The oil-air interface problem of fluid dynamic bearings in hard disk drives

Ferdi Hendriks

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Introduction to FDBs for HDDs

- **Fluid bearings have several advantages over ball bearings in spindles for modern hard disk drives.**
  - Quietness
  - Very low non-repeatable runout
  - Shock resistance

- **Well known issues are:**
  - Oil-air interface instability (Asada et al.)
  - Bubble ingestion (Asada et al.)
  - Leakage (Muijderman, Bootsma, Tielemans)
    - They used a homogenized Reynolds eqn (8 = infinity)
  - Numerical simulation of free boundary problem

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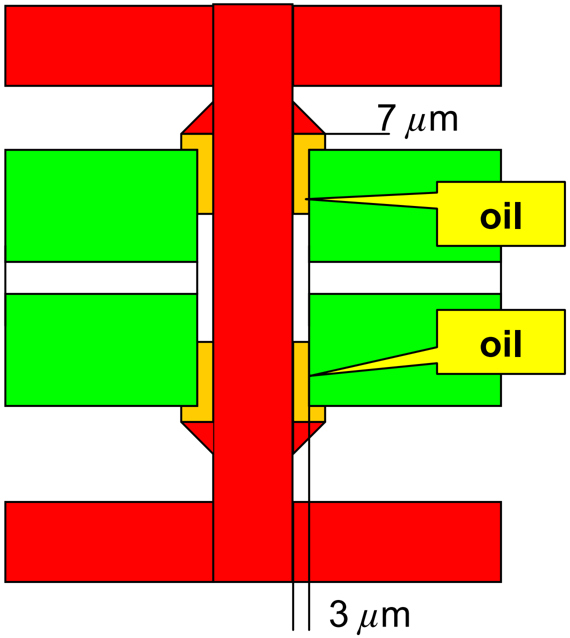
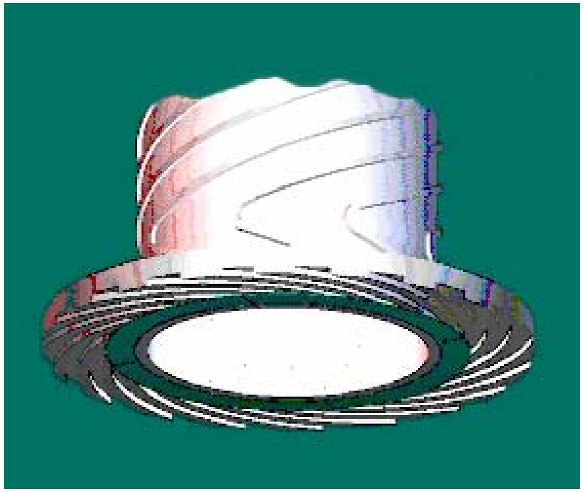


FDB of "stationary shaft design"



FDB's

N.V. Philips:

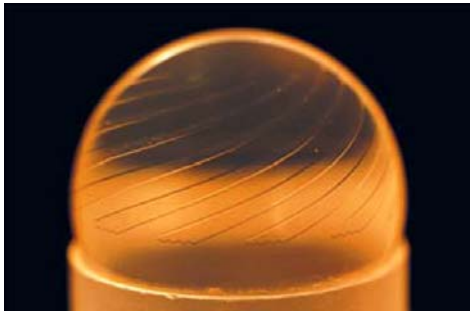
Evert Muijderman

Jan Bootsma

J. Tielemans

Ultracentrifuges 60 krpm,

Later HDDs



Stefan Risse

?

