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ANNULAR CLEARANCE GAS SEALS IN THE 21ST CENTURY:

A REVIEW OF THE EXPERIMENTAL RECORD ON LEAKAGE AND DYNAMIC FORCE COEFFICIENTS, INCLUDING COMPARISONS OF MODEL PREDICTIONS TO TEST DATA

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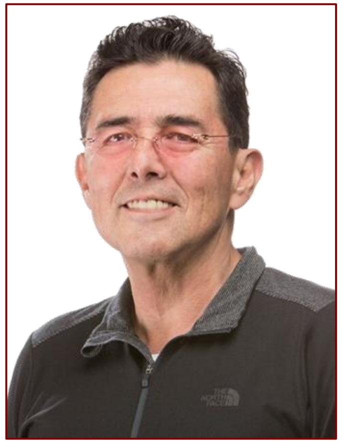


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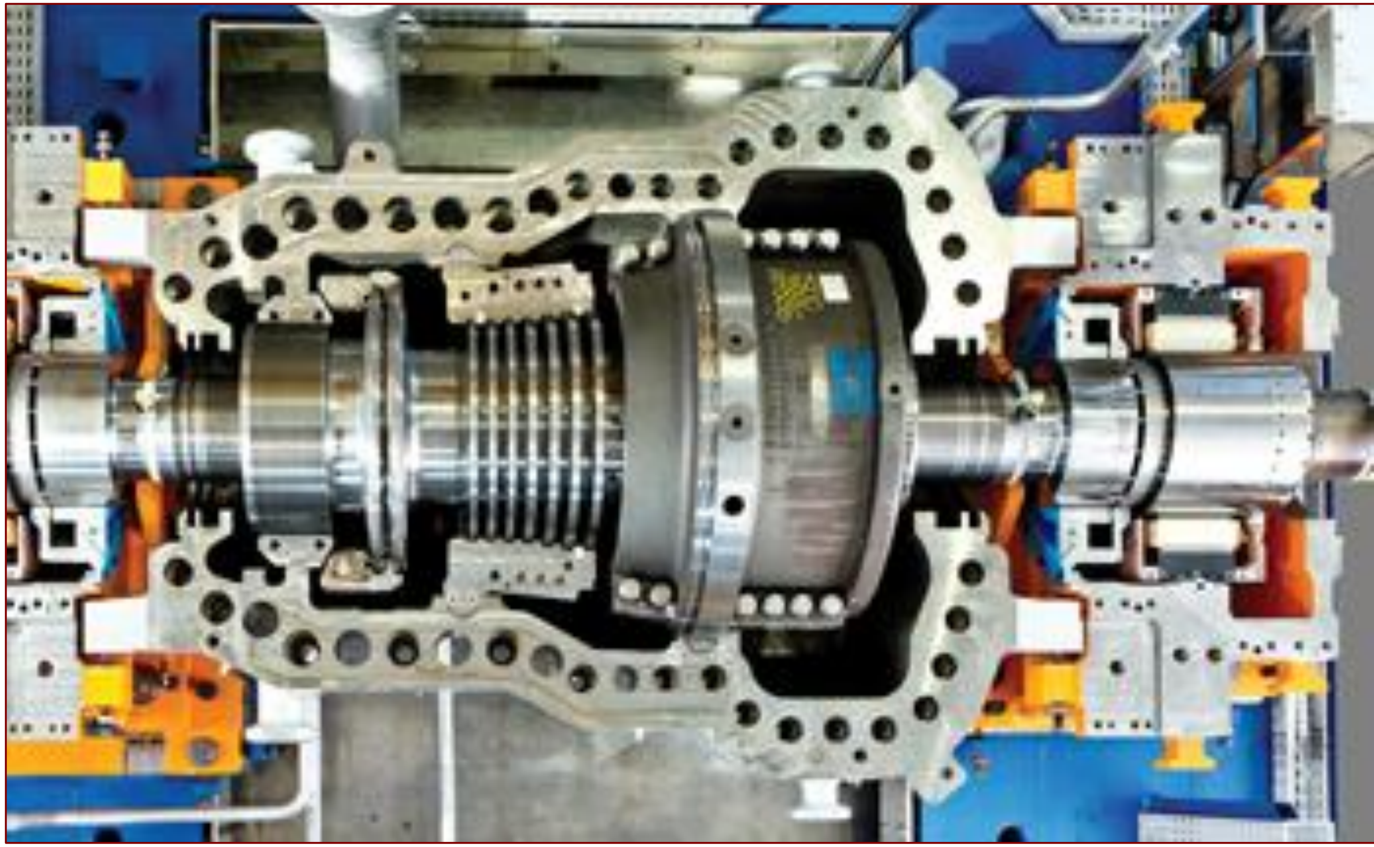
The ultimate aim

Machine efficiency & cost of operation rely on the accurate quantification of seals' leakage over the operating speed & pressure range, and life including wear of parts.



…. but are some seals better than others?

Childs, D. W., 1993, Turbomachinery Rotordynamics, Chap.5.

ABSTRACT

ANNULAR CLEARANCE GAS SEALS

MODELS AND MEASUREMENTS FOR LEAKAGE, FORCE COEFFICIENTS AND THEIR EFFECT ON ROTOR STABILITY

Turbomachinery seals are engineered to maintain efficiency and power delivery by minimizing leakage. Seals also appreciably affect the system rotordynamic behavior due to their relative position within a turbomachine. The tutorial reviews the experimental record on gas seals as published in the 21st century, and gives insight on the physical models predicting leakage and dynamic force coefficients. Unlike experiences in the past century, damper seals offer a remarkable opportunity to control the leakage and tailor the rotordynamic performance and stability of modern rotating machinery.

OUTLINE

ANNULAR CLEARANCE GAS SEALS

MODELS AND MEASUREMENTS FOR LEAKAGE, FORCE COEFFICIENTS AND THEIR EFFECT ON ROTOR STABILITY

1. Overview of annular clearance seals
2. Bulk-flow and CFD models for seal analysis
3. Seals leakage & their effective clearance
4. Seals force coefficients – an appraisal of the exp record
5. Closure – The road Ahead

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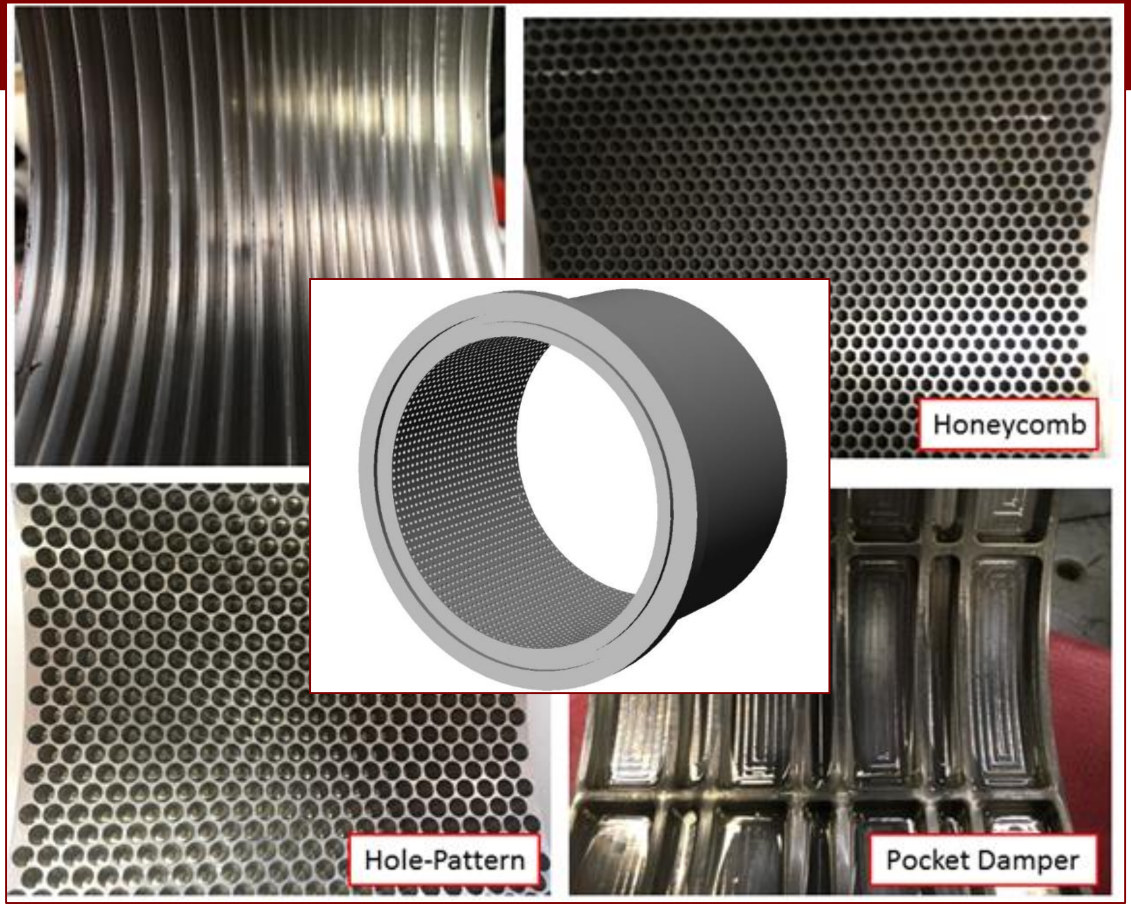
What will you learn today?

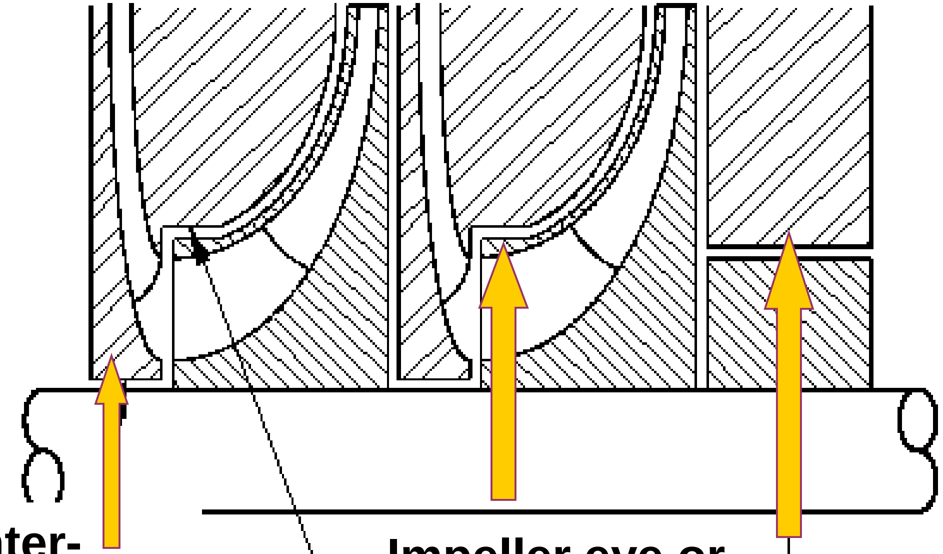
- **Types of clearance seals in turbomachinery and their characterization in terms of an effective clearance quantifying their leakage.**
- **Seals' force coefficients and their impact on rotordynamics and stability.**
- **Details on available models: accuracy and validation against test data.**
- **Opportunities to employ seals as load bearing elements with large energy dissipation ability.**

Annular clearance seals

Labyrinth seals, honeycomb seals, etc. separate regions of high pressure and low pressure. 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.

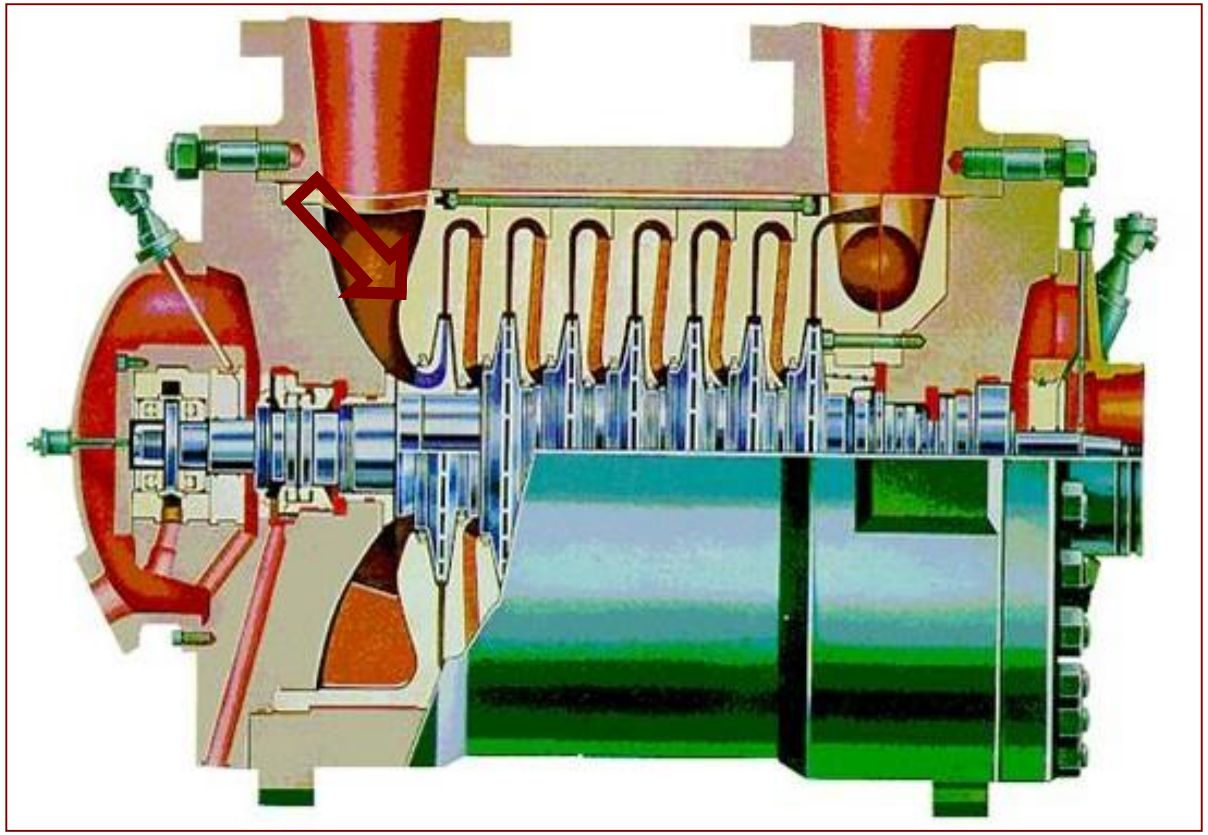


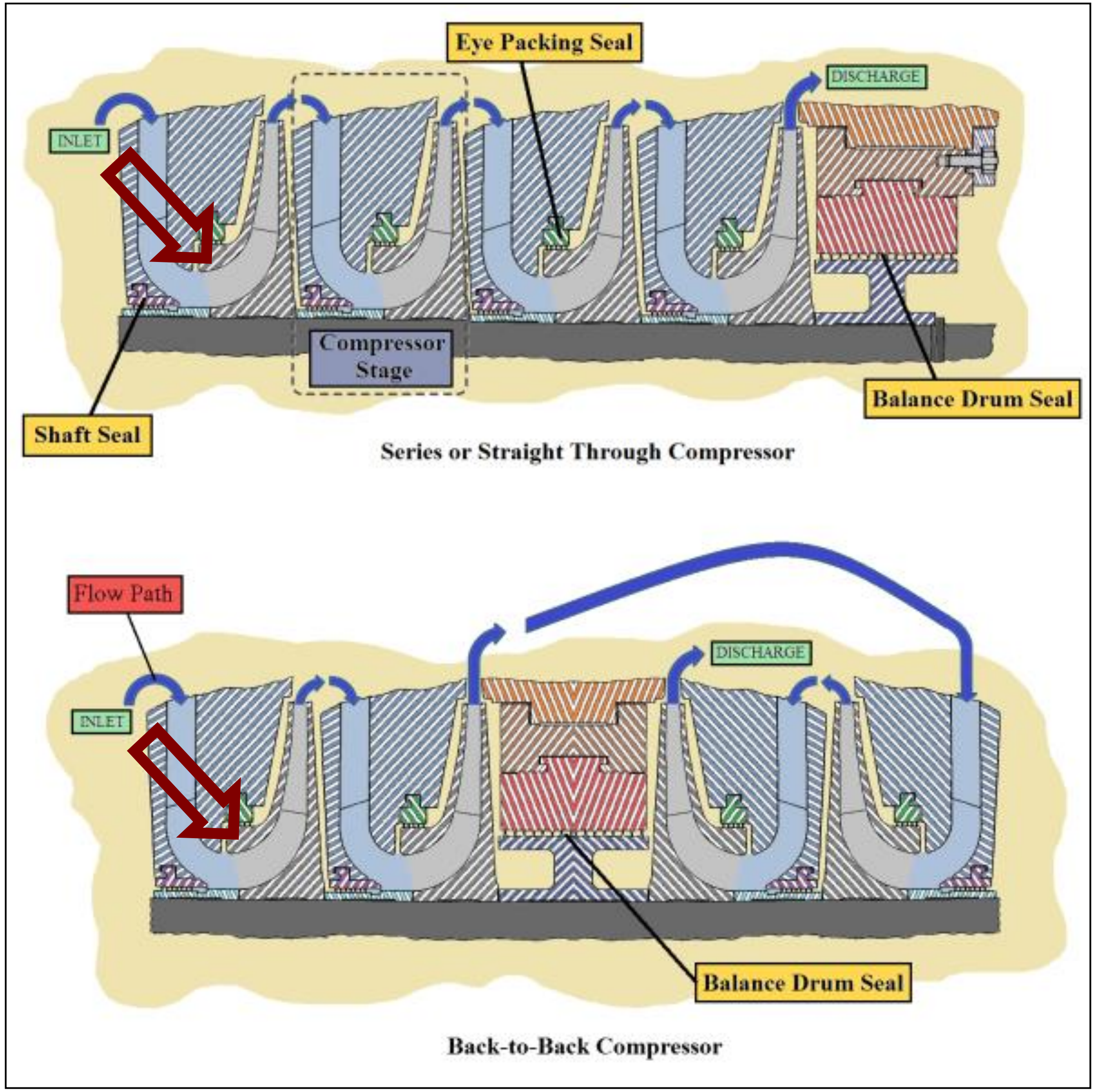
Common Seal Types



Seals in a Multistage Centrifugal Pump or Compressor

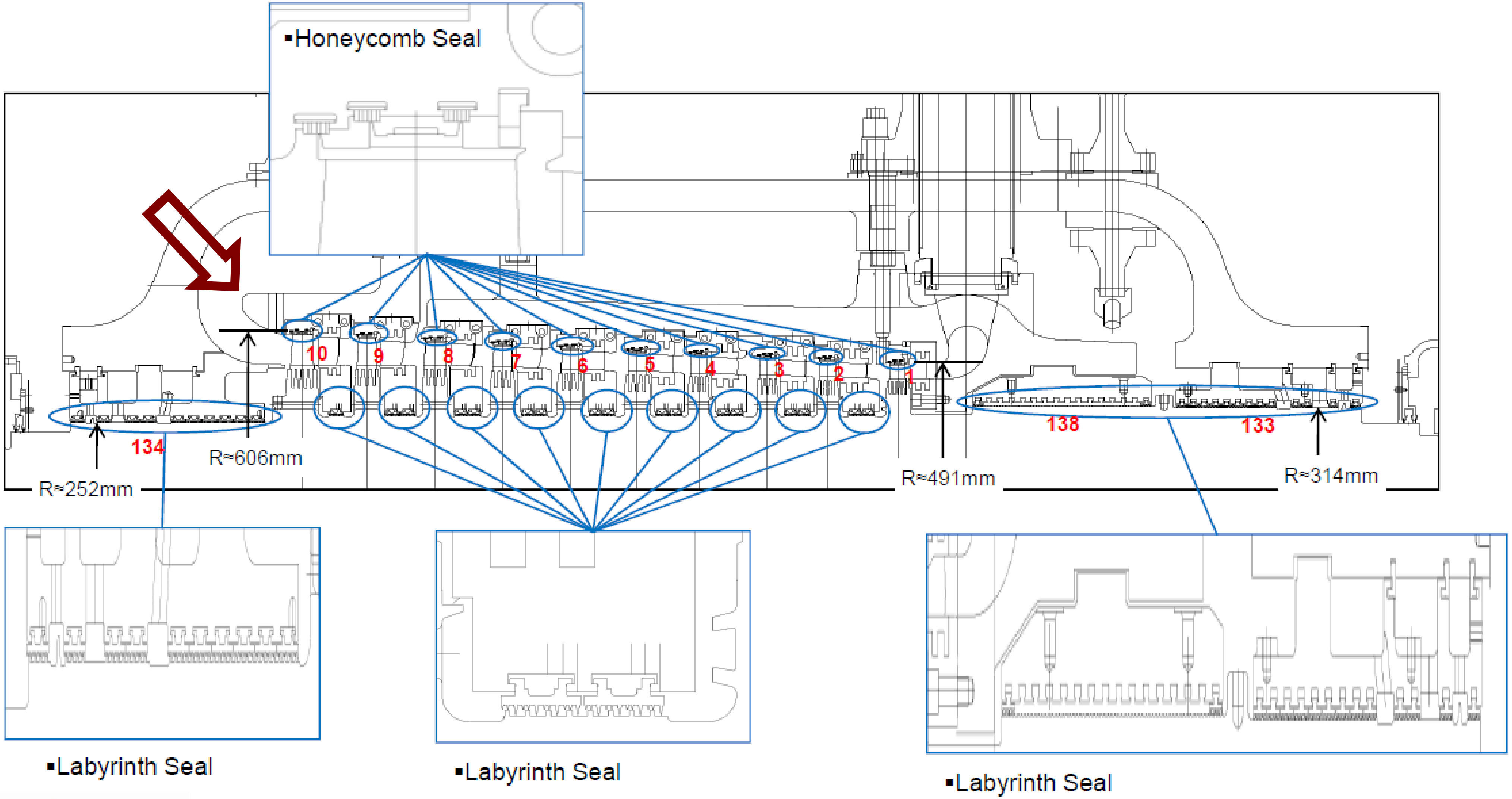
Count the seals in a barrel compressor…..





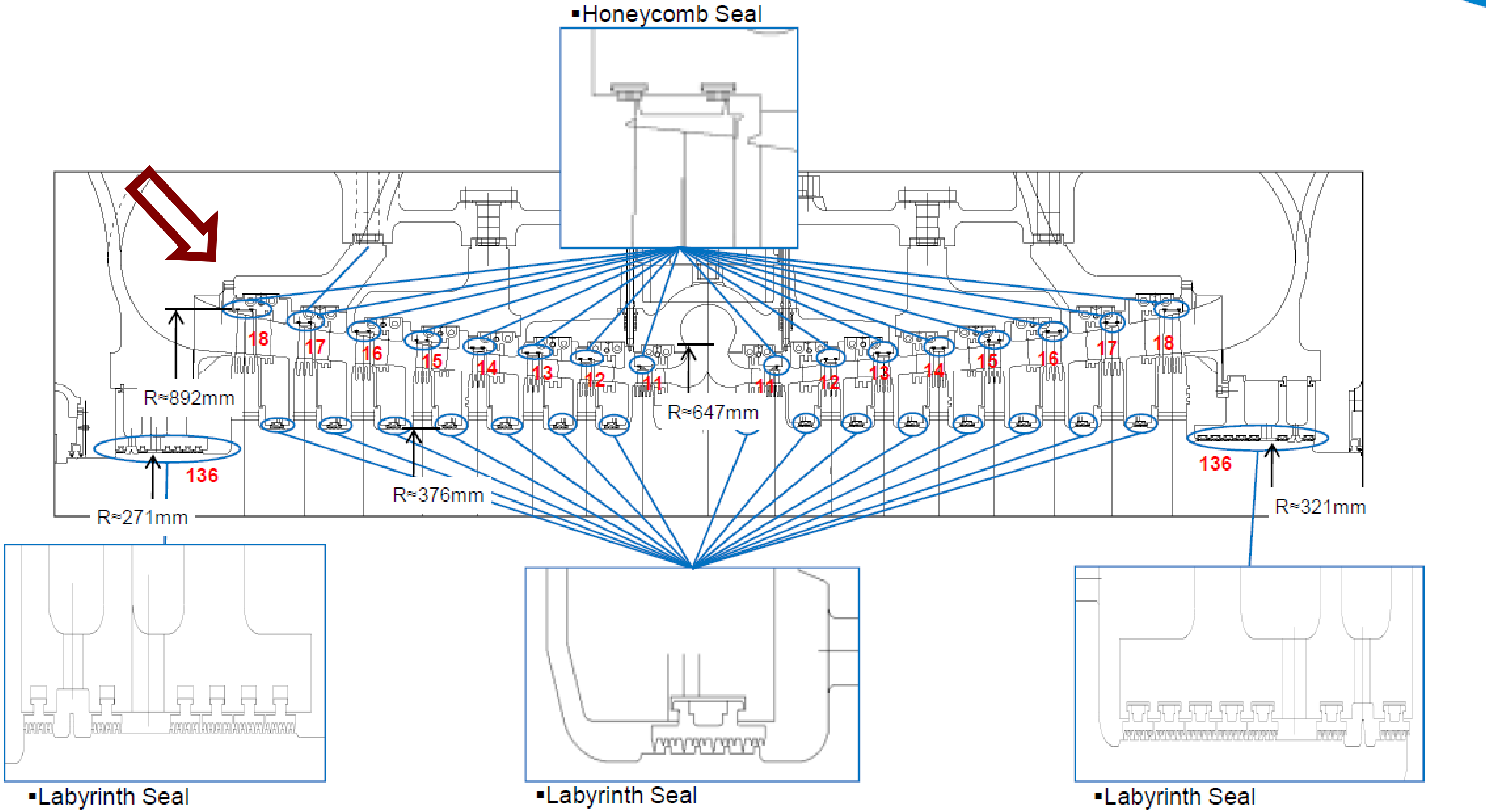
Count more seals in a HP steam turbine…..

▪50Hz 660MW – HP turbine

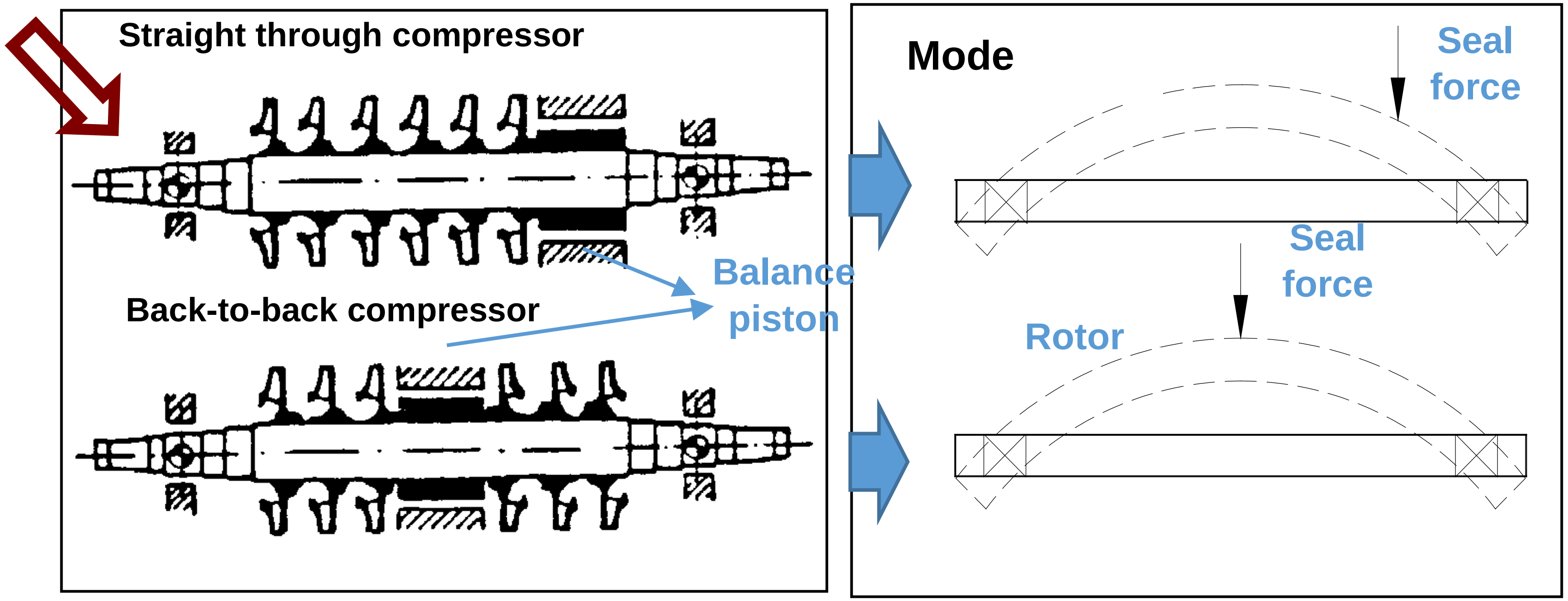


Keep counting… in an IP steam turbine

▪50Hz 660MW – IP turbine



Seals and their impact on rotordynamics



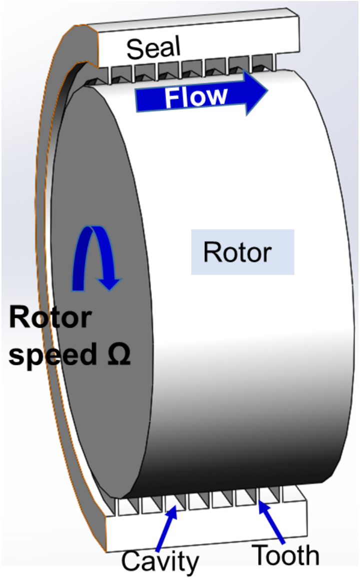
Straight-Through and Back-to-back Compressors and 1st Mode Shapes

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

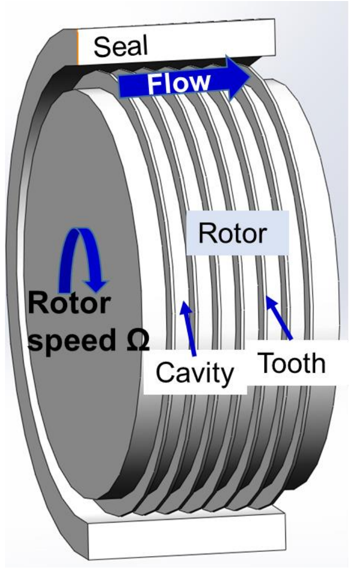
Labyrinth seals

Most common type of seal

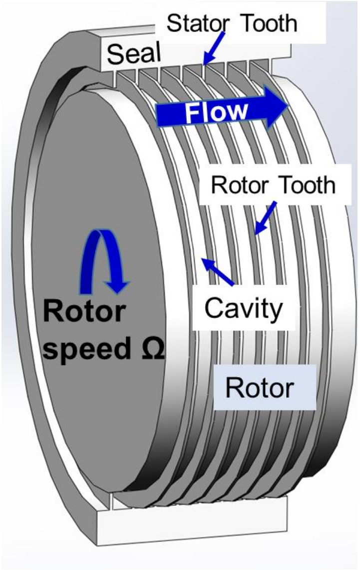
(1) TOS: all teeth on stator
(2) TOR: all teeth on rotor
(3) ILS : teeth on both rotor and stator



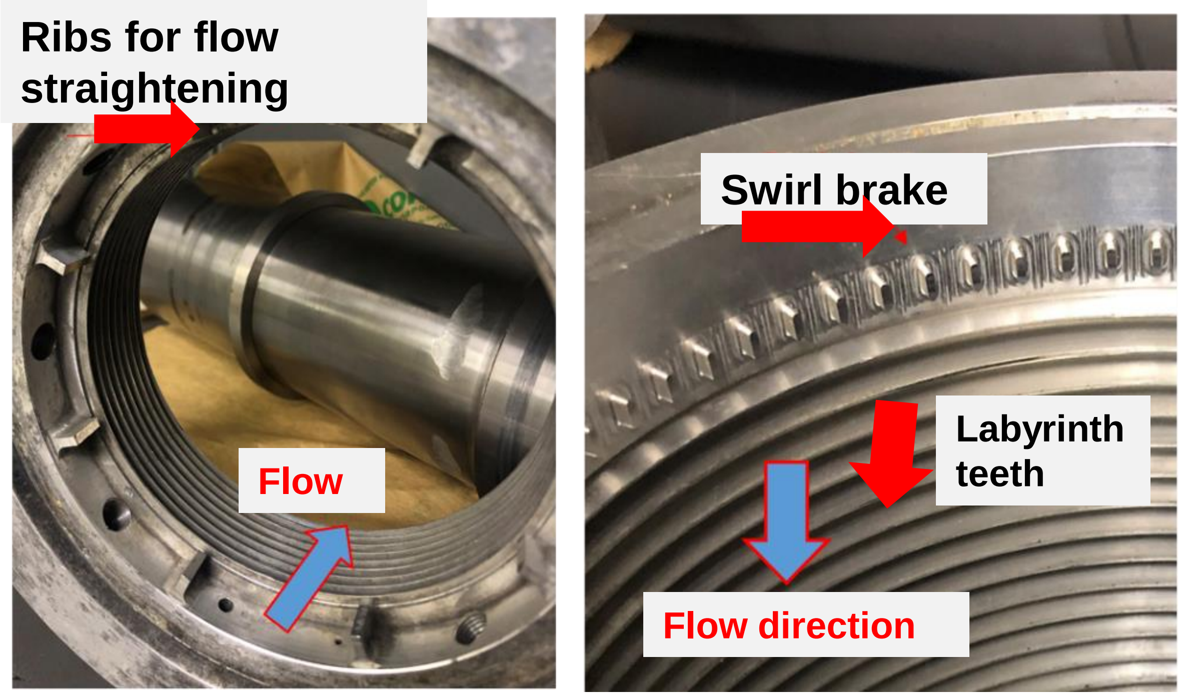
TOS



TOR



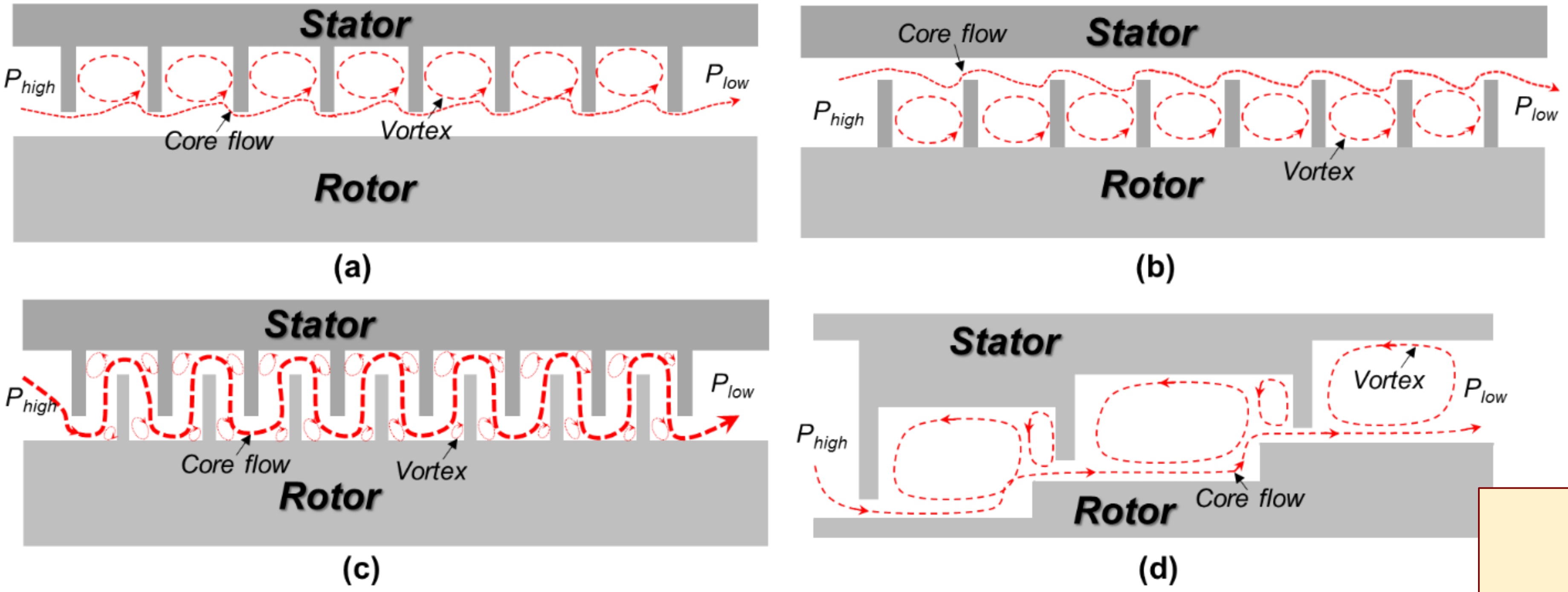
ILS



Labyrinth seals, one with a swirl brake

- Restrict secondary flow;
- Affect rotor system dynamic stability.

