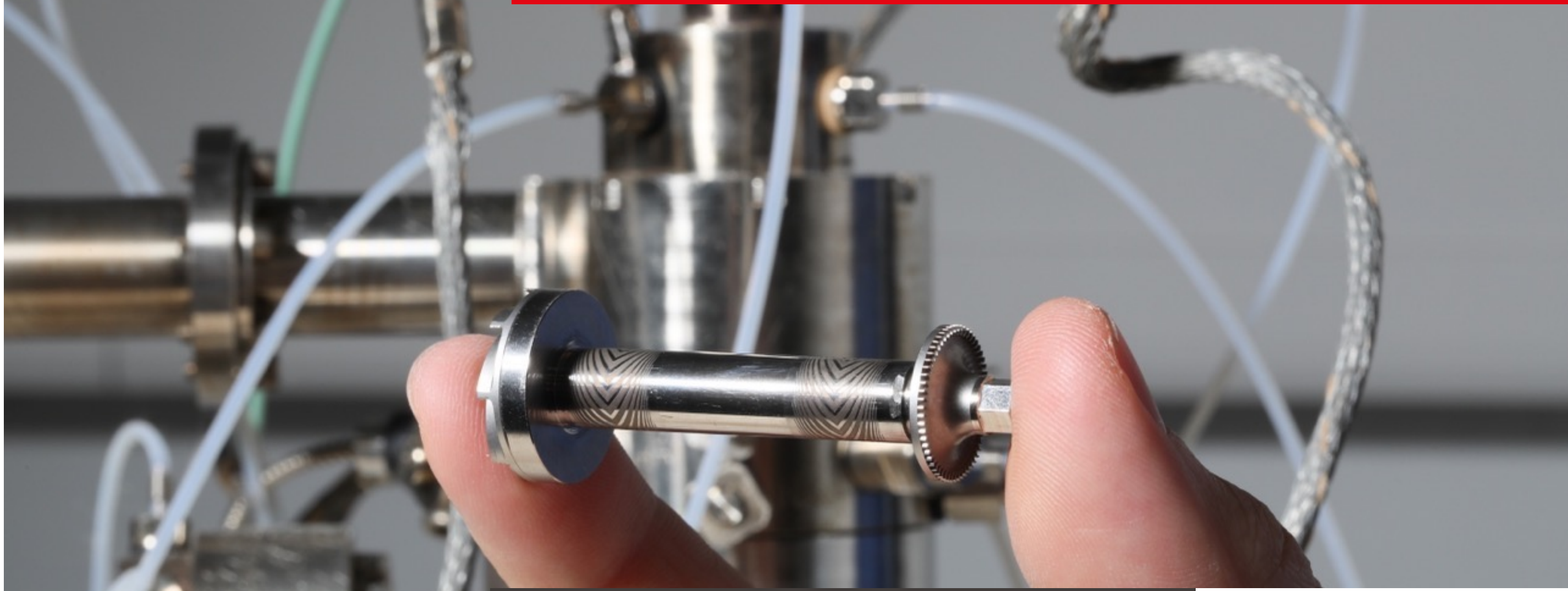
Gas Lubricated Bearings for Turbomachinery Applications



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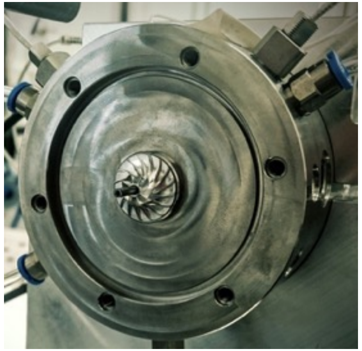
Overview

- Motivation & introduction
- Governing gas bearing equations
- Grooved gas lubricated bearings
- Practical implementation
- Conclusions

Selection of Examples

- Turbocompressor for domestic heat pumps improves performance by 20-30% compared to state of the art

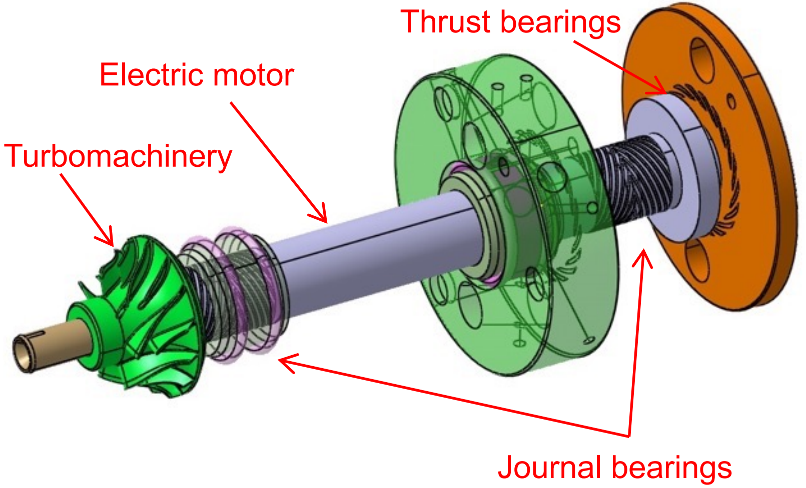


- Domestic scale solid oxide fuel cell (6 kW_{EL}) coupled to steam turbine driven anode off-gas fan achieves 66% efficiency

Common Denominator
Small-Scale Turbomachinery

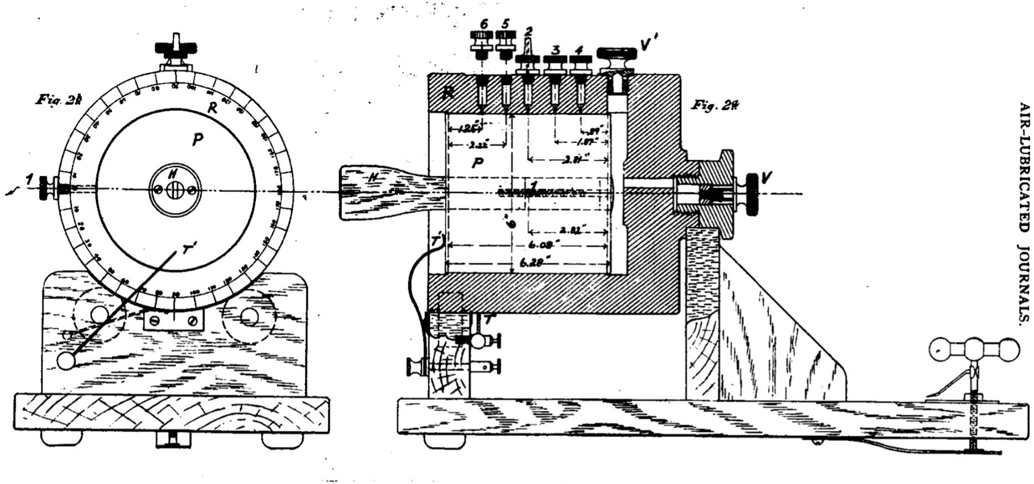
- Small-scale and oil-free turbomachinery is enabling technology for improving energy conversion efficiency

- Composed of several critical elements
  - Gas lubricated bearings
  - Turbomachinery aerodynamics
  - Integrated design approaches



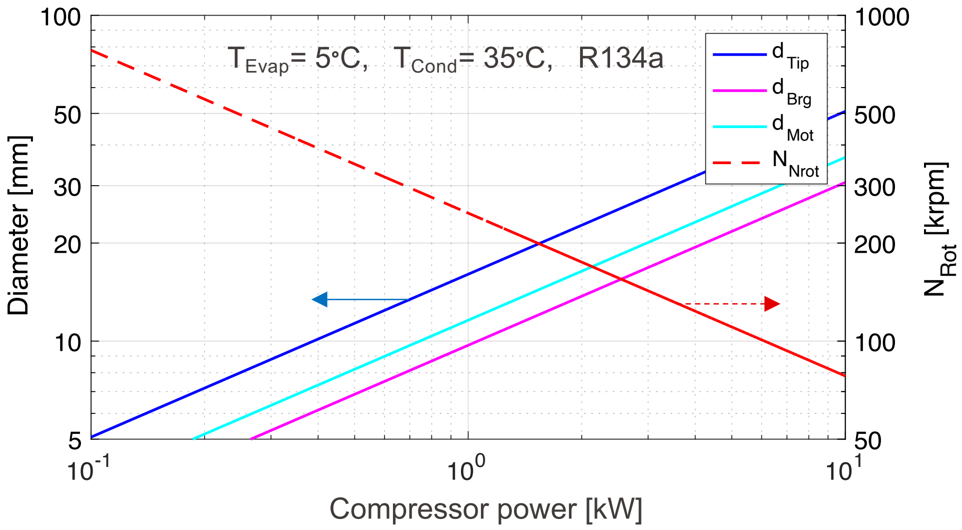
Common Denominator
Small-Scale Turbomachinery

- Gas lubrication : Discovered in 1895 (9 years after Reynolds equation), first paper in 1897.



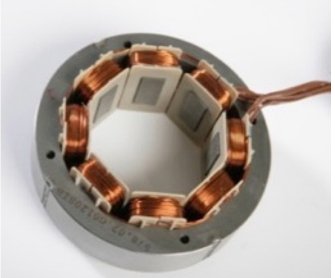
Turbocompressor Scaling Analysis

- Downscaling power increases speed (N_{rot}) and decreases siz
- Bearings
  - Need to support high speed (5 kHz)
  - High lifetime (>80'000 hours)
  - High efficiency
  - Low mechanical complexity

