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Challenges in Measuring the HTC for Turbine Cooling

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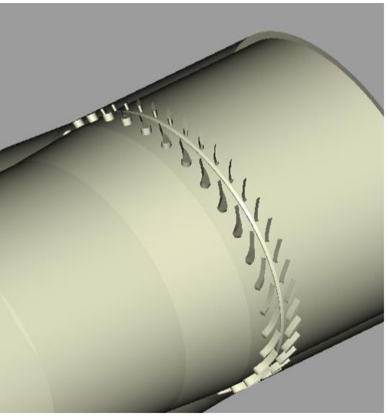
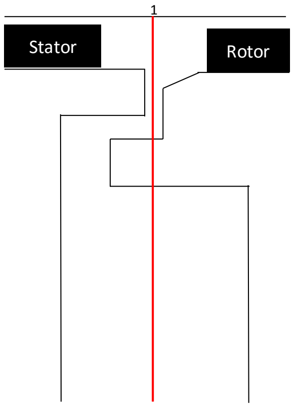
Our research group's focus is on turbine cooling.

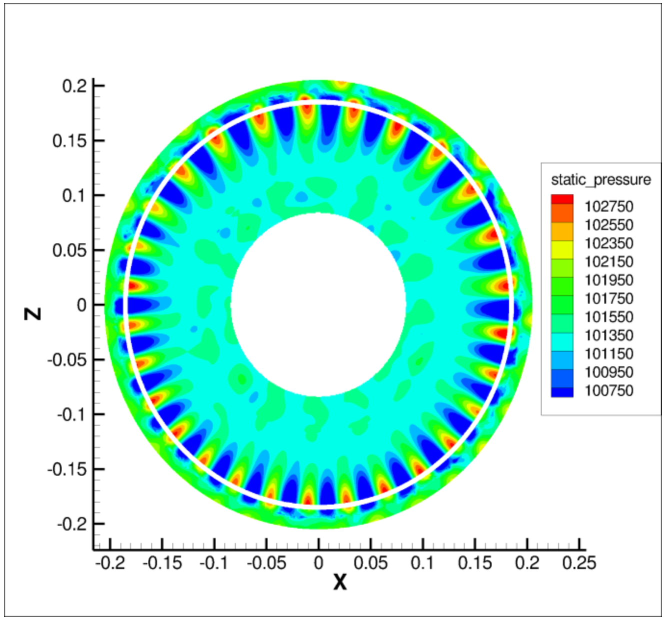
Current efforts include:

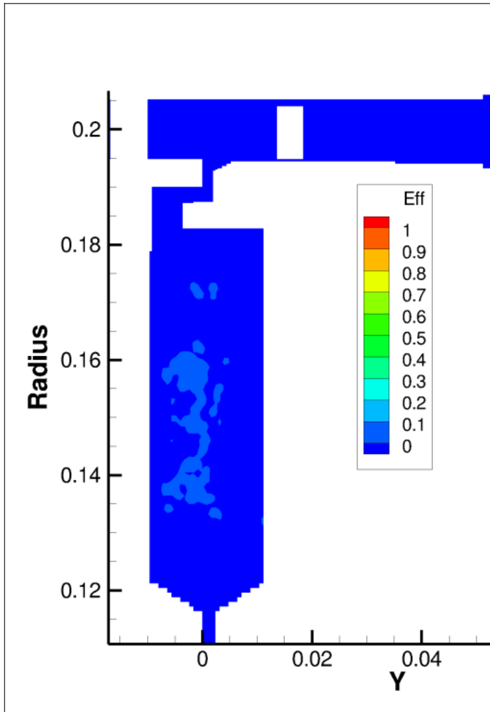
- LES of ingress/egress through rim seals that accounts for rotor-stator interactions.
- LES of internal & film cooling aimed at generating statistics to guide the development of RANS models.
- Inflow BC for LES and BC at the interface between LES and RANS for hybrid methods.
- Develop reduced-order methods from CFD for system-level tools.
- Examine fundamental issues in computing & measuring heat-transfer relevant to turbine cooling.

Liu & Shih









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Outline of Talk

Revisit

- HTC measured by transient methods
- HTC measured by steady-state methods

What else?

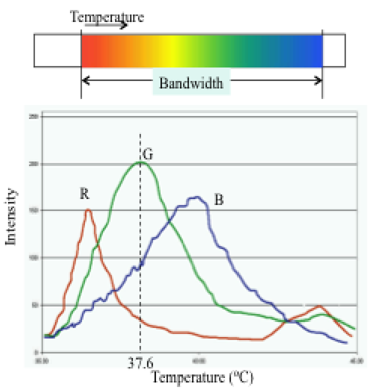
- Nu = F(Re, Pr, geometry)?

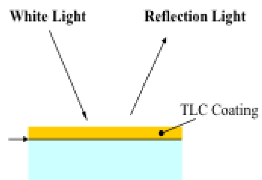
Summary & Current Work

Measurements of HTC by Transient Methods

Measurements are needed to **validate** CFD tools.

A widely used method is the **thermochromic liquid crystal technique**.





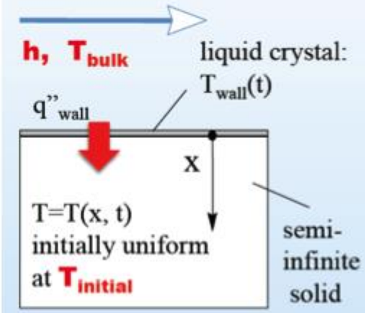
Thermochromic Liquid Crystal (TLC)

green light reflection gives the best intensity.

Surface temperature is measured as a function of time.

The heat transfer coefficient is then inferred from either 1-D or 0-D exact solutions.

1-D Model

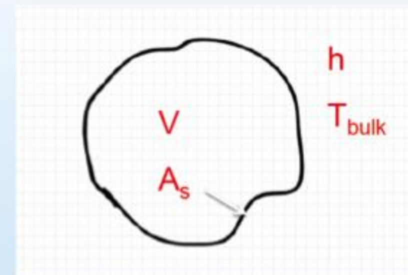


Assumptions:

- constant h, k, C_p
- constant T_{bulk}
- $h_{transient} = h_{steady\text{-}state}$

$$\frac{T_{wall}(t) - T_{initial}}{T_{bulk} - T_{initial}} = 1 - \exp\left(\frac{h^2\,\alpha\, t}{k^2}\right) \mathrm{erfc}\left(\frac{h\sqrt{\alpha\, t}}{k}\right)$$

0-D Model



Experimental Technique:

- Measures T = f(time)
- Calculate h at time when T = 37.6 °C

$$\frac{T_{wall}(t) - T_{bulk}}{T_{initial} - T_{bulk}} = \exp\left[-\left(\frac{h\, A_s}{\rho\, V\, C_p}\right) t\right]$$

One equation method

$$\frac{T_w - T_i}{T_b - T_i} = 1 - \exp\left[\frac{h^2 \alpha t}{k^2}\right] erfc\left[\frac{h\sqrt{\alpha t}}{k}\right]$$

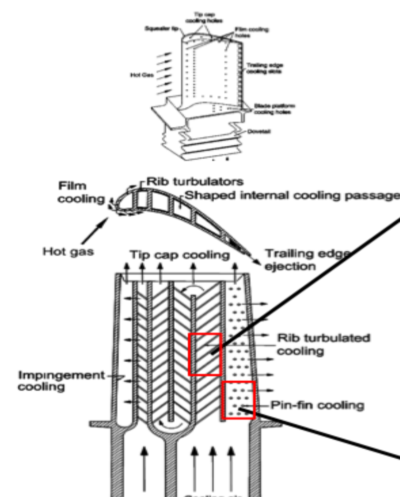
'h' can be calculated with known values of T_{bulk}

Two equation method

$$\frac{T_{w_1} - T_i}{T_b - T_i} = 1 - \exp\left[\frac{h^2 \alpha t_1}{k^2}\right] erfc\left[\frac{h\sqrt{\alpha t_1}}{k}\right]$$

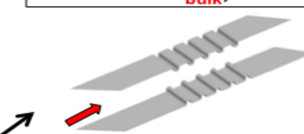
$$\frac{T_{w_2} - T_i}{T_b - T_i} = 1 - \exp\left[\frac{h^2 \alpha t_2}{k^2}\right] erfc\left[\frac{h\sqrt{\alpha t_2}}{k}\right]$$

'h' & 'T_{bulk}' can be found by solving the two equations



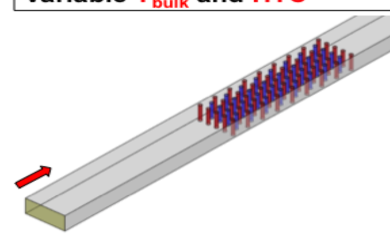
Schematic of a modern gas turbine blade with common cooling techniques

Constant T_{bulk}, Variable HTC

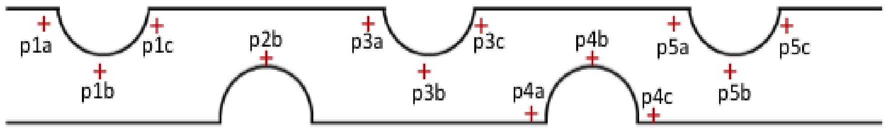


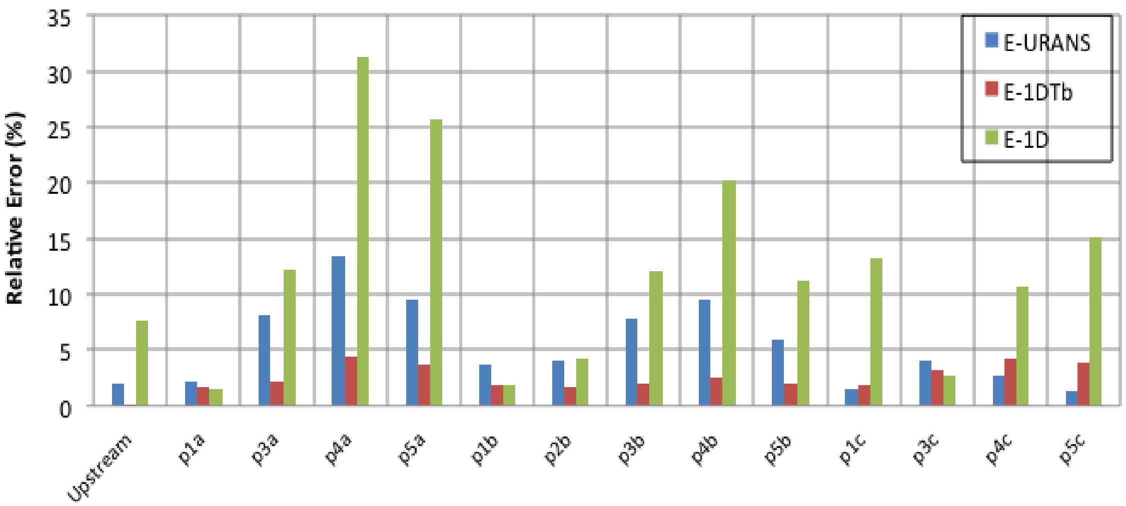
$T_{bulk,\ inlet} = 350$ K = 76.85 °C
$T_{surface,\ initially} = 300$ K = 26.85 °C

Variable T_{bulk} and HTC



HTC: 1-eq vs 2-eq vs URANS & RANS at the time $T_{surface}$ = 37.6 ºC



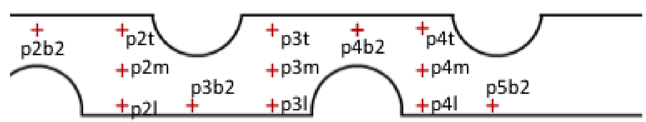


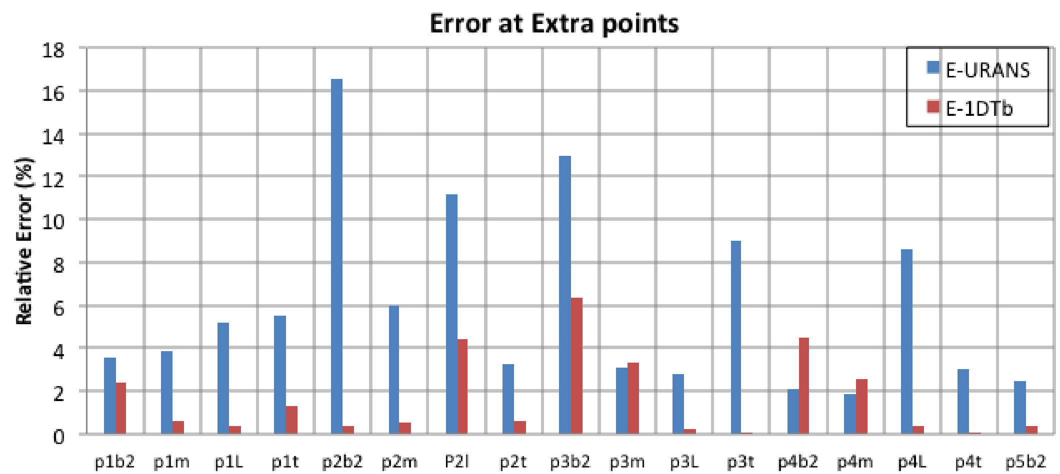
Error due to 3-D variation

- < 4% for 1-eq
- up to 30% for 2-eq.