How do Labyrinth Seals (LS) do their work?



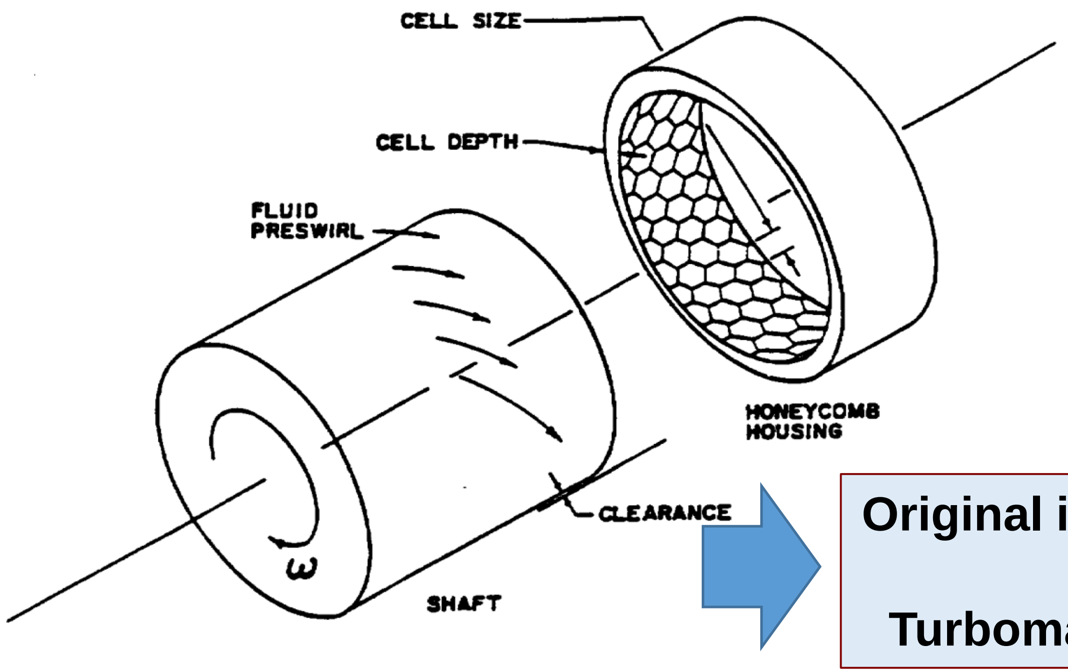
Core Flow: jet flow along leakage path plays dominant role. Pressure drop across sharp teeth dissipates kinetic energy.

Vortex Flow: Vortices (recirculation zones) in a cavity contribute to mechanical energy dissipation.

→ Increase flow resistance from vortical flow cells.

Textured surface seals

Honeycomb seal and round hole pattern seal →

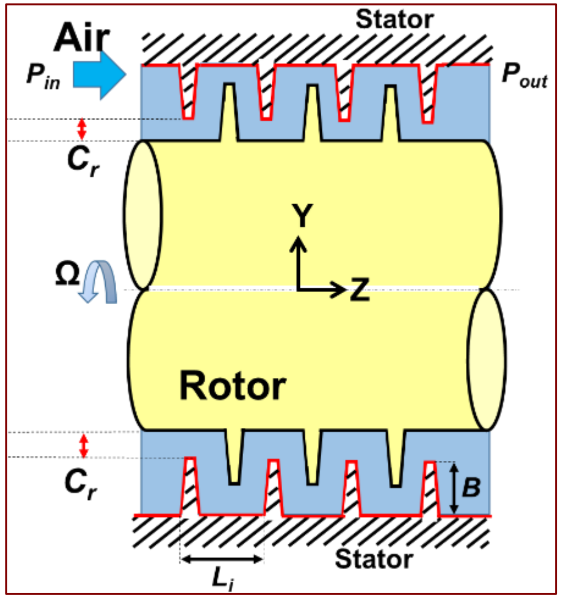


Original idea from G. Von Pragenau, 1982, "Damping Seals for Turbomachinery," NASA-TP- 19827

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

Damper seals provide large levels of damping to reduce vibrations and also generate large direct stiffness for added rotor support.

Models for prediction of seal leakage

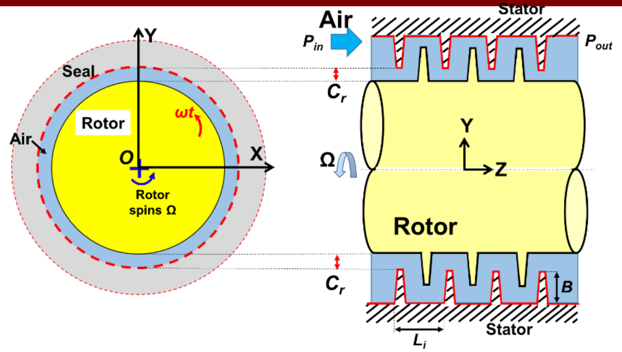


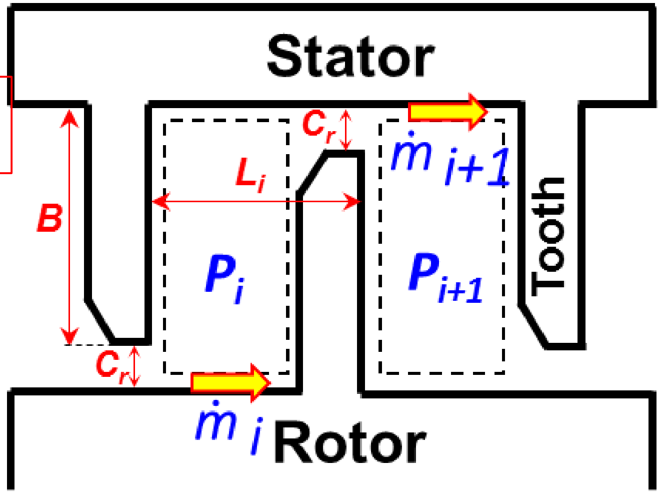
Predictive tools: BFM *vs.* CFD

	Pros	Cons
Bulk-Flow Model (BFM)	✓ Quick ✓ Easy set up	❑ Lacks accuracy ❑ Needs empirical coefficients
Computational Fluid Dynamics (CFD)	✓ High fidelity ✓ No empirical coefficients required	❑ Computationally expensive ❑ Requires knowledge on CFD (pre and post processing)

Available computational capability and desire for extreme fidelity push CFD analyses into common engineering practice

Bulk-Flow Model (BFM) for LS





$\dot{m}_i = \dot{m}_{i+1}$

Neumann's Leakage Eq. → Flow through clearance under a tooth

Used since 1930's

$$\dot{m} = \mu_{1i}\mu_{2i}\left(\pi D C_r\right)\sqrt{\frac{P_{i-1}^2 - P_i^2}{R_g T}}$$

Continuity Equation →

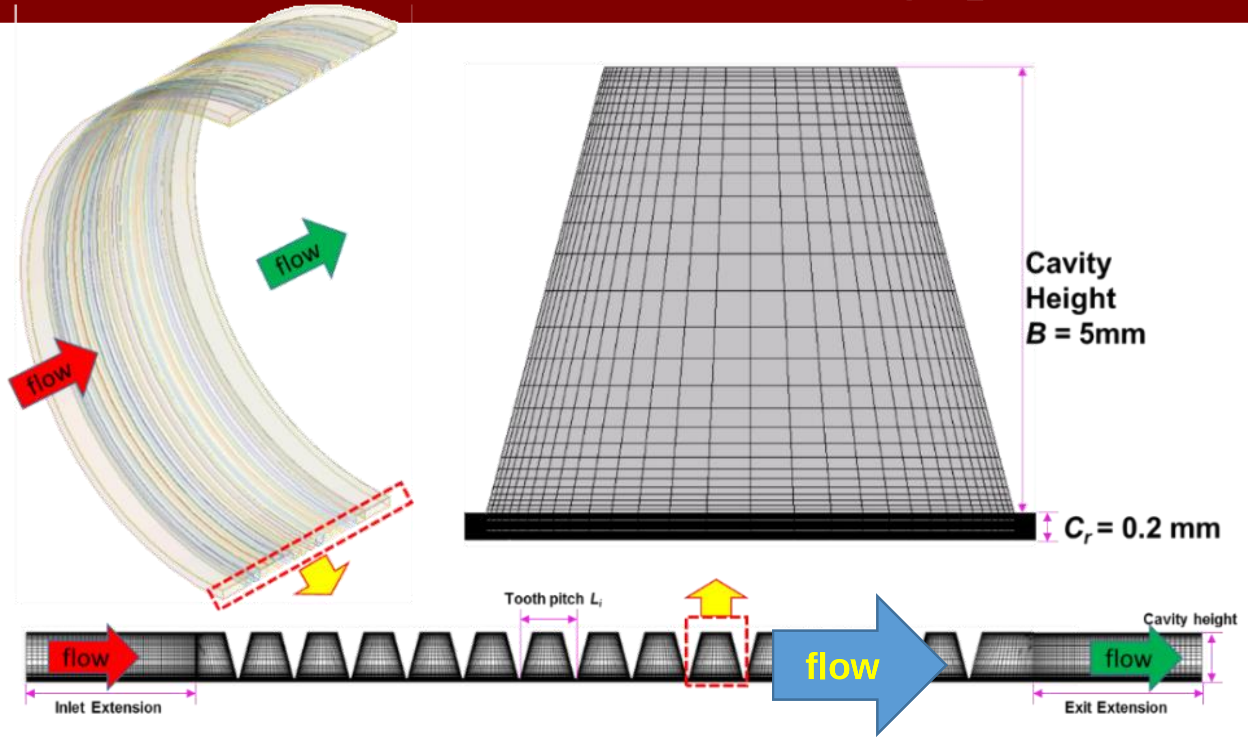
$$\frac{\partial(\rho_i A_i)}{\partial t} + \frac{\partial(\rho_i U_i A_i)}{R_s \partial\theta} + \bar{\dot{m}}_{i+1} - \bar{\dot{m}}_i = 0$$

Circumferential momentum in cavities →

$$\frac{\partial(\rho_i U_i A_i)}{\partial t} + \frac{\partial(\rho_i A_i U_i^{\,2})}{R_s\ \partial\theta} = -\frac{A_i}{R_s}\frac{\partial P_i}{\partial\theta} + (\tau_{r_i} a_{r_i} - \tau_{s_i} a_{s_i})L_i$$

CFD mesh and typical output

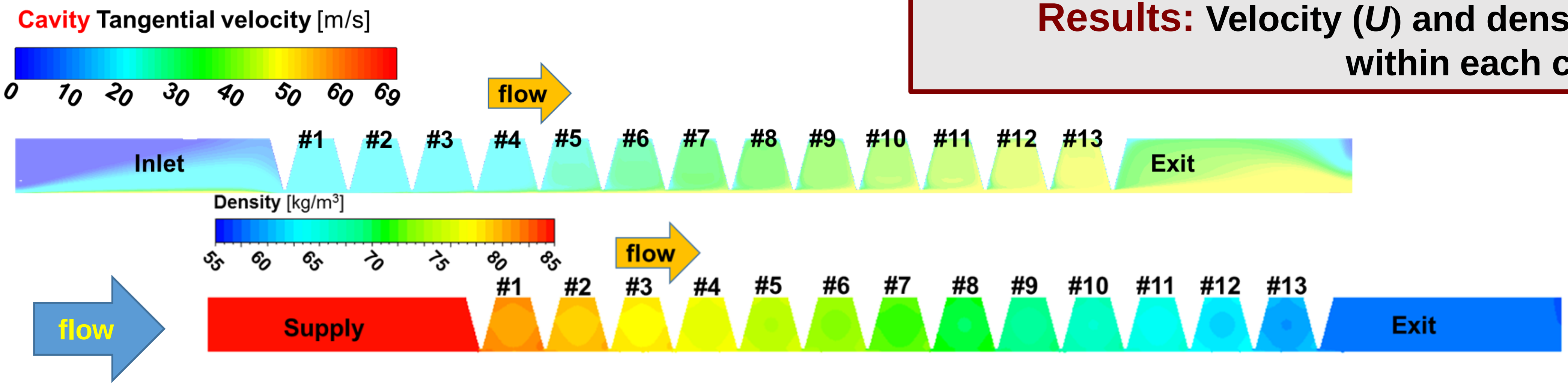
Mesh ~8 Million nodes



	Node number/ mesh size
Radial clearance	30
Tooth section	30
Cavity depth/length	30
Circumferential	180 (2° apart)
Min. mesh orthogonal quality	0.99

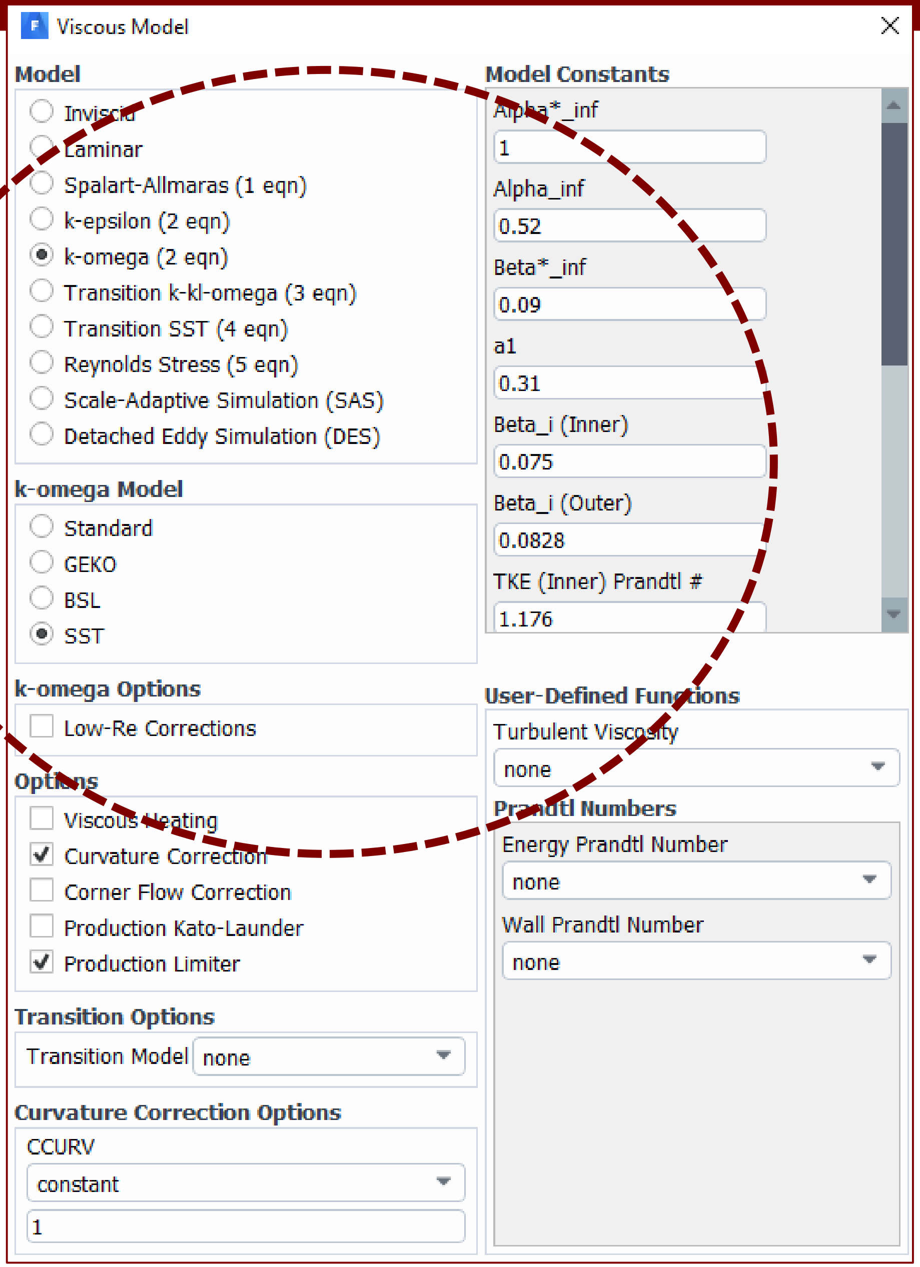
P_{in} = 7.3 MPa, P_{out} = 5.1 MPa, rotor speed 12 krpm

Results: Velocity (U) and density (ρ) within each cavity:

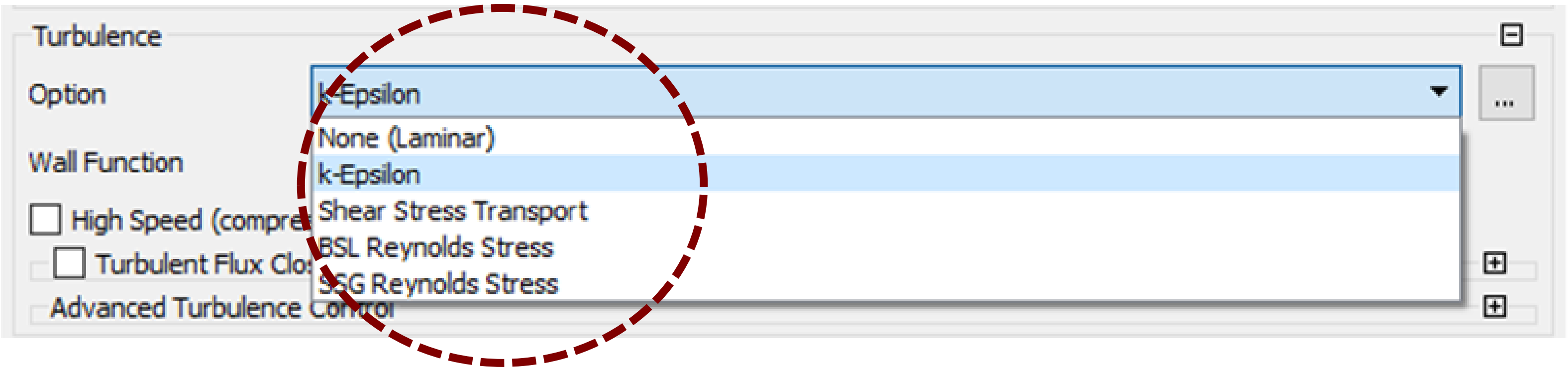


Commercial CFD model options

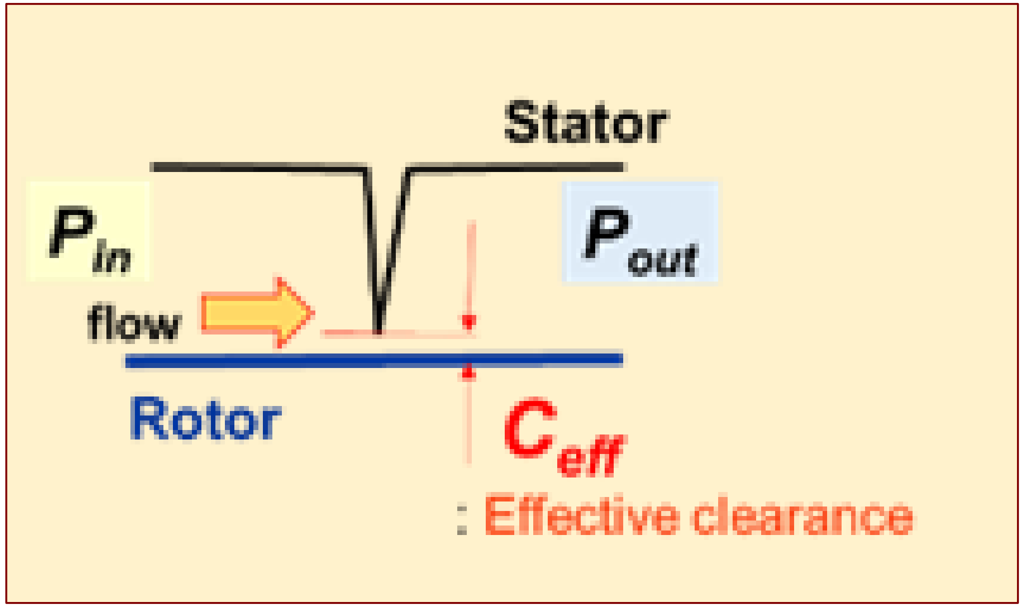
Too many options and little insight.



Recommend use (a) κ–ω SST (shear stress transport) model or realizable κ–ε model, both with a curvature correction function reducing model predictions sensitivity to streamlines' curvature and (shaft) rotation.



Characterization of seal leakage with an effective clearance



Flow Factor ϕ

demonstrates independence to seal size (diameter *D*) and inlet flow conditions in pressure (P_{in}) and temperature (*T*).

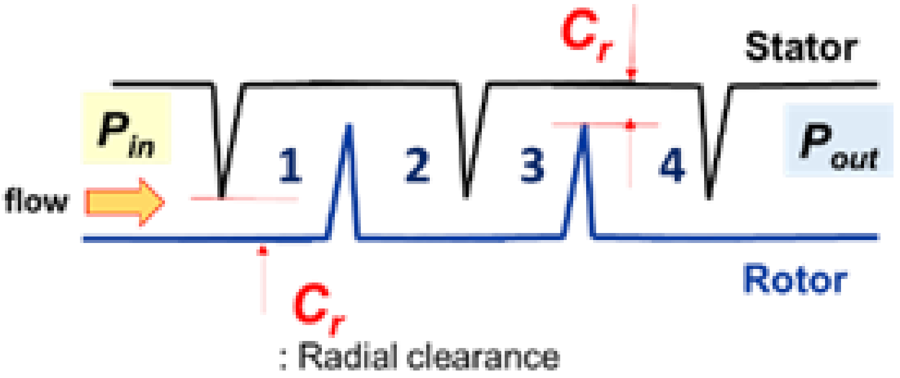
$$\phi = \dot{m}\sqrt{T} \big/ (P_{in} D) \quad \left[{}^{kg\sqrt{K}} \big/ {}_{MPa \cdot m \cdot s} \right]$$

…. but still a dimensional(ly odd) parameter

Delgado, I., and Proctor, M., 2006, AIAA–2006–4754.

Effective clearance for LS

Use **Neumann's equation** to define a **modified flow factor** and to represent seal as an equivalent **single tooth seal**.

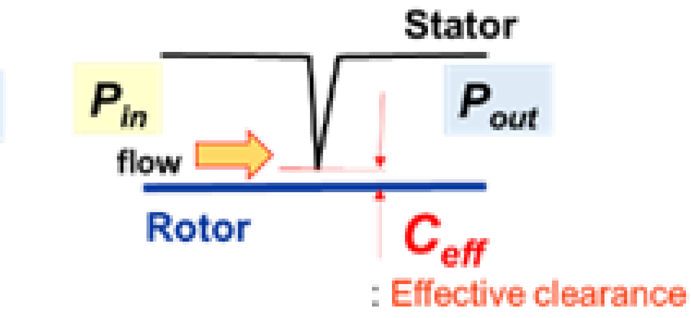


(a) Interlocking labyrinth seal

$$\dot m_1 = \mu_{11}\mu_{21}\left(\pi D C_r\right)\sqrt{\frac{P_{in}^2 - P_1^2}{R_g T}} =$$

$$\dot m_2 = \mu_{12}\mu_{22}\left(\pi D C_r\right)\sqrt{\frac{P_1^2 - P_2^2}{R_g T}} =$$

$$\dot m_N = \mu_{1N}\mu_{2N}\left(\pi D C_r\right)\sqrt{\frac{P_N^2 - P_{out}^2}{R_g T}}$$

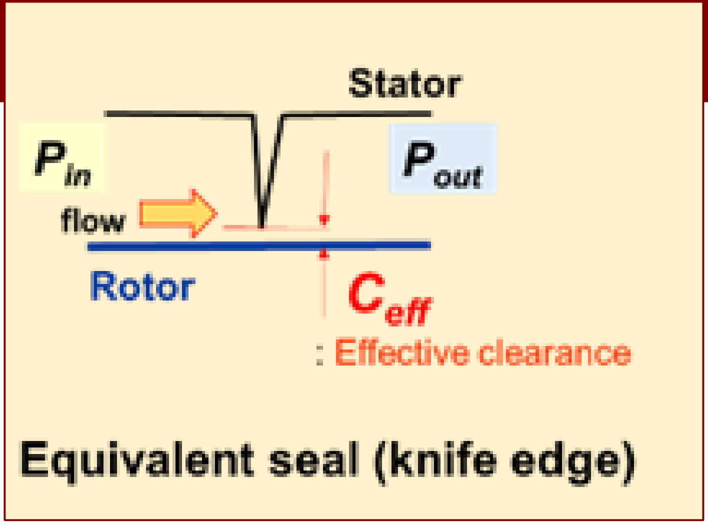


(b) Equivalent seal (knife edge)

$$\dot m = \left(\pi D C_{eff}\right)\sqrt{\frac{P_{in}^2 - P_{out}^2}{R_g T}}$$

C_{eff} is an effective clearance.

Modified Flow Factor $\bar{\Phi}$



$$\dot{m} \sim \left(\pi D C_{eff}\right)\sqrt{\frac{\left(P_{in}^2 - P_{out}^2\right)}{R_g T}} = \left(\pi D C_{eff}\right)\frac{P_{in}}{\sqrt{R_g T}}\sqrt{1-\left(\frac{P_{out}}{P_{in}}\right)^2}$$

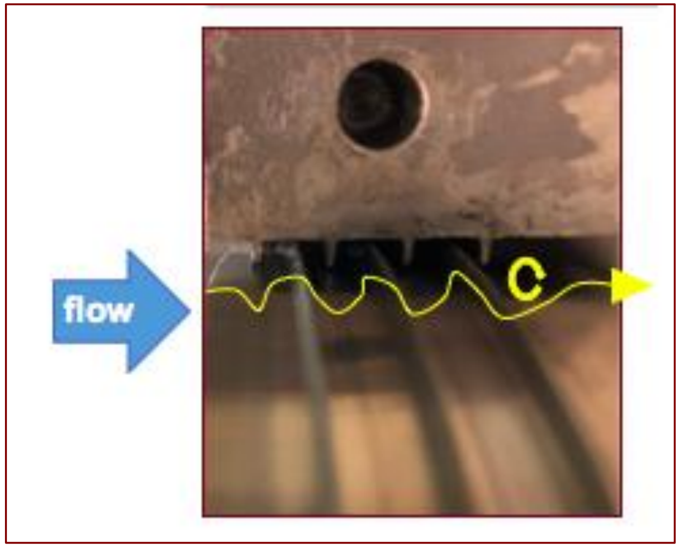
$$\phi = \dot{m}\sqrt{T} / (P_{in} D)$$

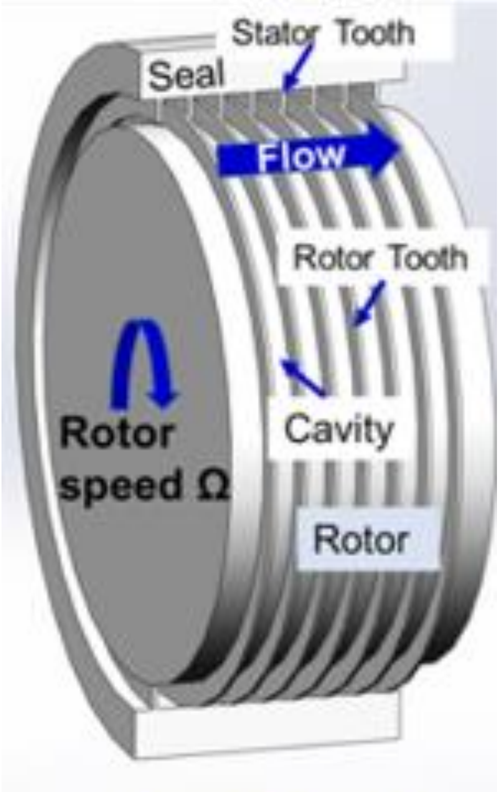
$$\bar{\Phi} := \frac{\phi}{\sqrt{1-PR^2}} = \frac{\dot{m}\sqrt{T}}{D\, P_{in}\sqrt{1-PR^2}} \sim \pi\, C_{eff}\, \frac{1}{\sqrt{R_g}}$$

C_{eff} = effective clearance = $c_d\, C_r$

c_d ← loss coefficient.

