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RANS and LES of Turbine Heat Transfer

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Our research group's focus is on turbine cooling.
Current efforts include:

- **Perform**
  - LES of ingress/egress into the wheelspace between the rotor and stator disks to understand seal design.
  - *LES of internal & film cooling aimed at generating statistics to understand physics and to guide development of RANS models.* ***POSTER***
- **Develop**
  - BC for LES and BC at the interface between RANS and LES for hybrid methods.
  - Reduced-order methods from CFD for system-level tools.
- **Examine**
  - Fundamental issues in computing & measuring heat-transfer relevant to turbine cooling: scaling of data measured in the lab (near 1 atm & room T) to engine conditions (high T & P).

Outline of Talk

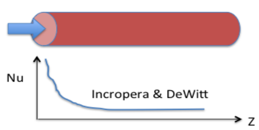
- **Scaling of Heat-Transfer Data**
- RANS, URANS, LES of Ingress-Egress into Rim Seals
- Q&A

Design of Experiments - Scaling: Nu = f(Re, Pr, T_w/T_b, ...)

Convective heat transfer from a surface is often described by Newton's law of cooling:

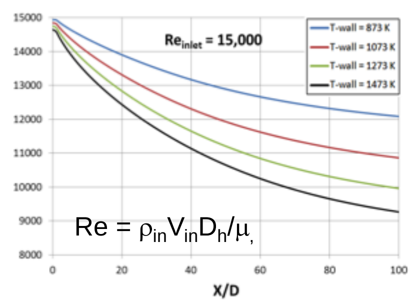
$$q'' = h\,(T_{wall} - T_b) \qquad T_b = \frac{\int_A \rho u C_p T\,dA}{\int_A \rho u C_p\,dA}$$
$$Nu = hD_h/k$$

All physics dumped into **h**, but $\mathbf{T_b}$ must be right!



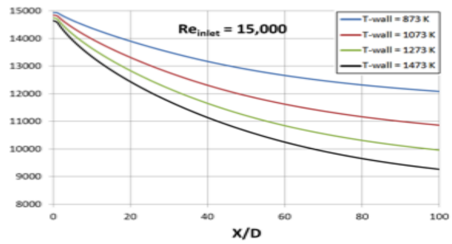
In turbine cooling,
- T_b can vary appreciably along the cooling duct → $T_w/T_b > 2$ because q" can be as high as 400 W/cm^2.

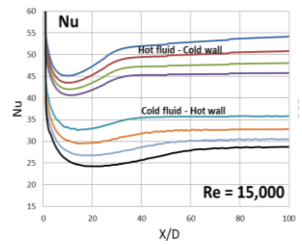
Thus,
- the local Re & Pr and hence Nu can vary appreciably along the duct



Design of Experiments - Scaling: Nu = f(Re, Pr, T_w/T_b, ...)

Currently, almost all experiments that measure the HTC are conducted at T_w/T_b near unity because realistic temperatures are hard to reproduce in the lab.





Right now, we all assume that Nu = f(Re, Pr, T_w/T_b, geometry, entrance region).

Since T_w/T_b and hence Re and Pr can vary appreciably along the cooling duct, should those variations be reproduced in the experimental set up in addition to Re, Pr, and T_w/T_b when measuring the HTC?

That is should Ld(Re)/dx and d(Pr)/d(x/L) be added to Nu = f(Re, Pr, T_w/T_b, geometry, x/L)?

Is this a reasonable question? In hypersonics, chemical kinetics, …, the rate is important

Design of Experiments - Scaling: Nu = f(Re, Pr, T_w/T_b, ...)

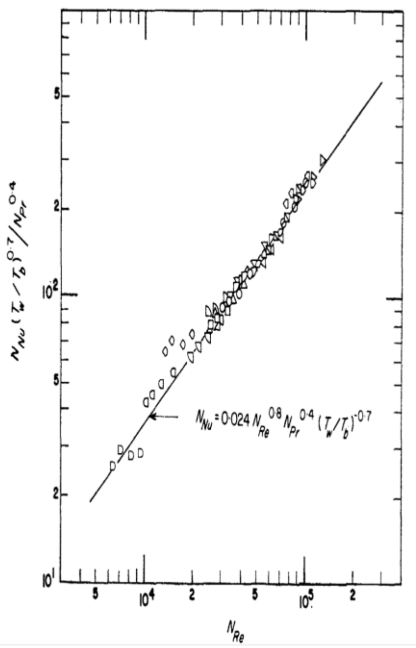
Literature Review:

Perkins & Worsoe-Schmidt, Int'l J. of Heat & Mass Transfer, Vol. 8, 1965, pp. 1011-1031:

- straight tube with circular cross section and L/D = 160 (vertical tube & upward flow)
- T_w/T_b up to 7, Re = 18,300 to 279,000
- provided local HTC based on 1-D analysis along duct to get T_b

They found that regardless how much T_w/T_b changed along the duct, all data collapse to a straight line except near the duct inlet and if Re is sufficiently high and the actual T_b is used (i.e ., no approximations to the T_b).

Thus, Nu = f(Re, Pr, T_w/T_b, geometry, entrance region) is correct - at least for circular ducts without features (e.g., ribs, pin fins, …).



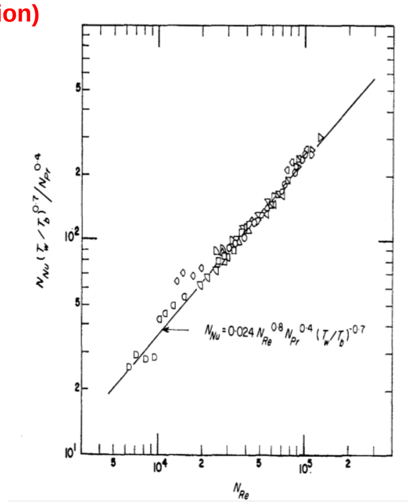
Design of Experiments - Scaling: Nu = f(Re, Pr, T_w/T_b, ...)

What does
Nu = f(Re, Pr, T_w/T_b, geometry, entrance region) mean to an experimentalist?

It means we can assume local thermodynamic equilibrium to hold for heat transfer so that **Nu = f(Re, Pr, T_w/T_b, geometry, entrance region)** is effectively an equation of state that depends only on the geometry and distance from the entrance region.

Thus, if
- The experimental setup is geometrically similar
- Re, Pr, T_w/T_b are the same

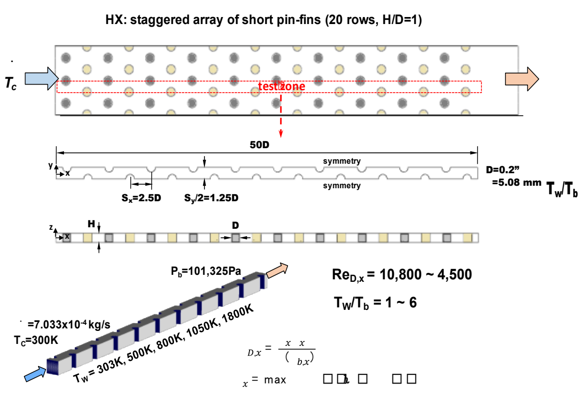
then
- $Nu_{exp} = Nu_{\text{real engine conditions}}$ except in the entrance region

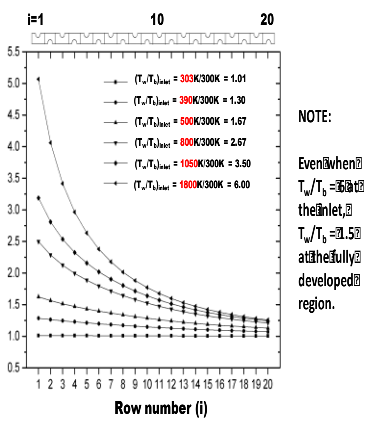


Since most measurements of HTC are made at T_w/T_b near unity, how to scale from T_w/T_b = 1 to T_w/T_b > 1?