Error due to unsteady T on wall

- up to 16%





E-URANS = (h-URANS – h-RANS) / h-RANS

E-1DTb= (h-1DTb – h-URANS) / h-URANS

E-1D = (h-1D – h-URANS) / h-URANS

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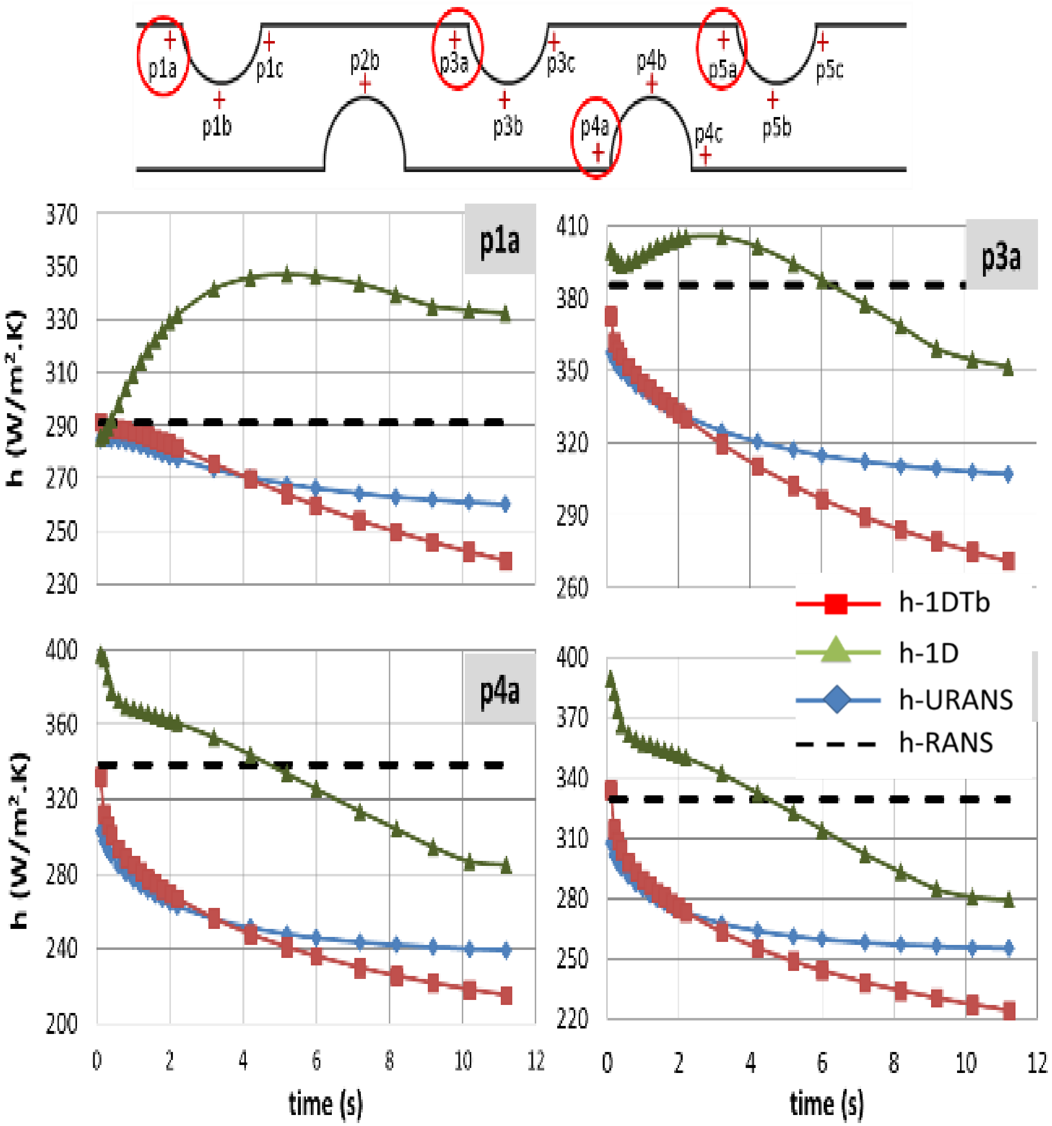
Measurements of HTC by Transient Methods



- HTC measured by transient methods **differ** from the HTC under steady-state conditions with isothermal walls (up to 17% for the pin-fin problem and up to 26% for the rib problem).
- HTC measured by the transient methods vary appreciably in time (up to 31% over 10 seconds). Thus, it is unclear which HTC measured is the correct or the meaningful one.

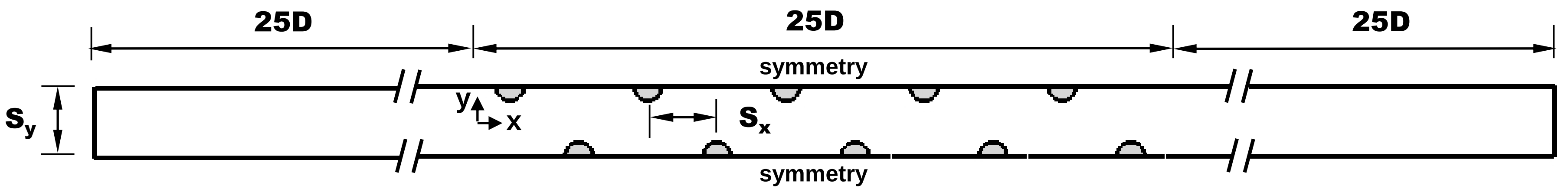
Sathyanarayanan, Ramachandran , Shih, et al. (AIAA 2015-1195)

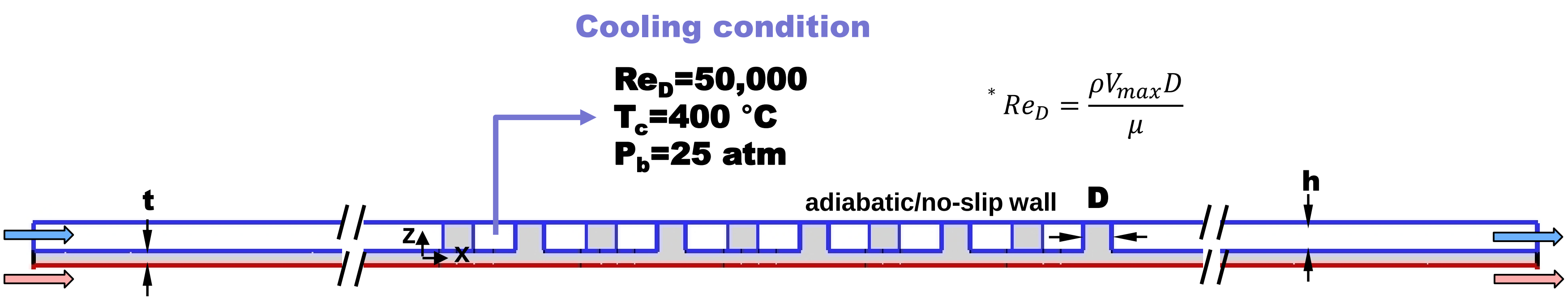
Measurements of HTC by Steady-State Methods

Issues:

- Lab vs engine-relevant conditions? Will data from lab conditions be useful?
- Since HTC = f(wall BC: T or q”), what are the effects of imposing constant q”, which is what is typically used?
- What are the effect of the plate thickness and its material properties on the HTC?

Problem Description





$$h/D = 1$$

$$S_x/D = 2.5$$

$$S_y/D = 5$$

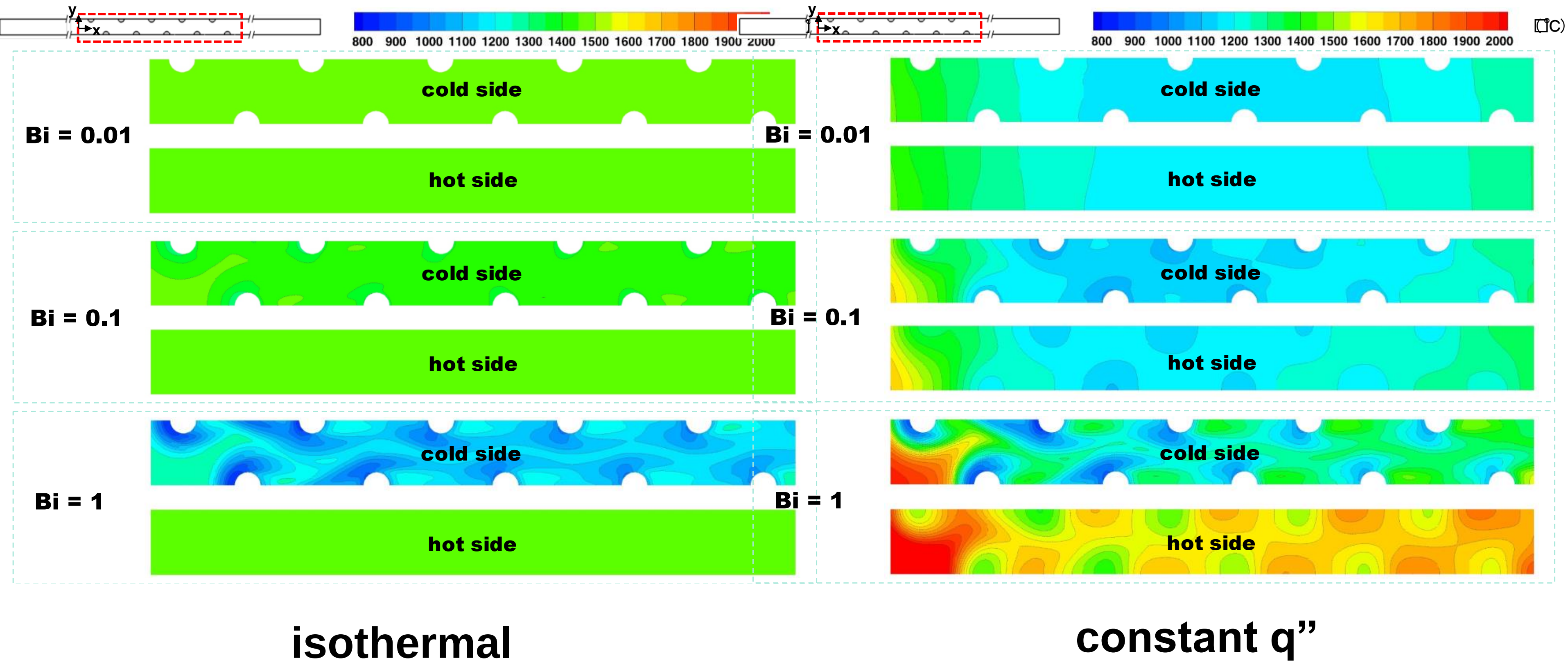
$$t/D = 0, 0.5, 1$$

Case	Heating BCs
I	HTC_hot = infinity (*Twall=Thot=1755K*)
II	HTC_hot = average of HTC_cold
III	specified q" without HTC_hot
IV	specified q", isothermal, or HTC_hot with different Bi (k or t/D varies)

