Predictions vs. Test Results for Leakage and Force Coefficients of a Fully Partitioned Pocket Damper Seal and a Labyrinth Seal – Limitations of the Current Computational Model

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TRC Project 32514/15196PD

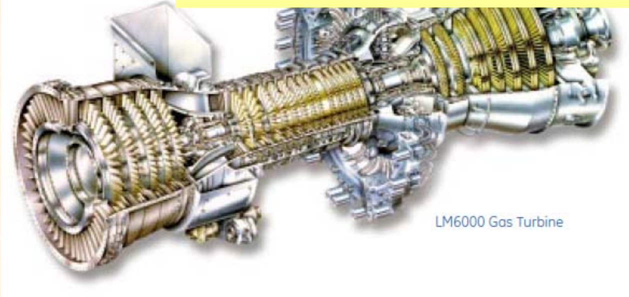
Year I

Justification

Trends in High Performance Turbomachinery

- Higher speeds & more compact units
- Extreme operating temperatures and pressures
- More efficient & reliable

Source: GE Energy



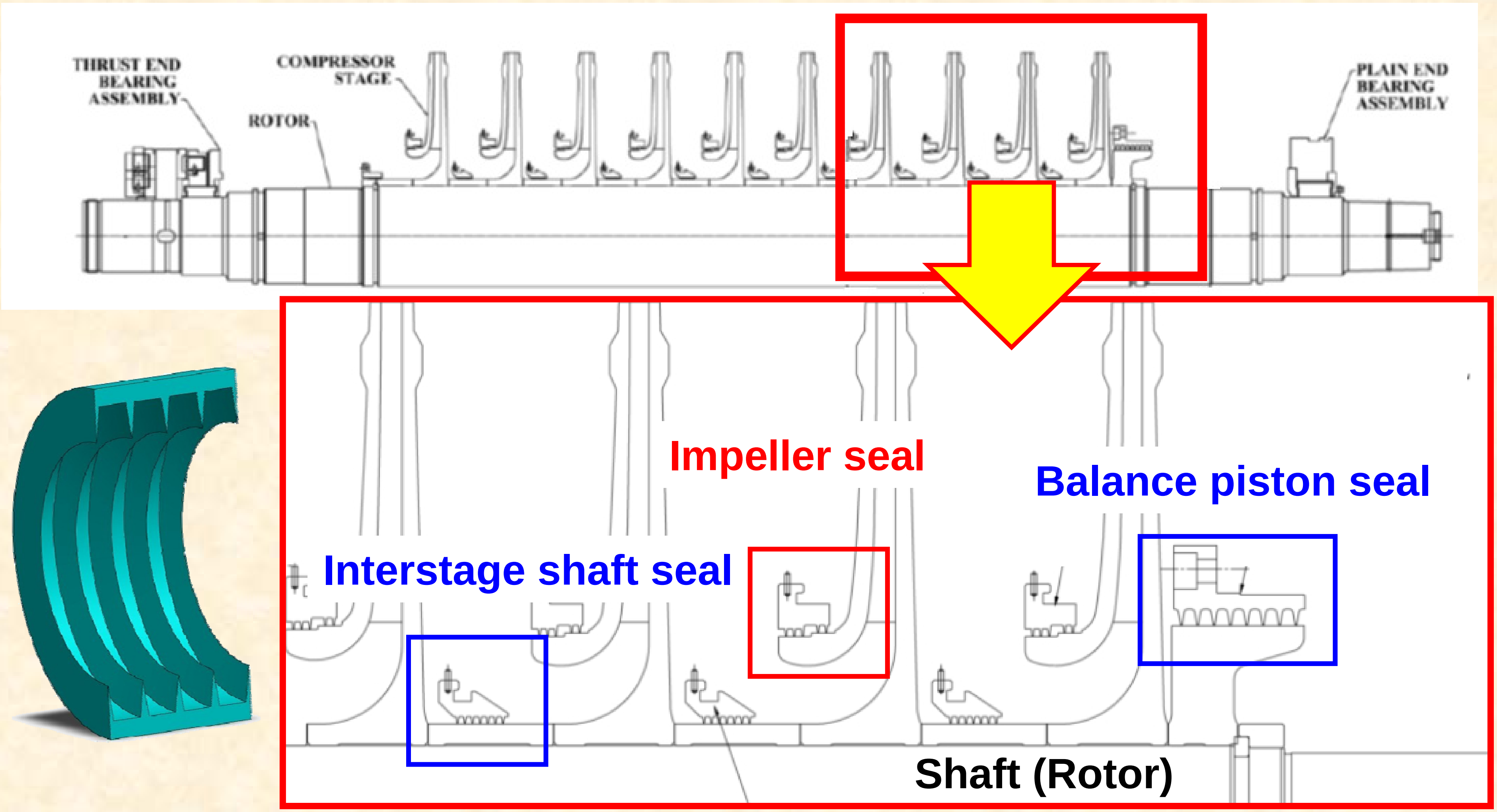
Issues of Importance

- Reduce secondary flows (parasitic leakage)
- Reduce specific fuel consumption
- Increase power delivery
- Eliminate potential for rotordynamic instability