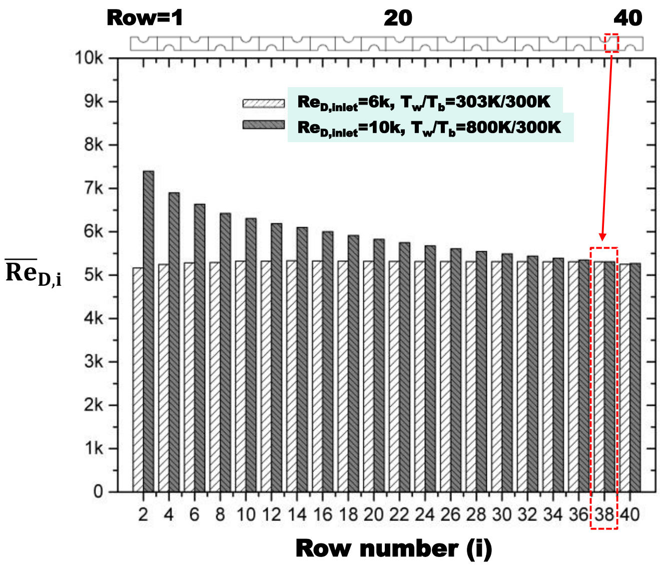
Design of Experiment / Scaling: How to scale lab to real?

How scale measurements of HTC at lab conditions (T_w/T_b = 1) to engine conditions (T_w/T_b >> 1)?



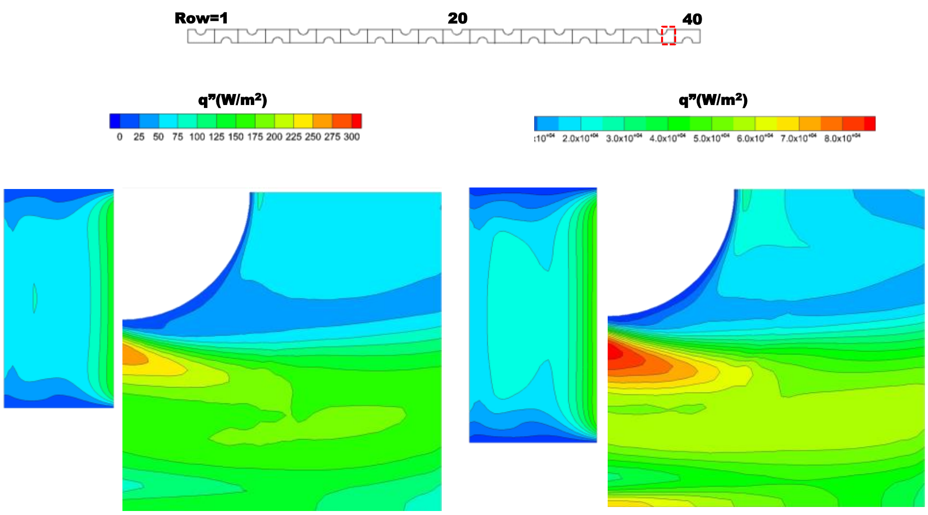


Zonally Averaged $Re_{D,i}$
(Area behind Pinfins, i=2, 4, …, 40 only)



For i=38:
$\overline{Re}_{D,i=38}$=5.3K

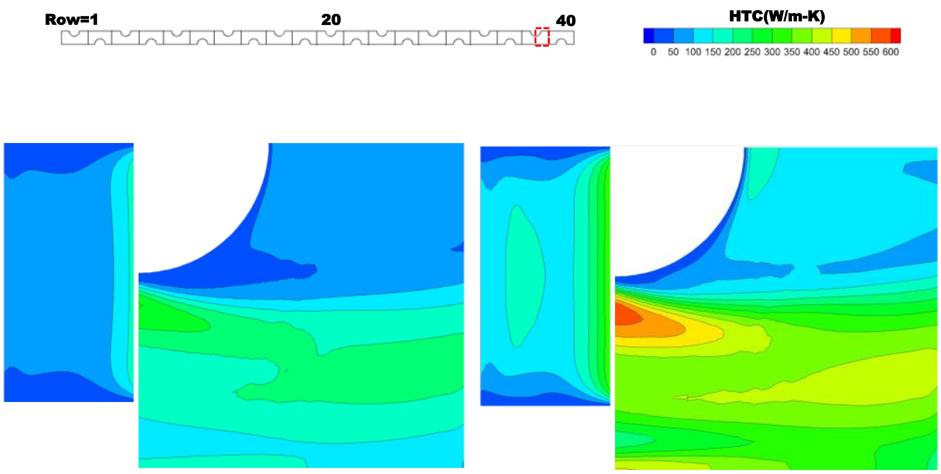
Local q” – One Patch (i=38)



$\overline{Re}_{D,i=38}$=5.3k, T_w/T_b=303K/300K

$\overline{Re}_{D,i=38}$=5.3k, T_w/T_b=800K/300K

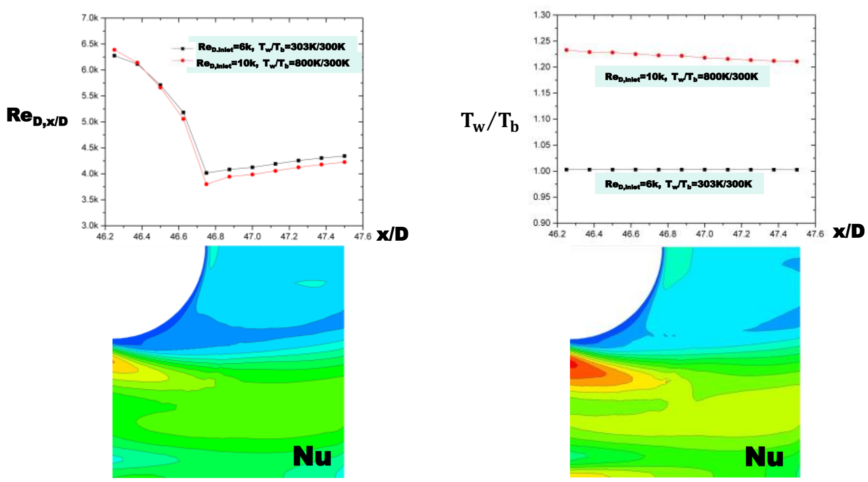
Local HTC – One Patch (i=38)



$\overline{Re}_{D,i=38}$=5.3k, T_w/T_b=303K/300K

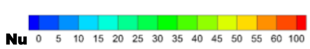
$\overline{Re}_{D,i=38}$=5.3k, T_w/T_b=800K/300K

Local Nu – One Patch (i=38)



$\overline{Re}_{D,i=38}$=5.3k, T_w/T_b=303K/300K

$\overline{Re}_{D,i=38}$=5.3k, T_w/T_b=800K/300K



Design of Experiments - Scaling: Nu = f(Re, Pr, T_w/T_b, ...)

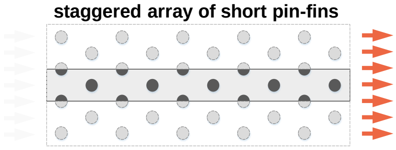
- Most previous experimental studies of the flow and heat transfer in channels with pin fins have focused on time-averaged data (**Metzger, Simoneau, Van Fossen, Chyu, Siw, Ames, Lawson, Ostanek, Thole, Ekkad, …**).
- Thus, most CFD studies have been based on RANS (**Delibra, Chi, Shih, Li, Jiang, Ligrani, …**).

Recently

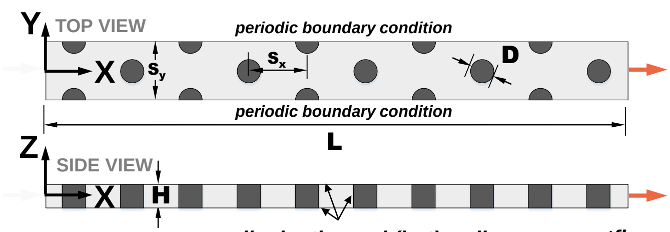
- **Ames & Dvorak (2006), Ames et al. (2005, 2007), Ostanek & Thole (2012)** measured the unsteady flow and heat transfer induced by the shedding of the wakes behind the pin fins.
- **Delibra et al. (2008, 2009), Paniagua et al. (2014)** reported URANS and LES studies to assess the capabilities of turbulence models for predicting the unsteady flow and heat transfer in a short pin-fin array.

