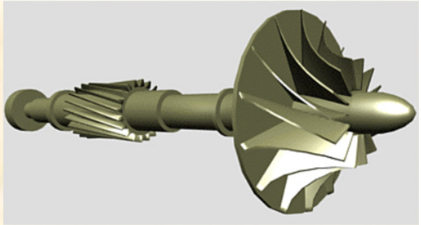
Modern Lubrication Theory FALL 2014 TAMU

Date:



Today's material

Notes

12a

Annular pressure (damper) seals

Stiffness principle. **Seals** as load support elements. Rotordynamic effects.

Notes

12b

Hydrostatic / hydrodynamic bearings

Stiffness principle. Effect of fluid compressibility. Force coefficients

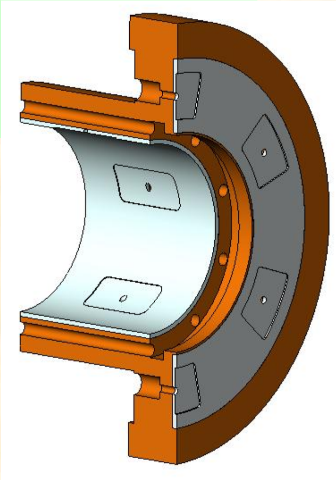
Observations/Announcements

Homework 4, Homework 5 due

Read:

Chupp et al., 2006, AIAA JPP, Childs, Chp 4, pp. 227-284

Damper Seals and Hydrostatic Bearings for Pump Applications



Dr. Luis San Andres
Mast-Childs Professor

Presentation for lectures 12(a) and 12(b)
Based on Lecture (3) delivered at Von Karman Institute (2006)

(Nov 2 update)

Damper Seals & Hydrostatic Bearings

LEARNING OBJECTIVES

a) Physical mechanism for generation of direct stiffness in annular pressure seals & hydrostatic bearings. Select design conditions to obtain maximum (optimum) stiffness

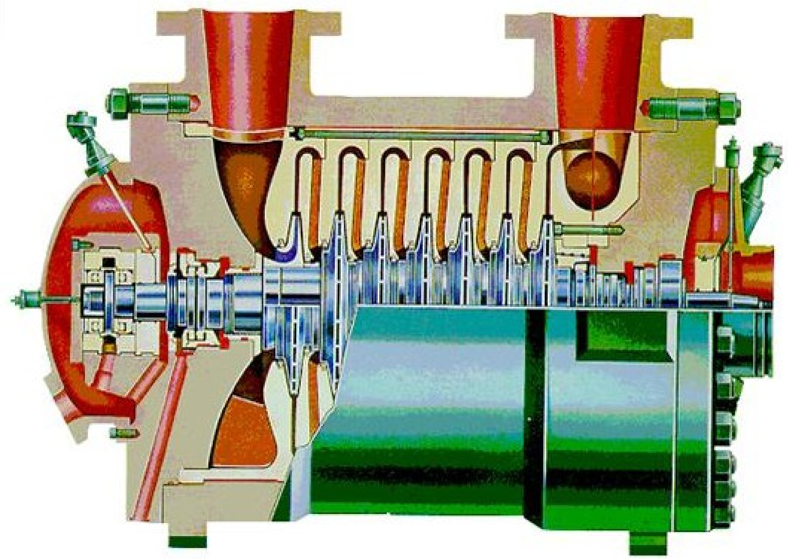
b) Bulk-flow equations for prediction of the flow and force coefficients in annular pressure seals and hydrostatic bearings

c) Predictions for two water seals, long and short, for application as neck ring and interstage seals.
Effect of seal length and inlet swirl on rotordynamic force coefficients

d) Discussion on design of hydrostatic bearing for water pump and to replace oil-lubricated bearings. Effect of angled injection and considerations for improvements in stability margin

Annular pressure seals

Seals **(annular, labyrinth or honeycomb) separate regions of high pressure and low pressure and their principal function is to minimize the leakage (secondary flow); thus improving the overall efficiency of a rotating machine extracting or delivering power to a fluid.**



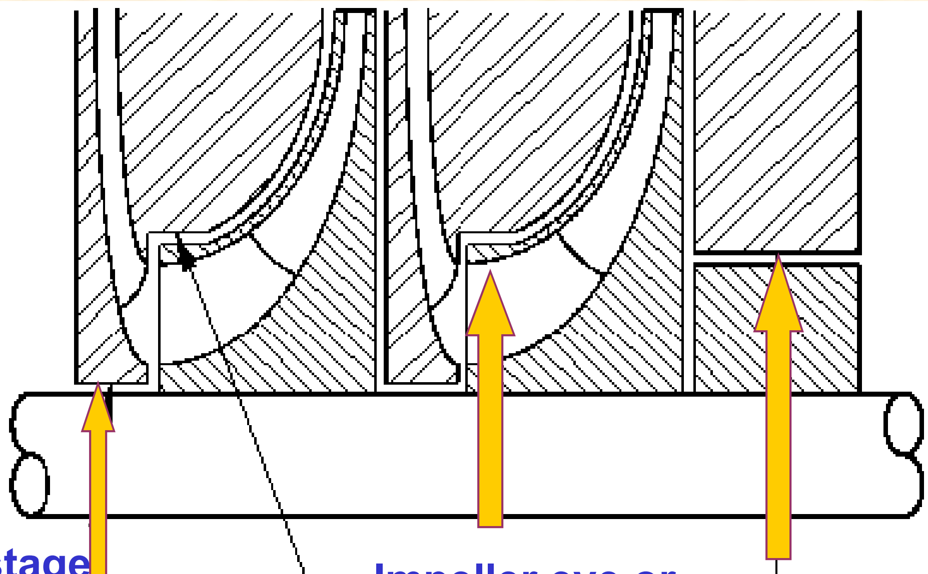
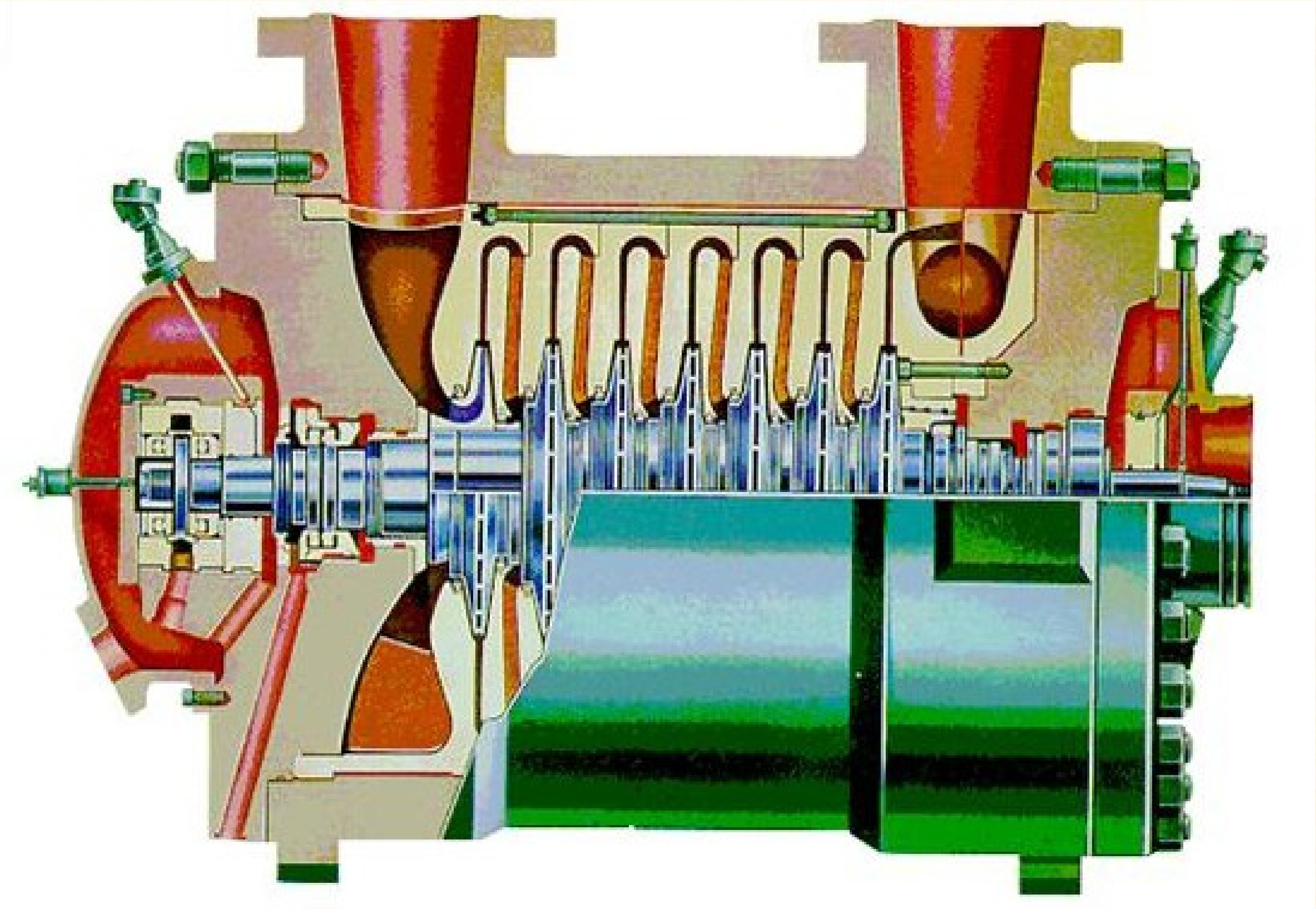
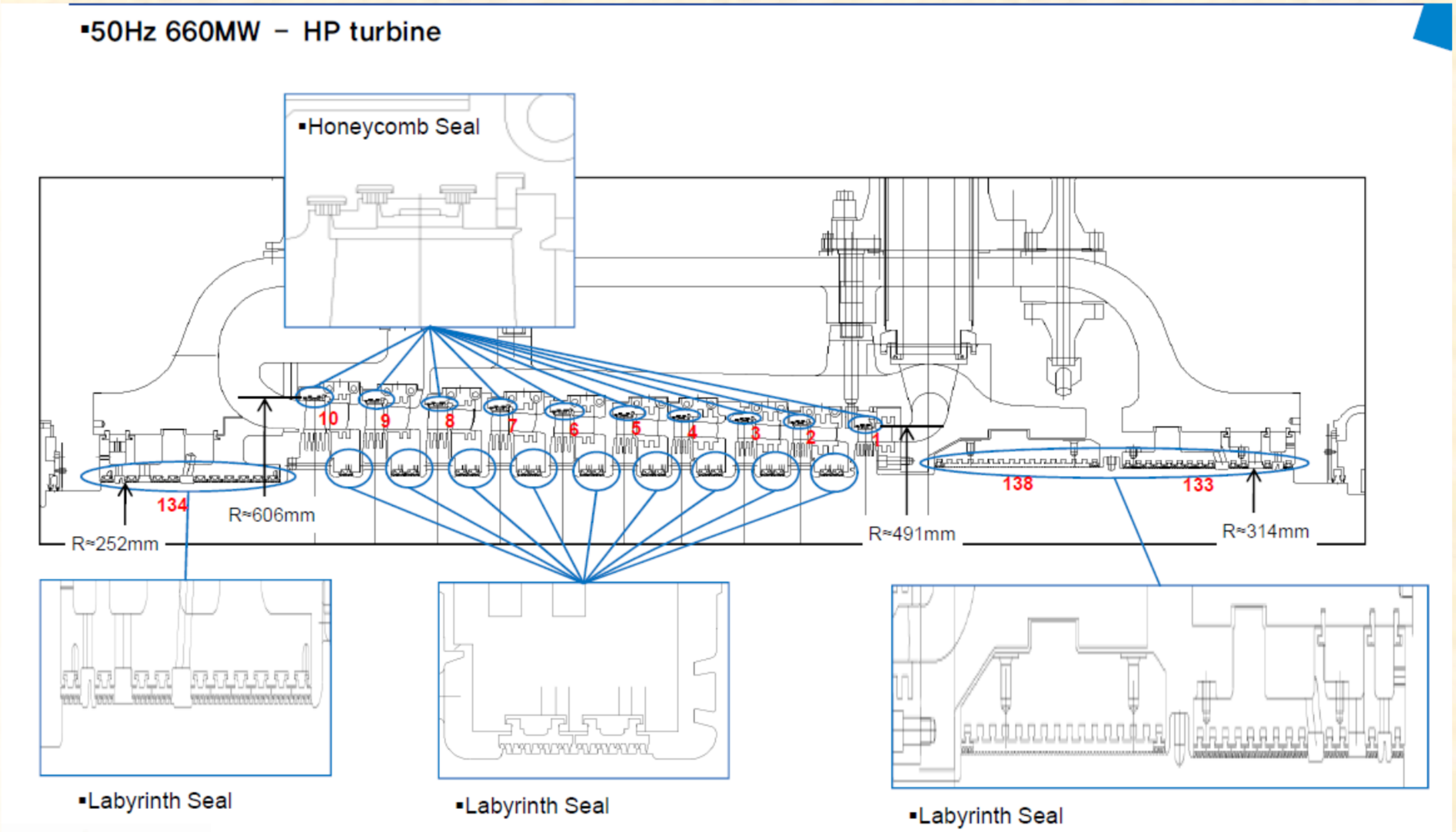


Figure 1 **Seals in a Multistage Centrifugal Pump or Compressor**

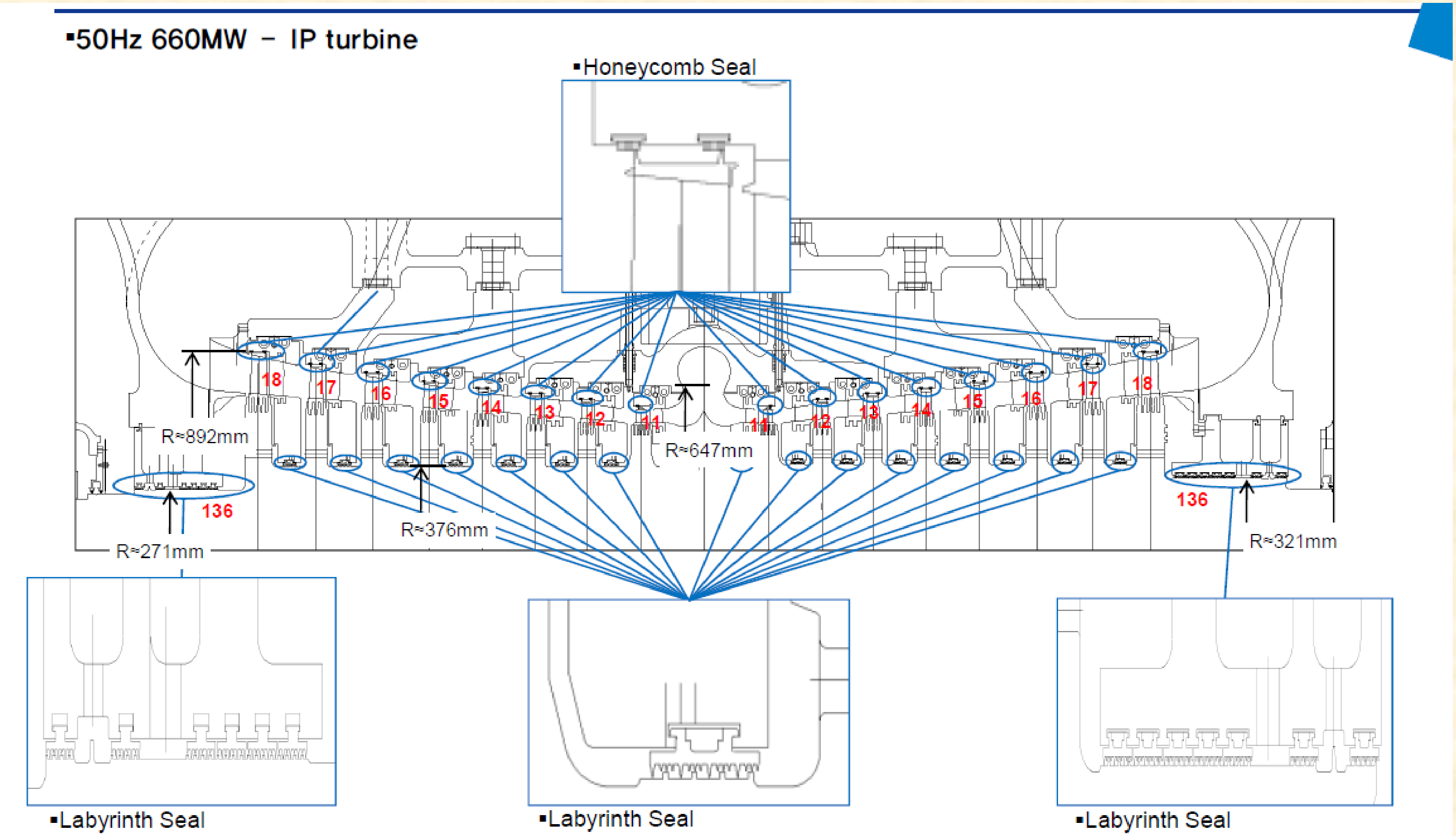
Count the seals…..



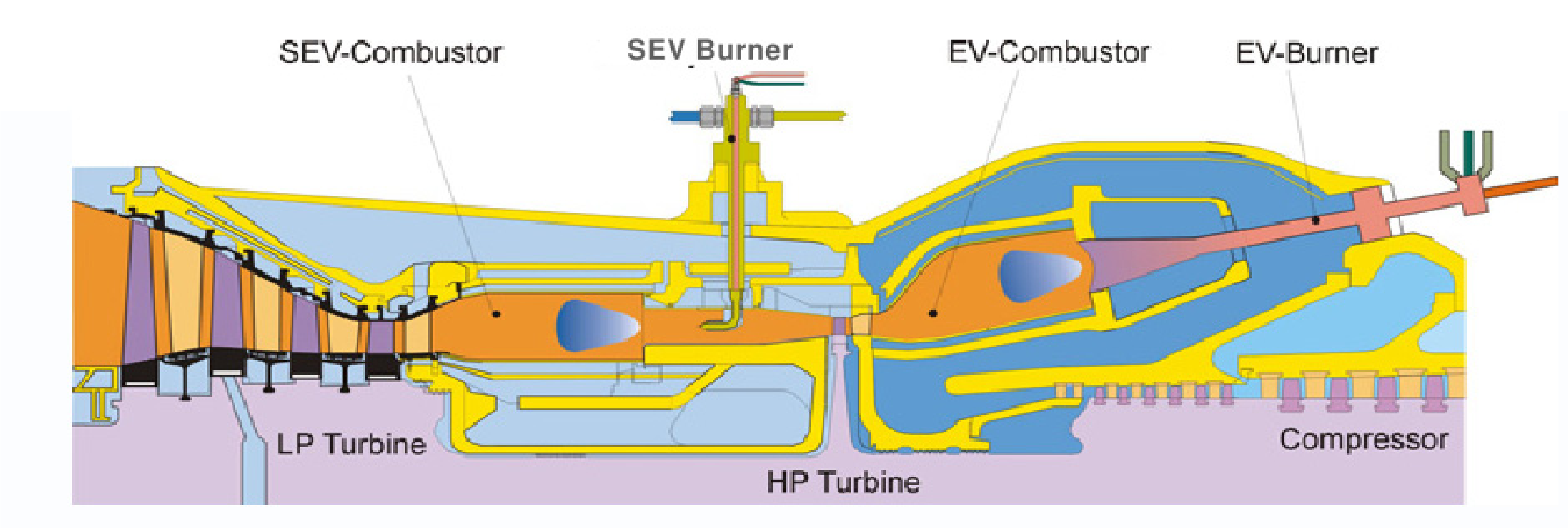
Keep counting seals…..

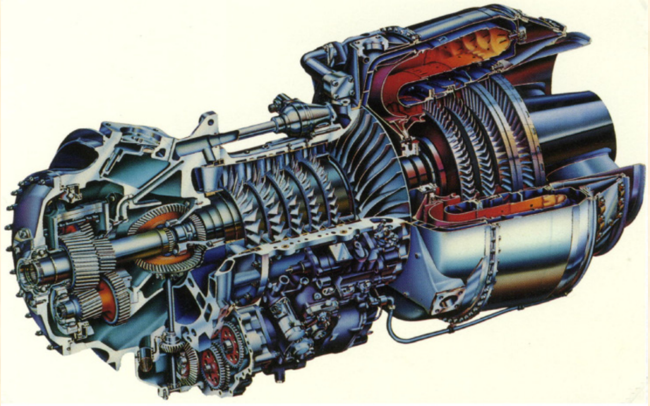


Keep counting more seals…..



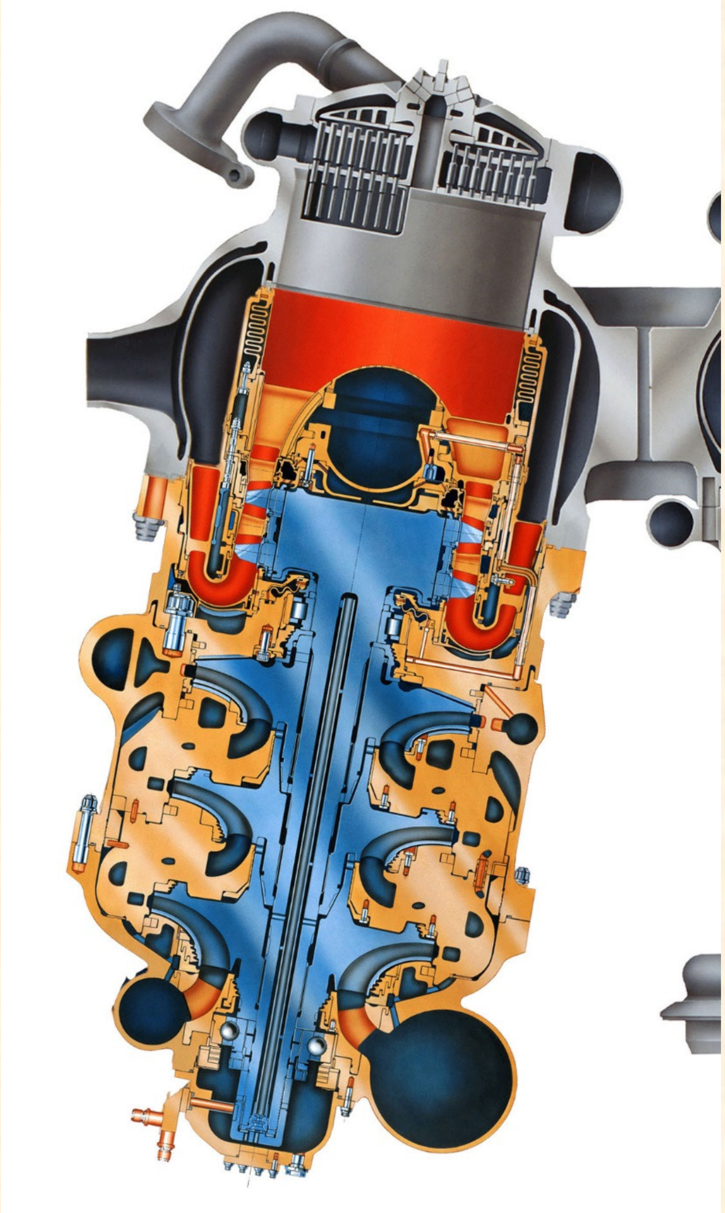
Keep counting………….