SEALS

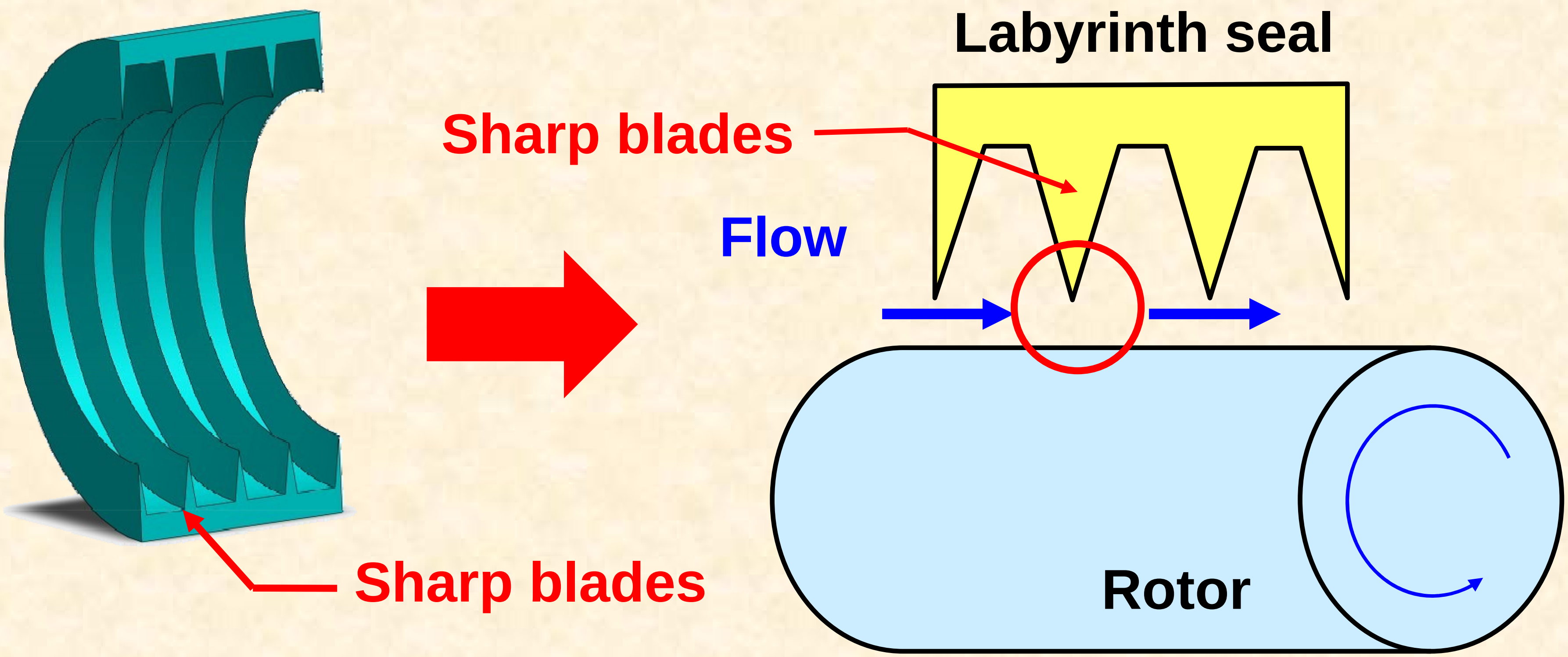
Background

Labyrinth seals (LS) in a straight-through compressor



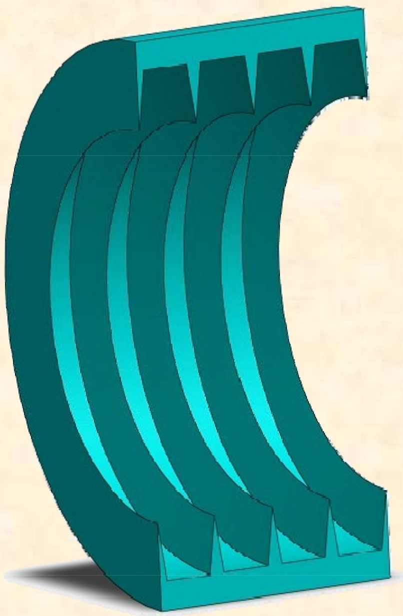
Ertas, B. H., 2005, Ph.D. Dissertation, Texas A&M University

Labyrinth seals reduce leakage



Leakage model between sharp blade and rotor treated as an orifice.

Disadvantages of labyrinth seals



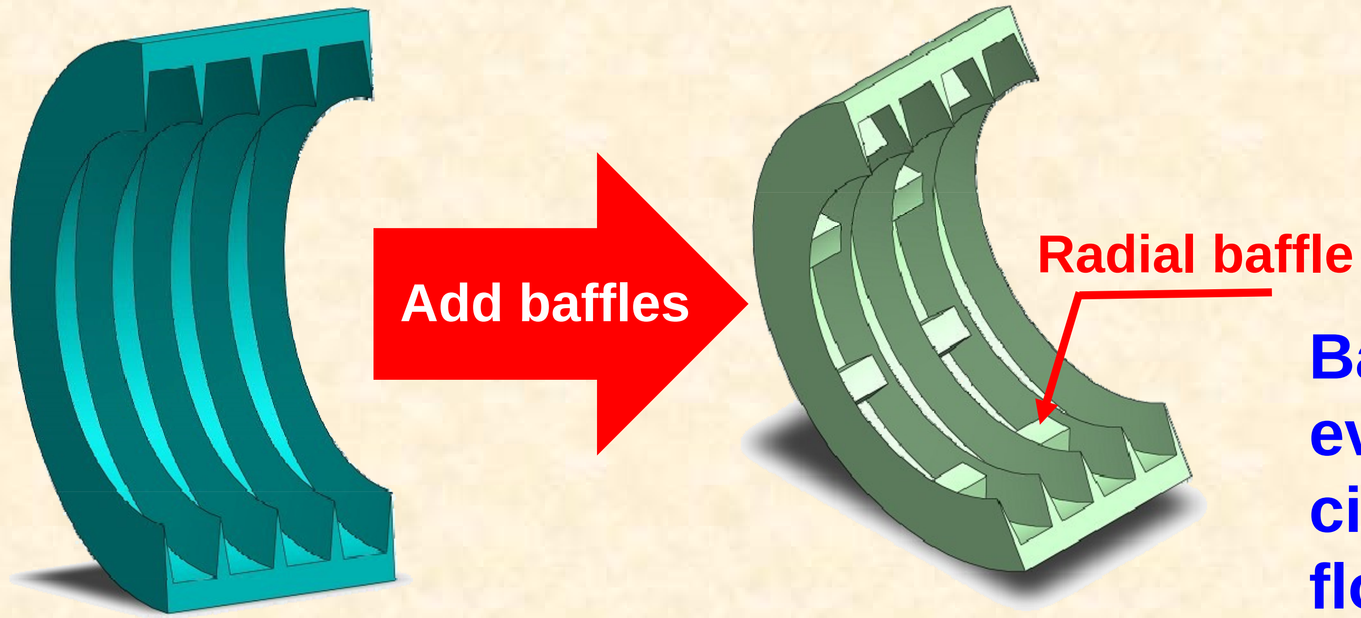
- **Direct damping coefficient is usually small, even negative.**
- **Large cross coupled stiffness drives rotor-bearing system instability.**

$$C_{eff} = C - \frac{k}{\omega}$$

LSs provide limited effective damping and could even destabilize a whole rotor-bearing system.

