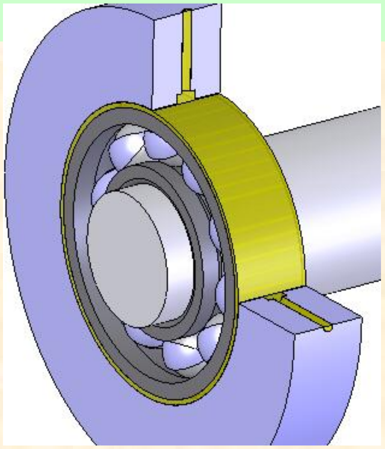
A Linear Fluid Inertia Model for Improved Prediction of Force Coefficients in Grooved Squeeze Film Dampers and Grooved Oil Seal Rings

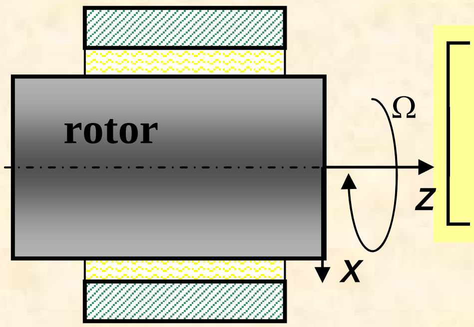


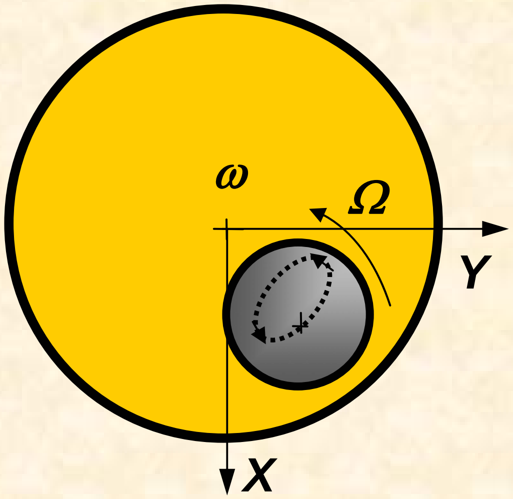
Luis San Andres

Adolfo Delgado

Texas A&M University

Bearing & seal dynamic reaction forces





Bearing coordinate system

$$\begin{bmatrix} F_X \\ F_Y \end{bmatrix} = -\begin{bmatrix} K_{XX} & K_{XY} \\ K_{YX} & K_{YY} \end{bmatrix}_B \begin{Bmatrix} X \\ Y \end{Bmatrix} - \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix}_B \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix}$$

Stiffness coefficients

Damping coefficients

Typically:
Small clearances (c/R < 0.001)
***Re**<<1**
No fluid inertia forces are included

Squeeze film Reynolds number

$$Re^* = \frac{\rho \omega c^2}{\mu}$$

Unsteady flow fluid inertia force

Viscous fluid force

Fluid inertia – When is it important?

-Large clearances and/or groove depths
-Long axial length
-High frequencies

$$Re^* = \frac{\rho \omega c^2}{\mu} >> 12$$

Grooved Oil Seals & Squeeze Film Dampers.

Dynamic reaction forces:

Oil seal:

$$\begin{bmatrix} F_X \\ F_Y \end{bmatrix} = -\begin{bmatrix} K_{XX} & K_{XY} \\ K_{YX} & K_{YY} \end{bmatrix}_S \begin{Bmatrix} X \\ Y \end{Bmatrix} - \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix}_S \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} - \begin{bmatrix} M_{XX} & M_{XY} \\ M_{YX} & M_{YY} \end{bmatrix}_S \begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix}$$

Stiffness coefficients

Damping coefficients

Inertia coefficients

SFD-CCO centered:

$$\begin{bmatrix} F_X \\ F_Y \end{bmatrix} = -\begin{bmatrix} K_{XX} & 0 \\ 0 & K_{YY} \end{bmatrix}_S \begin{Bmatrix} X \\ Y \end{Bmatrix} - \begin{bmatrix} C_{XX} & 0 \\ 0 & C_{YY} \end{bmatrix}_S \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} - \begin{bmatrix} M_{XX} & 0 \\ 0 & M_{YY} \end{bmatrix}_S \begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix}$$

from centering spring

Added mass coefficients:

- comparable to journal mass (i.e. can shift system natural frequency)
- larger in grooves (i.e. large clearances)

Fluid inertia – How large is it?

$$M_{fluid} := \rho \cdot \pi \cdot D \cdot L \cdot c \qquad M_{steel} := \rho_{steel} \cdot \pi \cdot \left(\frac{D}{2}\right)^2 \cdot L$$

$$M_{XX} := \rho \cdot \pi \cdot \left(\frac{D}{2}\right)^3 \cdot \frac{L}{c} \cdot \left(1 - \frac{\tanh\left(\frac{L}{D}\right)}{\frac{L}{D}}\right)$$

[1] Reinhardt, F., and Lund, J. W., **1975**, "The Influence of Fluid Inertia on the Dynamic Properties of Journal Bearings," ASME J. Lubr. Technol., 97(1), pp. 154-167.

SHORT SEAL ***L/D*=0.2**

.2

$M_{XX} = 2.91\ \text{kg}$

$M_{fluid} = 2.76 \times 10^{-3}\ \text{kg}$

$M_{steel} = 4.34\ \text{kg}$

$$\frac{M_{XX}}{M_{steel}} = 0.67$$

LONG SEAL ***L/D*=0.5**

0.5

$\boxed{M_{XX} = 42.03\ \text{kg}}$

$M_{fluid} = 6.9 \times 10^{-3}\ \text{kg}$

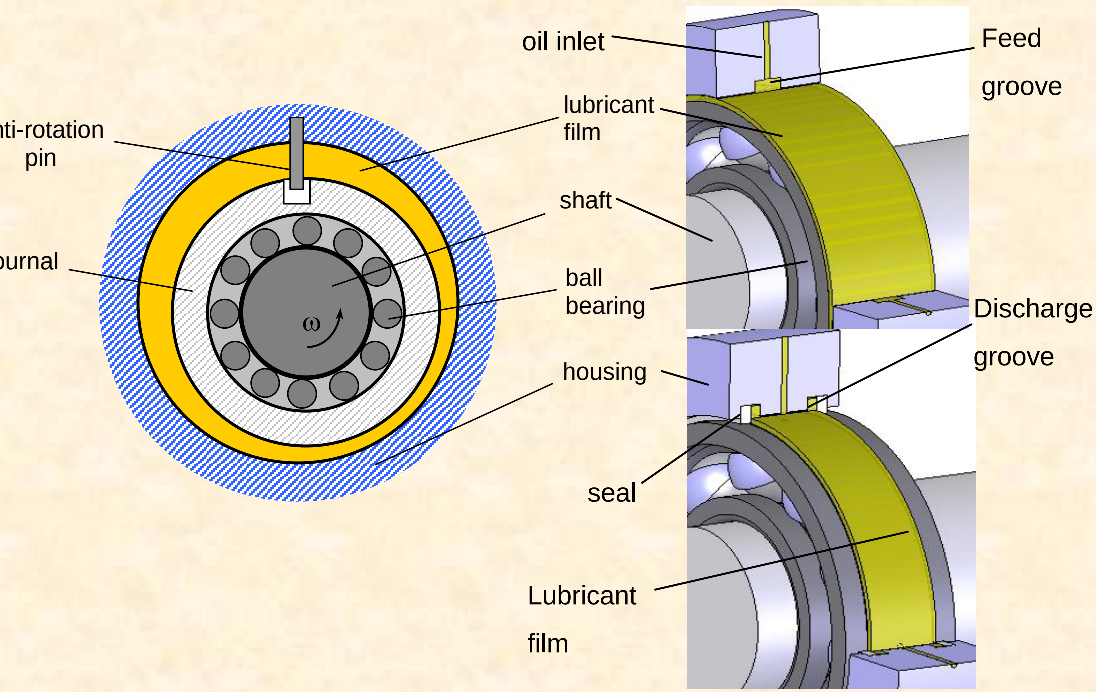
$M_{steel} = 10.84\ \text{kg}$

$$\frac{M_{XX}}{M_{steel}} = 3.88$$

Fluid mass inside film is just a few grams, but…. since added mass is proportional to Diameter and 1/clearance

Mxx >> mass of solid steel journal [1]

Squeeze film damper



In aircraft gas turbines and compressors, squeeze film dampers aid to attenuate rotor vibrations and provide mechanical isolation.

Common configurations include grooves and recesses to supply oil and to prevent oil starvation in squeeze film lands.

<u>Known issue:</u> Poor predictions of fluid film added masses for grooved SFDs

Added mass coefficient - Literature review

Grooved SFDs

San Andrés, 1992 (Predictions)

- SFD with a shallow groove behaves at low frequencies as a **single land damper**
- Groove influences dynamic force coefficients

Arauz and San Andrés, 1994 (Prediction and experiments)

- Dynamic pressure levels at the groove ($c_g/c < 10$) and film land are of the **same order of magnitude**
- Model **underestimates** the radial force and overestimates the tangential (damping) force at the damper film land.

Qingchang et al., 1998 (Predictions and experiments)

- Experimental results show that radial (inertial) force is underpredicted by a factor of **three**.

Added mass coefficient - Literature review (cont.)

Grooved SFDs

Lund et al., 2003 (Predictions and experiments)

Fluid inertia force coefficient is overpredicted (up to 70 %) for increasing groove volumes

Kim and Lee, 2005 (Predictions and experiments)

Predictions of the inertia coefficient correlate well experimental data while damping coefficients are underestimated (two-stage liquid seal)

San Andres and Delgado, 2006 (experiments)

Added mass coefficients are underpredicted by classical model (5 times smaller)

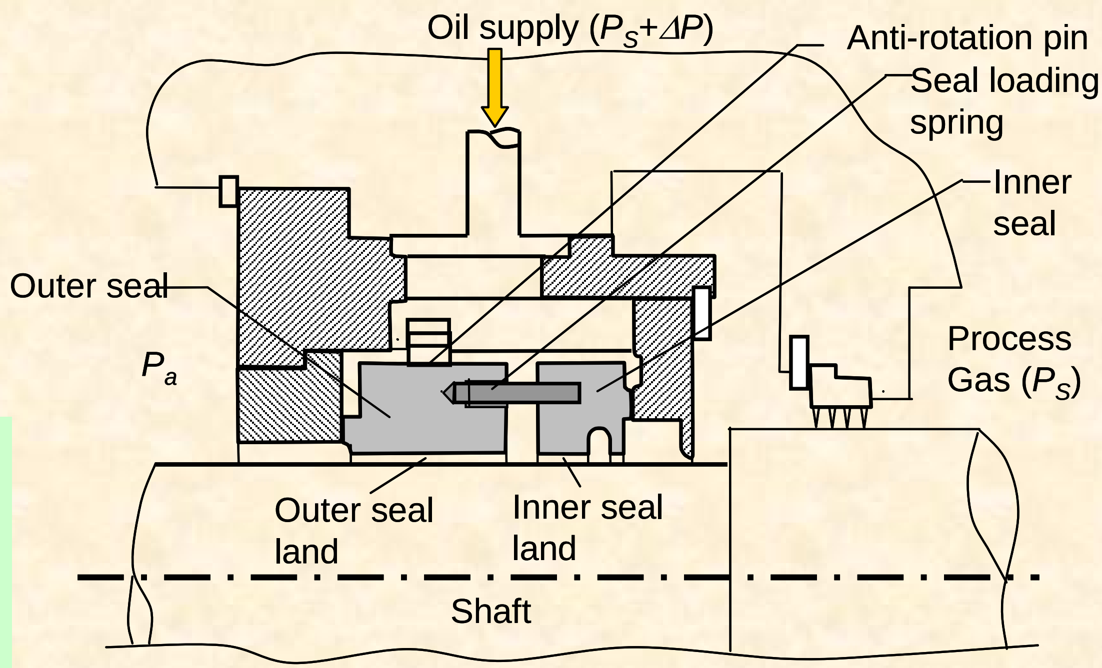
Current models do not properly predict both damping and added mass coefficients in grooved SFDs

Oil seal rings

In centrifugal compressors oil seals are commonly used to prevent leakage of process gas into ambient.

- Locked oil seal rings can induce instability in compressors.

- Seals are grooved to reduce cross-coupled stiffness and lower lock-up forces.