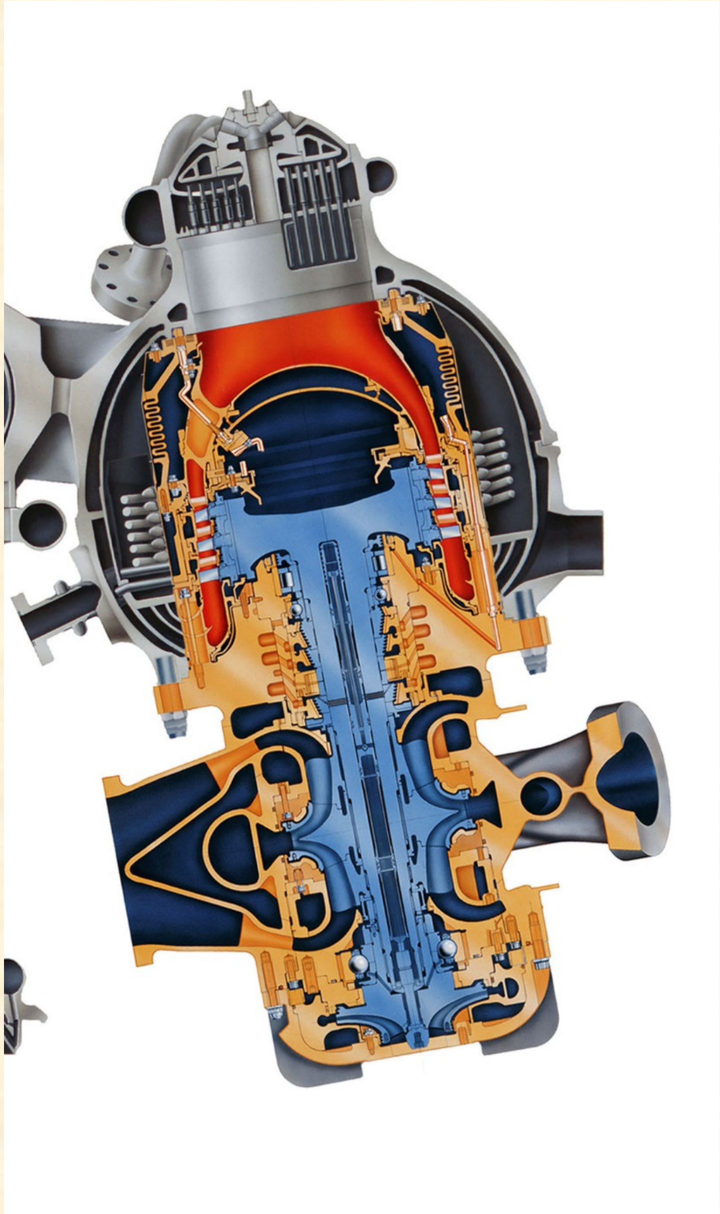




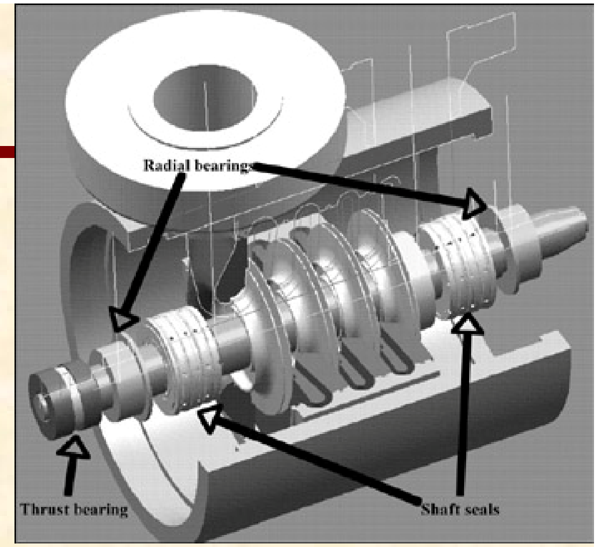
Source: Avco Lycoming

And keep counting seals…..





Annular Pressure Seals



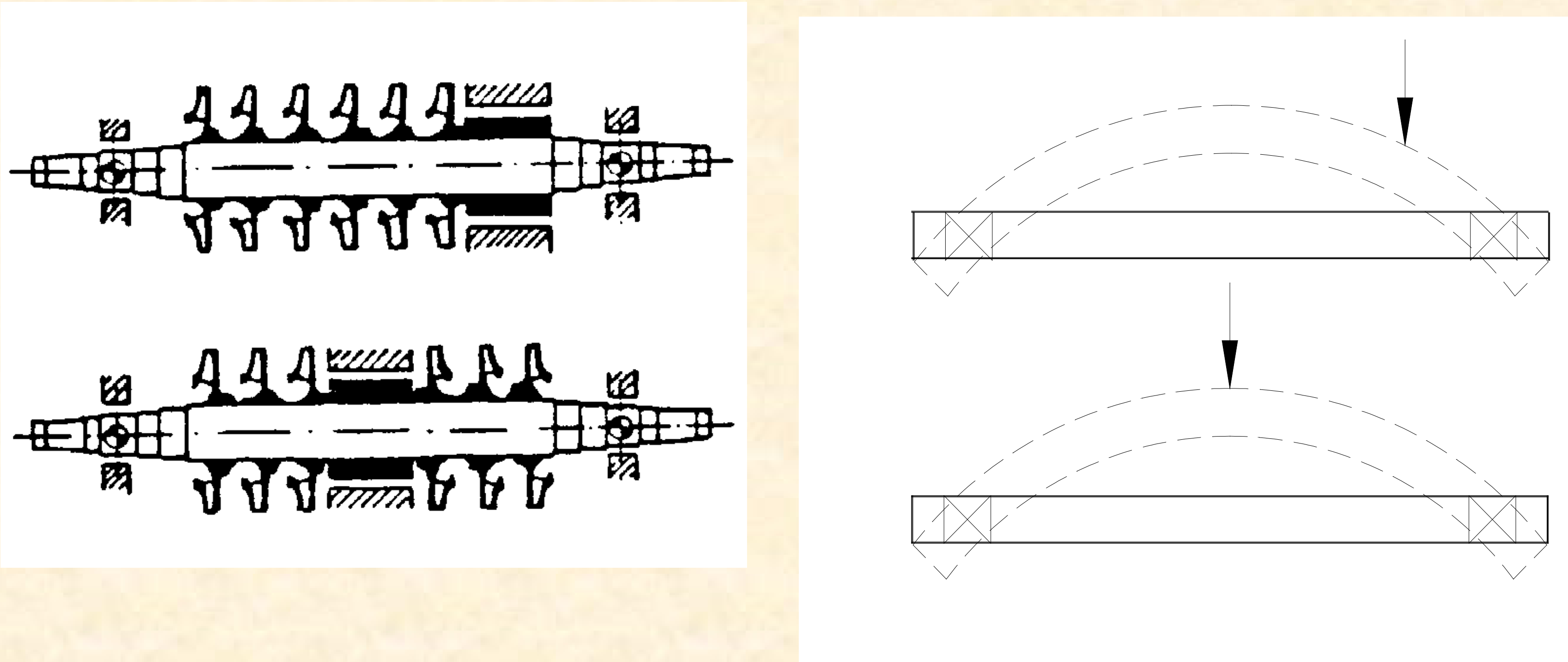
Non-contacting fluid seals are leakage control devices minimizing secondary flows in turbomachines. Seals "use" process liquids of light viscosity as the working fluid.

The dynamic force response of pressure seals has a primary influence on the stability response of high-performance turbomachinery.

Annular seals, although geometrically similar to plain journal bearings, **show a flow structure dominated by turbulence and fluid inertia effects.**

Operating characteristics **unique to seals are * large axial pressure gradients,**
*** large clearance to radius ratio (*R/c*) < 500, while**
*** the axial development of the circumferential velocity determines the magnitude of cross-coupled (hydrodynamic) forces.**

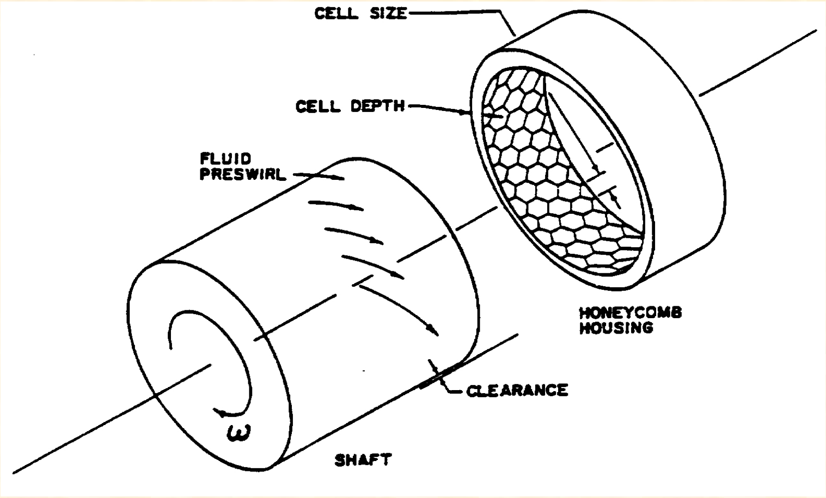
Annular Pressure Seals

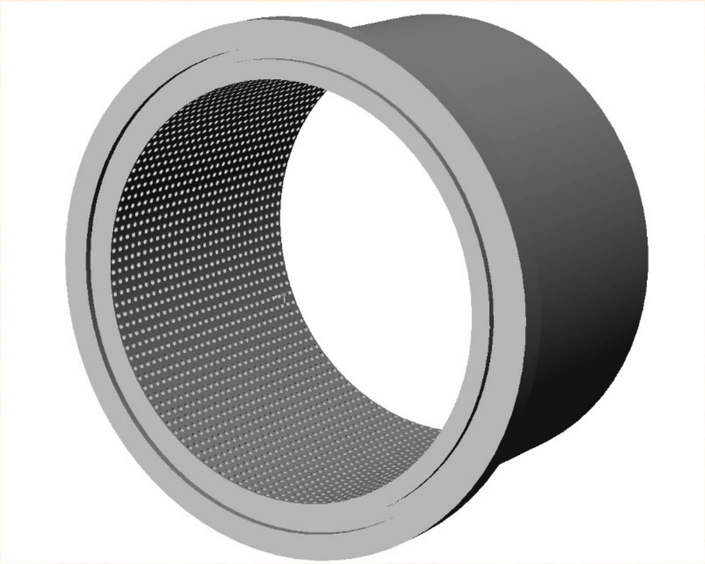


Due to their relative position within a rotor-bearing system, **seals** modify sensibly the system dynamic behavior. **Seals** typically "see" large amplitude rotor motions. This is particularly important in **back-to-back compressors** and **long-flexible multiple stage pumps**.

Figure 1 Straight-Through and Back-to-back Compressors and 1st Mode Shapes

Annular pressure seals





Intentionally roughened stator surfaces (macro texturing) reduce the impact of undesirable cross-coupled dynamic forces and improve seal stability.

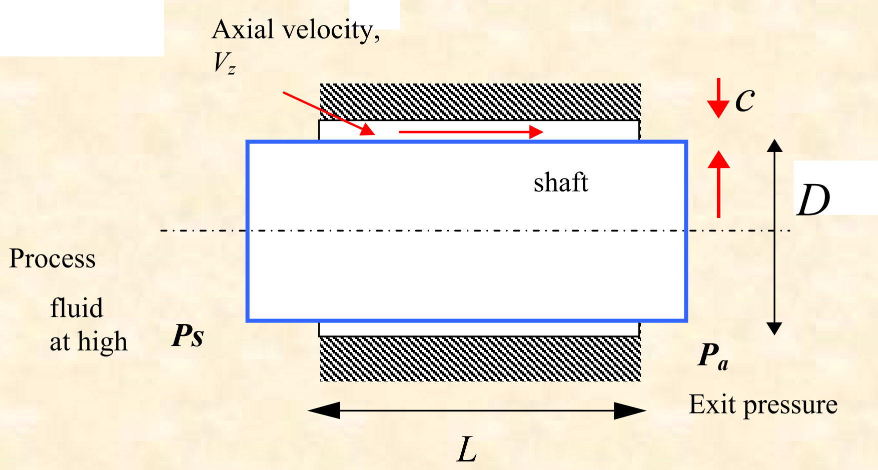
These seals are common practice in current damper seal technology for cryogenic turbo pumps.

Annular seals acting as Lomakin bearings have potential as support elements (damping bearings) in high speed cryogenic turbo pumps as well in process fluid applications.

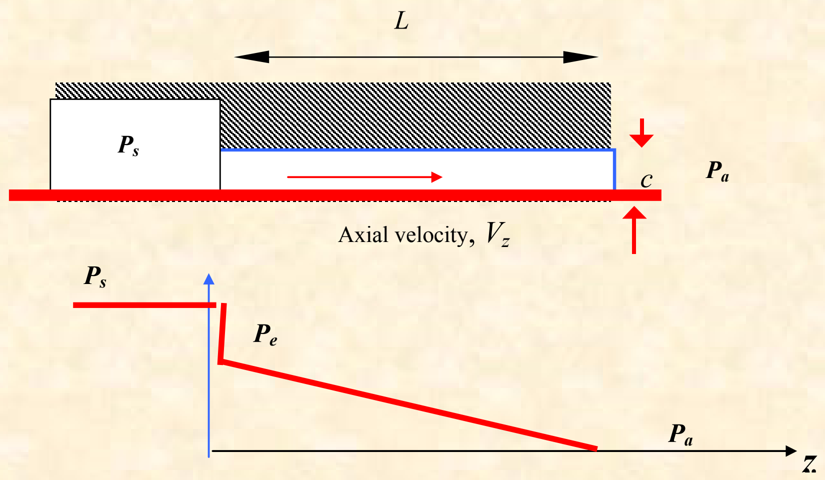
Figure 3 **Honeycomb and round hole pattern seals for turbopump**

The Lomakin effect in pressurized seals

The Lomakin Effect: generation of support (direct) stiffness



4: Geometry of an annular pressure seal



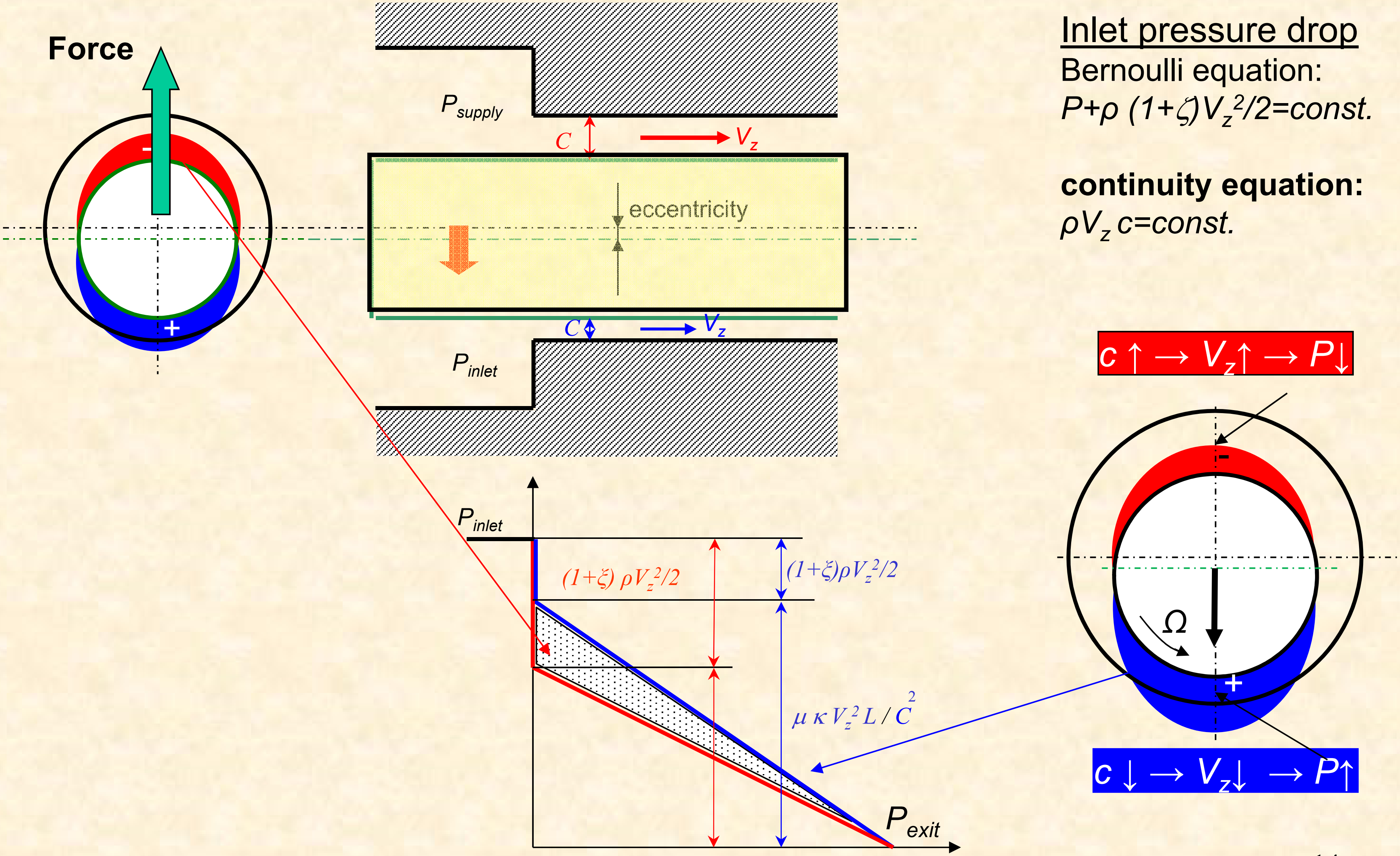
5: Inertial pressure drop due to a sudden contraction

The direct stiffness is due to the pressure drop at the seal inlet plane and its close interaction with the pressure drop (and flow resistance) within the seal lands (even without shaft rotation).

The entrance effect is solely due to fluid inertia accelerating the fluid from the upstream (stagnant) pressure supply to a flow with a high axial speed and reduced static pressure.

Figures 4 & 5

The Lomakin effect: a centering stiffness



<u>Inlet pressure drop</u>

Bernoulli equation:

$P+\rho\,(1+\zeta)V_z^2/2=const.$

continuity equation:

$\rho V_z\, c=const.$

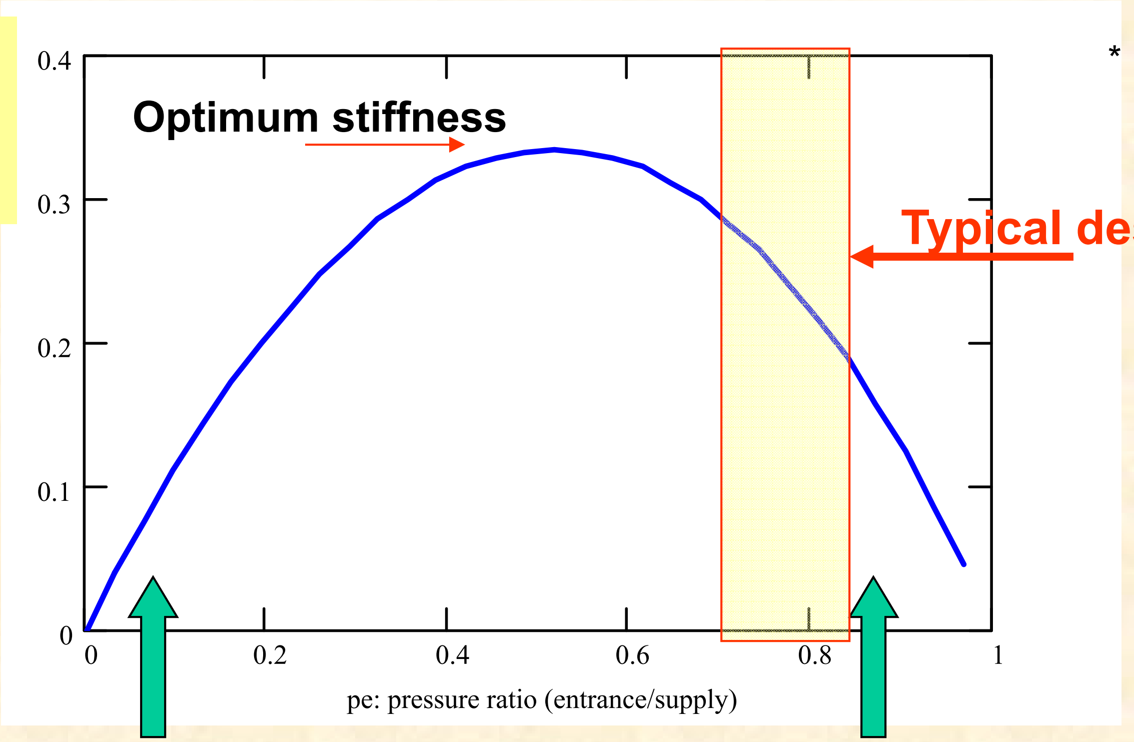
$c\uparrow \rightarrow V_z\uparrow \rightarrow P\downarrow$

$c\downarrow \rightarrow V_z\downarrow \rightarrow P\uparrow$

Original as per Dr. M. Arghir presentation (2006)

Centering stiffness from an annular seal

$$\overline{K} = \frac{K}{\frac{B\,L}{2c_o}\left(P_s - P_a\right)}$$



<u>pressure drop at inlet</u>

Too large clearance

Or short seal

<u>pressure drop in land</u>

Too tight clearance

Or Long seal

Seal wear enlarges clearance and increases leakage

Figure 6 **Stiffness versus entrance pressure ratio (simple model)**

Annular Pressure Seals

The Lomakin effect

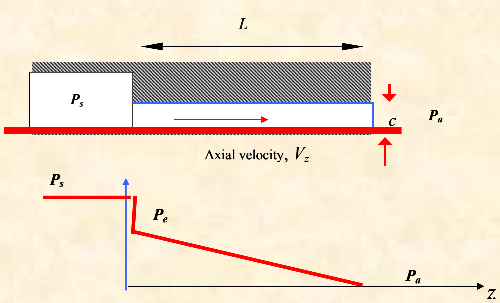
Incompressible fluid, turbulent flow

$$V_z^2 = \frac{P_s - P_a}{\frac{\rho}{2}(1+\zeta) + \rho \frac{L}{c} f_z}$$

Axial velocity (leakage) decreases with increase of inlet loss (ζ) and friction factor (f_z) and length of land in seal.

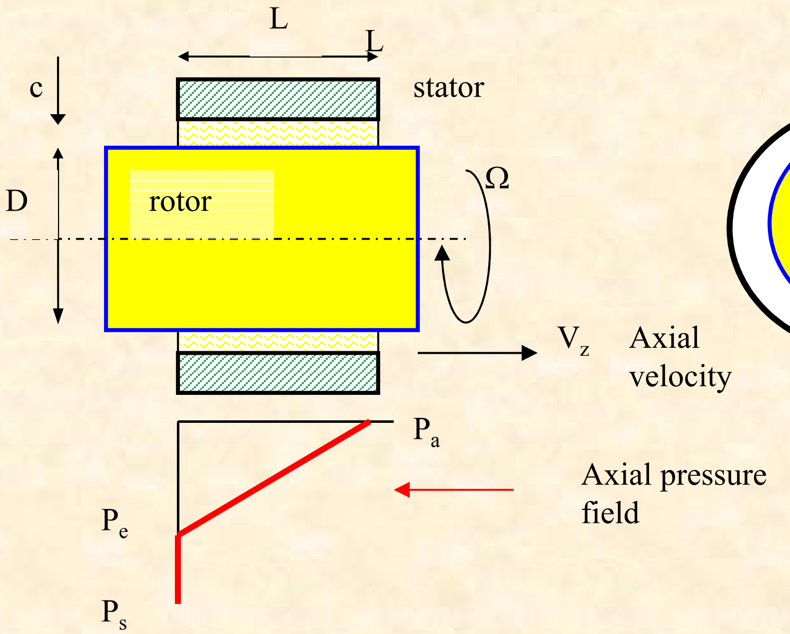
$$P_e - P_a = \frac{P_s - P_a}{\left\{1 + \frac{(1+\zeta)}{2 f_z} \frac{c}{L}\right\}}$$

Entrance pressure into seal decreases as inlet loss (ζ) increases and as land friction factor (f_z) ***decreases or land length increases***

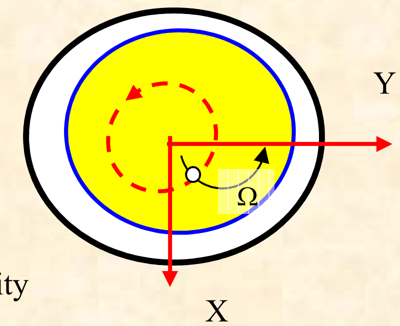


5: Inertial pressure drop due to a sudden contraction

Force Coefficients in Annular Seals



D Depiction of an annular pressure seal

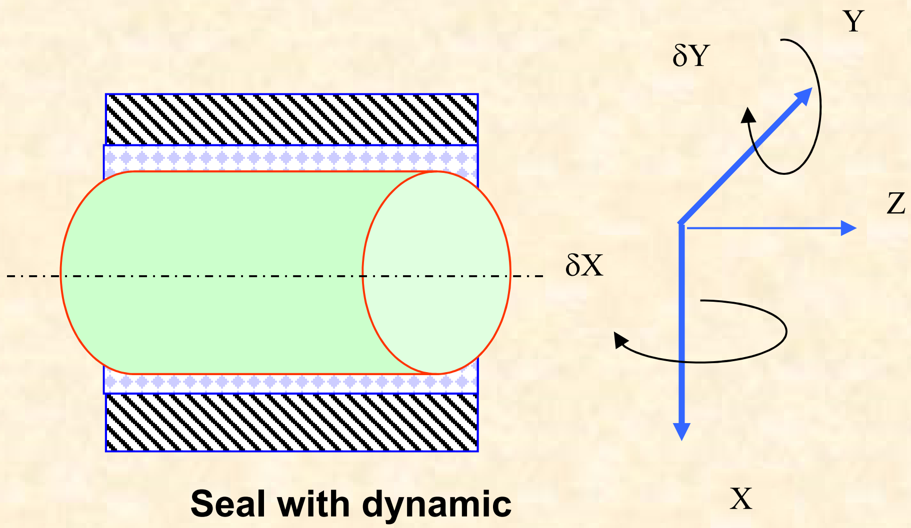


Seal reaction forces are functions of the fluid properties, flow regime, operating conditions and geometry.