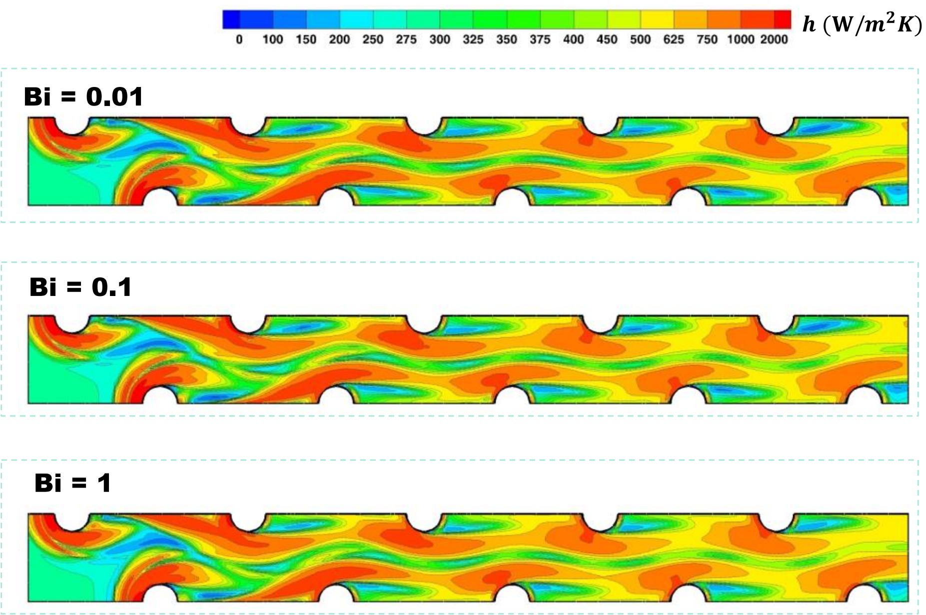
Heating BCs on Hot-gas Side

Case I	**HTC_hot = infinity**	
case I-1	*const. temperature*	$T_{w,hot}$=1,273 K
case I-2		$T_{w,hot}$=1,755 K
Case II	**HTC_hot = average of HTC_cold**	
case II-1		h_h=100 W/m^2-K, T_{hot}=1,755 K
case II-2	*const. convective environment*	h_h=1,000 W/m^2-K, T_{hot}=1,273 K
case II-3		h_h=1,000 W/m^2-K, T_{hot}=1,755 K
Case III	**specified q" without HTC_hot**	
case III-1	*adiabatic*	$q''_{w,hot}$=0 W/m^2
case III-2	*const. heat flux*	$q''_{w,hot}$=253,012 W/m^2
case III-3		$q''_{w,hot}$=462,483 W/m^2
Case IV	**specified heating BCs with Bi=0.01, 0.1, 1 (varied k)**	
IV-1	*const. heat flux (q"=462,483W/m^2)*	Bi=0.01
IV-2		Bi=0.1
IV-3		Bi=1
IV-4	*const. temperature (T_{hot}=1755K)*	Bi=0.01
IV-5		Bi=0.1
IV-6		Bi=1
IV-7	*const. convective environment (h=600 W/m^{2}K, T_h=1755K)*	Bi=0.01
IV-8		Bi=0.1
IV-9		Bi=1

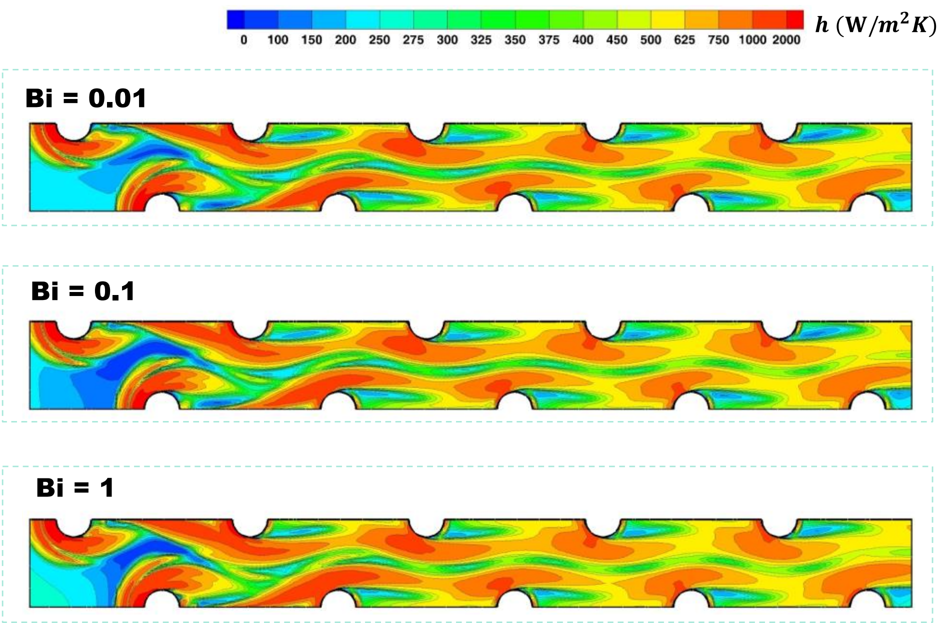
Temperature Distribution w/ Bi=0.01, 0.1, and 1



HTC on Cold Wall w/ Bi=0.01, 0.1, and 1



isothermal



constant q’’

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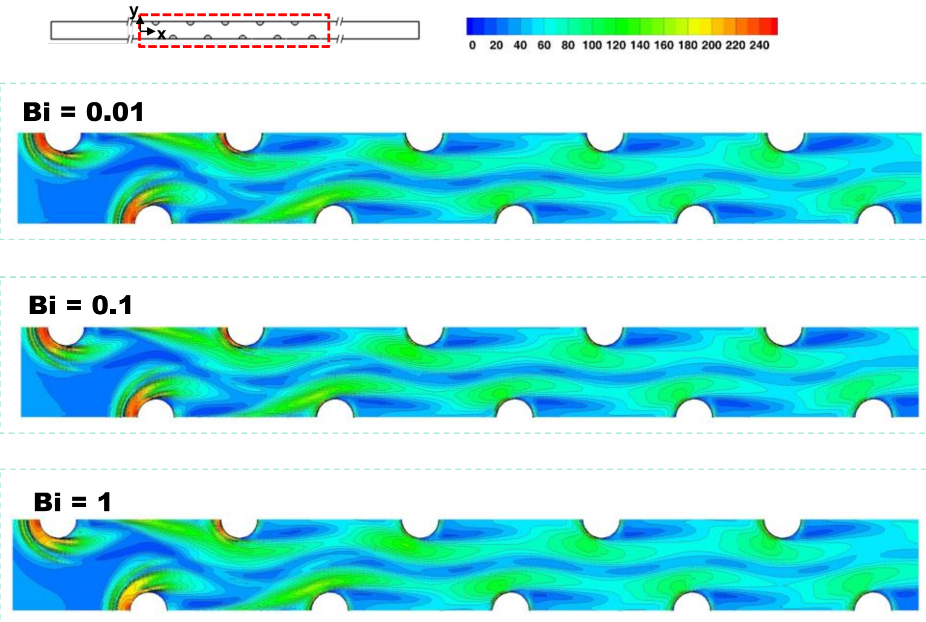
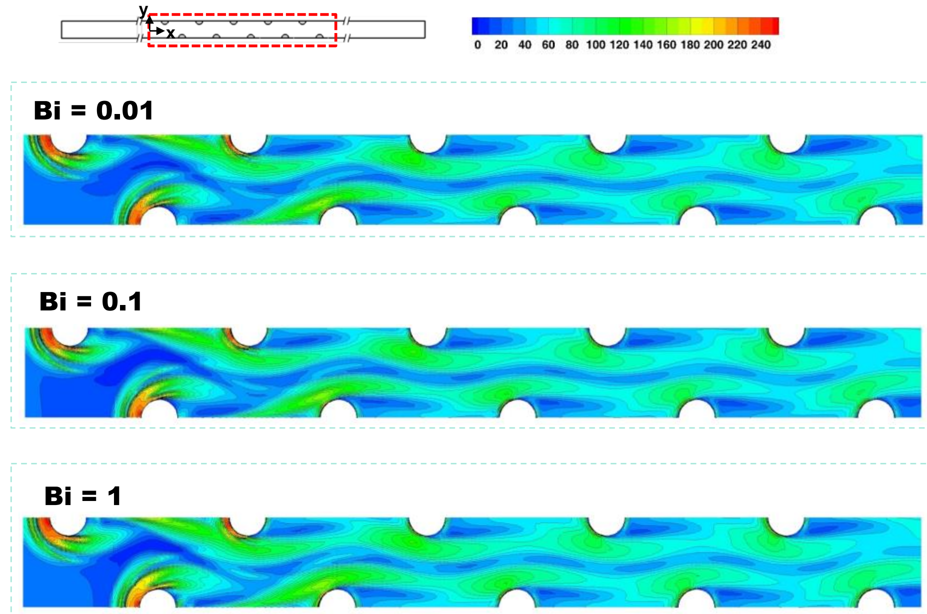
Nu on Cold Wall w/ Bi=0.01, 0.1, and 1

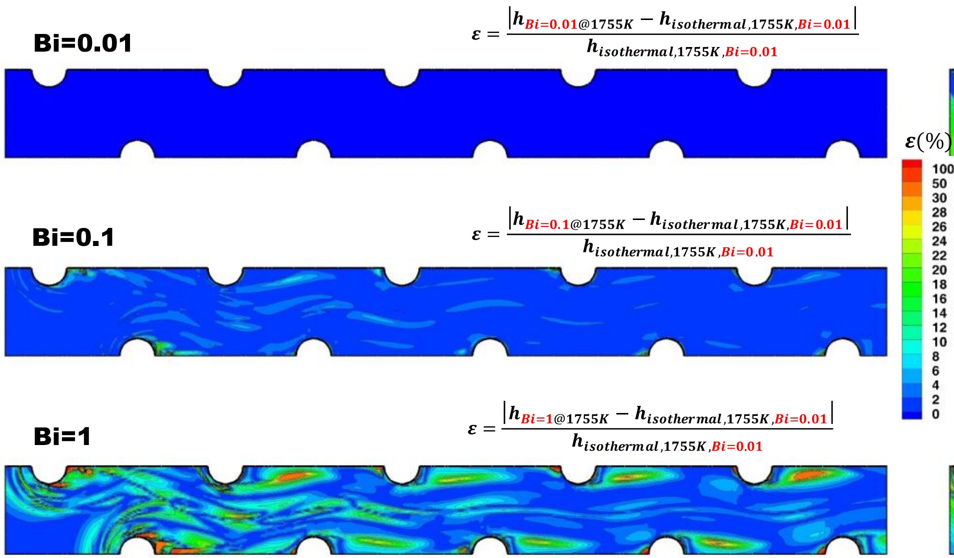


isothermal

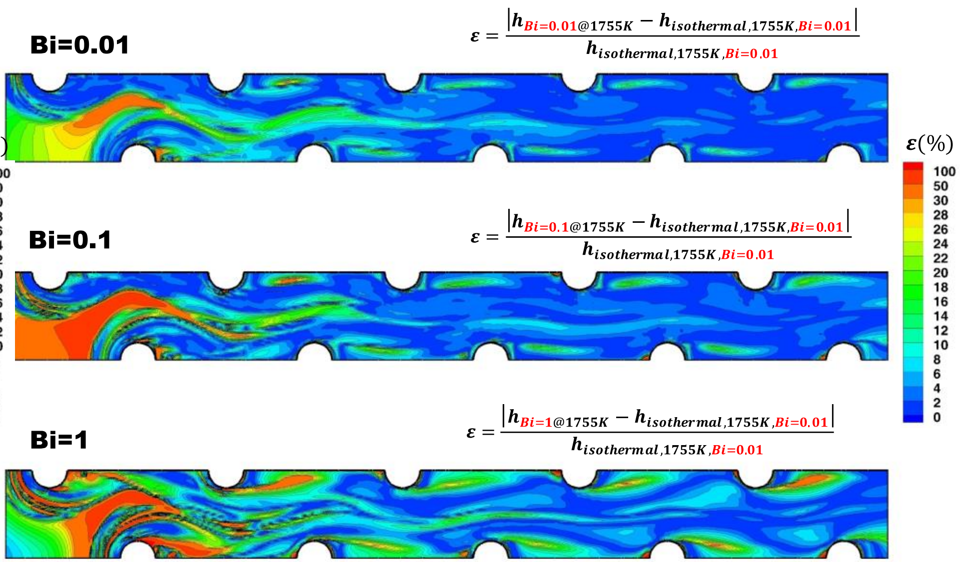


constant q”