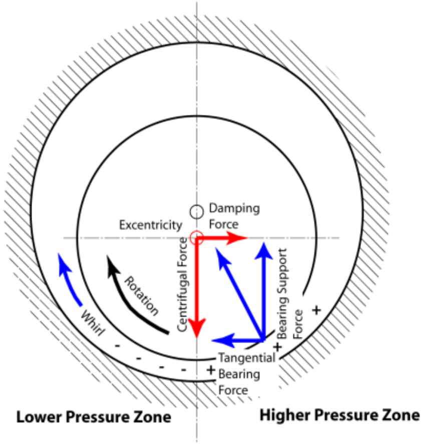


High Speed Bearing Classes

- Rolling element bearings
- Magnetic bearings
- Fluid film bearings

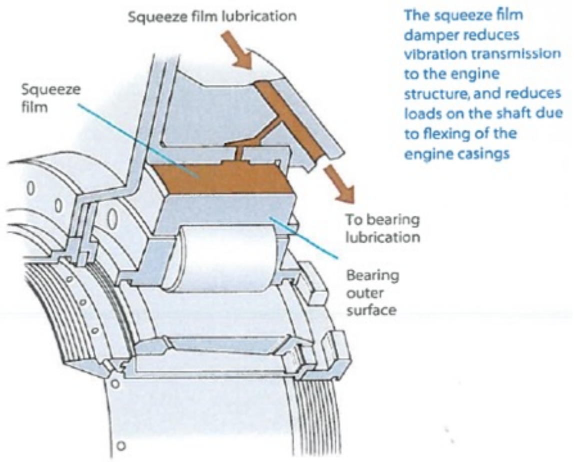


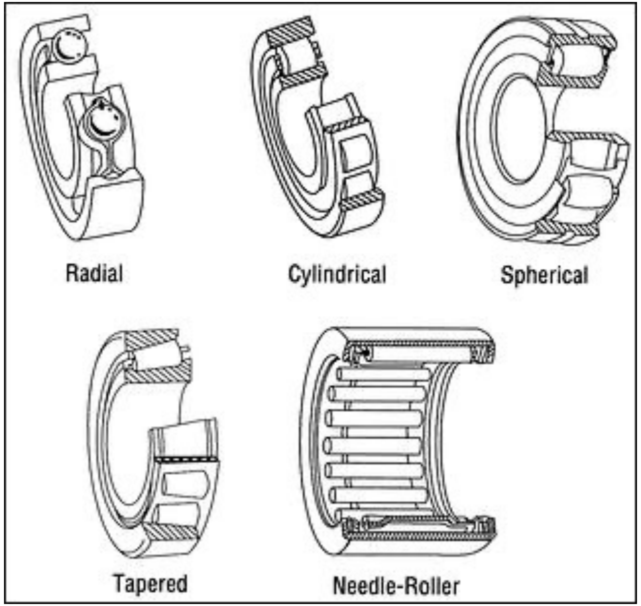
Courtesy of Mecos Traxler AG



Rolling Element Bearings

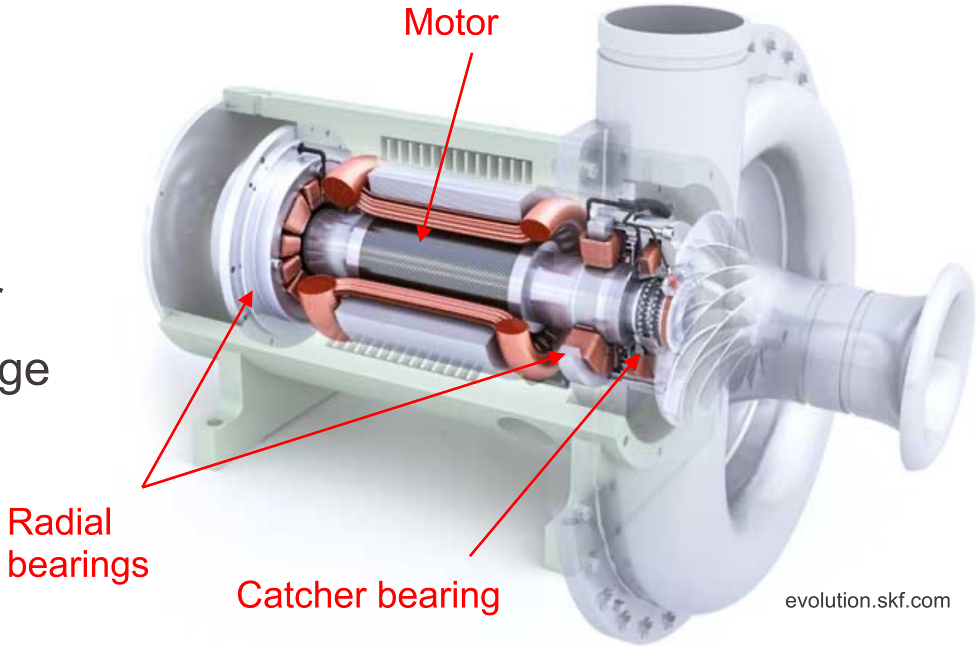
- Standardized and robust technology
- Needs controlled lubrication
- Limited lifetime
- Offers little damping





Magnetic Bearings

- No mechanical contact
- Work in vacuum
- Requires no lubrication
- Needs probes and controller
- Relatively expensive and large
- Requires catcher bearings



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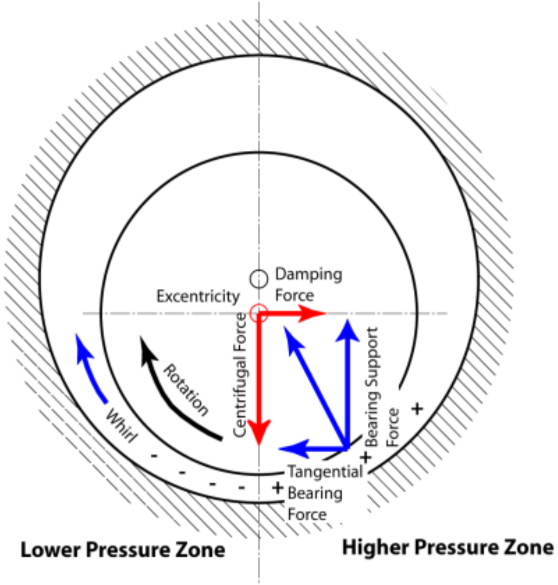
Fluid Film Bearings

- Features
  - Fluid film separates rotating from static part
  - Very simple geometry → ease of downscaling
  - No wear after liftoff
  - Low mechanical losses
  - No cycle contamination

- Challenges
  - Low specific load capacity and damping
  - Rotordynamic stability issues

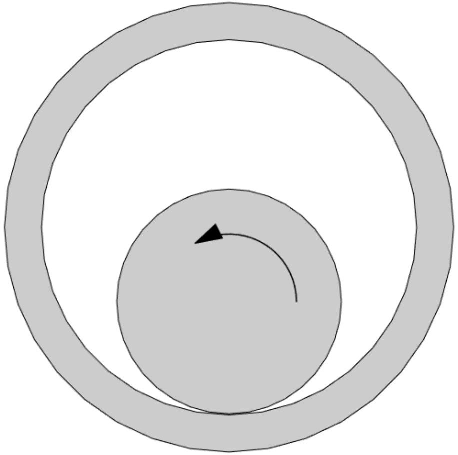


Fluid Film Bearing Classification

- Static (externally pressurized)
  - External source provides fluid film with pressurized fluid
  - Pressurization provides load capacity
- Dynamic (self-acting)
  - Relative motion between surfaces drags fluid into fluid film
  - Appropriate design generates load capacity
- Compressible or incompressible lubricant
  - Gas or liquid phase lubricant

Self-Acting Fluid Film Bearings

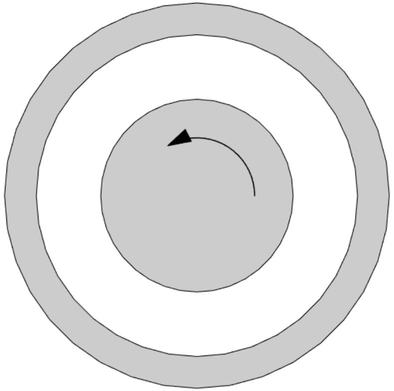
- Advantages
  - No external source of pressurized fluid required
  - Bearing is passive element
  - Requires little space
  - Simple geometry → ease of manufacturing
- Disadvantages
  - Contact at start & stop → potential wear
  - Sensitive to clearance distortion
  - Prone to instability



Instability

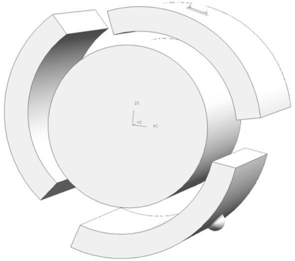
- Self-acting bearings are prone to modal unstability
  - Subsynchronous
  - Leads to limit cycles or contact at high speed
  - → Often means a destruction of the system

- Most of the R&D of the last 50 years aimed at improving the stability of gas bearings

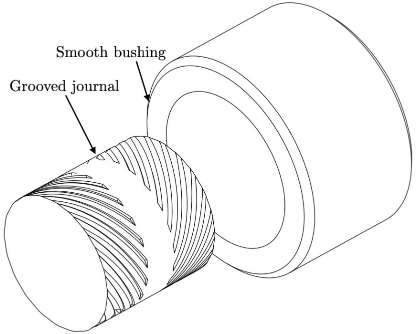


Example Self-Acting Fluid Film Bearings

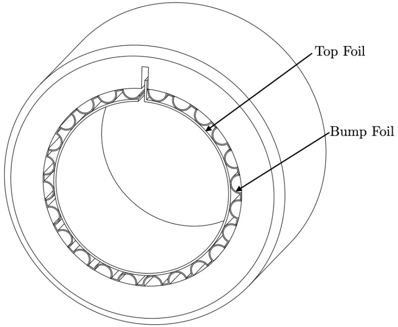
- Tilting pad



- Herringbone grooved



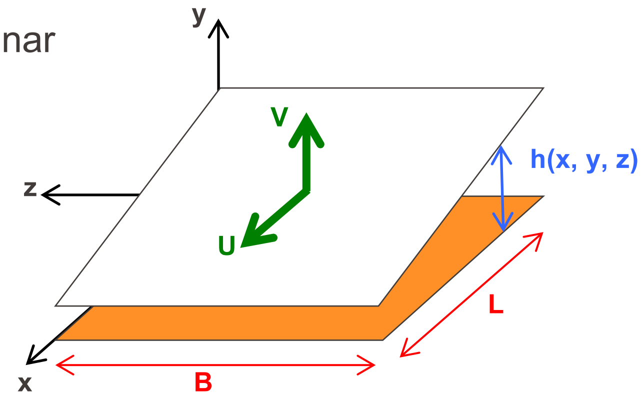
- Foil bearings



Reynolds Equations

Baseline hypothesis :

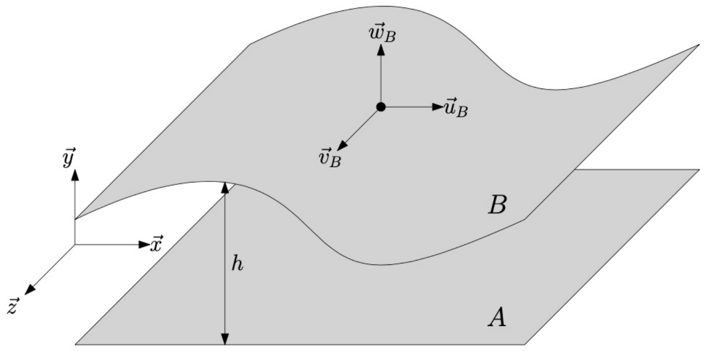
- The lubricant is a Newtonian fluid
- The film thickness is much smaller than the two other orthogonal directions (h/L<<1)
- The fluid characteristics (viscosity, pressure, density) are constant across the film thickness
- Viscous forces dominate the inertia and the flow is laminar



Reynolds Equations

Navier-Stokes boils down to :

$$\frac{\partial p}{\partial x} = \frac{\partial}{\partial y}\left(\mu \frac{\partial u}{\partial y}\right)$$
$$\frac{\partial p}{\partial z} = \frac{\partial}{\partial y}\left(\mu \frac{\partial v}{\partial y}\right)$$



Integrated twice :

$$u = \frac{y^2}{2\mu}\frac{\partial p}{\partial x} + A\frac{y}{\mu} + B$$
$$v = \frac{y^2}{2\mu}\frac{\partial p}{\partial z} + C\frac{y}{\mu} + D$$

With boundary conditions:

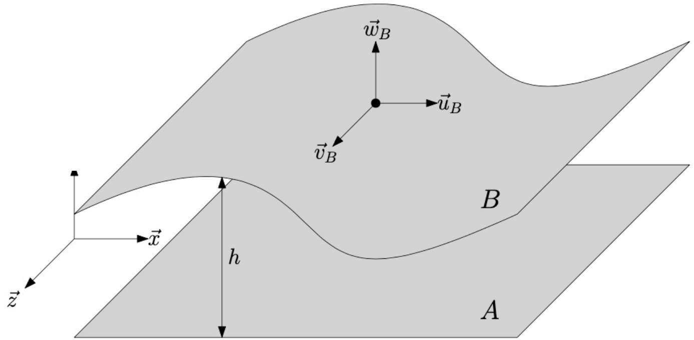
$$y = 0, u = 0, v = 0$$
$$y = h, u = u_B, v = v_B$$

→ Hypothesis :no-slip condition at the solid-lubricant interface

Reynolds Equations

Navier-Stokes boils Integration constants are determined:

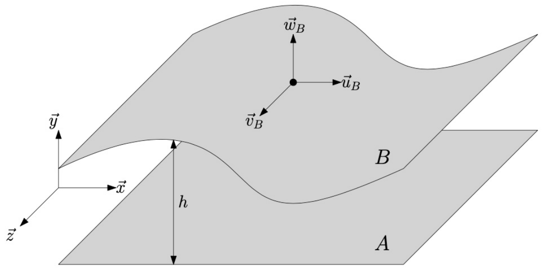
$$u = -y\left(\frac{h-y}{2\mu}\right)\frac{\partial p}{\partial x} + u_B\frac{y}{h}$$
$$v = -y\left(\frac{h-y}{2\mu}\right)\frac{\partial p}{\partial z} + v_B\frac{y}{h}$$



Mass flow rate per unit length:

$$\psi_x = \rho\int_0^h u\mathrm{d}y \quad = \rho\left(-\frac{h^3}{12\mu}\frac{\partial p}{\partial x} + \frac{u_b}{2}h\right)$$
$$\psi_y = \rho\int_0^h v\mathrm{d}y \quad = \rho\left(-\frac{h^3}{12\mu}\frac{\partial p}{\partial y} + \frac{v_b}{2}h\right)$$

Reynolds Equations



Integration of the continuity equation :

$$\int_0^h \left(\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial z}(\rho v) + \frac{\partial}{\partial y}(\rho w) \right) \mathrm{d}y = 0$$

$$\int_0^h \frac{\partial}{\partial y}(\rho w)\mathrm{d}y = \rho w_b$$

$$\int_0^h \frac{\partial}{\partial z}(\rho v)\mathrm{d}y = -\rho v_b \frac{\partial h}{\partial y} + \frac{\partial}{\partial y}\left(\rho \int_0^h v\mathrm{d}y \right)$$

$$\int_0^h \frac{\partial}{\partial x}(\rho u)\mathrm{d}y = -\rho u_b \frac{\partial h}{\partial x} + \frac{\partial}{\partial x}\left(\rho \int_0^h u\mathrm{d}y \right)$$

Recognize the mass flow rate per length expressions :

$$\frac{\partial}{\partial x}\underbrace{\left(\rho \int_0^h u\mathrm{d}y \right)}_{\psi_x} + \frac{\partial}{\partial z}\underbrace{\left(\rho \int_0^h v\mathrm{d}y \right)}_{\psi_z} + h\frac{\partial \rho}{\partial t} + \rho \underbrace{\left(-u_b \frac{\partial h}{\partial x} - v_b \frac{\partial h}{\partial z} + w_b \right)}_{\frac{\partial h}{\partial t}} = 0$$

Reynolds Equations

Following the development of material derivative of h:

$$0 = \underbrace{\frac{\partial}{\partial x}\left(-\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(-\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right)}_{Poiseuille} + \underbrace{\frac{\partial}{\partial x}\left(\frac{\rho h u_B}{2}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h v_B}{2}\right)}_{Couette} + \frac{\partial(\rho h)}{\partial t}$$

Reynolds Equation for Journal Bearing

- Change coordinate system (cartesian → cylindrical): $x = R\theta$

- Normalize with

$$Z = \frac{z}{R} \qquad P = \frac{p}{p_{Amb}} \qquad \mathrm{H} = \frac{h}{C} \qquad u_0 = R\omega_{Rot} \qquad T = \omega_{Ex} t$$

Journal radius (R), Rotor speed (ω_{Rot}), Ambient pressure (p_{Amb}), Nominal clearance (C), Excitation frequency (ω_{Ex})

- And assume a constant viscosity to lead to:

$$\partial_\theta\left(\rho H^3 \partial_\theta P\right) + \partial_Z\left(\rho H^3 \partial_Z P\right) = \underbrace{\frac{6\mu\omega_{Rot}}{p_{Amb}}\left(\frac{R}{C}\right)^2}_{\Lambda} \partial_\theta\left(\rho H\right) + \underbrace{\frac{12\mu\omega_{Ex}}{p_{Amb}}\left(\frac{R}{C}\right)^2}_{\sigma} \partial_T\left(\rho H\right)$$

Poiseuille terms; Compressibility number (Λ); Squeeze number (σ)

Simplifying Reynolds Equation

- Further assumptions
  - Isothermal fluid film
  - Ideal gas
  - Constant viscosity

$$\rho = \frac{P p_{Amb}}{R_G T}$$

Temperature

Spec. Gas constant

- Reynolds equation for journal bearing lubricated with a ideal gas & isothermal film

$$\partial_\theta\left(PH^3\partial_\theta P\right)+\partial_Z\left(PH^3\partial_Z P\right)=\Lambda\partial_\theta\left(PH\right)+\sigma\,\partial_T\left(PH\right)$$

Dynamic Bearing Properties

- Objective is prediction of dynamic stiffness and damping
- How to proceed?

1. Load bearing with static load → leads to static rotor eccentricity
2. Perturb rotor with periodic oscillation around static equilibrium
3. Integrate unperturbed pressure field to calculate static load capacity
4. Integrate perturbed pressure field to calculate stiffness and damping matrices

Dynamic Bearing Properties

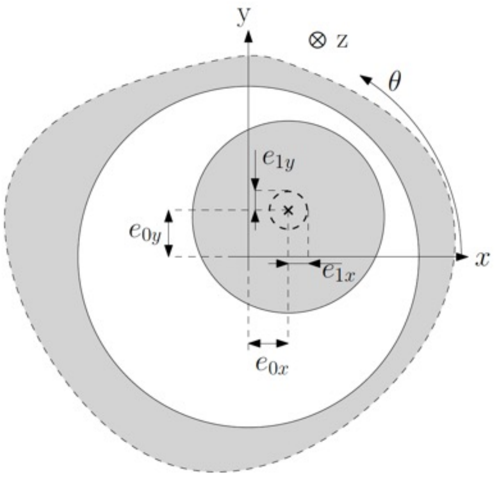
- Introduce static and dynamic perturbation

$$H = 1 + \varepsilon \cos\theta + \overline{x} e^{i\omega_{Ex} t} + \overline{y} e^{i\omega_{Ex} t}$$

$$P = 1 + \varepsilon P_1 + \overline{x} P_x e^{i\omega_{Ex} t} + \overline{y} P_y e^{i\omega_{Ex} t}$$

Dynamic perturbation in x

Static perturbation



- Introduce into Re-equation, collect 0th and 1st order terms and solve for static and dynamic pressure fields using Finite Difference Method or Finite Volume Method

- Integrate P on bearing domain to get the bearing reaction forces and identify load capacity, stiffness and damping coefficients K & C

Dynamic System Properties

- Use the linearized dynamic coefficients K & C to solve the eigenproblem of the rotor-bearing system

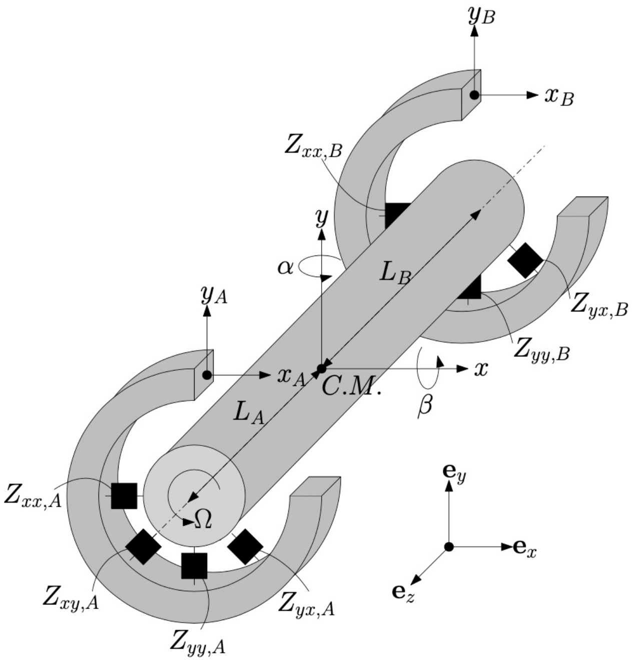
$$M\ddot{q} + C\dot{q} + Kq = 0$$

State-space problem $\dot{\mathbf{z}} + \mathbf{A}\mathbf{z} = 0$

where $\mathbf{z} = \begin{pmatrix} \dot{\mathbf{q}} \\ \mathbf{q} \end{pmatrix}$

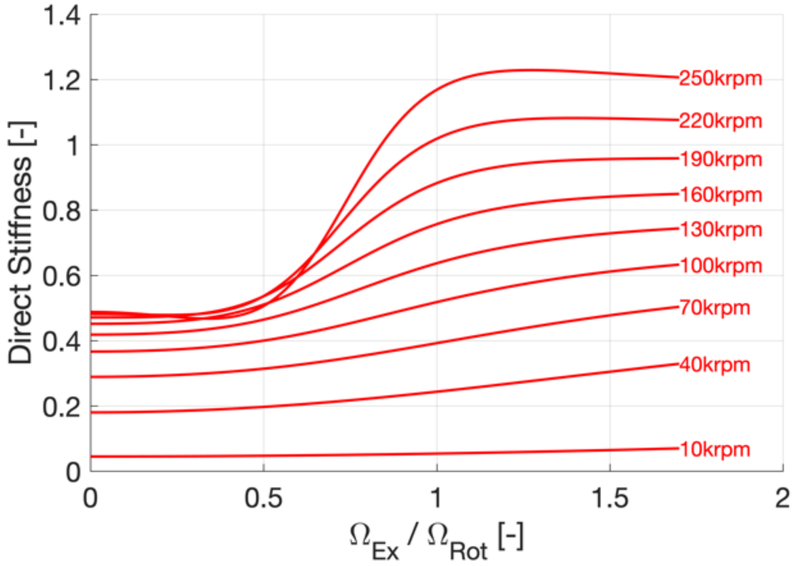
and $\mathbf{A} = \begin{pmatrix} \mathbf{M}^{-1}\mathbf{C} & \mathbf{M}^{-1}\mathbf{K} \\ -\mathbf{I}_n & \mathbf{0}_n \end{pmatrix}$

eigenproblem $\det(\mathbf{A} - \delta_p \mathbf{I}_{2n})\mathbf{b}_p = 0$



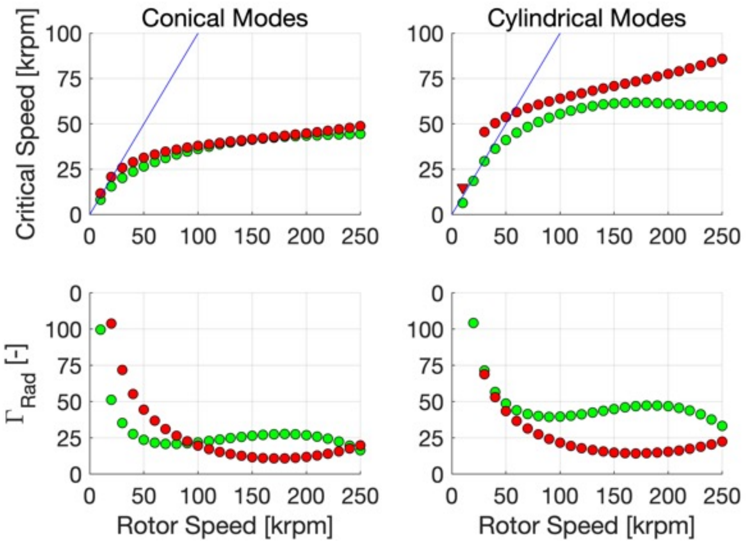
Particularity of Gas Lubricated Bearings

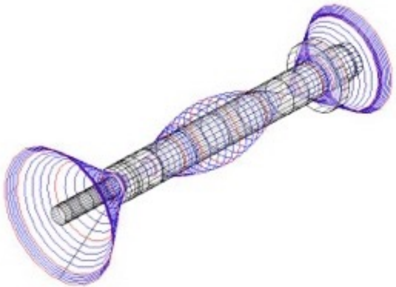
- Gas lubricated bearing properties dependent on rotor and excitation frequency → $K = K(f_{ex})$, $C = C(f_{ex})$



Dynamic Bearing Properties

- Compute the modal logarithmic decrement of all your eingenmodes
  → they must remain positive !