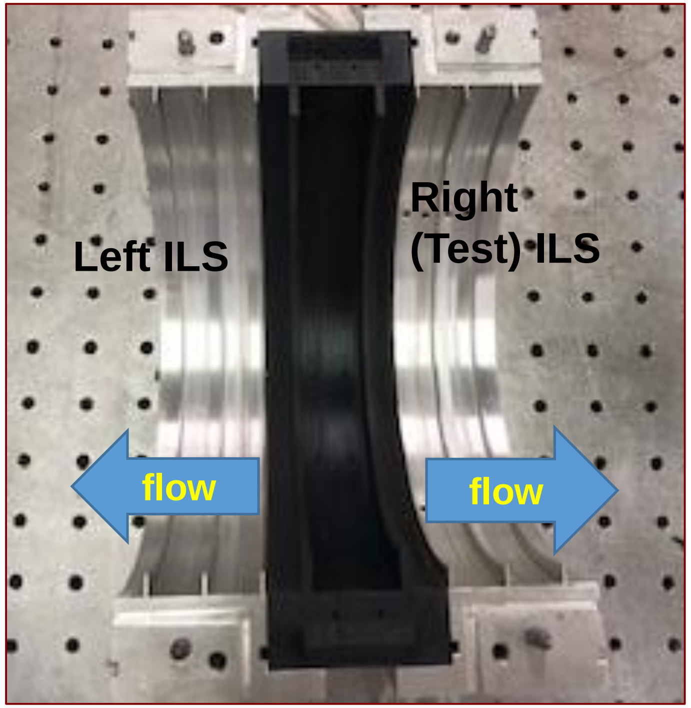
Leakage for interlocking labyrinth seal (ILS)





Effect of clearance on leakage

Seal Geometry	Rotor Diameter, D	150 mm
	Overall length, L	45 mm
	Radial clearance, C_r	0.13, 0.2, 0.3 mm
	Width at tip, B_t	0.25 mm
Air Properties	Density, ρ @25 °C	1.2 kg/m^3
	Temperature, T	297 K
	Sound speed, a_s	314 m/s
	Kinematic viscosity, ν	1.86×10^{-5} m^2/s
	Inlet pressure, P_{in}	292 ~ 1,150 kPa
	Pressure ratio, $PR = P_{out}/P_{in}$	0.3, 0.5, 0.8
	Rotor speed, Ω	0, 3, 5, 7.5, 10 krpm
	½ D Ω_{max}	0 ~ 79 m/s



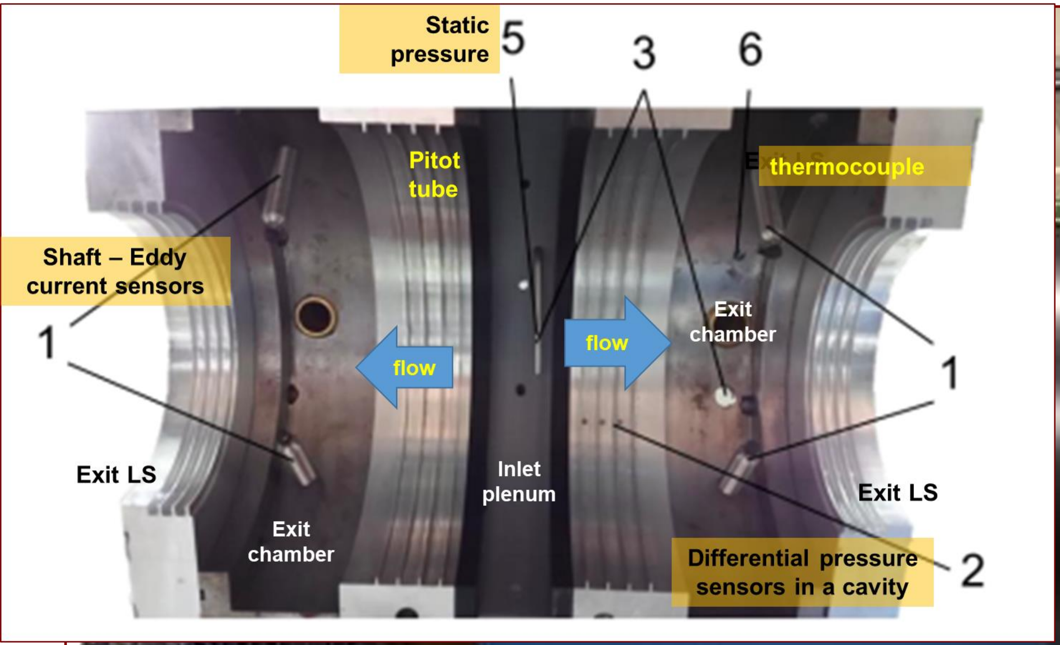
Five teeth ILS

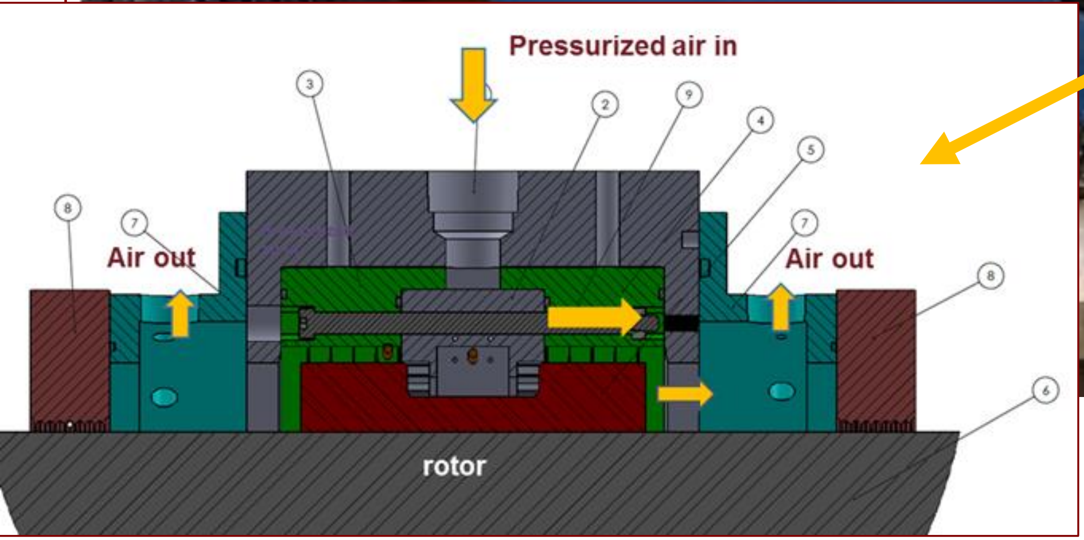
Clearance =
0.3 mm, 0.2 mm & 0.13 mm

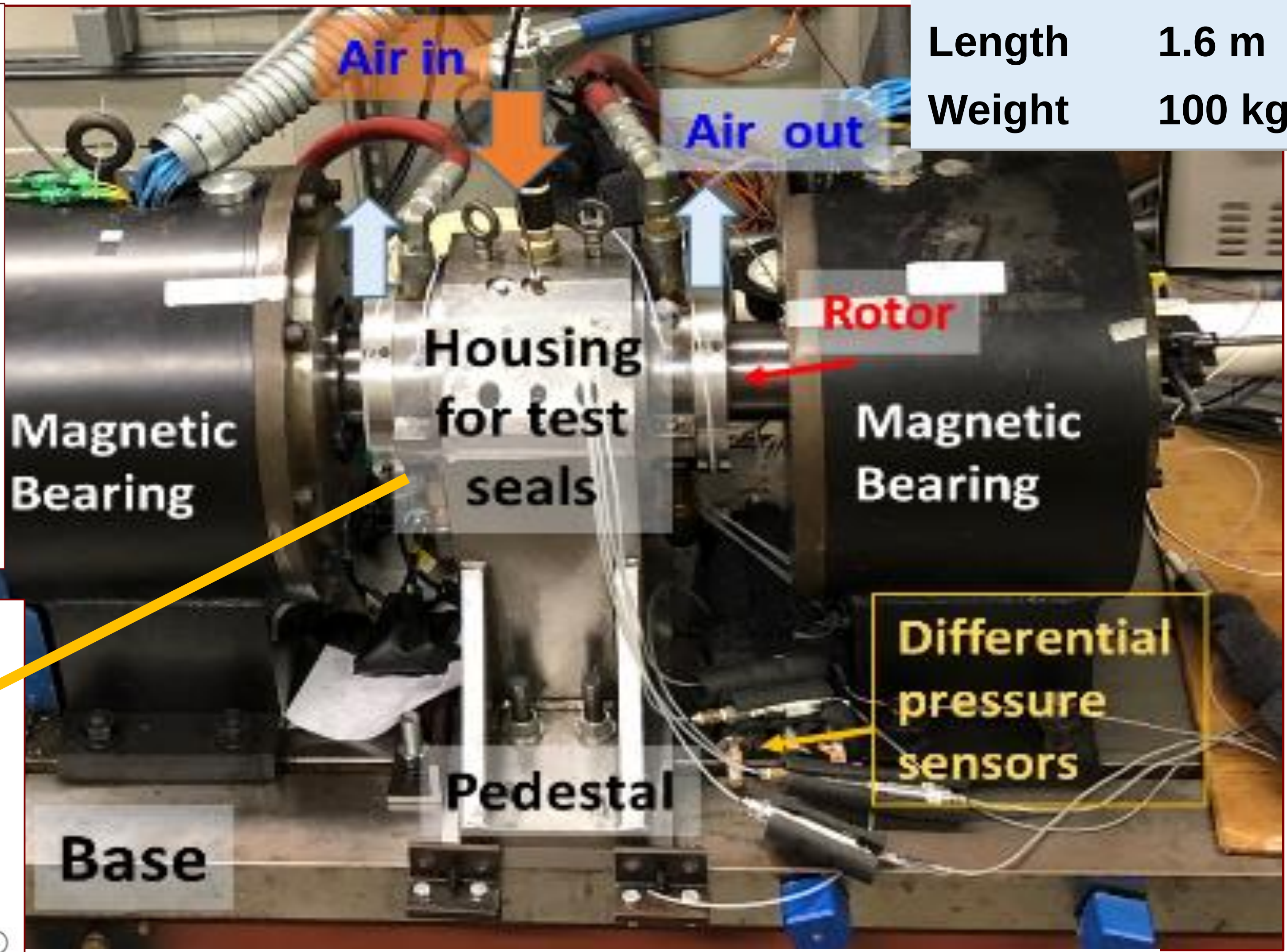
L. San Andrés, J. Yang and R. Kawashita, **2021,** "On the Effect of Clearance on the Leakage and Cavity Pressures in an Interlocking Labyrinth Seal Operating With and Without Swirl Brakes: Experiments and Predictions," ASME J. Eng. Gas Turbines Power, 143 (3).

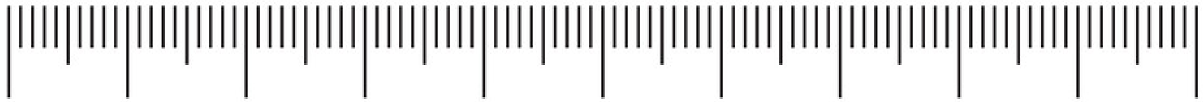
Test rig & seals

Steel rotor	
Diameter	150 mm
Length	1.6 m
Weight	100 kg



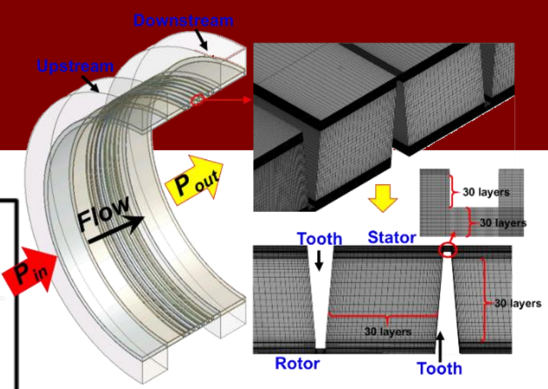


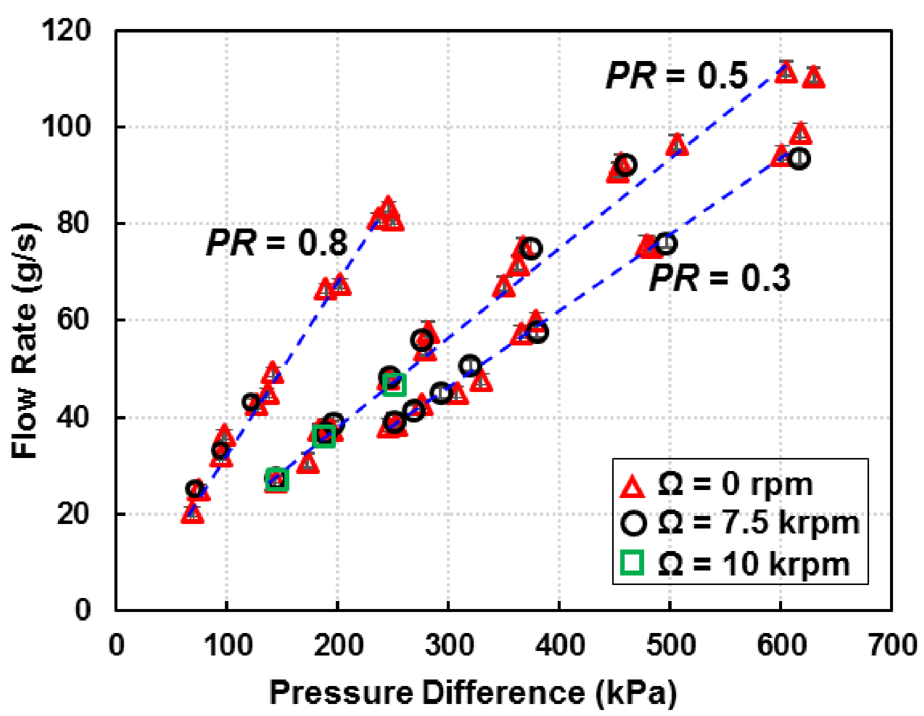


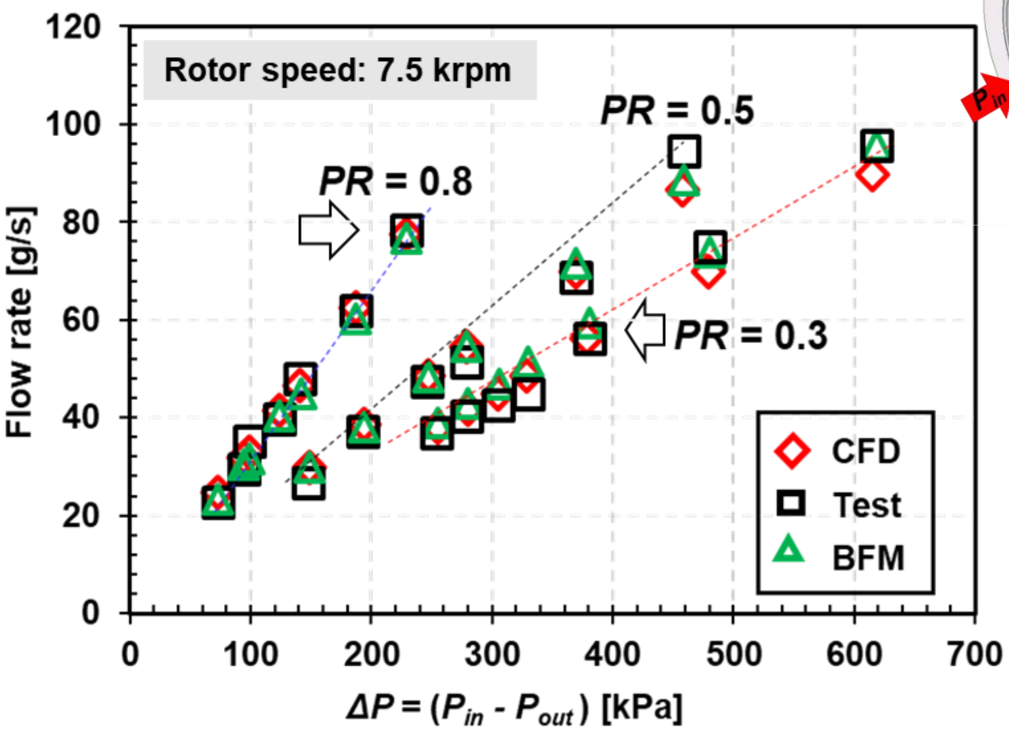


0 m 1 m

ILS Leakage vs. pressure difference







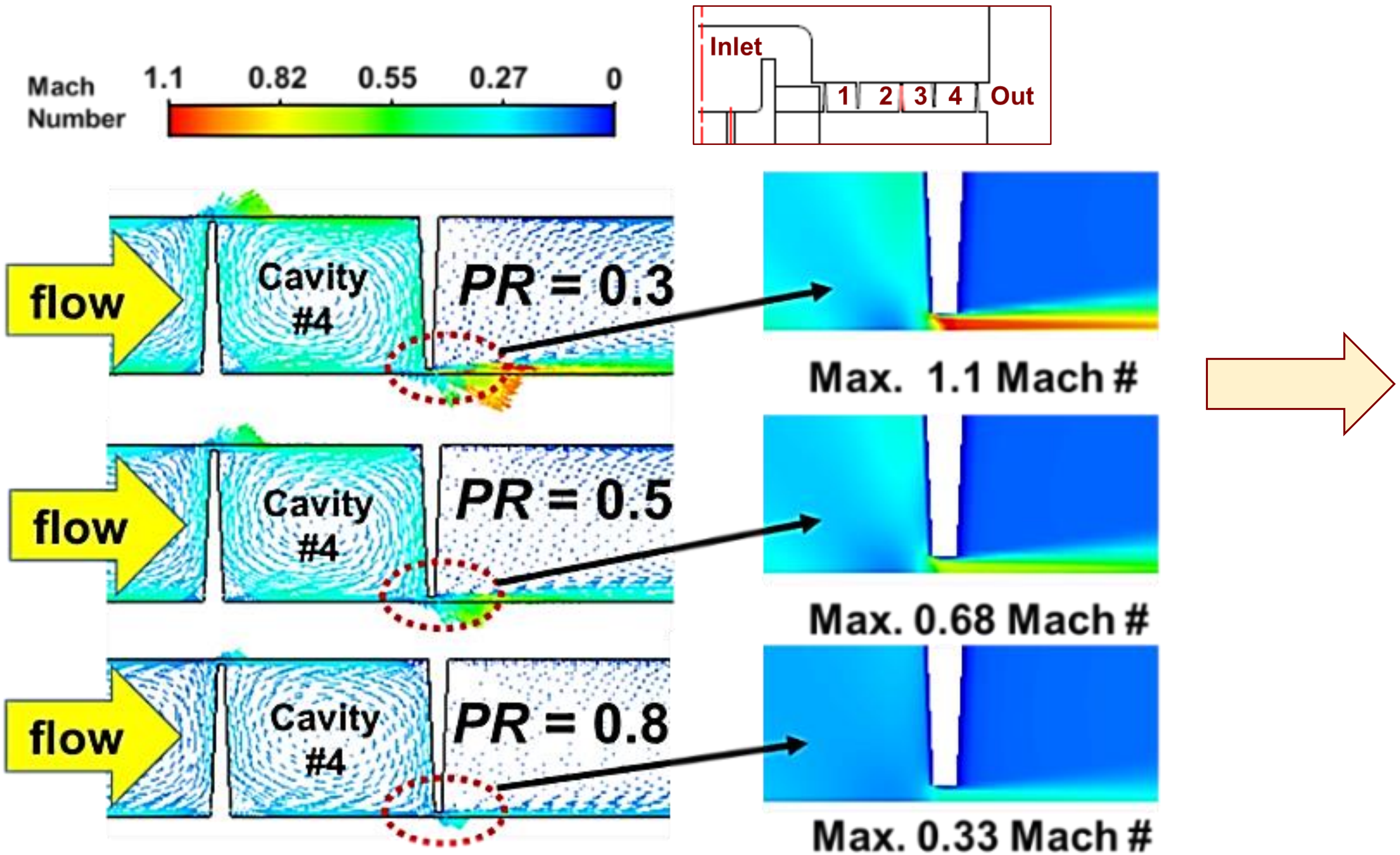
Three shaft speeds and three ***PR=(P$_{out}$ /P$_{in}$)***

Measured vs. CFD and BFM

Leakage ~ ΔP and not affected by shaft speed.
CFD & BFM agree + well with measurements.

Flow in last cavity – *PR* increases

Pin = 363 kPa, PR = 0.3, 0.5, 0.8, and 7.5 krpm (½DΩ = 59 m/s). Clearance = 0.2 mm

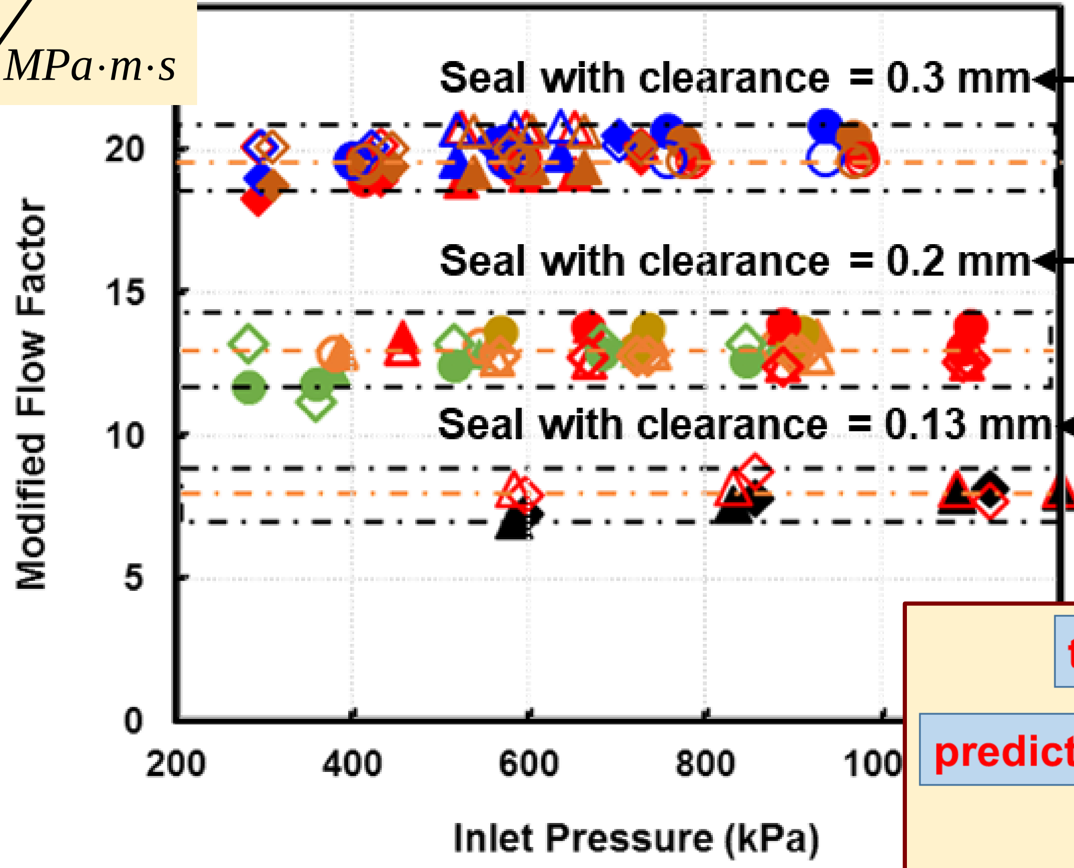


Reynolds number ~ 4,302 to 10,847 → turbulent flow.

For *PR*=0.3, flow chokes across last tooth (→ seal exit : cavity #4)

Modified flow factor for test ILS

$$\bar{\Phi} \quad kg\sqrt{K}\Big/_{MPa\cdot m\cdot s}$$



$$\bar{\Phi} = 19.6 \ kg\sqrt{K}\Big/_{MPa\cdot m\cdot s}$$

$$\bar{\Phi} = 13.3 \ kg\sqrt{K}\Big/_{MPa\cdot m\cdot s}$$

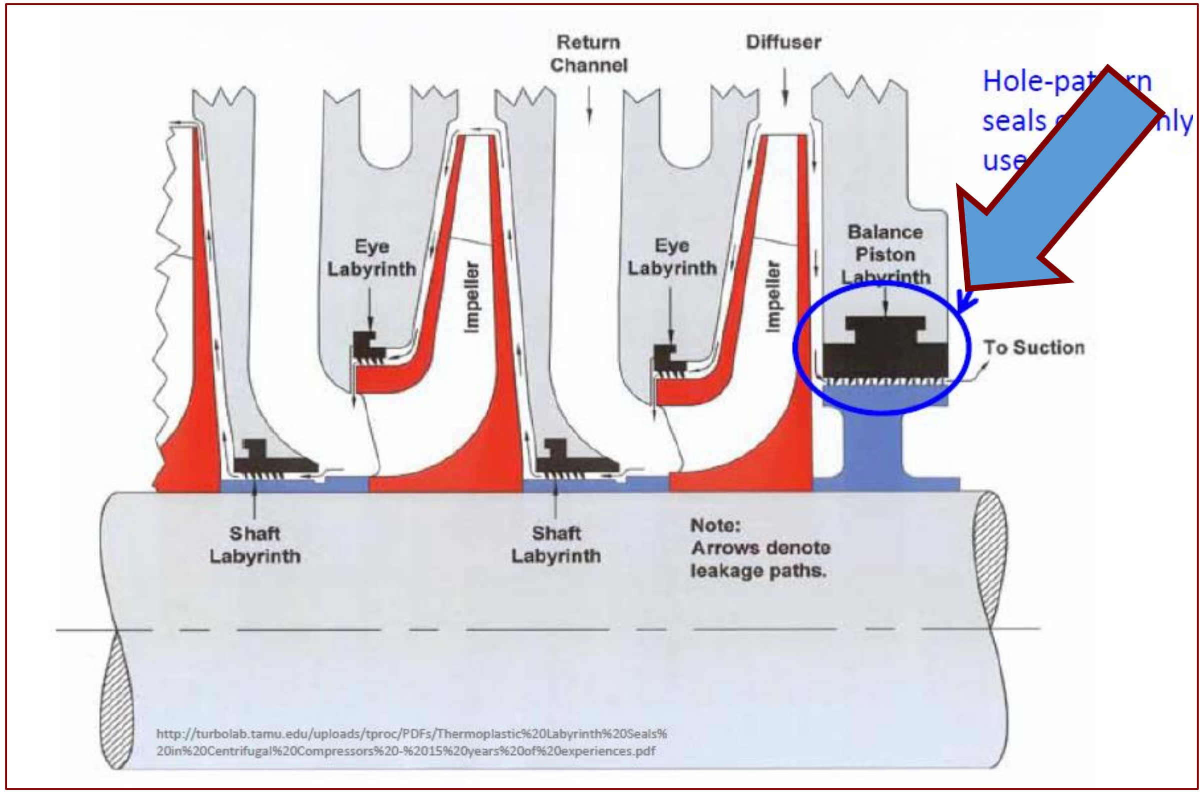
$$\bar{\Phi} = 7.8 \ kg\sqrt{K}\Big/_{MPa\cdot m\cdot s}$$

test	$c_d = 0.36 \pm 0.01$	$c_d = 0.36 \pm 0.01$	$c_d = 0.33 \pm 0.01$
prediction	$c_d = 0.37 \pm 0.01$	$c_d = 0.35 \pm 0.01$	$c_d = 0.34 \pm 0.01$
	clearance = 0.3 mm	clearance = 0.2 mm	clearance = 0.13 mm

c_d (a fraction of ILS clearance) is not a function of pre-swirl velocity, pressures P_{in} & P_{out}, rotor speed, or clearance!

Balance piston seals

Must withstand large pressure differences to balance axial thrust.



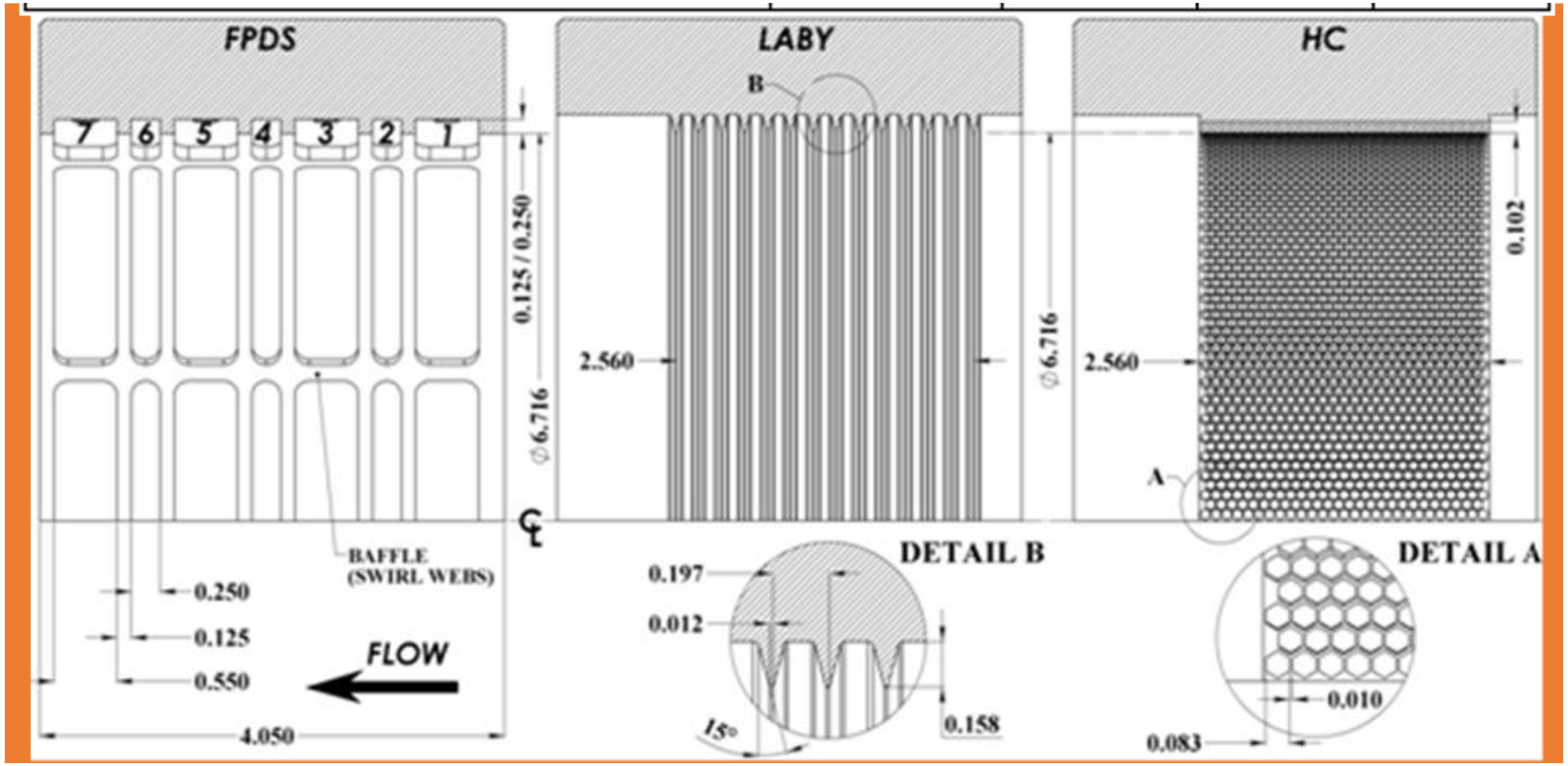
Picture from A Rimpel (2022).

Balance piston seals

Compare leakage:

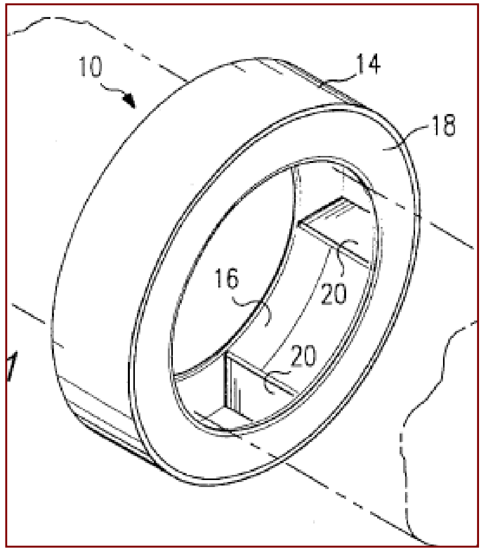
LS vs. honeycomb seal vs. pocket damper seal (PDS)

Radial clearance
C_r = 0.3 mm for all seals

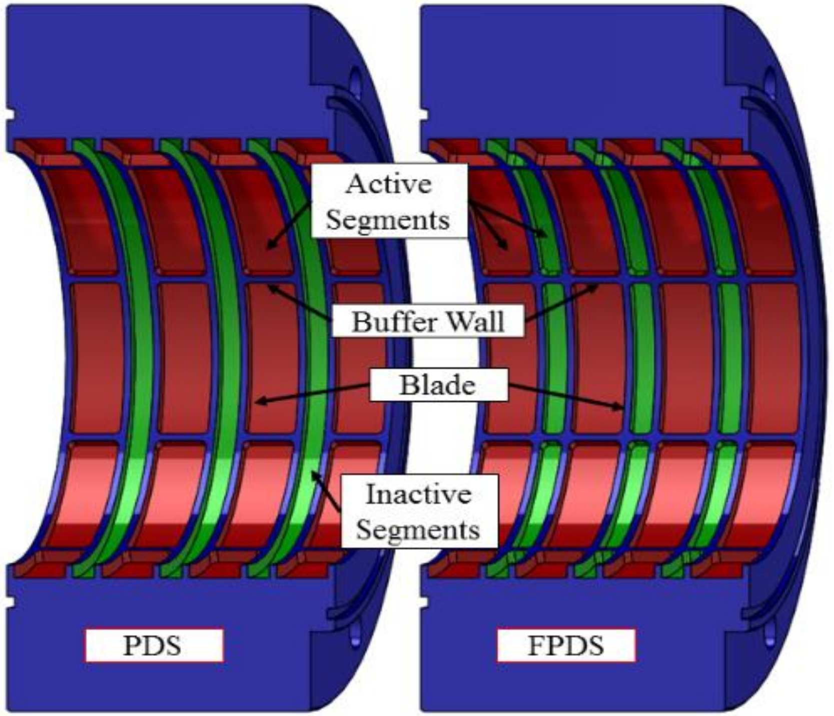


Ertas, B. H., Delgado, A., and Vannini, G., **2012,** "Rotordynamic Force Coefficients for Three Types of Annular Gas Seals with Inlet Pre-swirl and High Differential Pressure Ratio," ASME J. Eng. Gas Turbines Power, **134**(4).