About pocket damper seals

Labyrinth Seals (LS) **Pocket Damper Seals (PDS)**



Baffles brake the evolution of the circumferential flow velocity

- **PDS leaks more than LS.**
- **PDS provides ++ more effective damping and reduces rotor vibration amplitudes more effectively than a LS.**

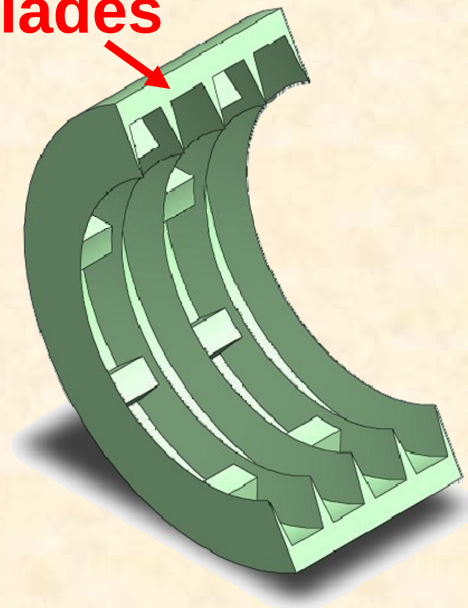
Vance, J. M., and Schultz, R. R., 1993
Vance, J. M., and Li, J., 1996

TAMU PDSeal© code (1999)

Neumman leakage model

$$\dot{m}_i = \frac{(C_k C_f H)_i}{R_g T}\sqrt{P_{i-1}^2 - P_i^2}$$

Orifice



Main flow equation

$$\frac{1}{R_g T}\left[\frac{\partial (PA)_i}{\partial t} + \frac{\partial (PAU)_i}{R_a \partial \Theta}\right] + \zeta_r(\dot{m}_{i+1} - \dot{m}_i) = 0$$

Circumferential momentum equation

$$\frac{1}{R_g T}\left[\frac{\partial (PAU)_i}{\partial t} + \frac{\partial (PAU^2)_i}{R_a \partial \Theta}\right] + \zeta_r(\dot{m}_{i+1}U_i - \dot{m}_i U_{i-1}) = -\frac{A_i}{R_a}\frac{\partial P_i}{\partial \Theta} + \Delta\tau_{xi}$$

Wall shear stress difference (Moody's friction factor)

Li, J., San Andrés, L., and Vance, J., 1999

TAMU PDSeal© code (1999)

- PDSeal over predicts leakage (4-10%) compared to test results.
- PDSeal predicts direct damping coefficients in agreement with test data.
- Direct stiffness & damping coefficients and leakage are weak functions of rotor speed. Cross-stiffnesses are typically small.

Li, J., San Andrés, L., Vance, J., Ransom, D., and Aguilar, R.

Progress in 2013

XLPDS© GUI created to interface with PDSEAL©

GUI linked to XLTRC^2 suite to predict performance of pocket damper seals (sharp blades)

(a) Leakage
(b) Stiffness and damping coefficients

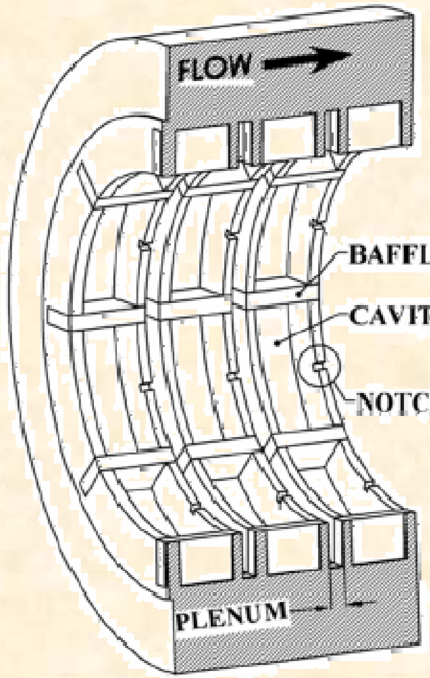
vs. pressure difference, rotor speed and excitation frequency.

Contact me for a demonstration on the use of the GUI.