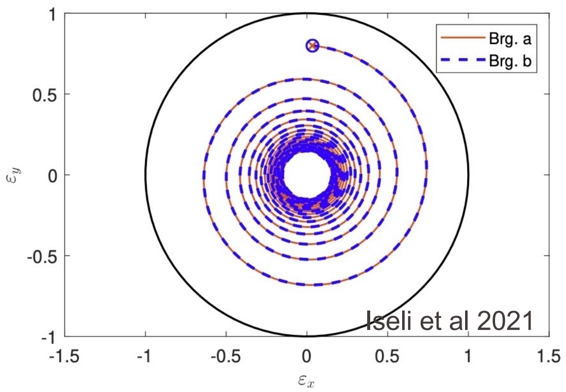
- This method is very fast but it relies on a linearization around an equilibrium point

Non-linear dynamic evaluation

- Alternatively, the Reynlods equation can be integrated <u>in time</u> and coupled with the rotordynamics equations of motion

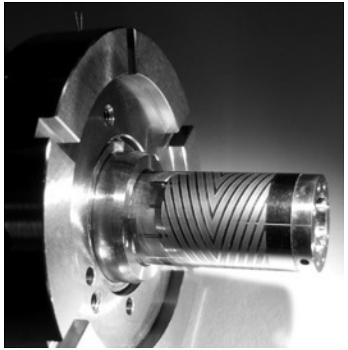
$$\partial_\theta\left(PH^3\partial_\theta P\right)+\partial_Z\left(PH^3\partial_Z P\right)=\Lambda\partial_\theta\left(PH\right)+\boxed{\sigma\partial_T\left(PH\right)} \qquad \dot{\mathbf{q}} = f(\mathbf{q},t)$$

- Very computationally costly, but captures non-linear effects

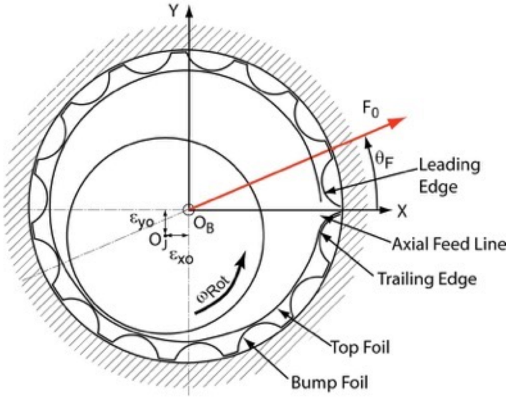


Grooved Gas Lubricated Bearings

Dynamic Gas Lubricated Bearings



- Herringbone groove journal bearings
  - Can achieve very high stability thresholds
  - Rigid bearing bushings need tight clearances and perfect alignment

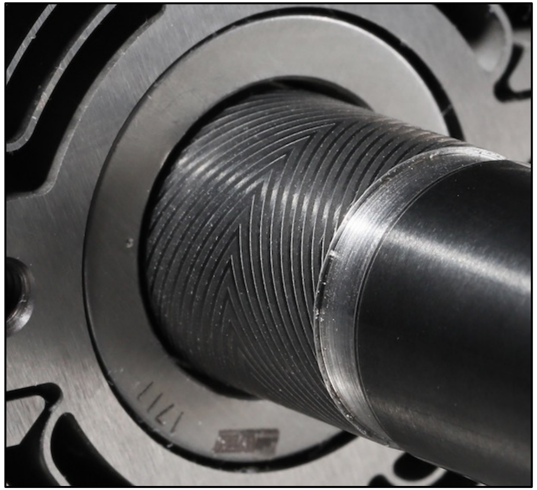


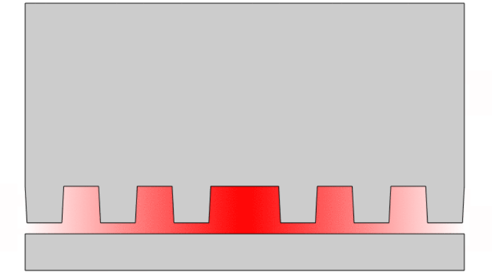
- Foil bearings
  - Fluid film operates in series with a compliant surface
  - Complex interaction between fluid film and soft structure
  - Compliant structure adds external damping
  - Tolerant to misalignment and thermal gradients

- Large number of models available but no validation data

Herringbone Grooved Journals

- Structured groove pattern improves stability
- Pumping grooves increase fluid film pressure
- Can achieve very high stability thresholds
- Rigid bearing bushings need tight clearances (5÷10 µm) and perfect alignment

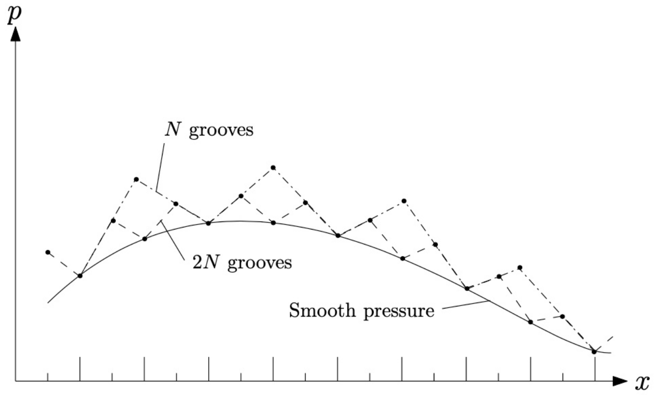




Modeling of Herringbone Grooved Journals

- Assumption of infinite number of grooves reduces saw-tooth pressure profile to smooth pressure

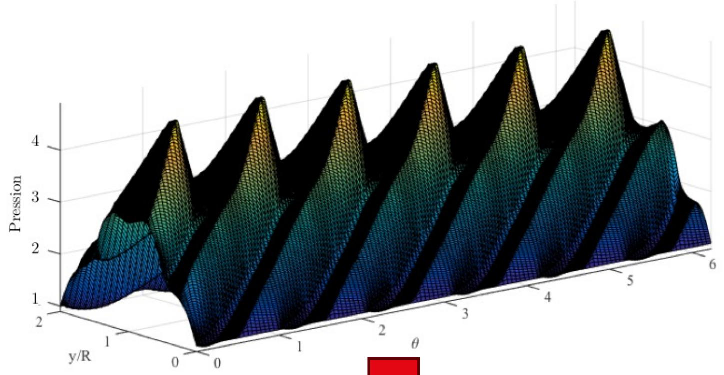
- Narrow Groove Theory (NGT) allows to express fluid film pressure evolution as modified Re-equation

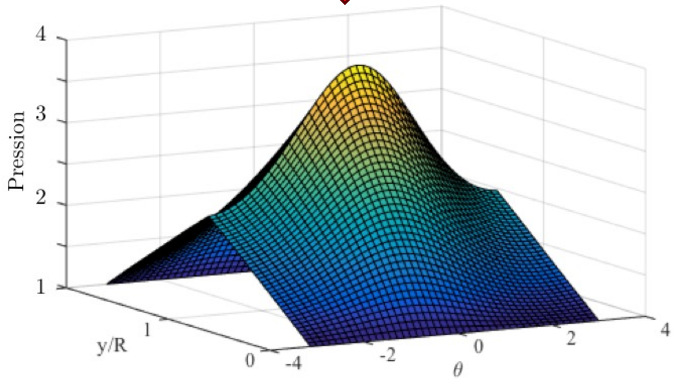


$$\partial_\varphi\left[\overline{P}\left(f_1\partial_\varphi\overline{P}+f_2\partial_{\bar{z}}\overline{P}\right)\right]+\partial_{\bar{z}}\left[\overline{P}\left(f_2\partial_\varphi\overline{P}+f_3\partial_{\bar{z}}\overline{P}\right)\right]+c_s\left[\sin\beta\partial_\varphi\left(\overline{P}f_4\right)-\cos\beta\partial_{\bar{z}}\left(\overline{P}f_4\right)\right]-\Lambda\partial_\varphi\left(\overline{P}f_5\right)-\sigma\partial_{\bar{t}}\left(\overline{P}f_5\right)=0$$

Modeling of Herringbone Grooved Journals

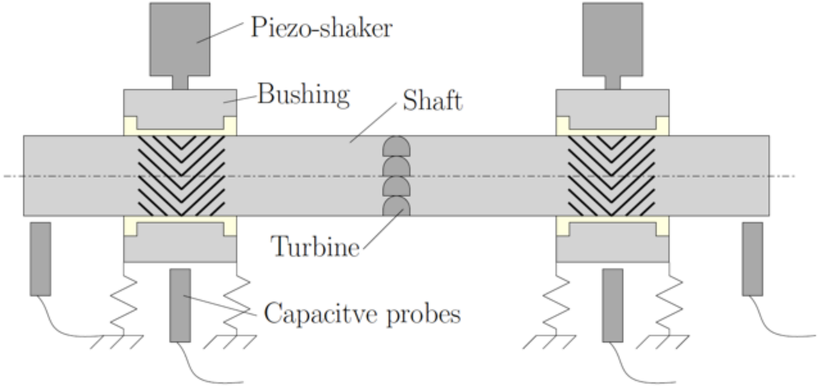
- NGT allows a very fast and elegant mathematical treatment of the lubricating effects due to the texturing
- No need to capture each groove individually in the numerical method
- Has been the dominating method for grooved bearing design for the last 60 years





Test-Rig for Herringbone Grooved Journals

- Excitation of softly supported bushings via piezo-shakers
- Measurement of relative motion between rotor and bushing allows to retrieve stiffness and damping matrices