Pocket Damper seal (PDS)



TAMSeal (1996)



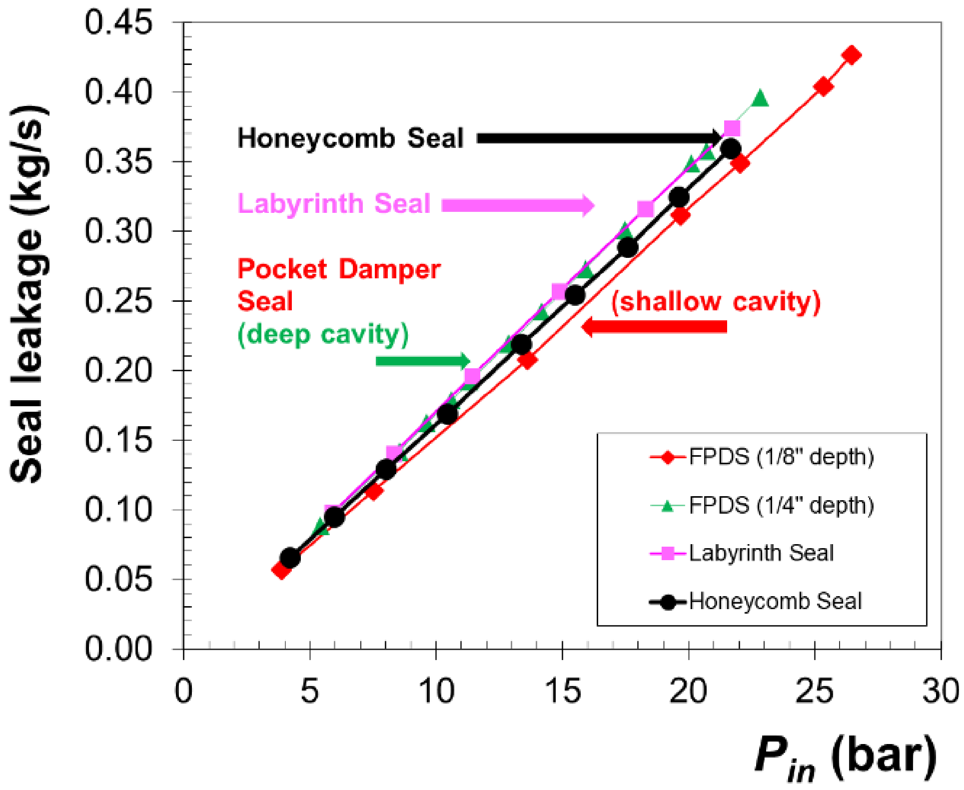
Left: conventional PDS;
Right: fully partitioned PDS.

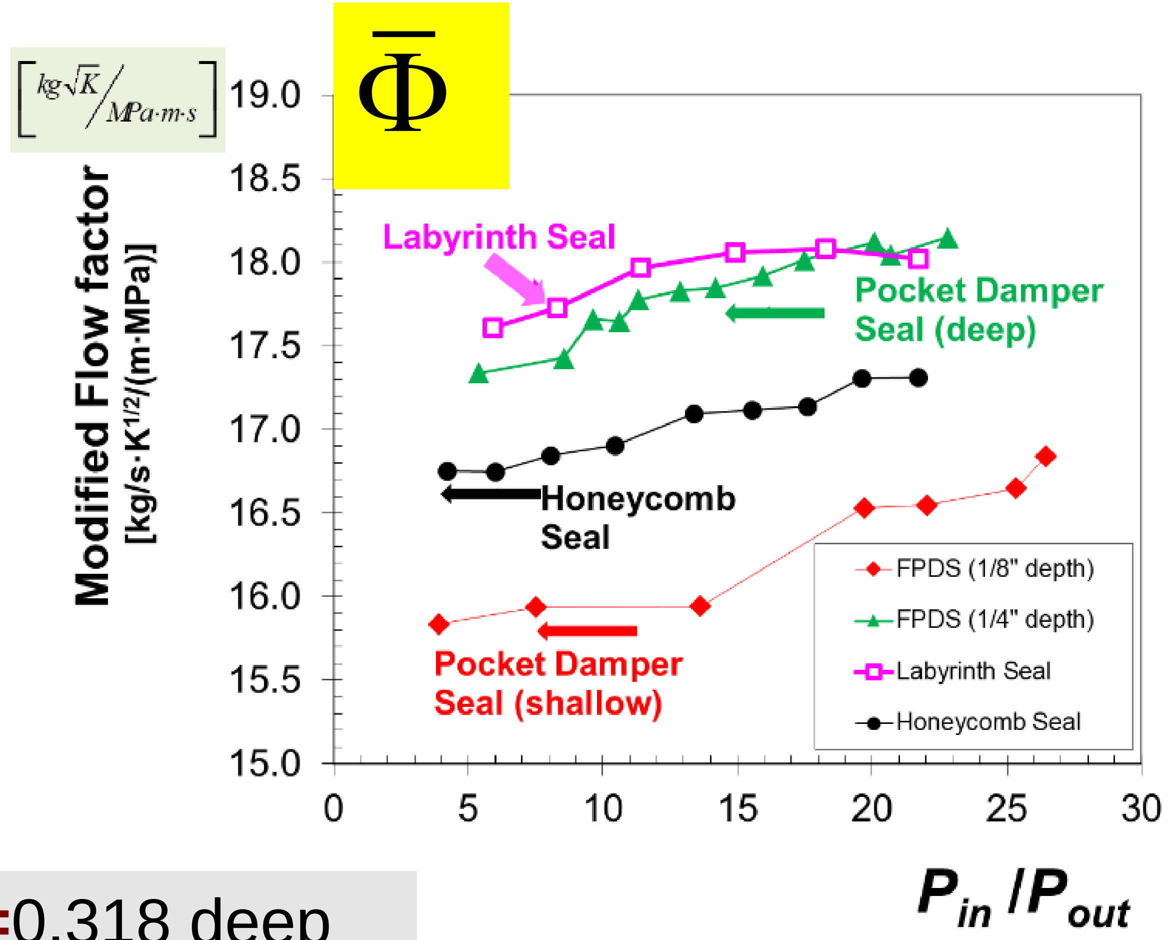
* Gamal, A., M., Ertas, B. H., and Vance, J. M., **2007,** "High Pressure Pocket Damper Seals: Leakage Rates and Cavity Pressures," ASME J. Turbomach., **129**(10).

* Ertas, B. H., Delgado, A., and Vannini, G., **2012,** "Rotordynamic Force Coefficients for Three Types of Annular Gas Seals with Inlet Pre-swirl and High Differential Pressure Ratio," ASME J. Eng. Gas Turbines Power, **134**(4).

Seals leakage and flow factor

P_{out} =1.01 bar. Inlet swirl velocity = 60 m/s, and rotor speed = 15 krpm (surface speed=133 m/s), ambient Temperature.

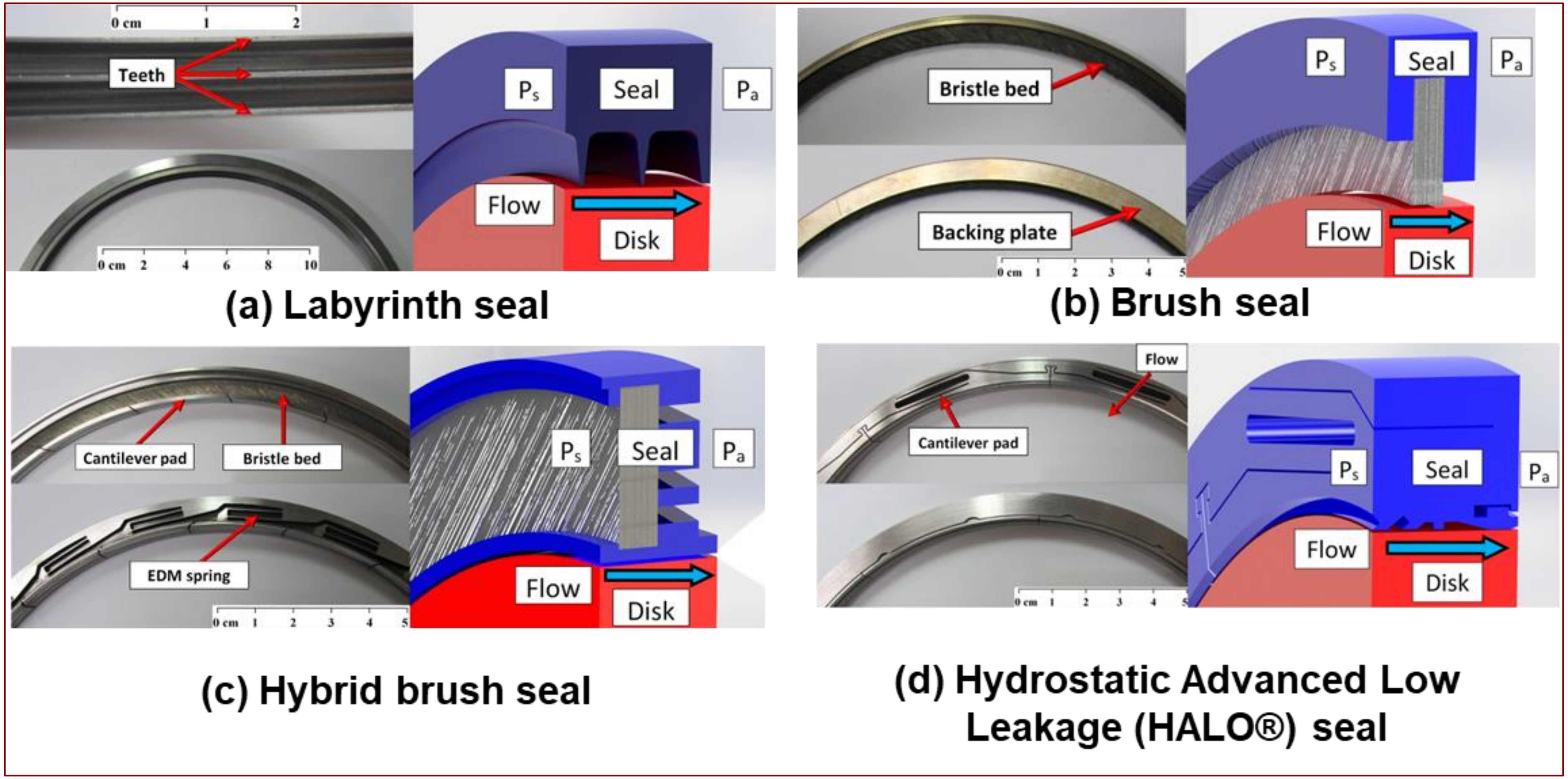




Loss coefficient c_d =0.32 for LS, ***c_d*** **=**0.318 deep pocket PDS, ***c_d*** =0.304 honeycomb seal, and ***c_d*** **=**0.291 for shallow pocket PDS.

Leakage for turbine rim seals

Compare leakage performance of four seals at high temperature.

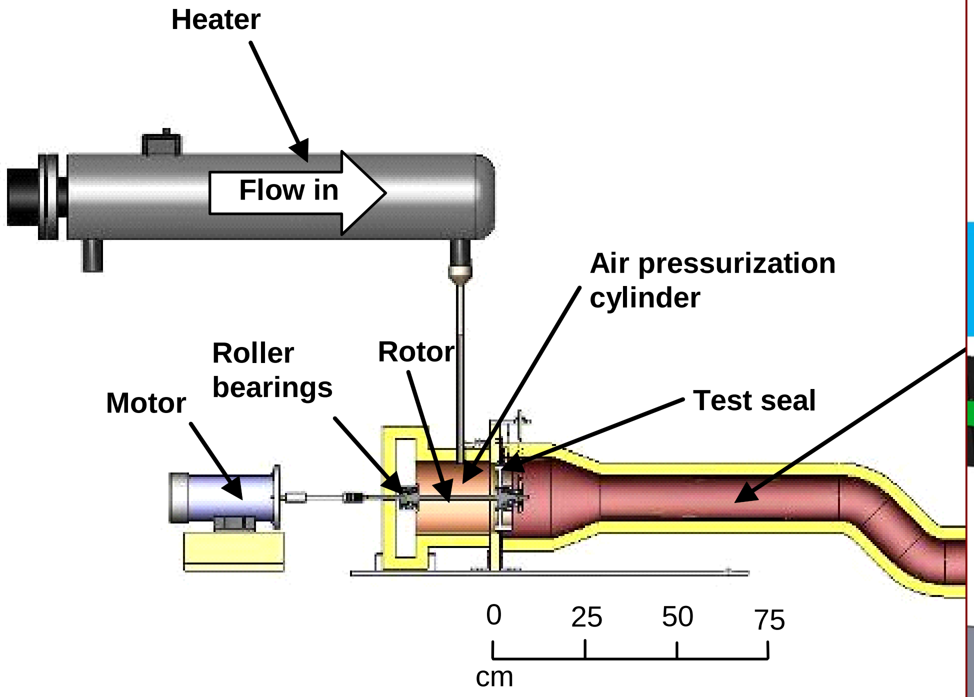


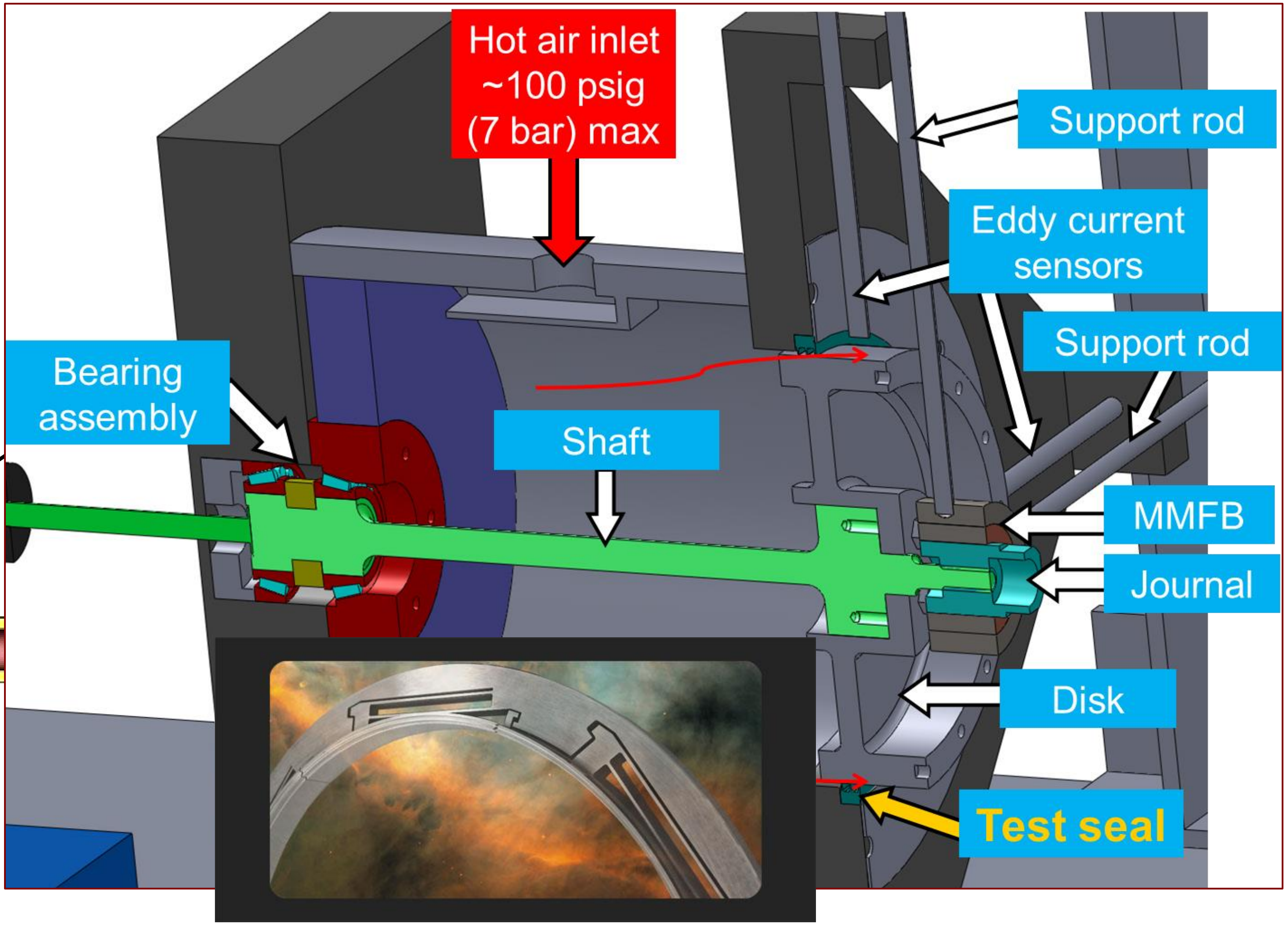
Metal brush seals are a known choice, while clearance control seals are novel.

L. San Andrés and A. Anderson, **2015**, "An All-Metal Compliant Seal Versus a Labyrinth Seal: A Comparison of Gas Leakage at High Temperatures," *ASME J. Eng. Gas Turbines Power,* vol. 137 (5)

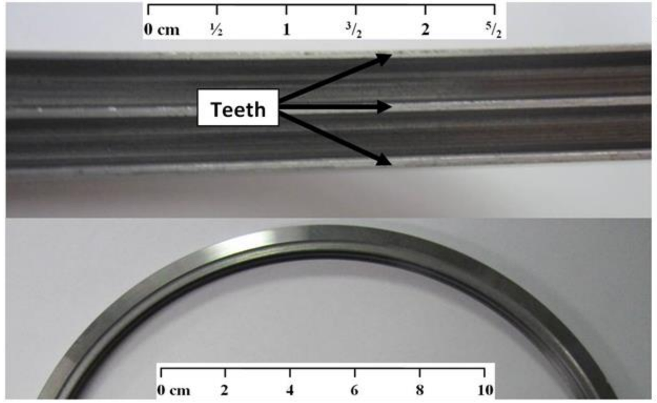
High temperature seal test rig

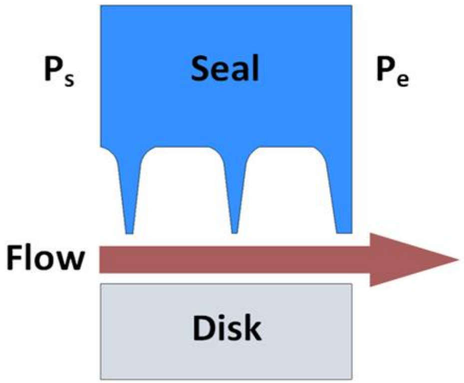
	Maximum		
	Voltage	Power	Output
Heater	240 V	12 kW	300°C
Motor	90 V	850 W	3,000 rpm



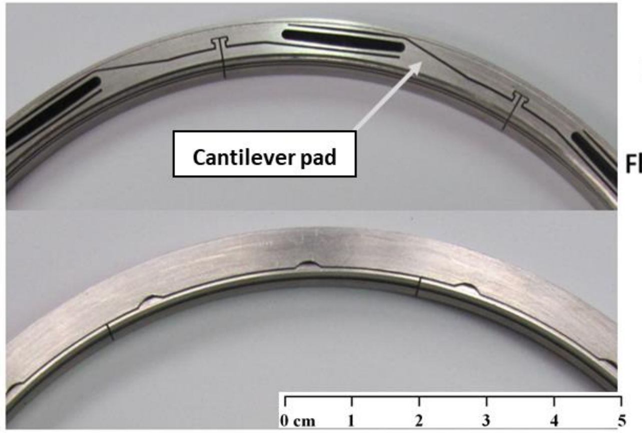


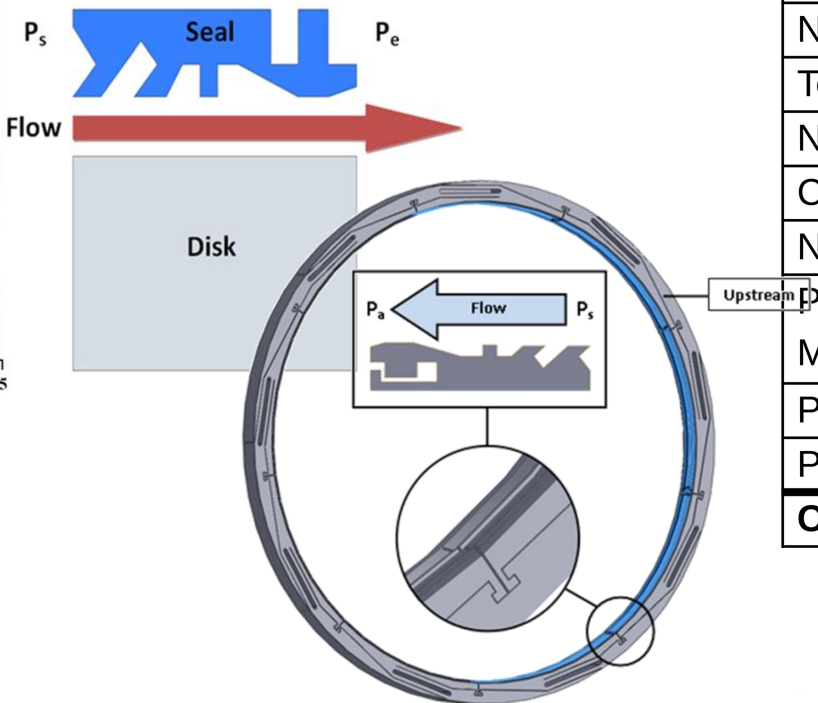
Test seals dimensions and materials





LS – three teeth





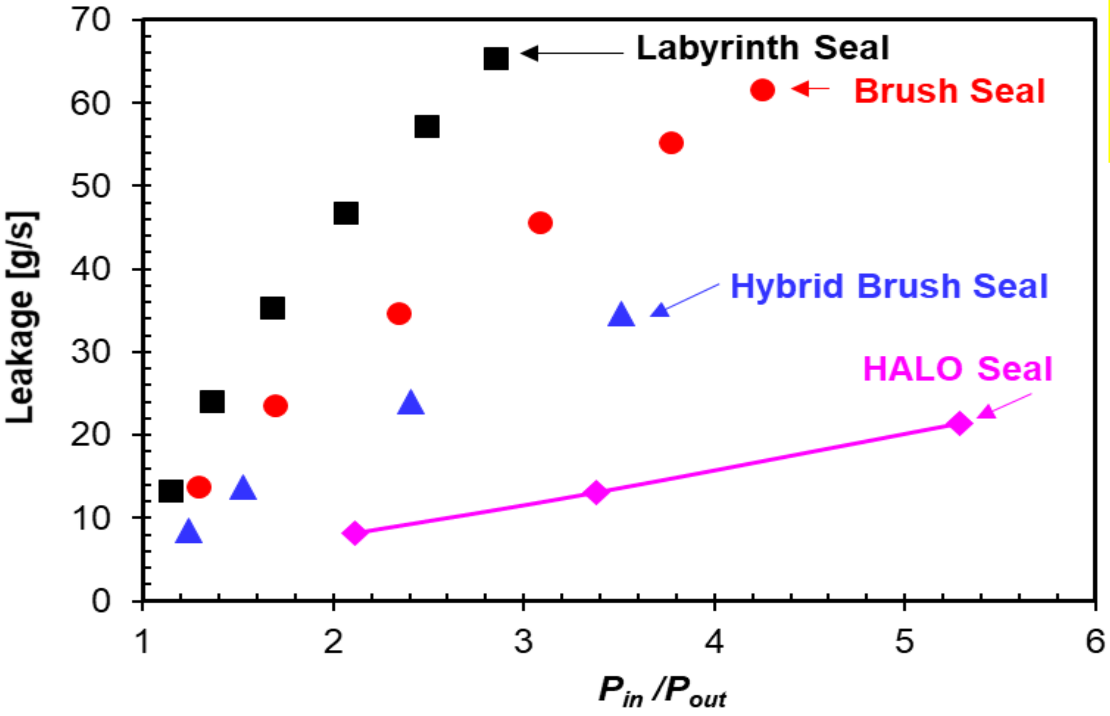
HALO® Seal

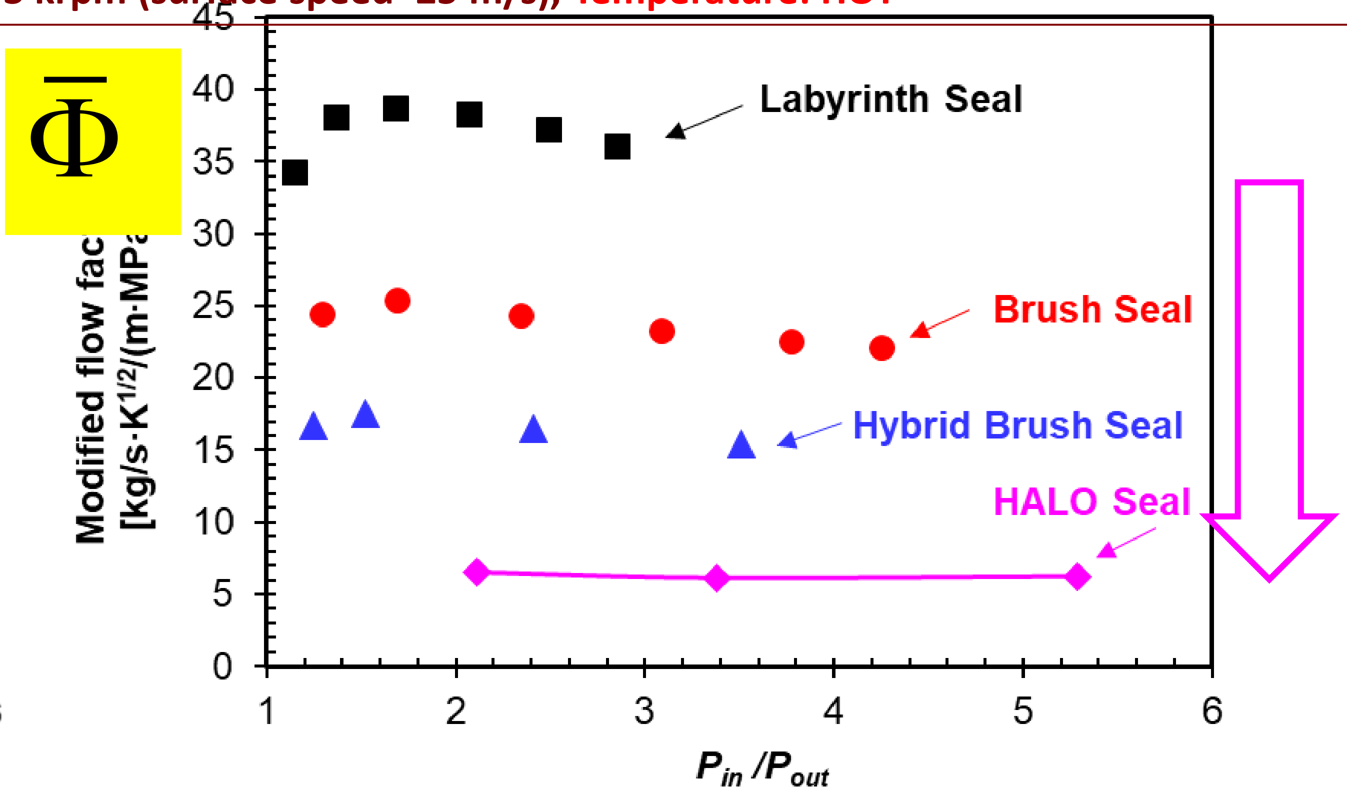
Disk Material	**4140 Steel**
Thermal Expansion coef., α	11.2 10^{-6}/°C
Outer Diameter, D	166.81 mm
Disk Thickness	44.45 mm

	Labyrinth Seal	**HALO^{TM} Seal**
Material	Steel	Inconel 718
Thermal Expansion coeff, α	11.2 10^{-6}/°C	12.0 10^{-6}/°C
Inner Diameter, S_{ID} (Downstream)	**167.36 mm**	**167.2 mm**
Seal Axial Length, l	8.40 mm	8.5 mm
Number of Teeth	3	
Teeth Tip Width	0.17 mm	
Number of Cavities	2	
Cavity Depth	3.0 mm	
Number of Pads		9
Pad Allowable Radial Movement		0.27 mm
Pad Axial Length, l		8.0 mm
Pad Arc Length (40°)		58.4 mm
Clearance (C_d=S_{ID}-D)	**0.51 mm**	**0.43 mm**

Seals leakage and flow factor at 300 ºC

P_{out} =1.01 bar. Inlet swirl velocity = 00 m/s, and rotor speed = 3 krpm (surface speed=25 m/s), Temperature: HOT





Loss coefficient c_d =0.78 for LS, c_d =0.49 for brush seal, c_d =0.34 for hybrid brush seal, and c_d =0.16 for HALO® seal

Other advanced concept seals include the Pressure Actuated Leaf Seal (PALS) and finger seals.

Knowledge ANNULAR CLEARANCE GAS SEALS

1. Overview
2. Bulk-flow and CFD models
3. **Seals’ leakage & their effective clearance**
4. Seals force coefficients
5. Closure – The road Ahead

ASIA TURBOMACHINERY & PUMP SYMPOSIUM

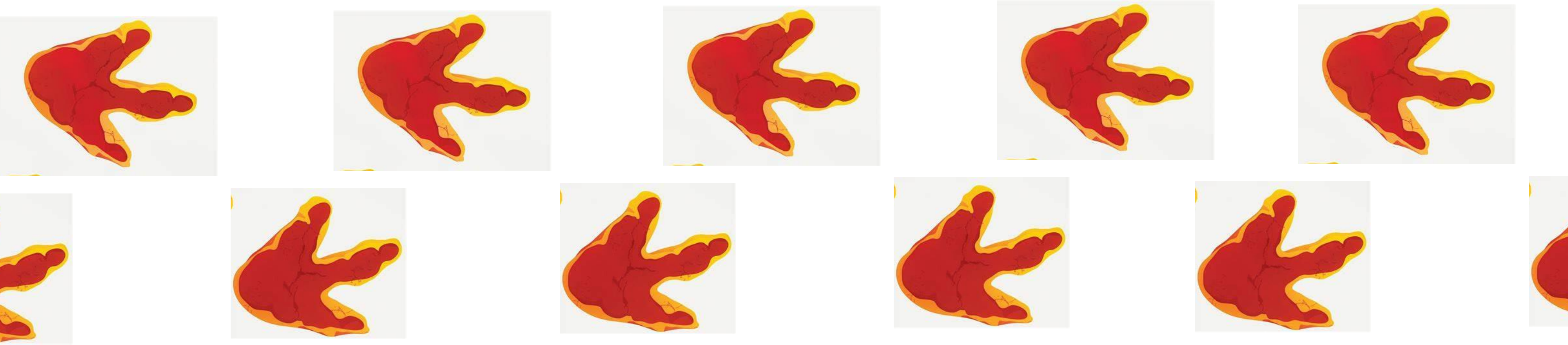
Knowledge on seals' leakage

Comparisons of measured and predicted seals' leakage demonstrate that (well designed and engineered) annular seals have effective clearance = fraction of seal operating clearance.

A typical range is $C_e/C_r = c_d$ ~ 0.30 - 0.40.

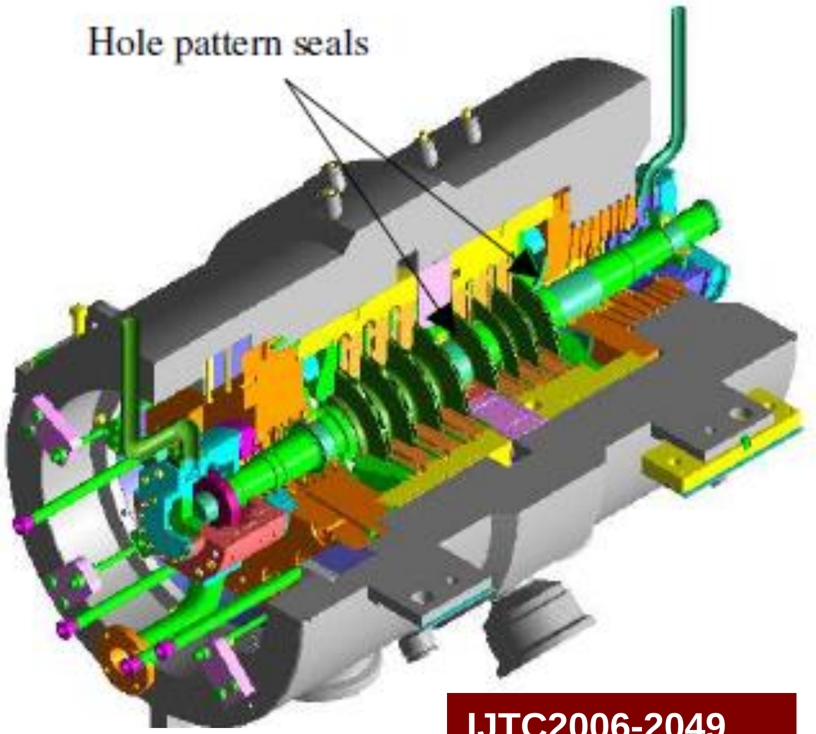
Most importantly, from the test data reviewed, c_d is not a function of either the inlet pressure, or the outlet pressure, or the shaft speed, or the inlet swirl, or the actual clearance!