Relative Error Distribution of HTCs on Cold Wall w/ Bi=0.01, 0.1, and 1

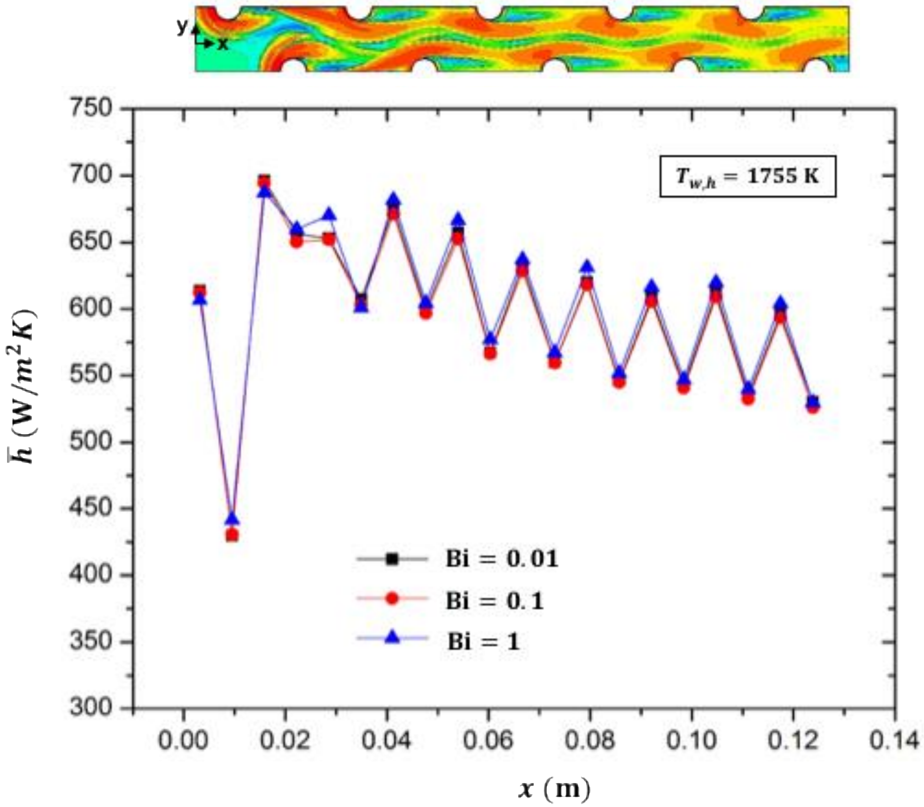


isothermal

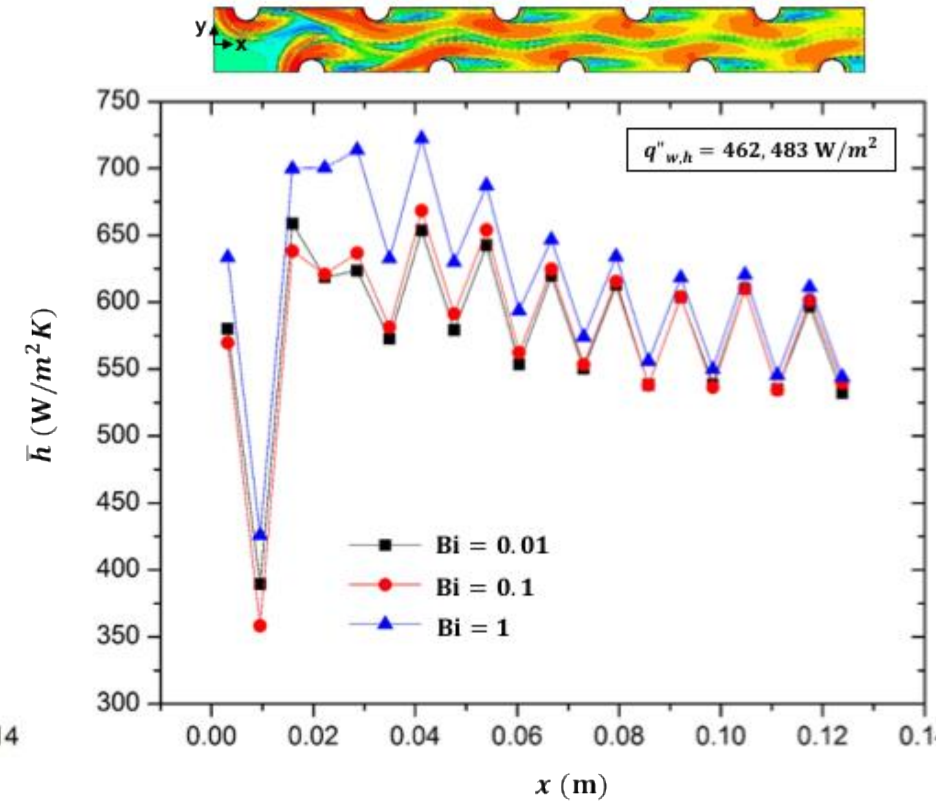


constant q''

Zonally Averaged HTC w/ Bi=0.01, 0.1, and 1

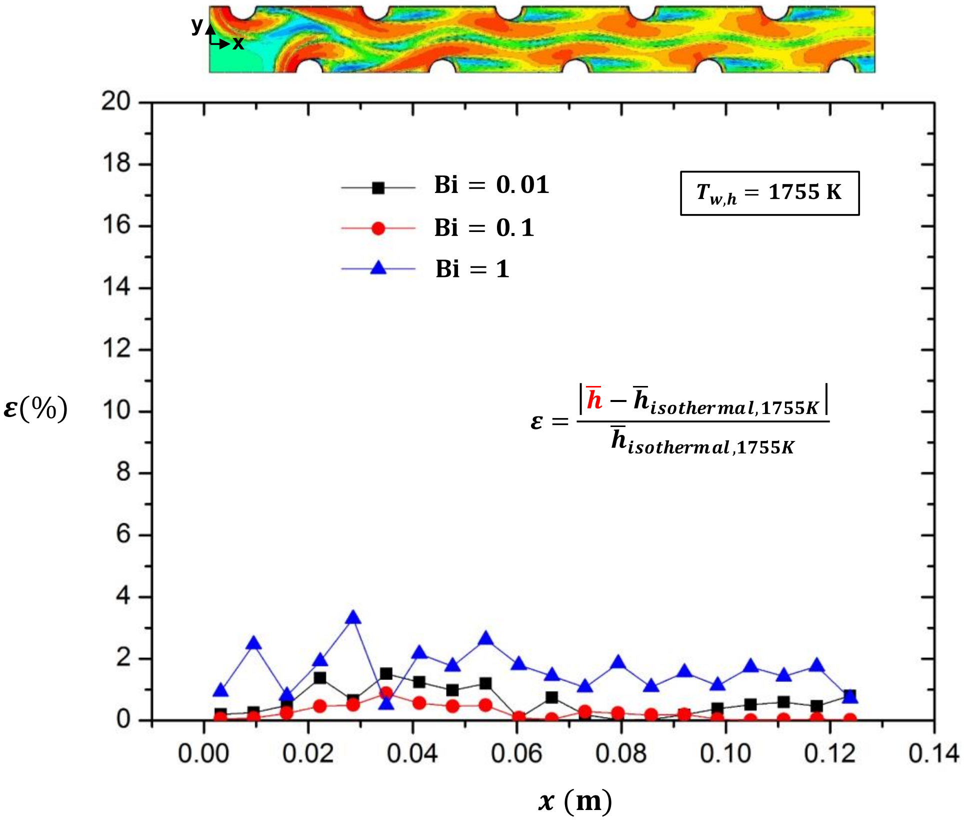


isothermal

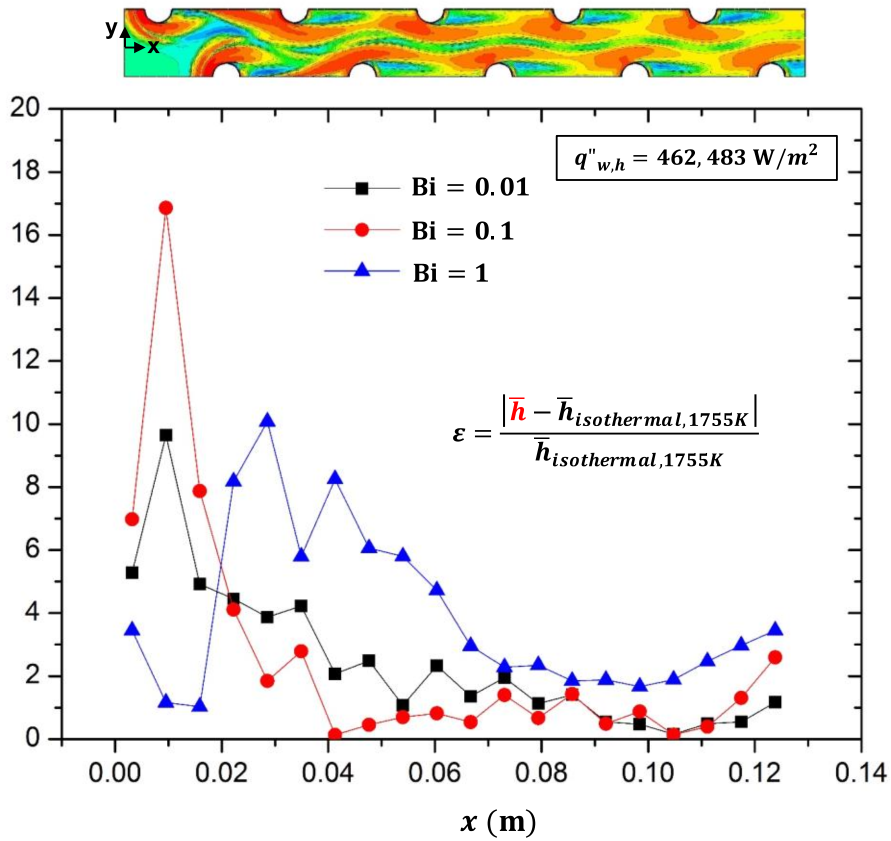


constant q"