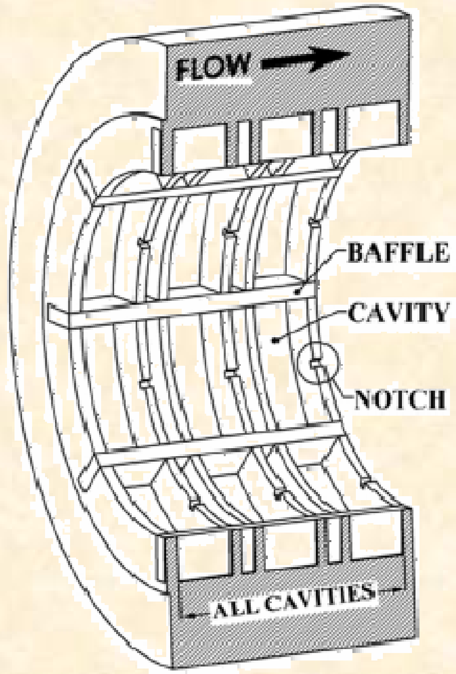
Commercial PDS & FPDS

Commercial PDS and FPDS have thick walls

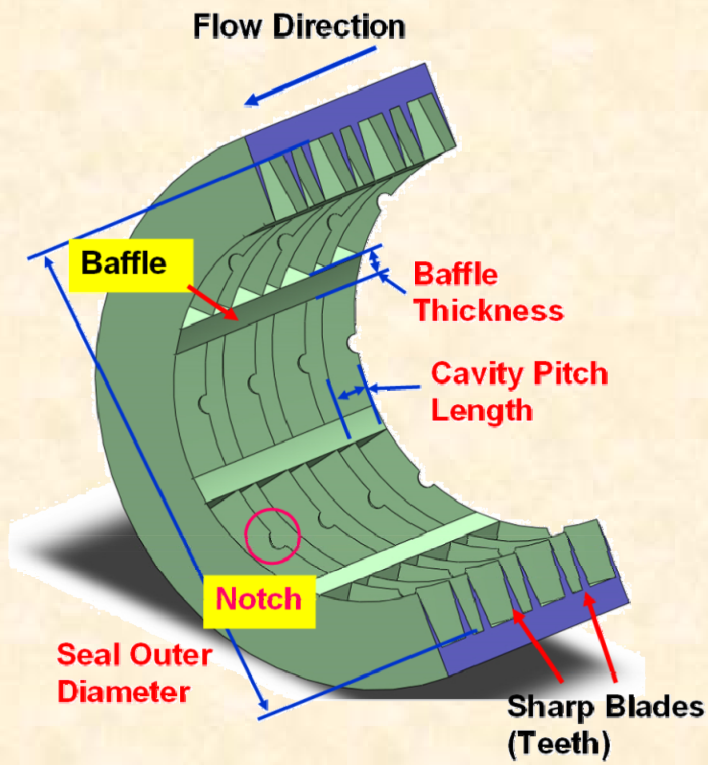
Original PDS had sharp blades



Pocket damper seal (PDS)



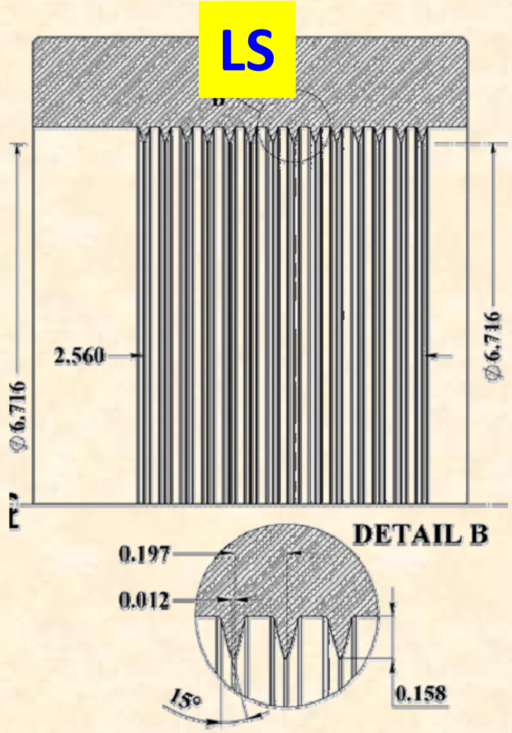
Fully partitioned pocket damper seal (FPDS)

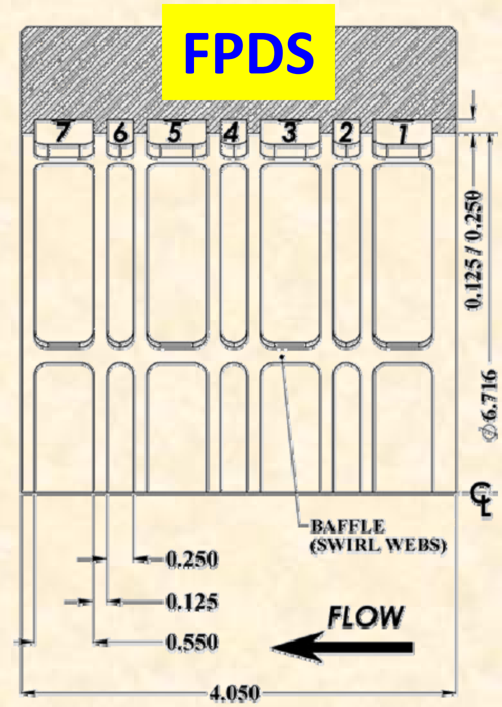


Ertas, B.H., Vance, J.M., 2007

Examples – seals geometry

Ertas, B.H., Delgado, A., Vannini, G., 2012





	14 bladed LS	8 bladed, 8 pocket FPDS
Blades properties	All active	Active / Inactive (without notch / with notch)
Cavity depth	4 mm	3.175 mm
Cavity axial length	5 mm	14 mm / 6.35 mm
Blade thickness (tip)	~ 0	6.35 mm / 3.175 mm
Radial clearance	0.3 mm	0.3 mm
Seal overall length	65 mm	103 mm
Rotor diameter	170 mm	170 mm

Examples: operating conditions

Inlet pressure	6.9 bar (Absolute pressure)			
Back pressure (Atmosphere)	1 bar (Absolute pressure)			
Excitation frequency	0 - 250 Hz			
Inlet temperature	286 K (13° C)			
Rotor speed	7 krpm	15 krpm	7 krpm	15 krpm
Rotor surface velocity	62 m/s	133 m/s	62 m/s	133 m/s
Inlet preswirl velocity	0	0	60 m/s	60 m/s
Preswirl ratio	0	0	0.96	0.45

Inlet preswirl ratio =inlet circumferential flow speed / rotor surface velocity

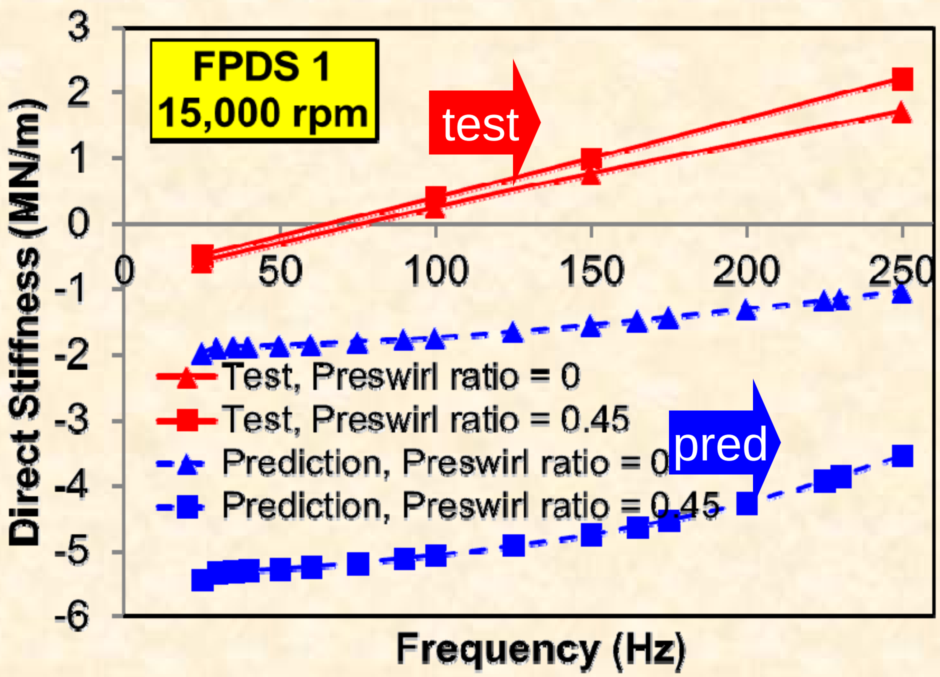
Gas	Air
Molecular weight	28.97
Gas compressibility factor	1
Specific heat ratio	1.4
Viscosity	18 μPa·s at 13° C