For small amplitudes of rotor lateral motion: forces represented with linearized stiffness, damping and inertia force coefficients:

$$-\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}\begin{bmatrix} X \\ Y \end{bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}\begin{bmatrix} \dot{X} \\ \dot{Y} \end{bmatrix} + \begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix}\begin{bmatrix} \ddot{X} \\ \ddot{Y} \end{bmatrix}$$

Force/Moment coefficients in seals



Seal with dynamic translations (X,Y) and angulations (δX, δY)

$$
\begin{bmatrix} F_X \\ F_Y \\ M_X \\ M_Y \end{bmatrix} = -
\begin{bmatrix}
K_{XX} & K_{XY} & K_{X\delta X} & K_{X\delta Y} \\
K_{YX} & K_{XX} & K_{X\delta Y} & K_{X\delta X} \\
K_{\delta XX} & K_{\delta XY} & K_{\delta X\delta X} & K_{\delta X\delta Y} \\
-K_{\delta XY} & K_{\delta XX} & K_{\delta X\delta Y} & K_{\delta X\delta X}
\end{bmatrix}
\begin{bmatrix} \Delta e_X \\ \Delta e_Y \\ \delta_X \\ \delta_Y \end{bmatrix}
-
\begin{bmatrix}
C_{XX} & C_{XY} & C_{X\delta X} & C_{X\delta Y} \\
C_{YX} & C_{XX} & C_{X\delta Y} & C_{X\delta X} \\
C_{\delta XX} & C_{\delta XY} & C_{\delta X\delta X} & C_{\delta X\delta Y} \\
C_{\delta XY} & C_{\delta XX} & C_{\delta X\delta Y} & C_{\delta X\delta X}
\end{bmatrix}
\begin{bmatrix} \Delta \dot{e}_X \\ \Delta \dot{e}_Y \\ \delta_{\dot{X}} \\ \delta_{\dot{Y}} \end{bmatrix}
-
\begin{bmatrix}
M_{XX} & M_{XY} & M_{X\delta X} & M_{X\delta Y} \\
M_{YX} & M_{XX} & M_{X\delta Y} & M_{X\delta X} \\
M_{\delta XX} & M_{\delta XY} & M_{\delta X\delta X} & M_{\delta X\delta Y} \\
M_{\delta XY} & M_{\delta XX} & M_{\delta X\delta Y} & M_{\delta X\delta X}
\end{bmatrix}
\begin{bmatrix} \Delta \ddot{e}_X \\ \Delta \ddot{e}_Y \\ \delta_{\ddot{X}} \\ \delta_{\ddot{Y}} \end{bmatrix}
$$

Seals also generate moment coefficients due to tilts (angulations) of rotor.

Most complete model:
16 stiffness, 16 damping and 16 inertia coefficients

Figure 7

Bulk-flow Analysis: governing equations

Flow Continuity

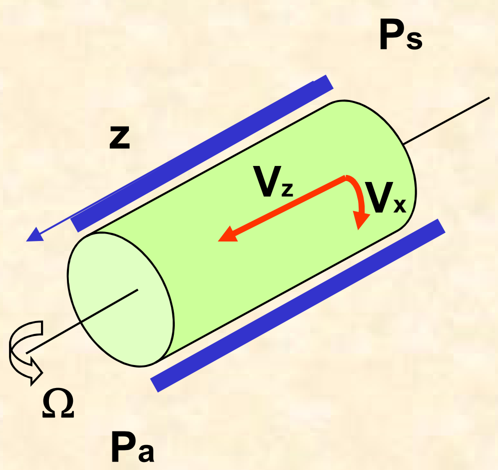
$$\frac{\partial}{\partial x}\left(hV_x\right)+\frac{\partial}{\partial z}\left(hV_z\right)+\frac{\partial h}{\partial t}=0$$

Circumferential Momentum transport

$$-h\frac{\partial P}{\partial x}=\frac{\mu}{h}\left(\kappa_x V_x-\kappa_J\frac{U}{2}\right)+\rho h\left\{\frac{\partial V_x}{\partial t}+\frac{\partial V_x^2}{\partial x}+\frac{\partial V_x V_z}{\partial z}\right\}$$

Axial momentum transport

$$-h\frac{\partial P}{\partial z}=\frac{\mu}{h}\kappa_z V_z+\rho h\left\{\frac{\partial V_z}{\partial t}+\frac{\partial V_x V_z}{\partial x}+\frac{\partial V_z^2}{\partial z}\right\}$$



- Turbulent flow with fluid inertia effects
- Mean flow velocities – average across film (h)
- No accounting for strong recirculation zones

Bulk-flow analysis: inlet conditions

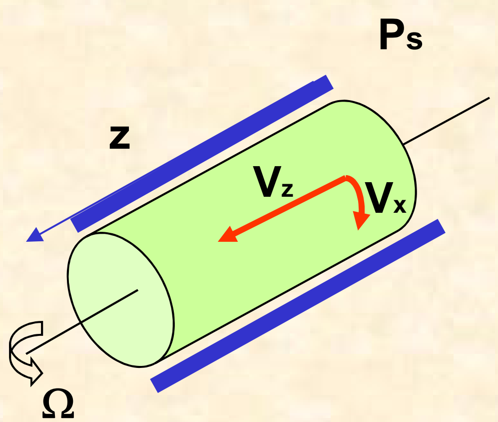
Boundary Conditions

- -Inlet pressure loss due to fluid inertia (Lomakin effect)
- - Inlet swirl determined by upstream condition (implementation of swirl-brakes)
- -Exit pressure without recovery loss, typically

$$P_e = P_s - \frac{1}{2}\rho(1+\xi)V_z^2, \quad V_x = \alpha\Omega R$$

Numerical Solution

- -Analytical solutions for short length seals (centered condition) – See Childs textbook
- -Numerical solutions for realistic geometries use CFD techniques (staggered grids, upwinding, etc) and predict (4 or 16) **K,C,M** force/moment coefficients.



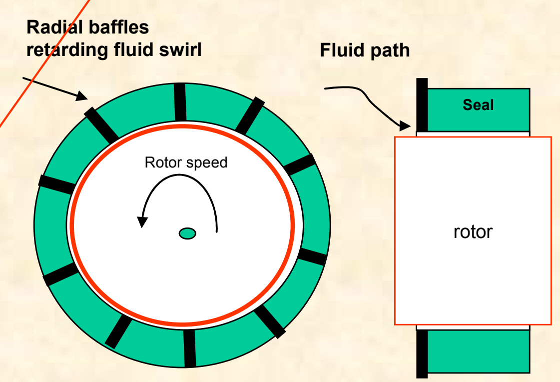


Figure 8 Anti swirl brake at inlet or pressure seal

Dynamic forced performance of annular seals

Example: (centered seal)

Centered seal *(e=0)*:
No static load

Fluid: water at 30°C (0.792 cPoise, 995 kg/m3)

D = 152. 4 mm (6 inch),

Short seal (neck ring seal) *L/D*=0.20
Long seal (inter stage seal) *L/D*= 0.50

neck ring seal &
inter stage seal or balance piston

c = 0.190 mm,
nominal clearance & worn clearance (2*c*)

Standard design practice: find influence of seal wear on leakage and rotordynamic coefficients

smooth rotor and stator surfaces

Nominal speed = 3600 rpm,
Pressure drop 34.4 bar

Pressure drop $\Delta P \sim RPM^2$

Inlet loss coefficient ξ=0.1

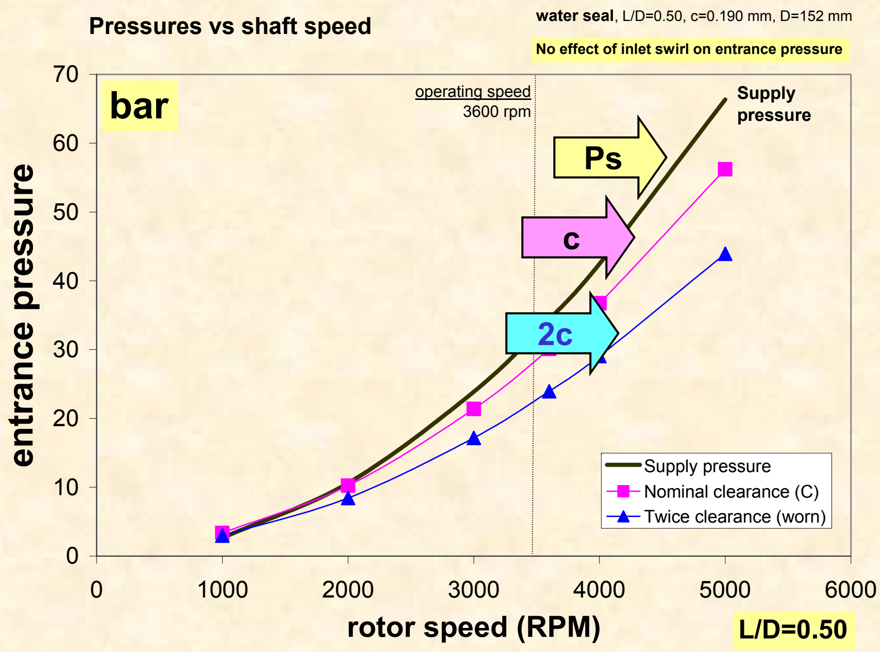
Inlet swirl α=0.5 and 0.0
(without and with swirl break)

Influence on cross-coupled stiffness and stability

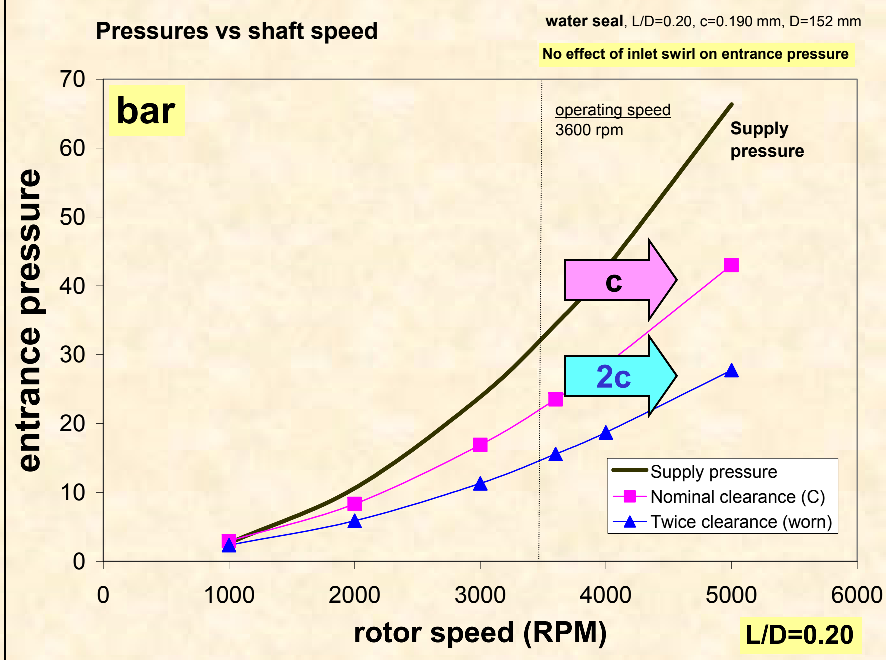
Table 1 Geometry and operating conditions of water seals in a liquid pump

Entrance pressure into seal

LONG SEAL



SHORT SEAL

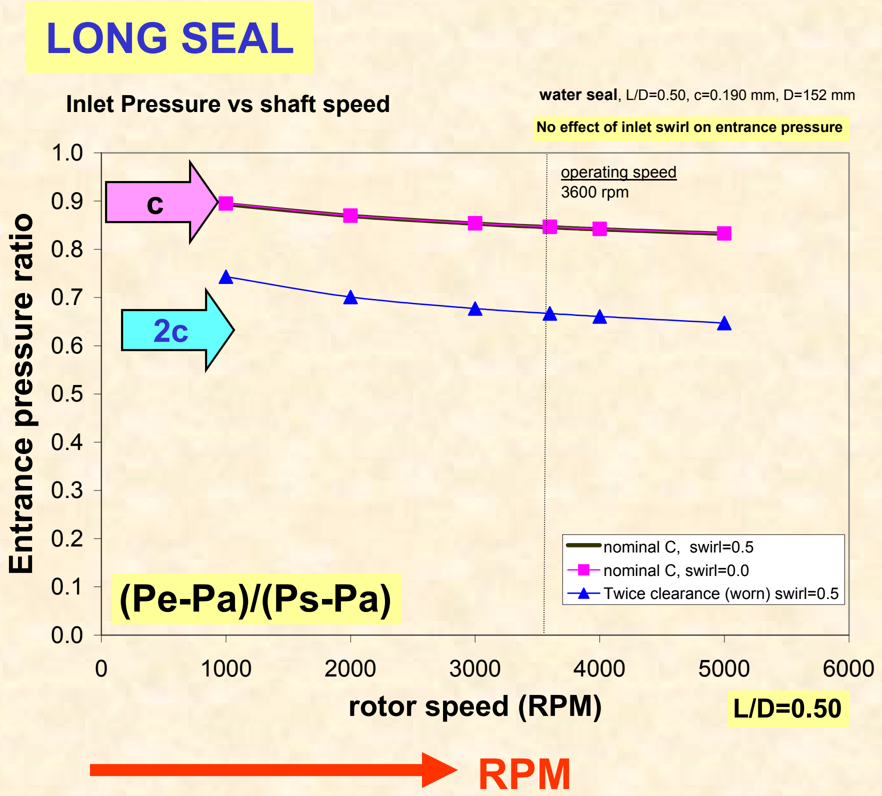


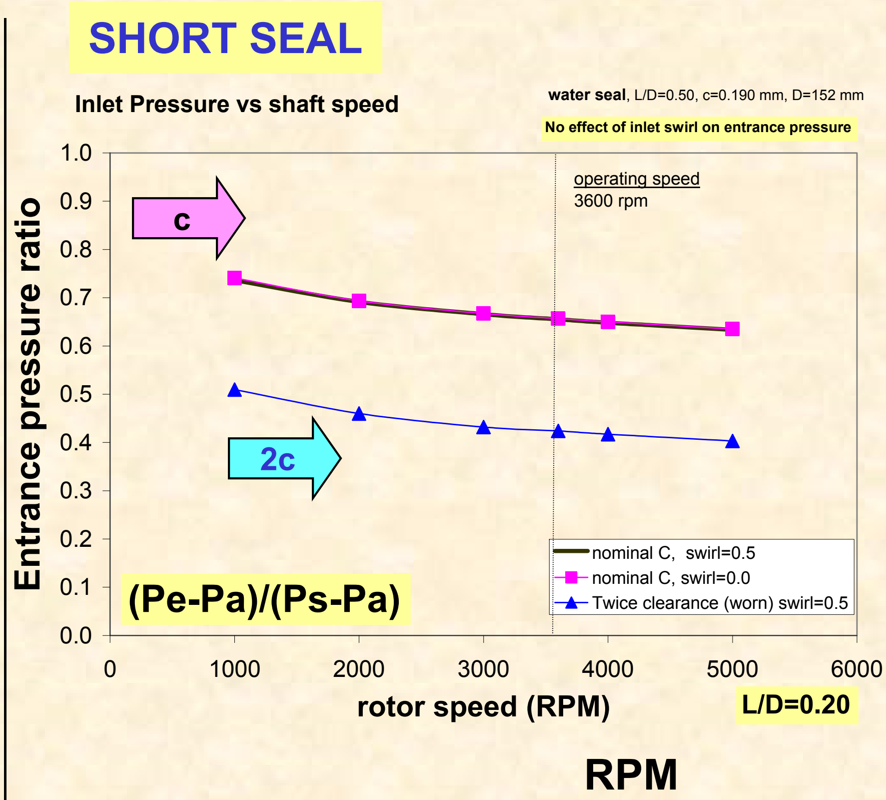
RPM

- Worn seal leads to lower entrance pressure loss due to increase in flow rate (reduction in land flow resistance)
- Short seal leads has lowest entrance pressure due to increase in leakage

Figure 9 Supply and entrance pressures for two water seals, *L/D*=0.50 and 0.20, and two clearances (*c* and 2*c*)

Entrance pressure ratio into seal



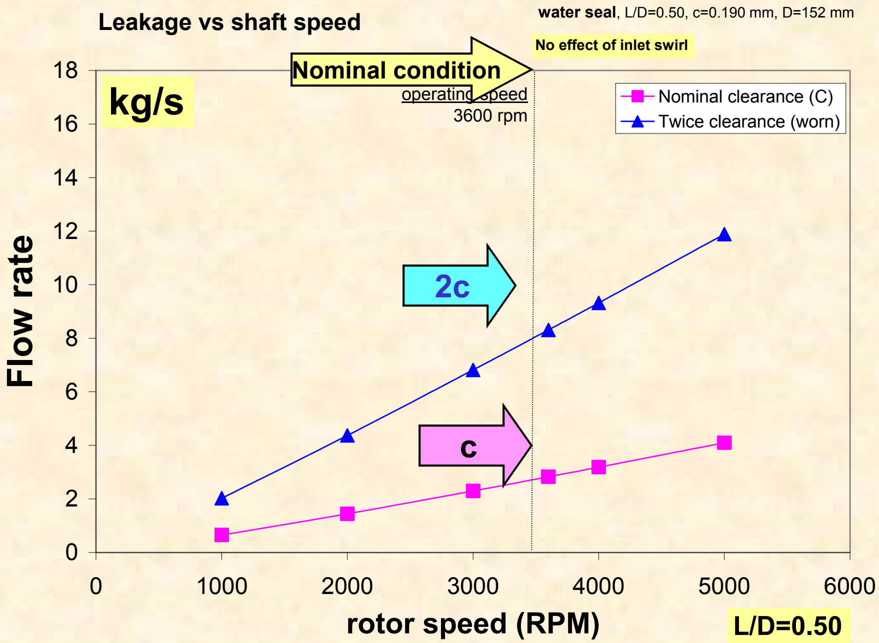


- Worn seal leads to higher entrance pressure loss due to increase in flow rate (reduction in land flow resistance)
- Short seal leads to even larger inlet pressure loss due to increase in leakage

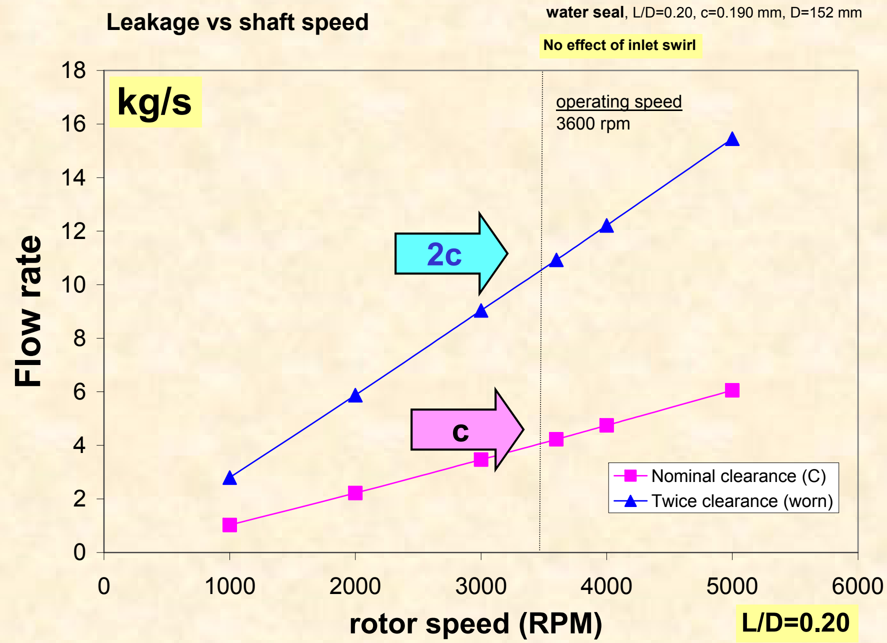
Figure 9 Entrance pressure ratio for two water seals and two clearances (*c* and 2*c*)

Seal leakage

LONG SEAL



SHORT SEAL



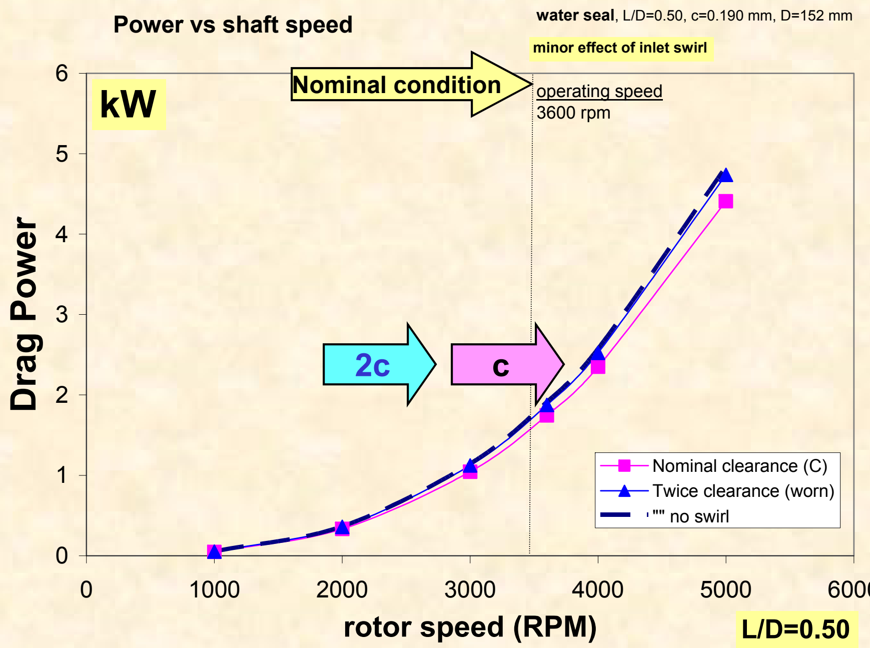
RPM & P_{supply}

- **Leakage is proportional to RPM ~ $\Delta P^{1/2}$ & not proportional to clearance**
- **Worn seal leaks more – Longer seal leaks less**
- **No influence of inlet swirl**

Figure 10 **Leakage (flow rate) for two water seals, *L/D*=0.50 & 0.20, and two clearances**

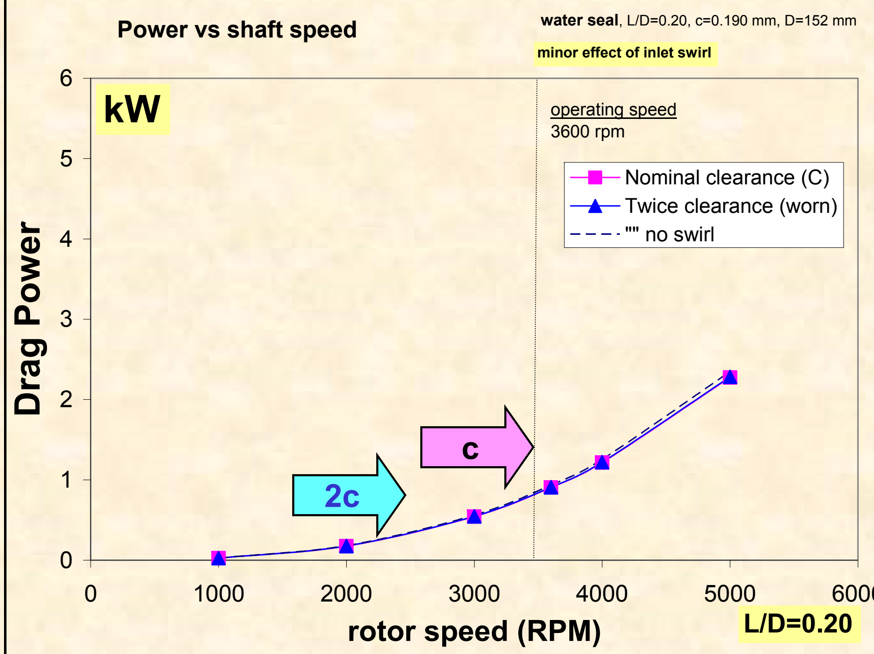
Drag power

LONG SEAL



RPM & P_{supply}

SHORT SEAL



- Drag Power ~ RPM^2
- Long seal draws more power (~ 5 times)
- No effect of clearance. As *C* increases, so does circ. Reynolds #
- No effect of inlet swirl