Relative Error of Zonally Averaged HTCs w/ Bi=0.01, 0.1, and 1

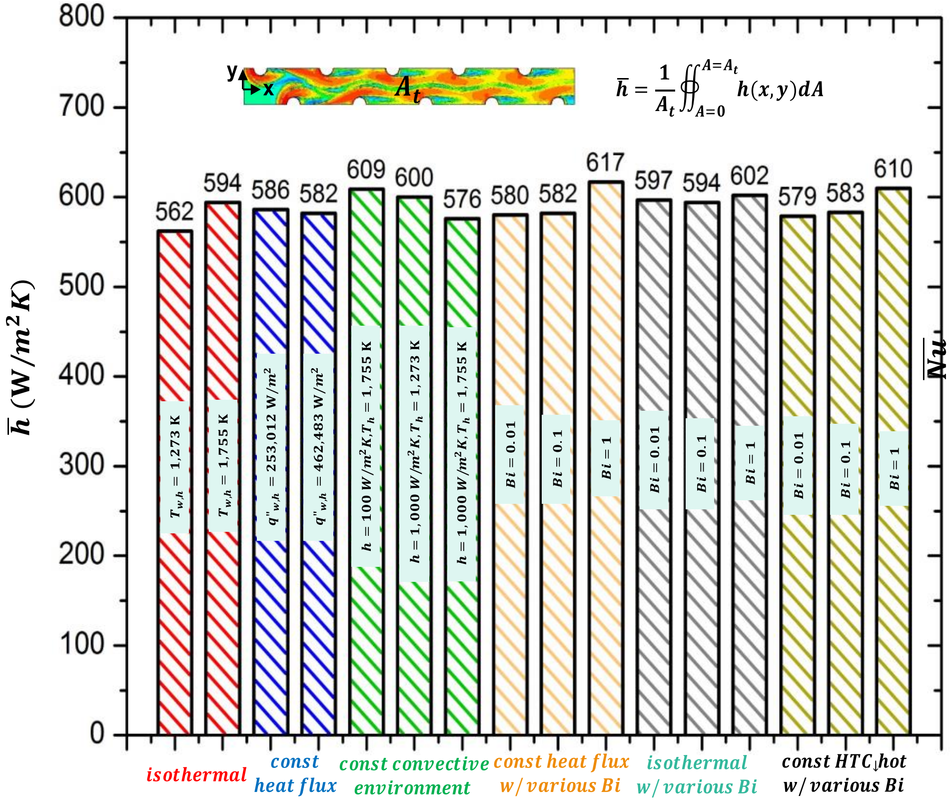


isothermal

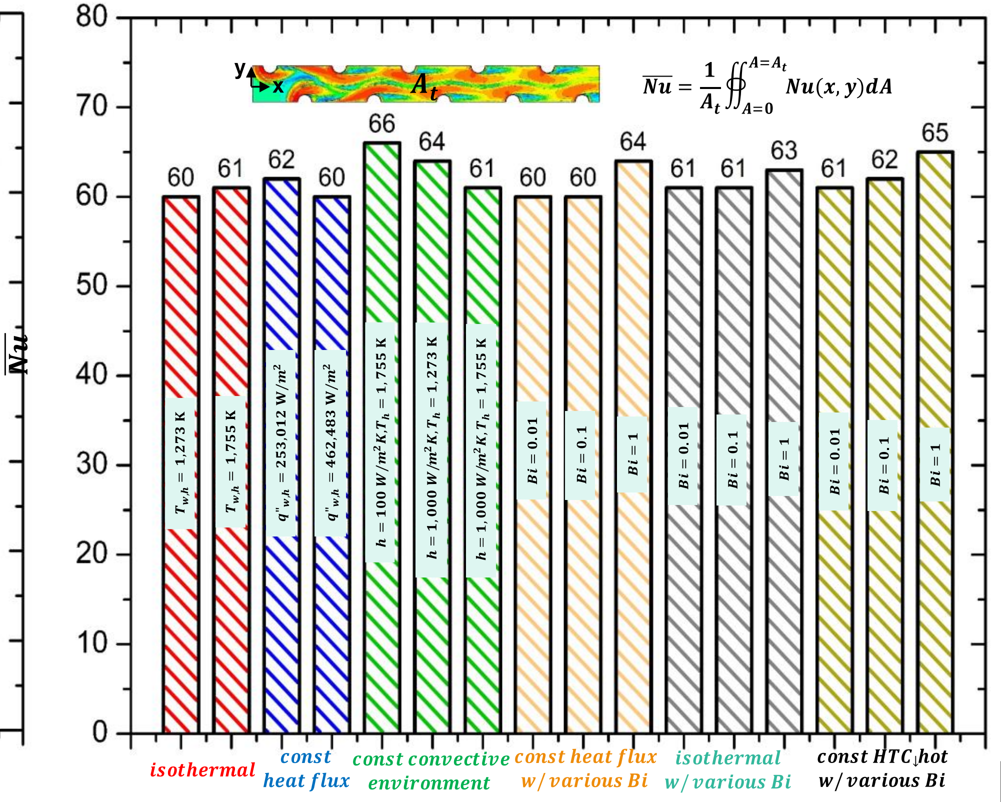


constant q''

Total Averaged HTC & Nu of the Entire Endwall Section



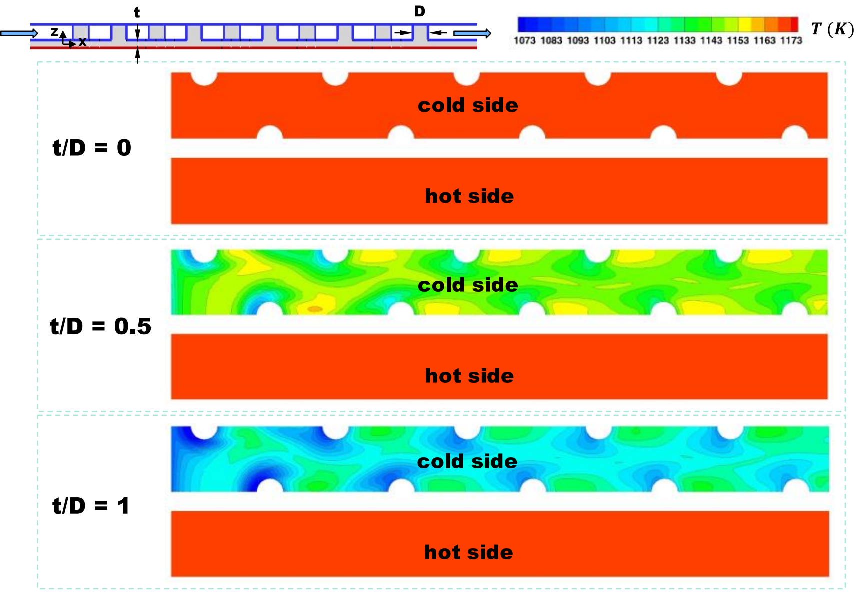
HTC



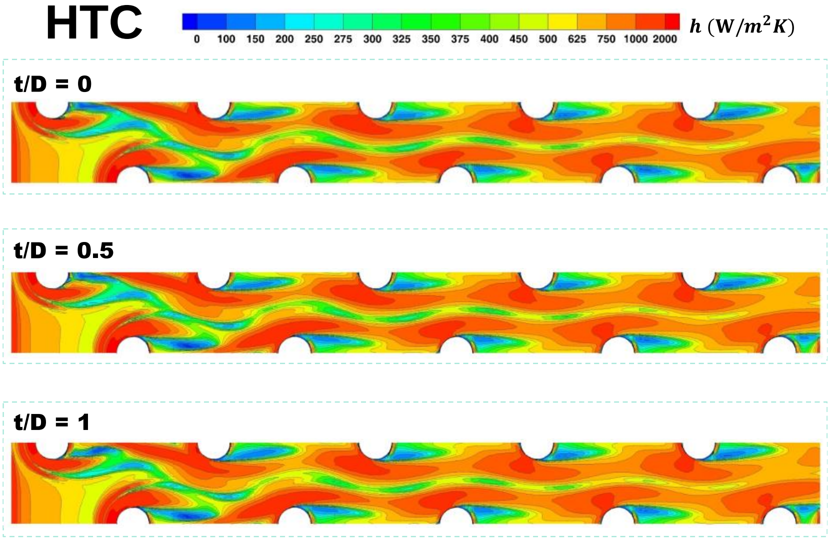
Nu

Temperature Distribution w/ t//D=0, 0.5, and 1 (isothermal)