Typical oil seal multi-ring assembly

Smooth oil seals- Experimental results

Childs et al., (2006, 2007)

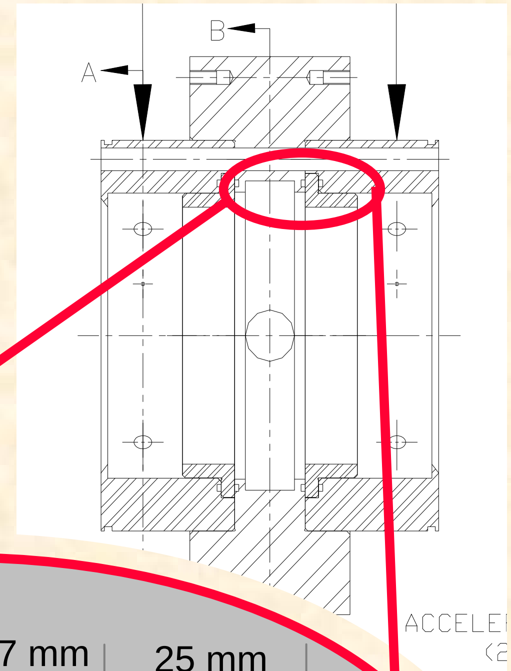
Parallel seal configuration **(balance thrust force due to pressure drop across the seals)**

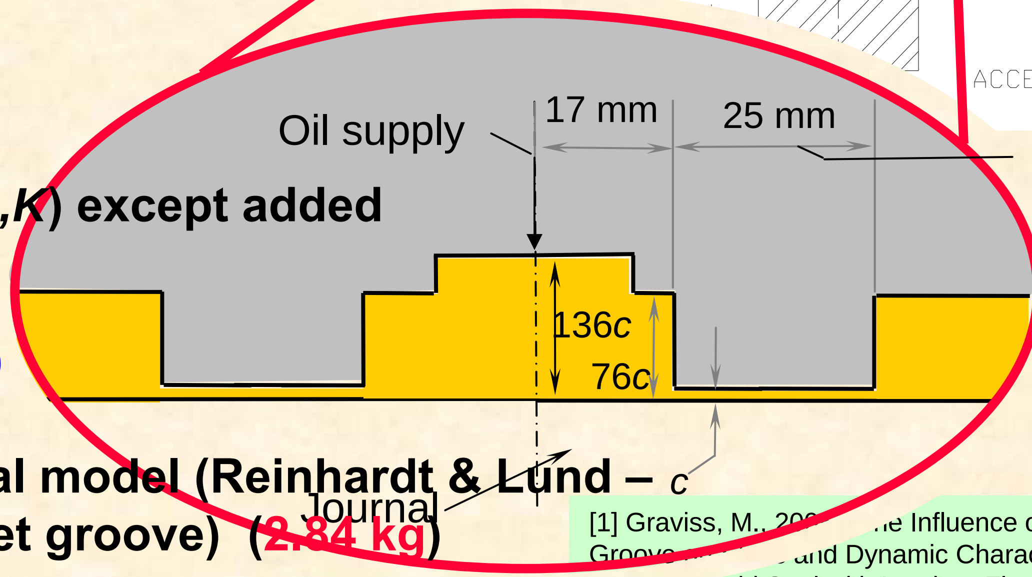
Includes 'deep' inlet (central) groove to feed seals

Parameter identification: $F_{SEAL} = 1/2\ F_{Test\ conf.}$

Predictions do not consider groove or fluid inertia effects
(Zirkelback and San Andrés 1996)

Parallel oil seal configuration [1]





Results

Force coefficients are well predicted (*C*,*K*) except added mass coefficients

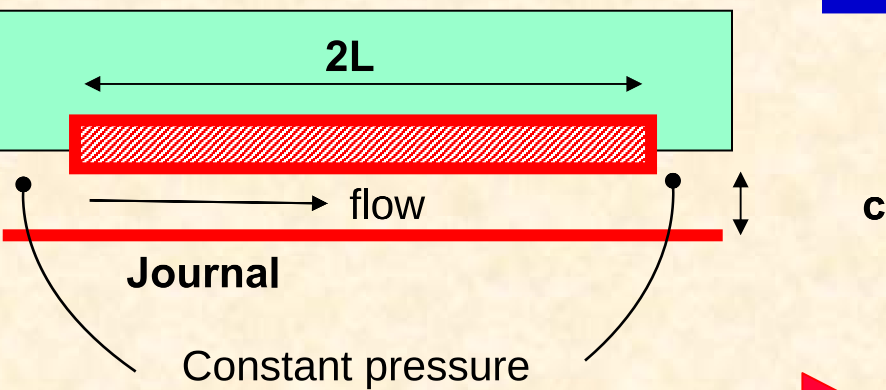
Large added mass coefficients (~15 kg)

Added mass predictions using Classical model (Reinhardt & Lund – 1975) (single land- i.e. not including inlet groove) (2.84 kg)

[1] Graviss, M., 200... ...e Influence of a Central Groove ... and Dynamic Characteristics of an ...nular Liquid Seal with Laminar Flow," M.S. Thesis, Texas A&M Univ., College Station, TX.

Compare smooth and grooved seals of short length

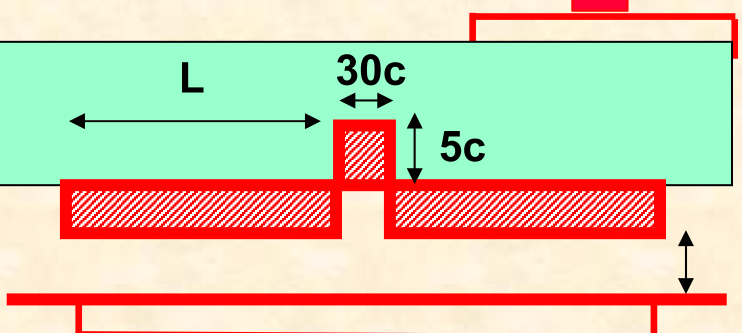
Damping, cross-coupled stiffness & added mass



$$C_{XX}=8\frac{\pi\,\mu\,D\,L^3}{4\,c^3};\;\; K_{XY}=\frac{1}{2}\Omega\,C_{XX};\;\; M_{XX}=8\frac{\pi\,\rho\,D\,L^3}{20\,c}$$

$$C_{XX}=\frac{\pi\,\mu\,D\,L^3}{4\,c^3};\;\; K_{XY}=\frac{1}{2}\Omega\,C_{XX};\;\; M_{XX}=\frac{\pi\,\rho\,D\,L^3}{20\,c}$$

2-land seal: (deep groove divides lands)



Coeffs are ¼ of original seal

$$C_{XX}=2\frac{\pi\,\mu\,D\,L^3}{4\,c^3};\;\; K_{XY}=\frac{1}{2}\Omega\,C_{XX};\;\; M_{XX}=2\frac{\pi\,\rho\,D\,L^3}{20\,c}$$

Grooved oil seals- Literature review

Predictive models

Semanate and San Andrés, (1993)

- Bulk flow equation model

- Grooves should reduce force coefficients by a factor of **four**, i.e.

K_{xy} (1 land)= 4 K_{xy}(2 lands)
C_{xx} (1 land)= 4 C_{xx}(2 lands)

-Fluid inertia effects not predicted (considered negligible)

Baheti and Kirk, (1995)

- Reynolds and energy equation (Finite element solution)
- Grooves should effectively isolate seal lands
- Cross-coupled stiffness and damping coefficients are reduced by ~60 % for grooved configurations

Grooved oil seals- Experimental results

Childs et al., (2006)

Single inner land groove and multiple groove oil seal (single clearance)

Childs et al., (2007)

One inner land groove with groove depths (5*c*,10*c*,15*c*)

Results

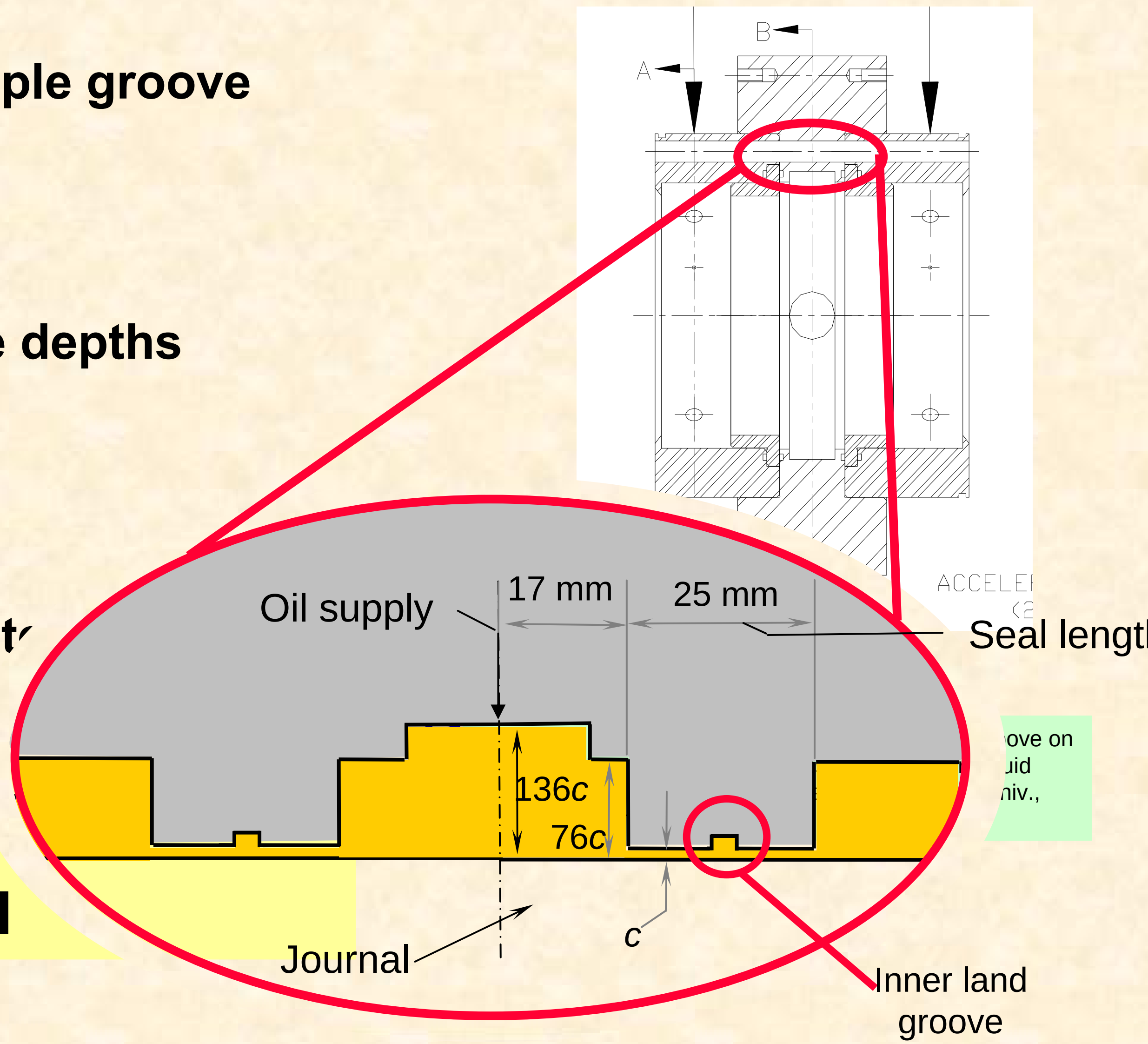
Force coefficients are underpredict

Inner land groove does not effecti

Large added mass coefficients (~

Higher than for 'smooth' seal

Parallel oil seal configuration [1]



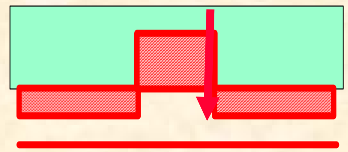
Test oil seal

Predictions

Experiments

Inlet groove not considered (null dynamic pressure)- Null added mass coefficients

≠

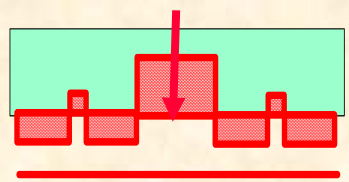


Large added mass coefficients

Inner land groove should reduce crossed-coupled stiffness and direct damping coefficients by a factor of four

K_{xy} (1 land)= 4 K_{xy}(2 lands)
C_{xx} (1 land)= 4 C_{xx}(2 lands)

≠



Groove does not effectively separate seal lands

At most:

K_{xy} (1 land)= 2 K_{xy}(2 lands)
C_{xx} (1 land)= 2 C_{xx}(2 lands)