Figure 11 Drag Power for two water seals, *L/D*=0.50 & 0.20, and two clearances

Rotordynamic coefficients – lateral motions

Seal reaction forces:

$$-\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}\begin{bmatrix} X \\ Y \end{bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}\begin{bmatrix} \dot{X} \\ \dot{Y} \end{bmatrix} + \begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix}\begin{bmatrix} \ddot{X} \\ \ddot{Y} \end{bmatrix}$$

Reduced model: Centered Condition

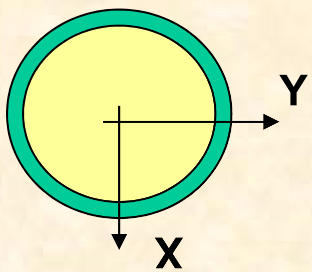
$K_{XX} = K_{YY}, \quad K_{XY} = -K_{YX}$

$C_{XX} = C_{YY}, \quad C_{XY} = -C_{YX}$

$M_{XX} = M_{YY}, \quad M_{XY} = -M_{YX}$

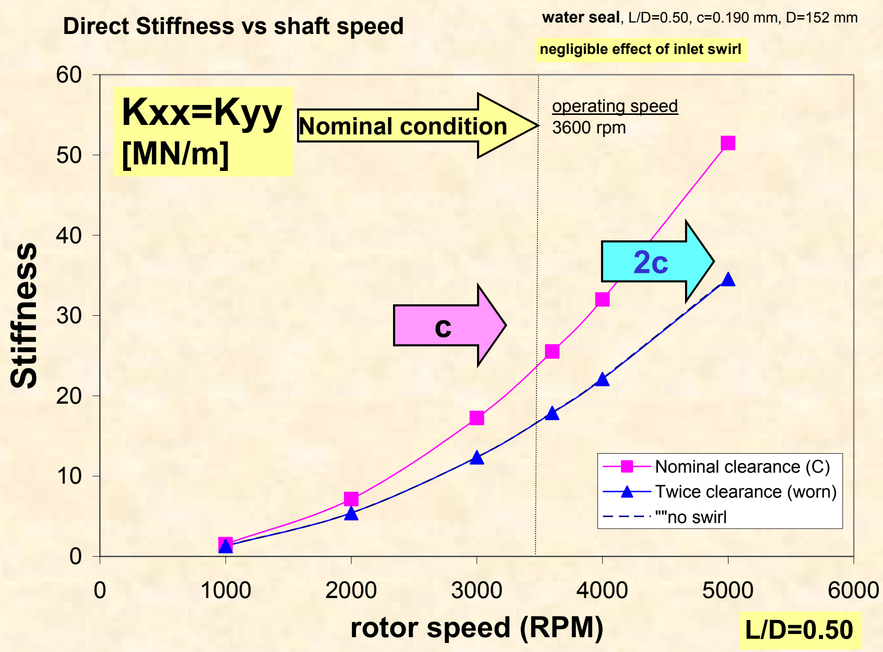
Assumes:
No static load capability
Circular centered orbit

Whirl frequency ratio **WFR** ~ $\dfrac{K_{XY}}{C_{XX}\,\Omega}$: measure of rotordynamic stability



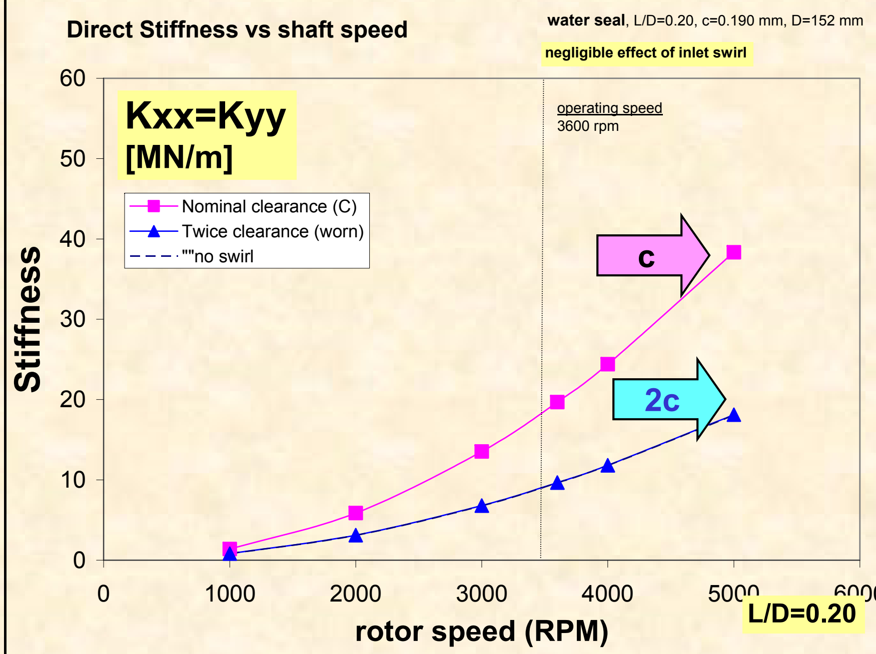
Seal direct stiffness

LONG SEAL



RPM & P$_{supply}$

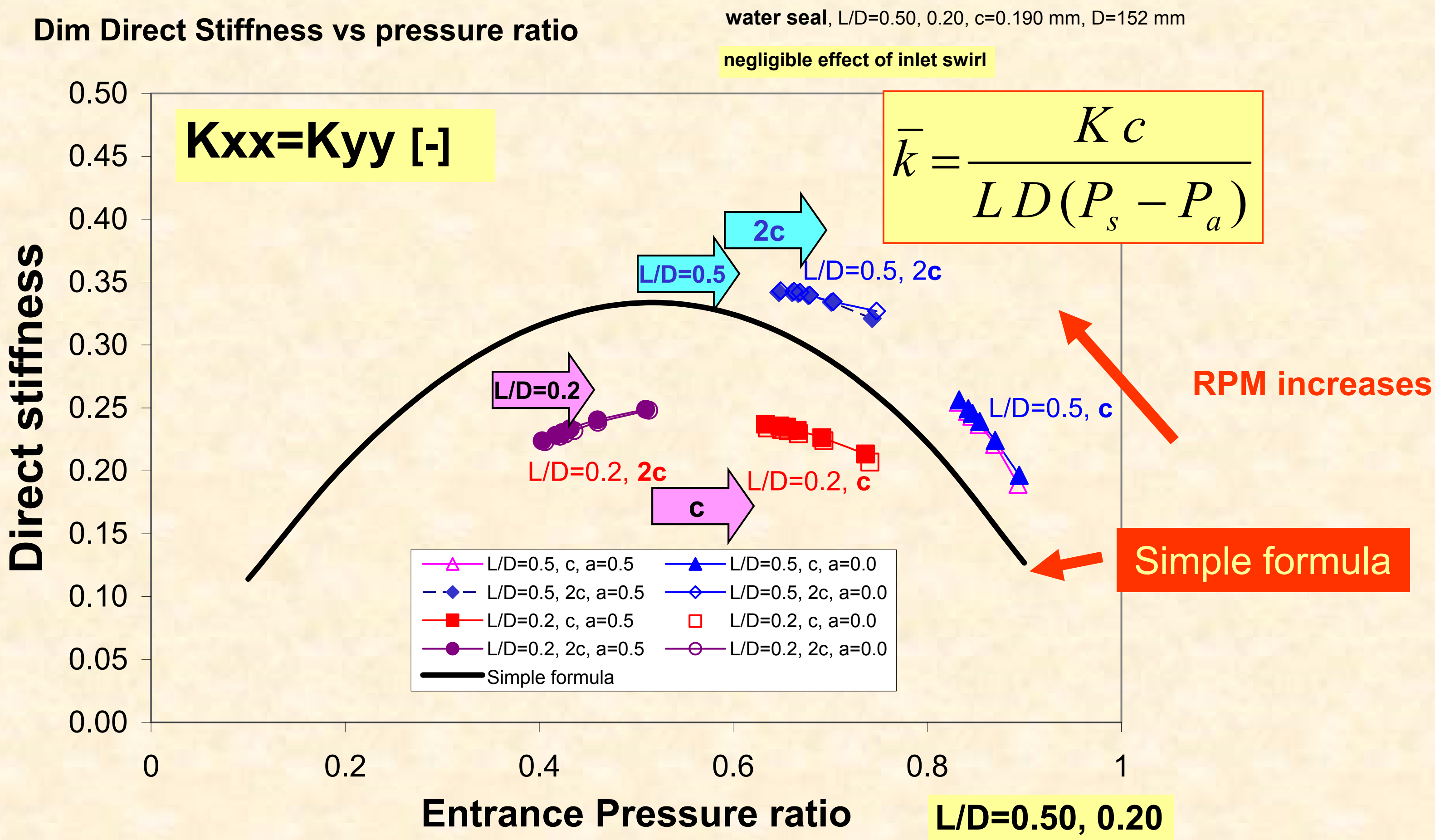
SHORT SEAL



- Direct stiffness ~ Pressure supply ~ RPM $^{1/2}$
- Long seal has ~ 2 x larger stiffness than short seal
- Worn clearance causes ~ 50% drop in direct stiffness. It will affect pump "WET" natural frequencies
- No effect of inlet swirl

Figure 12 Direct stiffnesses Kxx=Kyy for two water L/D seals & two clearances

Seal direct stiffness

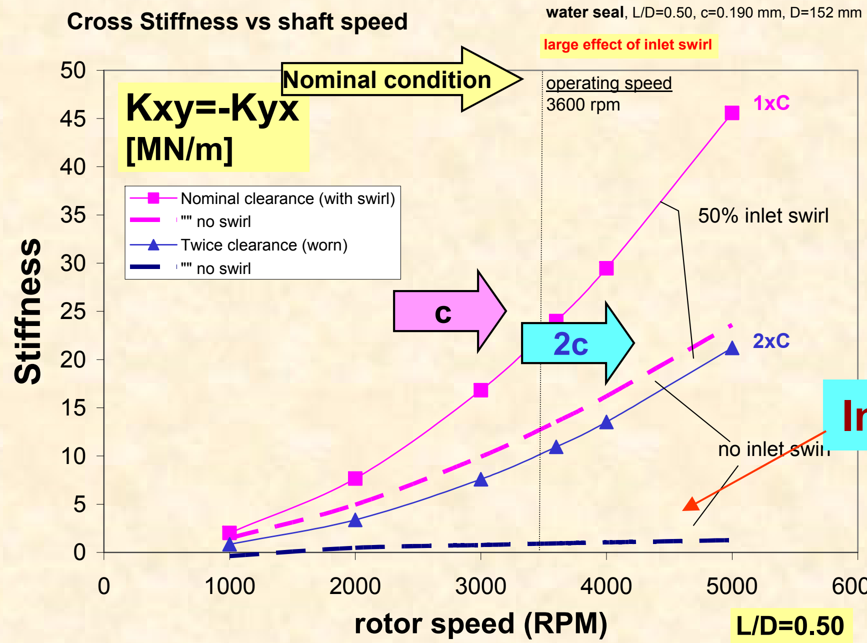


- **Direct stiffness follows simple formula**

Figure Dimensionless stiffnesses **Kxx=Kyy** for two water L/D seals & two clearances

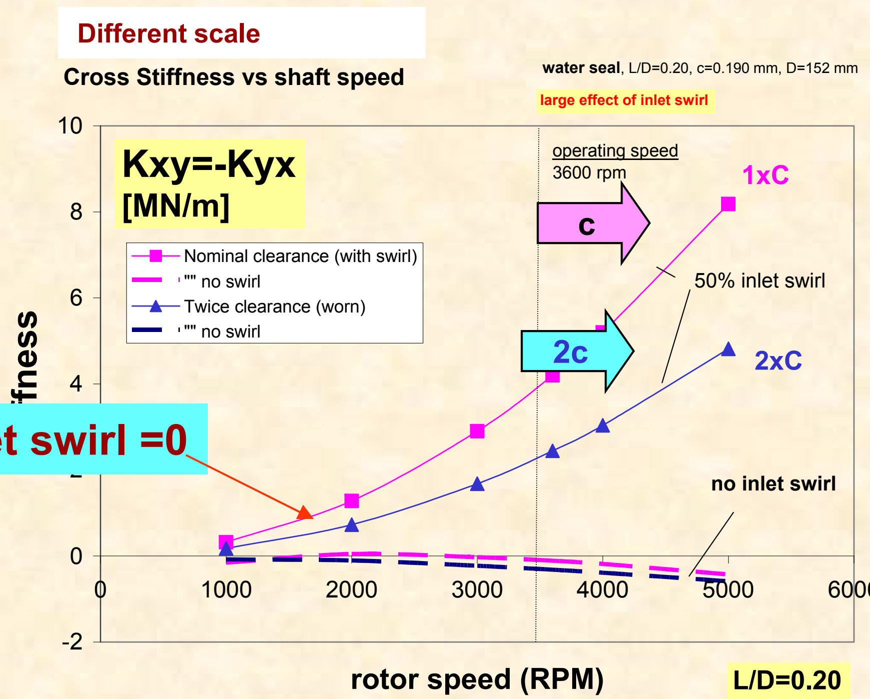
Seal cross-coupled stiffness

LONG SEAL



RPM & P_{supply}

SHORT SEAL

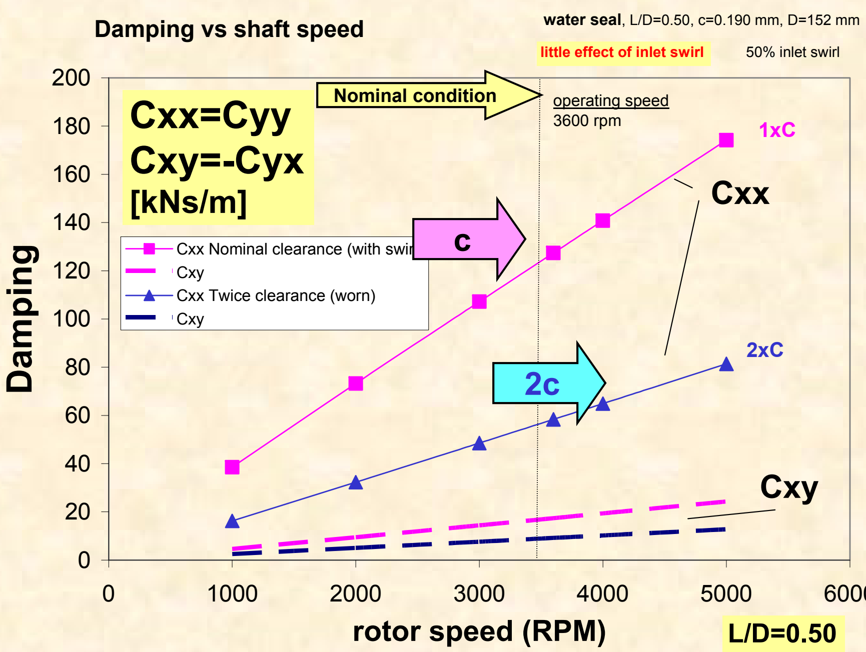


- Cross stiffness ~ RPM & 1/clearance
- Long seal has ~ 5x more cross-stiffness than short seal
- Worn clearance ➔ drop in cross-stiffness.
- Inlet swirl ~ 0 has most pronounced effect – (K_{XY} <0 favors stability)

Figure 13 Cross stiffnesses $K_{XY}=-K_{YX}$ for two water *L/D* seals and two clearances

Seal direct damping coefficients

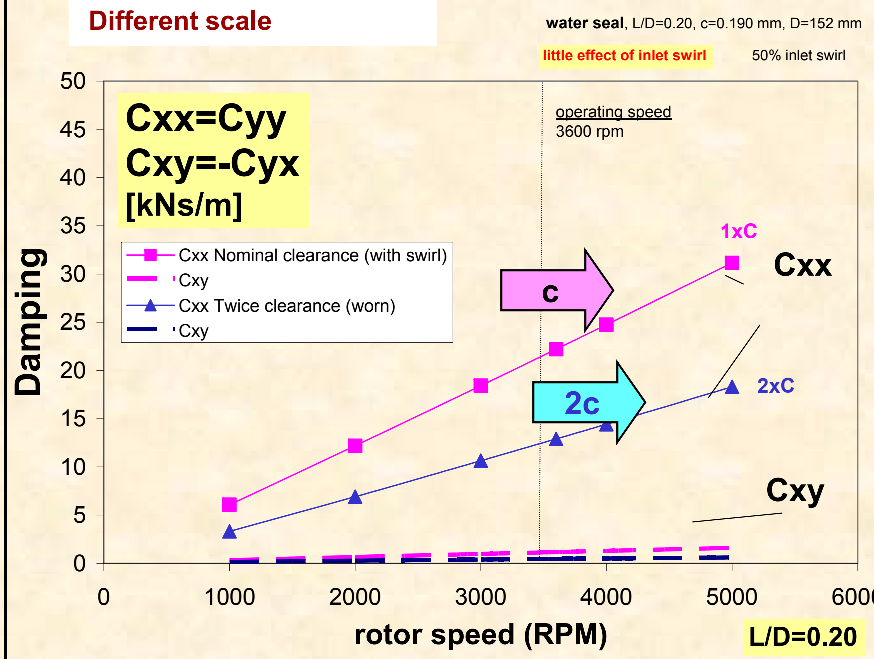
LONG SEAL



RPM & P_{supply}

SHORT SEAL

Different scale



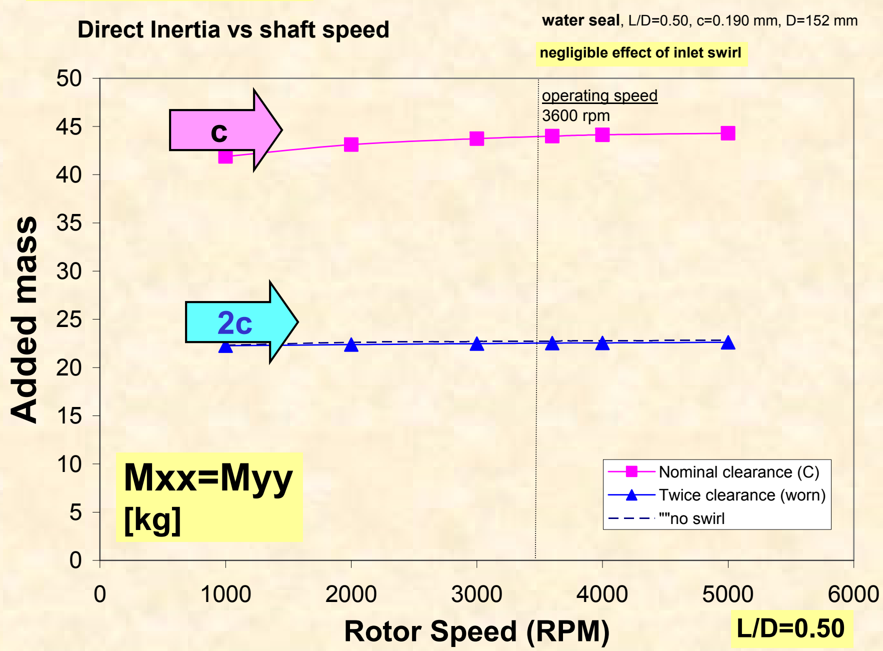
RPM

- Direct damping ~ effective turbulent flow viscosity & 1/clearance
- Long seal has ~ 5x more direct damping than short seal
- Worn clearance → drop in damping
- Inlet swirl no effect. $C_{XX} >> C_{XY}$

Figure 14 Damping $C_{XX}=C_{YY}$ & $C_{XY}=-C_{YX}$ for two *L/D* water seals and two clearances

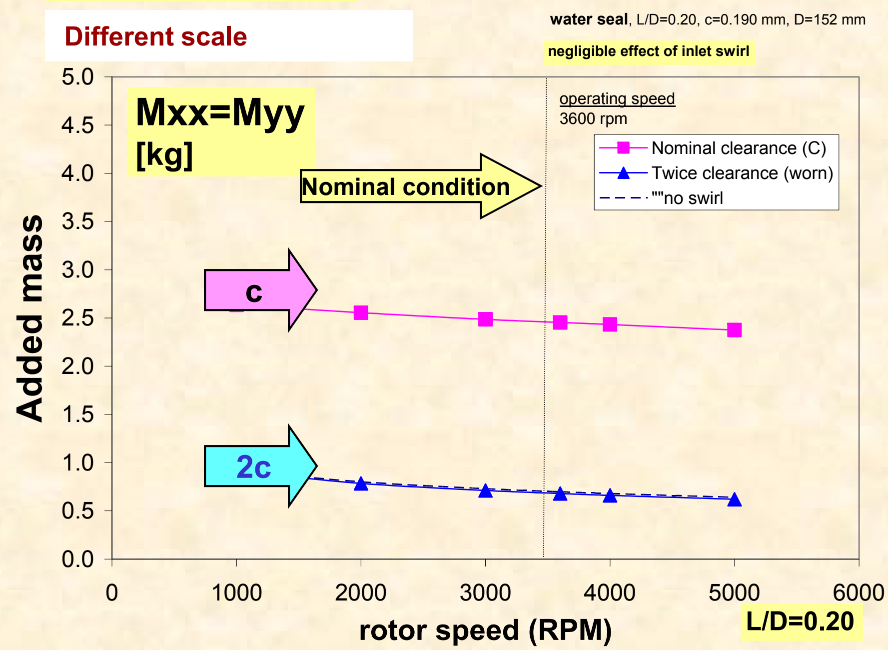
Seal added mass or fluid inertia coefficients

LONG SEAL



RPM & P_{supply}

SHORT SEAL



- Direct inertia invariant with speed & proportional to 1/clearance
- Long seal has ~ 20x more added mass than short seal
- Worn clearance → drop in added mass
- Inlet swirl no effect. $M_{XX} >> M_{XY}$
- Large inertia will affect "wet" pump critical speeds