Problem Description

staggered array of short pin-fins



H/D=1 (short pin-fins)
L/D=25
S_x/D=2.5 (streamwise pin spacing)
S_y/D=2.5 (spanwise pin spacing)
D=5.08mm (0.2")



no-slip, isothermal (hot) wall

cooling air inflow

$\dot{m} = 0.00636$ kg/s
$T_c = 400^o$C
$Re_D = 25{,}000$

Case	T_w (hot wall temp,°C)	T_w/T_c
1	420	1.03
2	1,000	1.89
3	1,500	2.63
4	2,000	3.38

outflow

$P_b = 25\ bars$

Formulation / Numerical Method of Solution

Code: ANSYS-Fluent v16.2

Formulation for Gas Phase:

- unsteady, compressible, thermally perfect gas with k(T), C_p(T), and μ(T)
- ensemble averaged continuity, momentum, and energy eqs. closed by the SST model

Algorithm:

- 2^{nd}-order accurate in time
- PISO with 2^{nd}-order upwind for the advection terms & central for diffusion terms

Convergence Criteria for Unsteady Solutions:

iterate until converged at each time step - normalized residual at the end of each time step for continuity< 10^{-5}, momentum< 10^{-5}, and energy<10^{-7}, turbulence quantities < 10^{-5}.

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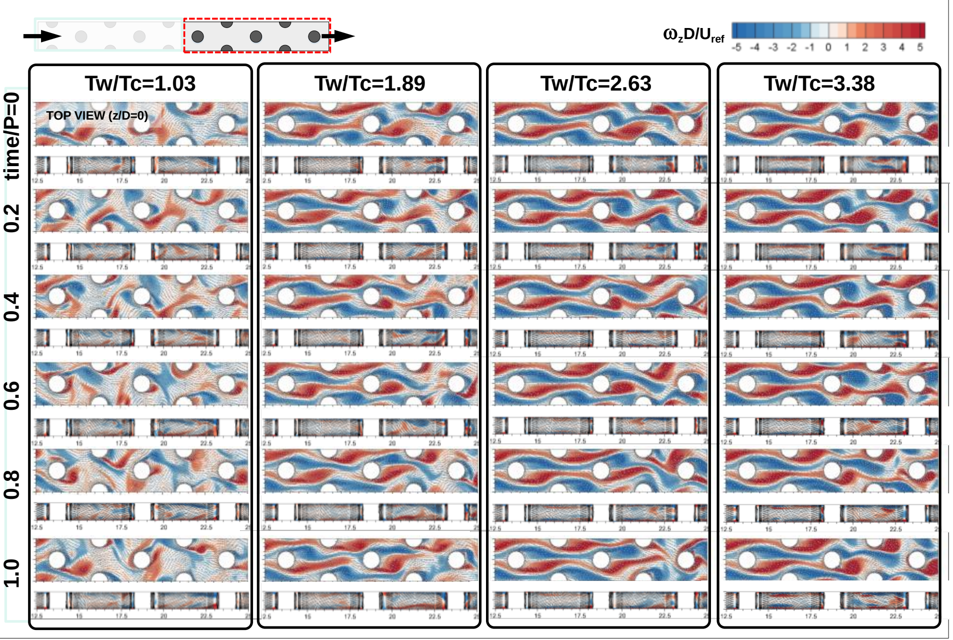
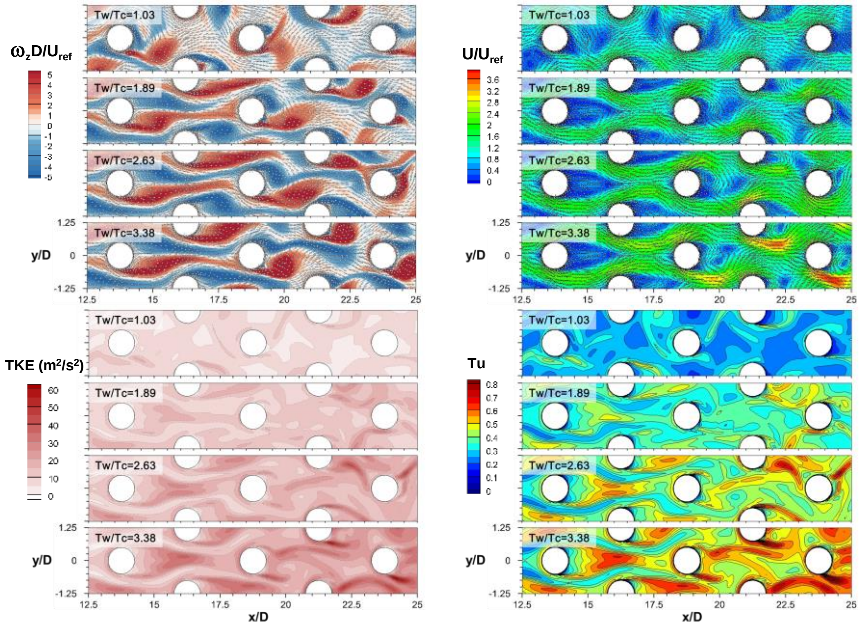
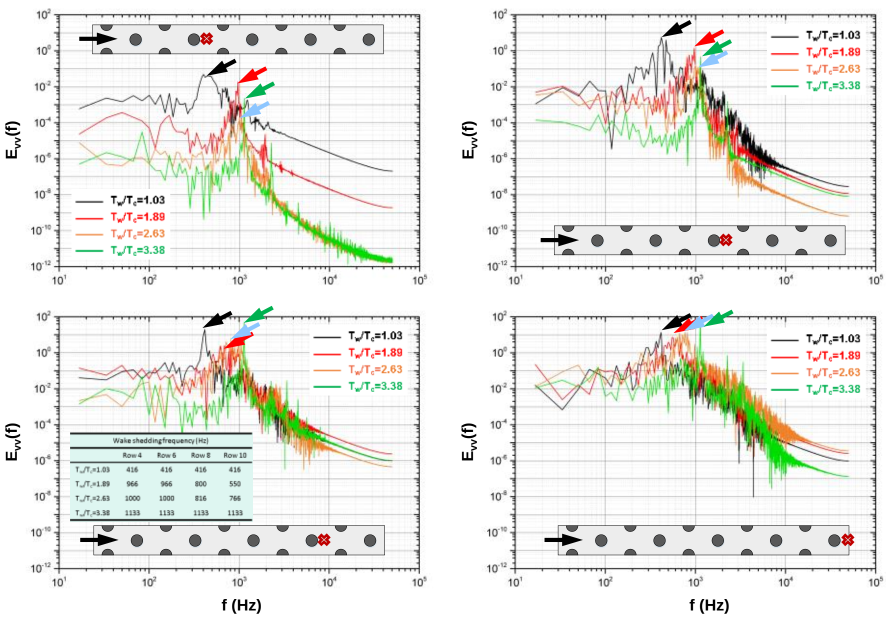
Evolution of Periodic Vorticity Shedding