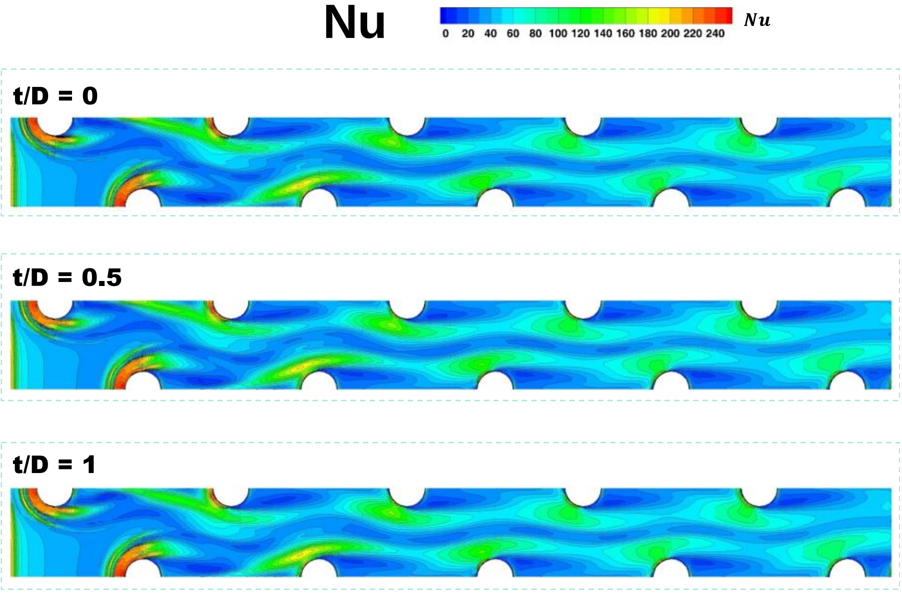
Temperature



HTC



Nu



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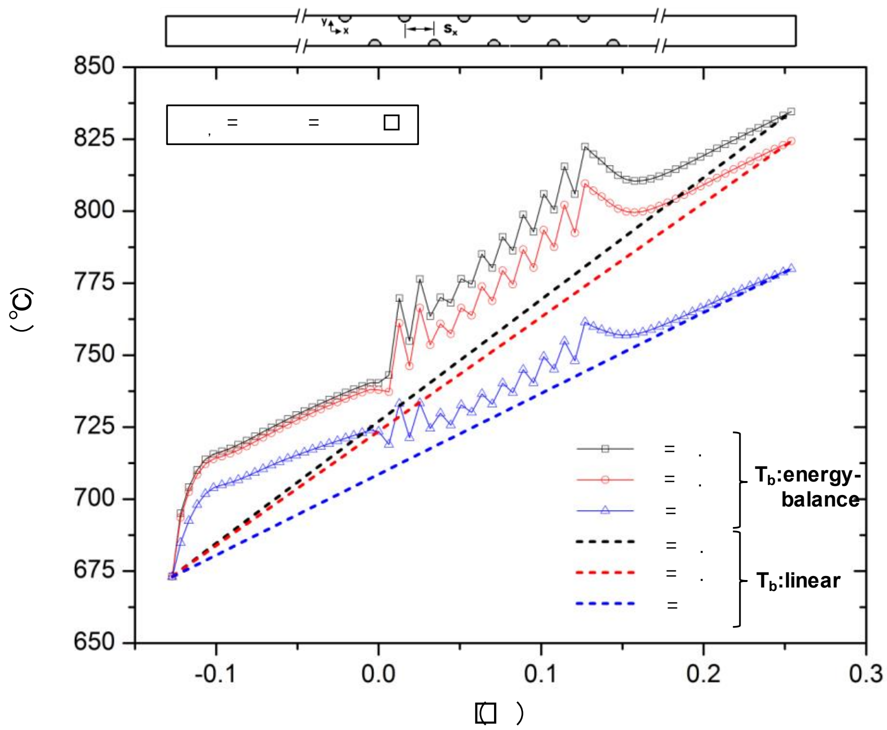
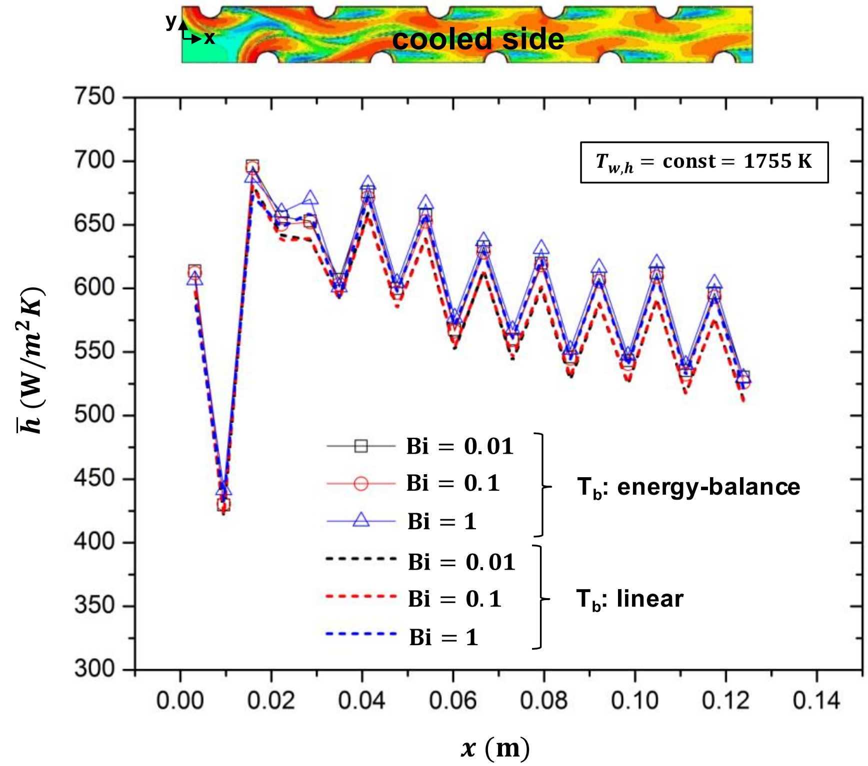
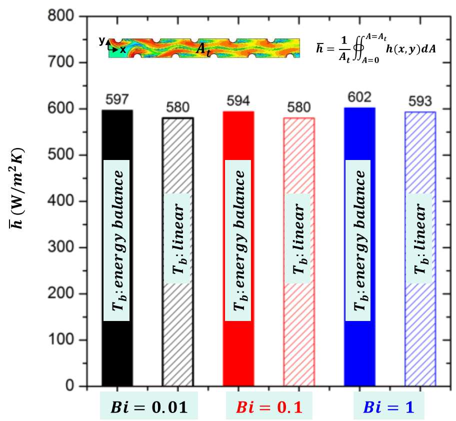
Energy-balance T_bulk v.s. linear T_bulk







For this problem, approximating T_bulk by linear interpolation is OK because the variation in T is small.

Outline of Talk

Revisit

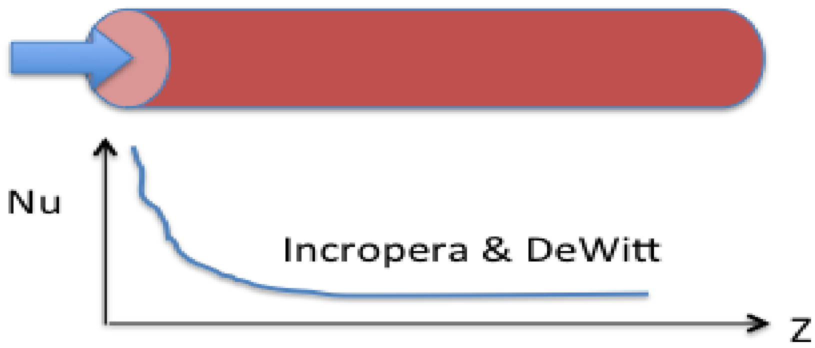
- HTC measured by transient methods
- HTC measured by steady-state methods

What else?

- Bulk Temperature?
- Nu = F(Re, Pr, geometry)?

Summary & Current Work

Example 1: Bulk Temperature and Heat-Transfer Coef.

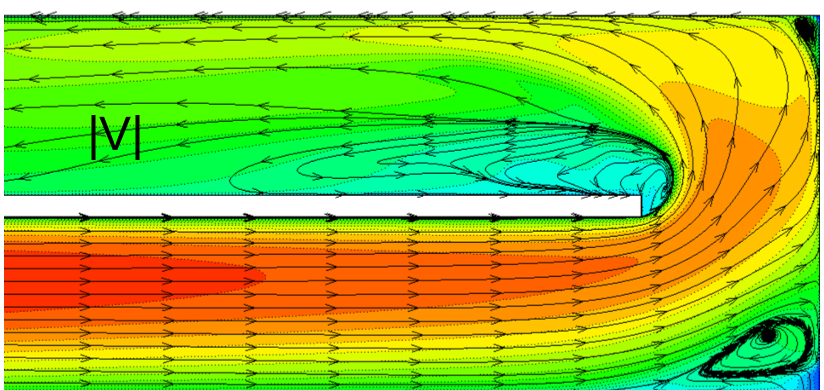


$$q'' = h\,(T_{wall} - T_b) \qquad T_b = \frac{\int_A \rho u C_p T\,dA}{\int_A \rho u C_p\,dA}$$

$$Nu = hD_h/k$$

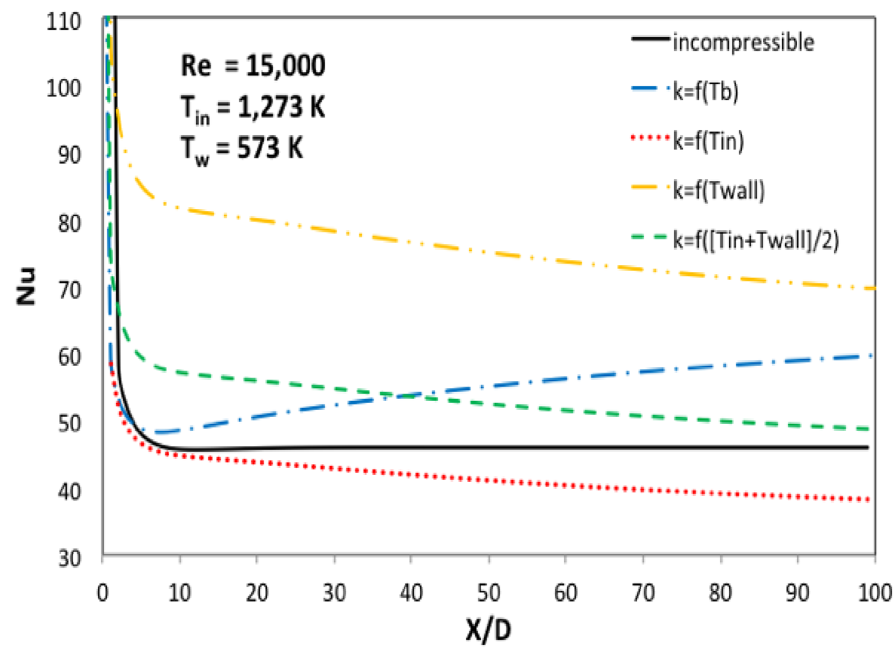
T_b is hard to measure. Why?

- an integral!
- Which cross section around the bend?
- How to handle flow separation, ribs, pin fins, …?



Bulk temperature is almost always approximated in EXPERIMENTS but rarely documented.

- What are the consequences of the approximations?
- How should HTC be measured and used in V&V and in design?

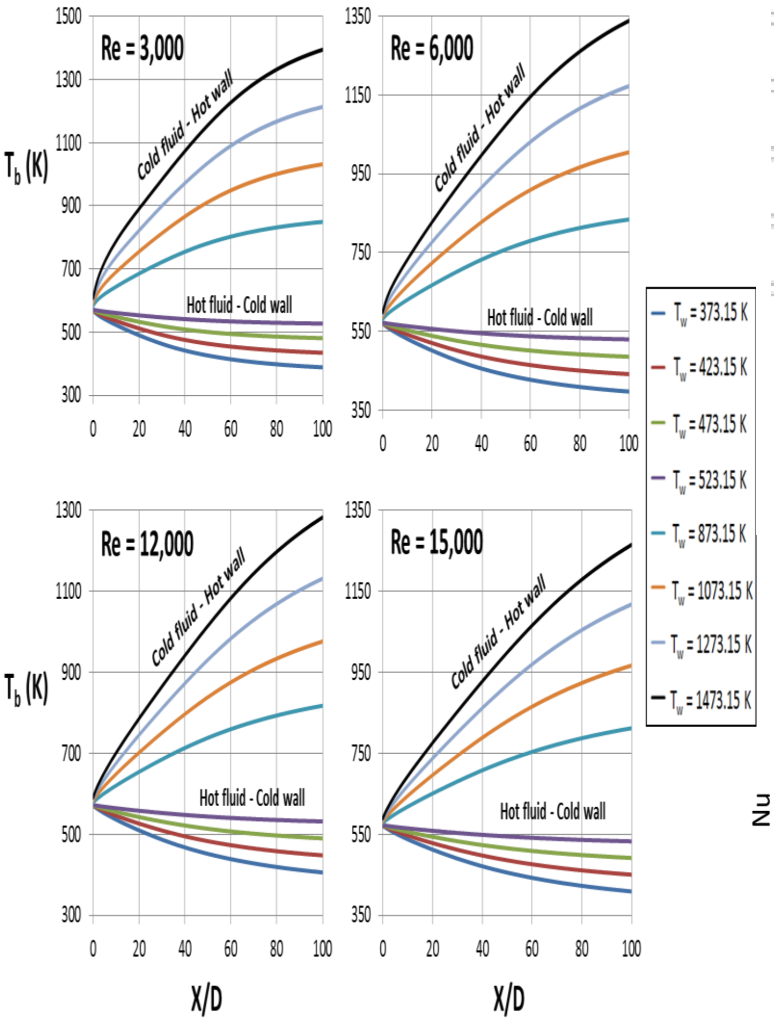


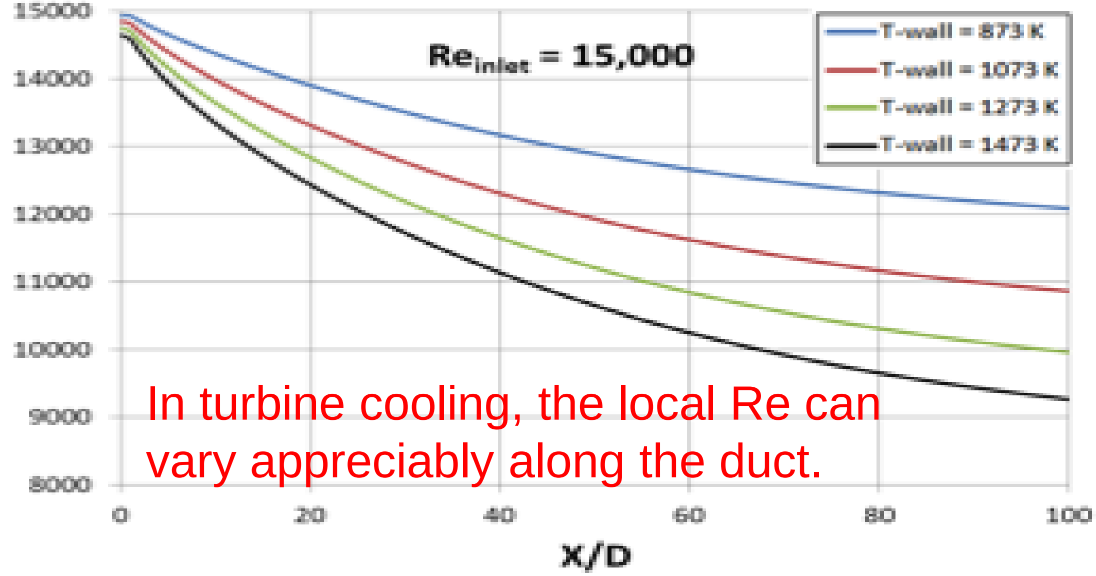
Chi & Shih (AIAA 2012-0807)

Example 2: Heat-Transfer Coef. & Reynolds No.