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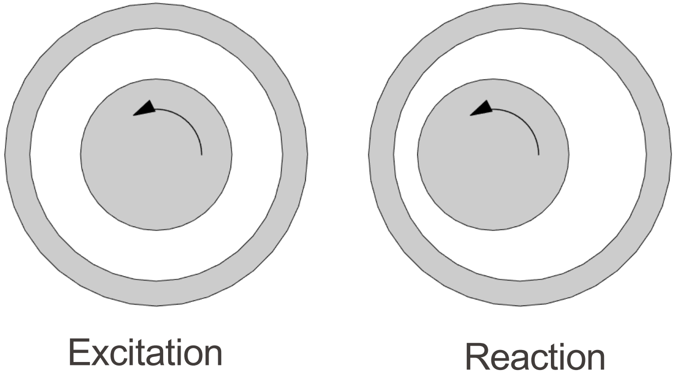
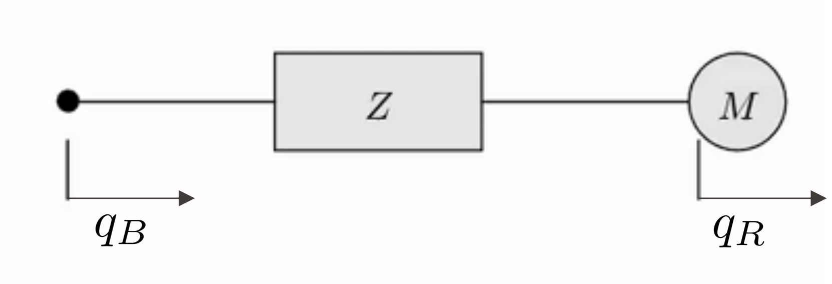
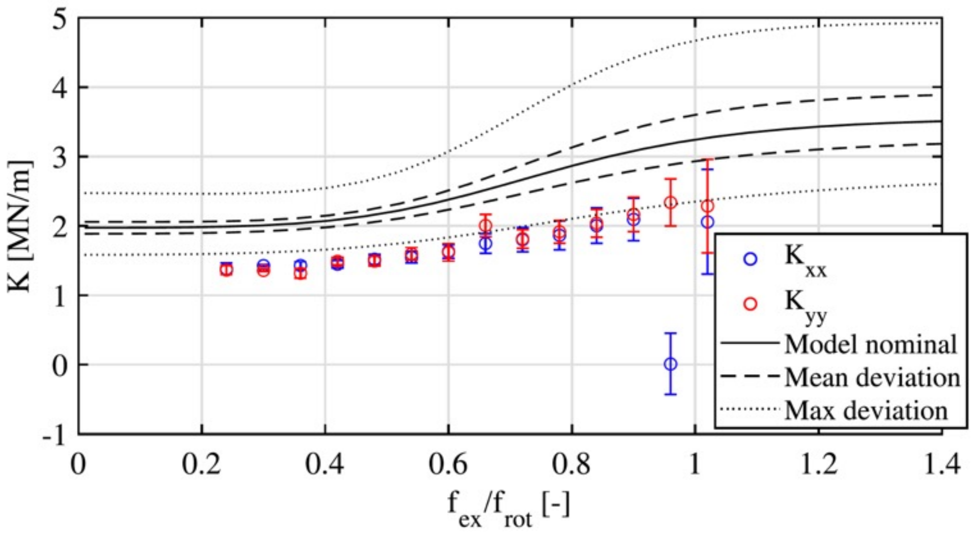
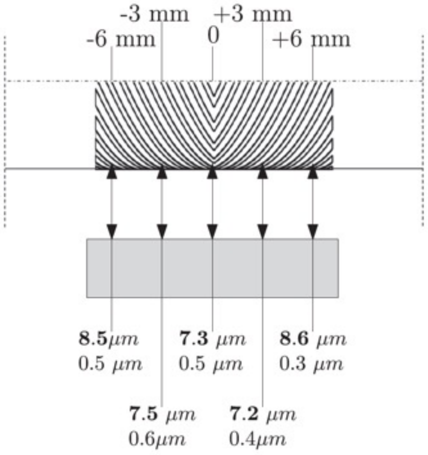




Measurement of Herringbone Grooved Journals

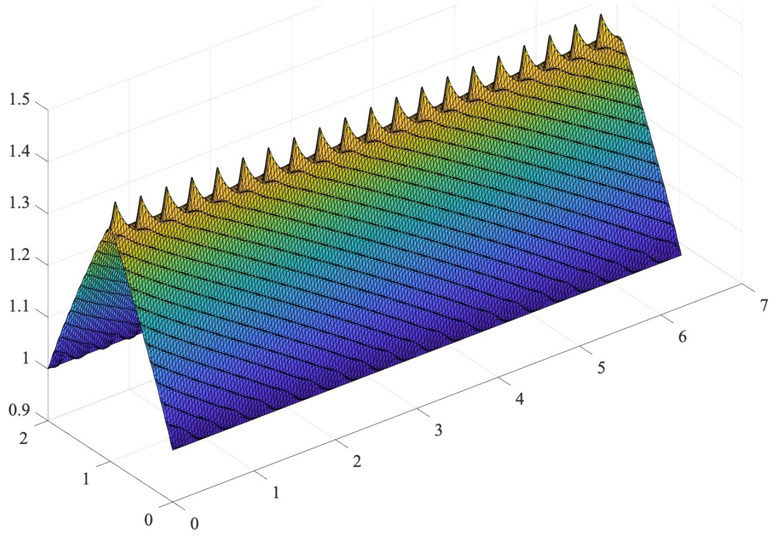




- Agreement between model and experimental data validates NGT
- Effects of manufacturing deviation significant

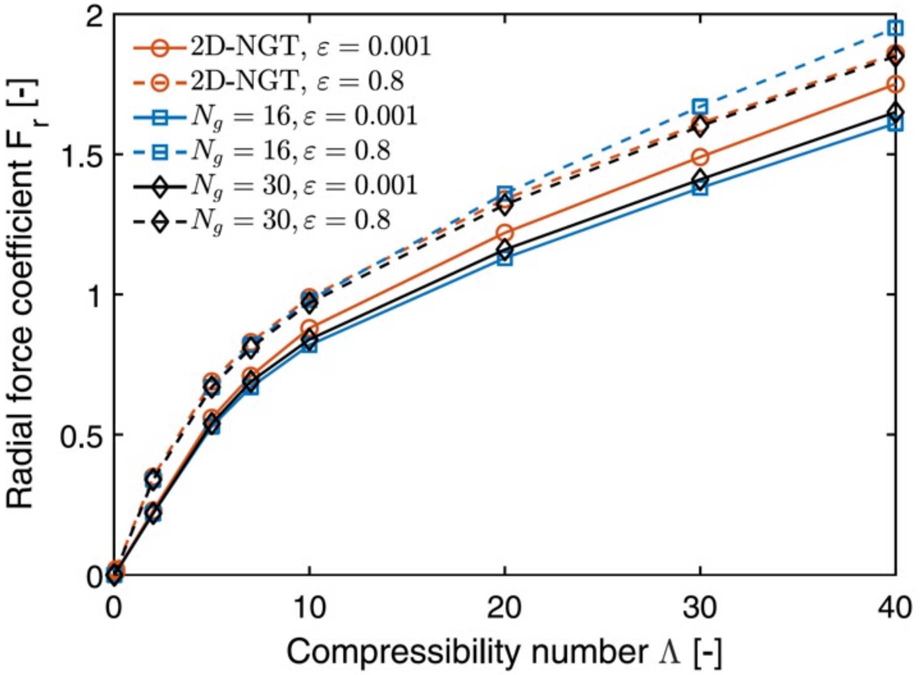
Modeling of Herringbone Grooved Journals

- NGT makes many questionable assumptions
- Objective was to introduce a solution of Re-equation for grooved bearings
- Modeling approach based on periodicity of pressure profiles
- Finite groove approach (FGA)
  - Finite Volume Method
  - Finite Element Method



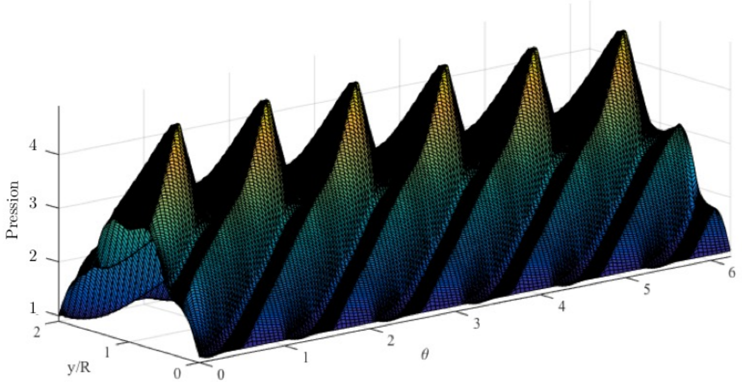
Static Performance (FGA)

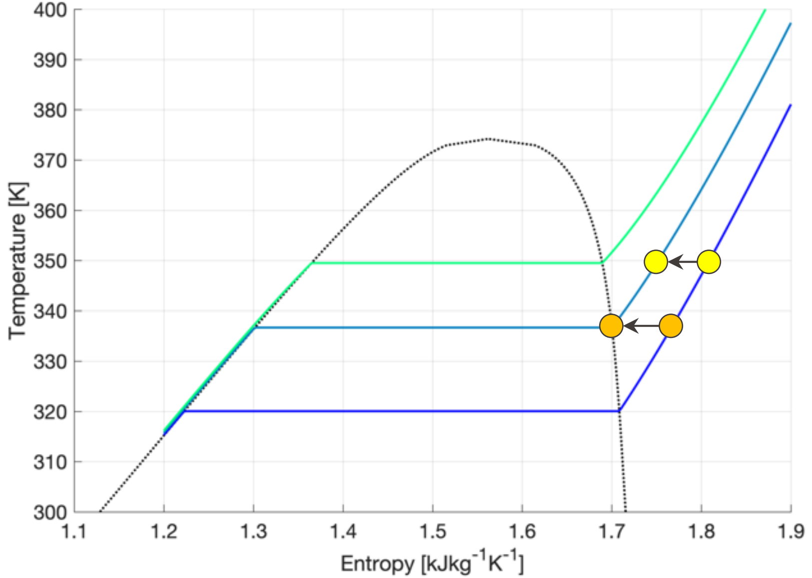
- NGT suggested to predict slightly larger static force coefficients than exact solution (<3%)
- Higher number of grooves decreases difference to NGT



Condensation in Gas Lubricated Bearings

- Bearings operated close to saturation line exposed to real gas behavior
  - Deviation from ideal gas behavior
  - Potential condensation





Capturing condensation

Under isothermal condensation : $\partial_{\theta|z} P = 0$

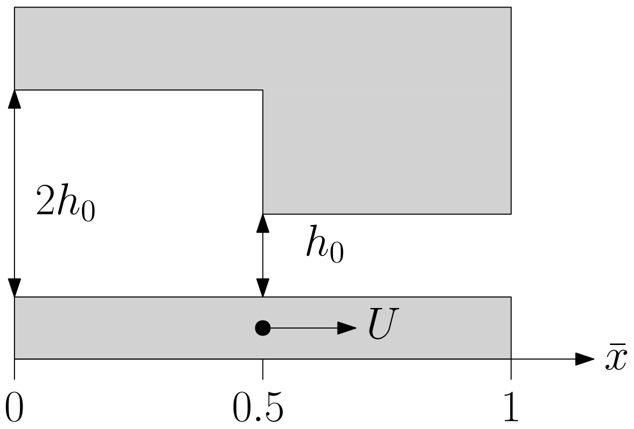
→ Re-equation changes from elliptic to hyperbolic → Information travels forward, not backward