Bulk-flow models and CFD accurately predict the leakage of annular clearance seals. CFD ~ BFM



Effect of seals on rotordynamic stability

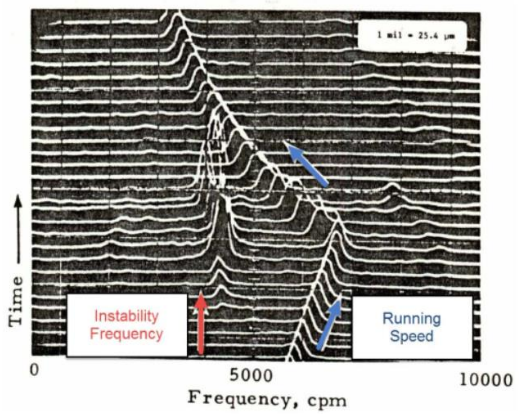
→ A review of their force coefficients

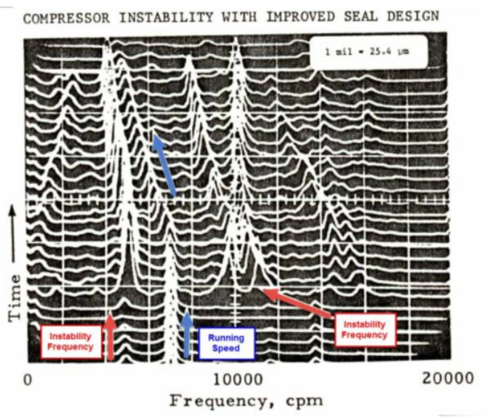


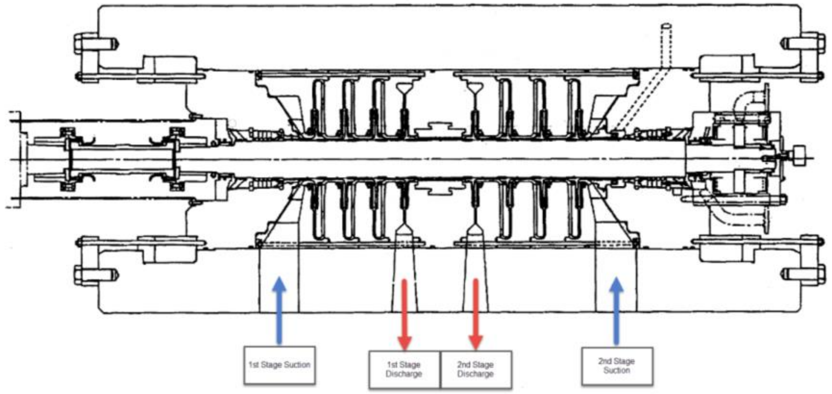
IJTC2006-2049

Seals are *bad actors* driving CCs/STs unstable

Forces in gas seals are roughly proportional to the pressure differential (DP) across the seals and the fluid density within the seal.







1997 Turbomachinery Symposium

ANNULAR GAS SEALS AND ROTORDYNAMICS OF COMPRESSORS AND TURBINES

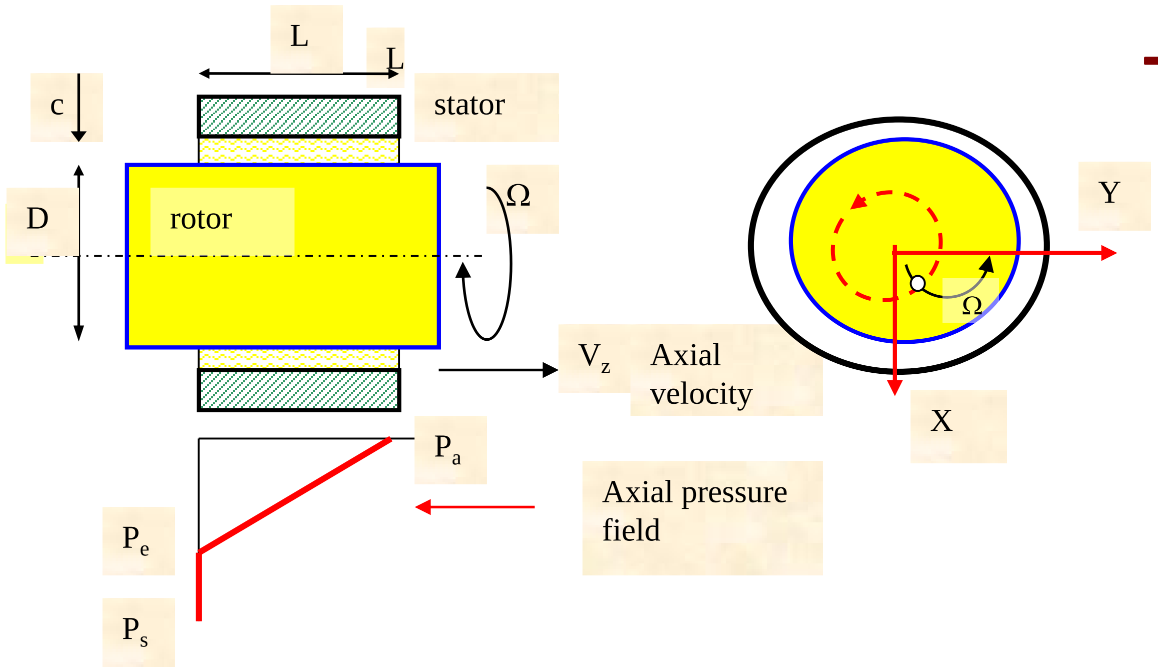
by

Dara W. Childs
Leland T. Jordan Professor
and
John M. Vance
Professor
Department of Mechanical Engineering
Texas A&M University
College Station, Texas

9,200 psi discharge pressure → 2000 psi higher than any compressor at the time

Cloud, H., Kokur, J., Pettinato, B., 2018, "PREDICTING, UNDERSTANDING AND AVOIDING THE EKOFISK ROTOR INSTABILITY FORTY YEARS LATER," Proc. Turbomachinery Pump Symposium.

Annular seals generate reaction forces



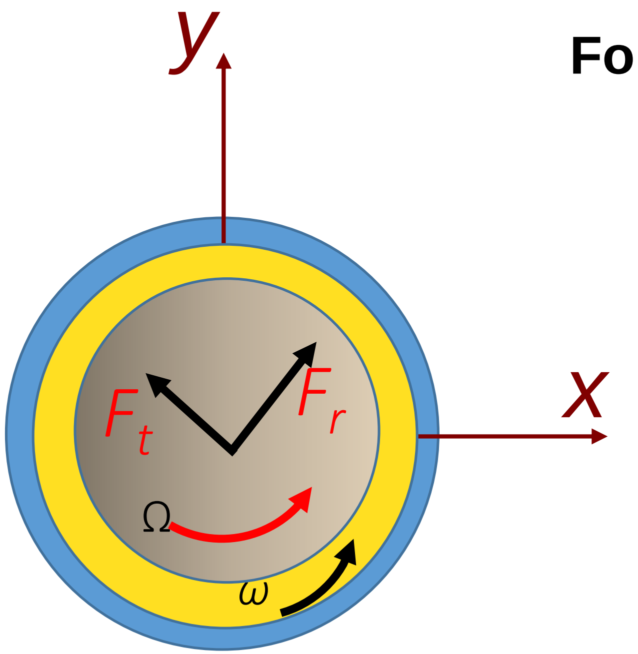
→ **function of the fluid properties, geometry, flow regime, and operating conditions:**

*** large axial pressure gradient,**
*** large diameter to radius ratio *(R/c)* < 500**
*** axial development of gas circumferential speed determines magnitude of cross-coupled (hydrodynamic) forces.**

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

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

Seal effective force coefficients



For circular whirl motions with amplitude *r* and frequency ω

$$-\begin{bmatrix} F_X \\ F_Y \end{bmatrix} = \left(K_{(\omega)} + \omega c_{(\omega)}\right)\begin{bmatrix} X \\ Y \end{bmatrix} + \left(C_{(\omega)} - {k}/{\omega}\right)\begin{bmatrix} \dot{X} \\ \dot{Y} \end{bmatrix}$$

Radial and tangential forces are

$$F_r = -K_{eff\,(\omega)}\, r \; ; \; F_t = -C_{eff\,(\omega)}\,(r\omega)$$

where the effective stiffness and damping coefficients $(K, C)_{eff}$ are:

$$K_{eff\,(\omega)} = \left(K_{(\omega)} + \omega c_{(\omega)}\right), \quad C_{eff\,(\omega)} = \left(C_{(\omega)} - {k_{(\omega)}}/{\omega}\right)$$

Gas seals show coefficients that are frequency dependent.

The coefficients are also a function of shaft speed and pressure drop across seal.

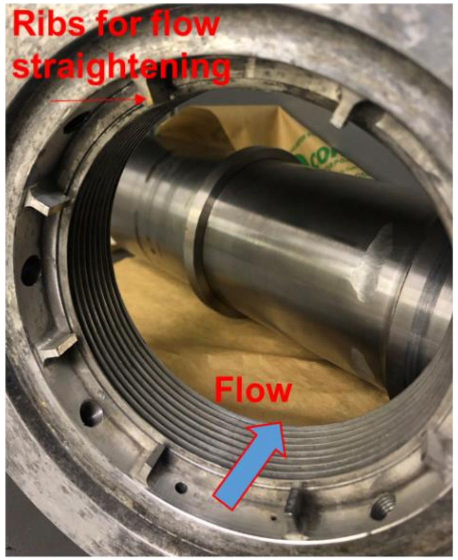
$C_{eff} > 0$ is highly desirable for rotor dynamic stability.

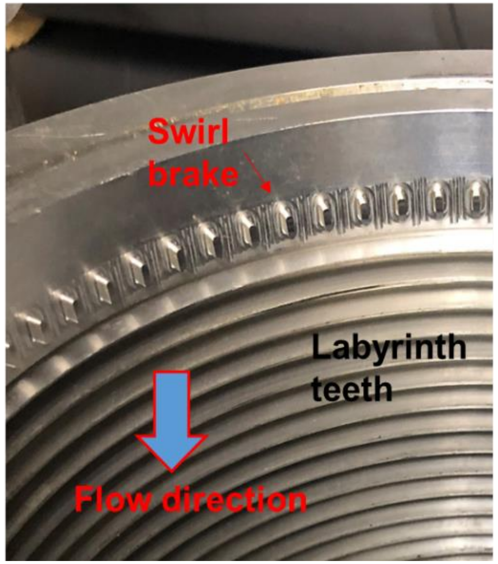
$K_{eff} > 0$ allows load carrying and rotor static stability.

Remedies to promote seal stability

Aim: to reduce cross-coupled stiffness (*k* →0) without affecting sealing ability.

Swirl brakes



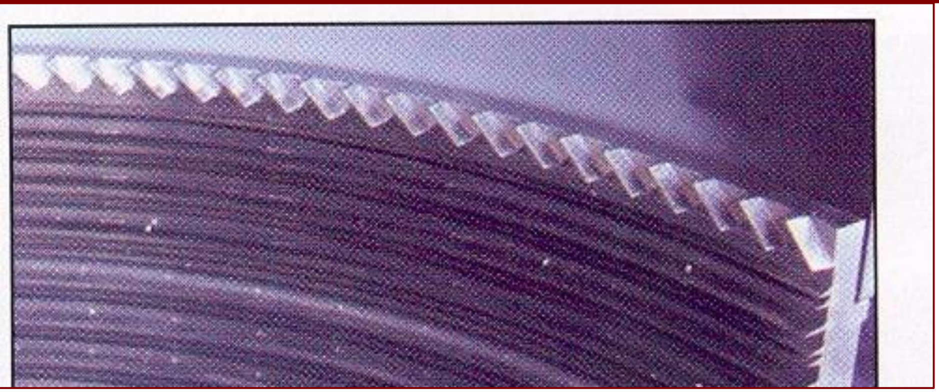


A series of vanes upstream a seal inlet plane to redirect the flow. Engineered since 1980

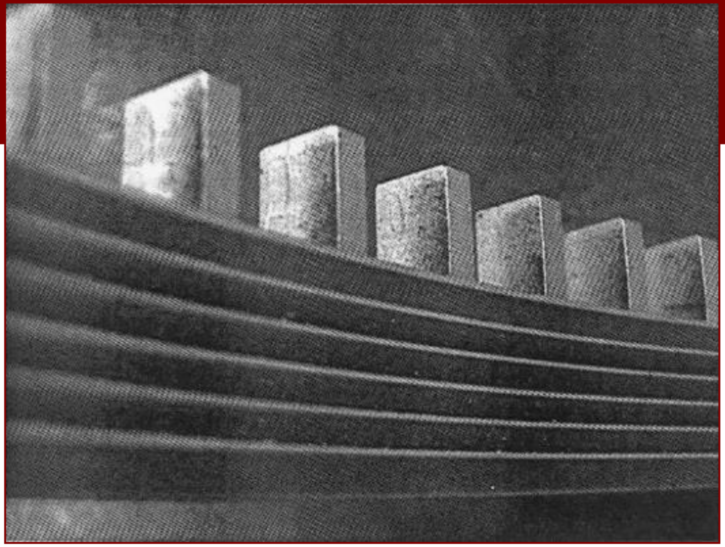
Benckert, H., and Wachter, J., **1980,** "Flow Induced Spring Coefficients of Labyrinth Seal for Applications in Rotordynamics," NASA CP-2133

→reduce inlet circumferential flow entering seal, hence also reducing the seal cross-coupled stiffness *(k)* to promote rotordynamic stability.

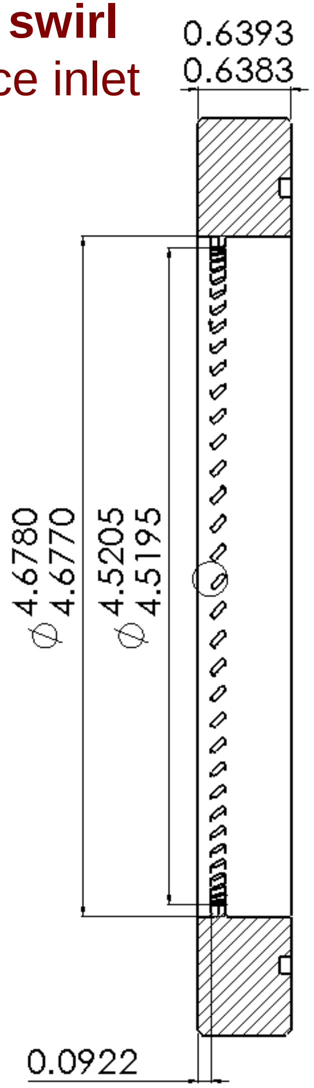
Swirl Brakes



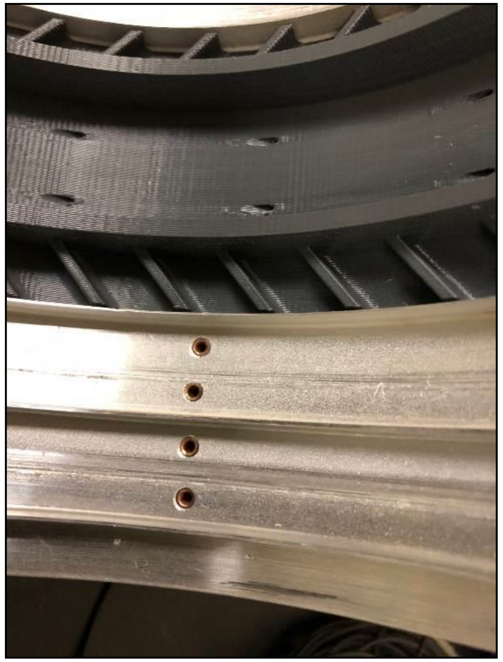
Turbomachinery International, September/October 2007

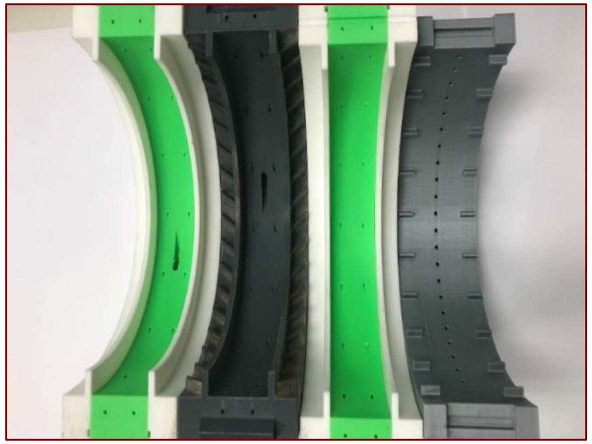


Conventional swirl brake → reduce inlet preswirl

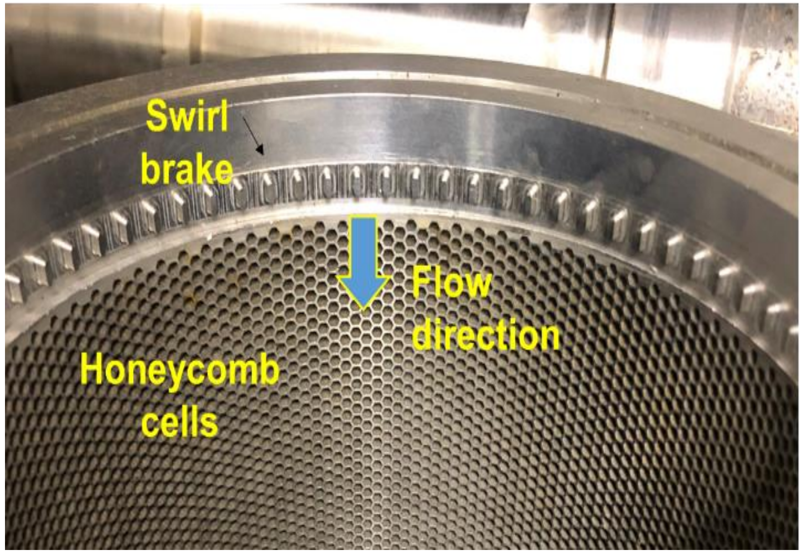


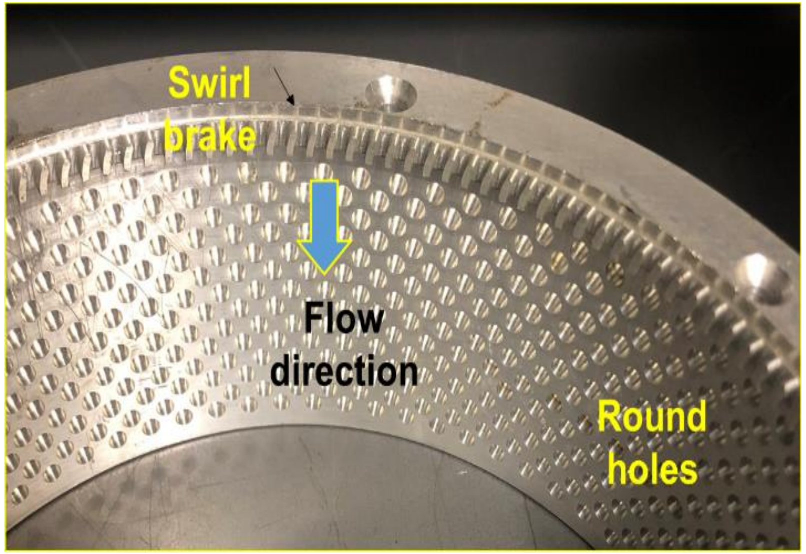
Swirl brakes for labyrinth seals (ADM)

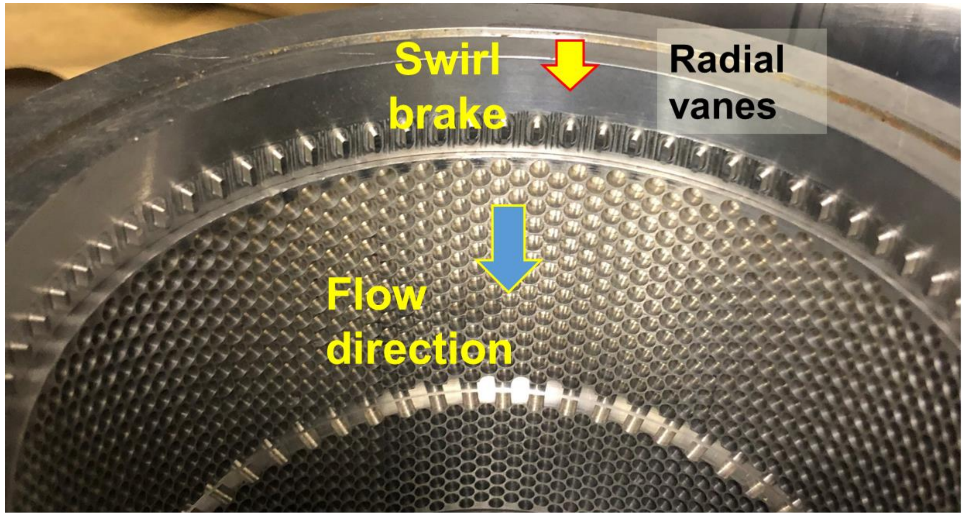


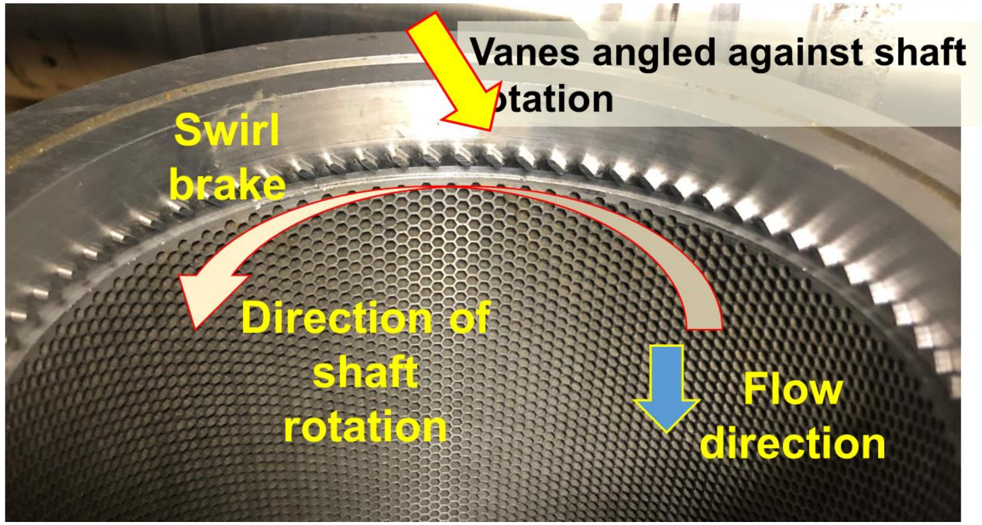


Honeycomb seal and hole-pattern seal with swirl brakes upstream of seal entrance









Shunt Injection also improves stability

WHY INJECT AGAINST ROTATION?

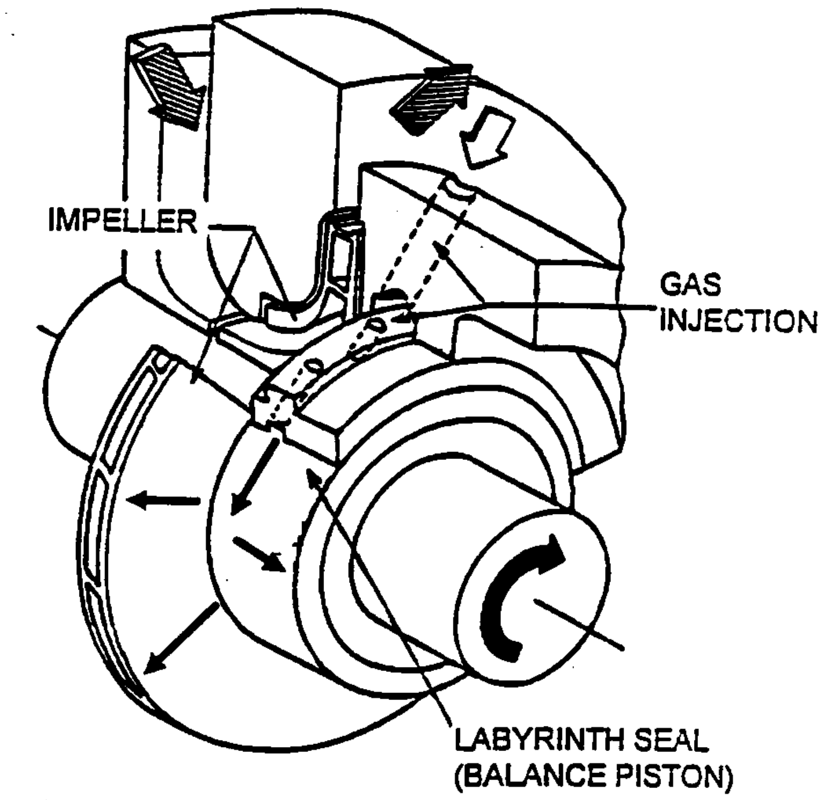
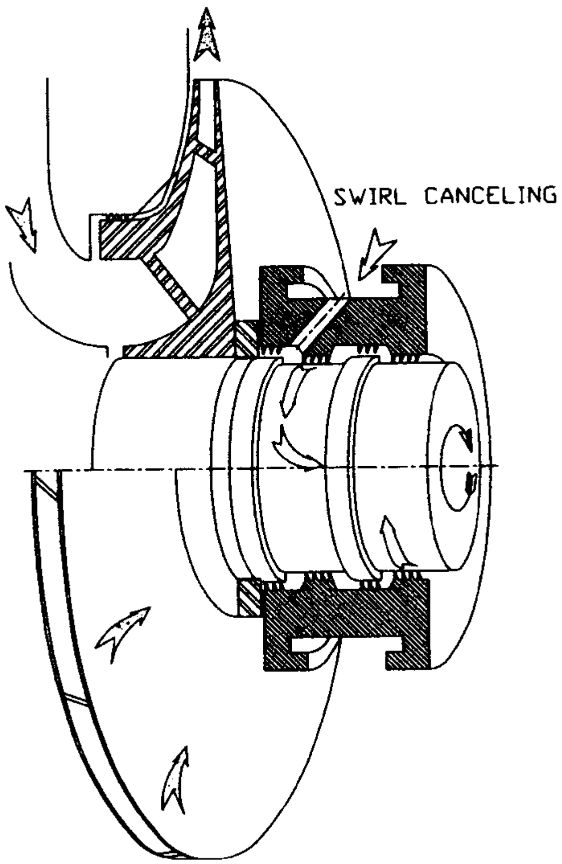


Figure 1. Labyrinth seal with shunt injection, adapted from Kanki et al. (1988)



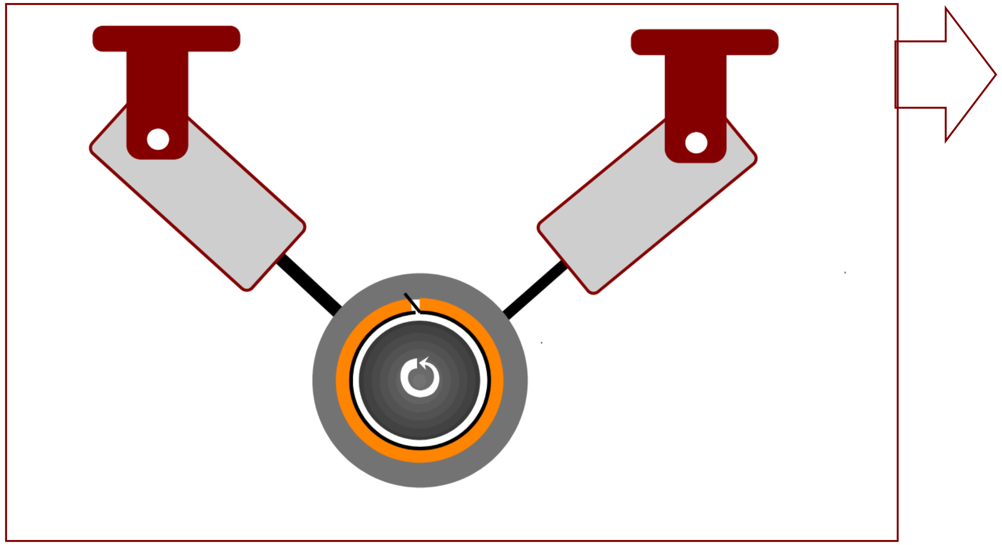
ure 6. Swirl Canceling Device.

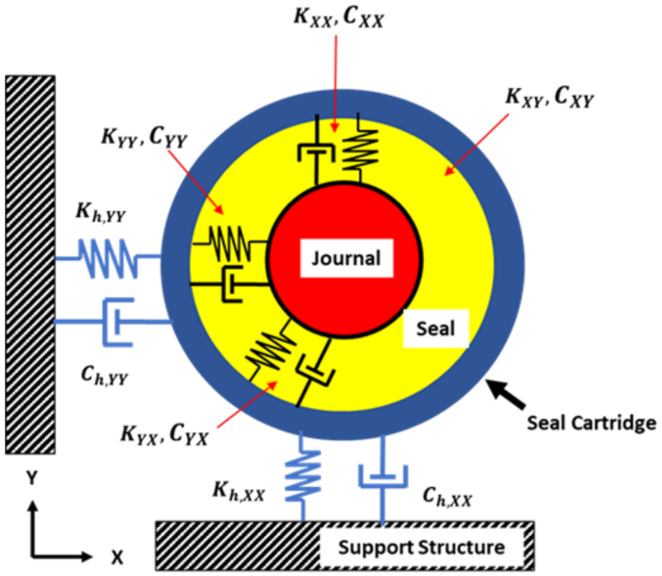
can change the sign of the cross-coupled stiffness *k*, which increases effective damping

$$C_{eff} = C_{(\omega)} - \frac{k_{(\omega)}}{\Omega}$$

Implementation is more difficult!

Identification of force coefficients





2-DOF system for seal and support structure.