Nature of Unsteady Flow

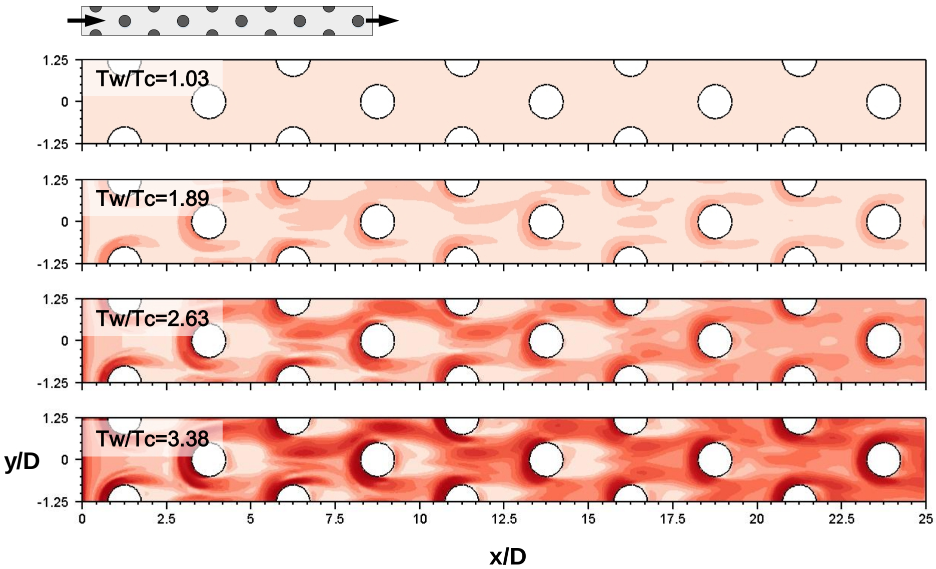


Vortex Shedding Energy Spectrum

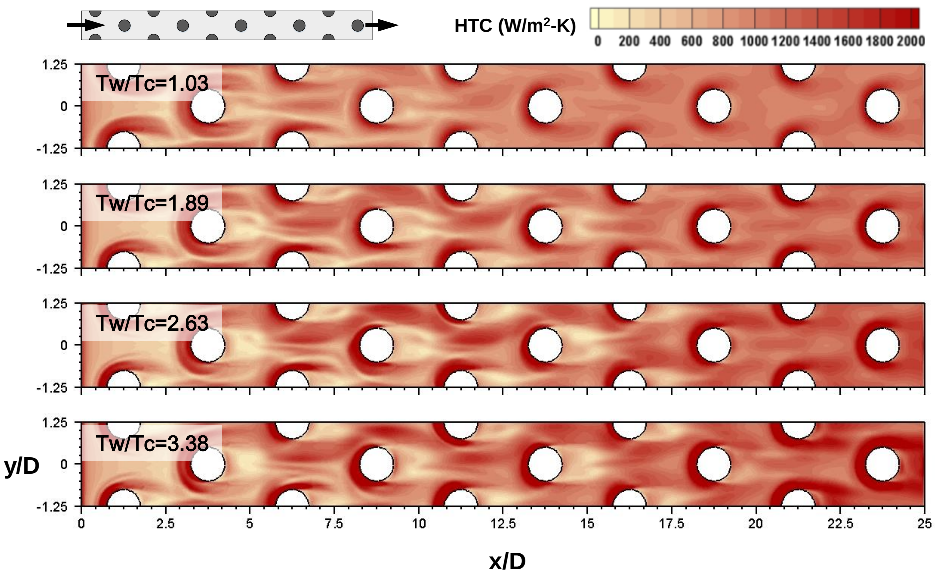


Wake shedding frequency (Hz)				
	Row 4	Row 6	Row 8	Row 10
T_w/T_c=1.03	416	416	416	416
T_w/T_c=1.89	966	966	800	550
T_w/T_c=2.63	1000	1000	816	766
T_w/T_c=3.38	1133	1133	1133	1133

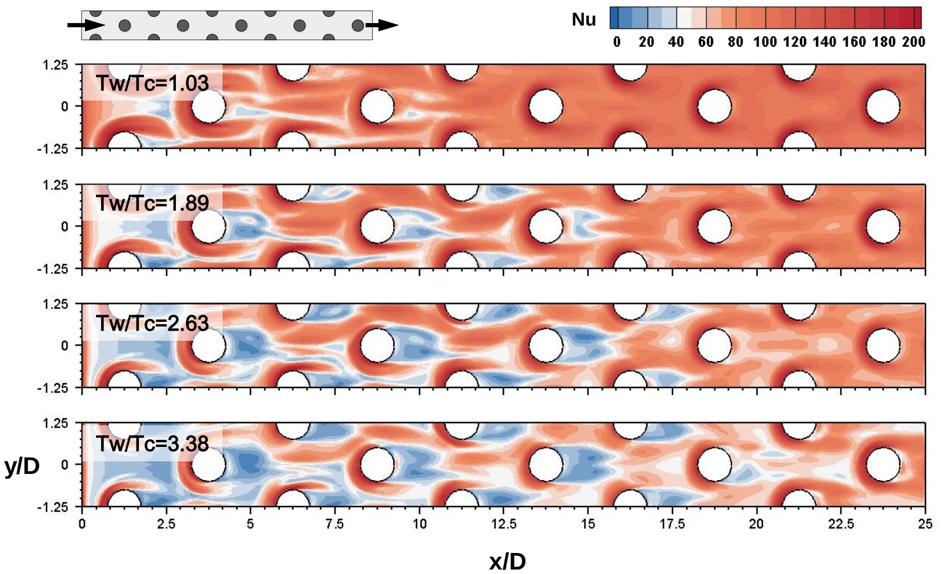
Mean Endwall Heat Flux



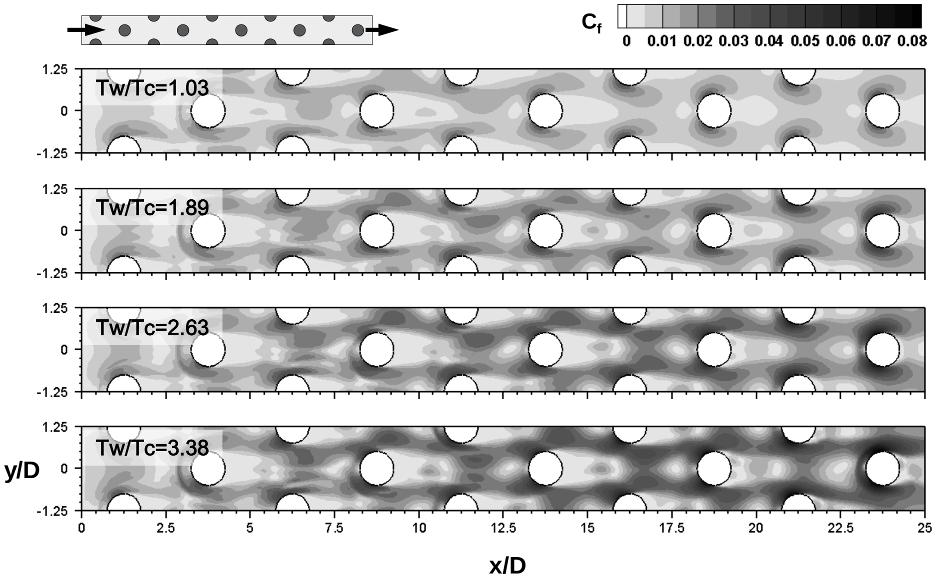
Mean Endwall HTC



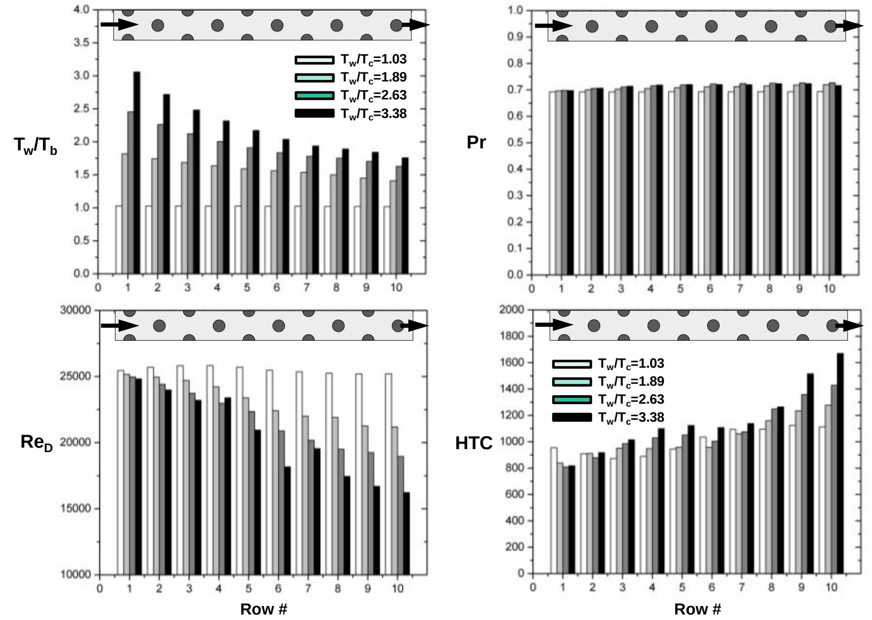
Mean Endwall Nu



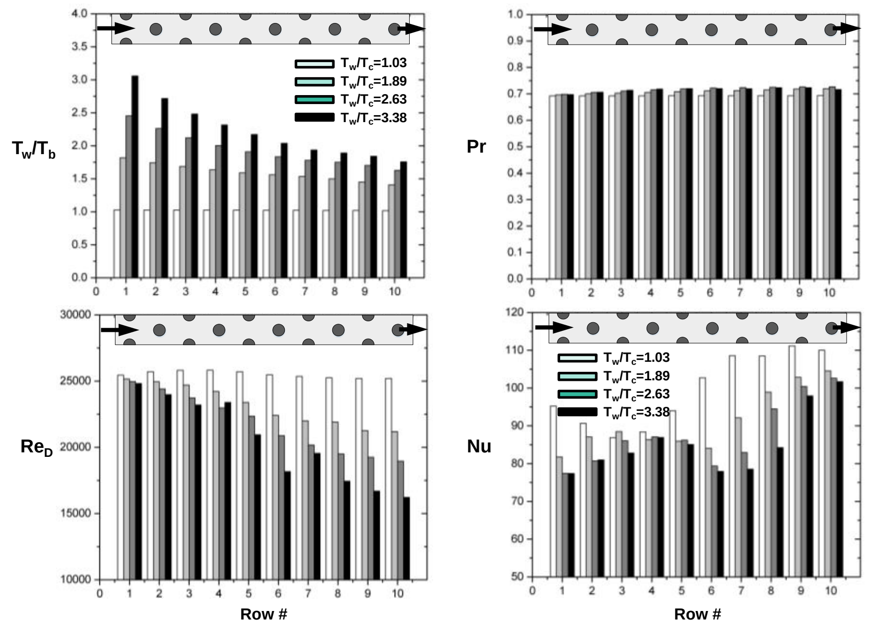
Mean Endwall Friction Coefficient (C_f)