Figure 15 Inertia $M_{XX}=M_{YY}$ for two *L/D* water seals and two clearances

Fluid inertia – Its magnitude

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

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

D=152.4 mm
c =0.190 mm
μ=0.792 cPoise
ρ=995 kg/m3

LONG SEAL L/D=0.5

0.5

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

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

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

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

SHORT SEAL L/D=0.2

0.2

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

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

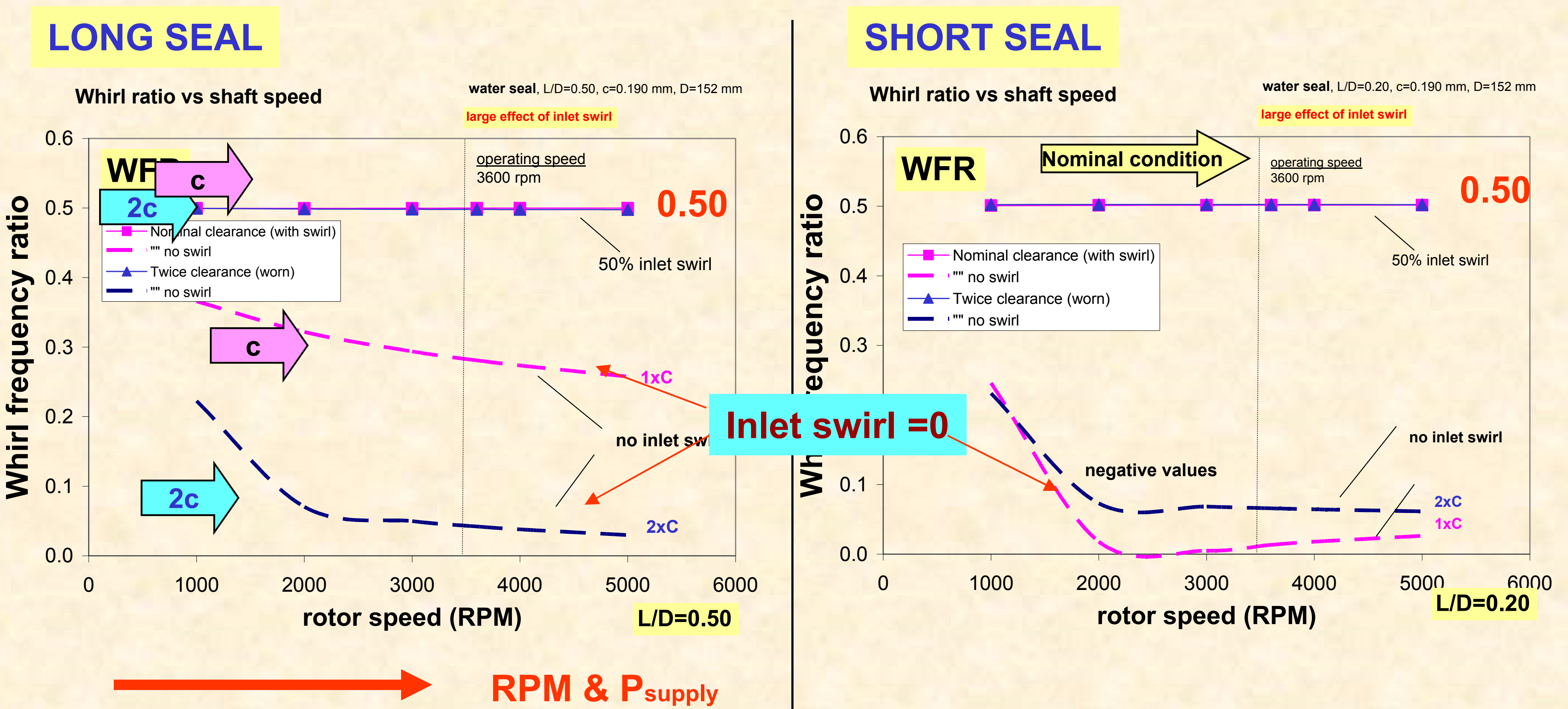
$$M_{steel} = 4.34\text{kg}$$

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

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

M_{XX} can be larger than mass of solid steel of same dimensions

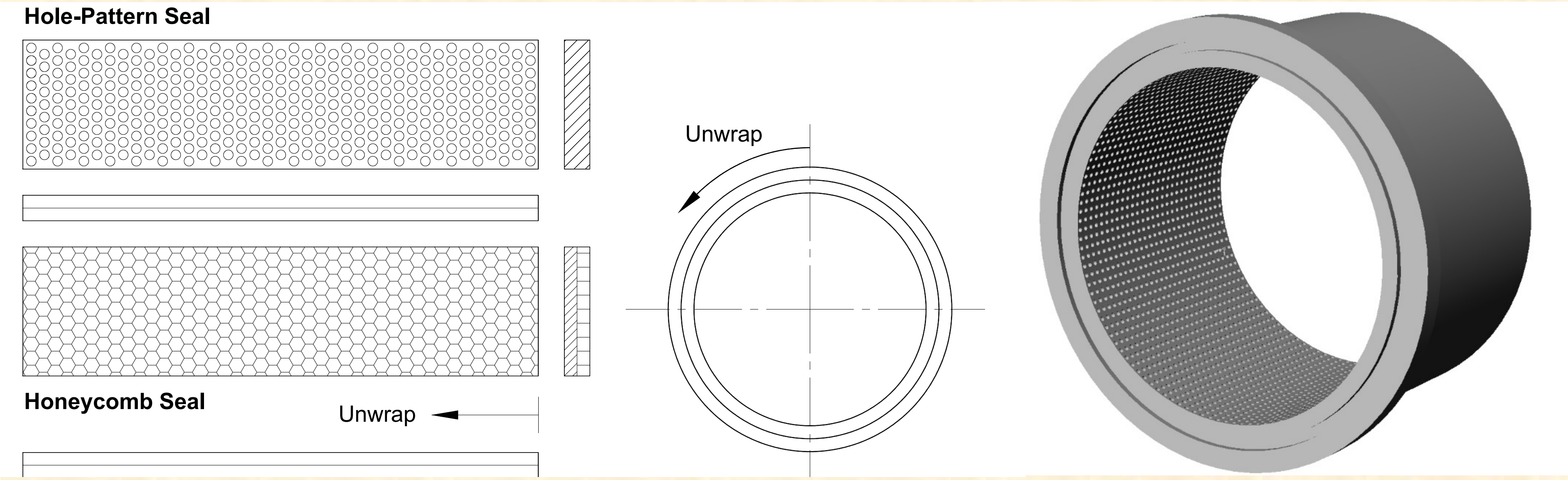
Whirl frequency ratio – stability indicator



- WFR always 0.50 for inlet swirl = 0.50 – Stable up to 2x critical speed
- Inlet swirl = 0.0 drops WFR, in particular for short seal.
- Inlet swirl =0.0 does not help greatly in long seal with tight clearance

Figure 16 Whirl frequency ratio for two *L/D* water seals & two clearances

Textured surface annular seals



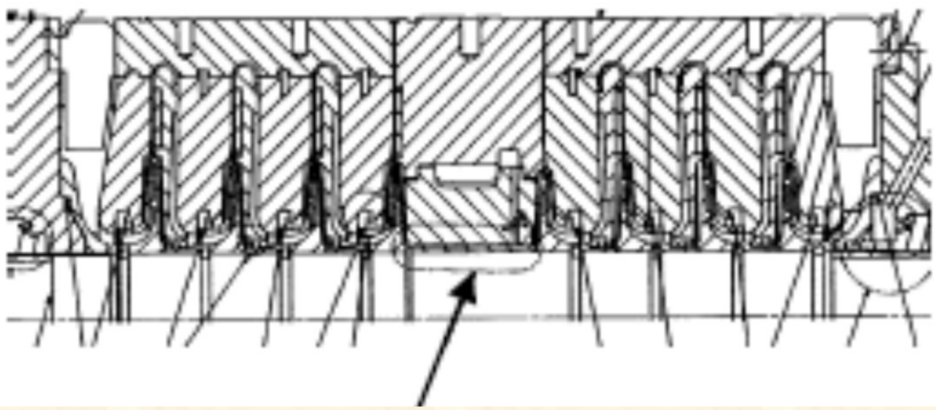
-Machined "roughness" in seal surfaces aids to increase friction thus reducing leakage.

- Surface texturing also reduces development of mean circumferential flow velocity, thus decreasing cross-stiffnesses.

-Proven improvement in stability margins in pumps and compressors

-Texturing works only on stationary surface. Opposite rotordynamic effect if rotor is "rough".

Hole pattern seal for compressors



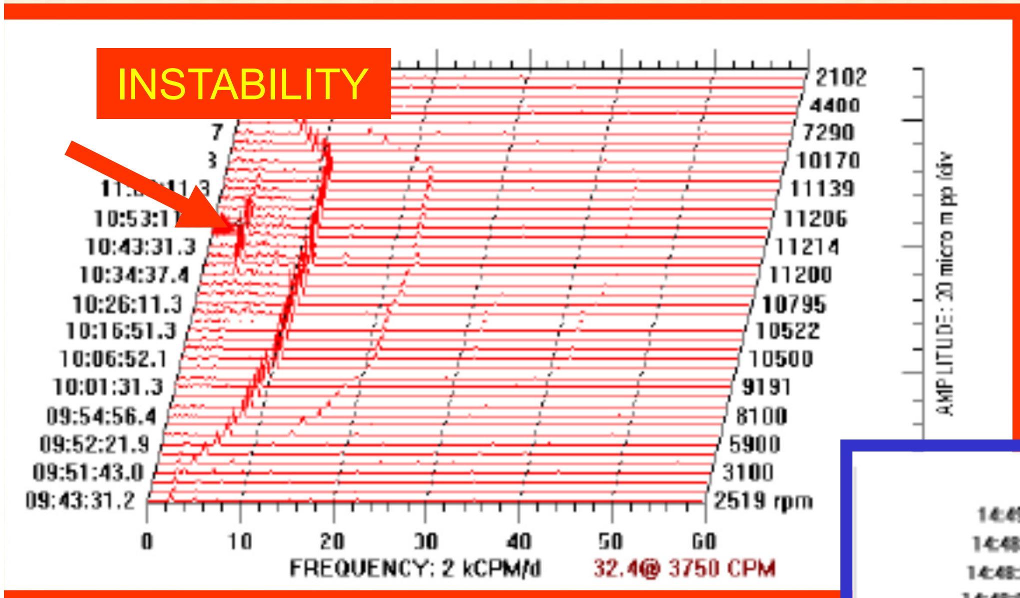
Hole damper seal replacing labyrinth seal in division wall of back-to-back compressor

2005 Turbomachinery Symposium

MECHANICAL UPGRADES TO IMPROVE CENTRIFUGAL COMPRESSOR OPERATION AND RELIABILITY - TP072

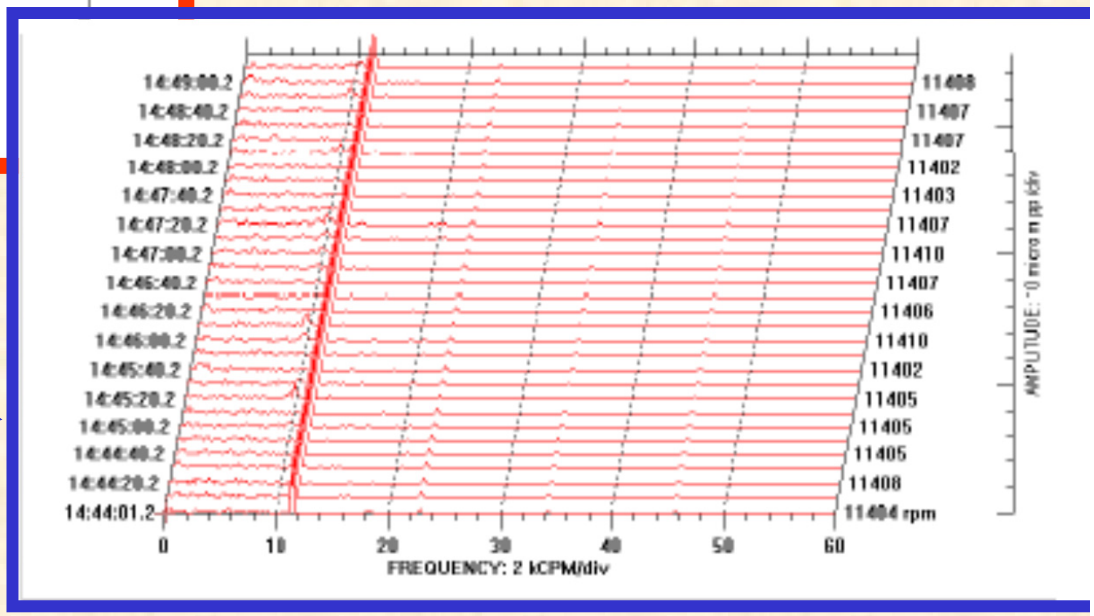
John Stahley
Dresser-Rand Company, Olean, NY, USA, 14870

Hole pattern seal



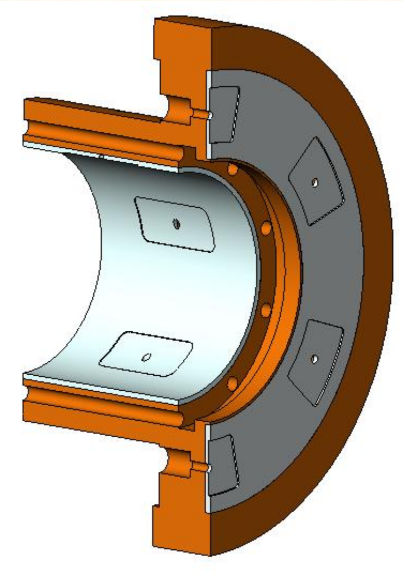
Waterfalls before (with original labyrinth seals) and after installation of hole-pattern seal in division wall

-



Questions?

Hydrostatic Bearings for pump applications



Hydrostatic Bearings for pump applications

External pressure source forces fluid to flow between two surfaces, thus enabling their separation and the ability to support a load without contact.

Advantages

Support very large loads. The load support is a function of the pressure drop across the bearing and the area of fluid pressure action.

Load does not depend on film thickness or lubricant viscosity

Long life (infinite in theory) without wear of surfaces

Provide stiffness and damping coefficients of very large magnitude. Excellent for exact positioning and control.

Disadvantages

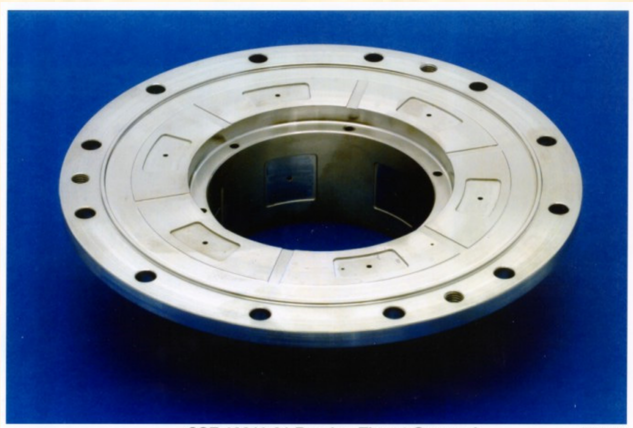
Require ancillary equipment. Larger installation and maintenance costs.
Need of fluid filtration equipment. Loss of performance with fluid contamination.
High power consumption: pumping losses.

Limited LOAD CAPACITY ~ $f(P_{supply})$

Potential to induce hydrodynamic instability in hybrid mode operation.

Potential to show pneumatic hammer instability with compressible fluids

Hydrostatic bearings for turbopumps



Low cost primary power cryogenic turbo-pumps (TP) are compact, operate at high speeds, and require of externally pressurized fluid film bearings to support radial and thrust loads.

Hybrid thrust & radial bearings enable smaller and lighter turbopumps with no DN life limitations

Large stiffness (accuracy of positioning) and damping force coefficients allow for unshrouded impellers with increased TP efficiency

Support stiffness in a Hydrostatic Bearing

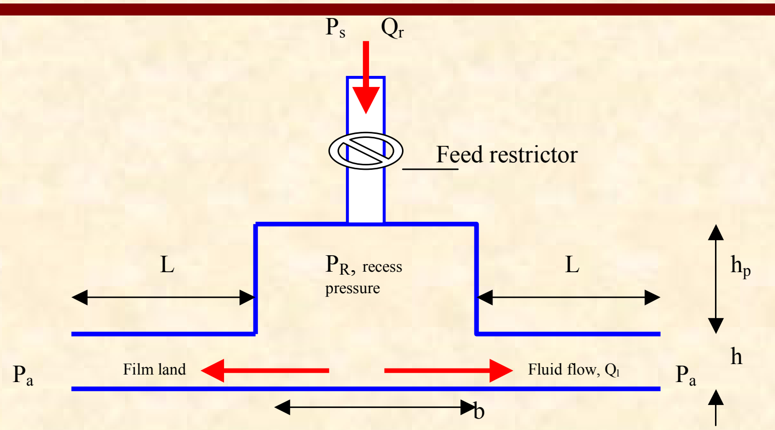


Figure 18: Geometry of a simplified 1-D hydrostatic bearing

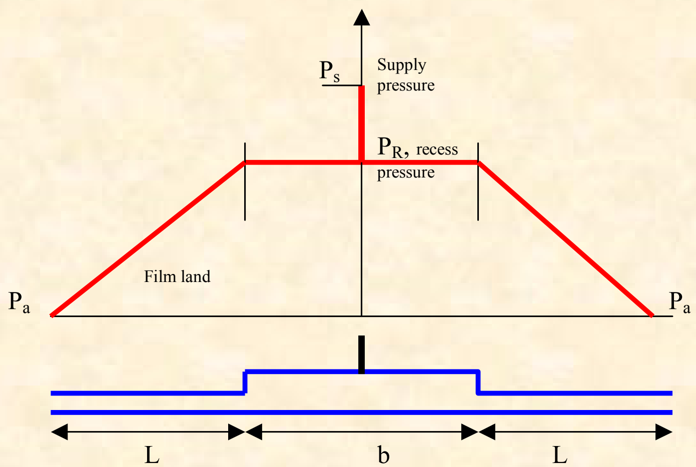


Figure 19: Typical pressure drop in a hydrostatic bearing (laminar flow without fluid inertia effects, incompressible fluid)

Flow through restrictor = Flow through film lands

$$Q_r = Q_o = A_o C_d \sqrt{\frac{2}{\rho}\left(P_s - P_R\right)}$$ **orifice**

$$Q_r = Q_c = \frac{\pi d^4}{128\,\mu\,\ell_c}\left(P_s - P_R\right)$$ **capillar**

$$Q_\ell = -\frac{B h^3}{12\mu}\frac{\partial P}{\partial x} = +\frac{B h^3\left(P_R - P_a\right)}{12\,\mu\,L}$$ **Land flow**

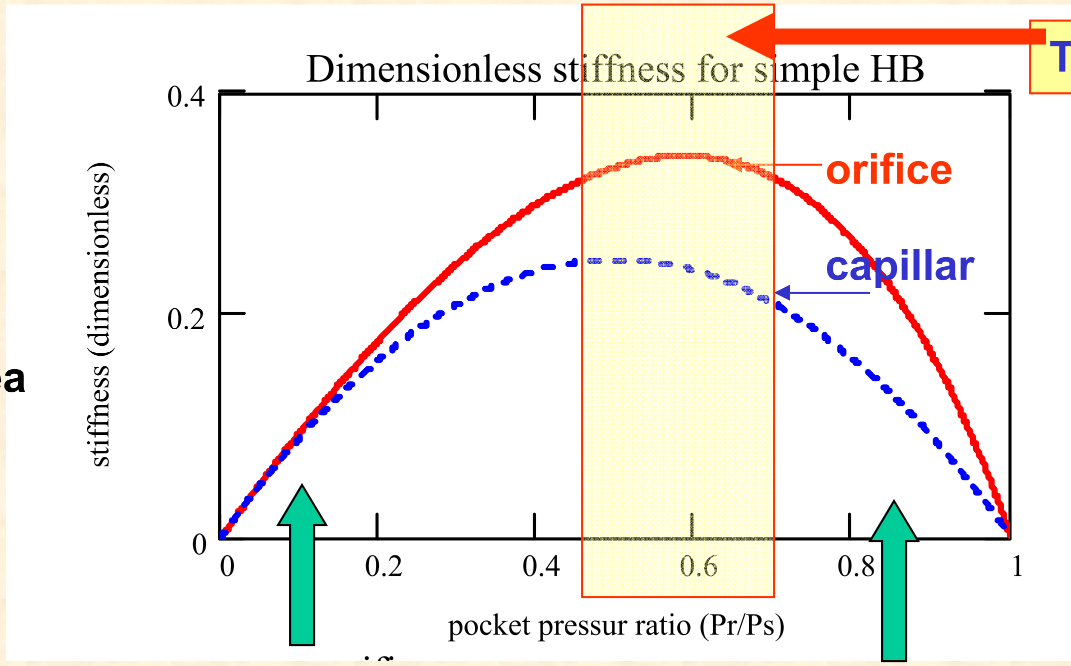
As film thickness decreases, resistance in land increases, thus reducing flow rate and increasing pressure in pocket or recess.

Stiffness generated from changes in pocket pressure

Figures 18 & 19

Static support stiffness in a hydrostatic bearing

$\frac{K\,c}{P_s\,Area}$



Typical design condition

Figure 20: Static stiffness for simple hydrostatic bearing (laminar flow w/o fluid inertia effects, incompressible fluid).

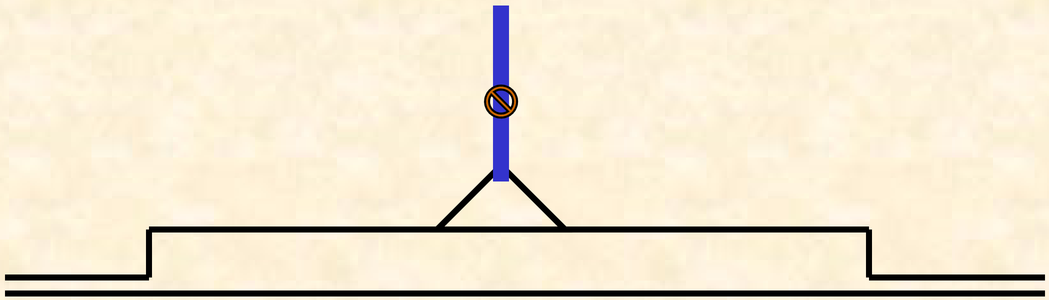
<u>Low recess pressure</u>
Too large clearance
Or too small orifice diameter

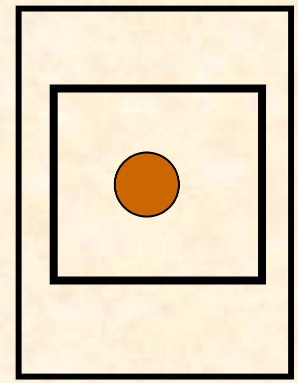
<u>High recess pressure drop</u>
Too tight clearance
Or too large orifice

- **No need of journal rotation**
- **Stiffness is proportional to supply pressure and bearing (pocket) area.**
- **Stiffness is inversely proportional to clearance**
- **Stiffness changes quickly with variations in pocket pressure**
- **Hydrostatic bearings have LIMIT load capacity**

Figure 20

Traditional hydrostatic bearing design:





* large pocket area (80-90 % of total area)
* deep pocket depth
* large orifice discharge volume

Applications:

low or null surface speed, low frequencies,
nearly **incompressible** fluids (water or mineral oil)
produces very large DIRECT Stiffness.

Warning: This design should NEVER be used with compressible liquids or gases

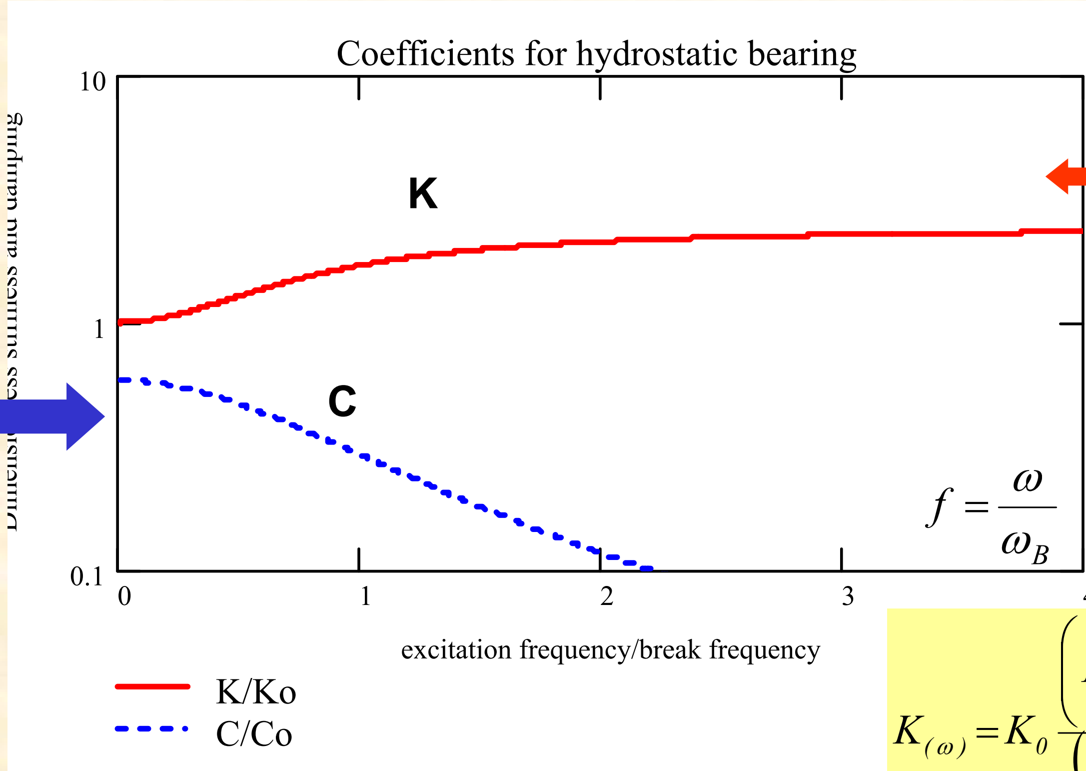
Fluid Compressibility Effects

Pneumatic Hammer is a self-excited instability (loss of damping) in **poorly designed** hydrostatic bearings for applications with compressible liquid and gases.