Steps to identify seal force coefficients

1) Apply Load **$F=F_o \sin(\omega t)$** →

Measure vectors of displacement $\mathbf{z} = \{x, y\}^T$, & acceleration $\mathbf{a} = \{a_x, a_y\}^T$

2) $\bar{\mathbf{F}}$, $\bar{\mathbf{A}}$, $\bar{\mathbf{Z}}$ ← Discrete Fourier Transforms of **F, a, z**

3) Equation of motion $\bar{\mathbf{F}} - \mathbf{M}_h \bar{\mathbf{A}} - [\mathbf{K}_h + i\omega \mathbf{C}_h]\bar{\mathbf{Z}}' = \mathbf{H}_{(\omega)} \bar{\mathbf{Z}}$

$[M, K, C]_h$ = mass, stiffness, damping of support structure

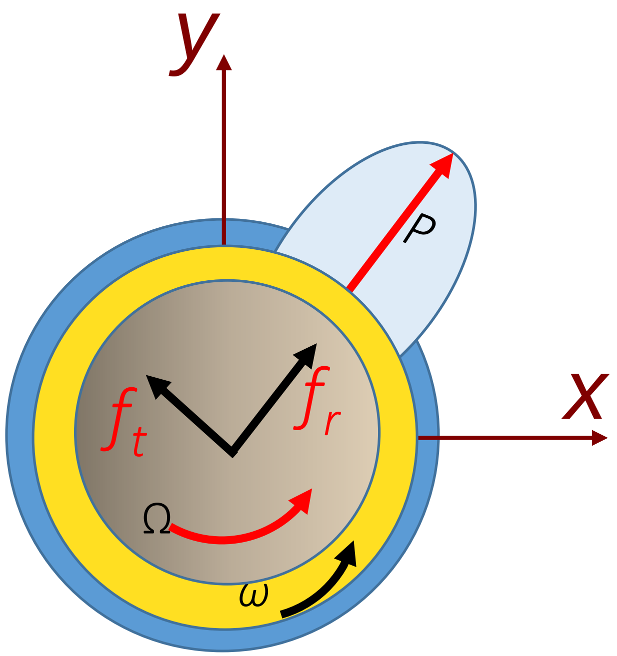
$\mathrm{Re}(\mathbf{H}_{(\omega)}) \rightarrow \boldsymbol{K}_{(\omega)}$ **Dynamic Stiffness**

$\mathrm{Im}(\mathbf{H}_{(\omega)}) \rightarrow \omega \boldsymbol{C}_{(\omega)}$ **Proportional to Damping**

$C_{eff} = C - k/\omega = [\mathrm{Im}(H_{xx}) - \mathrm{Re}(H_{xy})]/\omega$

Effective Damping

Bulk flow and CFD models for prediction of force coefficients



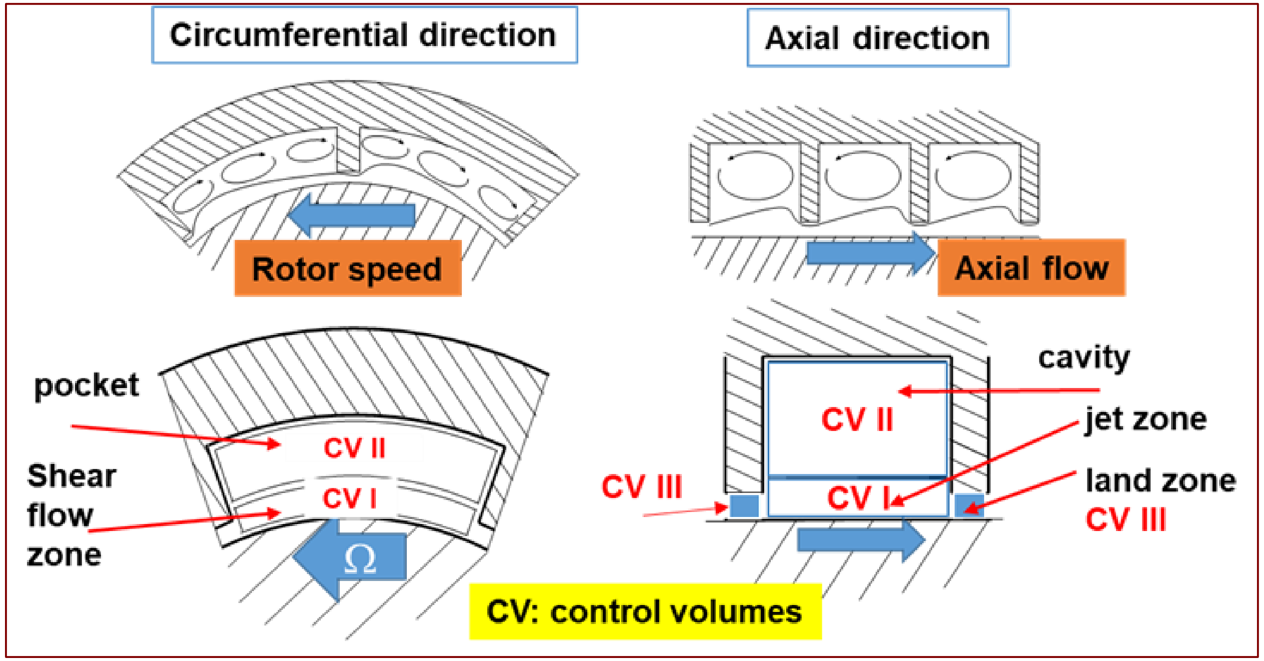
Rotor speed Ω

Whirl frequency ω

Calculate flow field to obtain forces along radial and tangential direction: f_r , f_t

BFM Bulk-flow model: typical governing equations

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



Flow Continuity

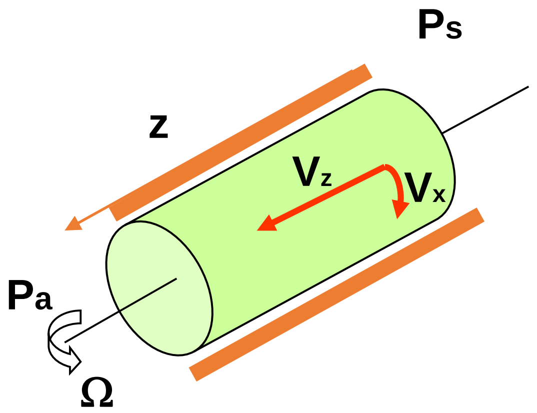
$$\frac{\partial}{\partial x}(hV_x)+\frac{\partial}{\partial z}(hV_z)+\frac{\partial h}{\partial t}=0$$

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

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



BFM procedure for predicting seal performance

2D bulk flow solver

Film thickness

$$h = h_{0(\theta,z)} + \delta e_X \, e^{i(\theta+\omega t)}$$

Small amplitude (δe) change displacement of rotor center

Fields: velocities and pressure

$$(P, V_x, V_z) = (P_0, V_{0x}, V_{0z}) + \delta e_X (P_\delta, V_{\delta x}, V_{\delta z}) e^{i\omega t}$$

Perturbed flow field solver

Available since 1980's

Frequency domain:

$$-\oint P_\delta \begin{Bmatrix} \cos\theta \\ \sin\theta \end{Bmatrix} R\, d\theta\, dz \Rightarrow \begin{Bmatrix} H \\ h \end{Bmatrix}_{(\omega)} = \begin{Bmatrix} K + i\omega C \\ k + i\omega c \end{Bmatrix}_{(\omega)}$$

steady state performance:

- ✓ Seal leakage (m);
- ✓ Static pressure (P);
- ✓ Axial and Swirl speeds;

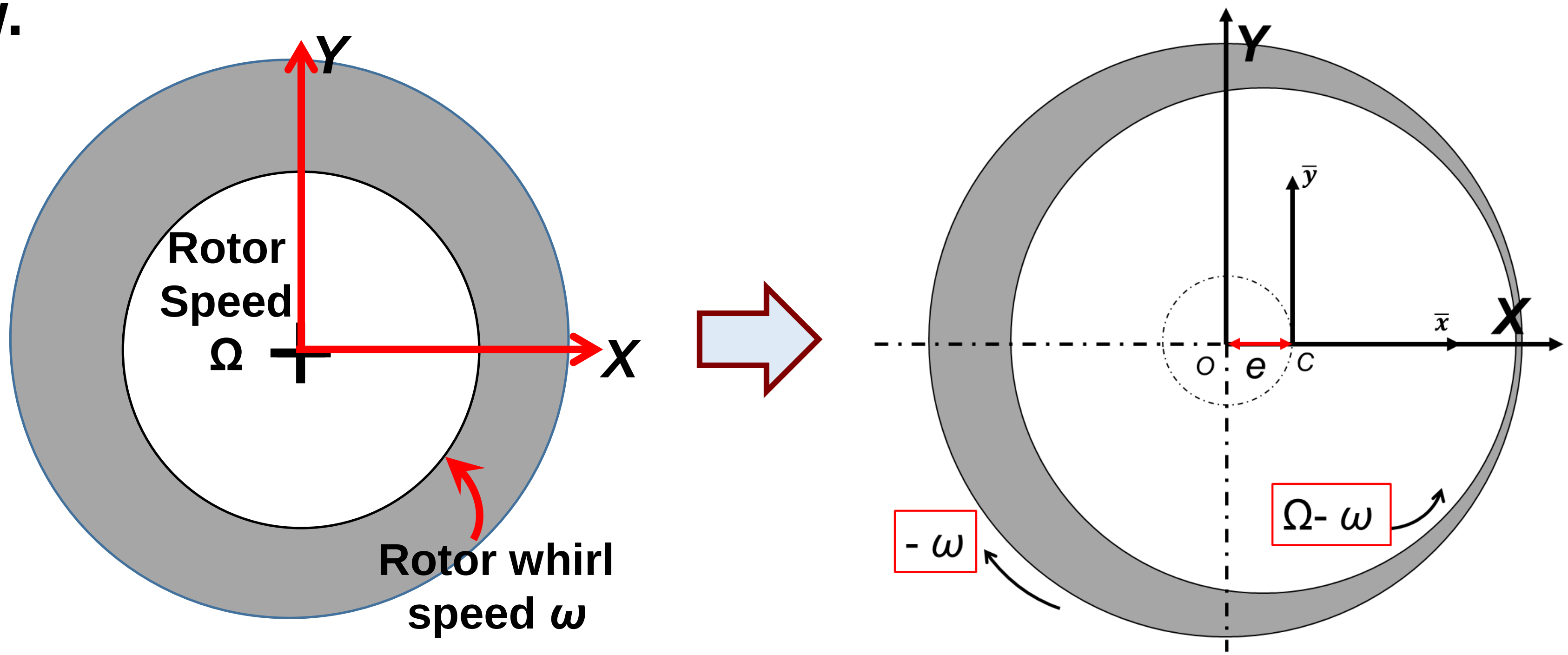
Seal complex dynamic stiffness:

- ✓ **Direct:** $H = K + i(\omega C)$
- ✓ **Cross-coupled:** $h = k + i(\omega c)$
- ✓ **Direct damping** C

Quantify how seal works!

CFD Coordinate transformation

Use a rotating coordinate frame to model a periodic (whirling) flow into a steady state flow.

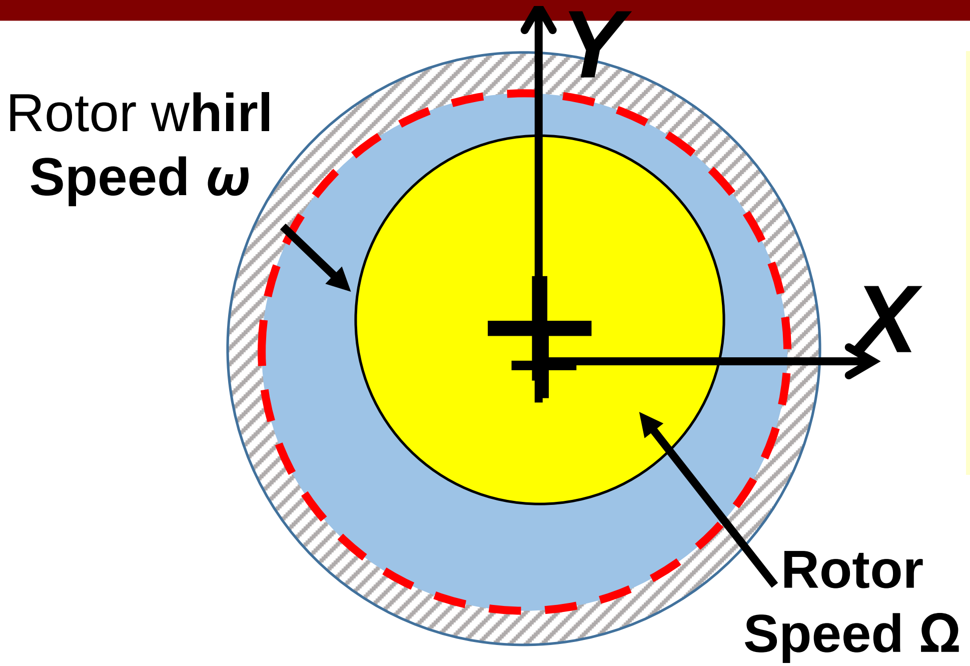


Periodic flow condition

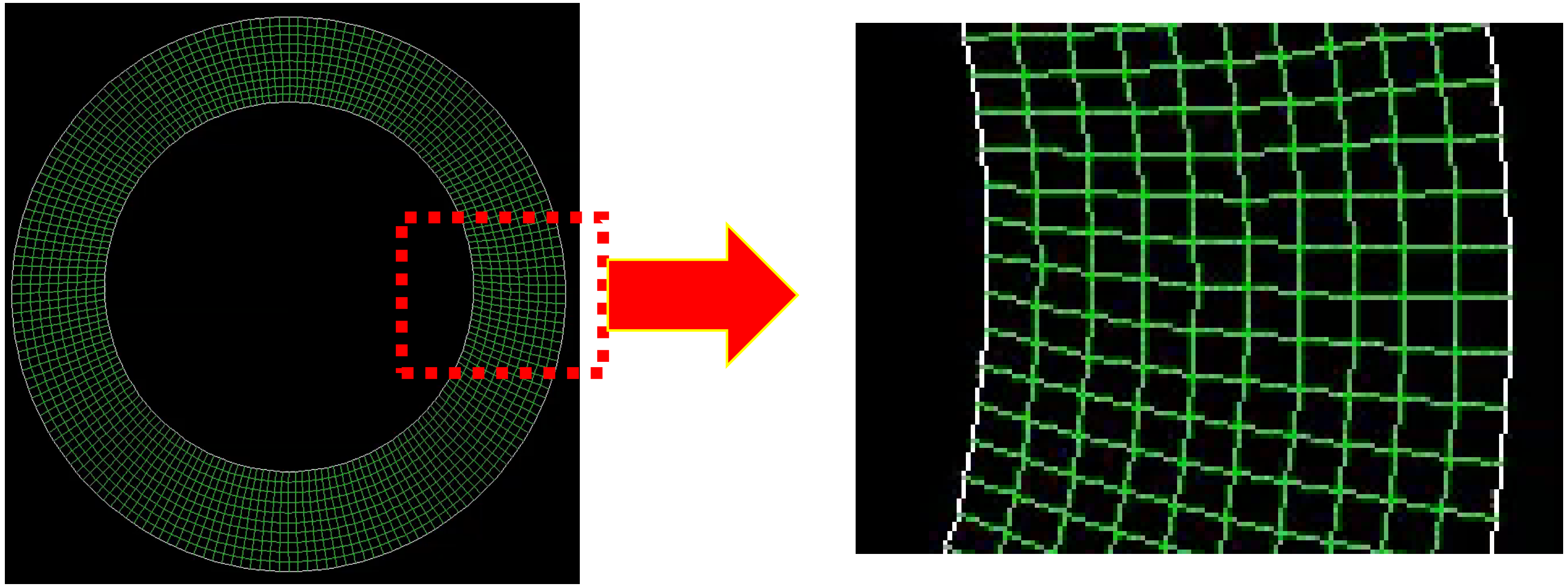
→ Steady flow state for observer in rotating frame

Gas seals produce frequency dependent force coefficients → analysis requires of multiple flow solutions at +/- whirl frequencies to extract seal force coefficients.

CFD Mesh deformation method

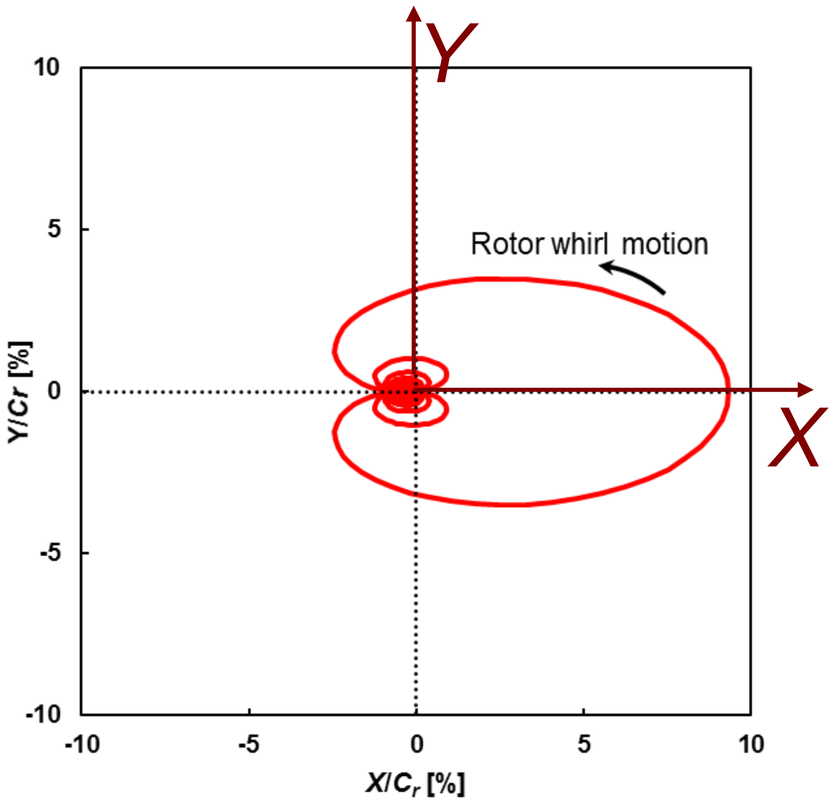


- **Rotor displaces with specified *X, Y* →**
- **Find transient response over a full period *(T)* delivers seal forces, f_r and f_t, on rotor surface.**

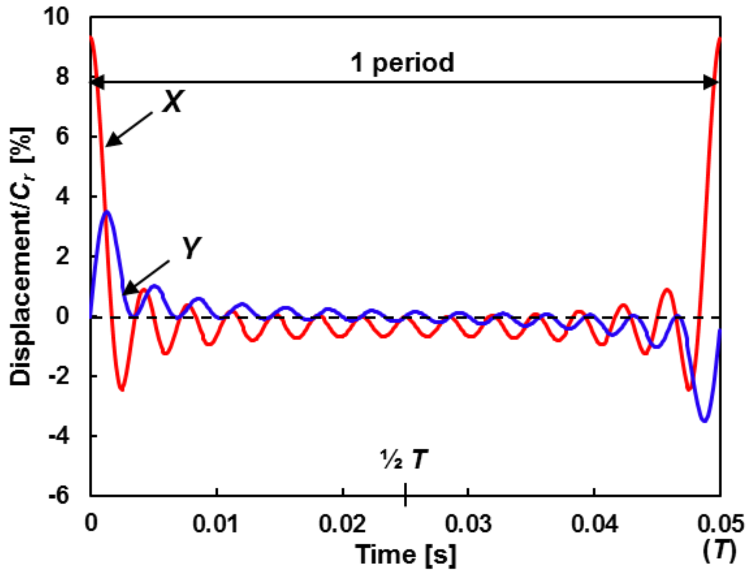


Rotor whirl motions with multiple-frequencies **CFD**

Method introduced by Li & Li (2014)



a = 1.5 μm ➔ X_{max} ~ 10%C_r
b = 1 μm ➔ Y_{max} ~ 5%C_r
ω_i = 20 - 280 Hz
(14 frequencies) ω_i/Ω = 0.12 – 1.68



$$X = a \cdot \sum_{i=1}^{N} \cos(\omega_i t) \ ; \ Y = b \cdot \sum_{i=1}^{N} \sin(\omega_i t)$$

Input: rotor displacements (*X,Y*) → output: seal reaction forces

CFD procedure for predicting seal performance

3D steady state flow solver

+

3D unsteady state flow solver

Multi-frequency, elliptic orbit method
Displacements $d_{x(t)}$, $d_{y(t)}$)

Solution:
Force ($f_{x(t)}$, $f_{y(t)}$)

Discrete Fourier Transform (DFT)

Frequency domain:
($F_{x(\omega)}$, $F_{y(\omega)}$)

Seal complex dynamic stiffness:

- ✓ **Direct:** $H = K + i\,(\omega C)$
- ✓ **Quadrature:** $h = k + i\,(\omega c)$
- ✓ **Direct damping** C

steady state performance:

- ✓ Seal leakage (m);
- ✓ Static pressure (P);
- ✓ Velocities ($U_{i,j,k}$);

Quantify how seal works!

Review of experimental record on seal force coefficients

→ **Do we know everything we should know?**

Representation of coefficients

Normalize coefficients with
Pressure drop **$(P_{in} - P_{out})$**,
Projected area **(LD)**,
Clearance **C_r**,
Shaft speed **Ω**

$$\begin{pmatrix} K^* & k^* \end{pmatrix} = \begin{pmatrix} K & k \end{pmatrix} \frac{C_r}{\left(P_{in} - P_{out}\right) LD}$$

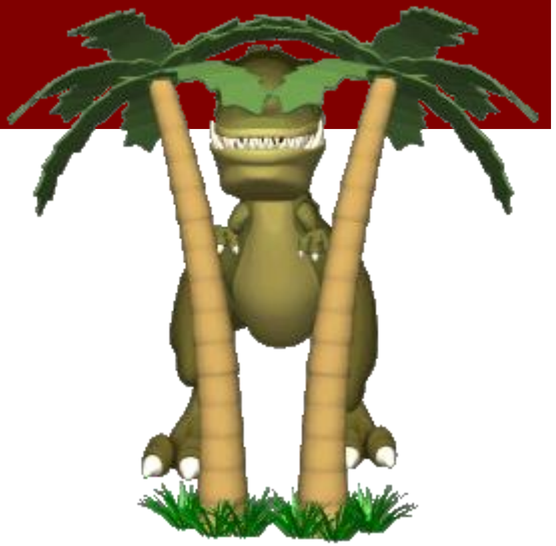
$$\begin{pmatrix} C^* & c^* \end{pmatrix} = \begin{pmatrix} C & c \end{pmatrix} \frac{C_r\,\Omega}{\left(P_{in} - P_{out}\right) LD}$$

Whirl frequency ratio: $WFR = \dfrac{k}{C\,\omega}$

Cross-over frequency: $C_{eff\,(\omega_{break})} = 0 \rightarrow \omega_{cross-over} = \left(\dfrac{k}{C}\right)$

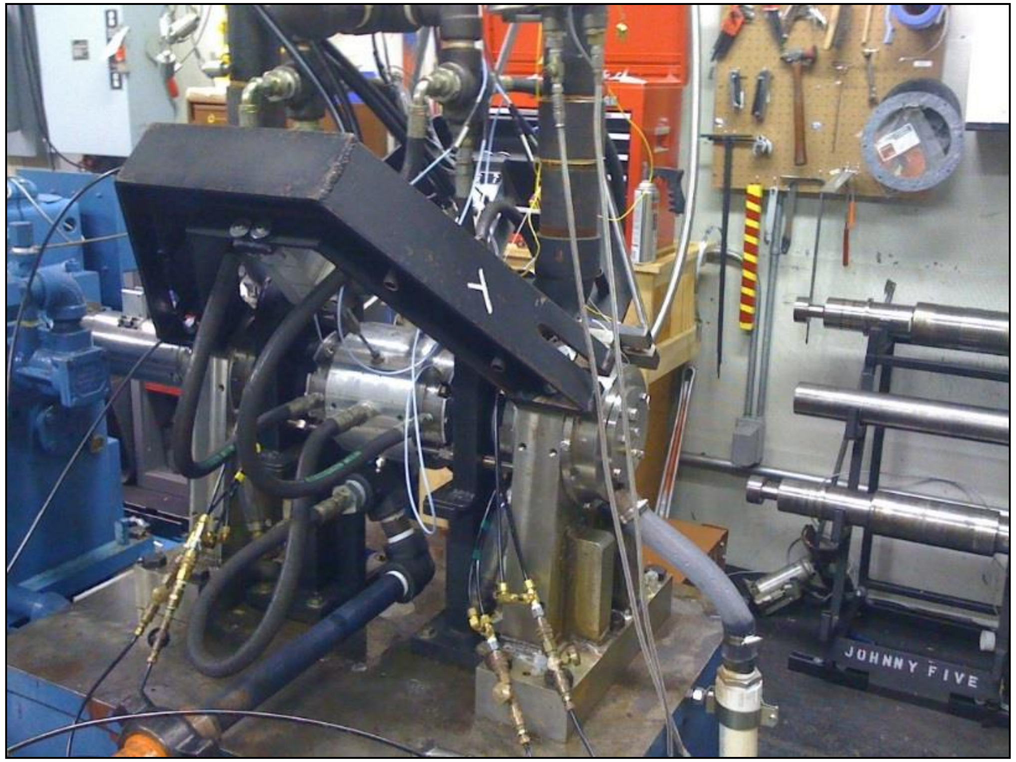
For comparison amongst known forced performance for various seal types.

Early record → 1989



1989: Compare honeycomb vs. labyrinth seal vs. uniform clearance

Childs, D., Elrod, D. and Hale, K., 1989, “Annular Honeycomb Seals: Test Results for Leakage and Rotordynamic Coefficients; Comparisons to Labyrinth and Smooth Configurations1,” ASME J. Tribol., **111** (4).



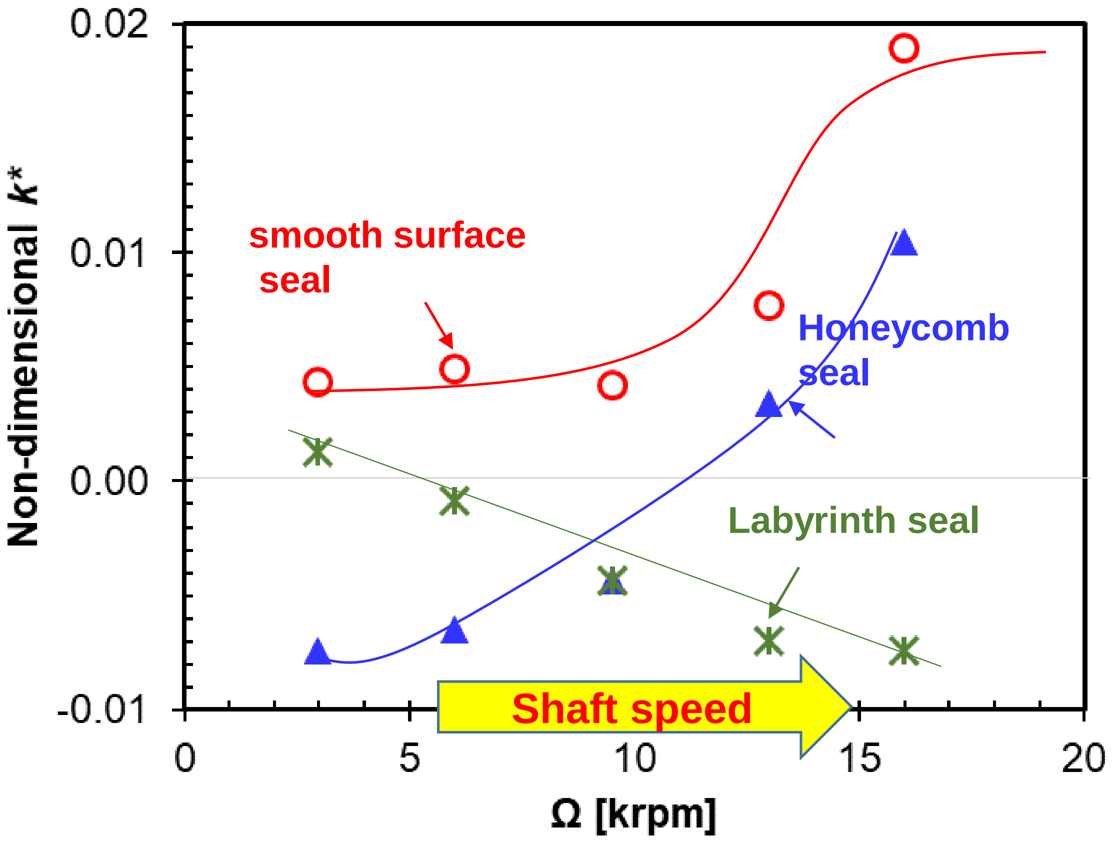
***L/D*= 0.34**	**HC #7** mm	**LS** mm
Radial clearance, C_r	0.41	0.41
Cavity/Cell width	1.57	3.2
Cavity/Cell depth	1.91	3.2
Number of blades	N/A	16

Length L = 50.8 mm, diameter D = 151.4 mm

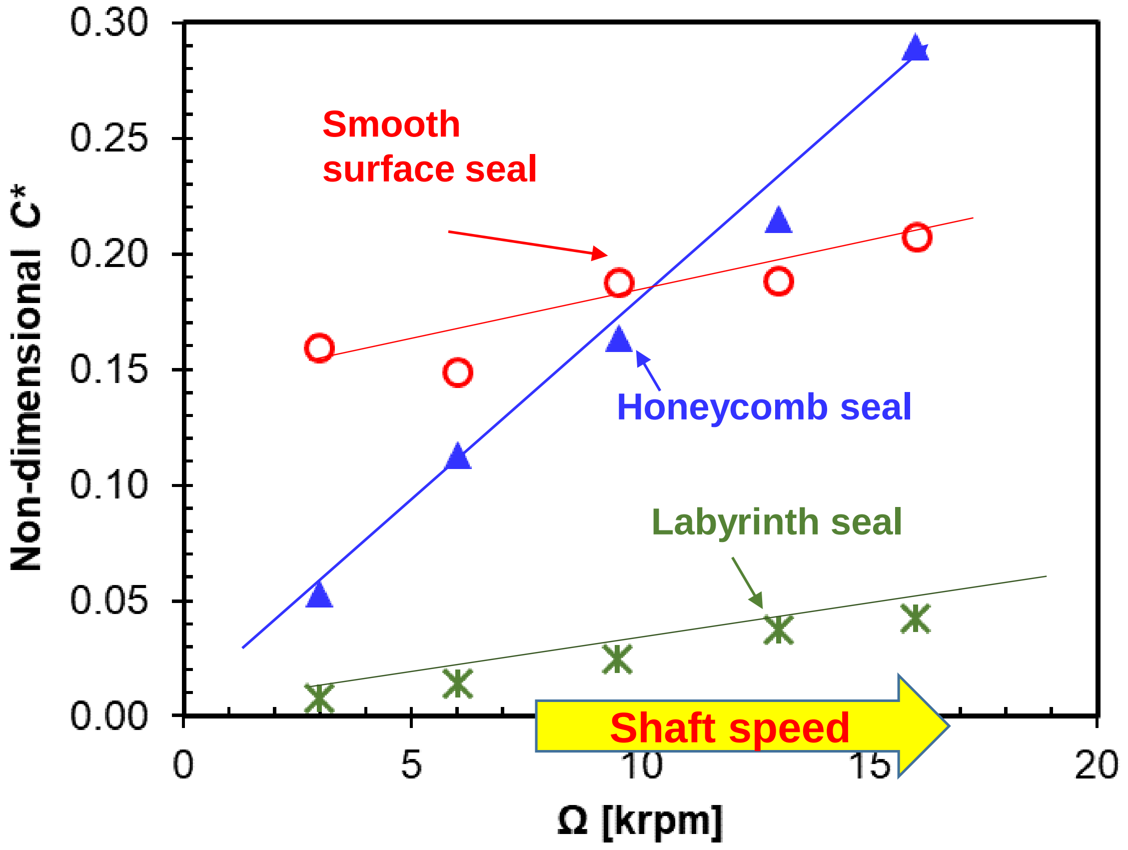
HC vs LS vs smooth surface seal

HC and LS: *Ps* = 3.08 bar, *Pa* = 1 bar.
No inlet preswirl

(a) Cross-coupled stiffness *k**



(b) Direct damping *C**



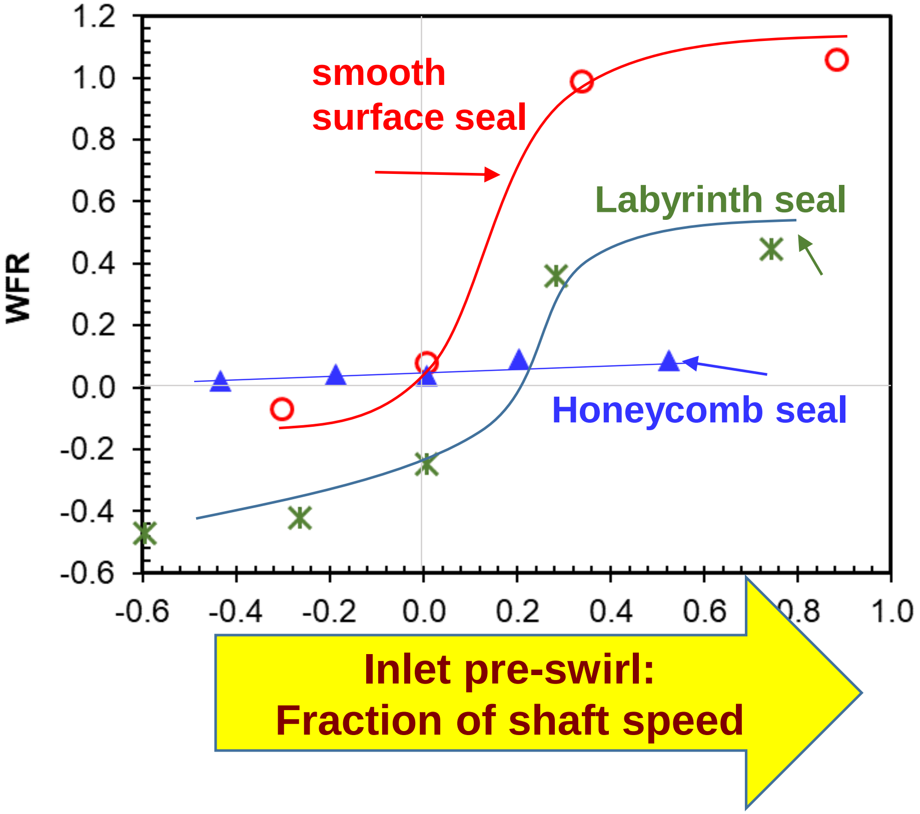
LS produces smallest damping and *k*<0. Uniform clearance seals offers largest *k* and direct damping *C*.