HTC


T_w/T_b
Pr
Re_D
HTC
T_w/T_c=1.03
T_w/T_c=1.89
T_w/T_c=2.63
T_w/T_c=3.38
Row #
Row #

Nu


T_w/T_b
Pr
Re_D
Nu
T_w/T_c=1.03
T_w/T_c=1.89
T_w/T_c=2.63
T_w/T_c=3.38
Row #
Row #

Outline of Talk

- Scaling of Heat-Transfer Data
- **RANS, URANS, LES of Ingress-Egress into Rim Seals**
- Q&A

RANS, URANS, and LES of Ingress/Egress into Rim Seals

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Outline

- Introduction
- Objective
- Problem Description
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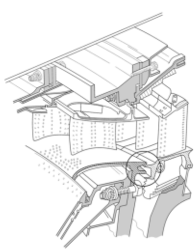
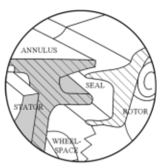
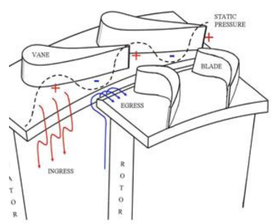
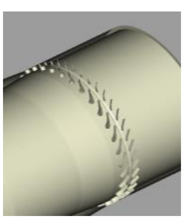
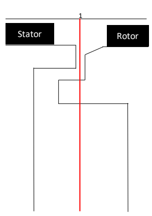
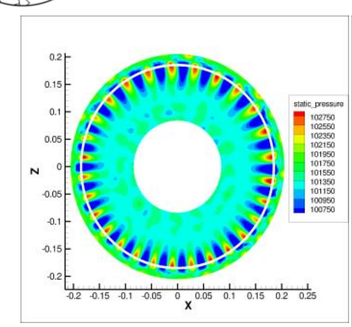
Ingress & Egress from Rim Seals

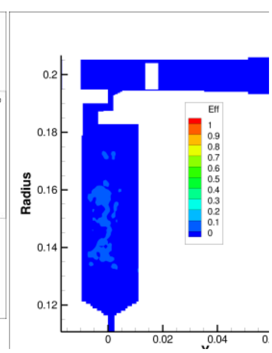
Hot gas can enter the gap between rotor and stator disks because of the high pressure







Many Previous Experimental Work on Seals

Experimental Studies:

- University of Bath: Owens, Lock, Sangan (2011, Bath
- General Electric Global Research Center: Bunker et al. (2009, GE)
- Arizona State University: Roy, Zhou, Balasubramanian (2005, 2007, 2015)
- Ohio State University: Dunn, Green, … (2014, …)
- **Penn State University: Karen Thole ← DoE & P&W**

Computational Studies:

- Laskowski, et al. (2009, GE)
- Zhou, et al. (2011, Beihang University)
- Wang et al. (2012, Pratt & Whitney)
- Teuber (2013, Siemans)
- Lalwani (2014, Bath)
- Green, Dunn, et al. (2014, Ohio State)
- Mirzamoghadam, et al. (2015, Honeywell)
- Liu, Weaver, Shih, Sangan, & Lock (2015, Purdue & Bath)

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- **Introduction**
- **Objective**
- **Problem Description**
- **Formulation / Numerical Method of Solution / Code**
- **Results**
- **Summary**

Objective

- Assess LBM method in predicting
  - pressure distribution after stator
  - swirl in the wheelspace
  - ingress through the seal

  by comparing results against EXP data & RANS results from CFX.
- Investigate the effects of vanes and blades on ingress.

Mathematical Approach

- Powerflow: URANS → LES

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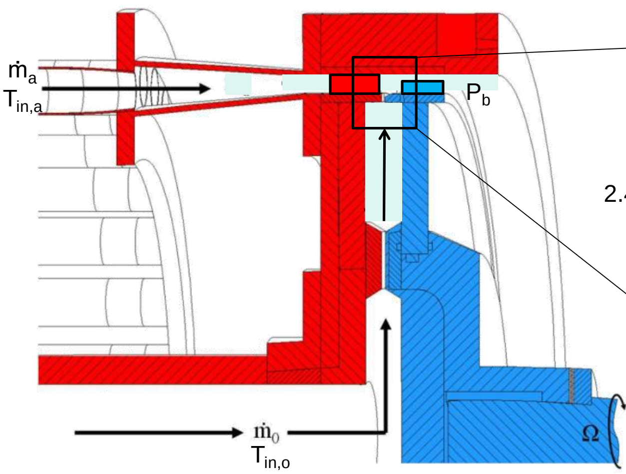
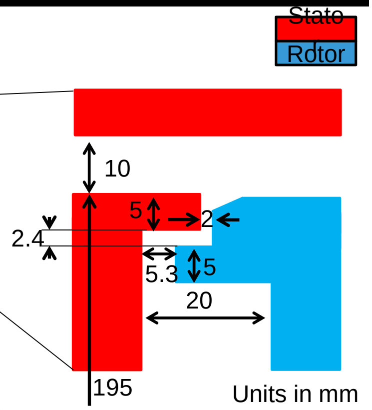
Problem Description - EXP

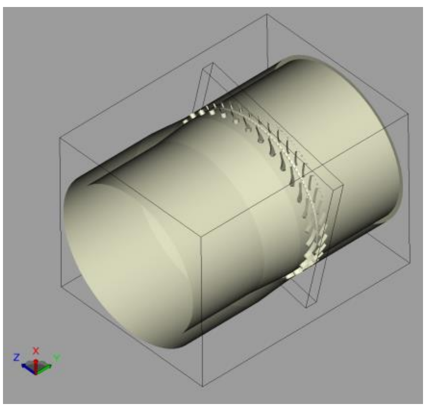


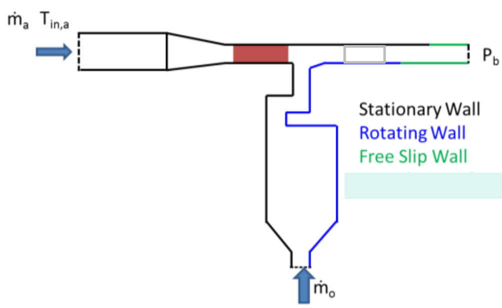


Geometry based on University of Bath experiment.
Rig consists of 32 stator vanes, 41 rotor blades
Seal is a radial seal (Bath Rig: GT2011-45310)

$\dot{m}_a$	340 g/s
$\dot{m}_o$	5.57 g/s (5% CO_2)
P_b	102,014 pa
Ω	2037 rev/min
$T_{in,\ a}$	23.34 C
$T_{in,\ o}$	23.34 C

Problem Description - CFD





Full 360 was simulated.
Instead of CO_2, the annulus flow was given a slightly higher temperature in order to track ingestion.