Ertas, B.H., Delgado, A., Vannini, G., 2012

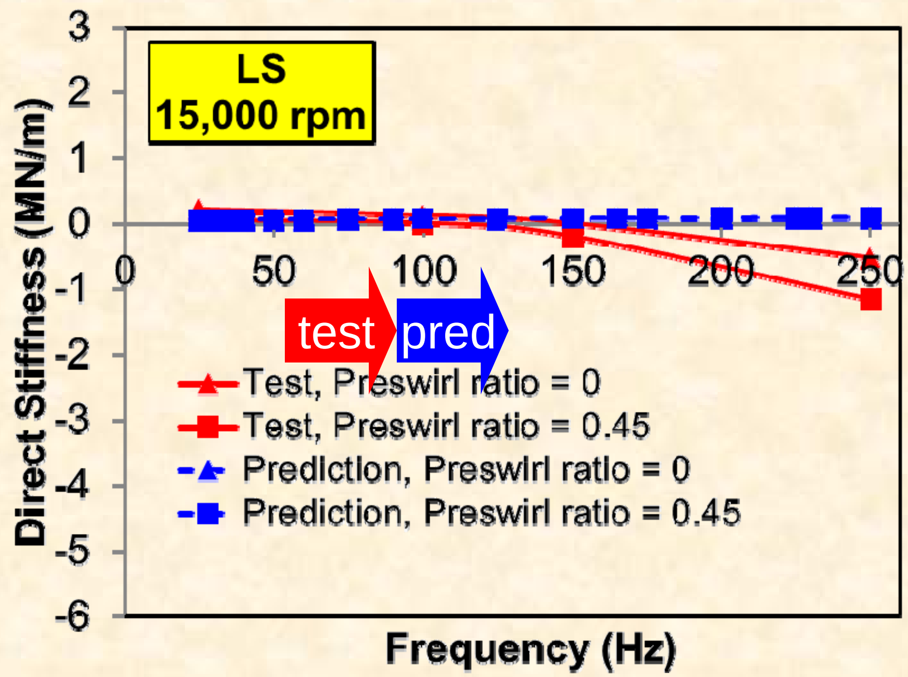
Direct Stiffness

rotor speed 15 kpm
preswirl ratios=0 & 0.45

Fully partitioned pocket damper seal



Labyrinth seal



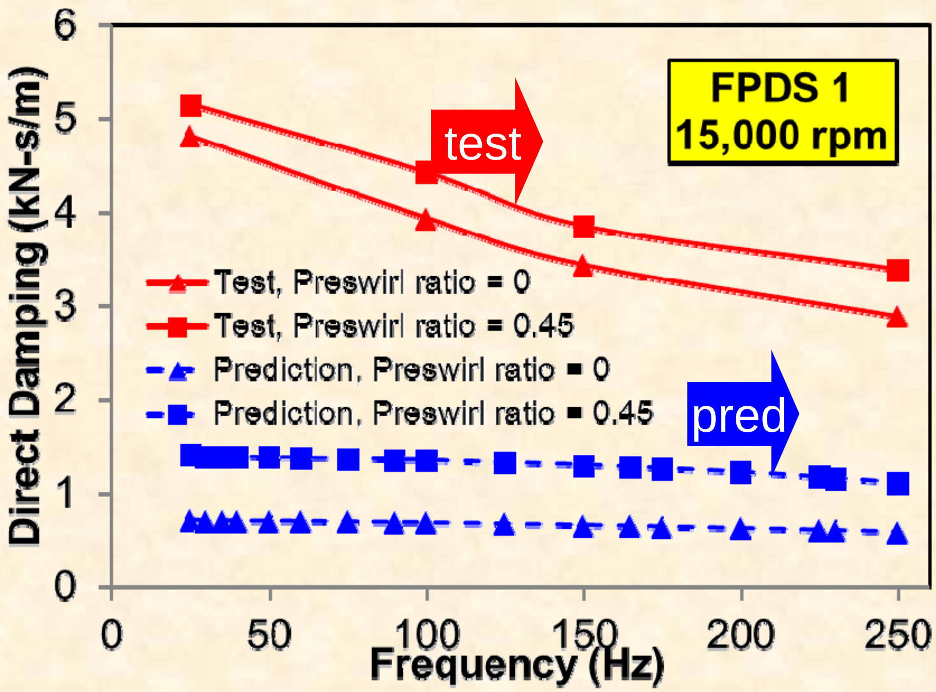
PDSeal© predicts well LS stiffness & misses stiffness for FPDS