The instability arises due to trapped fluid volume in the bearing pockets generating a dynamic pressure that lags journal motions by ~ (+)90 degrees.

Remedies to **AVOID** (reduce and even eliminate) pneumatic hammer are **WELL KNOWN** (documented analysis and operation verification) and can be implemented easily at the design stage.

Fluid compressibility Effects



Low frequency
Loss of damping if fluid is compressible

High frequency
Stiffness "Hardening" effect
Complete loss of damping

$$K_{(\omega)} = K_0 \frac{\left(1+\frac{f^2}{\alpha}\right)}{\left(1+f^2\right)}; \quad C_{(\omega)} = C_0 \frac{(1-\alpha)}{\left(1+f^2\right)}$$

$$\alpha = \frac{K_o}{\omega_B C_o}$$

Coefficients depend on BREAK frequency:

$$\omega_B = (Z+1)\frac{\kappa}{V_{rec_0}}\frac{h_0^3 B}{6\mu L}$$

A function of fluid bulk modulus (κ)/pocket volume

Figure 22 **Frequency dependent force coefficients for simple hydrostatic bearing**

Whirl frequency ratio of centered HJB

$$\phi = WFR = \frac{K_{XY}}{\Omega C_{XX_{f==0}}} = \frac{K_{XY}}{\Omega C_{XX_{=0}}(1-\alpha)} \approx 0.5\frac{1}{(1-\alpha)}$$

Whirl frequency ratio

$$\alpha = \frac{K_o}{\omega_B C_o}$$

$$\omega_B = \frac{Q_{r_0}}{P_{R_0}}\frac{(Z+1)\kappa}{V_{rec_0}} = (Z+1)\frac{\kappa}{V_{rec_0}}\frac{h_0^3 B}{6\mu L}$$

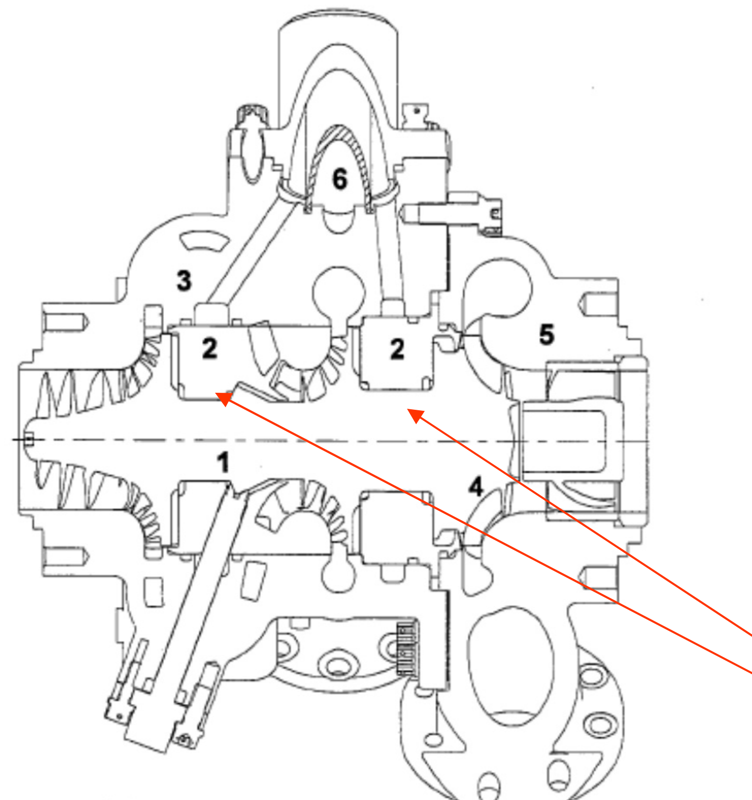
WFR in a hydrostatic bearing can be WORSE than that of a plain journal bearing due to fluid compressibility.

Worse conditions are for gases and liquid hydrogen in a cryogenic turbo pump.

In aerostatic bearings – pockets are not machined to reduce (eliminate) pneumatic hammer

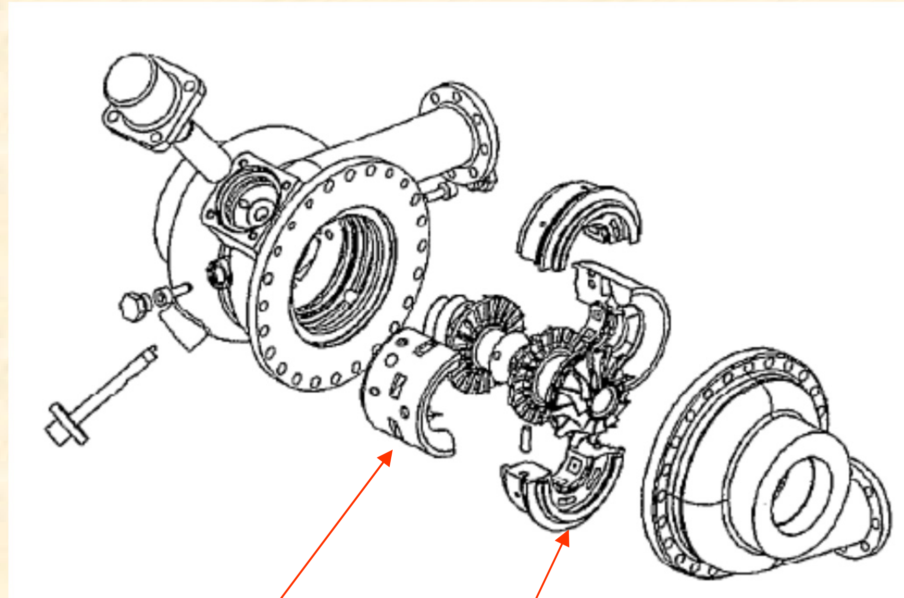
For LH2, very shallow pockets and reduced pocket area are recommended.

Hydrostatic Bearings for Cryogenic Turbo Pumps



Major Features:

1 One piece titanium rotor.
2 Split hydrostatic bearings.
3 Cast pump housing with integral crossover passages.
4 Radial inflow turbine.
5 Cast turbine housing with vaneless inlet volute.
6 Filtered bearing supply.



Radial hydrostatic bearings

Thrust hydrostatic bearing

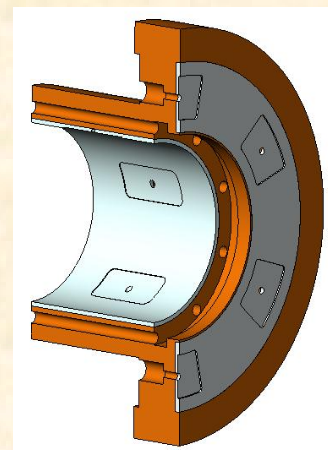
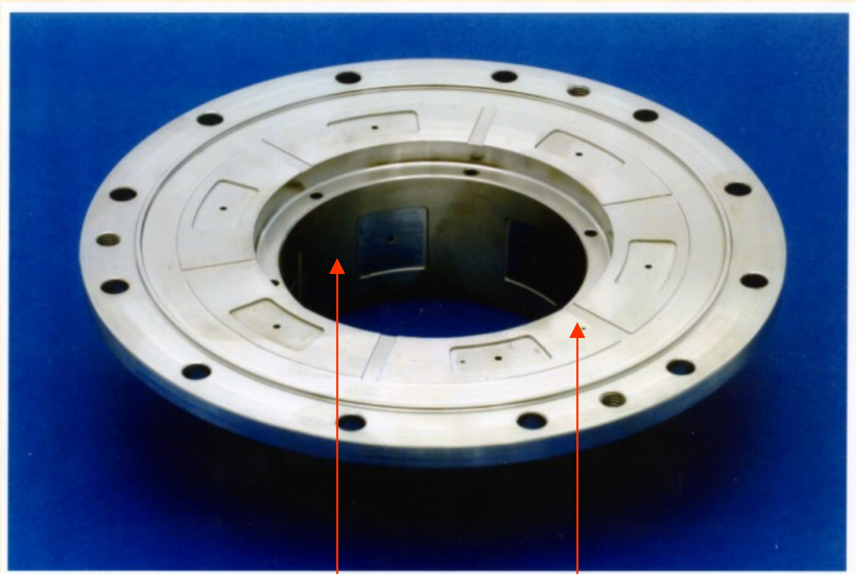
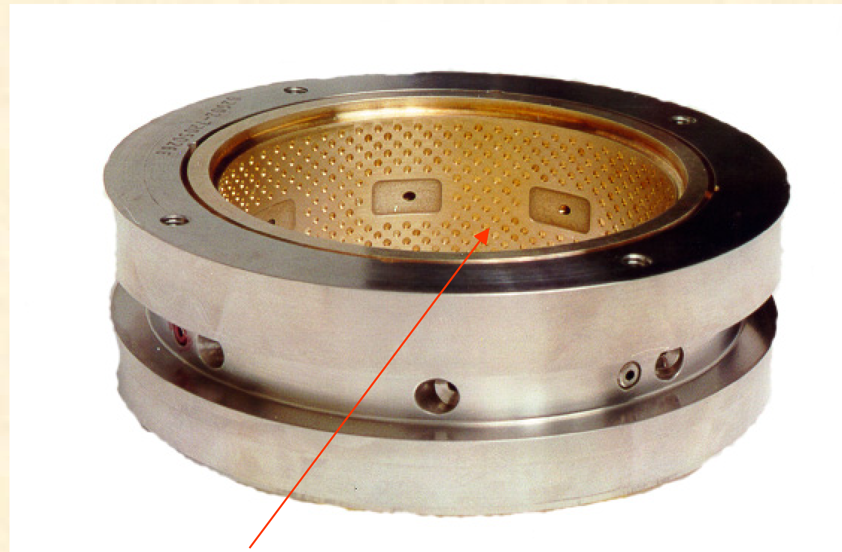


Figure 23 **Advanced Liquid Hydrogen Turbopump [22]**

Hydrostatic Bearings for Cryogenic Turbo Pumps





Radial hydrostatic bearing

Thrust hydrostatic bearing

Radial hydrostatic bearing for LH_2 TP - Knurled surface

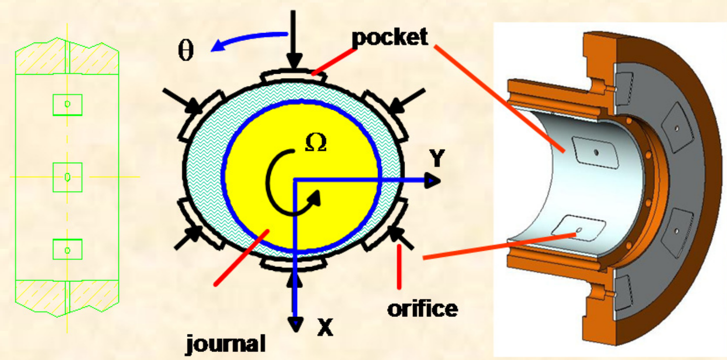


Figure 24 **Hydrostatic radial and thrust bearings for cryogenic turbopump**

Bulk-flow Analysis of Hydrostatic Bearings

Flow Continuity

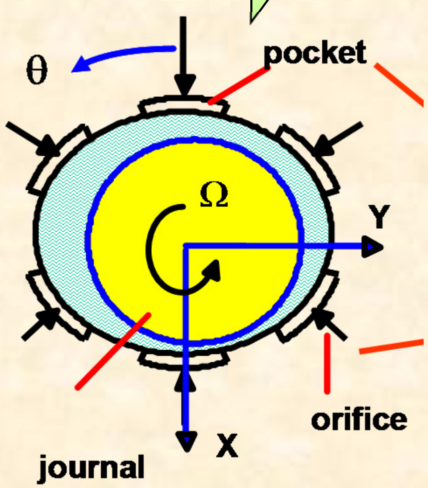
$$\frac{\partial}{\partial x}\left(hV_x\right)+\frac{\partial}{\partial z}\left(hV_z\right)+\frac{\partial h}{\partial t}=0$$

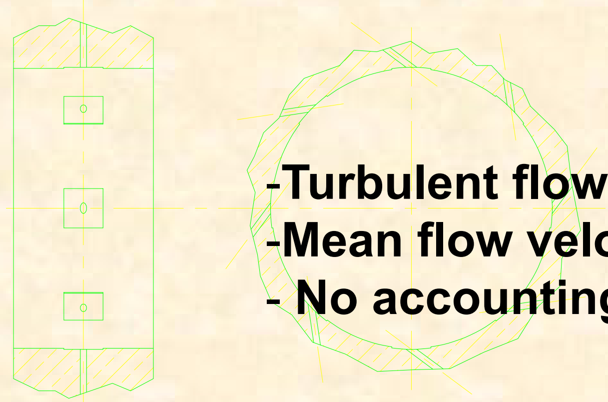
Circumferential Momentum transport

$$-h\frac{\partial P}{\partial x}=\frac{\mu}{h}\left(\kappa_x V_x-\kappa_J\frac{U}{2}\right)+\rho h\left\{\frac{\partial V_x}{\partial t}+\frac{\partial V_x^{\,2}}{\partial x}+\frac{\partial V_x V_z}{\partial z}\right\}$$

Axial momentum transport

$$-h\frac{\partial P}{\partial z}=\frac{\mu}{h}\kappa_z V_z+\rho h\left\{\frac{\partial V_z}{\partial t}+\frac{\partial V_x V_z}{\partial x}+\frac{\partial V_z^{\,2}}{\partial z}\right\}$$





-Turbulent flow with fluid inertia effects
-Mean flow velocities – average across film (h)
- No accounting for strong recirculation zones

Bulk-flow Analysis of Hydrostatic Bearings

Pocket pressure field with angled injection

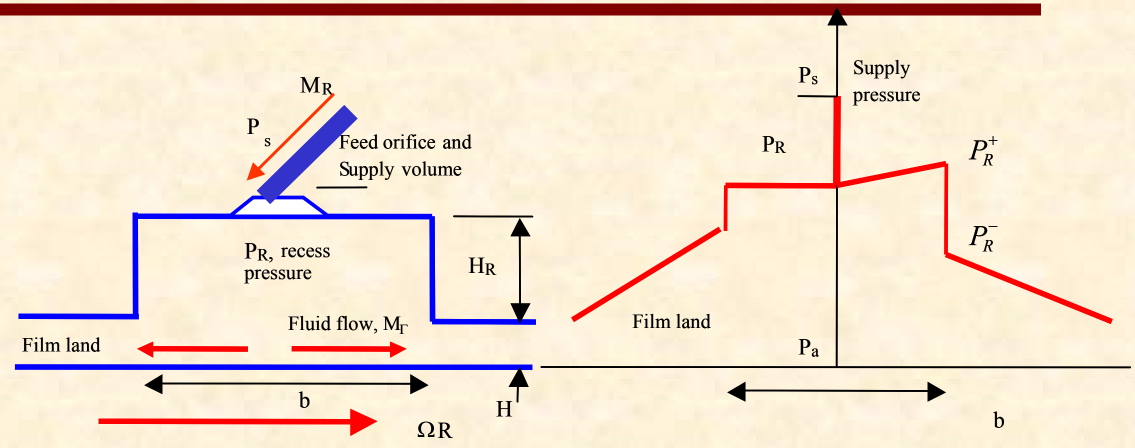
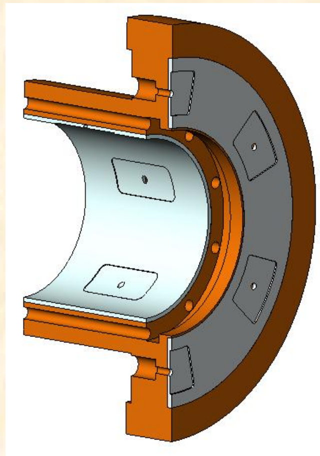


Figure 26: Turbulent flow pressure distribution in a pocket of a hybrid bearing

Flow continuity

$$M_R = C_d\, A_o \left(\frac{\rho}{2} \left[P_s - P_R \right] \right)^{1/2} = M_\Gamma + \frac{\partial}{\partial t}\left(\rho V_R \right)$$



Pressure rise before edge

$$P_R^+ = P_R + \mu \kappa_x \frac{b}{2\left(h+h_R\right)^2} \left\{ \frac{\Omega R}{2} - V_x \right\}_R$$

Pressure rise at edge

$$P_R^- = P_R^+ + \frac{(1+\xi)}{2} \rho \left[1 - \left(\frac{\rho_e^-}{\rho_e^+} \right) \left(\frac{h}{h+h_R} \right)^2 \right] V_{x,z}^2$$

Figure 26 **Pocket pressures: angled & radial injection**

Bulk-flow Analysis of Hydrostatic Bearings

Small amplitude radial motions (X,Y) at frequency (ω) to derive
zeroth order equations for equilibrium flow field:
flow rate, load capacity, power loss, temperature raise
first-order equations for perturbed flow field:
(stiffness, damping and inertia) force coefficients

$$\boxed{Z_{\alpha\beta} = -\oint_{A_B} \left\{ P_\beta \, H_\alpha \right\} R \, dz \, d\theta = K_{\alpha\beta} - \omega^2 M_{\alpha\beta} + i\,\omega\, C_{\alpha\beta}\,; \quad {}_{\alpha,\beta = X,Y}}$$

Numerical method of solution:

SIMPLEC method, extension of control volume method,
+
component method for superposition of hydrostatic and hydrodynamic effects.

Sensitivity Analysis

Bearing force coefficients and performance parameters are most sensitive to orifice entrance loss coefficient.

Dynamic Performance of Hydrostatic Bearings

Example: Water bearing replacing oil lubricated bearing

Fluid: water at 30°C (0.792 cPoise, 995 kg/m3)

D=L = 152. 4 mm (6 inch)
c=0.102 mm (4 mil), nominal clearance

5 pockets: l= 51 mm, arc 41° , depth= 0.381 mm

→ **c = 1.33x oil bearing**
Pocket depth/c = 3.75
Pocket area = 19%
to avoid hammer

Orifice diameter: 3.2 mm (Cd=0.80)

Static load =Wx = 5000 N → **STATIC LOAD FROM PUMP WEIGHT**

smooth rotor and stator surfaces

Nominal speed = 3600 rpm,
Pressure drop 34.4 bar

→ **Pressure supply ~ RPM^2**
(bleed off from pump discharge)

Inlet loss coefficient ξ=0.1
Inlet swirl α=0.50

RADIAL AND TANGENTIAL INJECTION → **Influence on cross-coupled stiffness and stability**

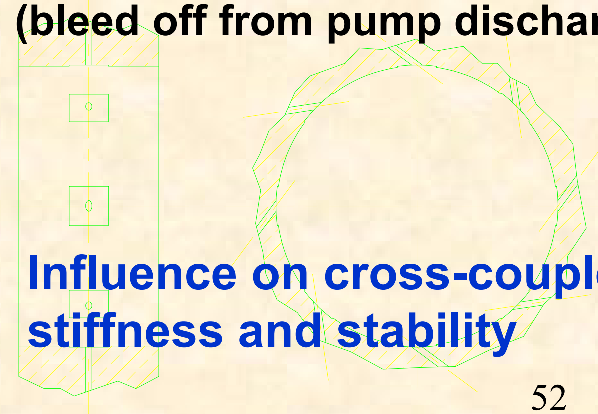
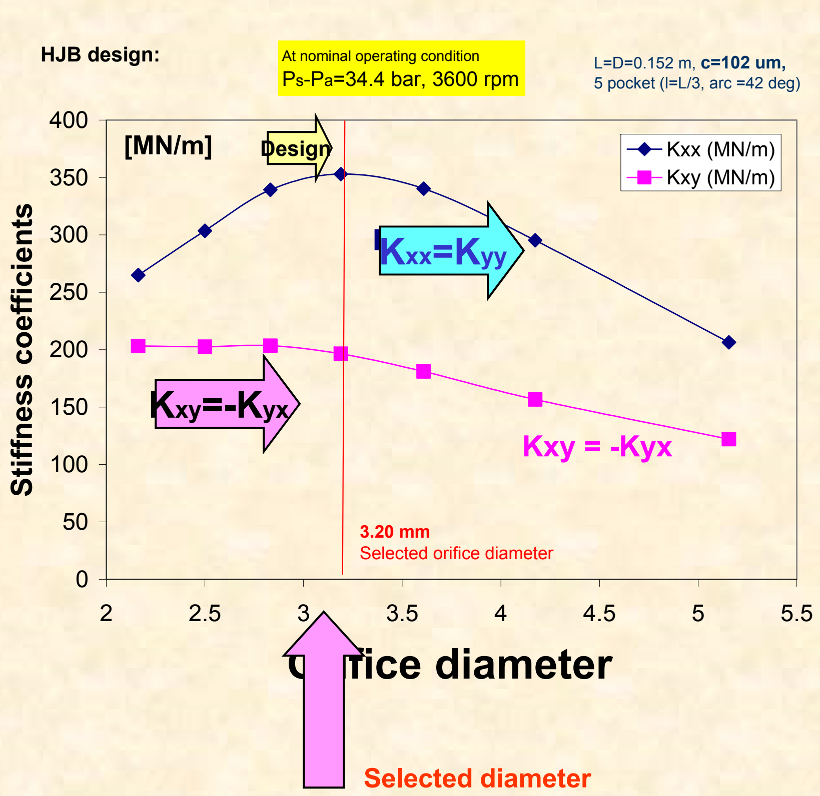
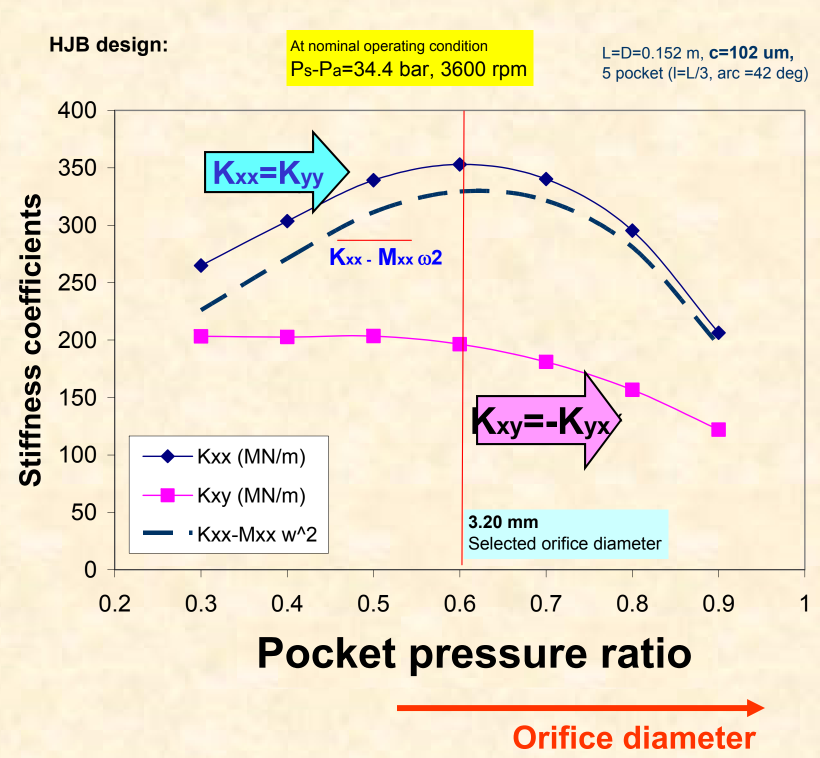