The Art of the Millstones, How They Work
by
Theodore R. Hazen

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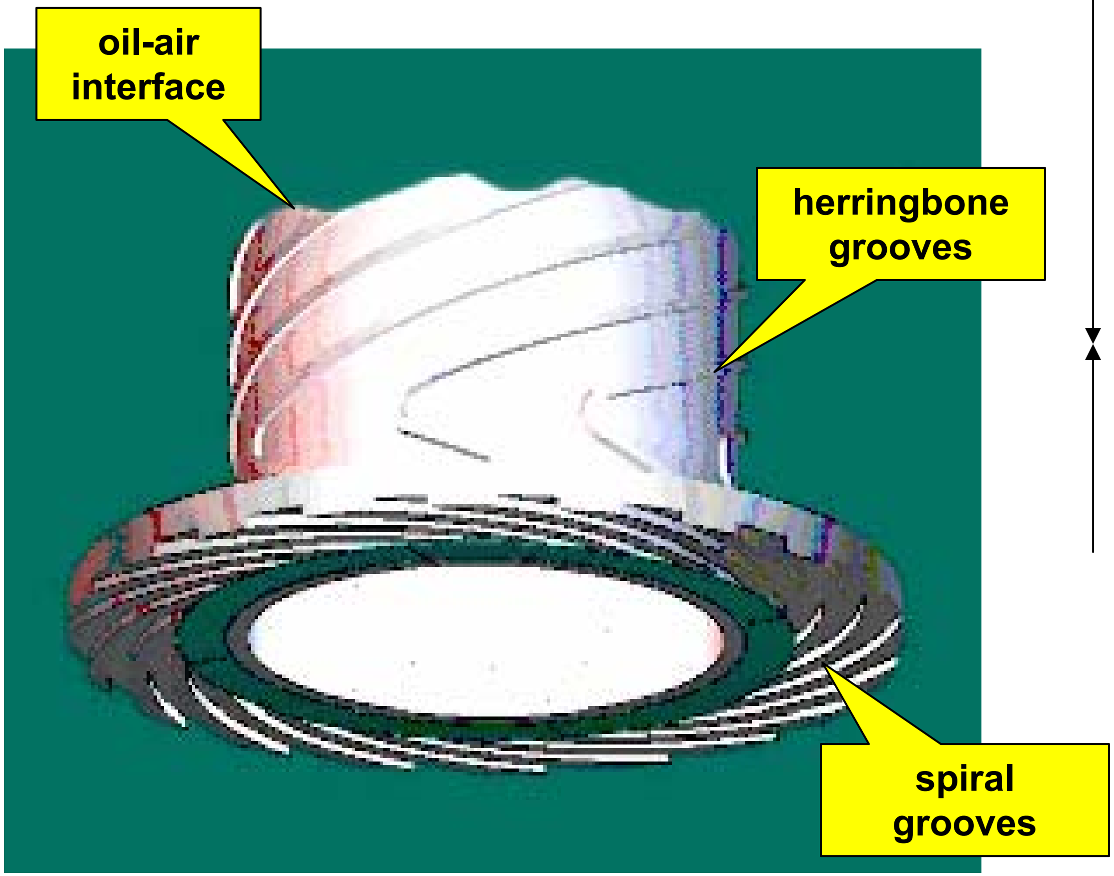
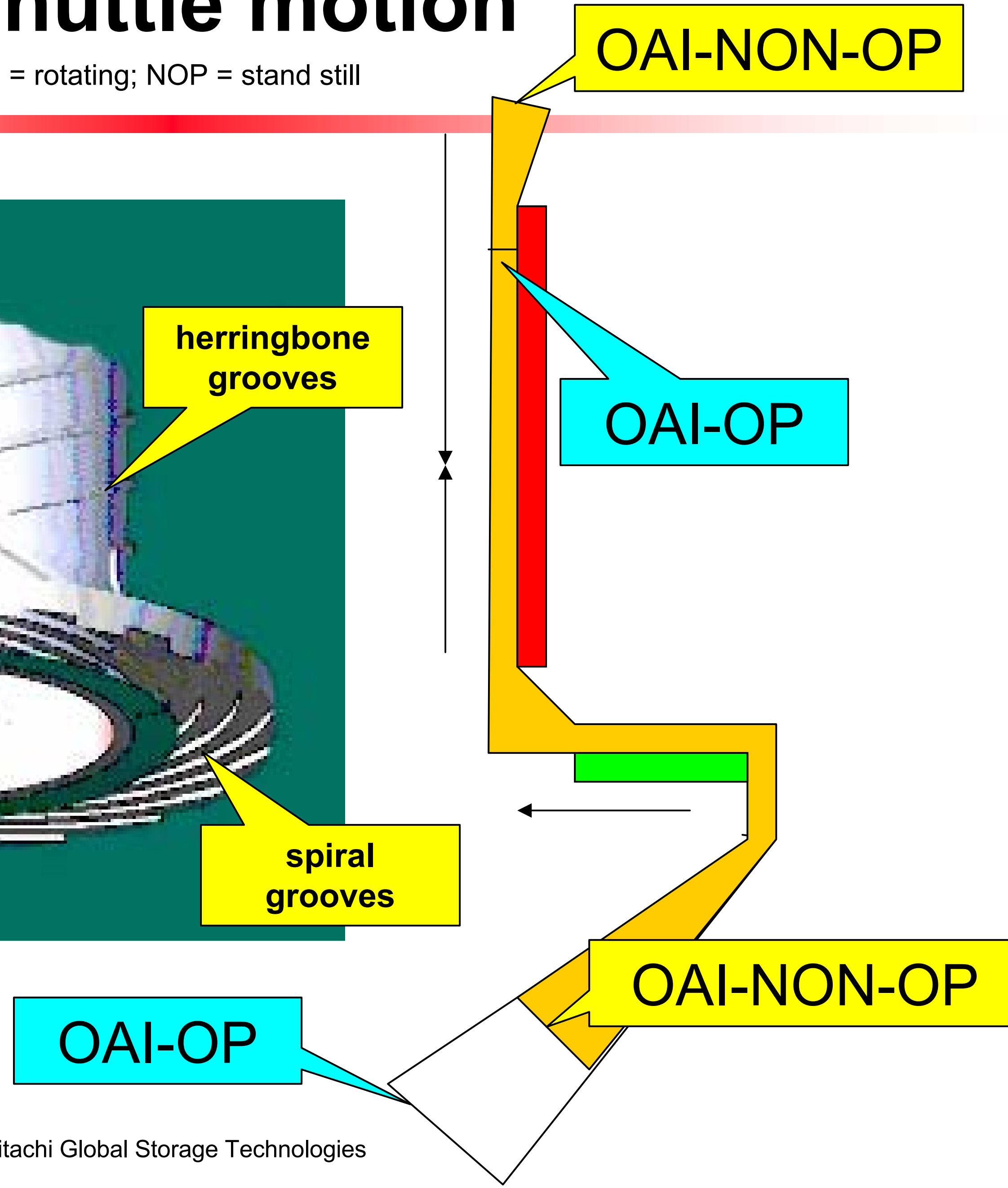
Oil shuttle motion

OP = rotating; NOP = stand still





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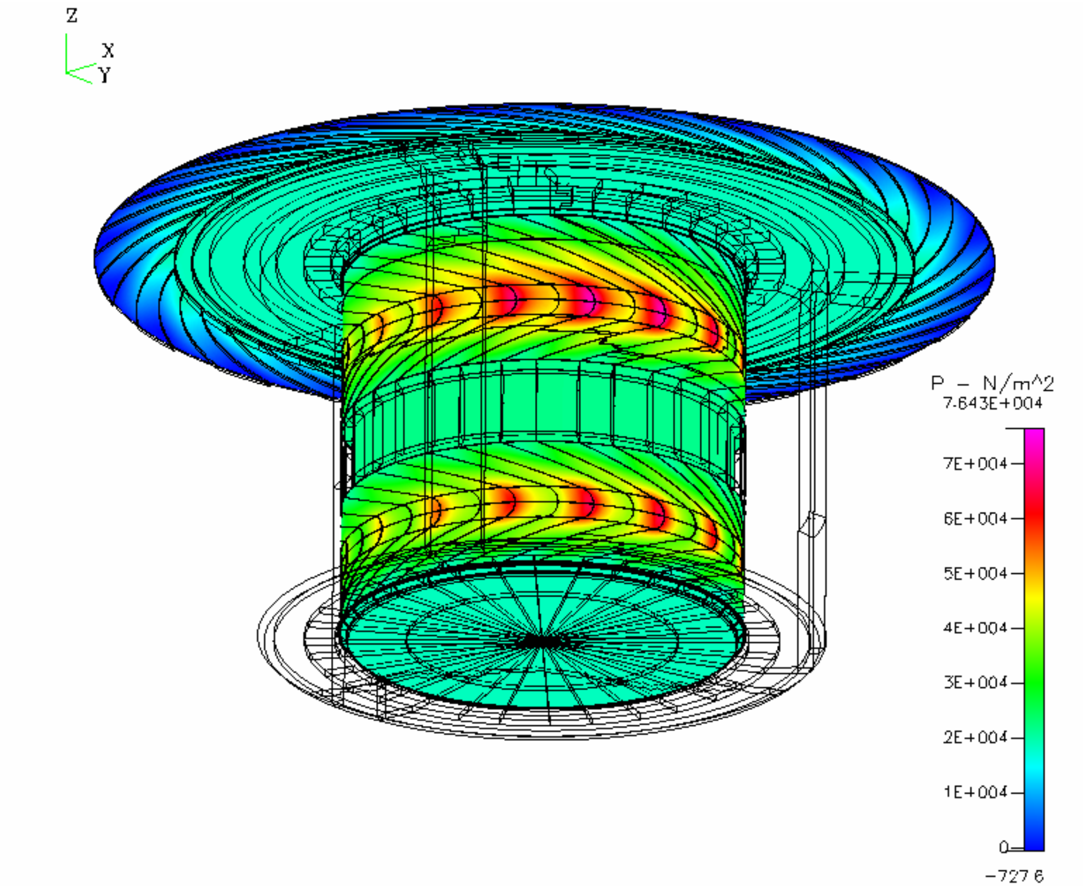
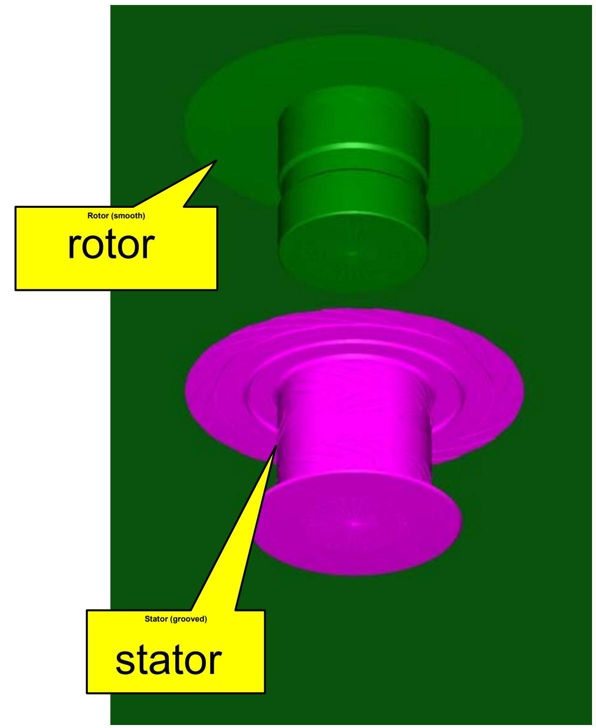
FDB's