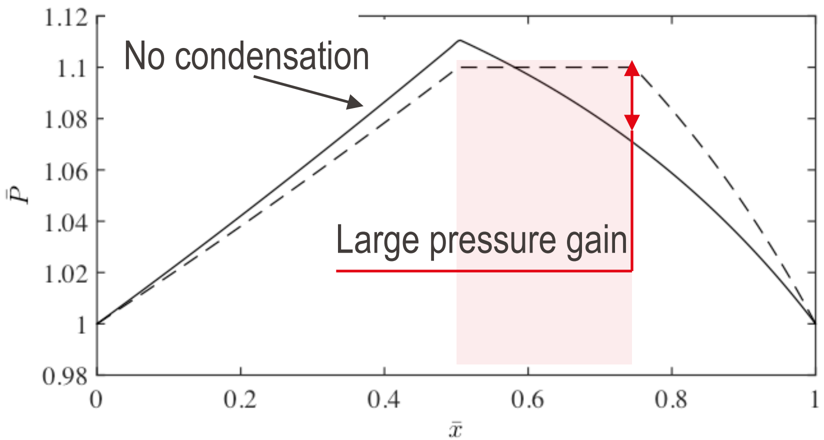
$$\partial_\theta(\bar{\rho}\bar{h}) + \sigma \partial_T(\bar{\rho}\bar{h}) = 0$$

- Numerical treatment inspired by work on cavitation in liquid bearings

Effects of Condensation

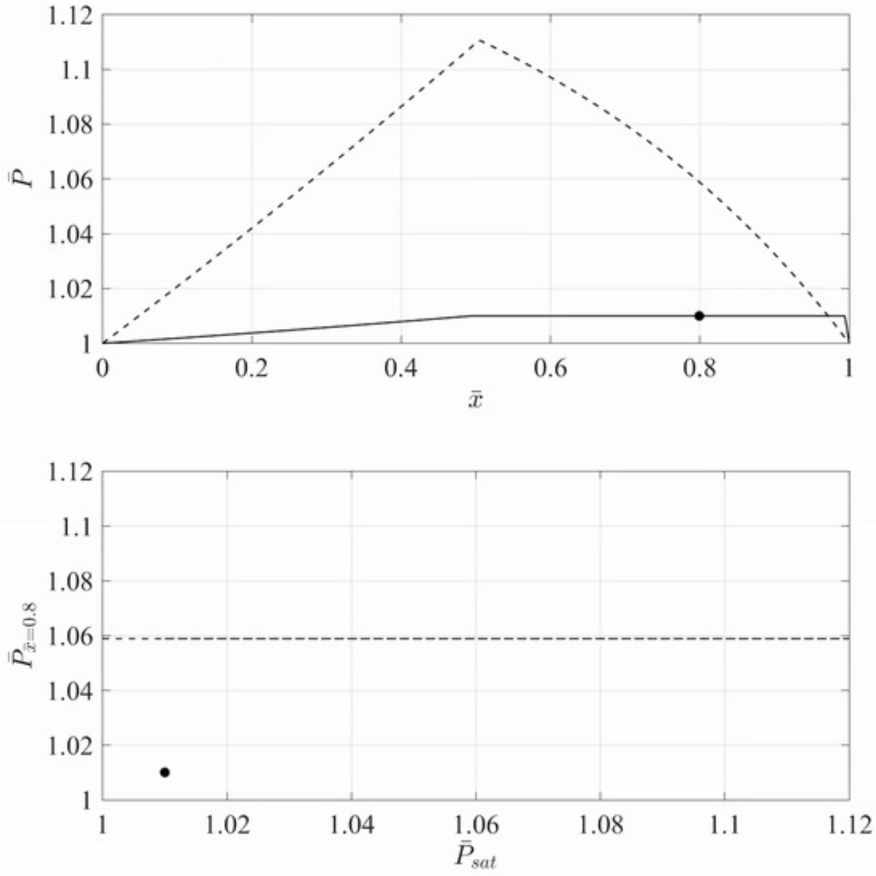
- Condensation links pressure to temperature and pressure governed purely by fluid film temperature (isothermal behavior)
- The closer to the saturation curve the wider the condensation zone





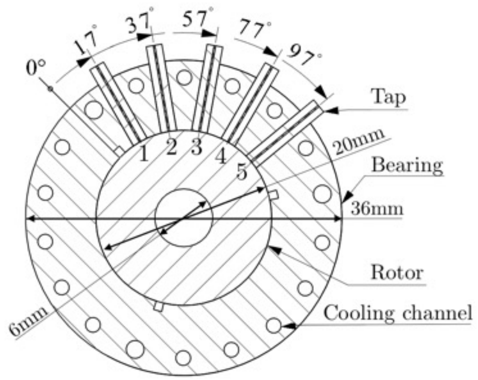
Effects of Condensation

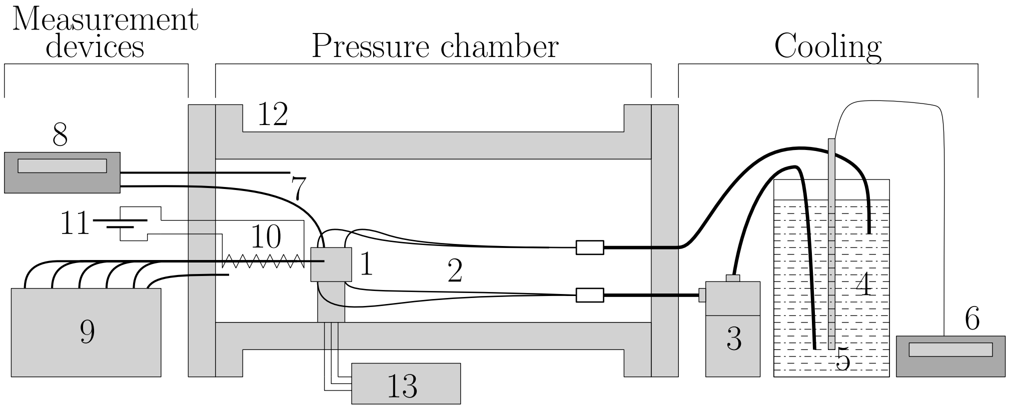
- As the temperature increases, so does the saturation pressure
- At x=0.8, the pressure experiences a pressure excursion <u>above</u> its single-phase gas value

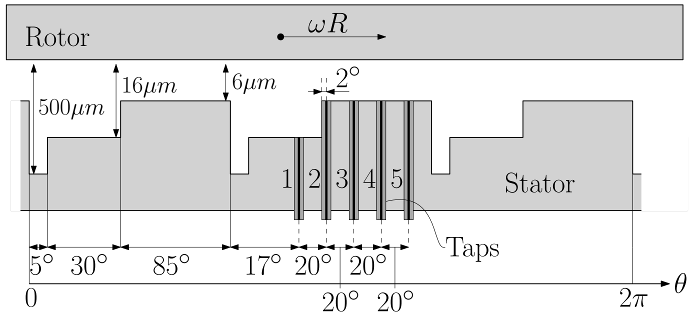


Experimental Setup for Condensation

- Instrumented 3-pad Rayleigh step bearing
- Vertical operation to avoid radial load
- Test performed in R245fa

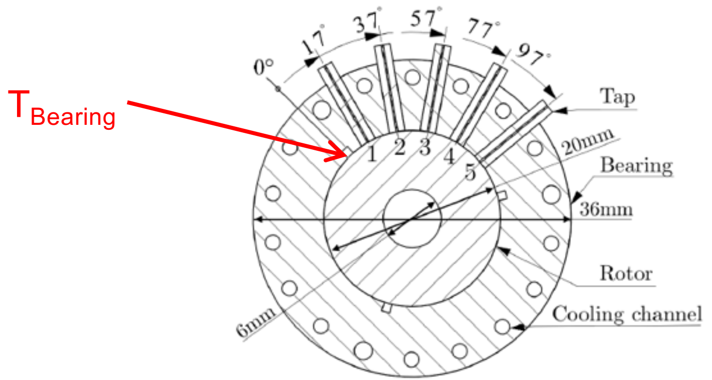


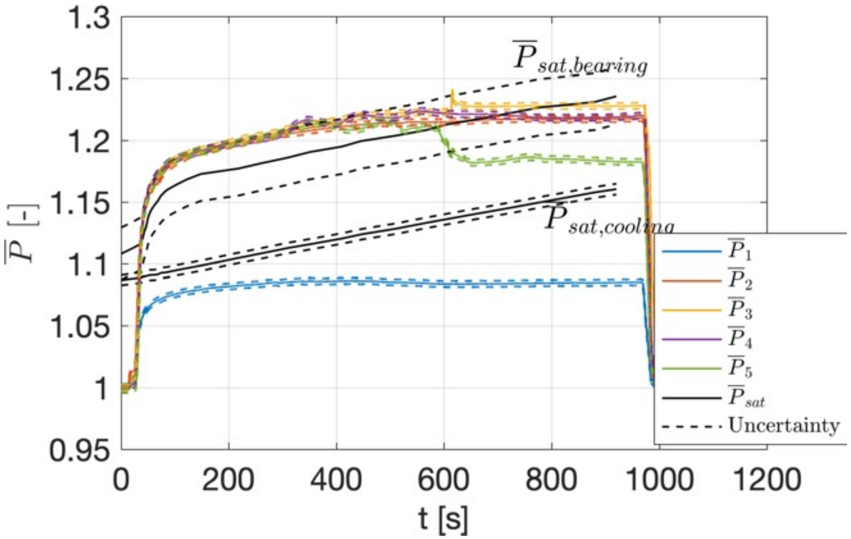




Experimental Results with Condensation

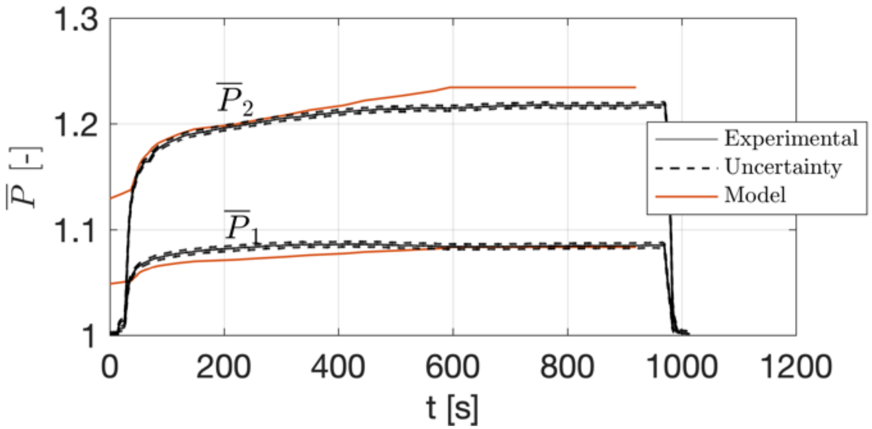
- Fluid film pressures evolve with temperature
- $P_{2\text{-}5}$ linked to and limited by condensation pressure in bearing
- P_1 is below saturation pressure and evolves slower than without condensation
  → upstream condensation

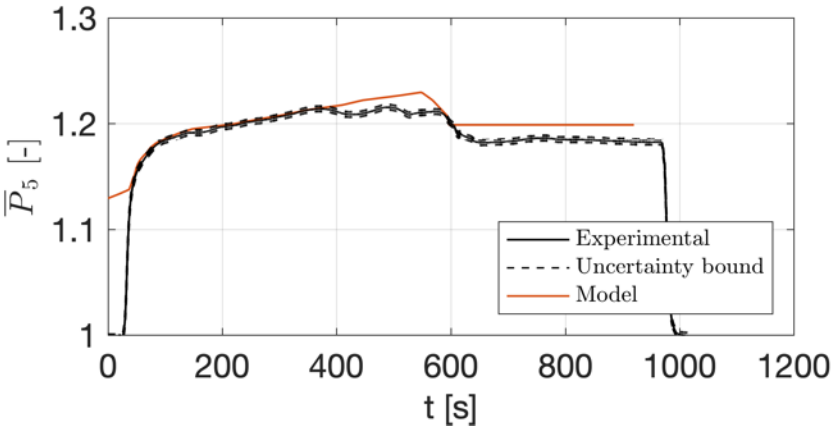




Experimental Results with Condensation (cont.)