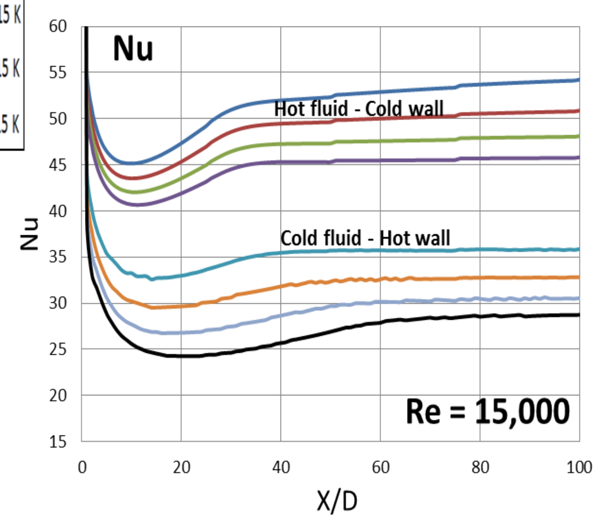
$$T_b = \frac{\int_A \rho u C_p T \, dA}{\int_A \rho u C_p \, dA}$$

$$Re = \rho_{in} V_{in} D_h / \mu, \quad \mu = f(T_b)$$





In turbine cooling, the local Re can vary appreciably along the duct.

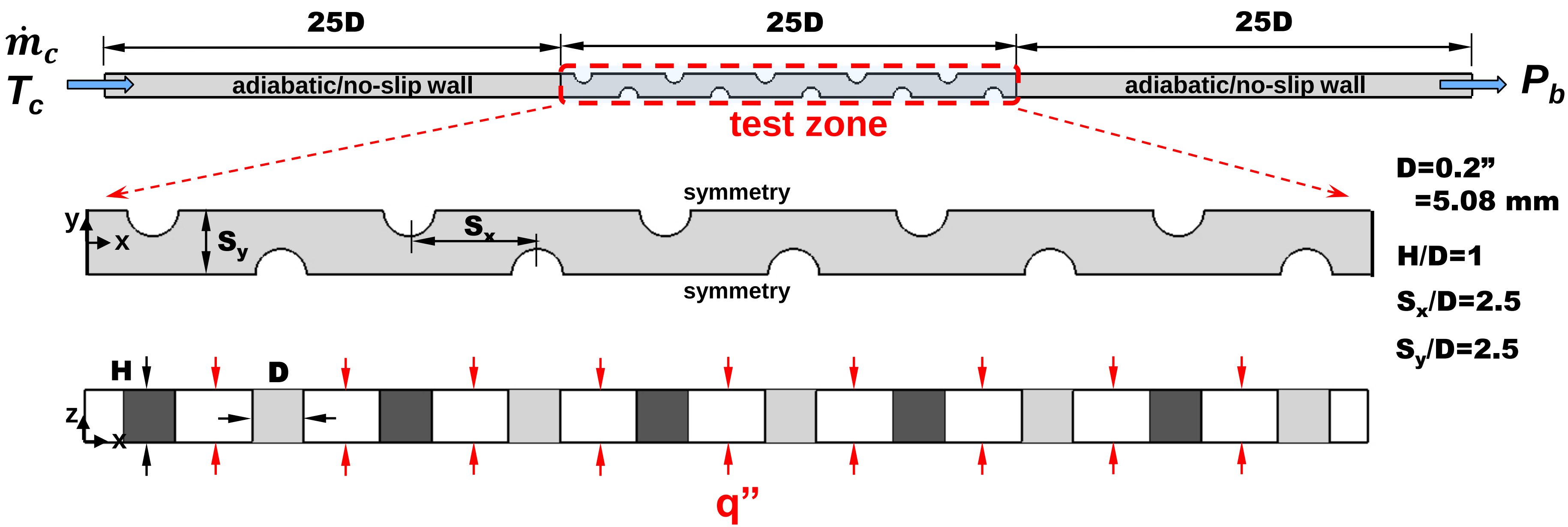


$$q'' = h(T_w - T_b)$$

$$Nu = hD/k, \quad k = k(T_b)$$

Shih, et al. (ASME HT-2013-17114)

Problem Description

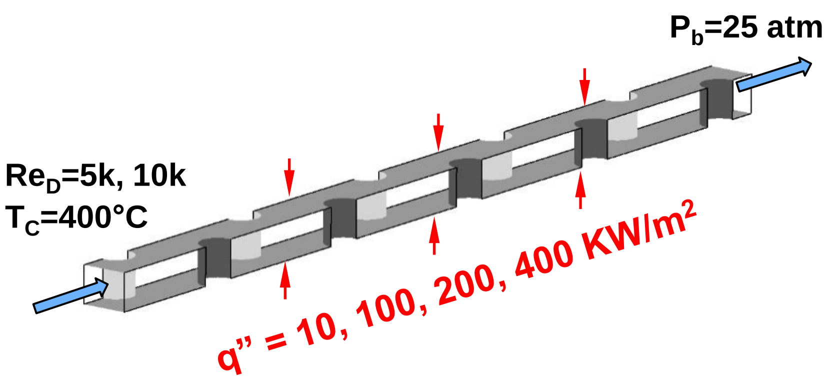


D=0.2"
=5.08 mm

H/D=1

$S_x/D=2.5$

$S_y/D=2.5$



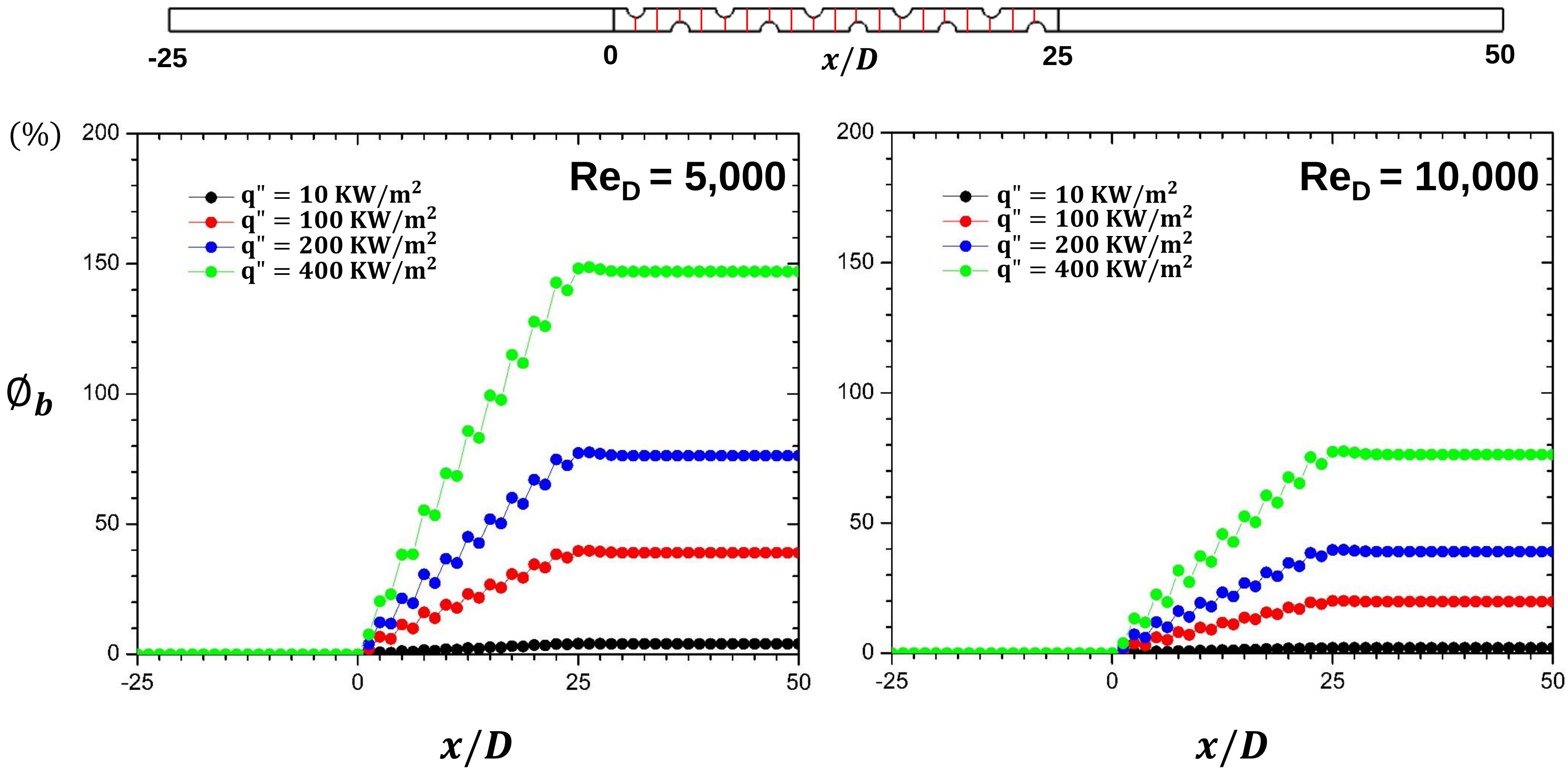
Cases	cooling condition	heating load
1		q" = 10 KW/m²
2	Re_D = 5,000 T_c = 400 °C P_b = 25 atm	q" = 100 KW/m²
3		q" = 200 KW/m²
4		q" = 400 KW/m²
5		q" = 10 KW/m²
6	Re_D = 10,000 T_c = 400 °C P_b = 25 atm	q" = 100 KW/m²
7		q" = 200 KW/m²
8		q" = 400 KW/m²

$$Re_D = Re_{D,x} \; at \; x = D, \quad Re_{D,x} = \frac{\rho_x U_x D}{\mu(T_{b,x})}$$

Dimensionless Bulk Temperature

$S_x/D = 2.5,\ S_y/D = 2.5,\ H/D = 1$

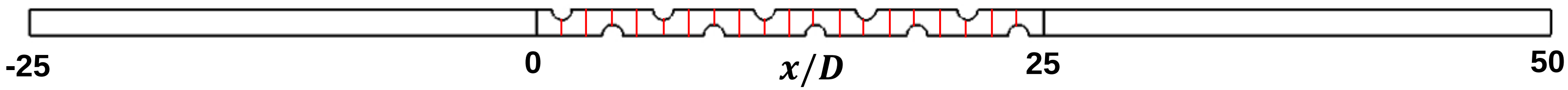
$$\emptyset_{b,x} = \frac{T_{b,x} - T_{c,in}}{T_{c,in}} \qquad T_{c,in} = 400°\text{C}$$