Pressure distribution in Hitachi's microdrive



Ca. 1 million cells



FDB of rotating shaft design

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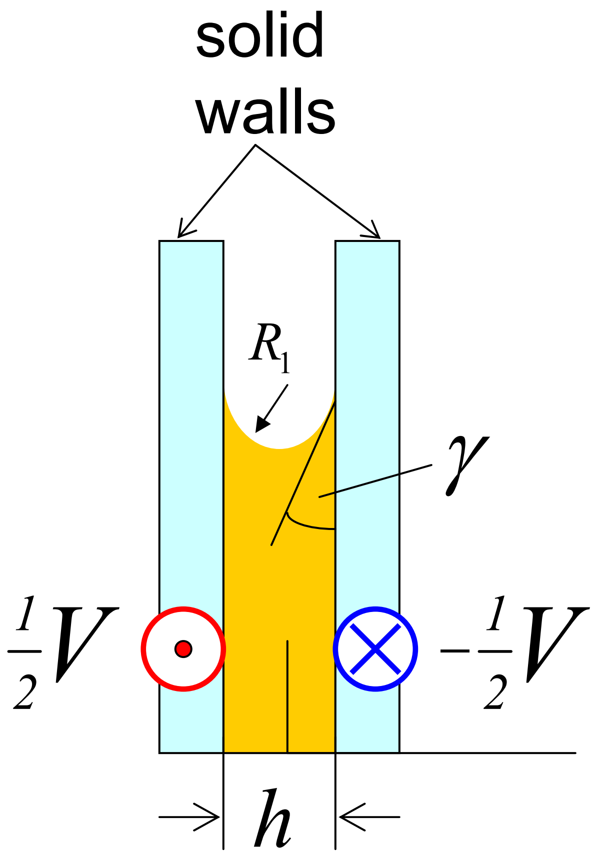
Last year's problem – MPI 2004

Capillary Couette flow with constant gap – is it stable?



$$Reynolds\ no:\quad Re = \frac{\rho V h}{\mu} \quad << 1$$

$$Capillary\ no:\quad Ca = \mu V / \sigma \quad O(1)$$

$$Wetting\ angle:\ \gamma \quad < 90^0$$

neglect gravity (Bond number)

neglect flow field in the air ($\mu_{oil} >> \mu_{air}$)

$$\frac{\partial \rho h}{\partial t} + \frac{1}{2}\frac{\partial \rho h U}{\partial x} = \frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right)$$

BC:

$$p_\Gamma = \sigma\left(\frac{1}{R_1} + \frac{1}{R_2}\right) \quad \textit{and tangential stress condition}$$

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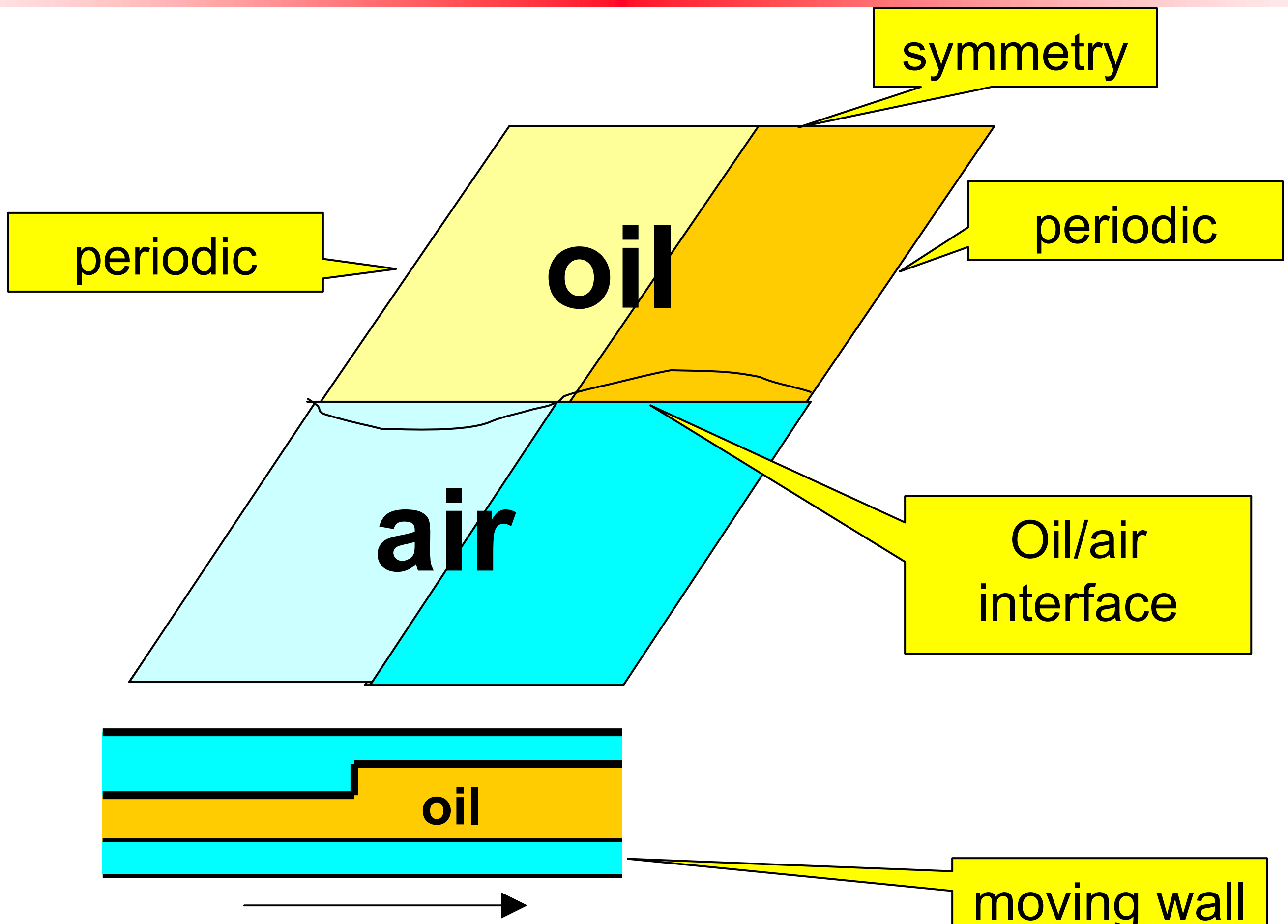
Average OAI deflection in a land-groove geometry with symmetry