$\dot{m}_a$	340 g/s
$\dot{m}_o$	5.57 g/s
P_b	102,014 pa
Ω	2037 rev/min
$T_{in,\ a}$	28.34 C
$T_{in,\ o}$	23.34 C

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Formulation/Numerical Method of Solution/Code

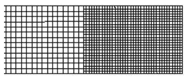
Code: Powerflow 5.1b

Formulation:
- Boltzmann Transport Equation
- Renormalized group form of the *k*-*ε* equations with proprietary extensions

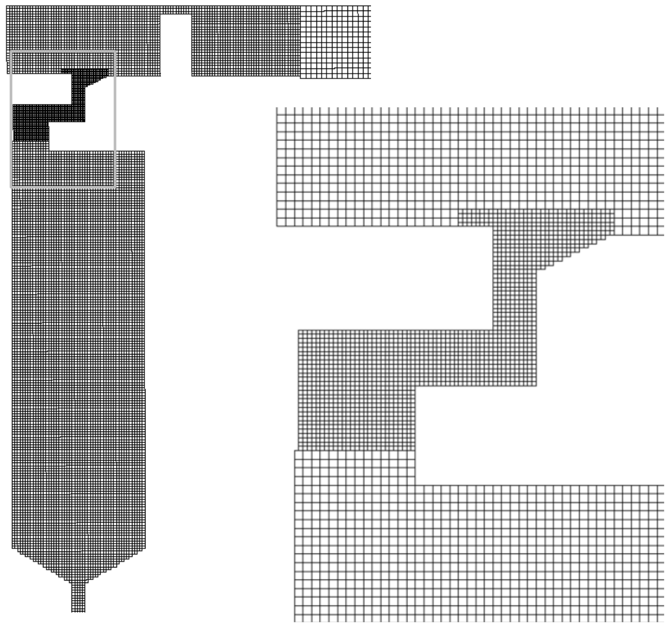
Algorithm:
- Lattice-Boltzmann method: $f_i(\vec{x} + \vec{c_i}\Delta t, t + \Delta t) = f_i(\vec{x}, t) + C_i(\vec{x}, t)$
- D3Q19 model: Discretizes the continuous velocity space $\vec{c}$ with a set of 19 discrete velocities in three dimensional space.
- Collision operator C is simplified using a patent pending “filter collison operator”

Convergence Criteria:
- compute until time-averaged
  - axial velocity in annulus converges and matches rotor-stator velocity triangle
  - pressure is steady in the annulus
  - Effectiveness is steady in the wheelspace

Mesh



Voxels:
- 30 million
- **42.8 million**
- 75 million (locally refined)



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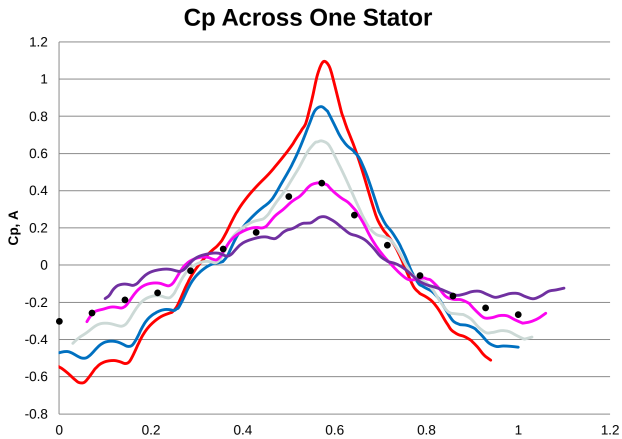
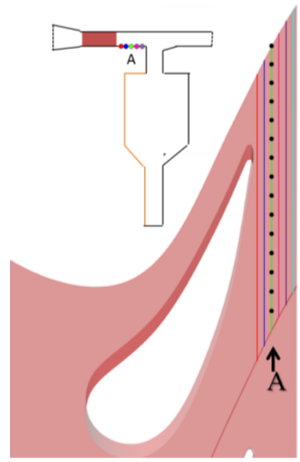
Pressure: CFX (RANS)

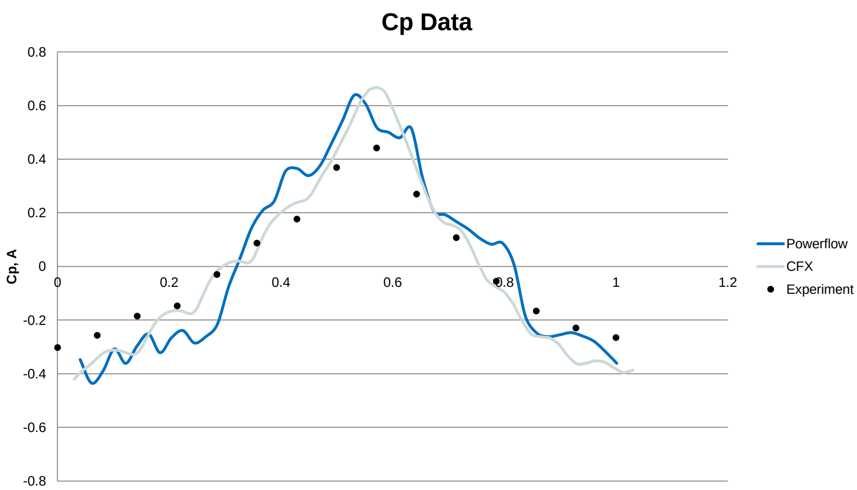
Pressure was measured and non-dimensionalized ($cp = \frac{p-\bar{p}}{\frac{1}{2}\rho\Omega^2 b^2}$). Cp is measured on the green line, marked A in the right figure and denoted as dots in the left figure. Cp was computed on all lines shown in the right figure and denoted as solid lines with the same color in the left figure. CFD results show (1) Cp to vary significantly downstream of vane and (2) Cp computed and measured match almost perfectly if location is 1 mm off.





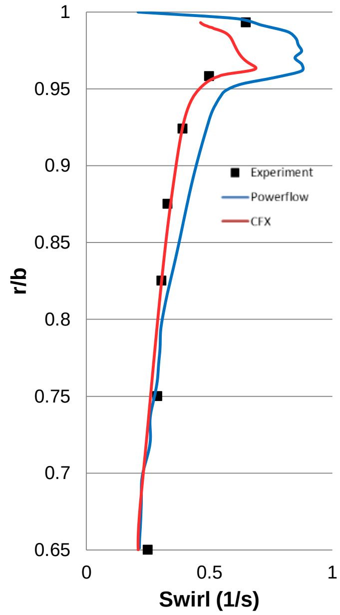
Pressure: CFX (RANS) vs Powerflow (LES)

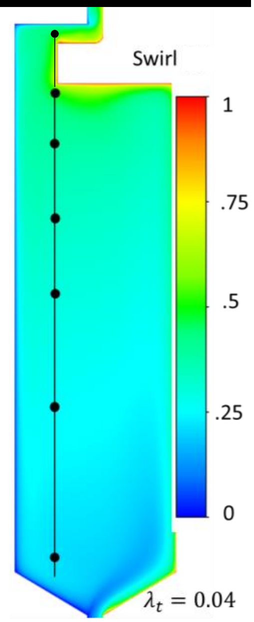
P from PowerFlow is averaged over 5 blade passes.
Min-Max predicted by PowerFlow is 8% more accurate than those predicted by CFX.



Swirl: CFX (RANS) vs Powerflow (LES)

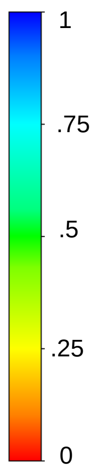
Powerflow matches the swirl measurements as well as CFX at lower radius. At higher radius, it starts to over-predict swirl. However, Powerflow predicts the highest radius point of swirl more accurately. CFX severely under-predicted ingress. This means that the high swirl fluid that would be ingested was not seen in CFX. Conversely, Powerflow better predicts ingestion, thus there is better agreement for the swirl near the seal.

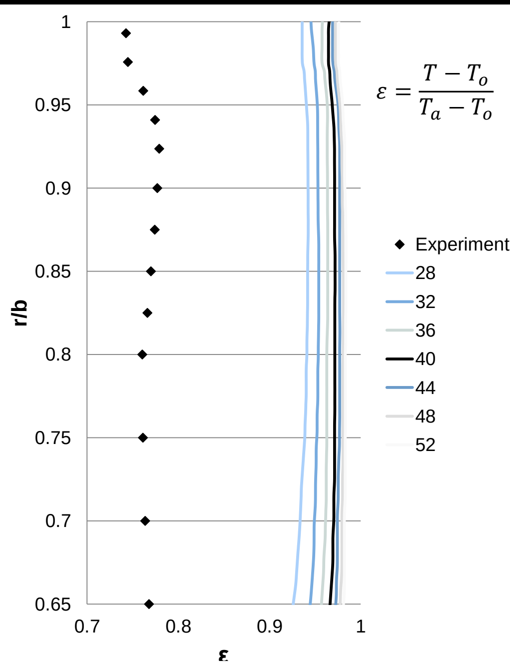




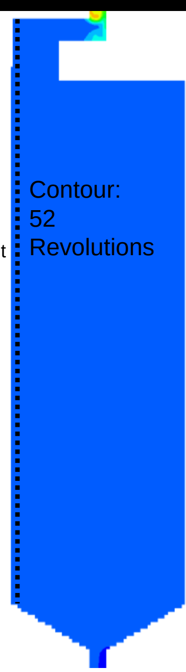
Effectiveness

Effectiveness lines plotted for revolutions 28 to 52.