Null (neglected) added mass coefficients

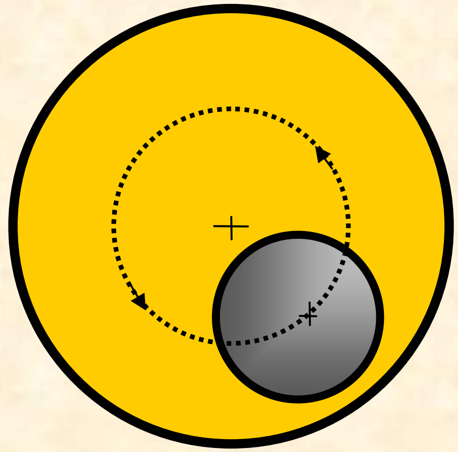
≠

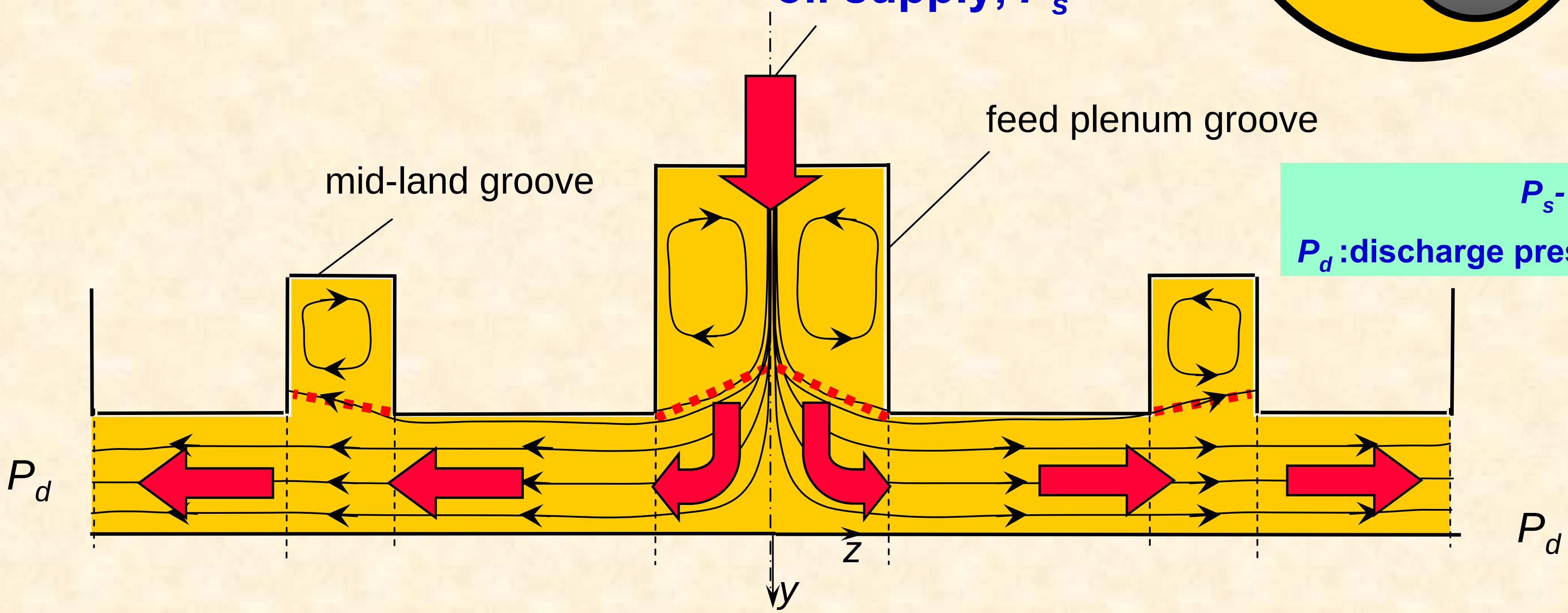
Large added mass coefficients, increasing with increasing groove depth

Need for better predictive models

Fluid flow predictive model

- **Bulk flow, centered operation, incompressible fluid**
- **Qualitative observations of laminar flow field**
  - **Boundary Conditions**
  - **Characteristic groove depth**





$P_s - P_d > 0$

P_d :discharge pressure

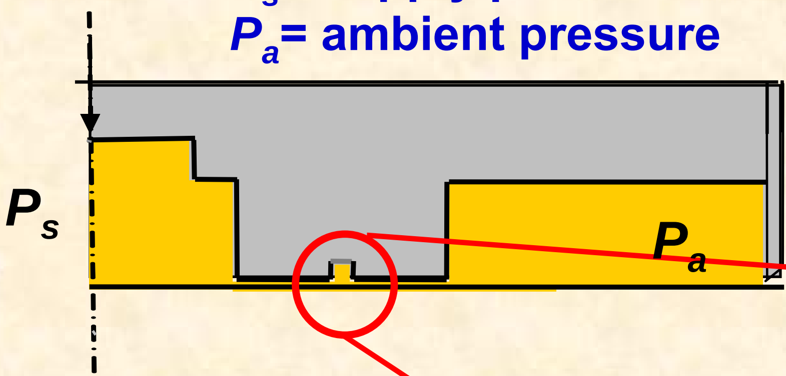
Streamlines in axially symmetric grooved annular cavity.

Groove effective depth

Childs, D. W., Graviss, M., and Rodriguez, L. E., 2007, "The Influence of Groove Size on the Static and Rotordynamic Characteristics of Short, Laminar-Flow Annular Seals," ASME J. Tribol, **129**(2), 398-406.

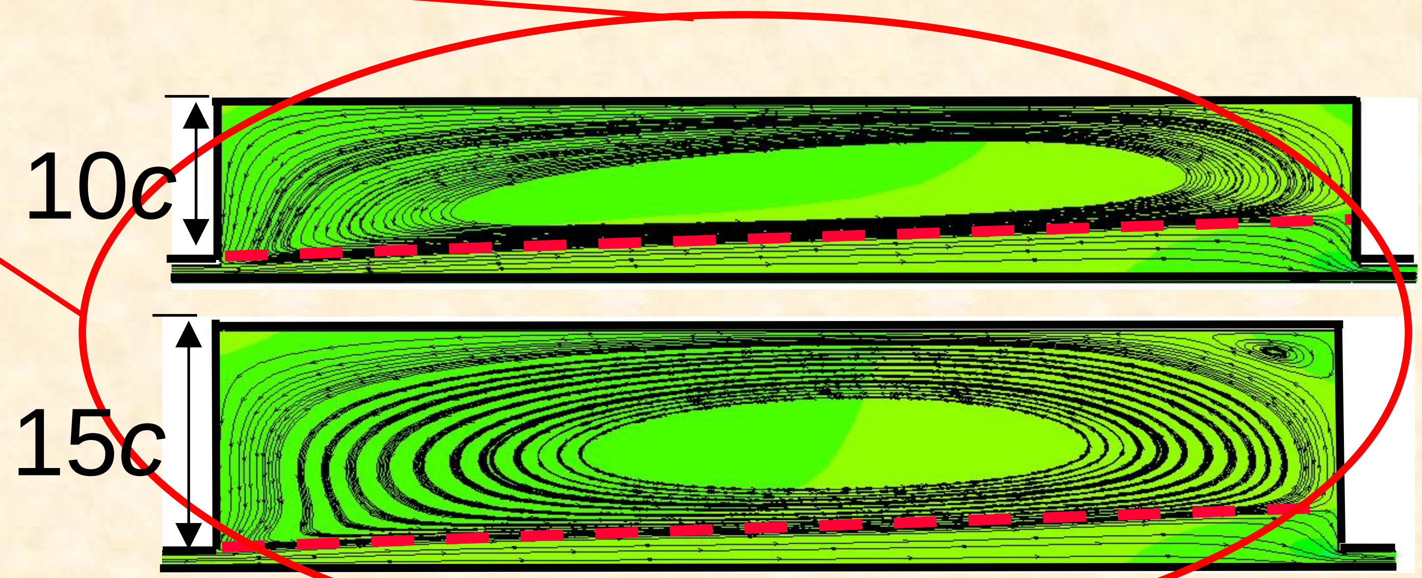
CFD simulations show: streamline separating flow regions IS a <u>physical boundary</u> delimiting the domain for squeeze film flow due to journal radial motions.

P_s= supply pressure
P_a= ambient pressure



Test seal

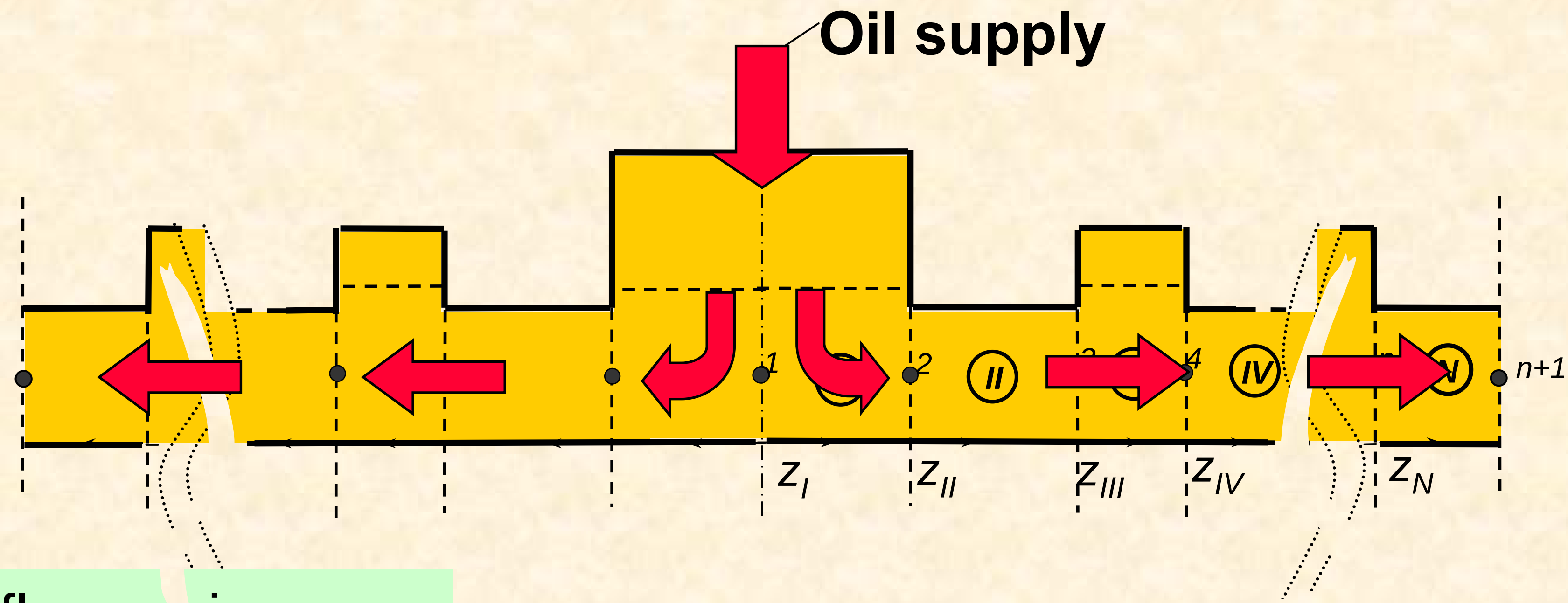
Laminar flow



Inner land groove close up (CFD -Pressure driven flow)

Linear fluid inertia model

No fluid inertia advection



In each flow region:

Reynolds equation with temporal fluid inertia

$$\frac{\partial}{\partial x}\left(h_\alpha{}^3 \frac{\partial P_\alpha}{\partial x} \right) + \frac{\partial}{\partial z_\alpha}\left(h_\alpha{}^3 \frac{\partial P_\alpha}{\partial z_\alpha} \right) = 12\,\mu \frac{\partial}{\partial t}\left(h_\alpha\right) + 6\mu R\Omega \frac{\partial}{\partial x}\left(h_\alpha\right) + \left(h_\alpha{}^2\right) \frac{\partial^2}{\partial t^2}\left(\rho h_\alpha\right)$$

$$\alpha = I, II, \ldots N$$

Perturbation flow analysis

Centered operation $\Delta e_X , \Delta e_Y << c_{\eta_\alpha}$

Film thickness

$$h_\alpha = c_{\eta_\alpha} + e^{i\omega t} \left\{ \Delta e_X \cos(\theta) + \Delta e_Y \sin(\theta) \right\}$$