State of the problem after 1 yr

- We know that capillary interfaces in land-groove geometries tend to form fingers in the grooves. The oil film rises over land regions.
- We know that the number of grooves plays a crucial role in oil-air interface deflection (analytical result)
- We do not know the stability as a function of Ca, Re and groove parameters: land/groove ratio and groove depth/clearance ratio.
- We know that averaging of the capillary interface across the fluid film is not (really) allowed. This is especially true in the grooves. I.e. There is no such thing as "the interface deflection."

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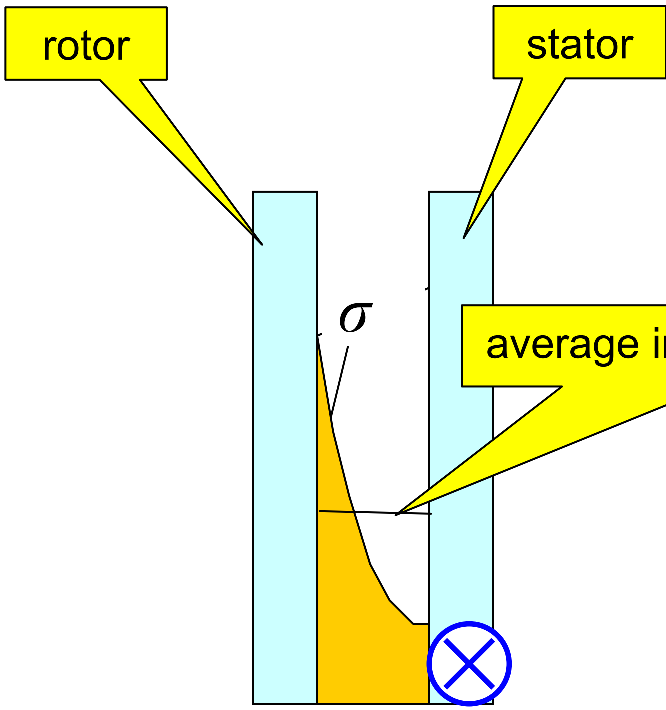
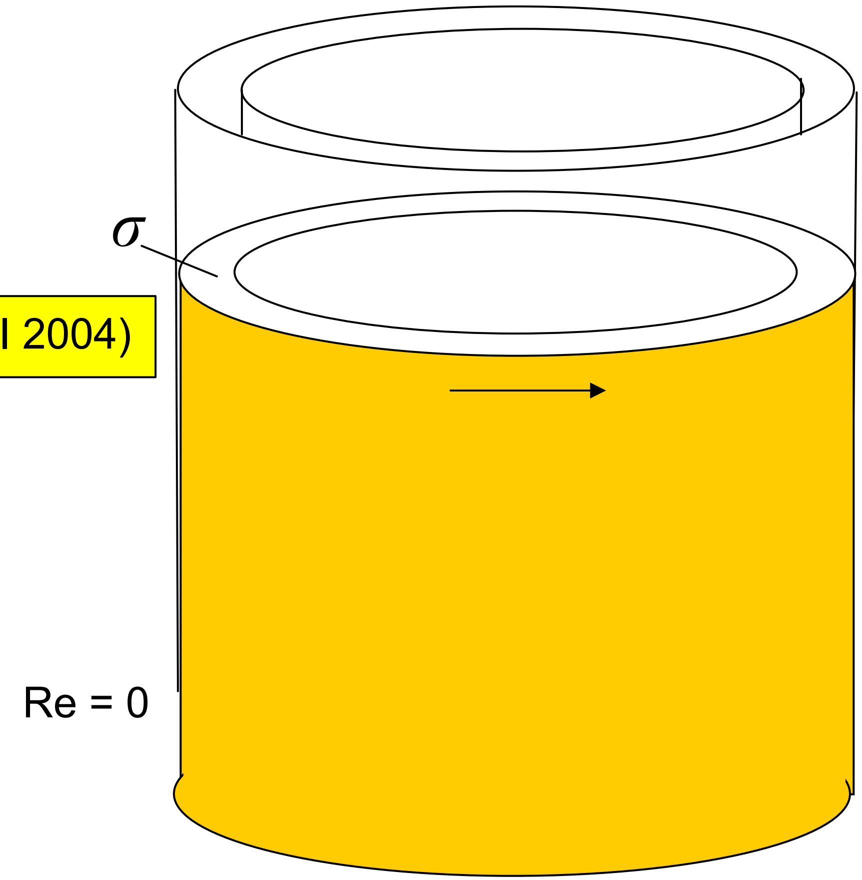
Oil film on rotor / Capillary Taylor- Couette flow



interface shape in the groove



MPI 2005 questions

- **Describe the capillary interface in a land-groove flow field. Relax (or drop) the averaging assumption.**
- **Investigate the stability of capillary Taylor-Couette flow**
  - Use average interface;
  - Eccentric, if the centric case is trivial.
- **Does one need to know the flow near the capillary interface to predict when bubble ingestion occurs?**
- **Does one need to know the detailed flow near the capillary interface to compute loads and torque of the bearing with "engineering precision"**

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Intermag presentation

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Main observation

- **Current HDDs use self-acting spiral groove and herringbone fluid dynamic bearings (FDBs) to achieve precise rotation of a disk pack.**

- **In some FDB ("self-sealing") designs oil-air interfaces occur.**

- **Oil-air interfaces become unstable under certain high stress conditions, expressed by**

  – The Capillary number — viscous stress / capillary pressure
  – The Reynolds number — inertial stress / viscous stress
  – The fractional eccentricity — eccentricity / clearance

- **Reynolds eqn with Half-Sommerfeld (Gümbel) or Reynolds BCs is not satisfactory to describe the oil / interface dynamics: Oil is not conserved.**