Table 2 Geometry and operating conditions of HJBs for a liquid pump

Hydrostatic Bearing – Orifice diameter selection



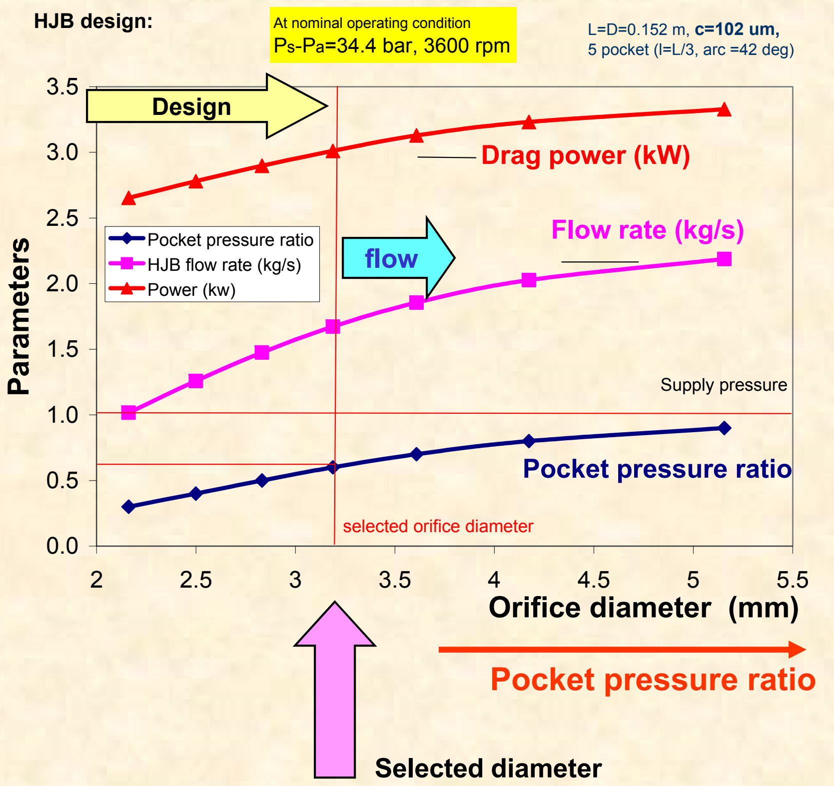


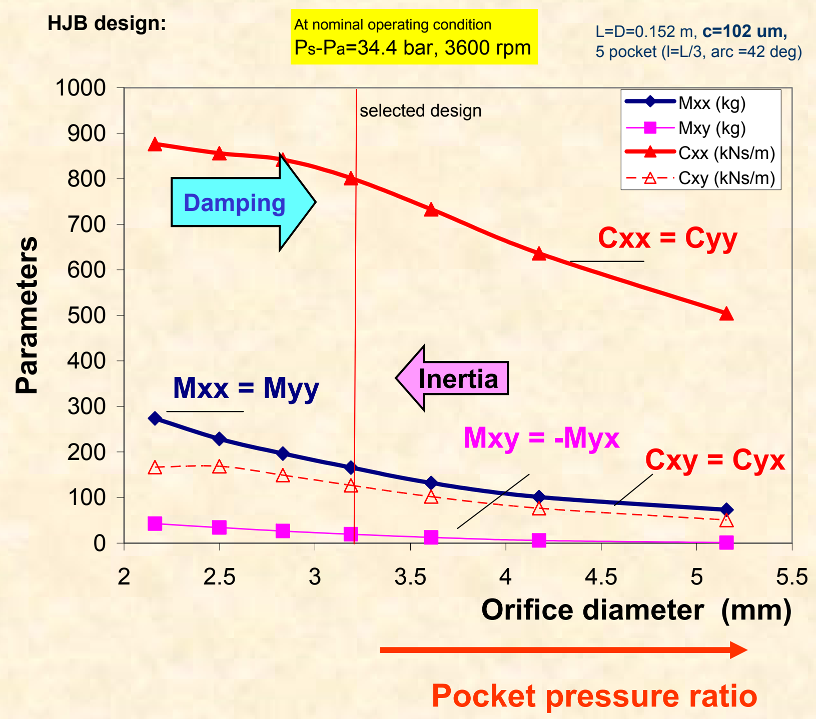
- **Operating condition: 3.6 krpm & 34.4 bar pressure supply (NO LOAD)**
- **Select orifice diameter to MAXIMIZE direct stiffness**
- No influence of angled injection on Kxx

Figure 27

Direct and cross-coupled stiffnesses versus orifice diameter and pocket pressure for water hydrostatic bearing. Nominal operating condition, centered bearing (no load)

Hydrostatic Bearing – Orifice diameter selection

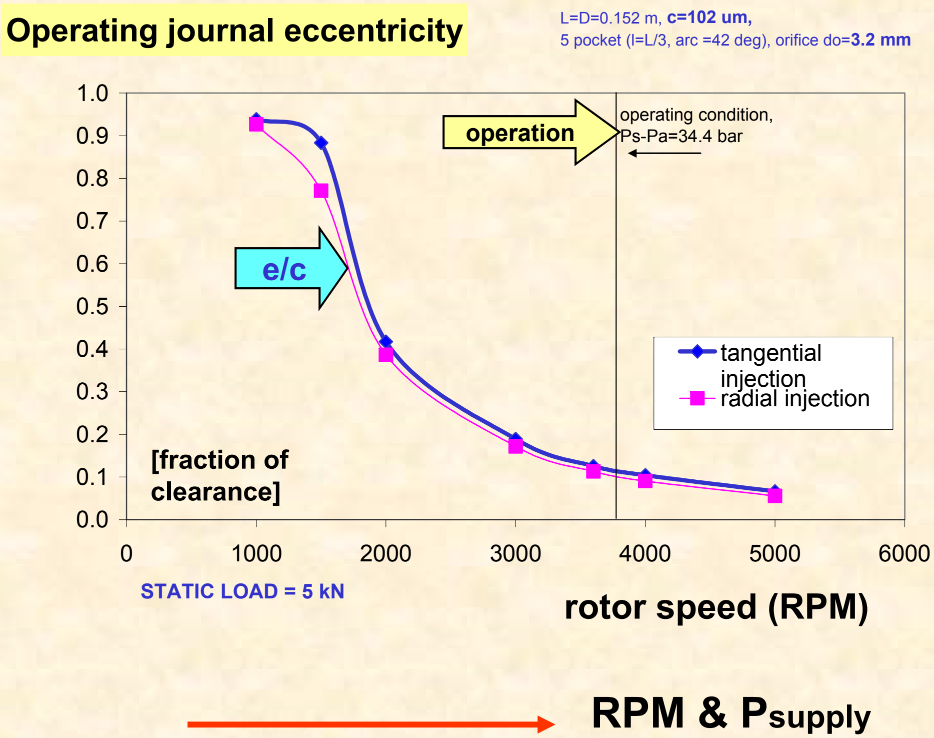


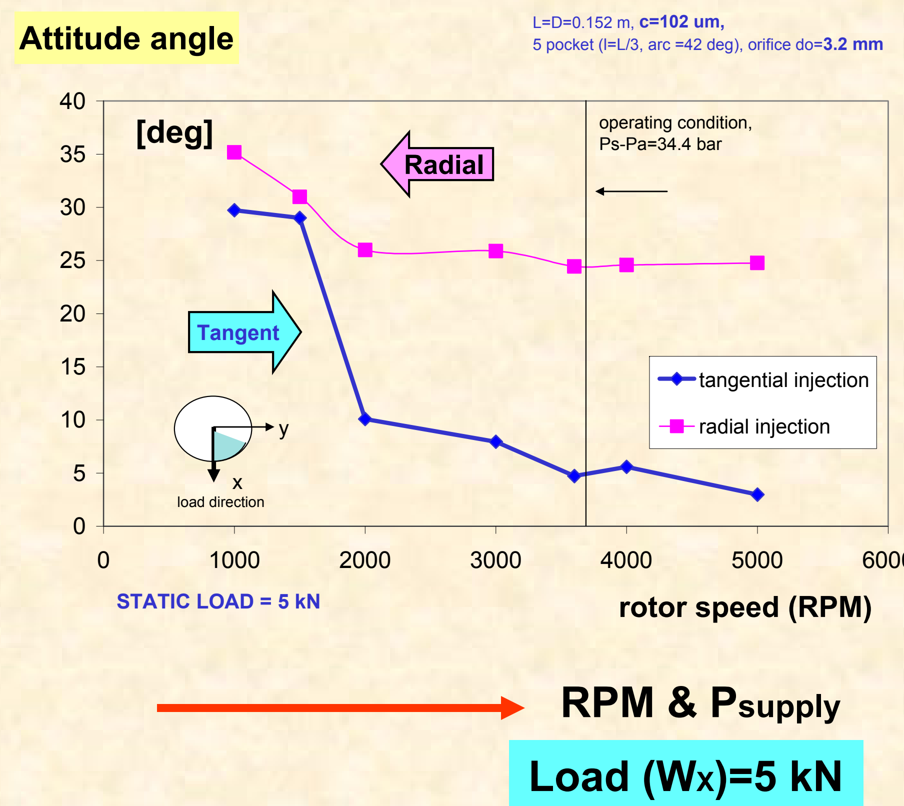


- Drag power, flow rate and pocket pressure increase with orifice diameter
- Direct damping and inertia coefficients decrease

Figure 28 **Performance parameters for water hydrostatic bearing versus orifice diameter.** Nominal operating condition, centered bearing (no load)

Hydrostatic Bearings – Static Load Performance

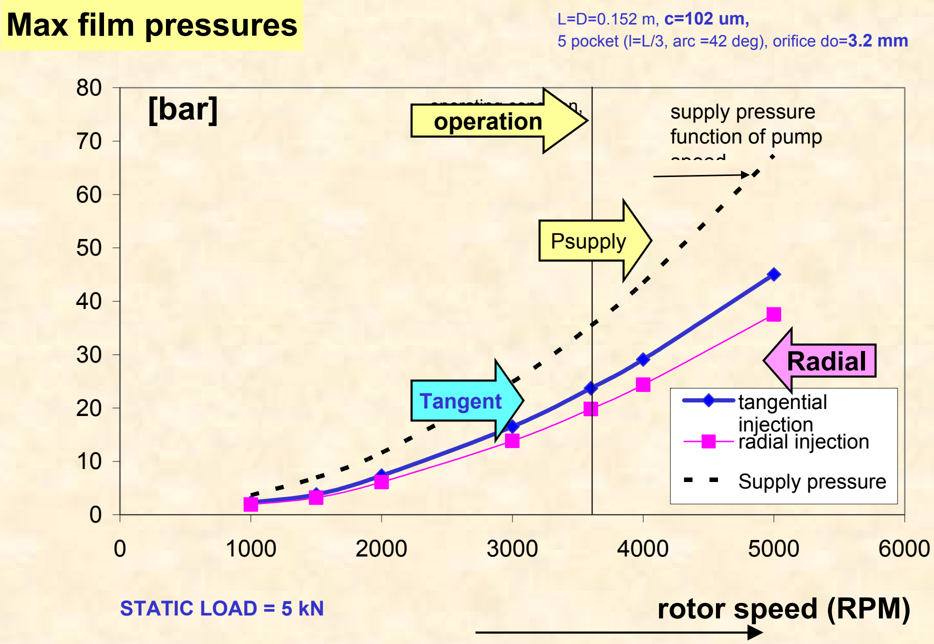


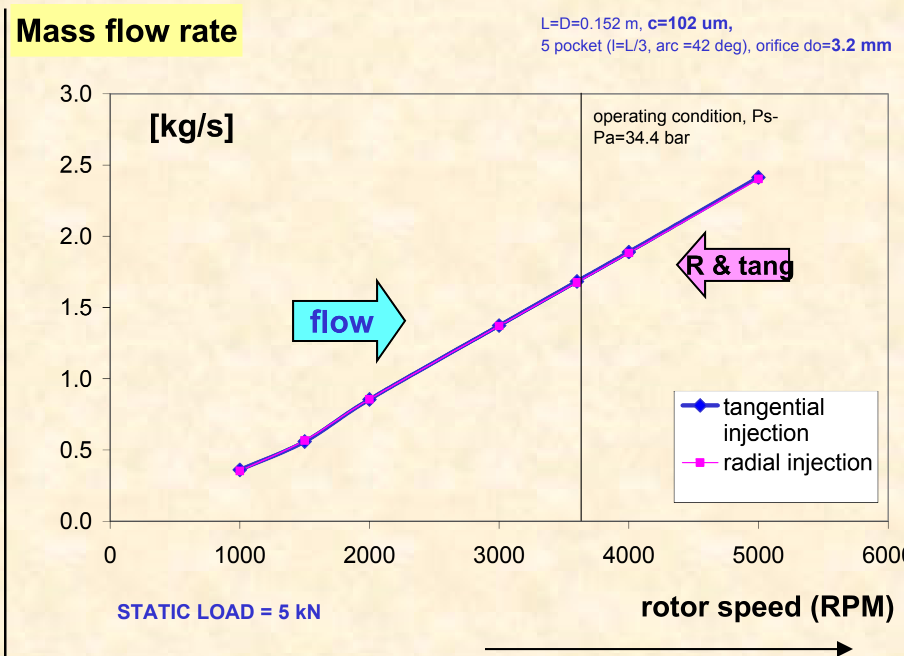


- At operating condition, static eccentricity is small, 10% of clearance (*c*=0.102 mm)
- Large eccentricities at low speeds because pressure supply is low (hydrodynamic operation)
-Tangential injection reduces attitude angle: will result in improved stability of bearing

Figure 29 **Journal eccentricity and attitude angle versus rotor speed.**
Water hydrostatic bearing with radial and tangential injection

Hydrostatic Bearings – Peak pressures



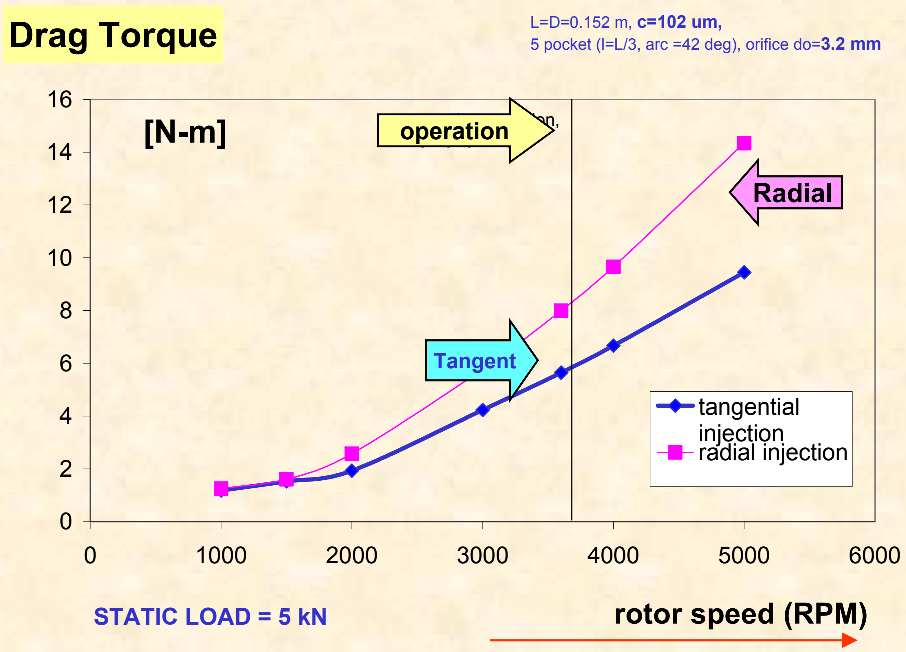


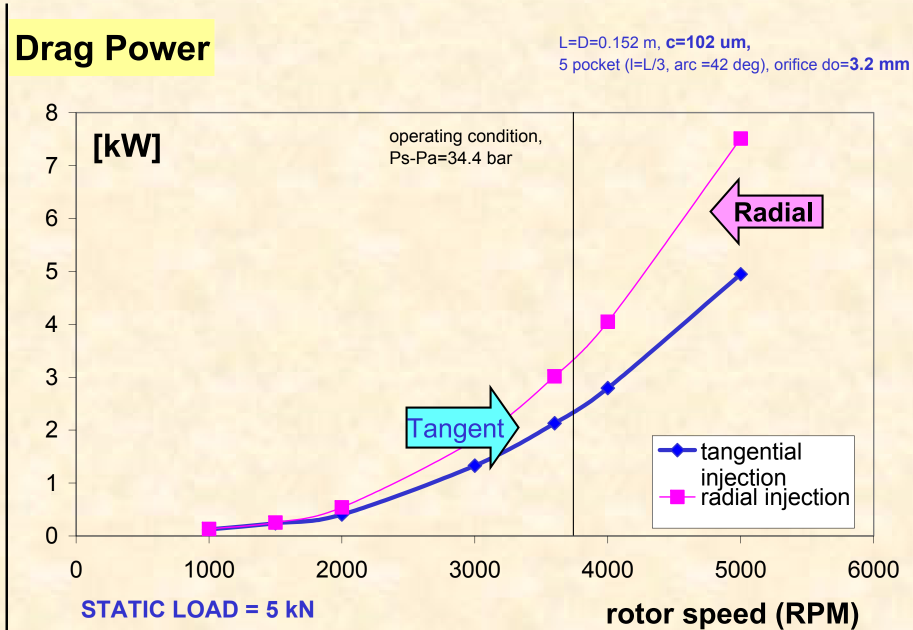
Load (Wx)=5 kN

- Flow rate is proportional to rotor speed since supply pressure (~ $RPM^{1/2}$)
- Max. film pressure is proportional to supply pressure, Max. Pressure ~ RPM^2
- Tangential injection increases edge recess pressure – no influence on flow rate
- Large flow rate (90 LPM) typical of application

Figure 30 **Maximum film pressures and flow rate versus rotor speed.**
Water hydrostatic bearing with radial and tangential injection

Hydrostatic Bearings – Drag Torque





Load (W_x)=5 kN

2

- Drag torque & power proportional to RPM
- Tangential injection decreases power & torque since it retards development of circumferential speed.

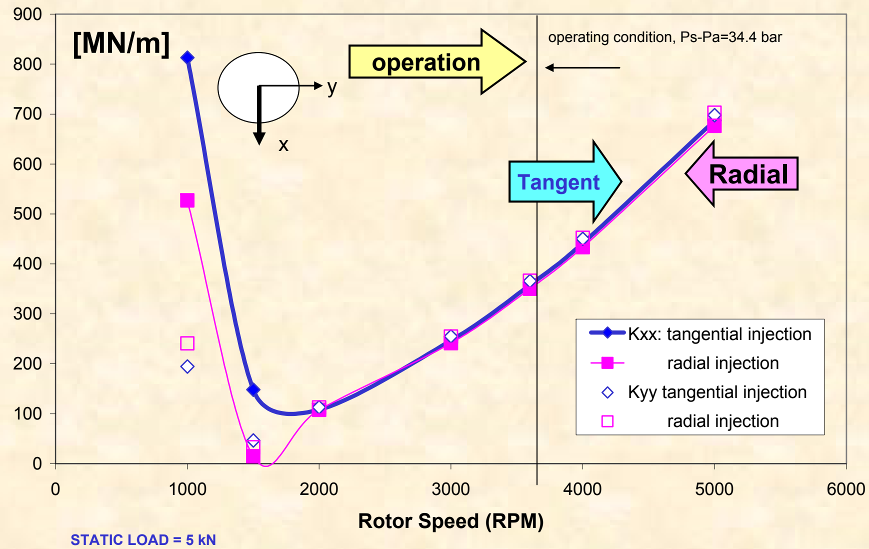
Care at low speeds and high pressures (reverse journal rotation)

Figure 31 **Torque and drag power versus rotor speed.** Water hydrostatic bearing with radial and tangential injection

Hydrostatic Bearings – STIFFNESSES

Direct stiffnesses **K_{xx}, K_{yy}**

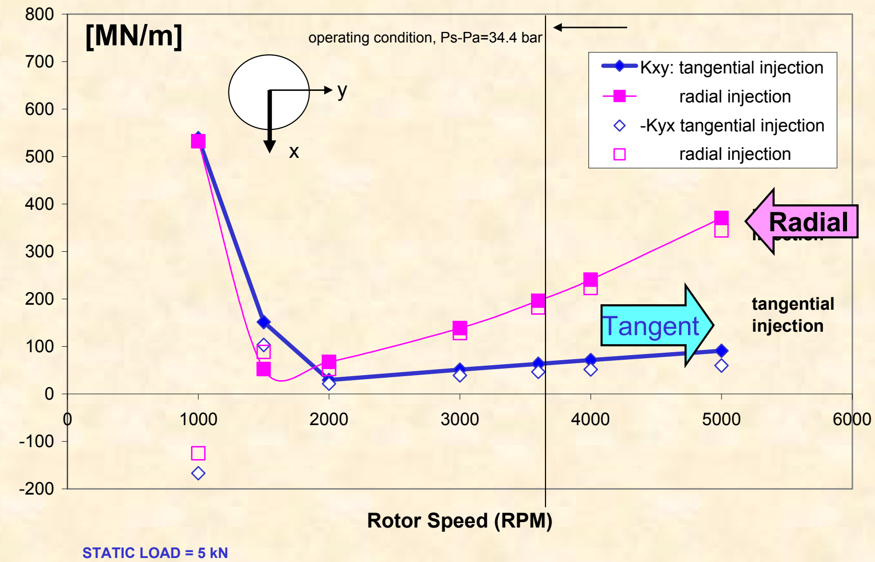
L=D=0.152 m, **c=102 um,**
5 pocket (l=L/3, arc =42 deg), orifice do=**3.2 mm**



RPM & P_{supply}

Cross stiffnesses **K_{xy}, $-K_{yx}$**

L=D=0.152 m, **c=102 um,**
5 pocket (l=L/3, arc =42 deg), orifice do=**3.2 mm**



RPM & P_{supply}

Load (W_X)=5 kN

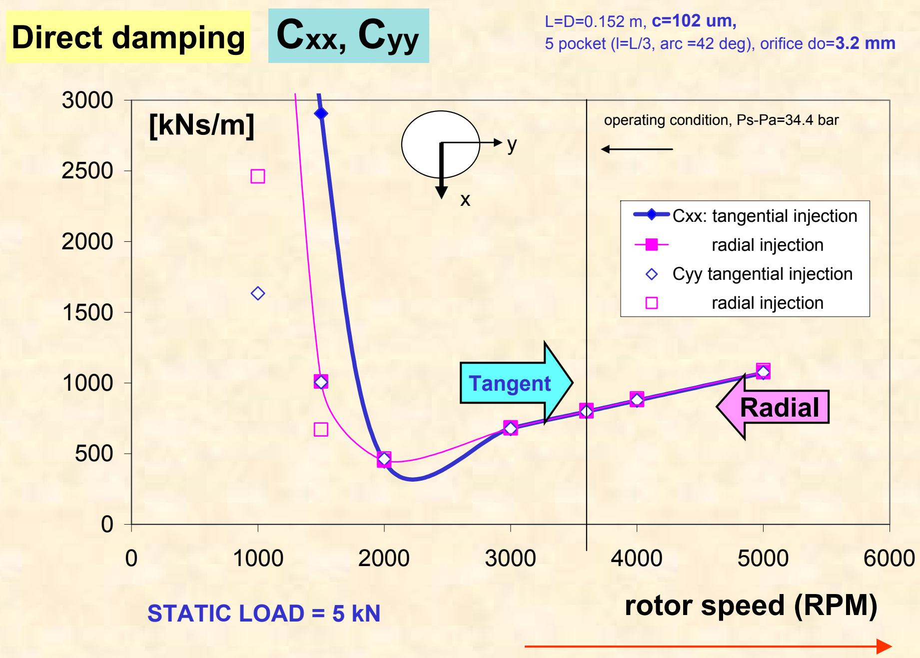
- <u>At low speeds</u>, bearing operates under hydrodynamic conditions since feed pressure is too low. All coefficients are large since operating eccentricity (*e/c*) is large

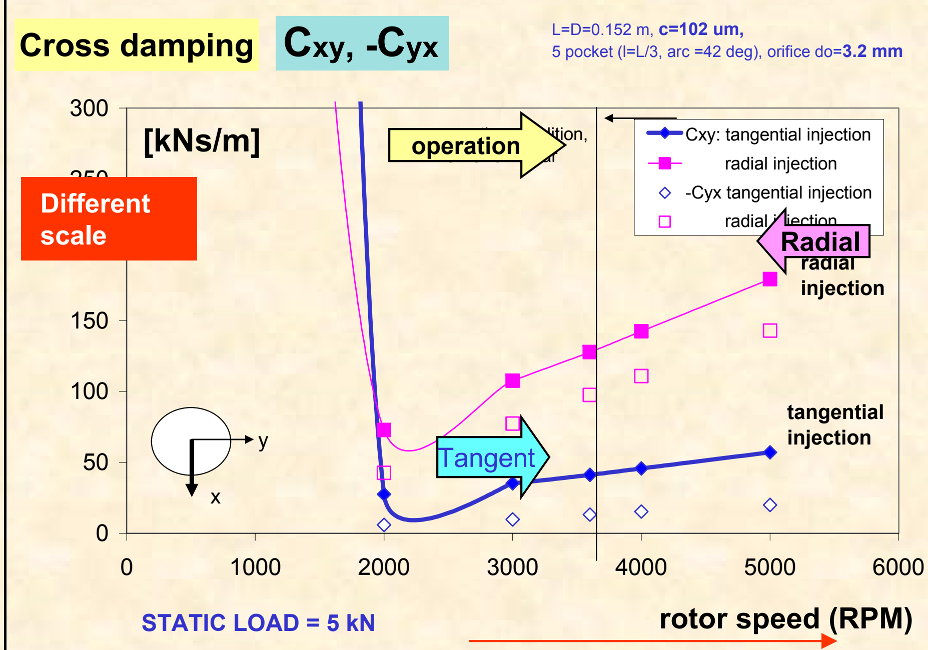
- <u>At operating speed</u>, direct stiffness K_{XX}~K_{YY} & K_{XY}=-K_{YX} since (*e/c*)~0 : near centered operation.