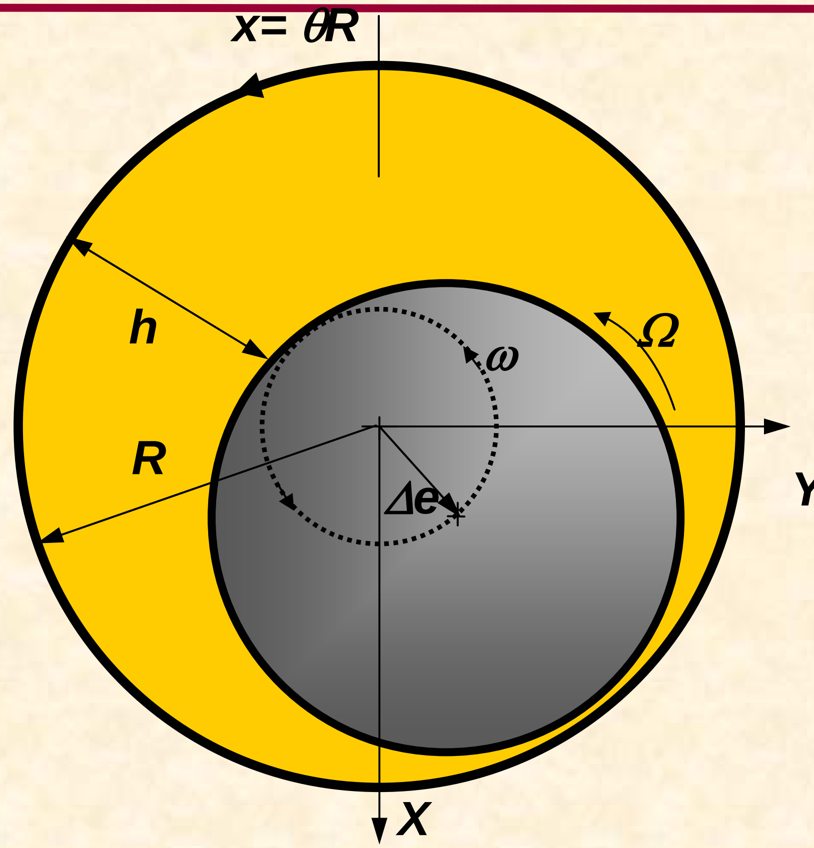
Pressure field

$$P_\alpha = P_{0_\alpha} + e^{i\omega t} \left\{ \Delta e_X \, P_{X_\alpha} + \Delta e_Y \, P_{Y_\alpha} \right\}$$

Flow field

$$\dot{q}_{x_\alpha} = \dot{q}_{x_{0_\alpha}} + e^{i\omega t} \left\{ \Delta e_X \, \dot{q}_{x_{X_\alpha}} + \Delta e_Y \, \dot{q}_{x_{Y_\alpha}} \right\}$$ Circumferential

$$\dot{q}_{z_\alpha} = \dot{q}_{z_{0_\alpha}} + e^{i\omega t} \left\{ \Delta e_X \, \dot{q}_{z_{X_\alpha}} + \Delta e_Y \, \dot{q}_{z_{Y_\alpha}} \right\}$$ Axial



View of rotating and whirling journal

For each individual flow region

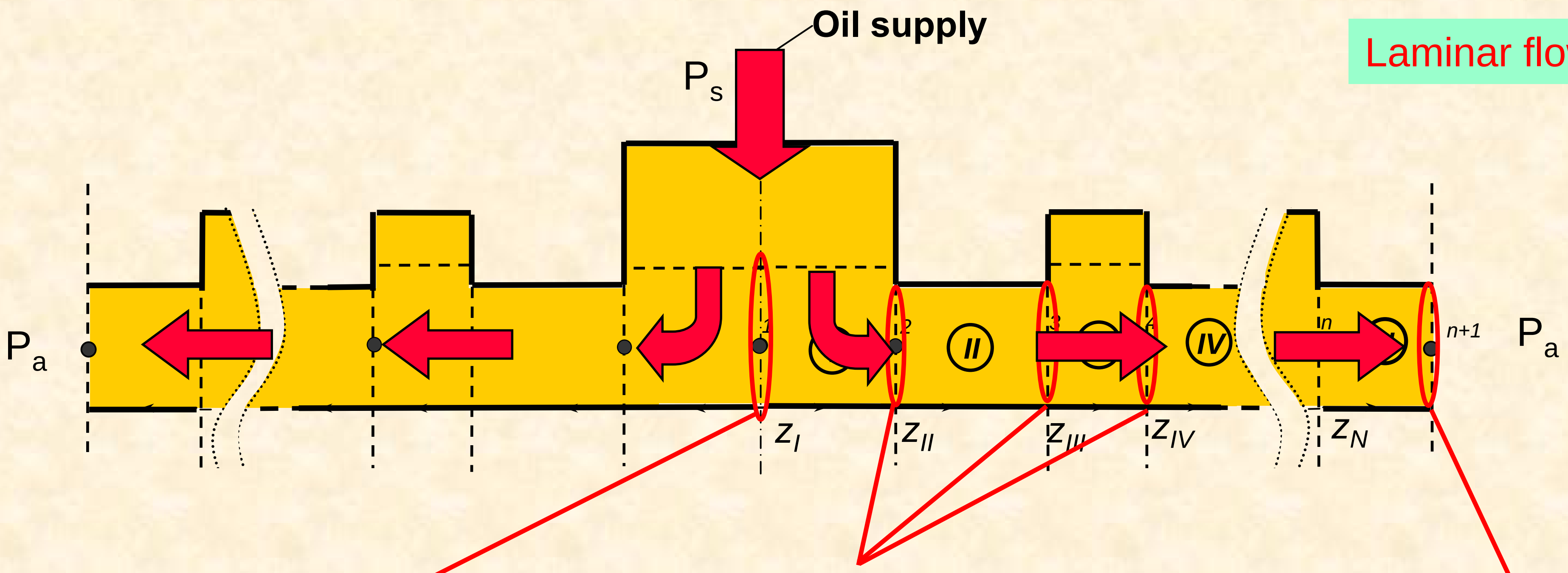
$$\alpha = I, II, \ldots N$$

Boundary conditions

Journal centered operation

P_s= supply pressure
P_a= ambient pressure

Laminar flow



Null axial flow rate (geometrical symmetry)

First-order pressures and axial flow rates must be equal

No generation of dynamic pressure

Pressure field (P_X)

Modified local squeeze film Reynolds number

$$\overline{\mathrm{Re}}_{*_\alpha} = \frac{\rho\omega c_{\eta_\alpha}^2}{12\mu}$$

Zeroth order (Static Pressure)

$$P_{0_\alpha} = a_\alpha + s_\alpha z_\alpha$$

First order (Dynamic Pressure)

$$P_{X_\alpha}(z) = f_{X_\alpha}(z_\alpha)\cos(\theta) + g_{X_\alpha}(z_\alpha)\sin(\theta)$$

First order solution

$$f_{X_\alpha}(z_\alpha) = c_{f_\alpha}\cosh\left(-{z_\alpha}/{R}\right) + s_{f_\alpha}\sinh\left({z_\alpha}/{R}\right) - 12i\frac{\mu\omega R^2}{c_{\eta_\alpha}^3}\left\{1 + i\,\overline{\mathrm{Re}}_{*_\alpha}\right\}$$

$$g_{X_\alpha}(z_\alpha) = c_{g_\alpha}\cosh\left(-{z_\alpha}/{R}\right) + s_{g_\alpha}\sinh\left({z_\alpha}/{R}\right) - \frac{6\mu\Omega R^2}{c_{\eta_\alpha}^3}$$

Force Coefficients $(c_{f_\alpha}, c_{g_\alpha}, s_{f_\alpha}, s_{g_\alpha})$

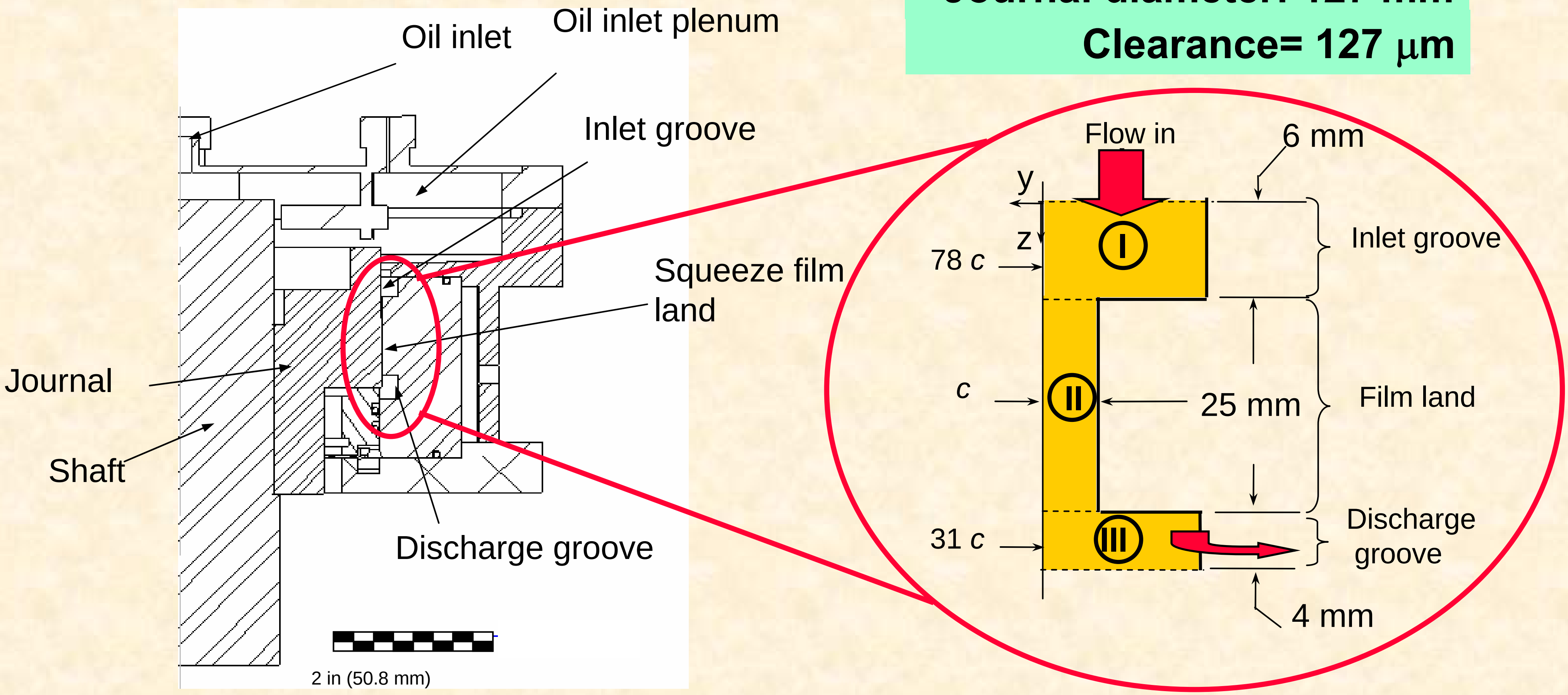
Obtained from boundary conditions

$$-\begin{bmatrix} K_{XX} - \omega^2 M_{XX} + i\omega C_{XX} \\ K_{YX} - \omega^2 M_{YX} + i\omega C_{YX} \end{bmatrix} = 2\sum_{\alpha=1}^{N}\int_0^{L_\alpha}\int_0^{2\pi R}\left\{\begin{Bmatrix} f_{X_\alpha}(z_\alpha)\cos(\theta)^2 \\ g_{X_\alpha}(z_\alpha)\sin(\theta)^2 \end{Bmatrix} dx\right\} dz_\alpha = 2\sum_{\alpha=1}^{N}\begin{Bmatrix} \pi R\int_0^{L_\alpha} f_{X_\alpha}(z_\alpha)\,dz_\alpha \\ \pi R\int_0^{L_\alpha} g_{X_\alpha}(z_\alpha)\,dz_\alpha \end{Bmatrix}$$