- **Modified "true cavitation" approaches are also problematic.**

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Bootsma and Tielemans' work

- In 1977 Bootsma and Tielemans already suggested that the stability of the oil-air interface involves the Capillary number and the Weber Number. Because

$$We = Ca\,Re \qquad \text{(we care!)}$$

this is equivalent to involvement of the Reynolds number

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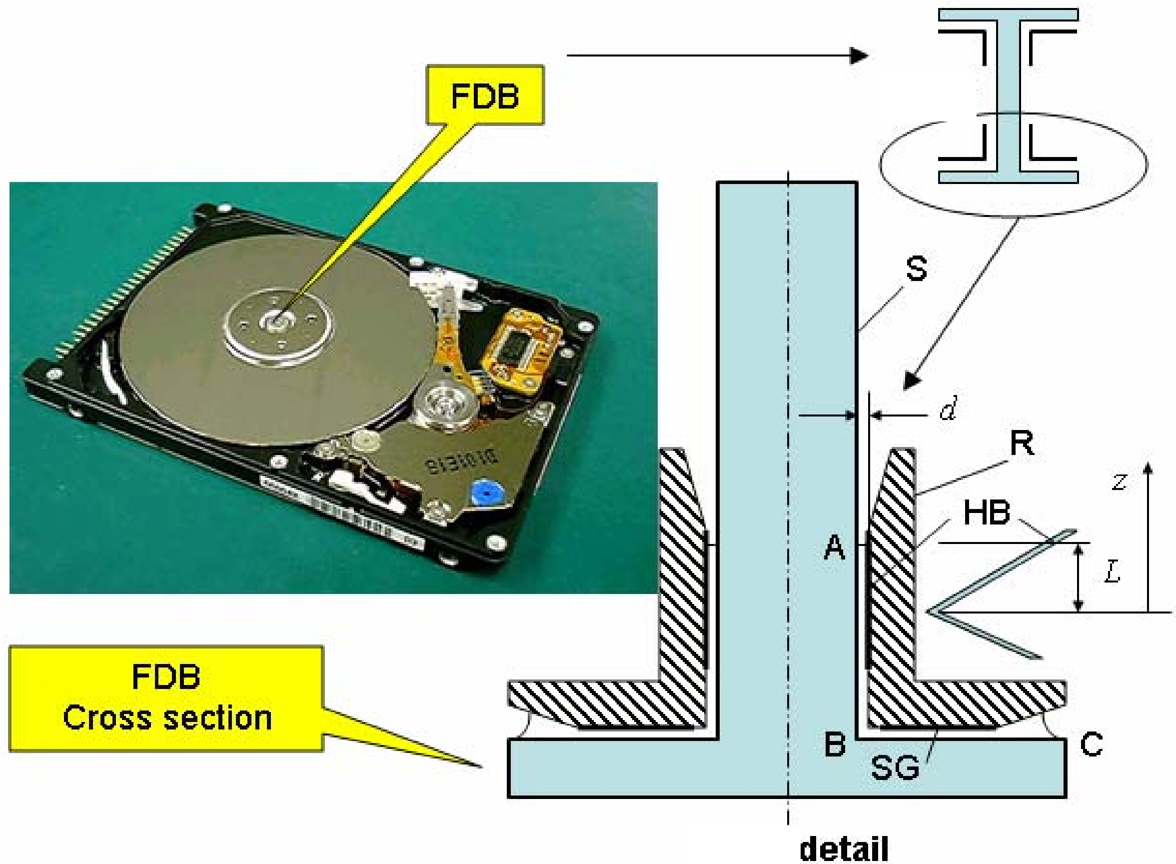
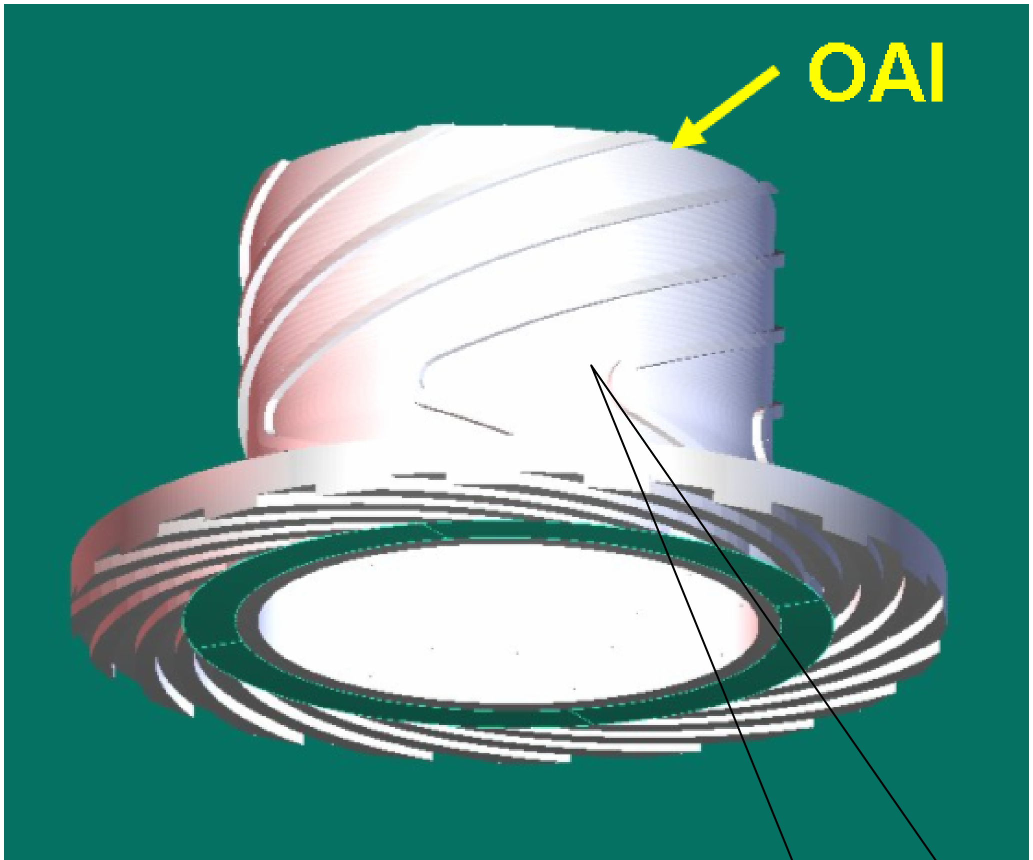
Fluid bearing with stationary shaft



Lower spool of the bearing of stationary shaft design



The oil-air interface (OAI) is located among the rotating herringbone grooves.

We wish to determine its evolution

$Z(\theta,t)$

This is a CAD model of the liquid! Not the solid.