- Reynolds equation with real gas effects captures evolution of pressures
- Sudden drop of P_5 due to switch from two to single phase lubrication
- Occurrence of condensation proven experimentally
- Sustained operation of bearing under condensation technically feasible





Practical Implementation of Gas Lubricated Bearings

Implementation of Grooved Bearings

- Gas bearings require very small clearances (5-10 μm for ø10 mm) to ensure stable operation and sufficient load capacity
- Stringent manufacturing and alignment tolerances yield high manufacturing cost
- Design to alleviate manufacturing challenges
- Increase bearing clearance
  - Flexible bushing support with damping
  - Modify fluid film behavior

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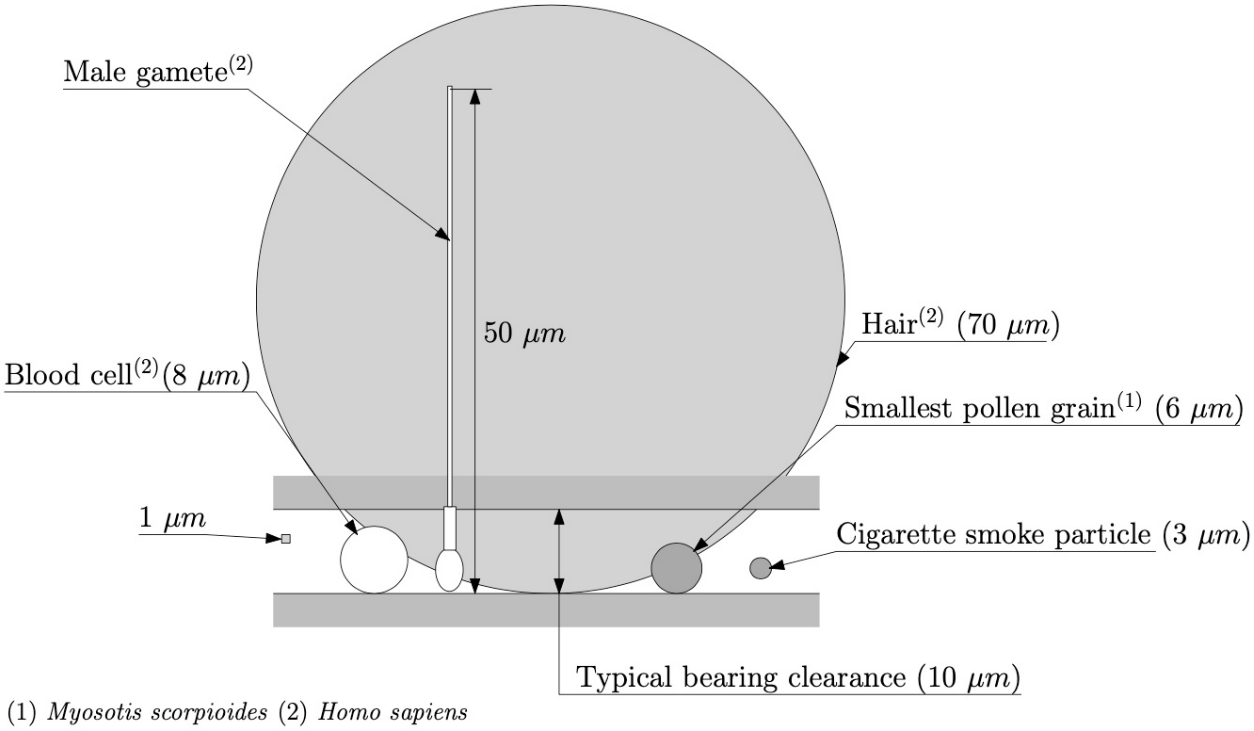
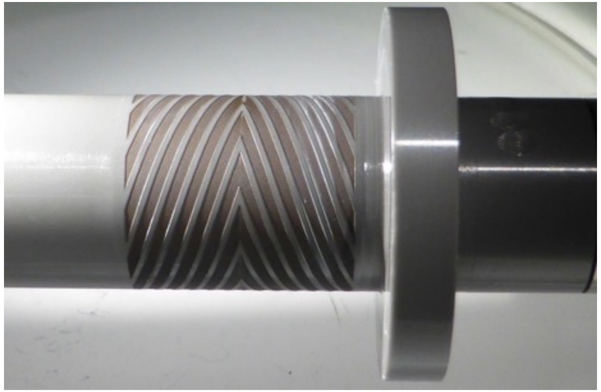
Implementation of Grooved Bearings

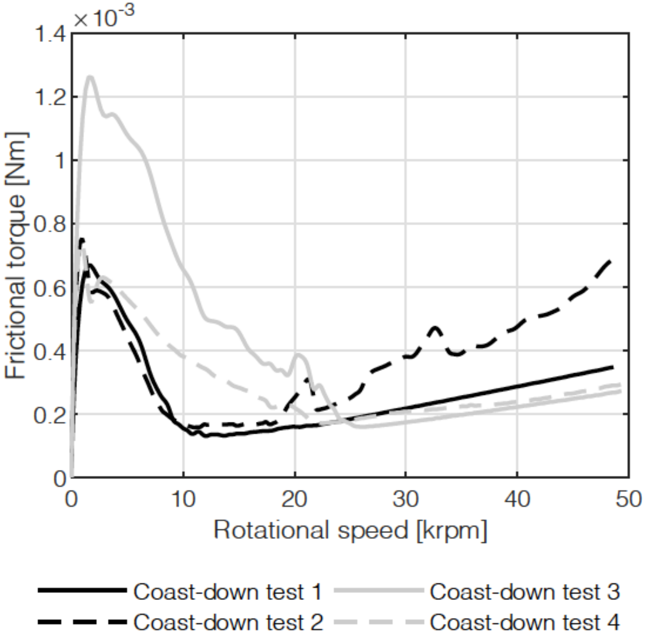
Manufacturing tolerance : ~1 µm



Flexible Bushing Support

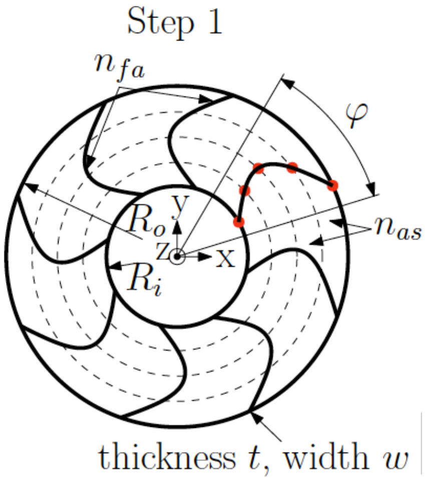
- O-ring as a flexible bushing support with self-aligning features
  - Cheap, compact, availability, high damping
  - Dynamic properties difficult to measure / tune
  - Limited lifetime
  - Non-repeatable assembly

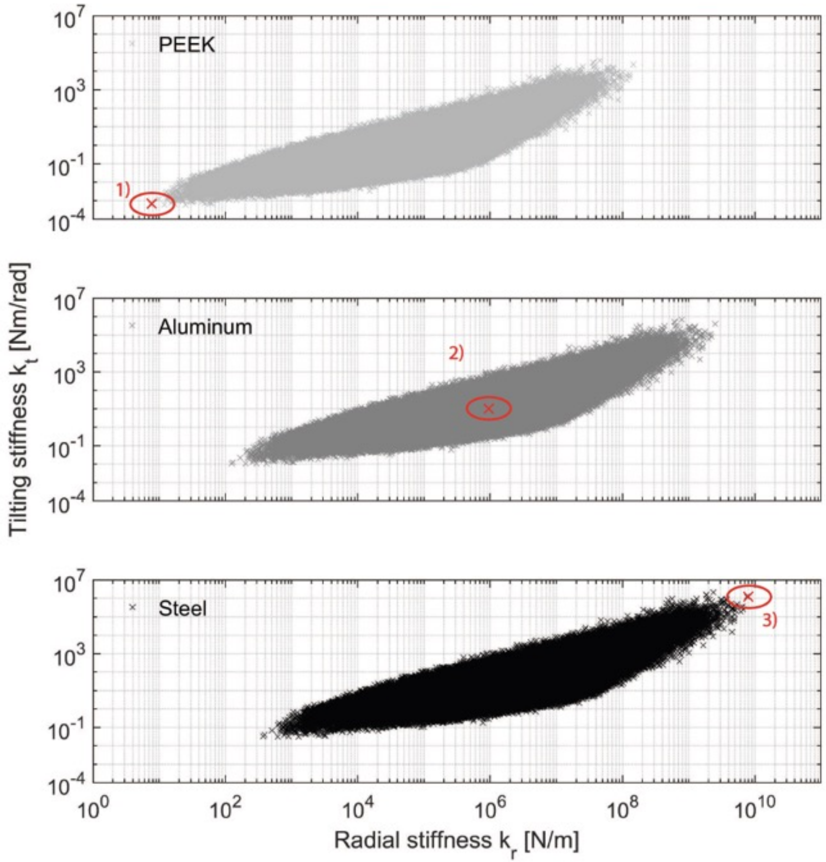




Tunable Membrane Based Support

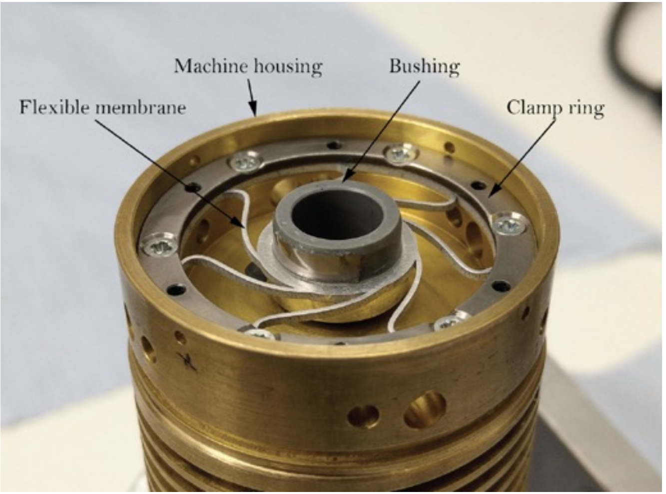
- Membrane design with flexible arms
- Tuning radial and tilting stiffness
- Possibility to introduce damping