Test Squeeze film damper

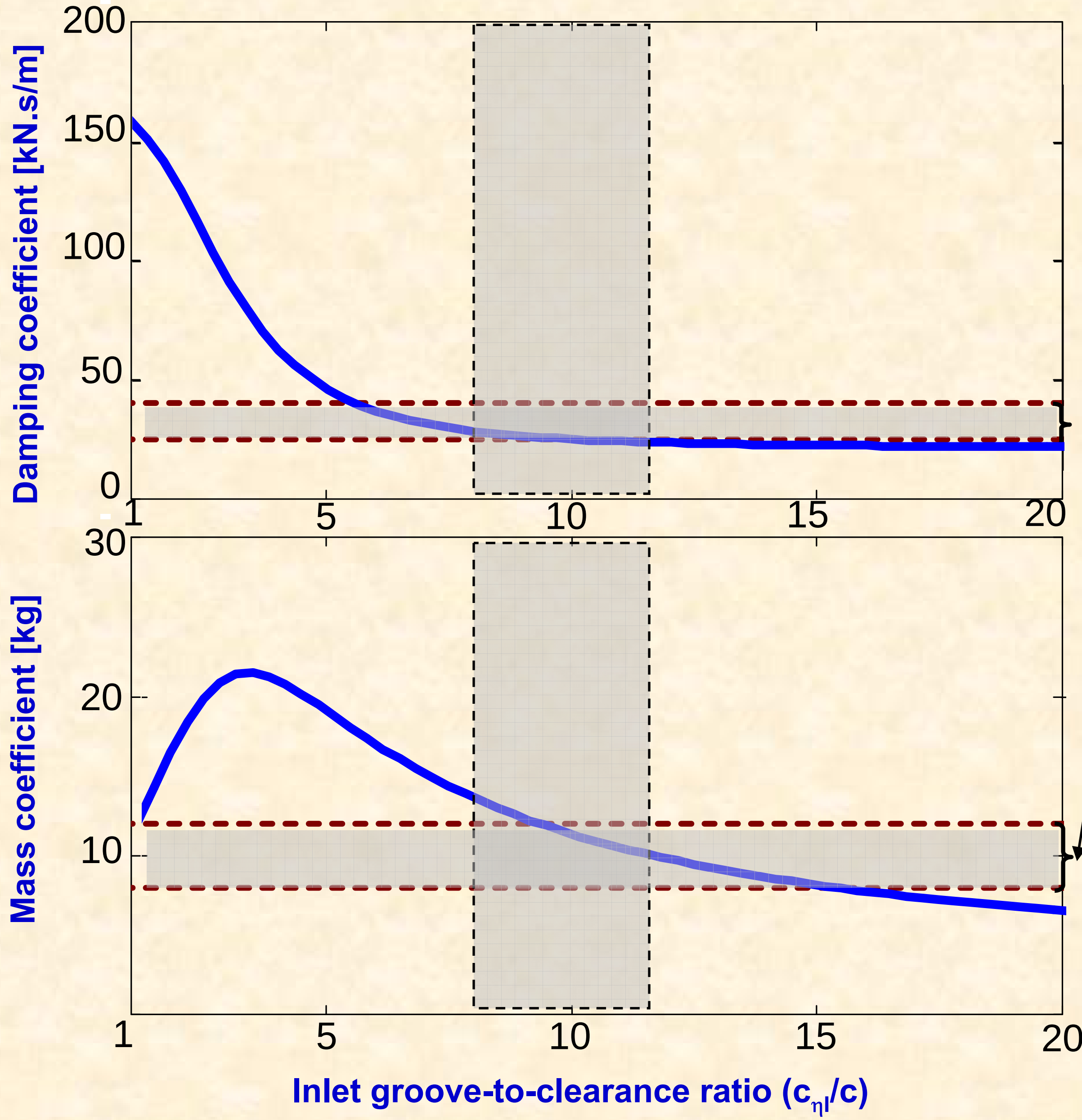
Journal diameter: 127 mm

Clearance= 127 μm



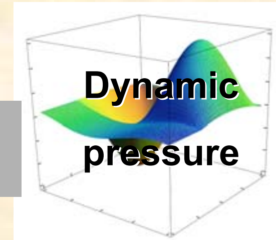
Sealed-end SFD assembly cut view.

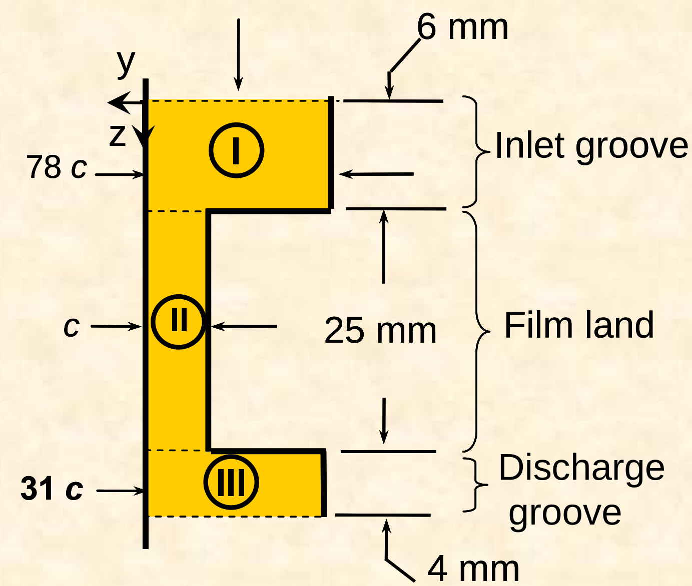
Grooved SFD



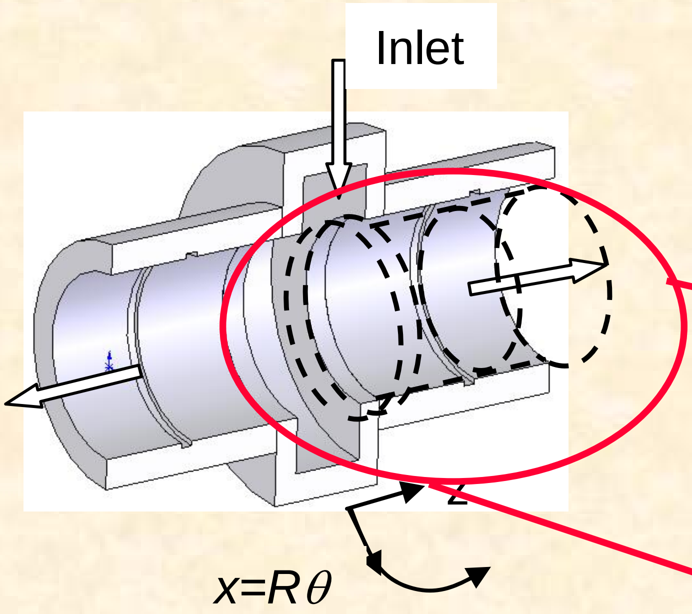
Good correlation with experimental results for both damping and added mass coefficients

Experiments



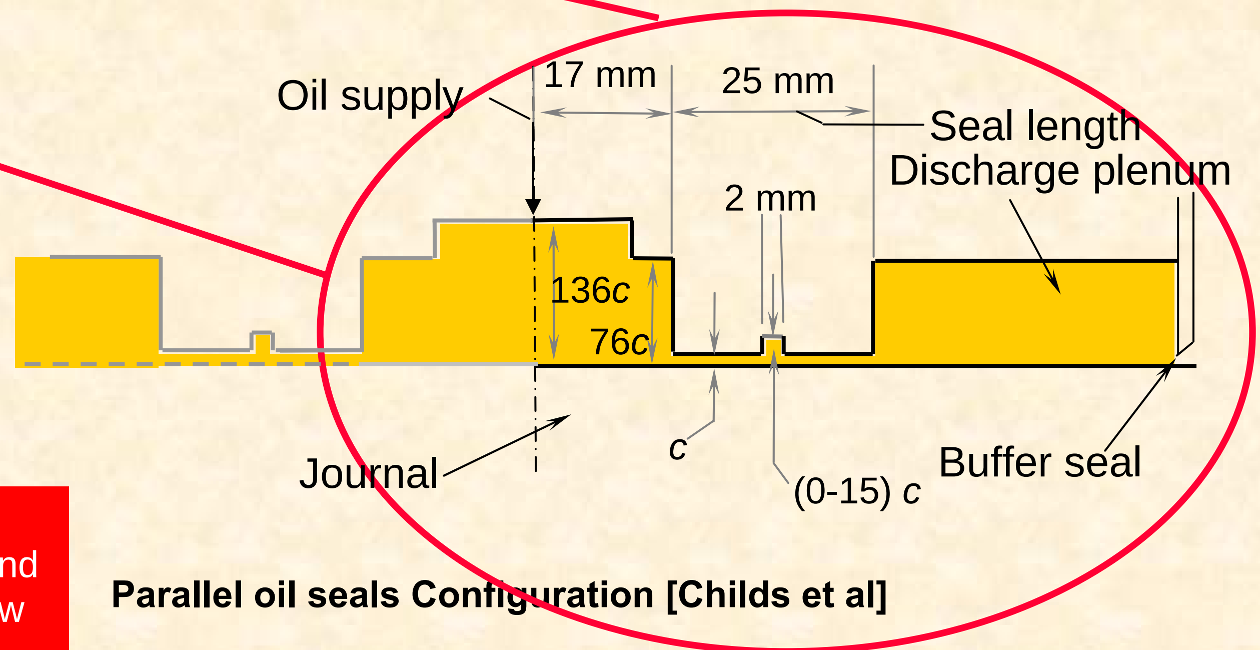


Test grooved oil seal



Clearance= 86 μm

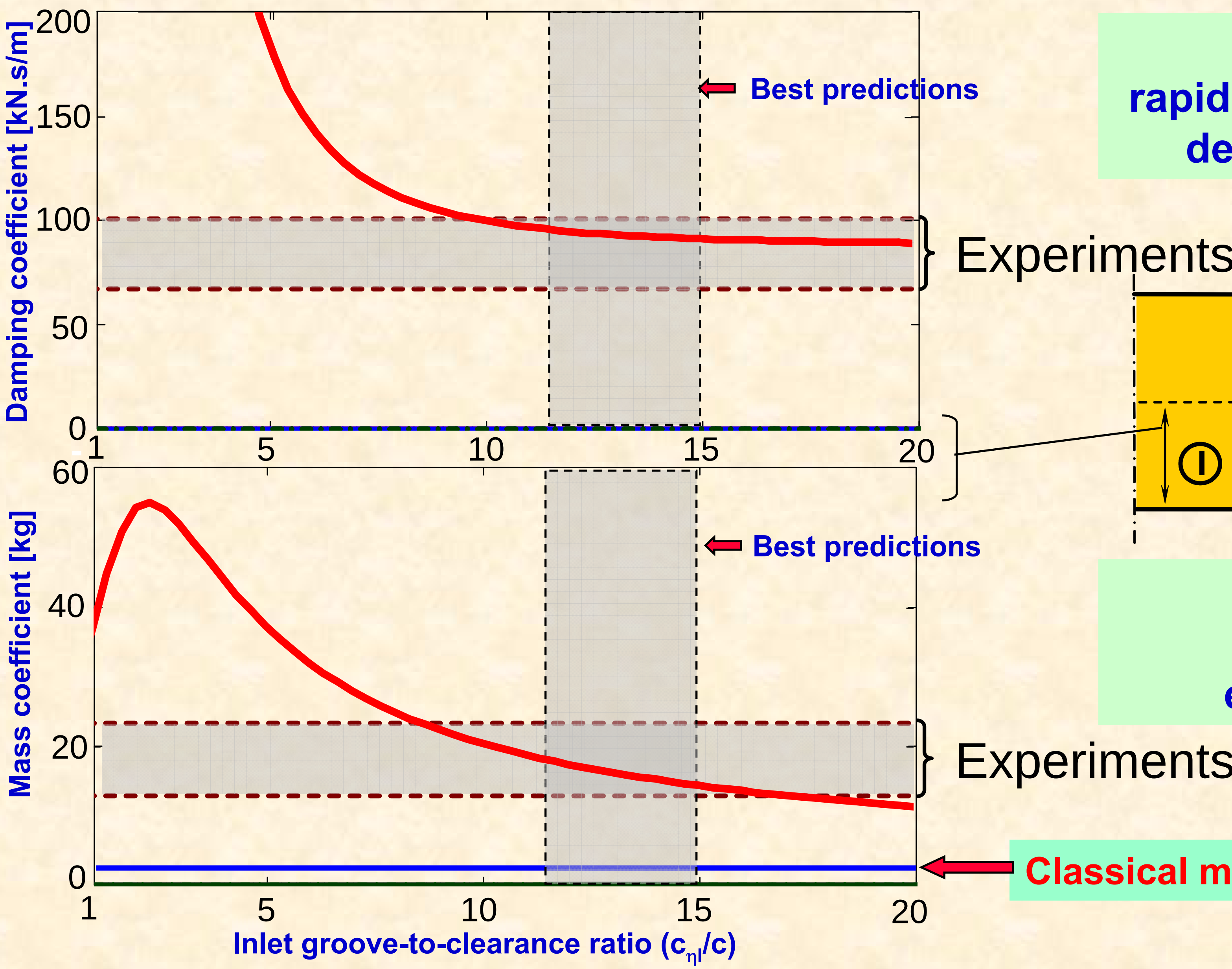
Journal diameter: 117 mm



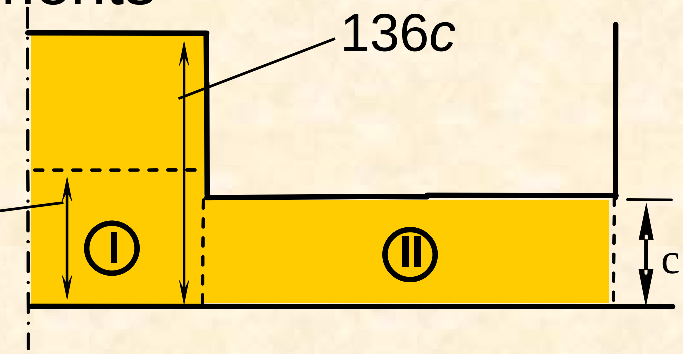
Parallel oil seals Configuration [Childs et al]

Childs, D. W., Graviss, M., and Rodriguez, L. E., 2007, "The Influence of Groove Size on the Static and Rotordynamic Characteristics of Short, Laminar-Flow Annular Seals," ASME J. Tribol, **129**(2), 398-406.