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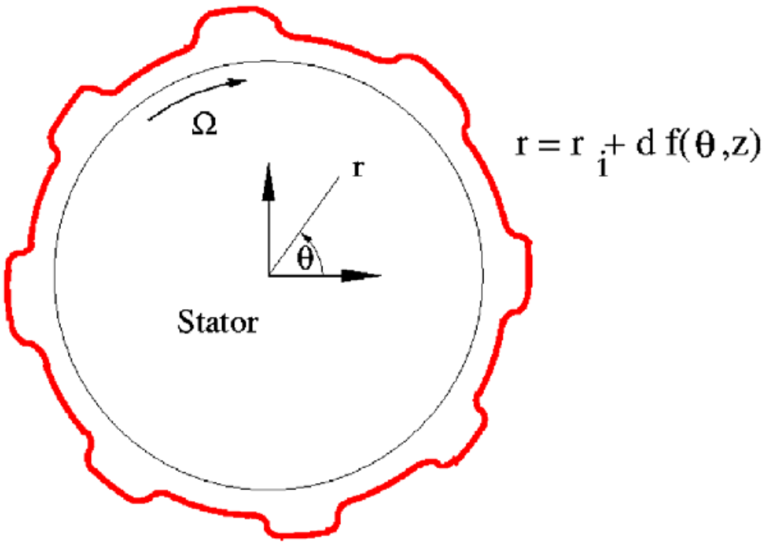
Groove-fixed (rotating) coordinate system



r_i : *inner (shaft) radius*

d : *clearance*

$f(\theta, z)$: *groove profile*

we do not consider eccentricity, rather, we are focused on the details of a single land/groove pair

Continuity / Navier-Stokes / Interface

$$\nabla . \mathbf{u}^* = 0 \qquad (1)$$

$$\rho \frac{D\mathbf{u}^*}{Dt} = -\nabla p^* + \mu \nabla^2 \mathbf{u}^* - 2\rho\Omega\, \hat{\mathbf{k}} \times \mathbf{u}^* - \rho \Omega^2 r \hat{\mathbf{r}} \, . \qquad (2)$$

$$\frac{\partial Z^*}{\partial t} - \mathbf{u}^* . \mathbf{n} = 0 \qquad (3)$$

$$\sigma \kappa^* = p^* - \hat{\mathbf{n}} . \mathbf{T}^* . \hat{\mathbf{n}} \qquad (4)$$

$$\left[\hat{\mathbf{n}} . \mathbf{T}^* \right] \times \hat{\mathbf{n}} = \mathbf{0} . \qquad (5)$$

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Reynolds' eqn / compact OAI

$$\frac{\partial}{\partial \theta}\left\{ f^3 \frac{\partial p}{\partial \theta} + 6f \right\} + \frac{\partial}{\partial z}\left\{ f^3 \frac{\partial p}{\partial z} \right\} = 0 \qquad (6)$$

<u>Averaging</u> the flow velocity at the interface, we obtain the evolution equation of the interface:

$$\frac{\partial Z}{\partial t} + \bar{u}_\theta \frac{\partial Z}{\partial \theta} - \bar{u}_z = 0 \qquad (7)$$

<u>This relies on the existence of a single, compact oil-air interface. The average velocity at the OAI is</u>

$$\bar{\mathbf{u}} = -\frac{1}{12} f^2 \nabla p - \frac{1}{2}\hat{\theta} \qquad (8)$$

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Oil-air interface evolution eqn.

$$\frac{\partial Z}{\partial t} - \left[\frac{1}{2} + \frac{f^2}{12}\frac{\partial p}{\partial \theta}\right]\frac{\partial Z}{\partial \theta} + \frac{f^2}{12}\frac{\partial p}{\partial z} = 0 \qquad (9)$$

$BC's:$

$p = 0 \ along \ z = Z(\theta, t) \ and$

$p \ and \ Z \ are \ 2\pi \ periodic \ in \ \theta$

Simplify further by considering shallow grooves

$$(\theta, z) = 1 + \delta \sin\left(n\left[\theta - k z\right]\right) \qquad (10)$$

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Shallow sine groove OAI evolution, result of linearized theory

$$Z_0(\theta,t) = Z_{in}\left(\theta + \frac{t}{2}\right) -$$

$$\frac{1}{n\left(1+k^2\right)\cosh n}\left\{\sinh n \sin\left[n(\theta-k)\right] + k\cosh n \cos\left[n(\theta-k)\right] - k\cos n\theta\right\} \tag{11}$$

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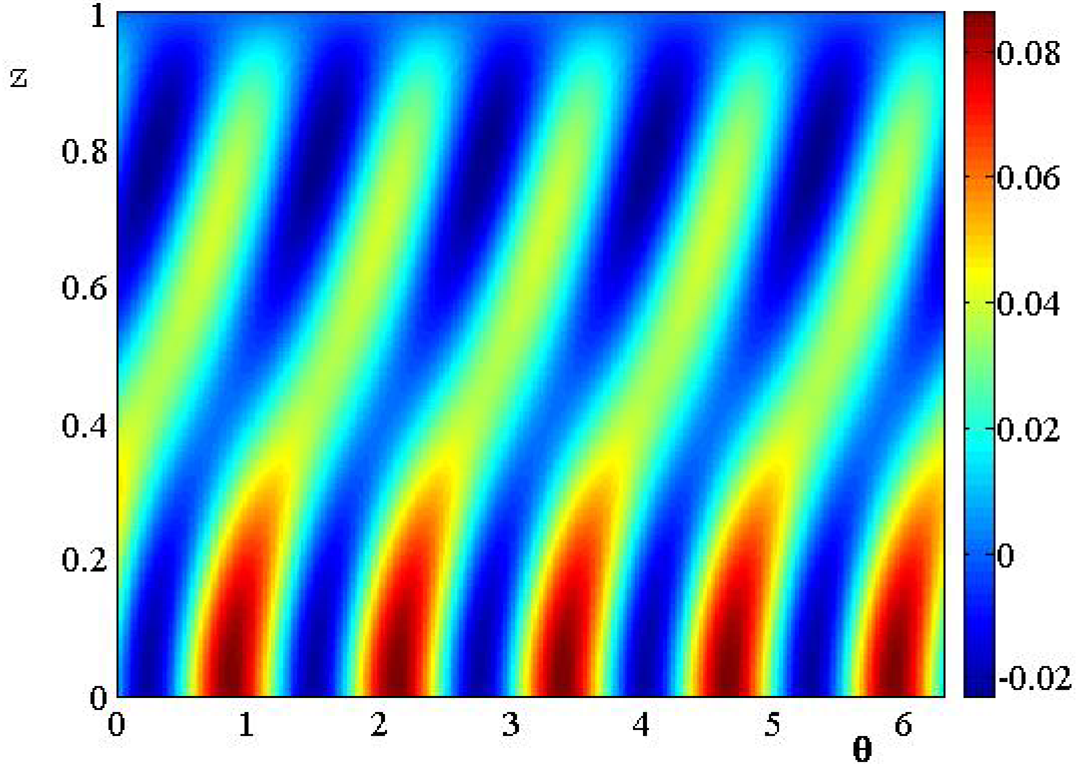
Linearized pressure distribution in a shallow, sinusoidally grooved herringbone



