mm	r_i=4.80 mm
C_0=6.0 μm	Ω=2094.4 rad/s	μ=1.8 (10)$^{-5}$ N·s/m^2
P_i=1.0 (10)^5 Pa	P_o=2.0 (10)^5 Pa	m^*=1.0 kg
I_t^*=1.8 (10)$^{-3}$ kg·m^2	N_g=12	α=160 deg
β=0.5	δ=12 (10)$^{-6}$ m	k_{sZ}=5.0 (10)^5 N/m
d_{sZ}=300.0 N·s/m	$k_{s\gamma}$ =900.0 N·m/rad	$d_{s\gamma}$=0.54 N·m·s/rad
γ_r= 0.2 mrad	γ_m= 0.5 mrad	

Transient operation conditions

$$f = 0 \qquad t < 0,\ t > t_3$$

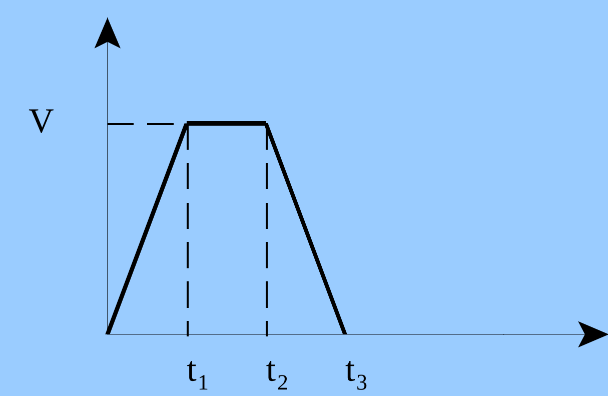
$$f = V\frac{t}{t_1} \qquad 0 \le t \le t_1$$

$$f = V \qquad t_1 \le t \le t_2$$

$$f = V\left(1 - \frac{t - t_2}{t_3 - t_2}\right) \qquad t_2 \le t \le t_3$$

where V is a desired steady-state value

$$\{V\} == \{\dot{\psi}_r = \omega,\ p_i,\ or\ \ p_o\}$$



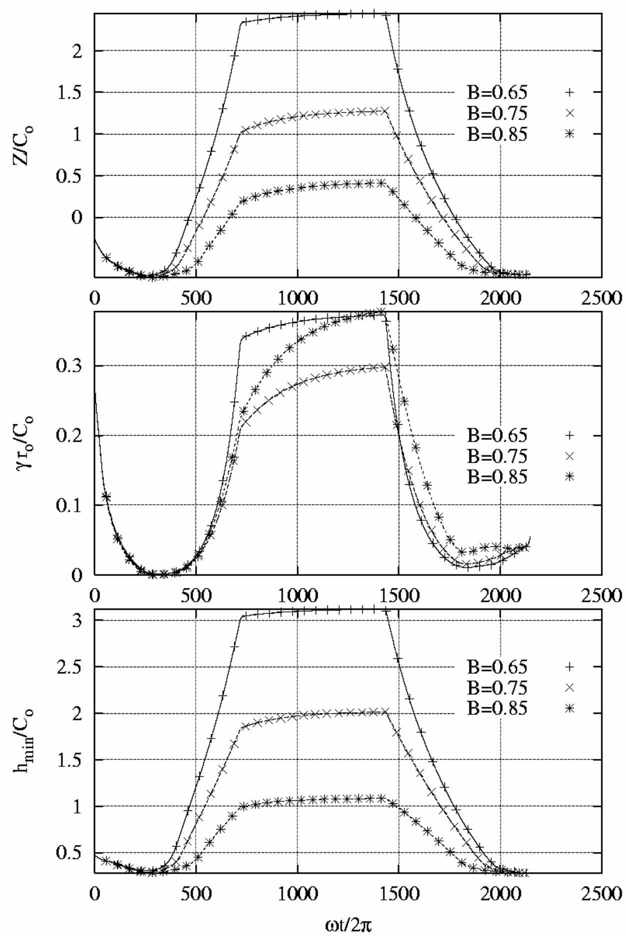


Fig. 4a: Balance ratio effects upon transient response (τ = 2 s)