Smooth seal (inlet groove)



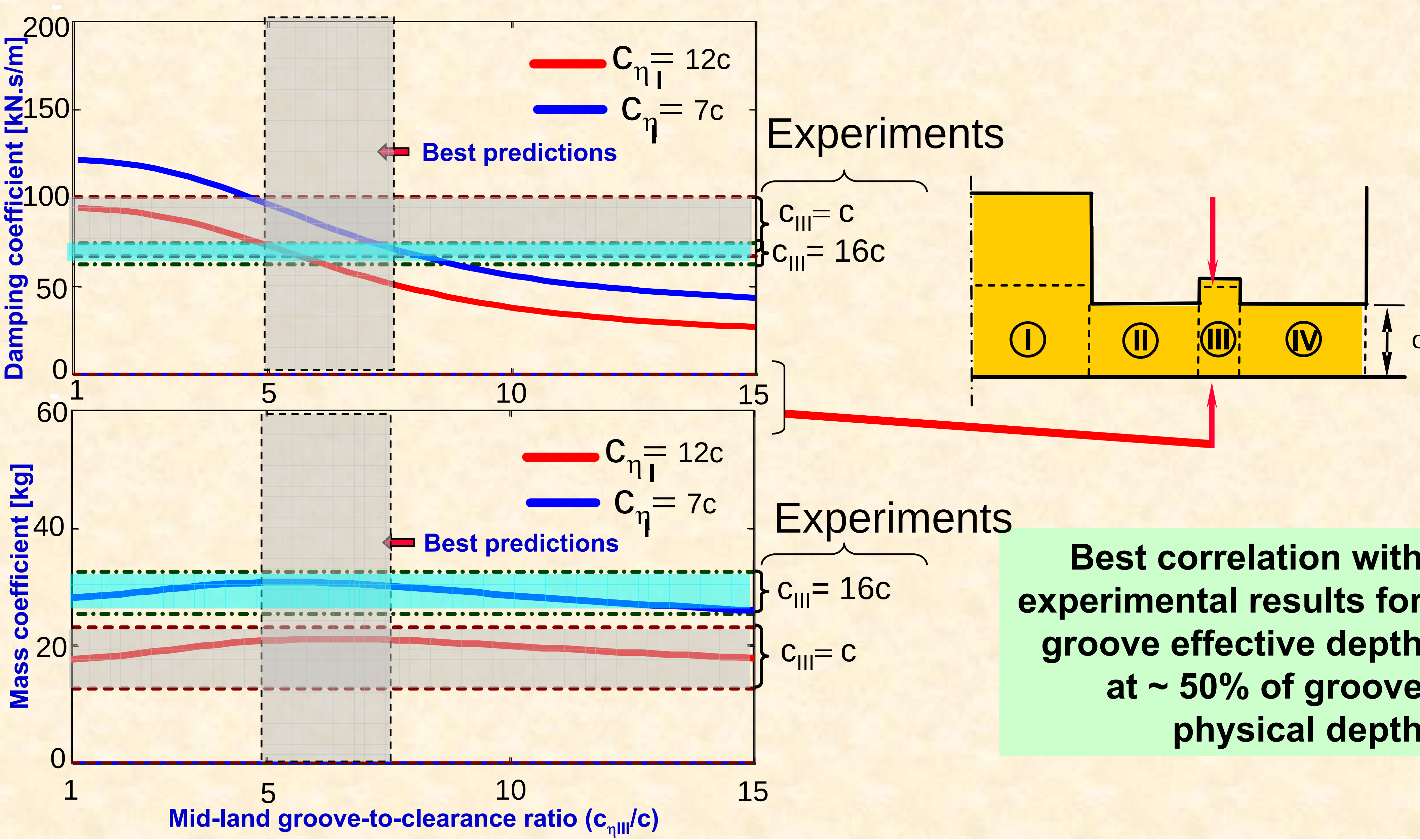
Damping decreases rapidly as groove effective depth increases ($\sim 1/c_\eta^3$)



Mass coefficient less sensitive to groove effective depth ($\sim 1/c_\eta$)

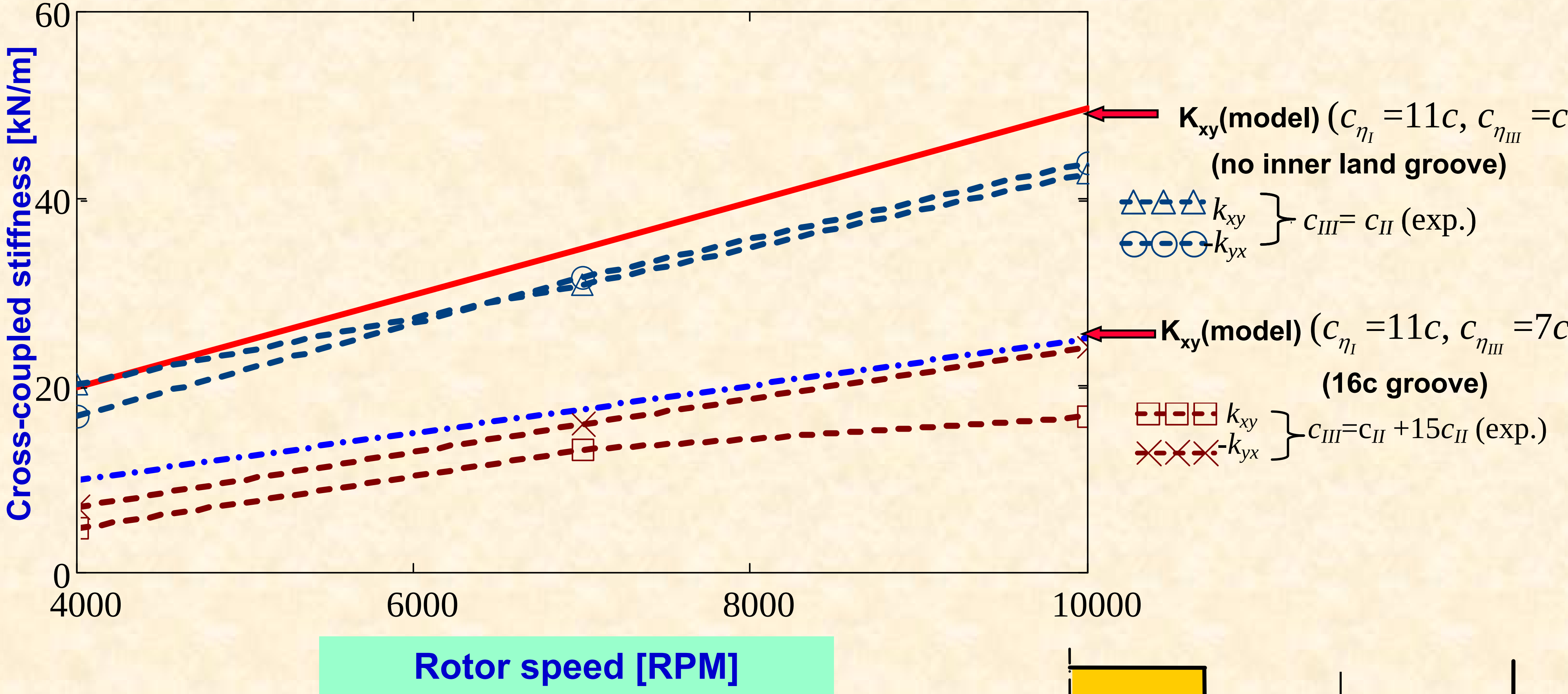
Classical model prediction

Grooved seal (mid or inner land groove)

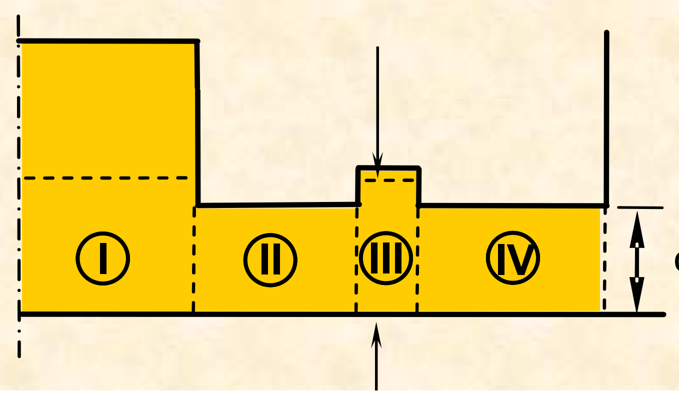


Best correlation with experimental results for groove effective depth at ~ 50% of groove physical depth

Oil seal (grooved and ungrooved)



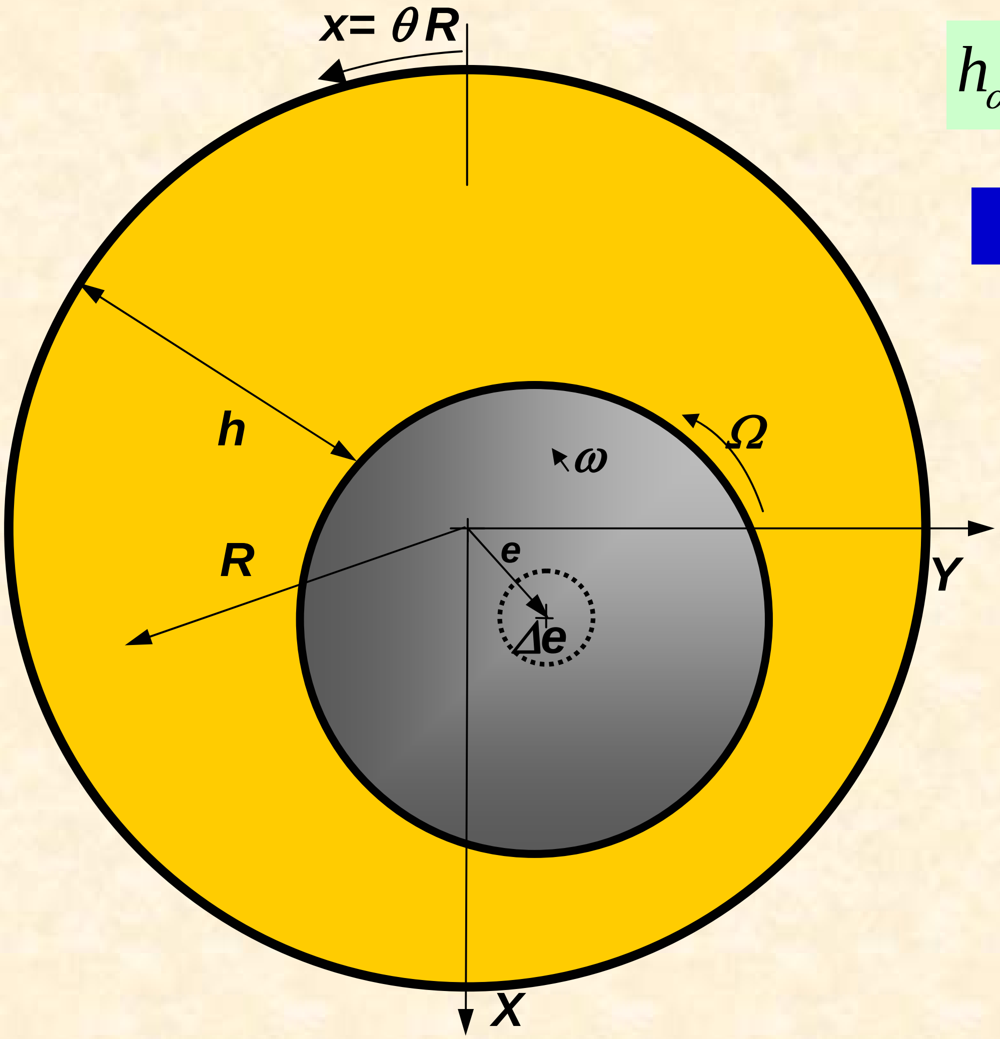
Model effectively predicts reduction of cross-coupled stiffness due to inner land groove.