- Tangential injection reduces greatly cross-coupled stiffness.

Figure 32 **1X Stiffness coefficients versus rotor speed.** Water hydrostatic bearing with radial and tangential injection

Hydrostatic Bearings – DAMPING



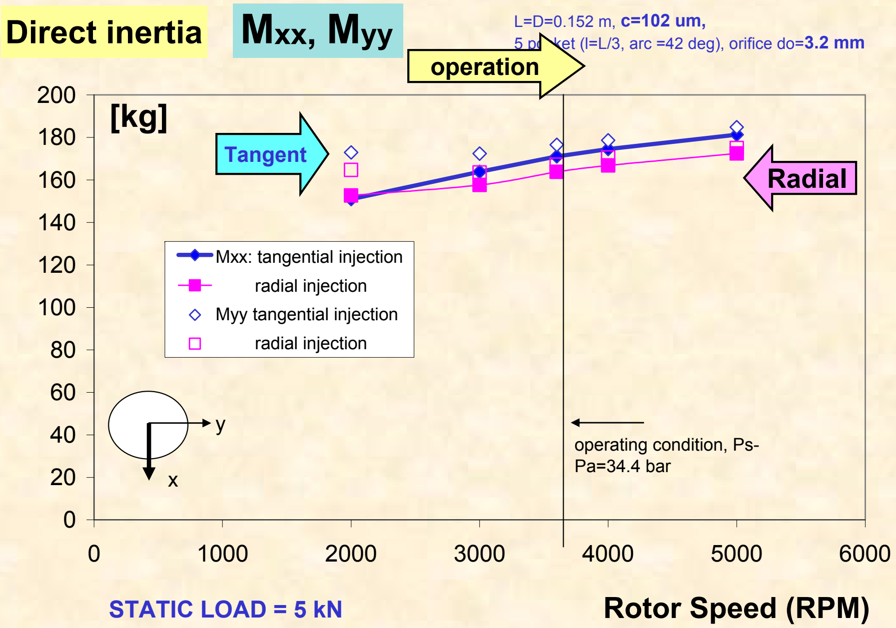


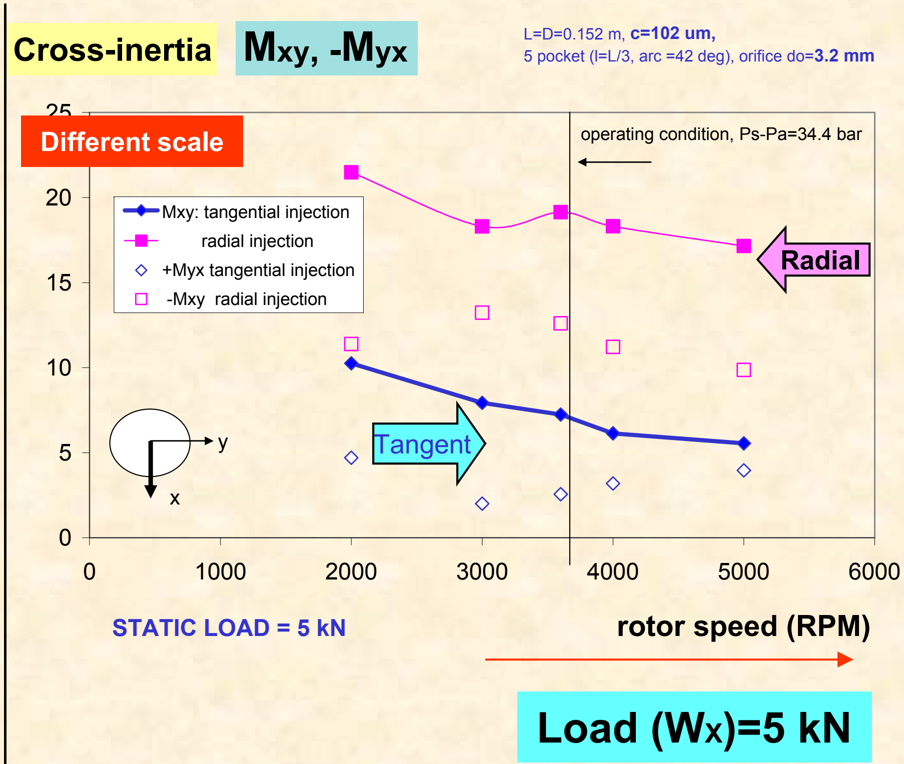
Load (W_X)=5 kN

- **At low speeds, all coefficients are large since operating eccentricity (*e/c*) is large (Hydrodynamic operation)**
- **At operating speed, direct damping $C_{XX} \sim C_{YY}$ & $C_{XY} = -C_{YX}$ since (*e/c*)~0 : near centered operation.**
- **From moderate to high speeds, $C_{XX} > C_{XY}$**
- **Tangential injection reduces cross-coupled damping –** small effect on rotordynamics

Figure 33 **1X Damping coefficients versus rotor speed.** Water hydrostatic bearing with radial and tangential injection

Hydrostatic Bearings – ADDED MASS

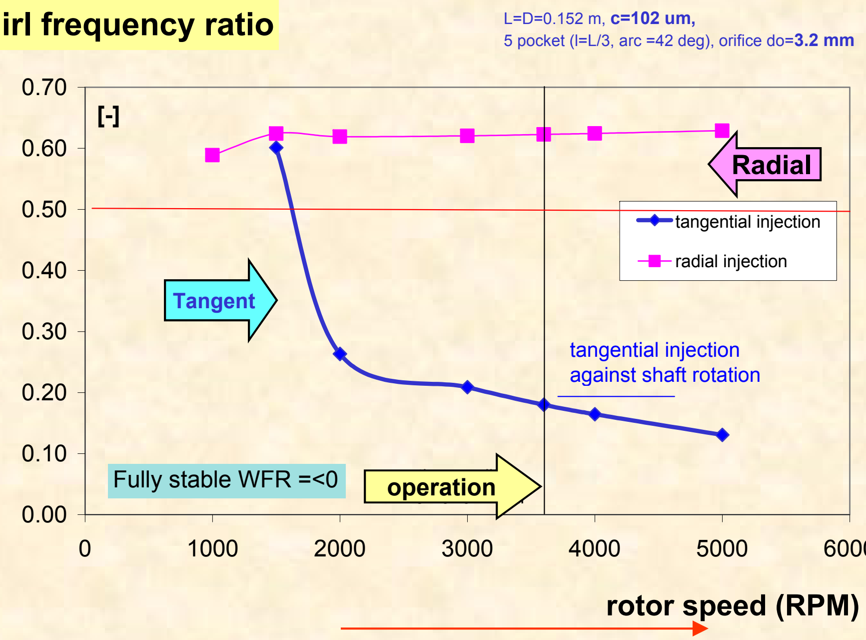


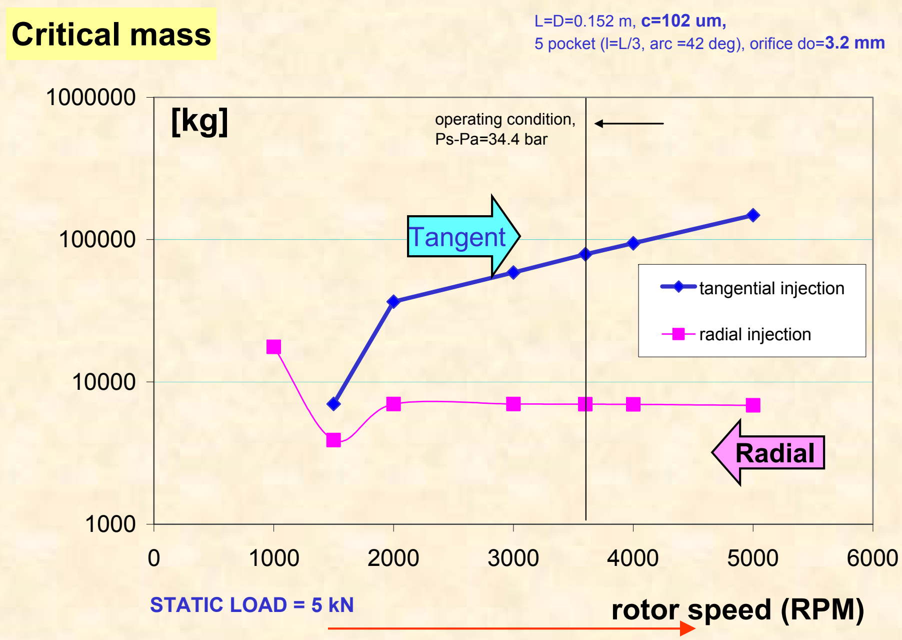


- **Large fluid inertia effect (~ 160 kg) >> 22 kg = solid steel piece (*L,D*)**
- **Direct inertia changes little with rotor speed**
- **<u>At operating speed</u>, direct inertia M_{XX}~M_{YY} & M_{XY}=-M_{YX} since (*e/c*)~0 : near centered operation.**
- **For all speeds, M_{XX} > M_{XY}**
- **Tangential injection reduces cross-coupled inertia –** small effect on rotordynamics

Figure 34 **1X Fluid inertia coefficients versus rotor speed.** Water hydrostatic bearing with radial and tangential injection

Whirl Frequency Ratio & Critical mass





Load (W_X)=5 kN

- **WFR ~ 0.60 for radial injection bearing, Critical mass ~ 8,000 kg**
- **<u>With tangential injection:</u> WFR drops rapidly towards 0.10 at operating point and beyond. Critical mass at least 10x larger than for radial injection**
- **In other applications (ex: LOx bearing), benefit of tangential injection is lost at very high speeds when hydrodynamic effects dominate flow**

-Critical mass results applicable to RIGID ROTOR only

Figure 35 **Whirl frequency ratio & Critical Mass versus rotor speed.** Water hydrostatic bearing with radial and tangential injection

Hydrostatic Bearings – Recommendations

Hydrostatic bearings have WFR > ~ 0.50 limiting their application to ~2x critical speed. Limiting speed condition can be worse if fluid is compressible and pockets are too deep & large area.

To reduce risk of hydrodynamic instability & increase bearing stability margin:

-Texture bearing surface

Proven with macro "rough" surfaces such as
Knurled, round hole and tire truck pattern tested successfully

-Angled injection against rotation

Retards circumferential flow swirl, effectiveness reduces at high rotor speeds,
Can induce backward whirl. Tested successfully

-Bearing asymmetry

Geometrically induce stiffness orthotropy ($K_{XX} > K_{YY}$)
Axial feed grooves, mechanical preload, etc. Tested & patented!

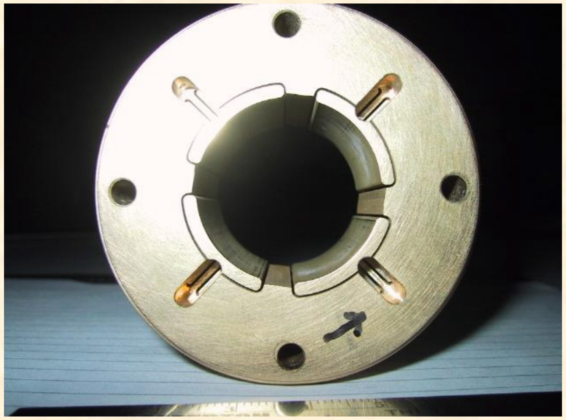
Hydrostatic Bearings – Recommendations

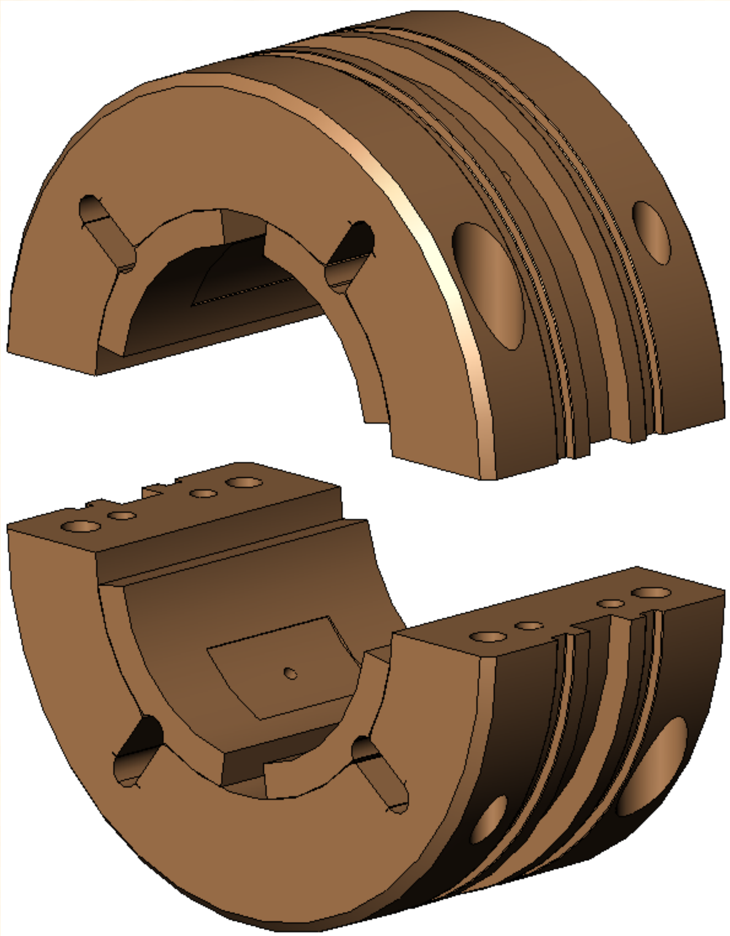
Flexure-pivot Tilting pad hybrid bearing

Tilting pads accommodate shaft motions to load direction, no generation of cross coupled stiffnesses, $K_{XY} \sim K_{YX} \sim 0$. No stability margin.

Wire EDM construction allows for reliable hydrostatic feeding.

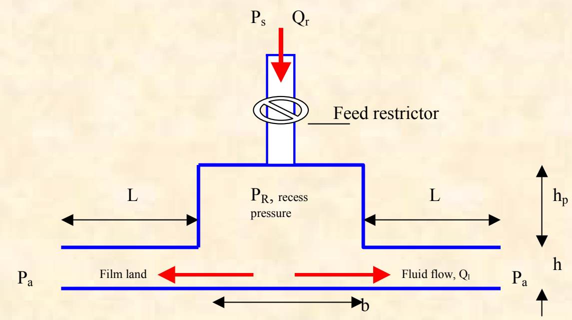
Tested in water, oil and gas applications





To avoid pneumatic hammer:

* reduce area pocket/land area to ~ 15-25%
* pocket depth to clearance ratio ~ 10 or less
* design and construct orifices without supply discharge volume (no pressure recovery zone)



Design parameter, $\Delta P << 1$, TYP **0.1** or less

$$DP = \frac{(P_R - P_A)}{K_{fluid}} \frac{3}{(Z+1)} \frac{N_{pocket}\,(V_{pocket} + V_{supply\ orifice})}{A_{rea\_bearing} \times C_{radial\_clearance}}$$

K_{fluid}: Fluid Bulk modulus

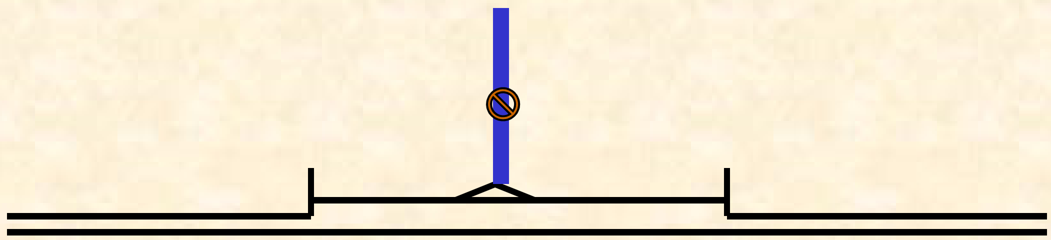
$A_{rea\_bearing} \sim \pi D L$

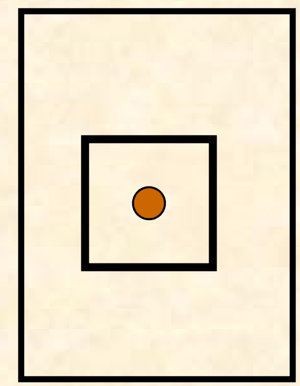
$$Z = \frac{(P_R - P_A)}{a\,(P_S - P_R)}$$

a=2 for orifice restrictors

P_S, P_R, P_A: supply, recess or pocket, discharge pressures

Hydrostatic Bearings for Cryogenic Turbo Pumps





Cryogenic fluid hydrostatic bearing design:

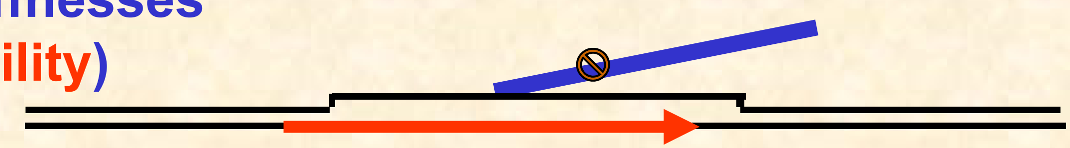
* small pocket area (15-25 % of total area)
* shallow pocket depth
* small or null orifice discharge volume

Application:

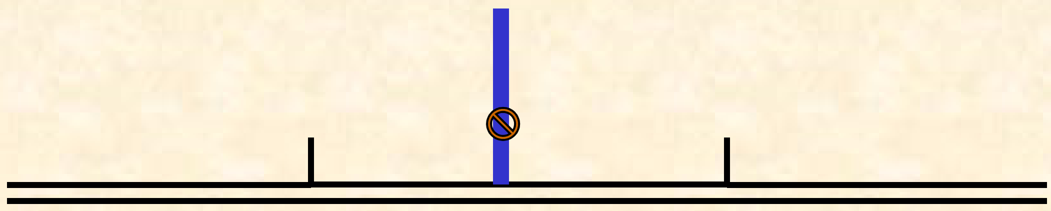
high surface speeds, low and high frequencies, **compressible** liquids (LO_2, LH_2, LN_2)

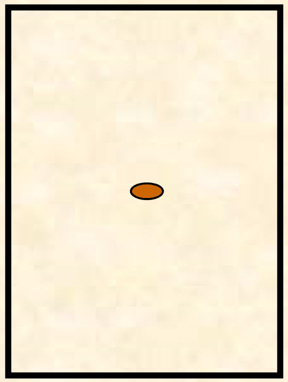
+ Angled injection against rotation to reduce cross-coupled stiffnesses (avoid hydrodynamic instability)

Nearly inherent restrictor type, i.e. orifice coefficient regulated by clearance



Gas hydrostatic bearing design

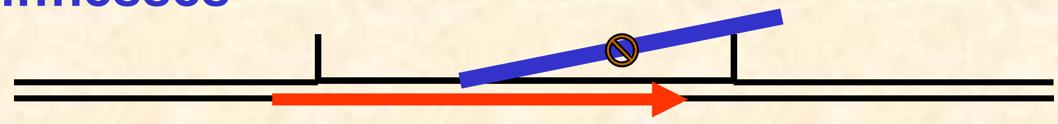




* Null pocket area (0 % of total area)
* **Inherent restrictor type, i.e. orifice coefficient regulated by clearance**

Applications:

low and high surface speeds, low and high frequencies,
All Gases (air, GO_2, GH_2, GN_2)
+ angled injection against rotation
to reduce cross-coupled stiffnesses



Hydrostatic Bearings – Model Validation

HYDROJET – radial hydrostatic bearings

Tests at TAMU with water (1000 psi (70 bar) max, 25 krpm max).
+ 20 bearings x 3 clearances & 2 pocket depths, different pocket shapes, macro-roughness (surface textured) bearings, angled injection.

Gas Honeycomb seals

Water Lomakin Bearings (Snecma-SEP)

Oil tilting and flexure pivot journal bearings

HYDROTHRUST – axial thrust hydrostatic bearings

NONE available in literature for high speed, high pressure (turbulent flows)

Concerns: centrifugal and advection fluid inertia cause severe fluid starvation in bearing and reduced axial stiffness coefficients due to effect of added mass coefficients

TESTs verified predictions

Needs in turbomachinery

largest power output to weight ratio, → *Rotordynamics*

reliability and performance, → *low friction and wear*

compact with low number of parts, → *automated agile processes*

extreme temperatures and pressures, → *coatings: nanopowders*
surface conditioning

environmentally safe and friendly. → *oil-free machinery*
inert gas buffer sealing

Desired features of support elements (bearings and seals)

High stiffness and damping coefficients
Linearity with respect to amplitudes of rotor motion
Avoidance of rotordynamic instability due to hydrodynamic effects
Controllable features to avoid surge/stall, etc.

Hydrostatic Bearings – Learn more

See

http://rotorlab.tamu.edu

Publications

For complete list of computational model predictions and comparisons to test data (over 30 journal papers and 10+ technical progress reports to NASA, USAF, P&W, Rocketdyne, Snecma-SEP, Northrop Grumman)

Bulk-flow Analysis of Hydrostatic Bearings

At Texas A&M: **Hydrojet® & Hydrothrust®**

Equations for flow in film lands of bearing	Mass conservation, Bulk-Flow momentum in circumferential and axial directions (2D), **Energy transport for mean flow temperature** Various surface temperature models Fluid inertia effects at entrance and exit flow regions.
Equations for flow in pockets of hydrostatic bearing	Global mass conservation: orifice inlet flow, flow from recess towards or from film lands, and rate of accumulation of fluid within pocket volume, Global momentum in circumferential direction due to angled injection. Global energy transport with adiabatic heat flow surfaces
Flow conditions	**Laminar, laminar to turbulent transition and fully developed turbulent bulk-flow model.** Turbulent flow closure model: Moody's friction factor including surface roughness. Fluid with variable properties *f*(P,T)

Model for Cryogenic Hydrostatic Bearings

Hydrostatic Bearings – Funding & Work to Date

240 k, Rocketdyne (1988-1991),
110 k, Pratt & Whitney (1991-92),
360 k, NASA GRC (1993-1996),
120 k, NASA MSFC (1998/99-2001/2)
229 k, Norhtop Grumman (2005-2007) - (USET Program)

Separate funding 1988-1995 for performance evaluation and rotordynamics measurements for validation of radial bearing tool prediction

All US turbo pump manufacturers and NASA, including SNECMA-SEP, use Hydrojet® and Hydrothrust® to model cryogenic fluid film bearings and seals. Other industries and Universities have benefited from technology.

USET Program (2005-2008)

CLIN 4.2.1.3.2 (a) non-linear forced response of fluid film bearing (98.5 k)
CLIN 4.2.1.3.2 (b) mixed flow regime – lift off response (130.5 k)
CLIN 4.2.1.3.7 Experimental Study of Hydrostatic / Hydrodynamic Thrust Bearings (788 k)

Hydrostatic Bearings – CFD models

3D CFD modeling of flow and pressure in POCKETS carried out by P&W (90's) and also by Universite de Poitiers funded by Snecma-SEP (France, 2001-2005).

OBJECTIVEs:

Predict complex flow field in pocket and extract "empirical" parameters for pressure loss at inlet to film lands and for development of pressure profile for angled injection bearings.

Insert empirical parameters into 2D-bulk-flow codes for prediction of bearing rotordynamic force coefficients.

Best published work: Mihai ARGHIR, Université de Poitiers, France.

FINDINGS:

3D-CFD predictions are NOT better than 2D-bulk flow predictions when compared to test data (TAMU-Childs). TAMU codes are still unsurpassed in terms of accuracy of predictions and speed of execution.