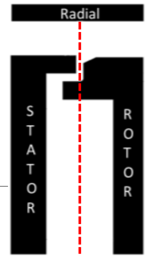


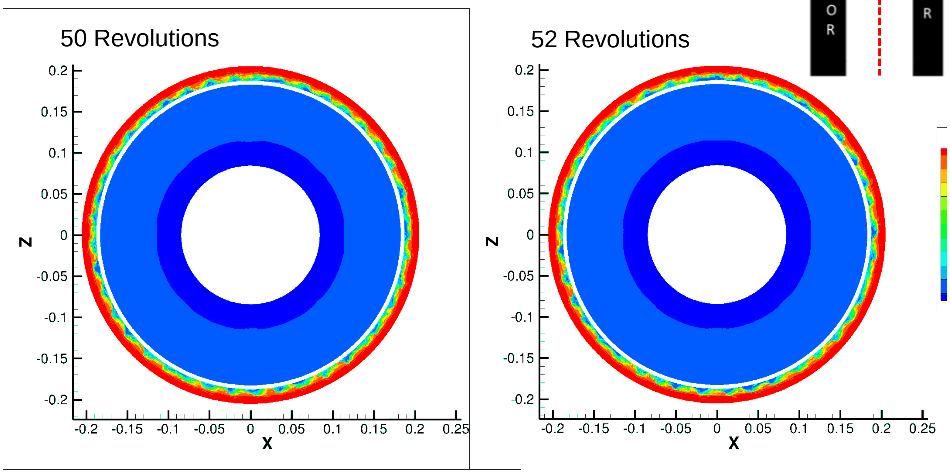
$$\varepsilon = \frac{T - T_o}{T_a - T_o}$$

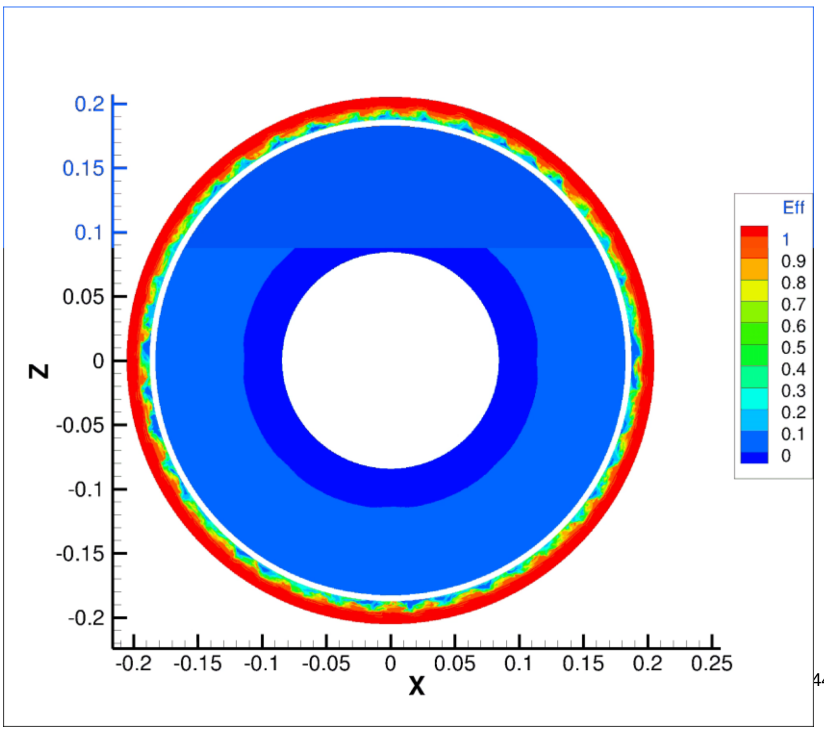


Effectiveness

Looking at an axial cut plane through the middle of the seal, the sinusoidal pattern on the temperature plots shows the initial ingestion that occurs at the seal. There are 41 times where the hot gas enters into the seal. This corresponds with the number of Rotor Blades, suggesting that the rotor blades drive the initial ingestion.

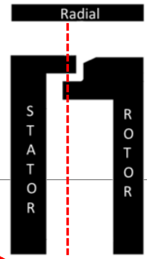


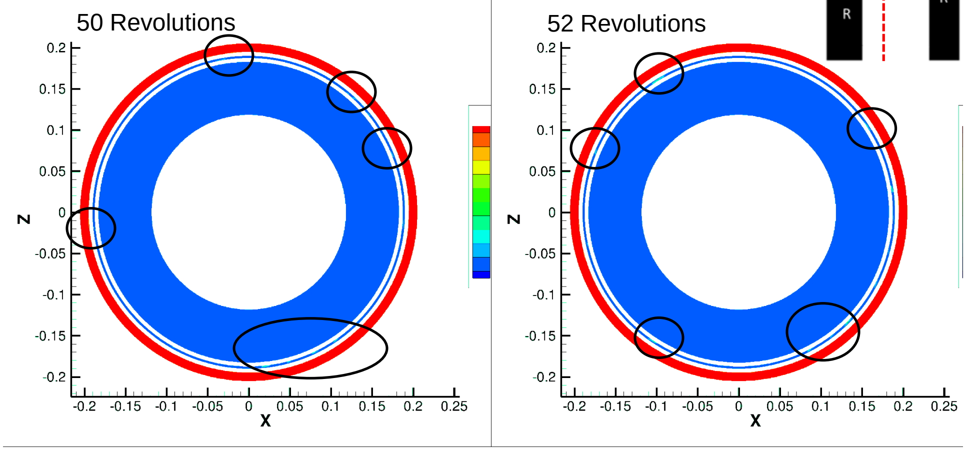




Effectiveness

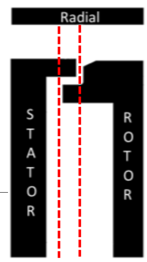
Looking at an axial cut plane through the bend of the seal, it visible that not all of the fluid that initially ingests makes it into the wheelspace. Only some of the hot fluid penetrates past the first bend. So while the number of rotor blades determines the initial ingestion points, the interactions between the rotor, stator and the rpm determine how far the hot fluid actually penetrates.