Cross-coupled stiffness

Finite element solution

Off-centered journal operation



Film thickness

$$h_\alpha = h_{0_\alpha} + e^{i\omega t}\left\{ \Delta e_X \cos(\theta) + \Delta e_Y \sin(\theta) \right\}$$

Reynolds eqn. with temporal fluid inertia

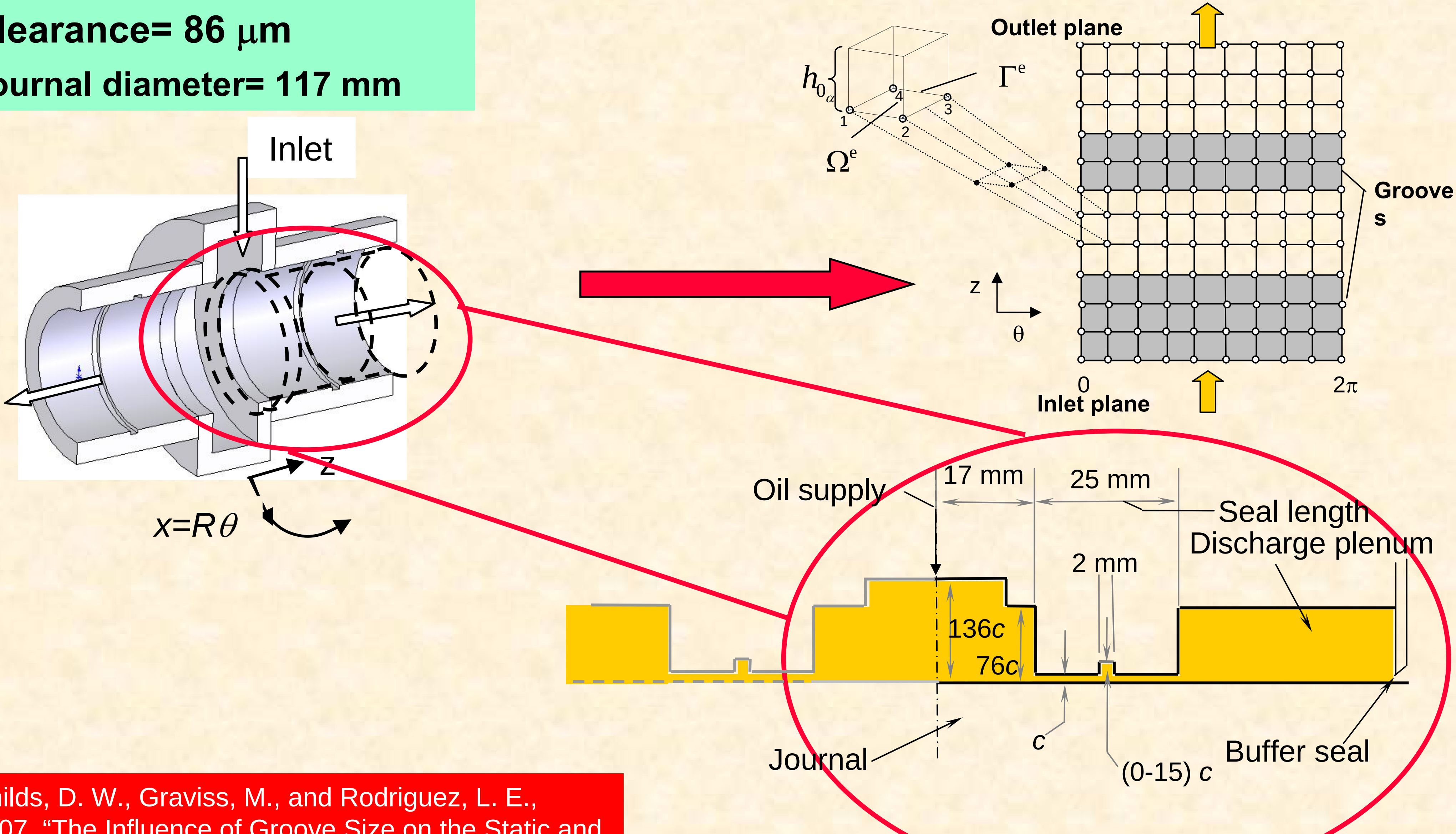
$$\frac{\partial}{\partial x}\left(h_\alpha{}^3 \frac{\partial P_\alpha}{\partial x} \right) + \frac{\partial}{\partial z_\alpha}\left(h_\alpha{}^3 \frac{\partial P_\alpha}{\partial z_\alpha} \right) =$$

$$12\mu \frac{\partial}{\partial t}(h_\alpha) + 6\mu R\Omega \frac{\partial}{\partial x}(h_\alpha) + \left(h_\alpha{}^2\right) \frac{\partial^2}{\partial t^2}(\rho h_\alpha)$$

Test grooved oil seal

Clearance= 86 μm

Journal diameter= 117 mm



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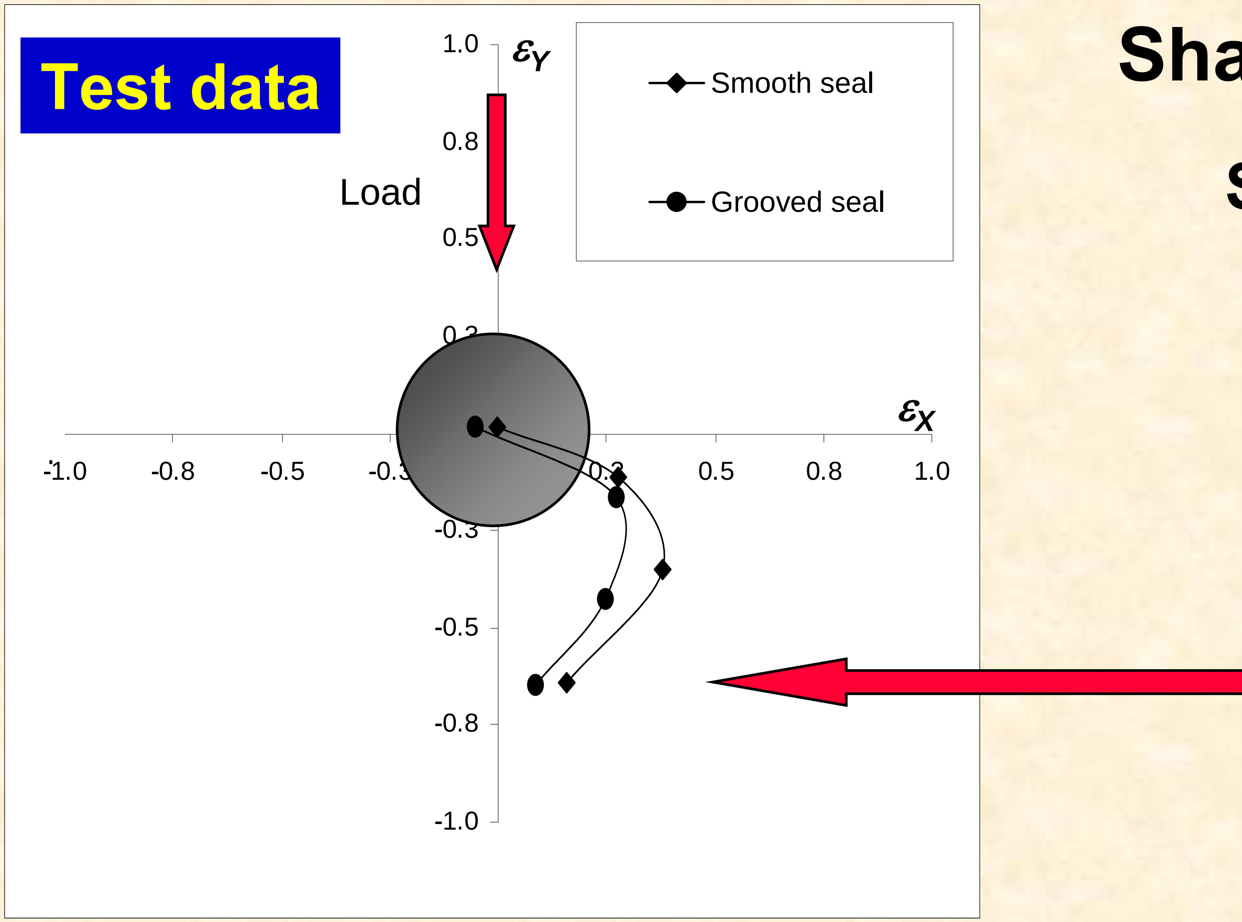
Seal operating conditions



Journal locus

Supply pressure: 70 bar

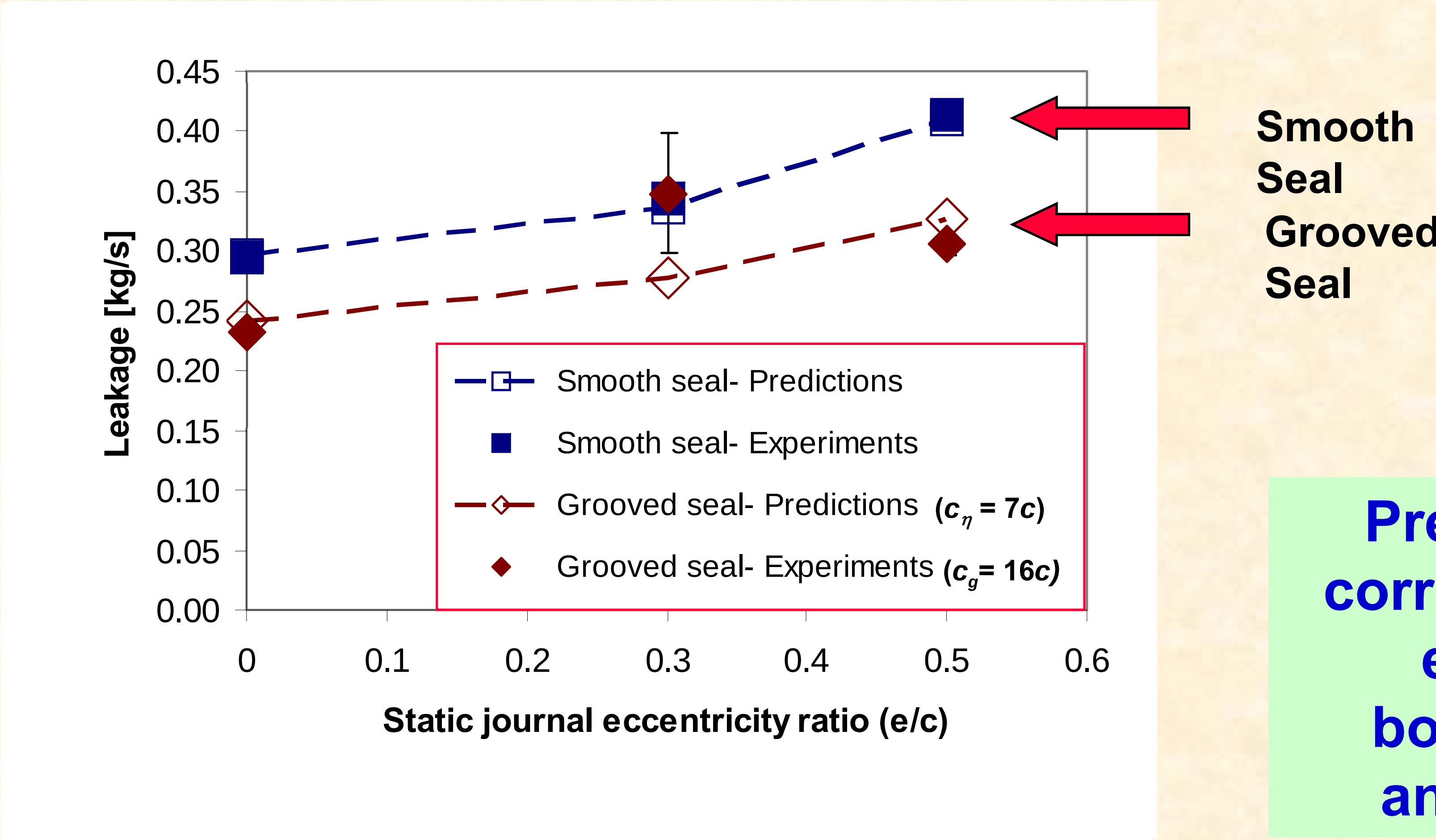
Shaft speed: 10,000 rpm

Static eccentricity ratio:
0, 0.3, 0.5, 0.7

Journal center locus indicates seal operates with oil cavitation at the largest test eccentricities