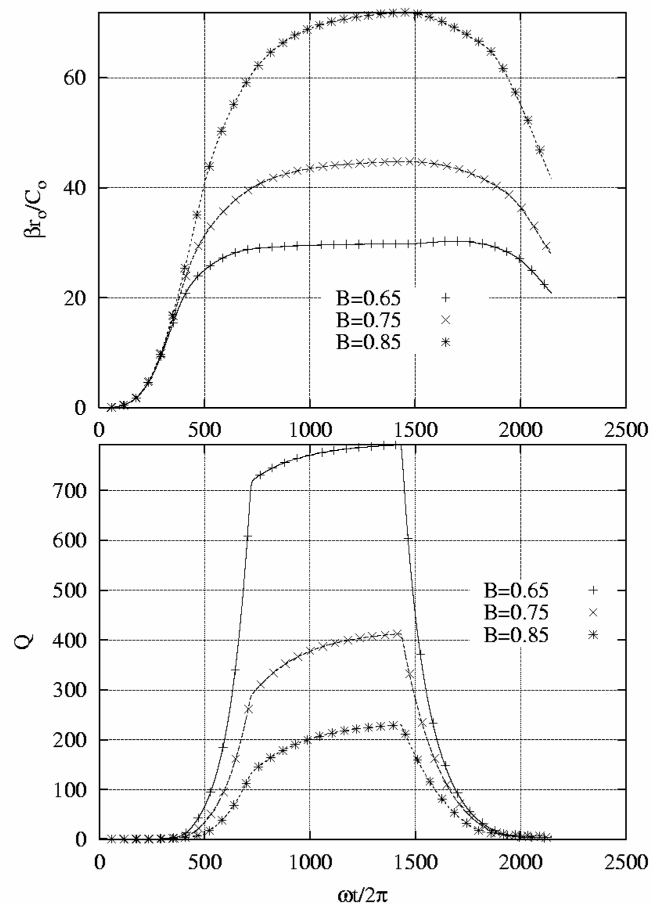


Fig. 4b: Balance ratio effects upon coning and flow (τ=2 s)

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Mechanical Seal Codes

	INCOMP	COMP	SEPARATE	MIXED3D	TAU	TAU-G	Comments
Transient Dynamic Analysis	yes	yes	no (1a)	no	yes	yes	(1a) can predict analytically separation speed, (1n) separation speed is obtained from numerical simulation
Degrees of Freedom	3 (non-axisymmetric)	3 (non-axisymmetric)	2 (axisymmetric)	3 (non-axisymmetric)	3 (non-axisymmetric)	3 (non-axisymmetric)	
Incompressible	yes	no	yes	no	yes	no	
Compressible	no	yes	yes	yes	no	yes	
Noncontacting	yes	yes	yes(2)	yes(2)	yes(2)	yes(2)	(2) seamless transition from contacting to noncontacting modes of operation; thus, classification of conttact/noncontact is no longer needed.
Contacting	no	no	yes(2)	yes(2)	yes(2)	yes(2)	
Coning	yes(3)	yes(3)	yes(3)	yes(4)	yes(3)	yes(4)	(3) linear coning; (4) cubic coning
Wavy	no	no	no	yes(5)	no	yes(5)	(5) periodic; or arbitrary (read from file)
Spiral grooves	no	no	no	yes(6)	no	yes(6)	(6) Includes a sector solution (as an option)
Thermal Face Deformation	no	no	no	no	yes(7)	yes(7)	(7) using time-dependent ad hoc differential equation (allows complete transient analysis)
Separation Speed	no	no	yes(1a)	no	yes(1n)	yes(1n)	
Wear Model	no	no	no	no	yes(8)	yes	(8) Under development

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