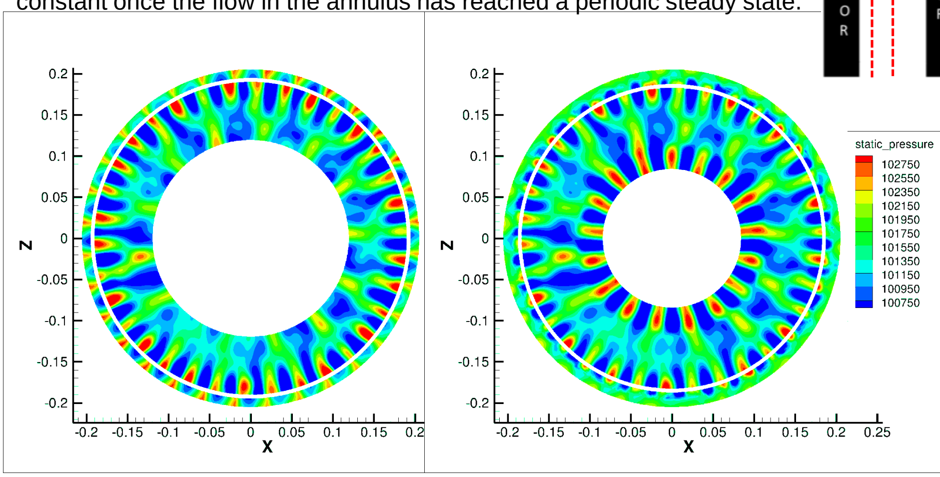




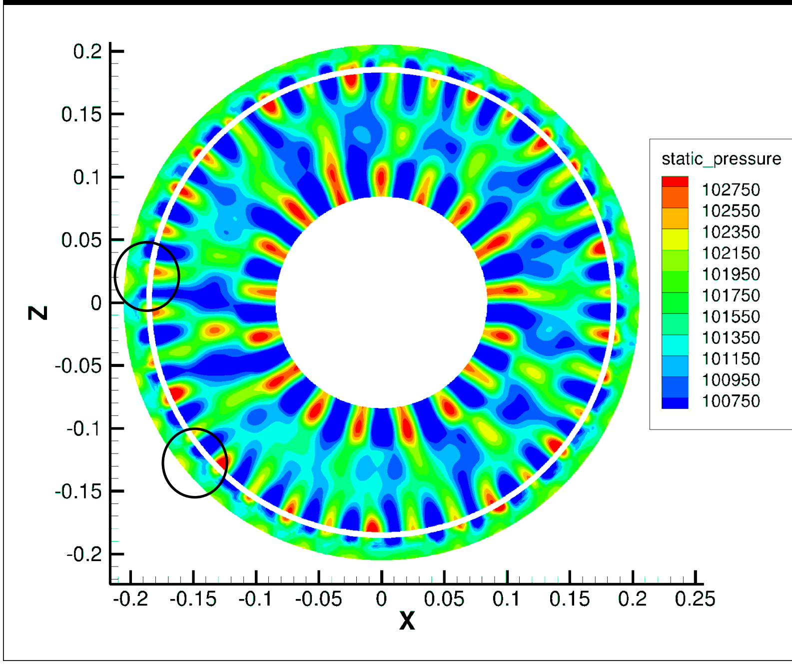
Pressure

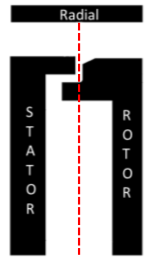
Looking at an axial cut plane 2 mm after the stator and a plane through the seal, it is clear that immediately after the blade, the pressure distribution is a result of the stator blades in the annulus; the 32 alternating high and low pressure spots in the annulus after the stator correspond with the 32 stator blades. However, counting the pattern at the seal, reveals a different number of pressure pairs, 34. This number at the seal occurs and remains constant once the flow in the annulus has reached a periodic steady state.





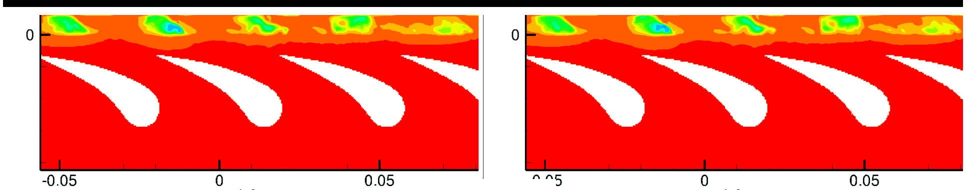
Pressure





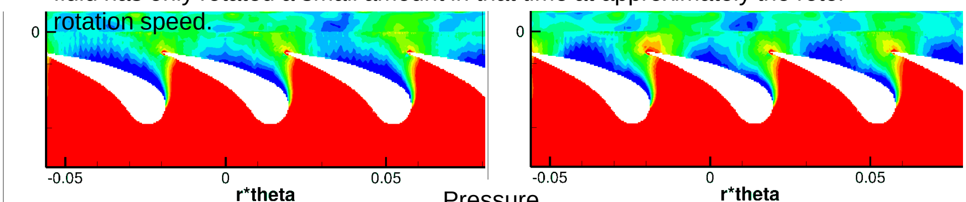
Because there is a mismatch in the number of high-low pressure pairs, there will be mismatches between the stator and the seal high-low pressure pairs.

Radial Contours



Temperature

The time difference between the frame on the left and the frame on the right is that 1/30 of a blade passing has gone by. Thus, the movement of the hot ingested fluid has only rotated a small amount in that time at approximately the rotor rotation speed.

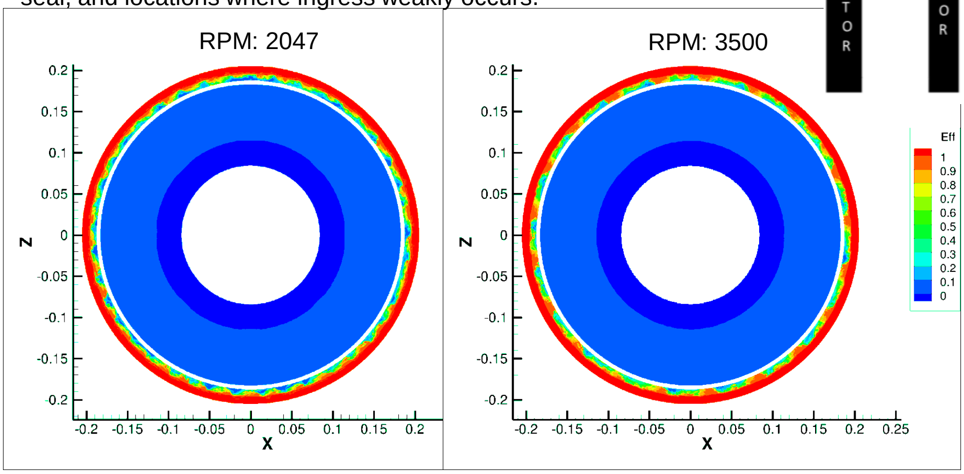


Pressure

However, on the stator, the pressure responds more quickly. For the same time, we see much greater pressure changes. In part, this is because the stator blades reduce the area and increase the velocity between the blades. Thus, the changes that occur are more rapid between the blades than at the seal.

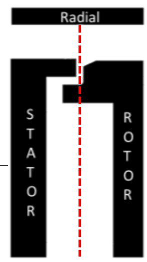
Effectiveness: Different RPMs

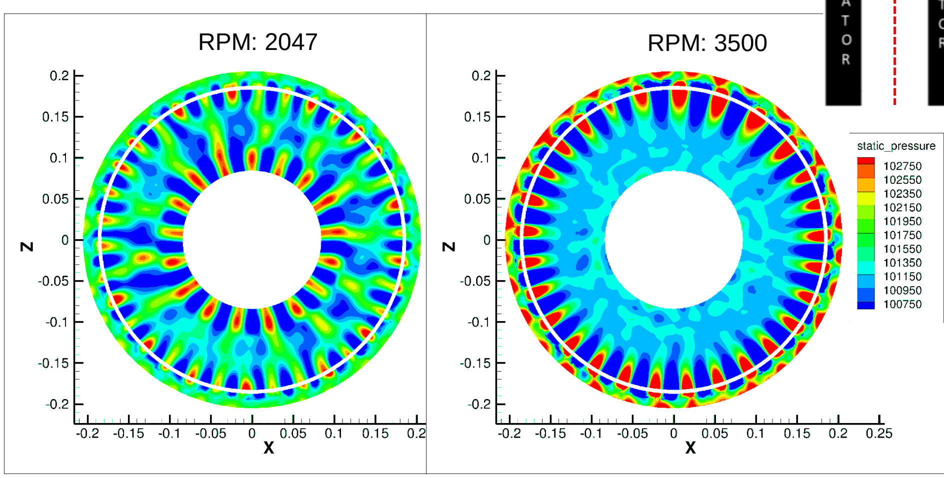
Below, the RPM was increased. The simulation done is still on design, with the same sealing flow rate. The spacing of the ingestion pattern suggests it is still a function of the rotor blades, as the angle between ingestion points corresponded with 41 blades. However, the penetration depth has changed, which is evident in both locations that penetrate fully through the seal, and locations where ingress weakly occurs.



Pressure: Different RPMs

The change in rpm also dramatically changes the pressure distribution over the seal. At the higher rpm, there are now 28 pressure pairs over the seal, as opposed to the 34 seen in the lower rpm case. Since the blade and vane numbers remain the same, this difference may be a contributing factor to the change in ingestion patterns seen previously.





Outline

- Introduction
- Objective
- Problem Description
- Formulation / Numerical Method of Solution / Code
- Results
- Summary

Summary

- A Powerflow simulation based on a Rotor-Stator case run at the University of Bath was completed.
- Powerflow was able to predict the pressure distribution as well, if not better, than CFX.
- Powerflow also better matched swirl in the wheelspace closest to the seal. It also matched well at lower radius, but had poorer matching between the seal and wheelspace inlet.
- The final ingested amount did not match the experiment. However, unlike the CFX steady simulations in the correct reference frame, Powerflow did show that hot fluid was ingested.
- Ingestion was seen to be initially a function of the rotor blades.
- However, how far the ingress penetrates into the wheelspace was a function of both the rotor and the stator.
- Across the seal, there is a number of high low pressure pairs that does not correspond to either the number of vanes or blades. This mismatch affects how far the hot gas penetrates into the wheelspace.
- The timescale of the pressure variations at the seal differs from the rotation speed, which makes predicting the ingestion pattern as a function of vane and blade relative locations difficult.
- At different rpms, the locations where the ingestion occurs can still be seen as a function of the rotor. However, the depth of penetration has changed. This difference is a consequence of the different pressure distribution that occurs at the seal.

Outline of Talk

- Scaling of Heat-Transfer Data
- RANS, URANS, LES of Ingress-Egress into Rim Seals
- **Q&A**