Ertas, B.H., Delgado, A., Vannini, G., 2012

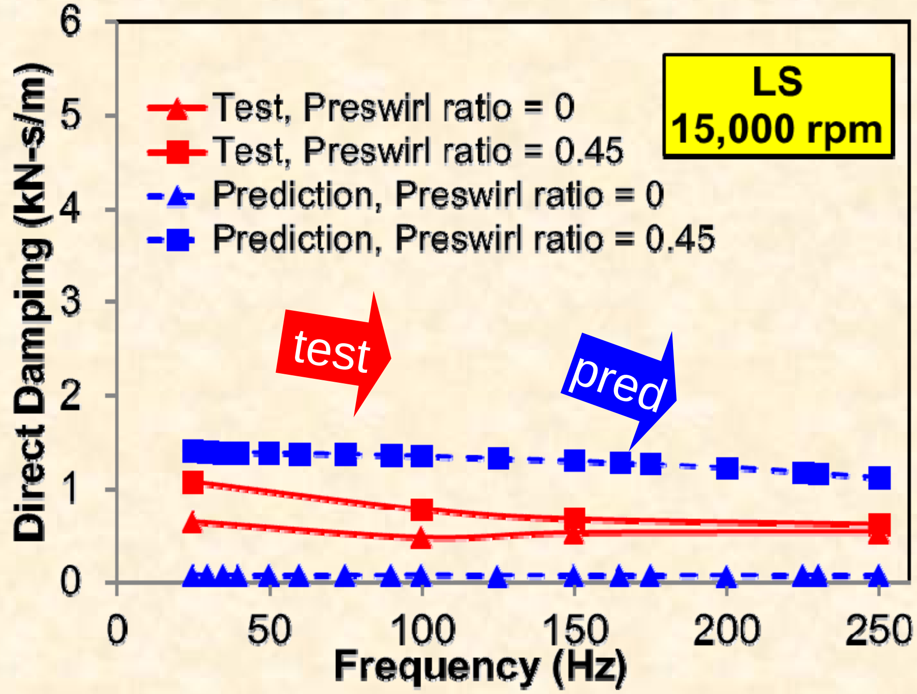
Direct Damping

rotor speed 15 kpm
preswirl ratios=0 & 0.45

Fully partitioned pocket damper seal



Labyrinth seal



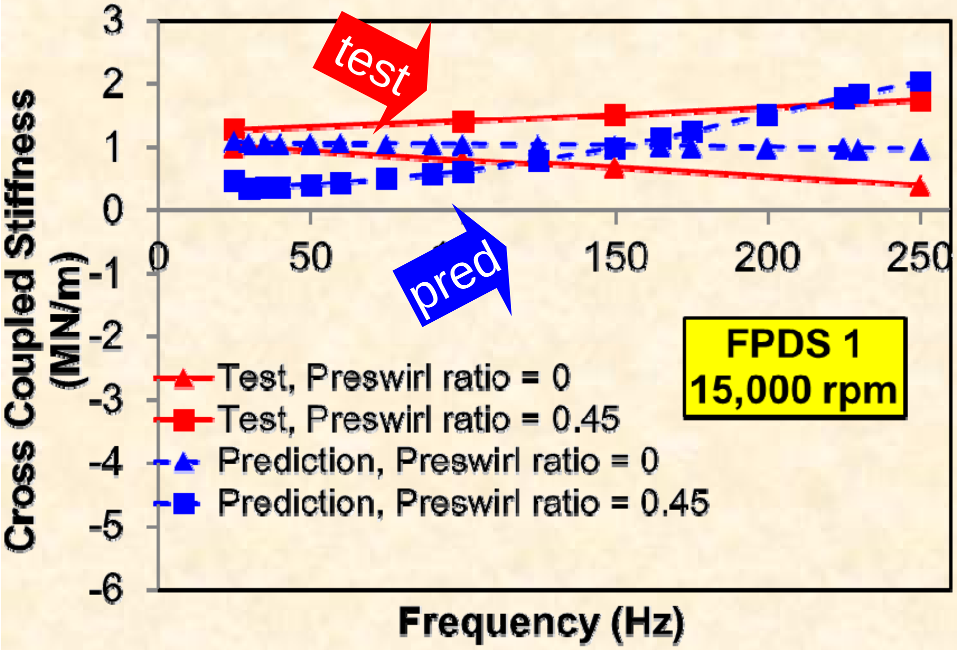
PDSeal© predicts well LS damping & gives too little damping for FPDS