HC vs LS vs smooth surface seal

Ps **= 8.288 bar,** ***Pa*** **= 1 bar**
Speed = 16 krpm (35 m/s)

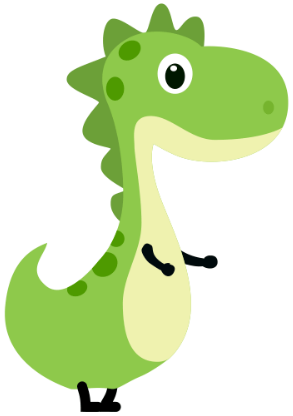
(c) Whirl Frequency Ratio

$$WFR = \frac{k}{C\,\omega}$$



Effect of whirl frequency on force coefficients (yet) unknown in 1989

Honeycomb seal offers lowest WFR ~ 0, whereas smooth surface seal shows ***WFR*** **→ 1 (worst for stability in spite of largest damping).**



More recent test data

Balance piston seals

2012

LS vs. honeycomb seal vs. pocket damper seal (PDS)

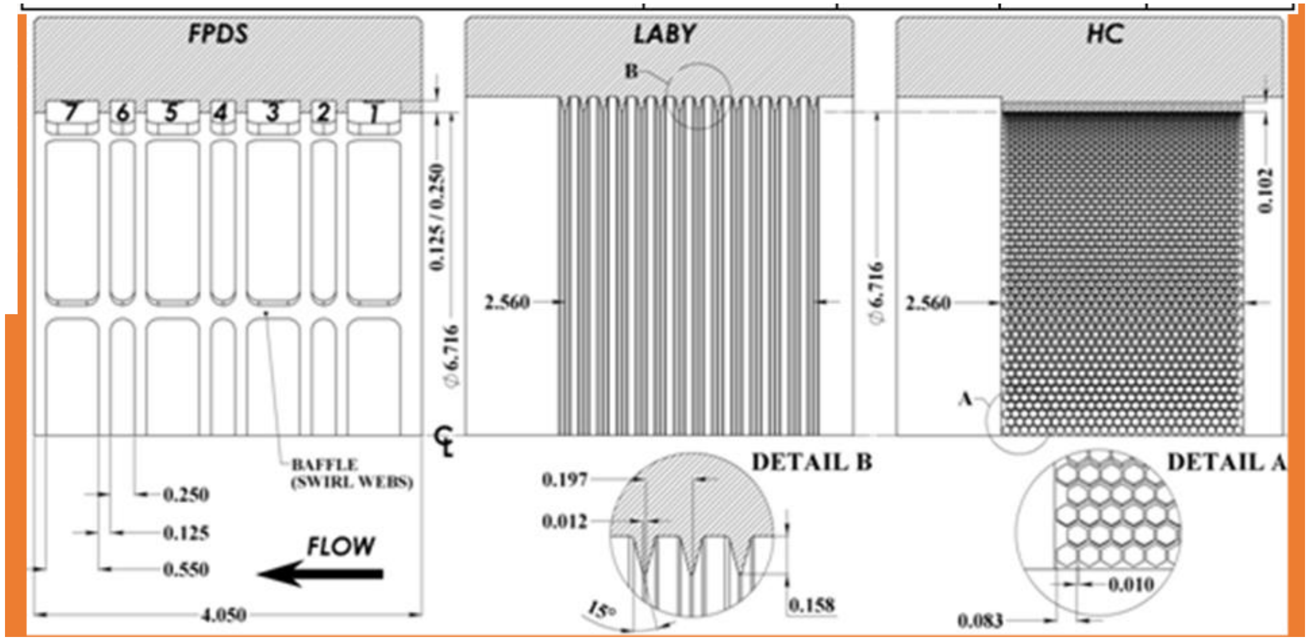
Diameter=170 mm	FPDS mm	HS mm	LS mm
Seal length, *L*	102	65	65
Cavity/Cell width	13.3/5.7	0.79	4.3
Cavity/Cell depth	3.1	3.2	4.3
Number of blades	8	N/A	20

Radial clearance C_r = 0.3 mm for 3 seals

15 krpm: surface speed 133 m/s

MODERATE ΔP

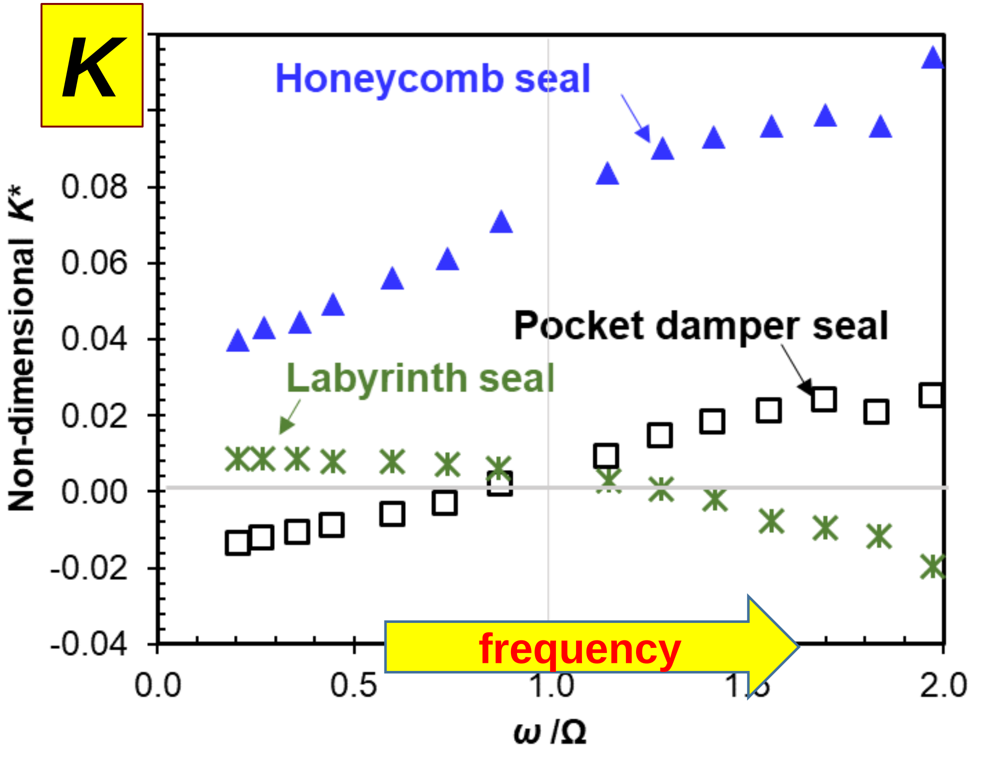
P_{in} = 6.9 bar, P_{out} = 1 bar



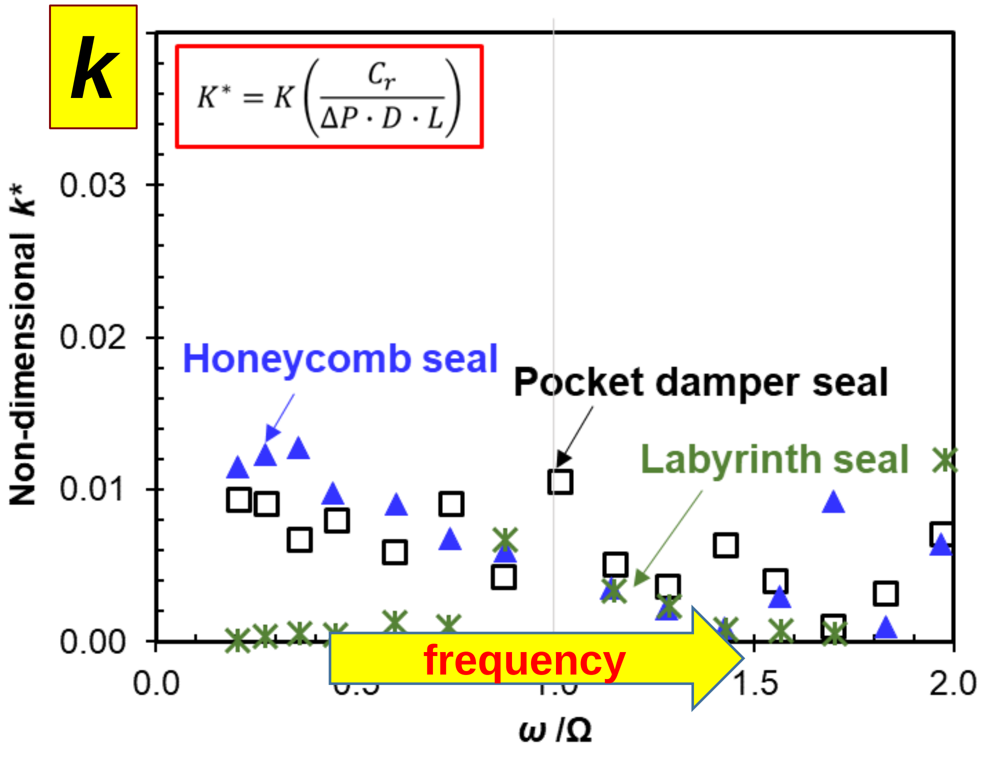
Ertas, B. H., Delgado, A., and Vannini, G., **2012,** "Rotordynamic Force Coefficients for Three Types of Annular Gas Seals with Inlet Pre-swirl and High Differential Pressure Ratio," ASME J. Eng. Gas Turbines Power, **134**(4).

Direct and cross stiffnesses

ΔP = 5.9 bar, rotor speed = 15 krpm, zero pre-swirl ratio



(a) Direct stiffness K^*

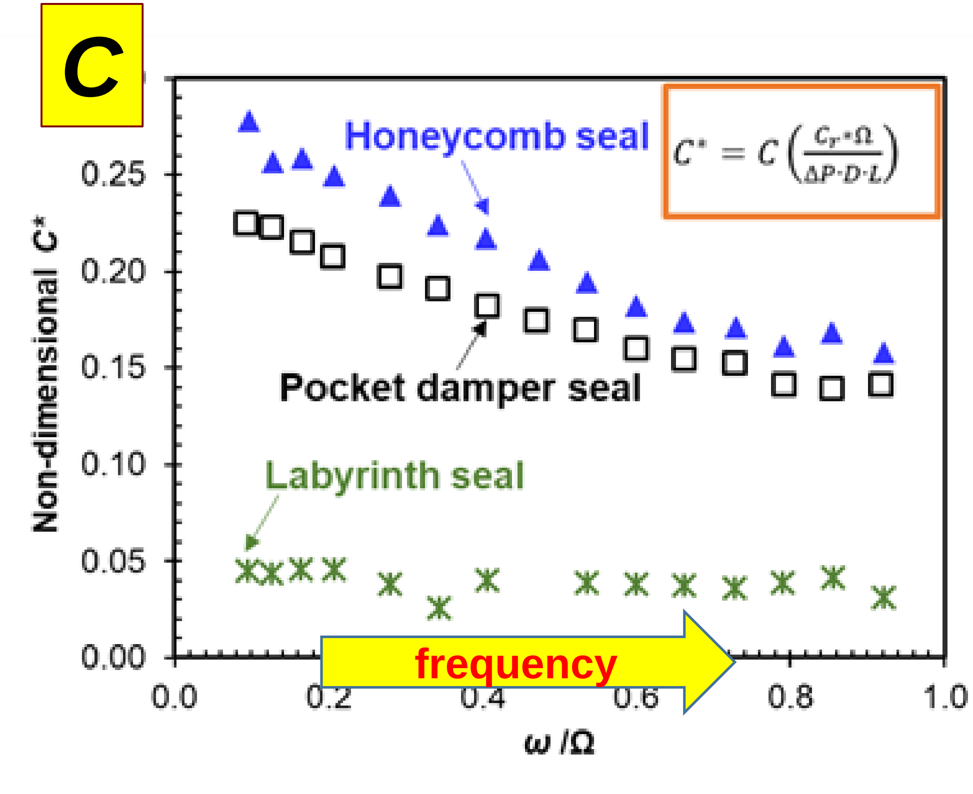


(b) Cross-coupled stiffness k^*

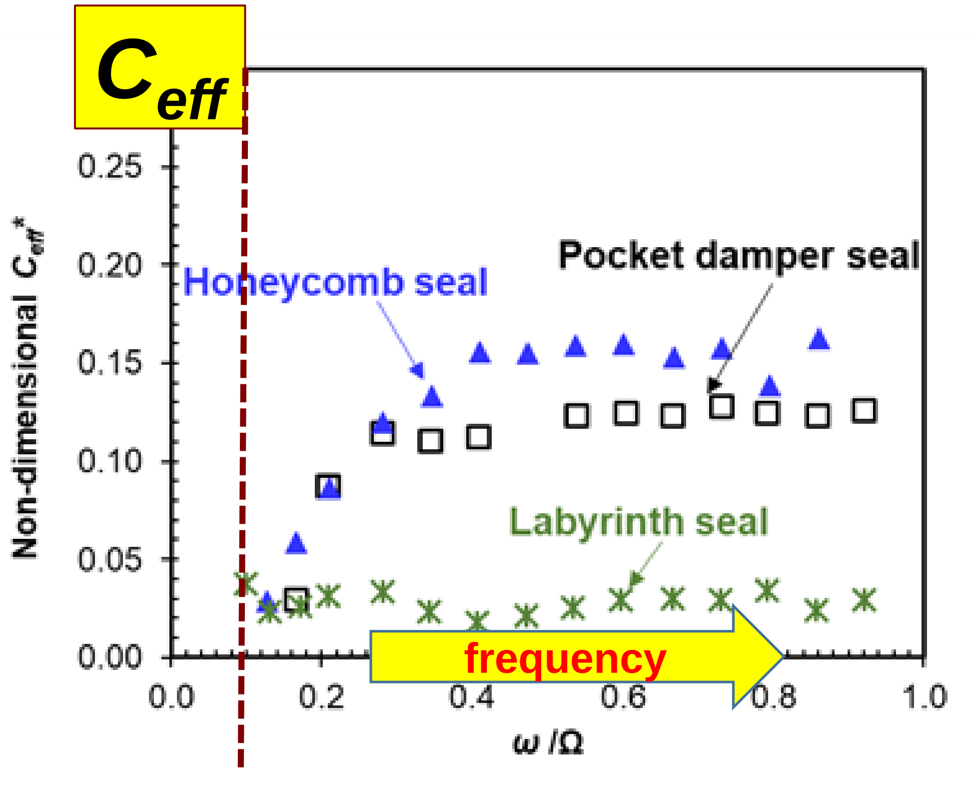
LS offers negligible coefficients. Honeycomb seal (HCS) and PDS show *K* growing with frequency. (*k*/*K*) is small.

Direct and effective damping

ΔP = 5.9 bar, rotor speed = 15 krpm, zero pre-swirl ratio



(c) Direct damping C*



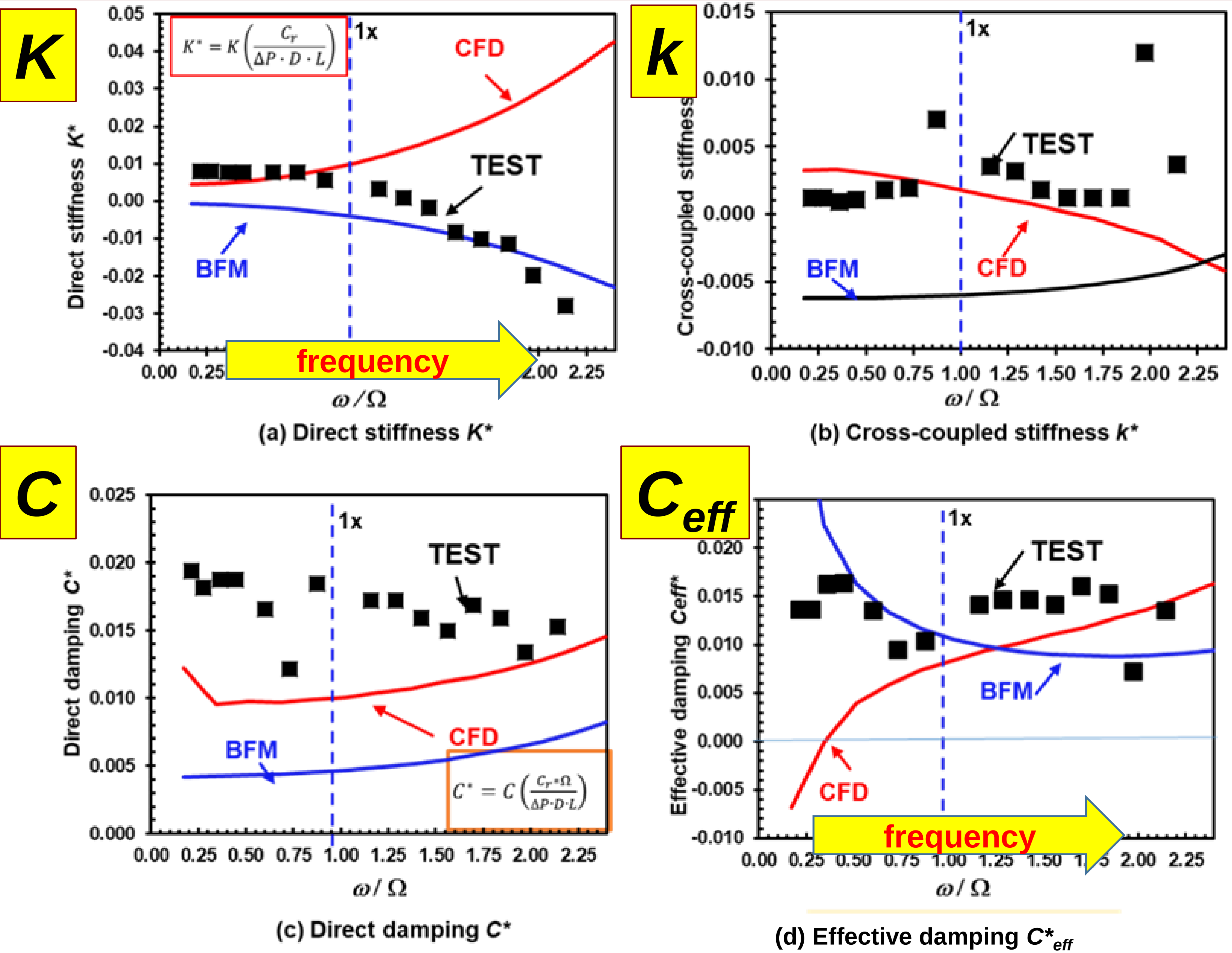
(d) Effective damping C^*_{eff}

In damper seals, the cross-over frequency defines the transition from negative to positive C_{eff}.

Honeycomb seal (HCS) and PDS show +++ larger damping than LS. At high frequencies, HCS has slightly + effective damping. Cross-over frequency is low ~ 0.1 of running speed.

LS BFM and CFD predictions

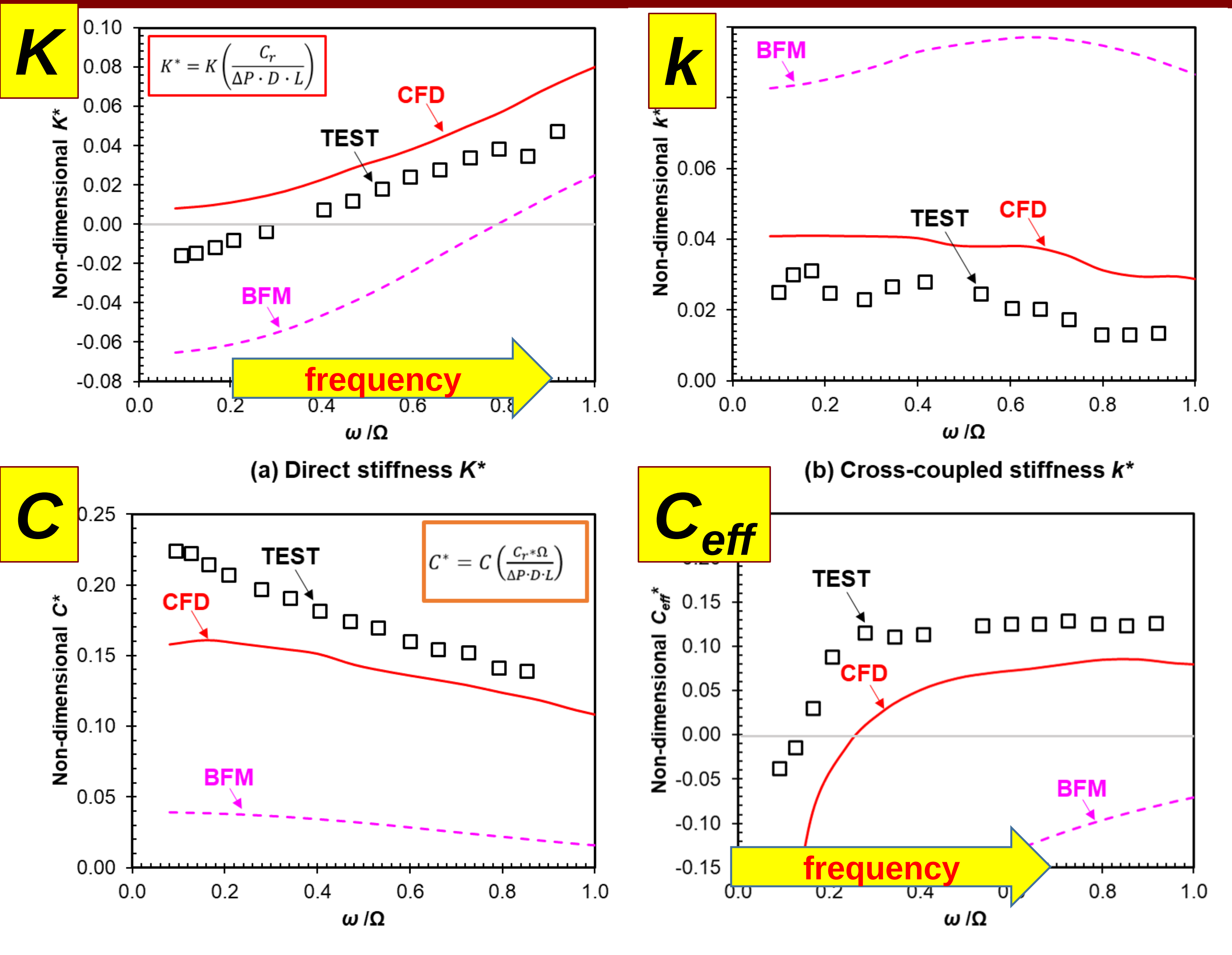
ΔP = 5.9 bar, rotor speed = 7 krpm, zero pre-swirl ratio



Compared to test data, **BFM does +++ better than CFD** to predict stiffness (*K*). Both methods do poorly for direct damping (*C*)

PDS BFM and CFD predictions

ΔP = 5.9 bar, rotor speed = 15 krpm, zero pre-swirl ratio



(a) Direct stiffness K*

(b) Cross-coupled stiffness k*

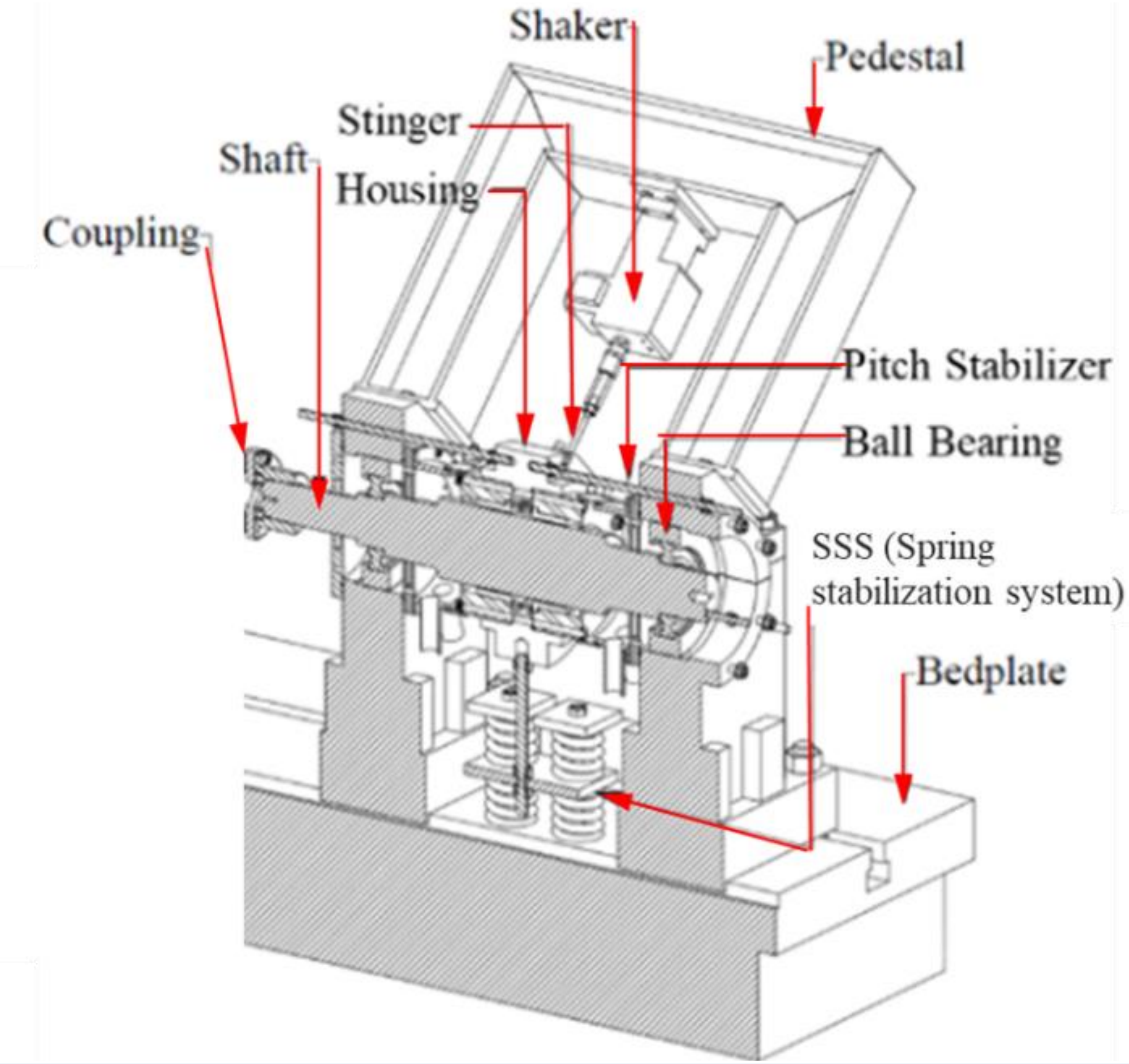
(c) Direct damping C*

(d) Effective damping C^*_{eff}

Compared to test data, CFD does +++ better than BFM to predict direct stiffness (*K*), cross-stiffness (*k*) and direct damping (*C*).

Fully partitioned pocket damper seal (FPDS)

2020



Comparison vs. honeycomb seal and labyrinth seal

A. Delgado, L. San Andres, J. Thiele, J. Yang and F. Cangioli, **2020,** "Rotordynamic Performance of a Fully-Partitioned Damper Seal: Experimental and Numerical Results," Proc. of the 49th Turbomachinery Symposium, Houston (**Also ASME GT2022-83164**).

FPDS vs. honeycomb seal and labyrinth seal

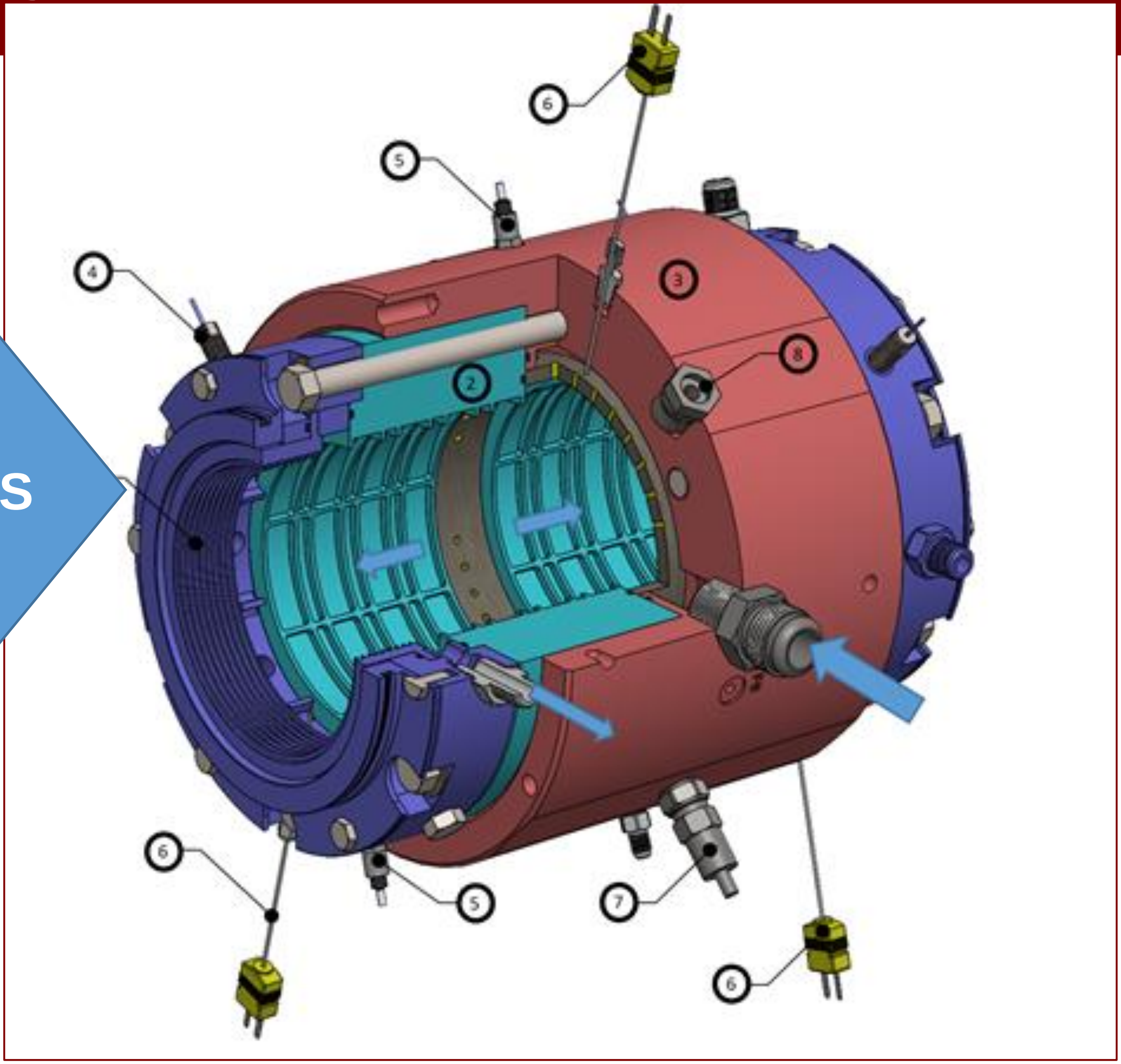
Diameter=115 mm	FPDS	HS	LS
	mm	mm	mm
Seal length, *L*	85	86	86
Cavity/Cell width	13.3/5.7	0.79	4.3
Cavity/Cell depth	3.6	3.2	4.3
Number of blades	8	N/A	20

Radial clearance (3 seals) C_r = 0.2 mm

10 krpm: surface speed 60 m/s

LARGE ΔP **P_{in} = 70 bar, P_{out} = 35 bar**

FPDS



HS and LS data:

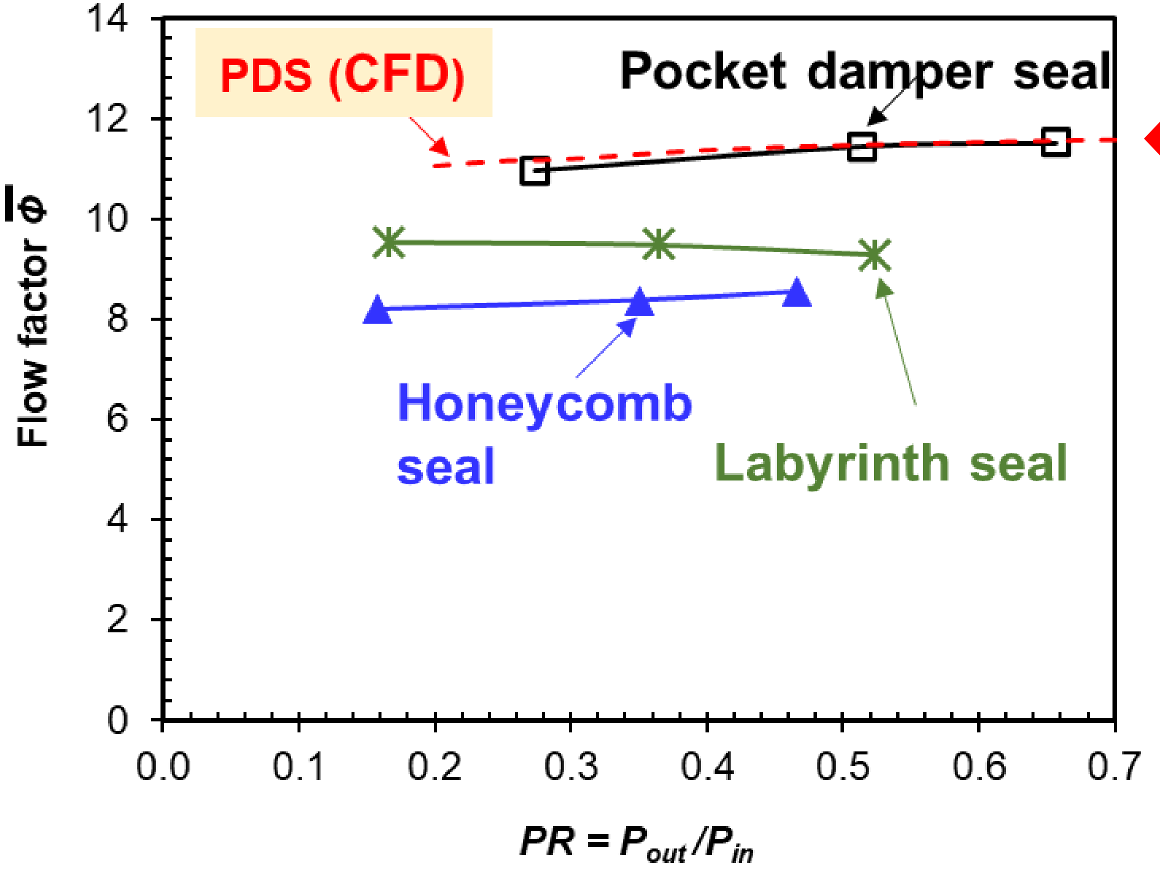
Sprowl, T. B., **2003**, “A Study of the Effects of Inlet Pre-swirl on the Dynamic Coefficients of a Straight-bore Honeycomb Gas Damper Seal,” Master thesis, Texas A&M University, College Station, TX.