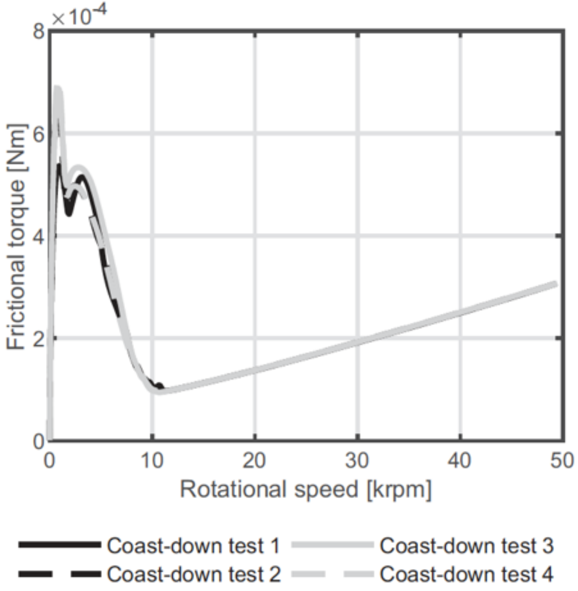




Tunable Membrane Based Support

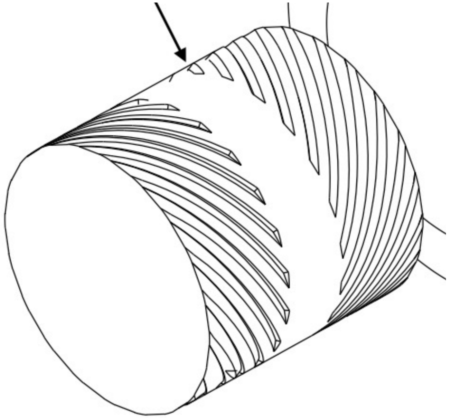
- Assembly procedure with perfect alignment
- Yields totally repeatable results
- Identified unstable bushing tilting mode

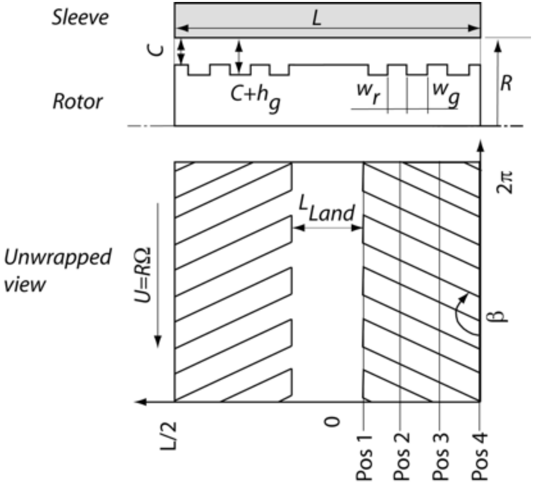




Enhanced Herringbone Grooves Journal Bearings

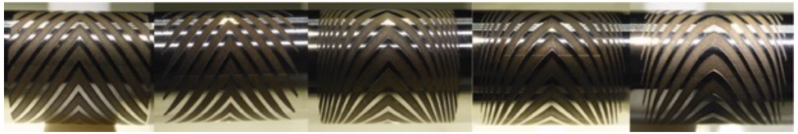
- Traditional grooves of herringbone journal bearing follow helicoid curve with constant angle with constant cross-sectional area
- Idea: introduce variable groove geometry and identify promising patterns by combining models with optimization algorithms

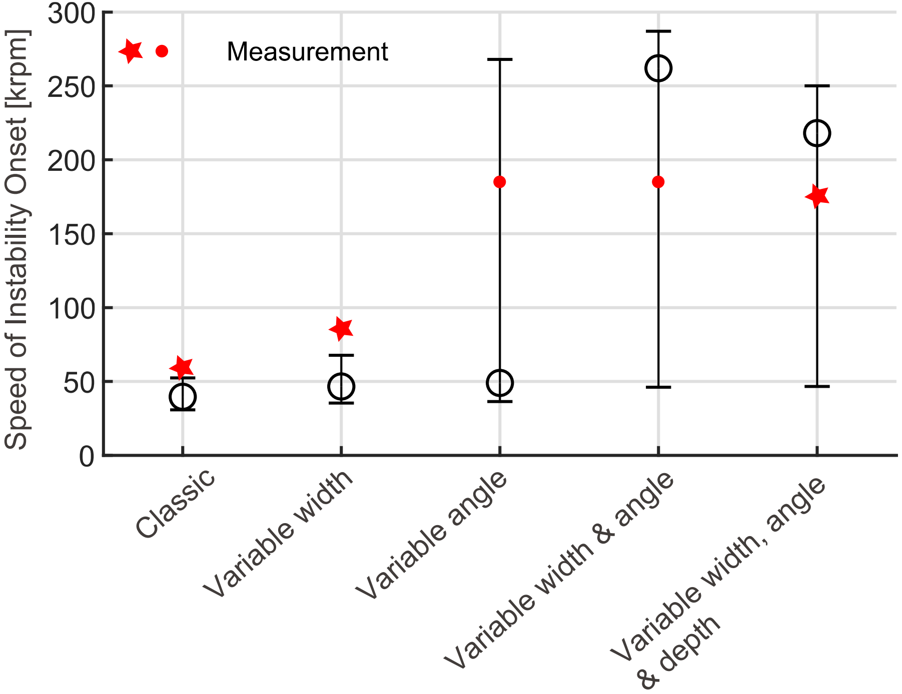




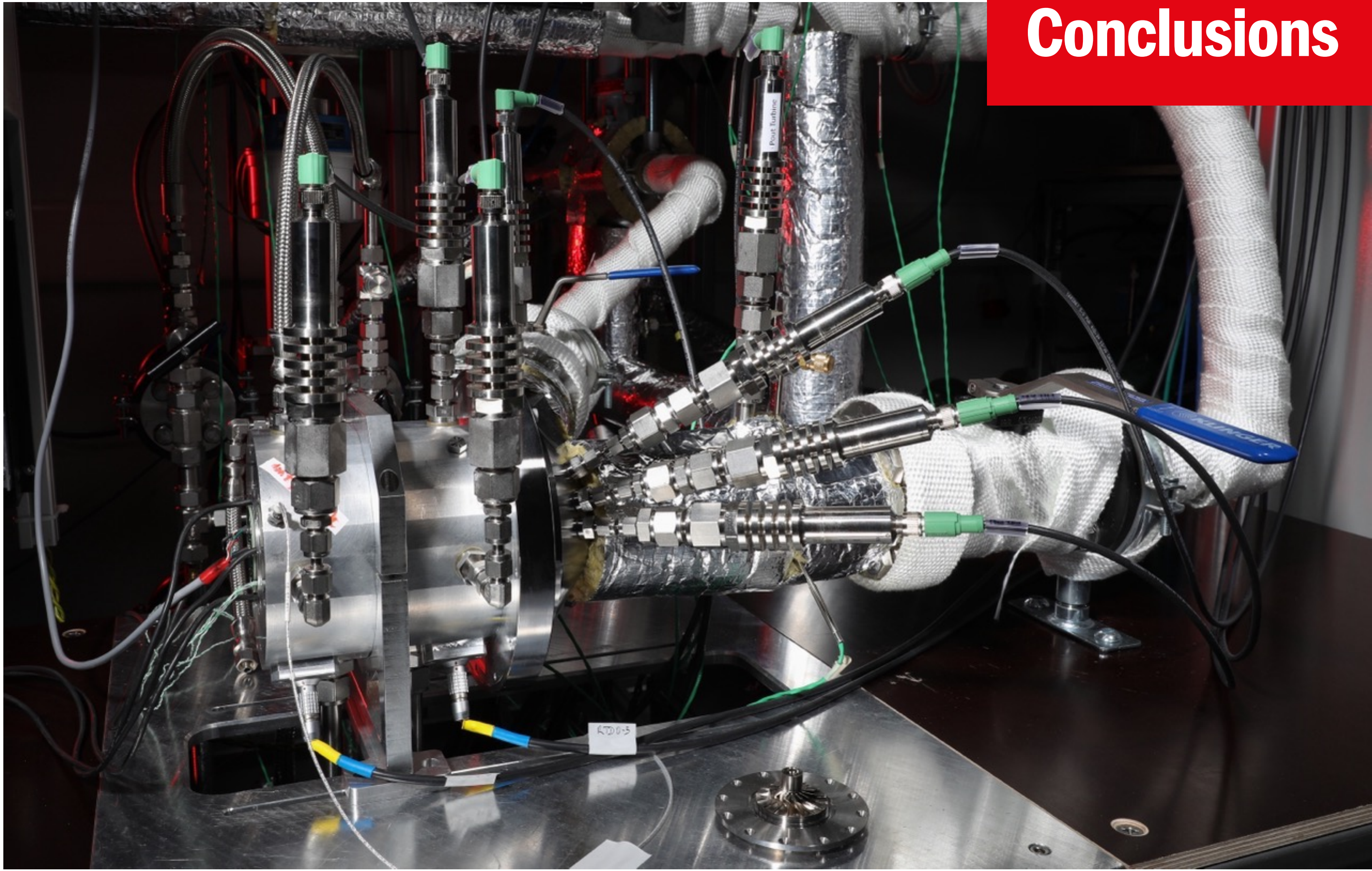
Enhanced Herringbone Grooves Journal Bearings

- Optimized patterns tested on same rotor with same bearing clearance
- Enhanced patterns allows to increase rotor speed x3 compared to classical grooves
- New patterns allow 50% clearance increase





Conclusions



Conclusions

- High rotor speeds and high life time expectations call for gas lubricated bearings
- Gas lubricated bearings can be modeled via Re-equation. Perturbation allows prediction of linearized stiffness and damping matrices
- Grooved bearings with high stability threshold can be approximated by Narrow Groove Theory
- Stringent manufacturing tolerance remains a challenge, additional external damping is helpful

Thank you for your attention

Laboratory for Applied Mechanical Design

Department of Mechanical Engineering

Reynolds Equations

Following the development of material derivative of h:

$$\frac{\mathrm{D}h}{\mathrm{D}t} = \frac{\partial h}{\partial t} + \frac{\partial h}{\partial x}\frac{\mathrm{d}x}{\mathrm{d}t} + \frac{\partial h}{\partial z}\frac{\mathrm{d}z}{\mathrm{d}t}$$

With :

$$\frac{\mathrm{d}x}{\mathrm{d}t} = u_b \quad \frac{\mathrm{d}y}{\mathrm{d}t} = v_b \qquad \frac{\mathrm{D}h}{\mathrm{D}t} = w_b$$

$$\rightarrow \quad \frac{\partial h}{\partial t} = \left(-u_b\frac{\partial h}{\partial x} - v_b\frac{\partial h}{\partial z} + w_b\right)$$