Ertas, B.H., Delgado, A., Vannini, G., 2012

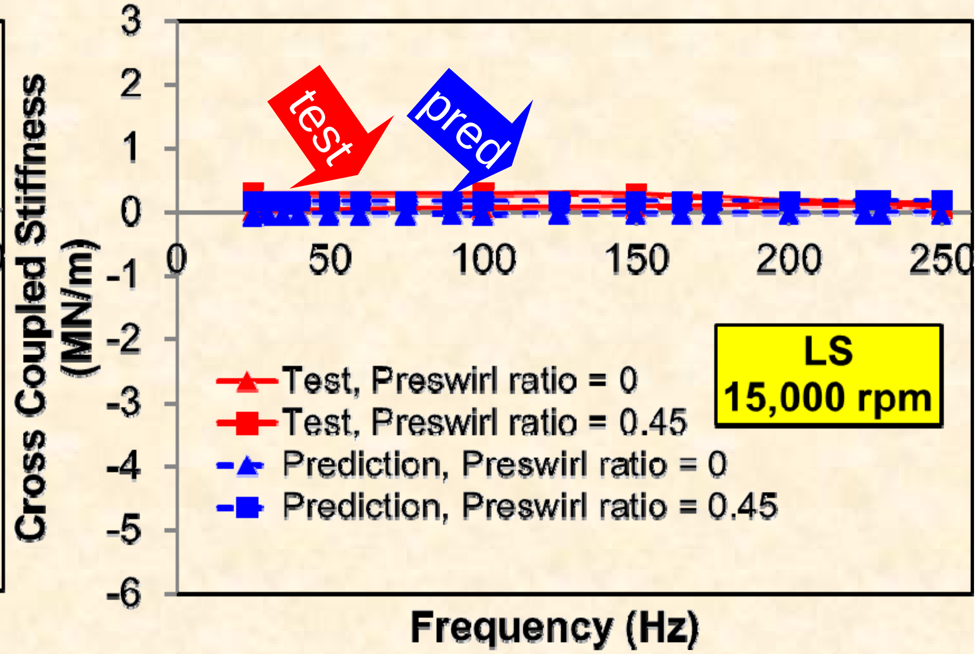
Cross Coupled Stiffness

rotor speed 15 kpm
preswirl ratios=0 & 0.45

Fully partitioned pocket damper seal



Labyrinth seal



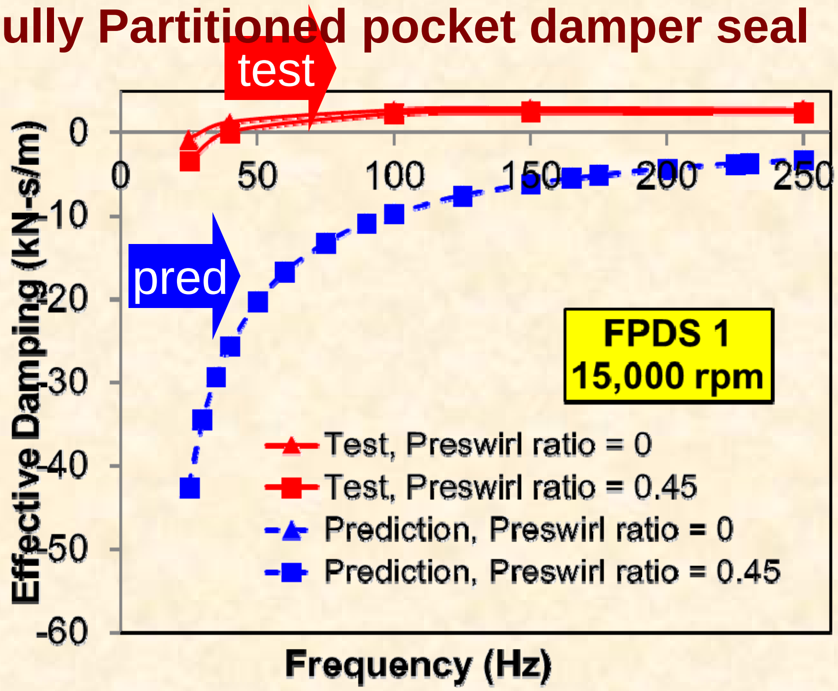
PDSeal© predicts well cross stiffness for both seals

Ertas, B.H., Delgado, A., Vannini, G., 2012

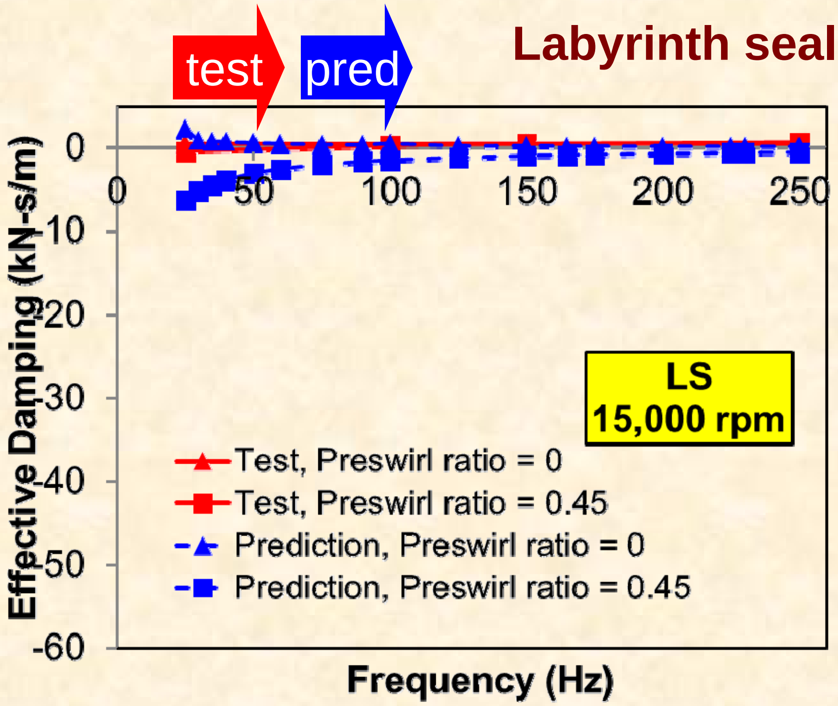
Effective Damping

rotor speed 15 kpm
preswirl ratios=0 & 0.45

Fully Partitioned pocket damper seal



Labyrinth seal



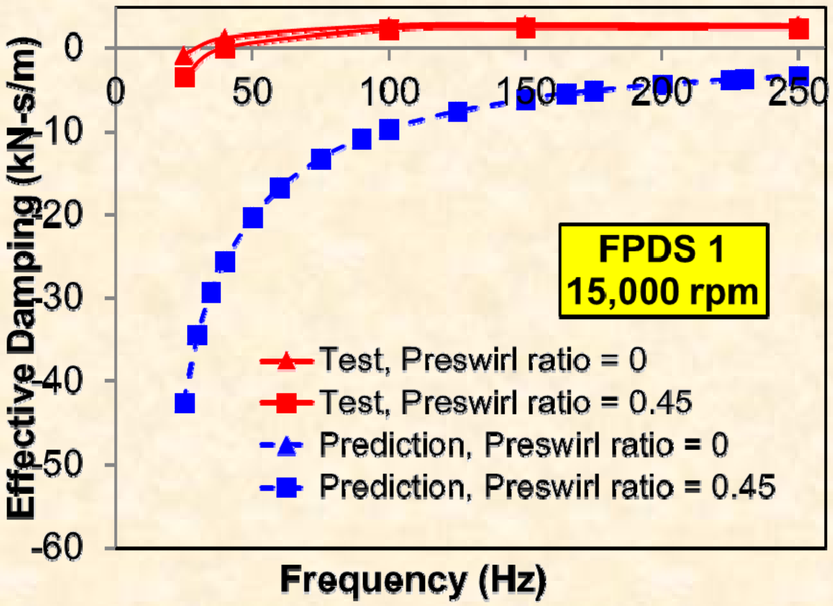
$$C_{eff} = C - \frac{k}{\omega}$$

PDSeal© does a poor job in predicting the effective damping of a FPDS

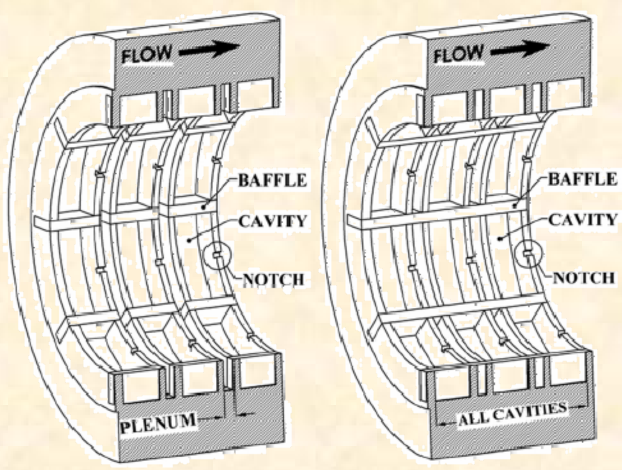
Ertas, B.H., Delgado, A., Vannini, G., 2012

Conclusions

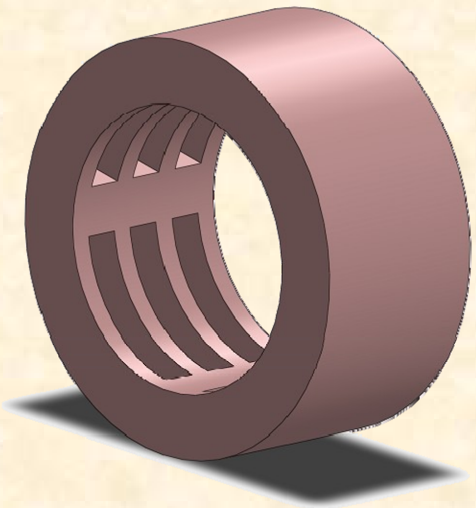
Predicted effective damping for FPDS is distinct from test data.