$$f(\theta, z) = 1 + \delta \sin\left[n(\theta - k z)\right]$$

$$k = 2, \quad n = 5$$

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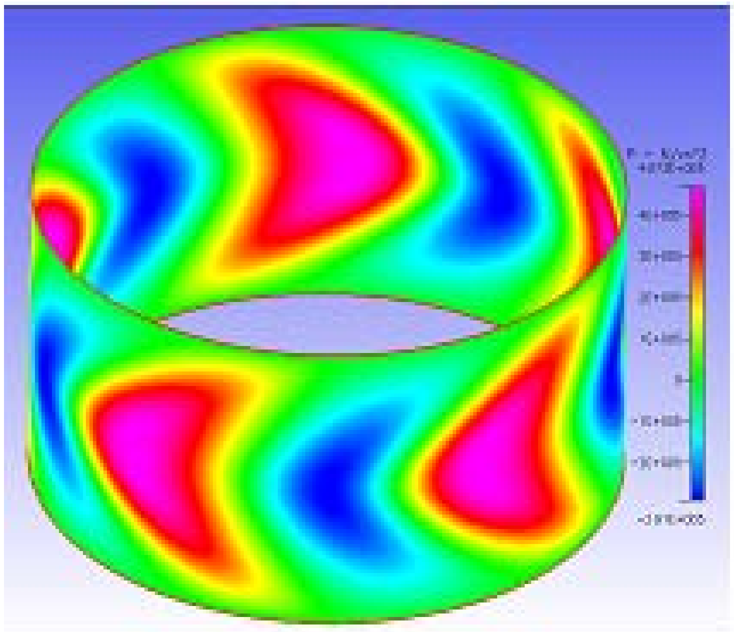
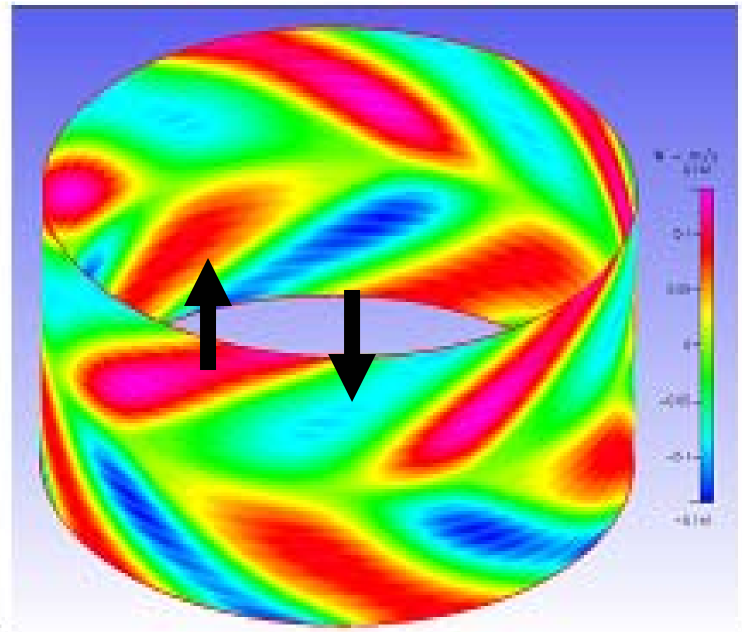
Herringbone with sinusoidal groove



(a)



(b)

pressure

axial flow

$$f(\theta, z) = 1 + \delta \sin\left[n(\theta - k z)\right]$$

$$k = 2,\ n = 5,\ \delta = .1$$

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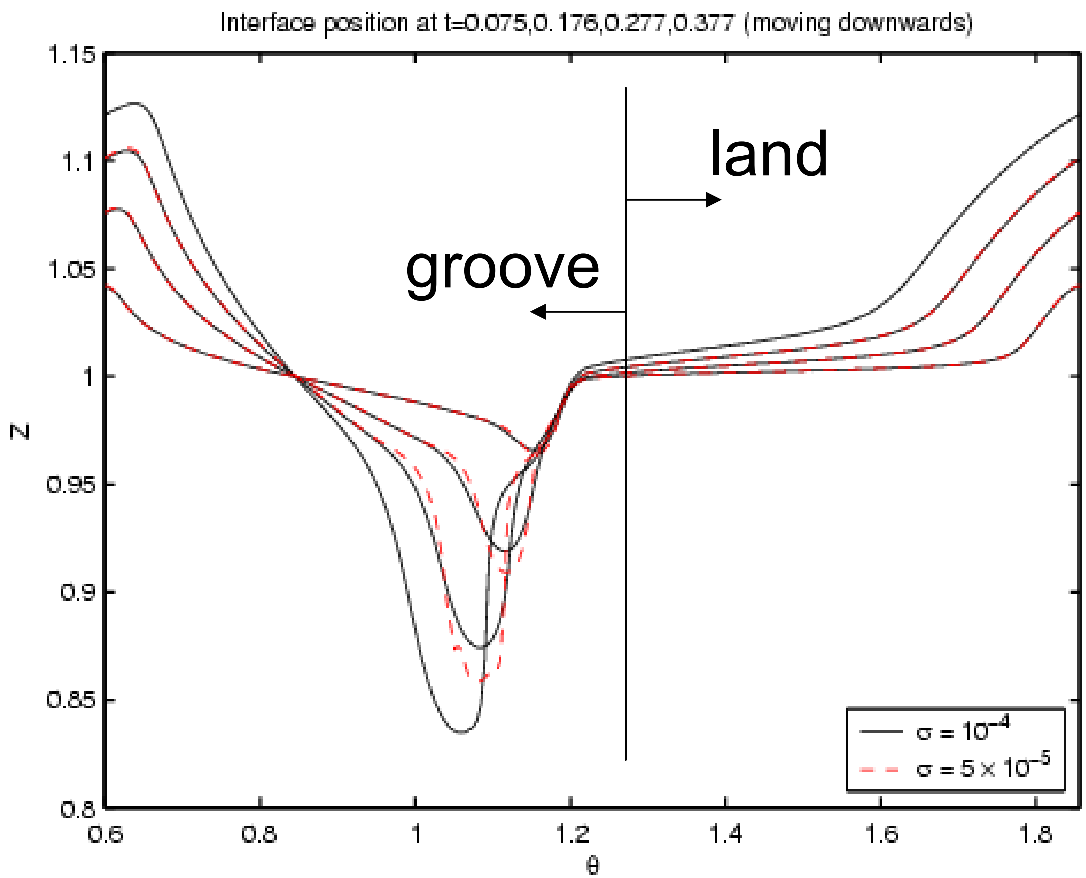
OAI evolution for a “tanh” groove profile



$$f(\theta, z) = 1 + \frac{H}{2}\left\{ 1 + \tanh\left[s \left[n(\theta - k z) \right] \right] \right\}$$