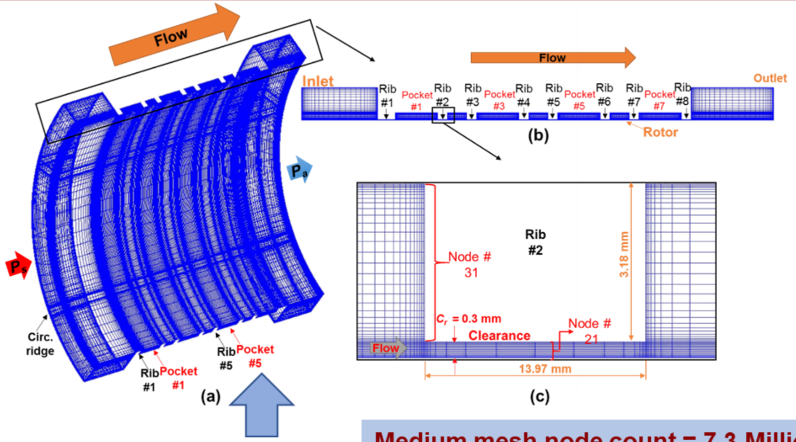
Picardo, A. M., **2003**, “High Pressure Testing of See Through Labyrinth Seals,” Master thesis, Texas A&M University, College Station, TX.

Seals leakage: test & CFD

$$\bar{\Phi} = \frac{\dot{m}\sqrt{T}}{D\,P_{in}\sqrt{1-PR^2}} \sim \pi\, C_{eff}\, \frac{1}{\sqrt{R_g}}$$

P_{out} =70 bar. Inlet swirl velocity = 60 m/s, and rotor speed = 10-20 krpm, inlet swirl ratio =0.7-1.3. Temperature=12 C





Medium mesh node count = 7.3 Million

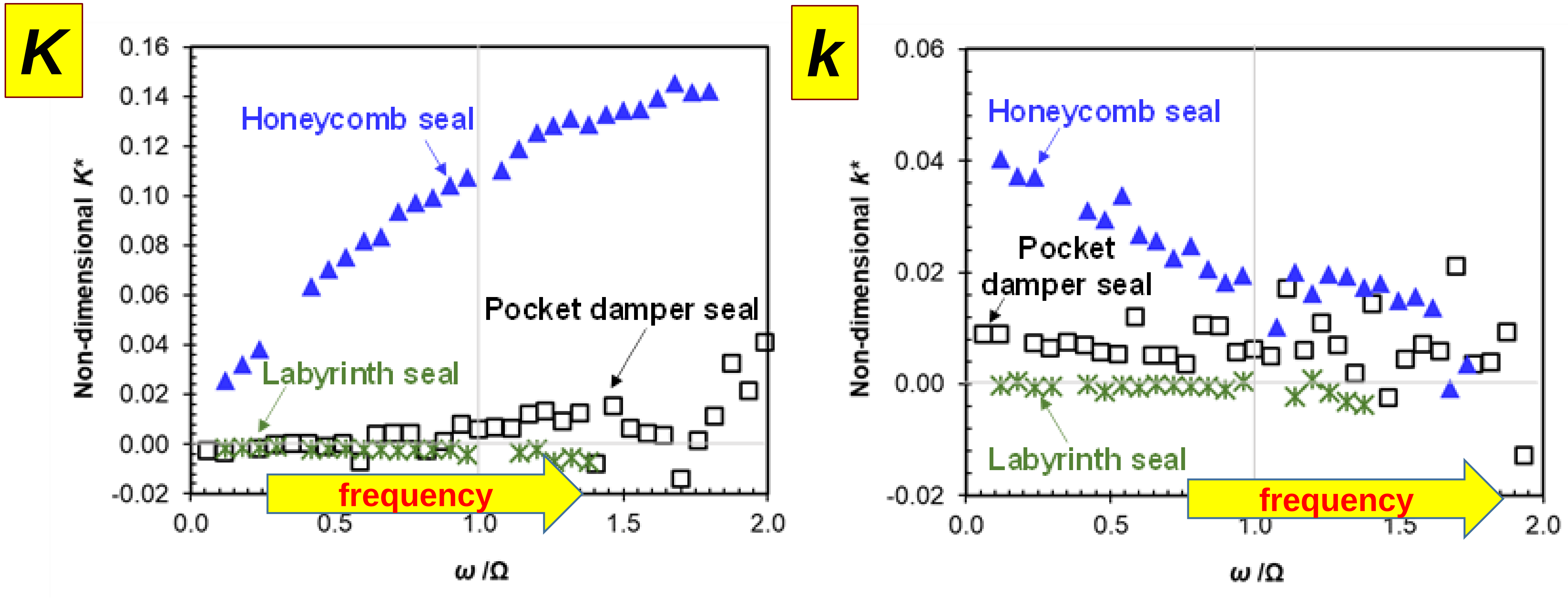
Loss coefficient c_d =0.22 for honeycomb seal, c_d =0.25 for labyrinth seal, and c_d =0.30 for fully-partitioned PDS.

→ **Honeycomb seal has smallest effective clearance.**

Radial clearance (3 seals) C_r = 0.2 mm

Direct and cross stiffnesses

ΔP = 35 bar, rotor speed = 10 krpm,
Low pre-swirl ratio



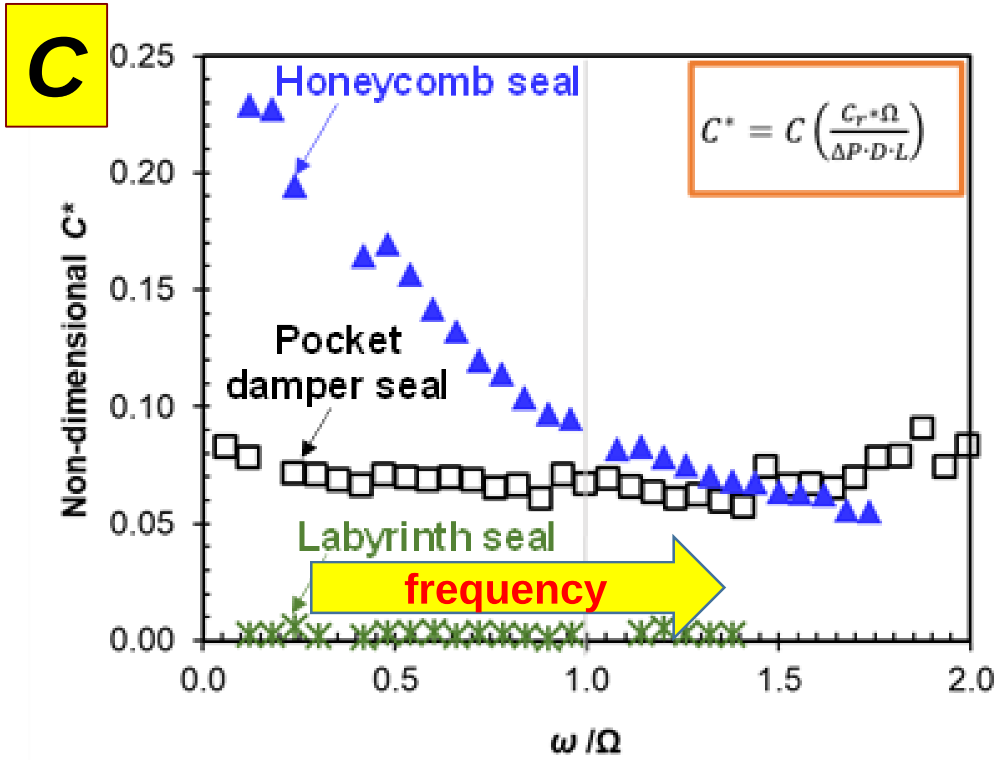
(a) Direct stiffness K^*

(b) Cross-coupled stiffness k^*

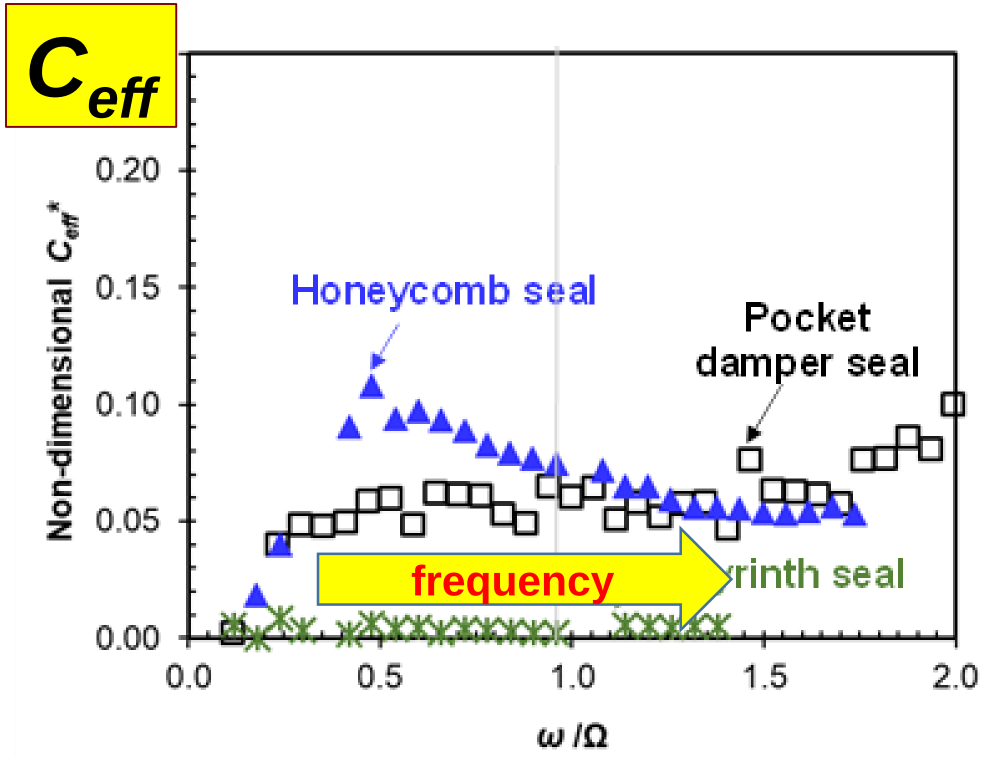
LS offers negligible coefficients. Honeycomb seal (HCS) shows largest *K* that grows with frequency.

Direct and effective damping

ΔP = 35 bar, rotor speed = 10 krpm,
Low pre-swirl ratio



(c) Direct damping C^*

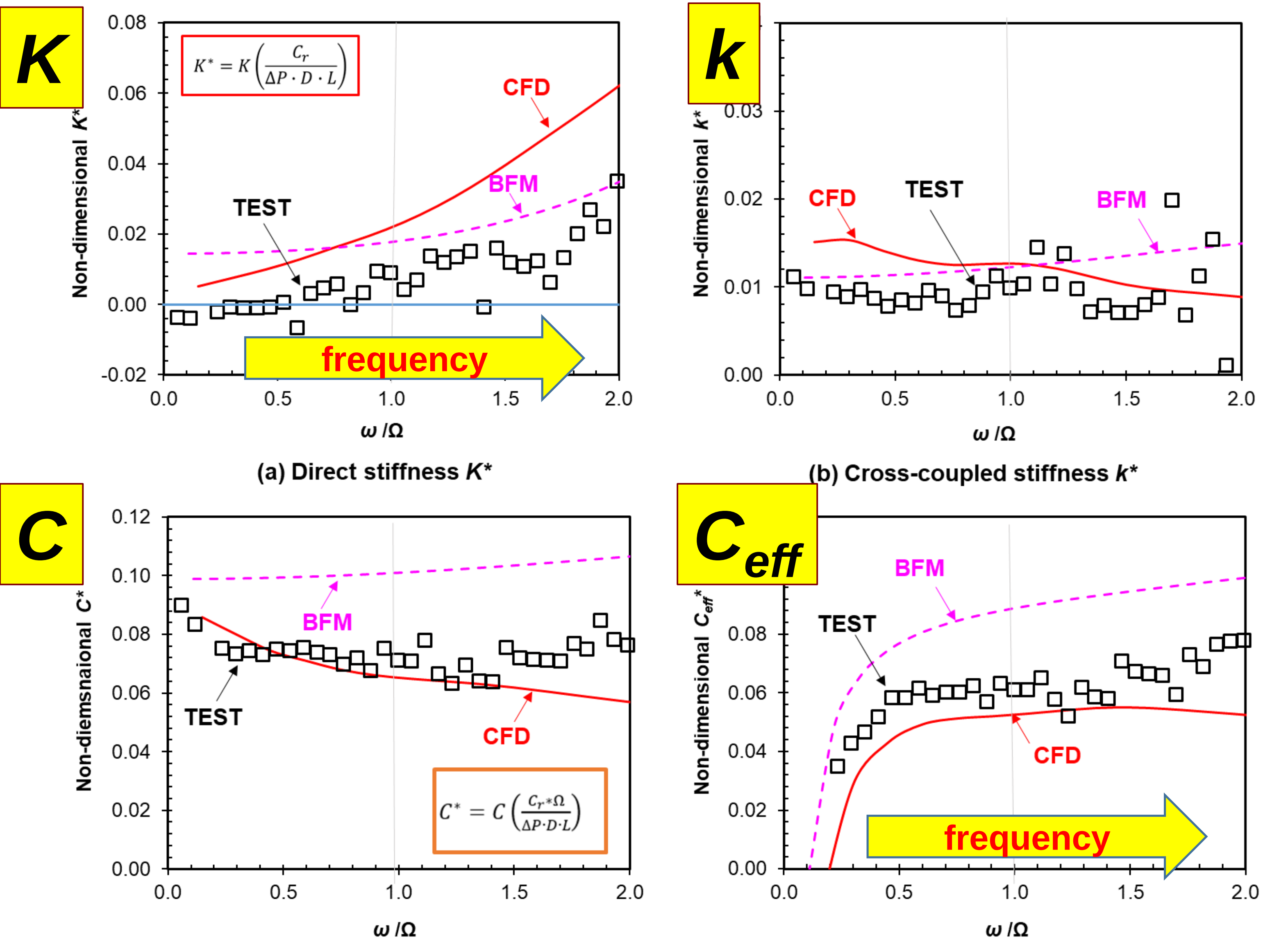


(d) Effective damping C^*_{eff}

Honeycomb seal (HCS) and PDS show +++ larger damping than LS, in particular at low frequencies. HCS and PDS show same effective damping for frequencies = synchronous or higher. Cross-over frequency is low ~ 0.01 of running speed.

PDS BFM and CFD predictions

ΔP = 5.9 bar, rotor speed = 15 krpm, zero pre-swirl ratio



(a) Direct stiffness *K**

(b) Cross-coupled stiffness *k**

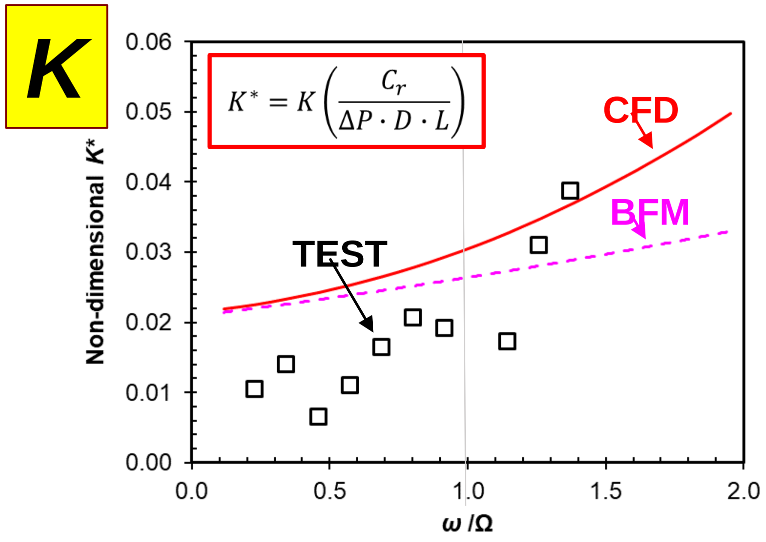
(c) Direct damping *C**

(d) Effective damping C^*_{eff}

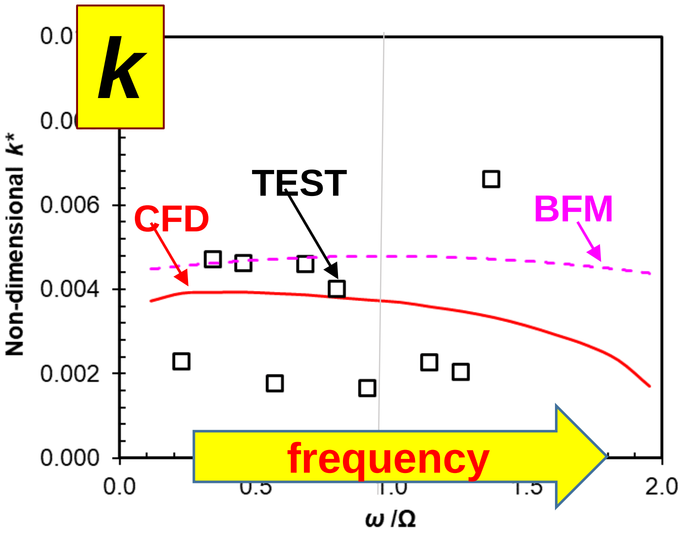
Compared to test data, **BFM does better than CFD** to predict direct & cross-stiffnesses (*K*,*k*). CFD does better for direct damping (*C*). **Methods deliver more or less C_{eff} than test data.**

PDS: CFD & BFM vs Test data

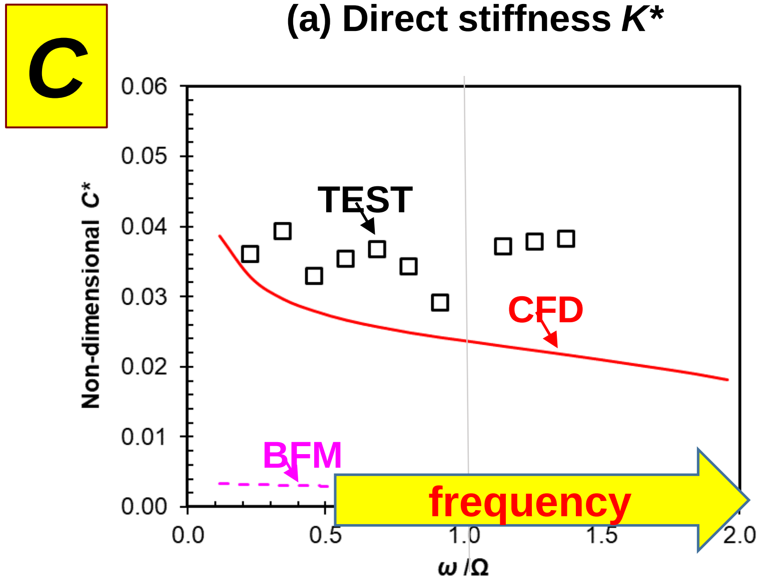
ΔP = 2.3 bar, rotor speed = 5.2 krpm, Null pre-swirl ratio



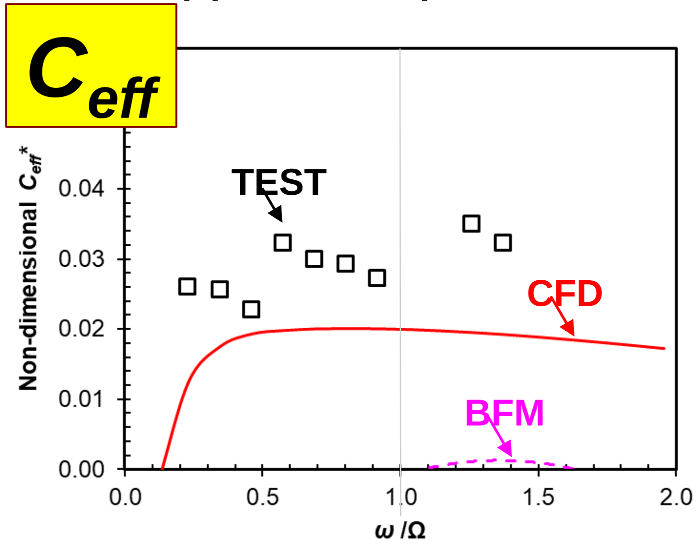
(a) Direct stiffness *K**



(b) Cross-coupled stiffness *k**



(c) Direct damping *C**



(d) Effective damping C^*_{eff}

CFD does ++ better than BFM to predict direct damping but over predicts cross-stiffness. BFM does poorly predicting the PDS effective damping (C_{eff} ~0)

Knowledge ANNULAR CLEARANCE GAS SEALS

1. Overview of annular clearance seals
2. Bulk-flow and CFD models for seal analysis
3. Seals leakage & their effective clearance
4. **Seals' force coefficients**
5. Closure – The road Ahead

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Knowledge on seals' force coefficients

The whirl frequency affects the forced response of gas damper seals (honeycomb & pocket damper seals). Tests and predictions show seal "stiffness hardening" and loss of damping at high frequencies.

Large direct stiffness (K>>0) enable to design/operate seals as load bearing elements and affecting placement of critical speeds.

Damper seals offer large effective damping (C_{eff} >> 0) with break frequencies at a fraction of operating speed.

Bulk-flow models and CFD do not always predict accurately force coefficients of gas annular seals.

Rules of thumb

Good until changed!

The review of the experimental record show that the direct stiffness and effective damping of damper seals can be (safely) estimated from

At synchronous frequency $\omega=\Omega$

Short length Seal

$$K \sim [0.05 \rightarrow 0.10]\frac{(P_{in} - P_{out})LD}{C_r}; \; C_{eff} \sim [0.04 \rightarrow 0.12]\frac{(P_{in} - P_{out})LD}{C_r \Omega}$$

Honeycomb seal

Swirl ratio decreases

Seal with incompressible fluid

$$K \sim [0.20 \rightarrow 0.40]\frac{(P_{in} - P_{out})LD}{C_r}; \; C_{eff} \sim [0.04 \rightarrow 0.08]\frac{(P_{in} - P_{out})LD}{C_r \Omega}$$

Knowledge ANNULAR CLEARANCE GAS SEALS

1. Overview of annular clearance seals
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5. **Closure – The road ahead**

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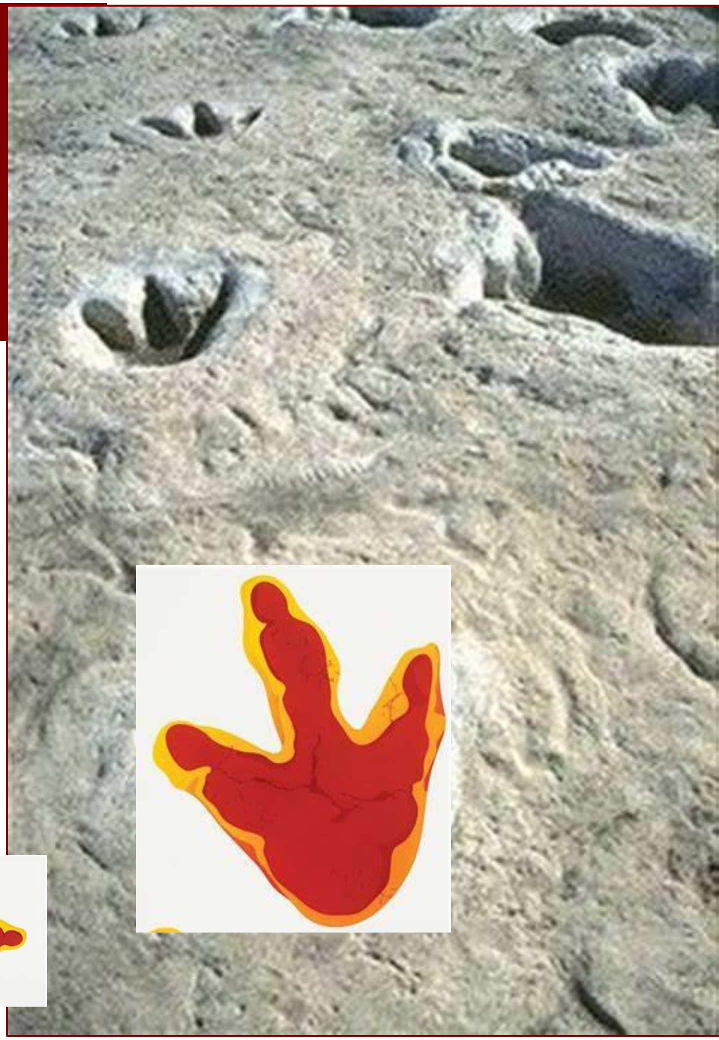
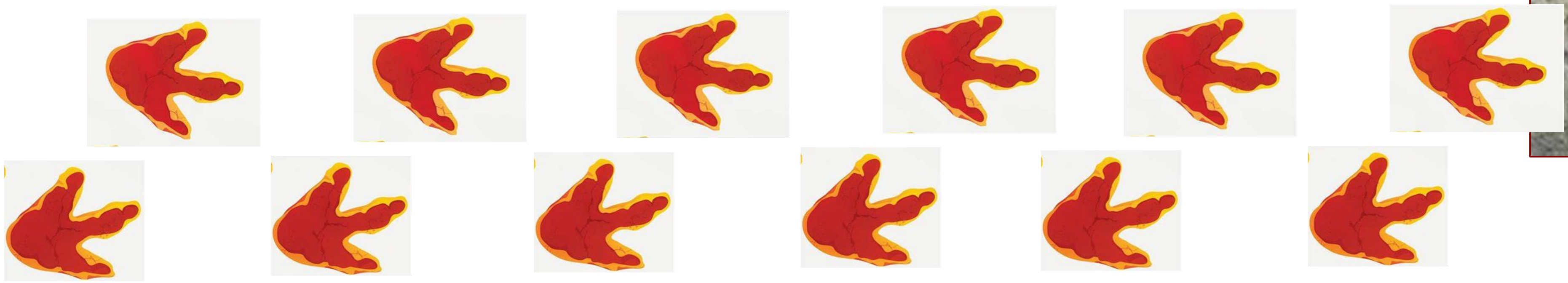
The road ahead

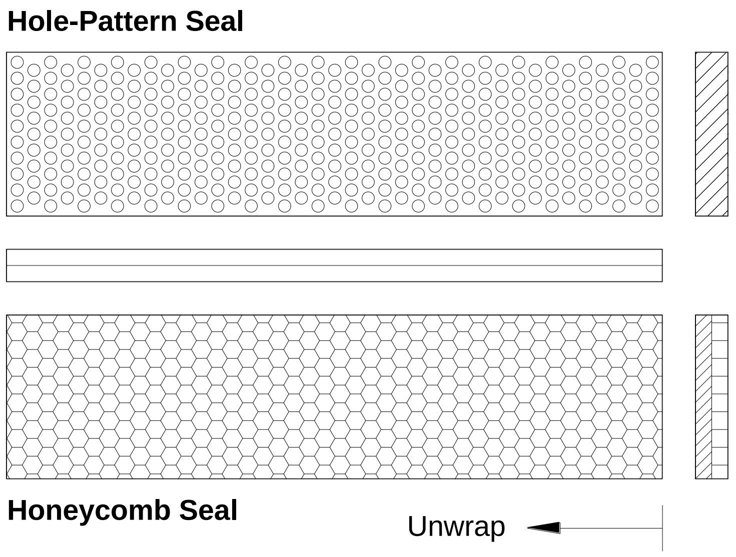
To seal or not to seal....

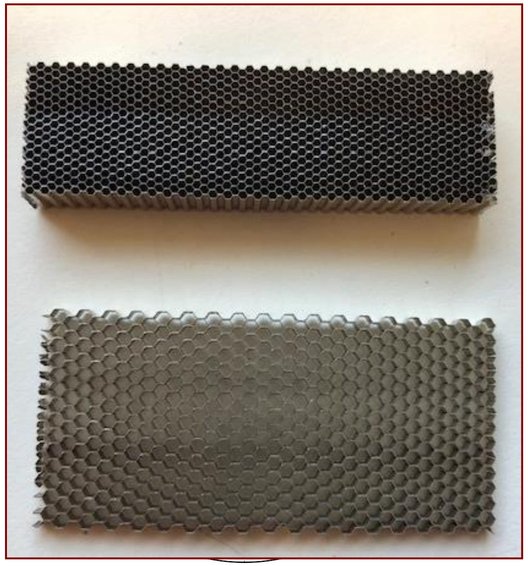




The bottom line…

The experimental record, field practice & physical models (CFD & BFM) show DAMPER SEALS offer a much better dynamic forced response than labyrinth seals.



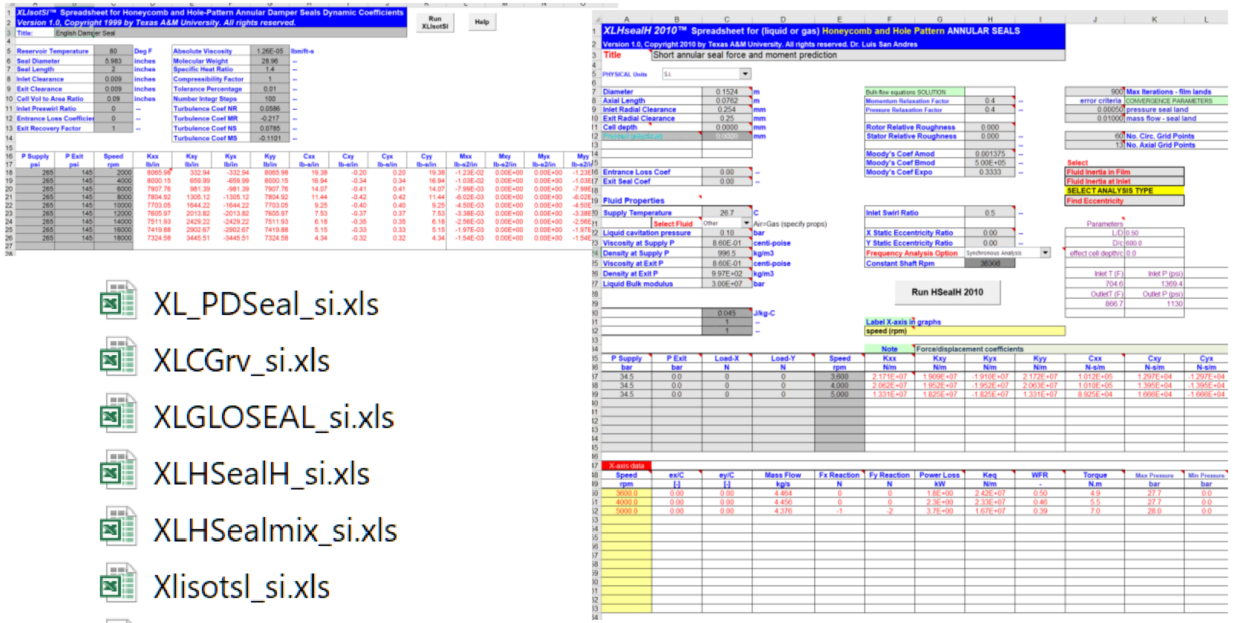


Damper seals can be designed to reduce synchronous amplitude rotor motions and to control placement of critical speeds.

Novel developments include 3D-configurations ADM (printed) with greater damping & stiffness coefficients.

TURBOLAB IMPACT

Funded by industry, work at the Turbo Lab since the early 1980s' has been instrumental to the development of predictive models anchored to test data.



- XL_PDSeal_si.xls
- XLCGrv_si.xls
- XLGLOSEAL_si.xls
- XLHSealH_si.xls
- XLHSealmix_si.xls
- XIisotsI_si.xls
- XLLaby_si.xls
- XLLUBGT_si.xls

XLTRC2® includes a comprehensive set of codes to model seals leveraging data and physical insight from experimental work. This feature differentiates XLTRC2® from other rotordynamic software packages.

The multiple contributions of Dr. Dara Childs (former TL Director) and Dr. John Vance to the rotordynamics of turbomachinery are acknowledged with gratitude.

The continued support of the Turbomachinery Research Consortium, Mitsubishi Co, Siemens Power, GE, and many other industries is acknowledged. Thanks to countless graduate and under-graduate students working diligently to advance the state of the art.

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