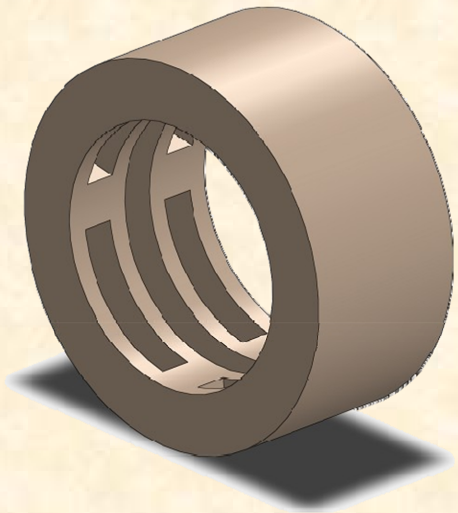
PDSeal© needs to be improved for better prediction for FPDS with thick walls.



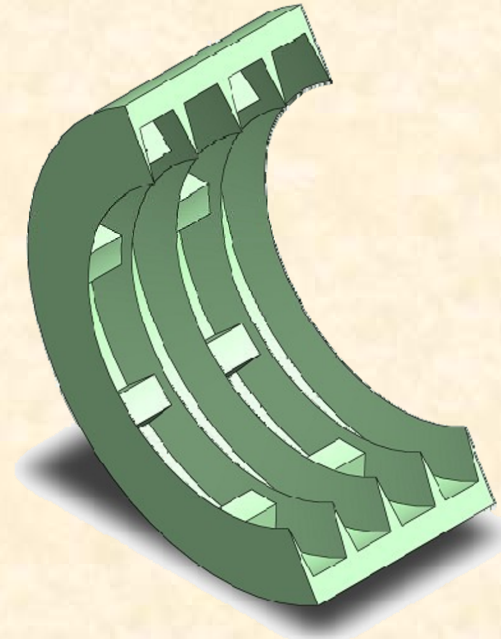
Why the differences?



FPDS with 4 pockets and 3cavities



PDS with 4 pockets and 3cavities



Original model of PDS with sharp teeth in TAMU PDS code

PDSeal© does not consider axial thickness of the partition walls

Engineering Analyses for Pocket Damper Seals and Combined Labyrinth-Brush Seals

Luis San Andrés
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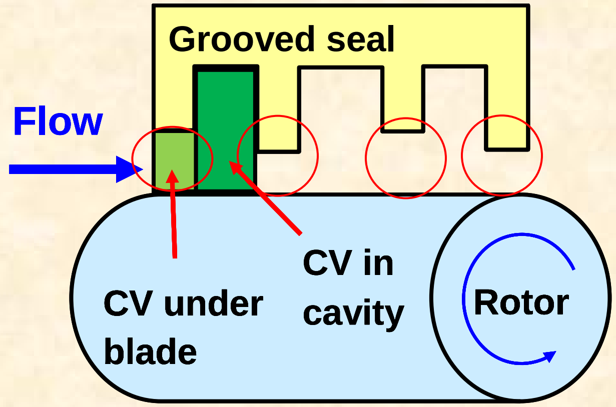
Weilian Shan
Graduate Research Assistant

May 2013

Proposed work 2013-2014 Year II

- **Update bulk-flow flow model for PDS and FPDS.**
- **Model will include real gas properties including supercritical CO2 and steam.**
- **Perform more code calibrations: compare predictions to test data for leakage and force coefficients.**
- **Begin extensions of the model to include two-component mixtures (liquid and gas).**

Model PDS as a grooved seal



Continuity equation

Replaces empirical leakage equation

$$\frac{L}{\xi_r}\frac{\partial(dP)_i}{\partial t}+\frac{\partial(HPV)_i}{\partial z}+\frac{\partial(dPU)_i}{R_r\partial\theta}=0$$

Circumferential momentum equation

$$-\frac{1}{R_r}\tau_{\theta,i}-\frac{\partial(dLP)_i}{R_r\partial\theta}=\frac{1}{ZR_gT}\left[\frac{1}{\xi_r}\frac{\partial(dLPU)_i}{\partial t}+\frac{\partial(dLPU^2)_i}{R_r\partial\theta}+\frac{\partial(HLPUV)_i}{\partial z}\right]$$

Axial momentum equation

Considers blade thickness

$$-\left[\frac{1}{R_r}\tau_{z,i}+\frac{\partial(HLP)_i}{\partial z}\right]=\frac{1}{ZR_gT}\left[\frac{1}{\xi_r}\frac{\partial(dLPV)_i}{\partial t}+\frac{\partial(HLPV^2)_i}{\partial z}+\frac{\partial(dLPUV)_i}{R_r\partial\theta}\right]$$

Kim, C. H., Childs, D. W., 1987

TRC Budget 2013-2014 Year II

	Year II
Support for graduate student (20 h/week) x $ 1,950 x 12 months	$ 23,400
Fringe benefits (0.6%) and medical insurance ($185/month)	$ 2,360
Travel to (US) technical conference	$ 1,200
Tuition & fees three semesters ($362/credit hour x 24)	$ 8,686
Others (Mathcad® and portable data storage)	$ 220
Total Cost:	**$ 35,866**

Year 2: Develop computational models for predictions of leakage, drag power and force coefficients of FPDS, and combined labyrinth-bush seals for gas and steam turbines

Thank you !

More information at:

http://rotorlab.tamu.edu