Seal Leakage

10,000 rpm, 70 bar

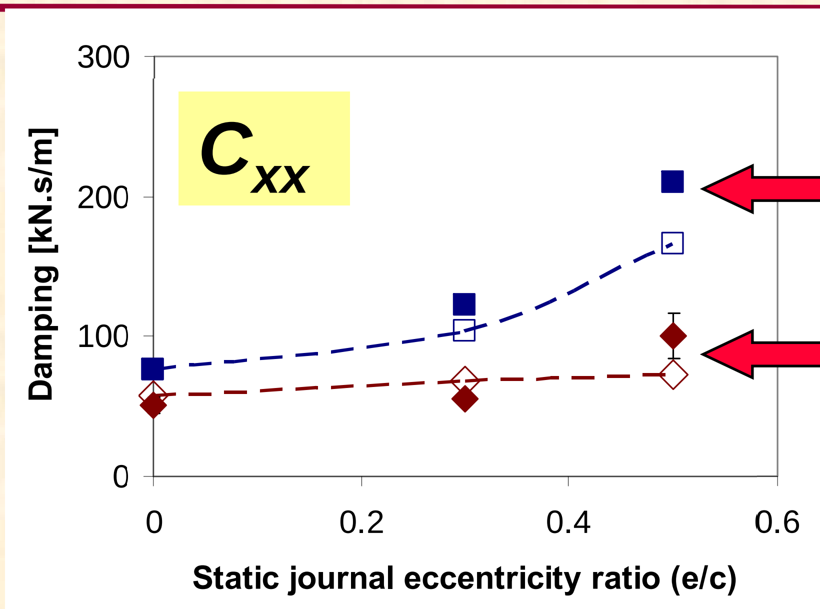


Predicted leakage correlates well with experiments for both smooth land and grooved seal

Direct damping

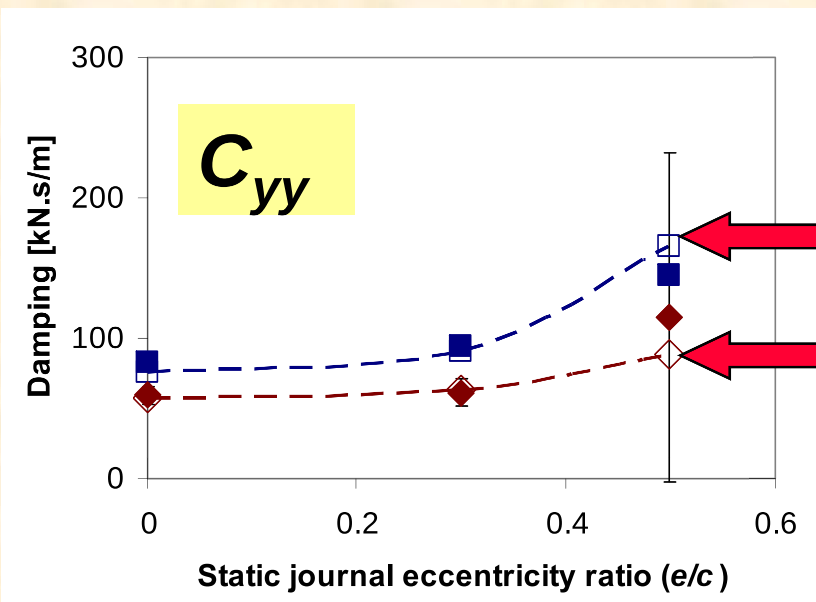
10,000 rpm, 70 bar

Model predicts accurately reduction in direct damping due to inner land groove.



Smooth Seal

Grooved Seal



Smooth Seal

Grooved Seal

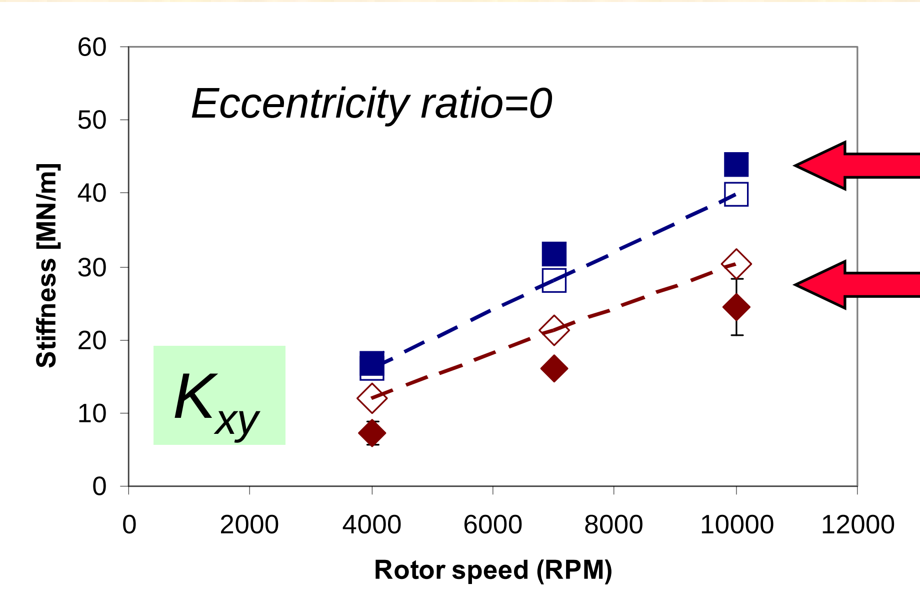
Smooth seal- Predictions

Smooth seal- Experiments

Grooved seal- Predictions **(c_η = 7*c*)**

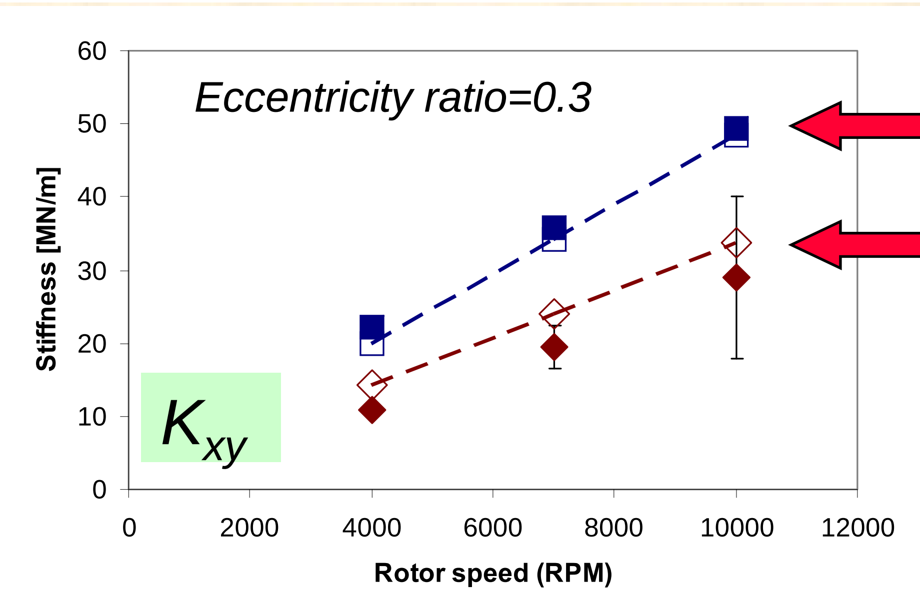
Grooved seal- Experiments **(c_g= 16*c*)**

Cross-coupled stiffness



Smooth Seal

Grooved Seal



Smooth Seal

Grooved Seal

70 bar

ε = 0, 0.3

Model effectively predicts reduction in cross-coupled stiffness due to mid-land groove.

Smooth seal- Predictions

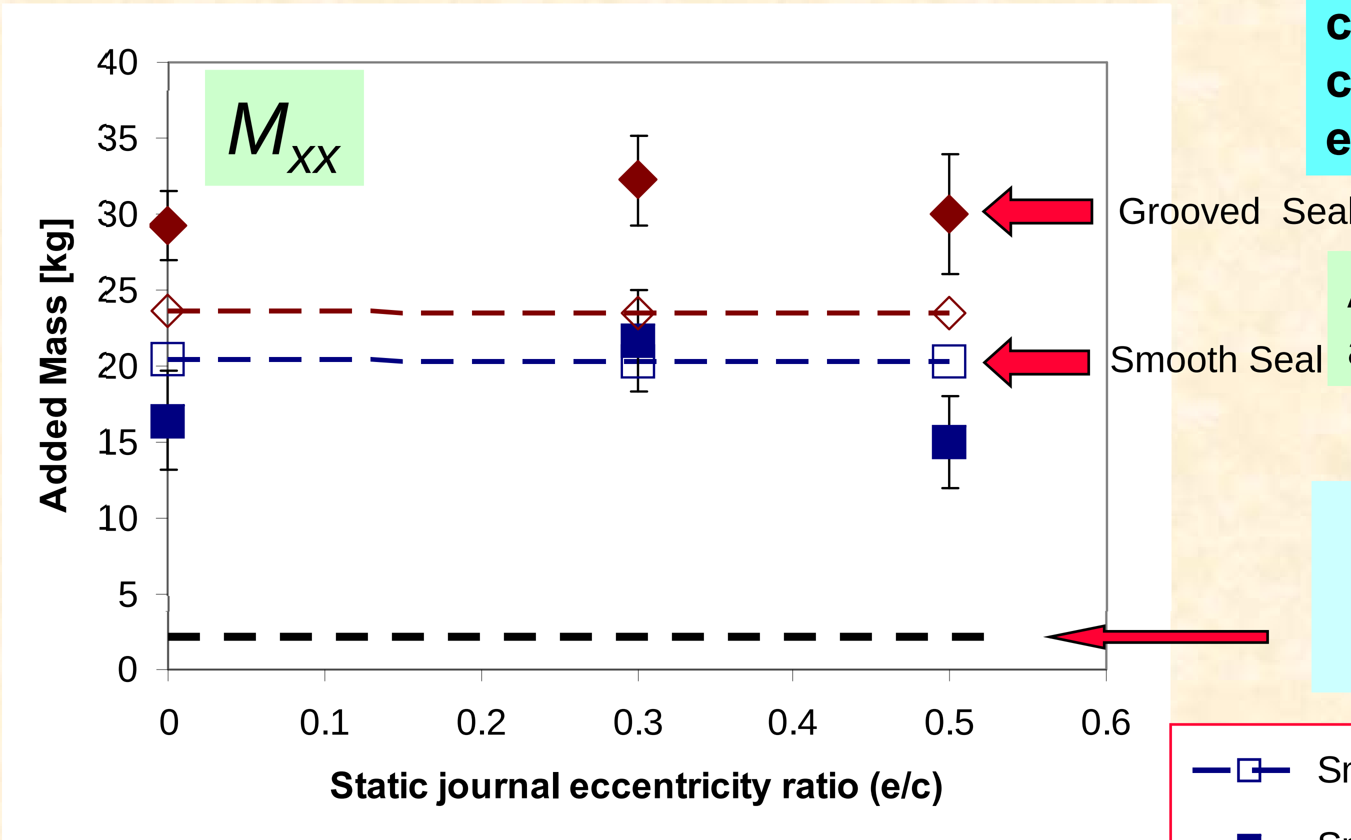
Smooth seal- Experiments

Grooved seal- Predictions **(c_η = 7c)**

Grooved seal- Experiments **(c_g= 16c)**

Added mass

10,000 rpm, 70 bar



Experimental data shows relatively large added mass coefficients. Predictions correlate well with experimental results.

Added mass coefficients are larger for grooved seal

Classical theory [1] predicts ~ 1/10 of test value

- Smooth seal- Predictions
- Smooth seal- Experiments
- Grooved seal- Predictions **(c_η = 7*c*)**
- Grooved seal- Experiments **(c_g= 16*c*)**

[1] Reinhardt, F., and Lund, J. W., 1975, "The Influence of Fluid Inertia on the Dynamic Properties of Journal Bearings," ASME J. Lubr. Technol., **97**(1), pp. 154-167.

Conclusions:

- **Damping and cross-coupled stiffness decrease rapidly as the effective groove depth increases(~$1/c^3$).Added mass coefficients less sensitive to effective depth (~$1/c$).**
- **Predicted force coefficients (*K,C,M*) correlate well with experimental data for a narrow range of groove effective depths**
- **For shallow and short mid-land groove (oil seal) predicted (*K,C,M*) correlate best with test data when using 50% of actual groove depth.**
- **In oil seals, an inner land groove does not uncouple adjacent film lands!!**

Conclusions (cont):

- The dynamic pressure field in deep grooves may be difficult to measure for most practical excitation frequency ranges. (fluid inertia pressures are proportional to ω^2).
- Deep grooves do generate dynamic pressures of mainly inertial nature, which lead to large added mass coefficients.
- There is a specific groove depth where the added mass coefficient peaks. This groove depth value for the studied configurations is < 10c.
- Force coefficients in test configurations, like SFDs and oil seals, are a function of the ancillary geometries like feeding/discharge arrangements.