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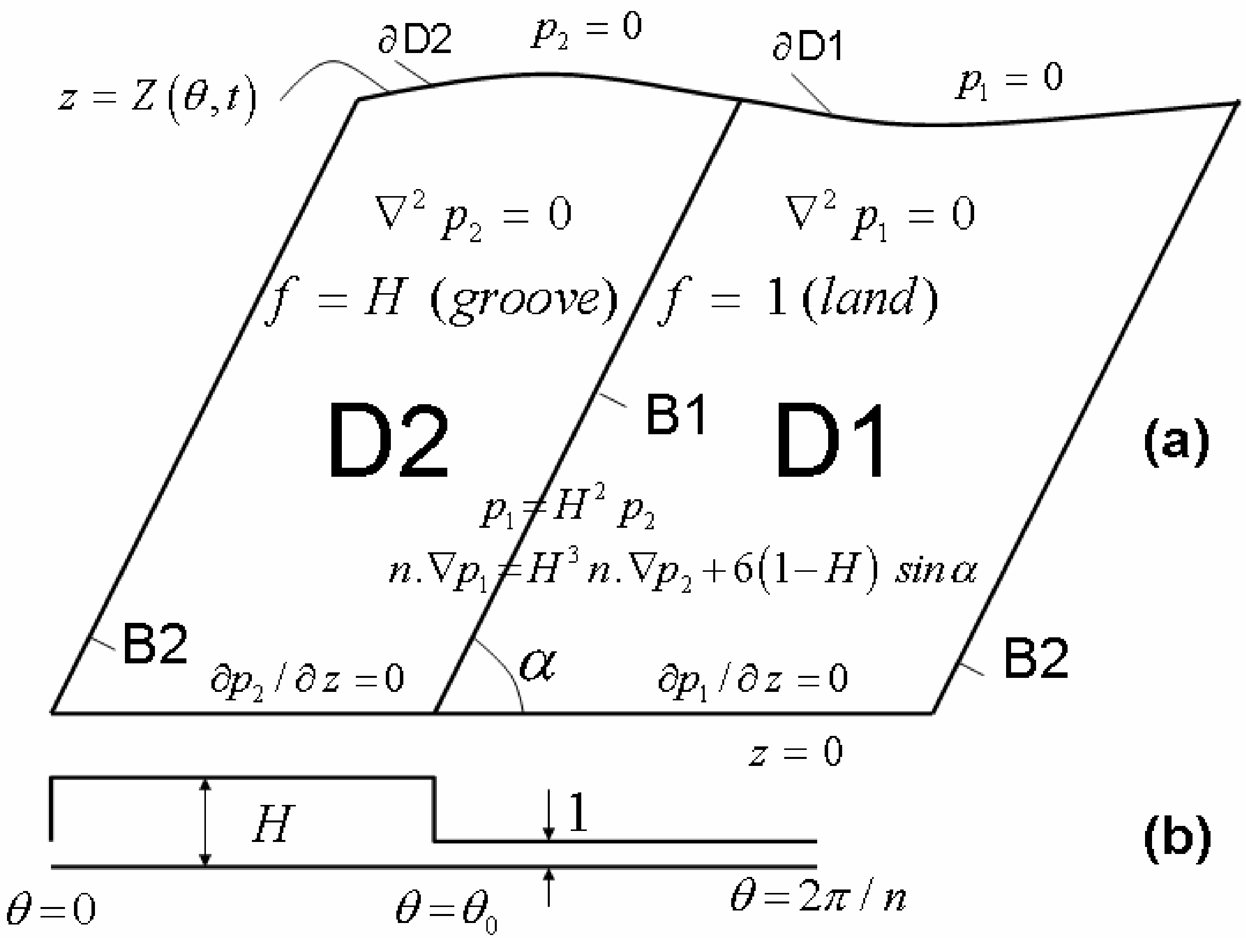
BEM problem setup: step groove



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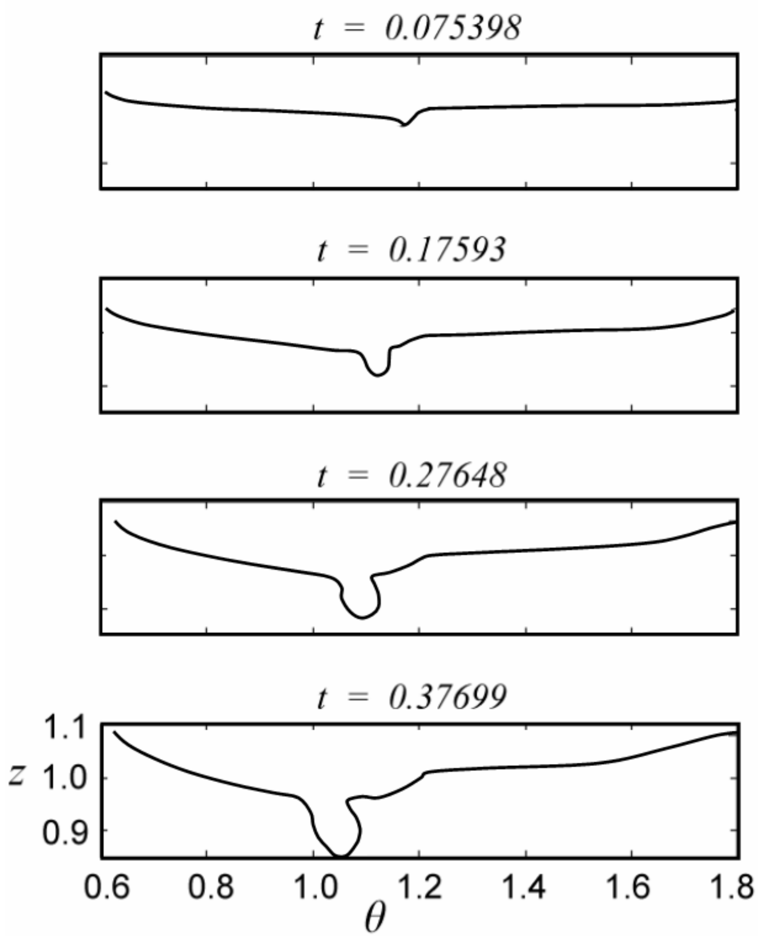
“Fingering” in a step groove profile BEM solution



Conclusions

- In fluid dynamic bearings of hard disk drives the oil-air interface deforms largely in response to the flow in the bearing interior. Surface tension has a regularizing effect.
- The OAI is drawn down into the grooves and squeezed upward in lands.
- Interfacial fingering develops, possibly leading to tip streaming. The step groove has the strongest fingering tendency.
- According to shallow groove theory the forced interfacial deflections are reduced ***exponentially*** as the **number of grooves** increases while they are reduced ***algebraically*** as the **groove angle** decreases. This agrees with experiments by Asada.

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