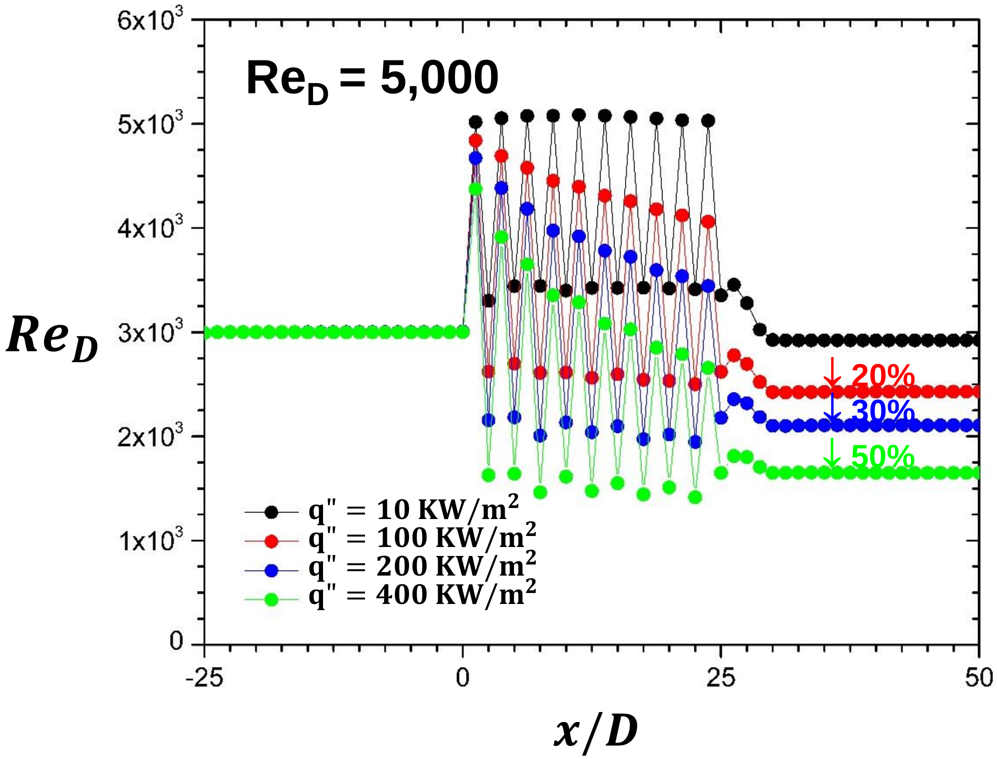


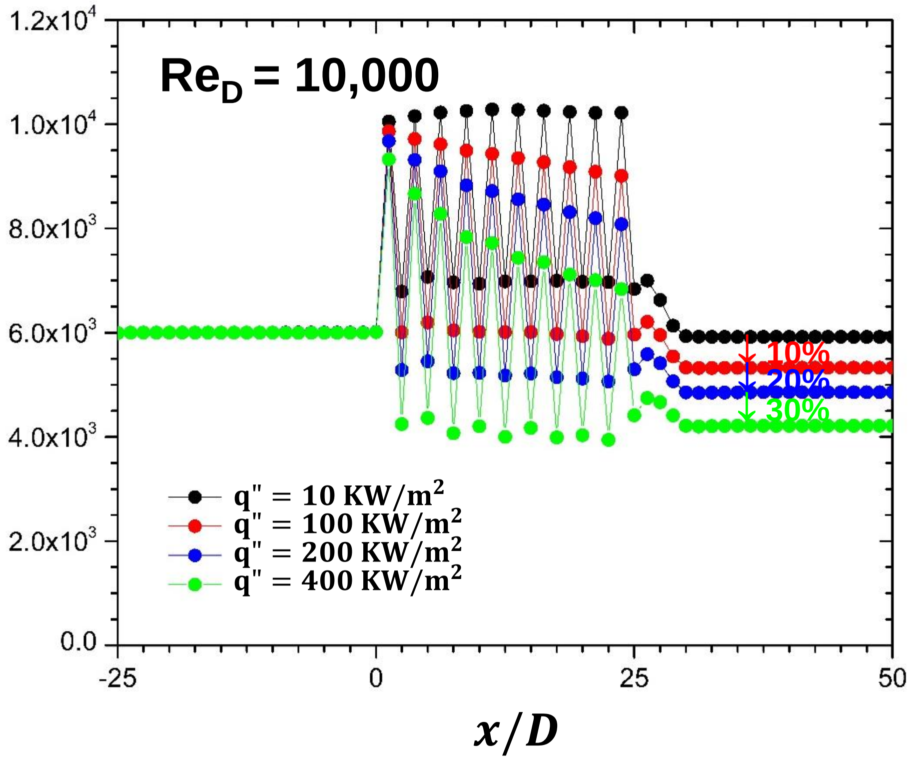
Local Reynolds Number = Re(T_b)

$S_x/D = 2.5,\ S_y/D = 2.5,\ H/D = 1$

$$Re_{D,x} = \frac{\rho_x U_x D}{\mu(T_b)}$$



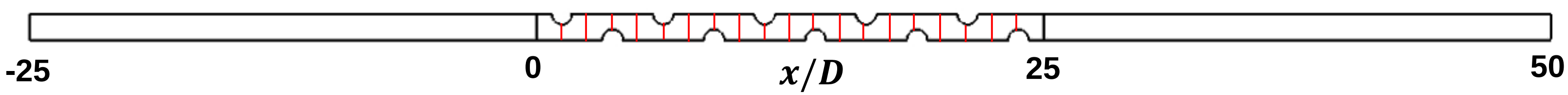


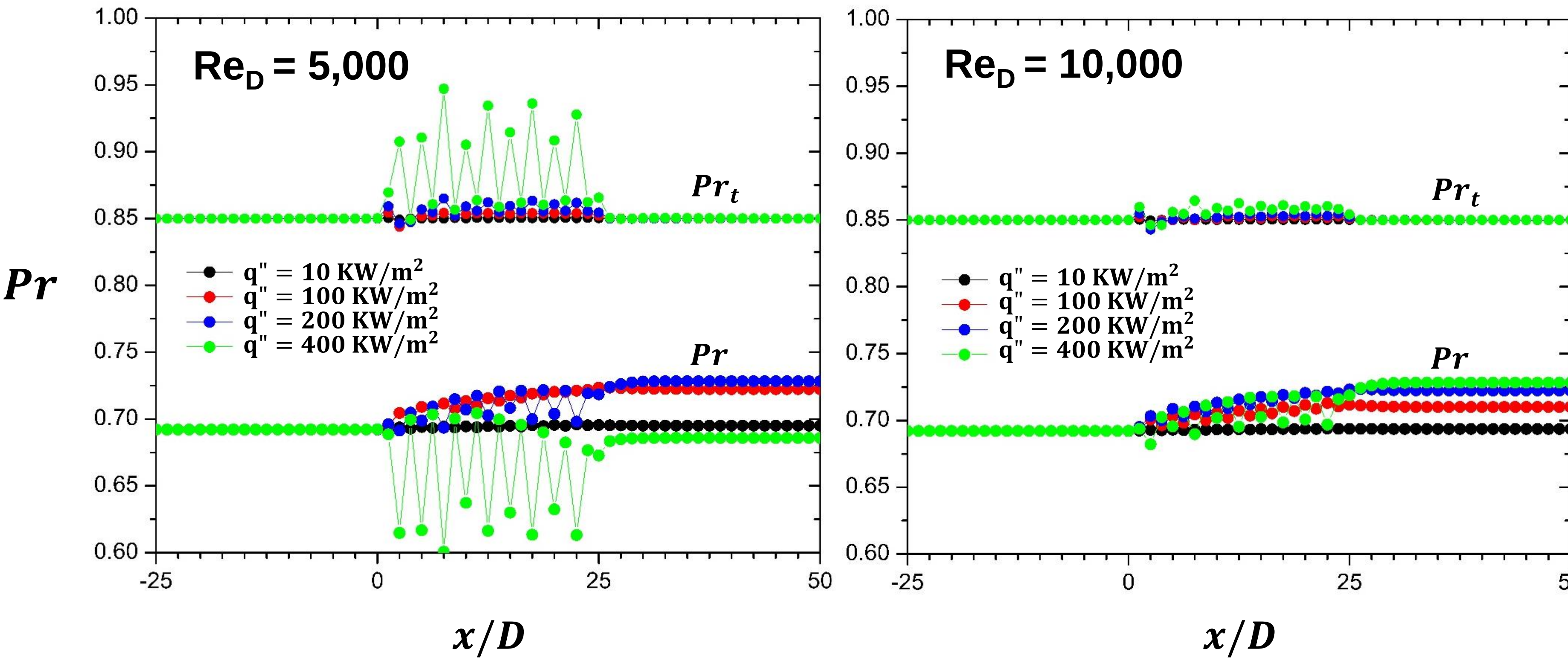


Local Prandtl Number = $Pr(T_b)$

$S_x/D = 2.5,\ S_y/D = 2.5,\ H/D = 1$

$$Pr = \frac{\mu(T_b)C_P(T_b)}{k(T_b)} \qquad Pr_t = \frac{\mu_t C_P(T_b)}{k_t}$$



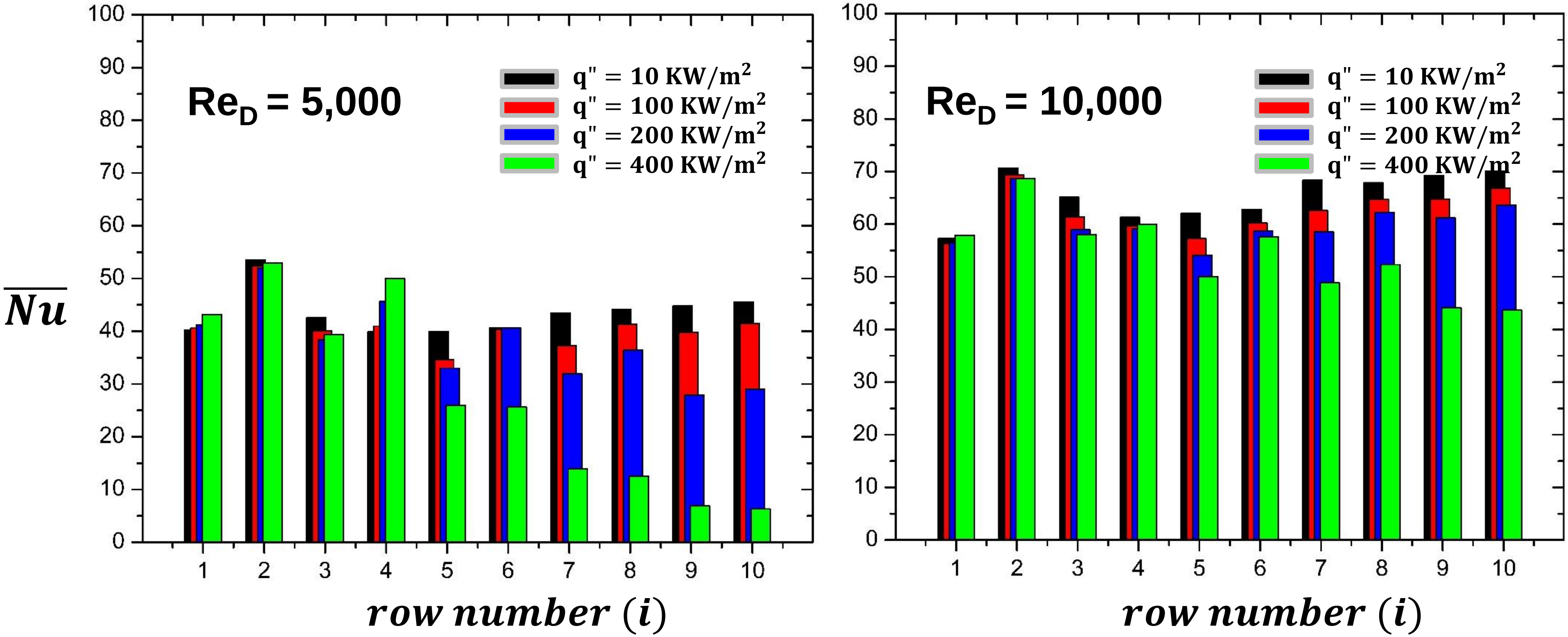


Lateral Averaged Nusselt Number

$$\overline{Nu}_i = \frac{\bar{h}_i D}{k}$$

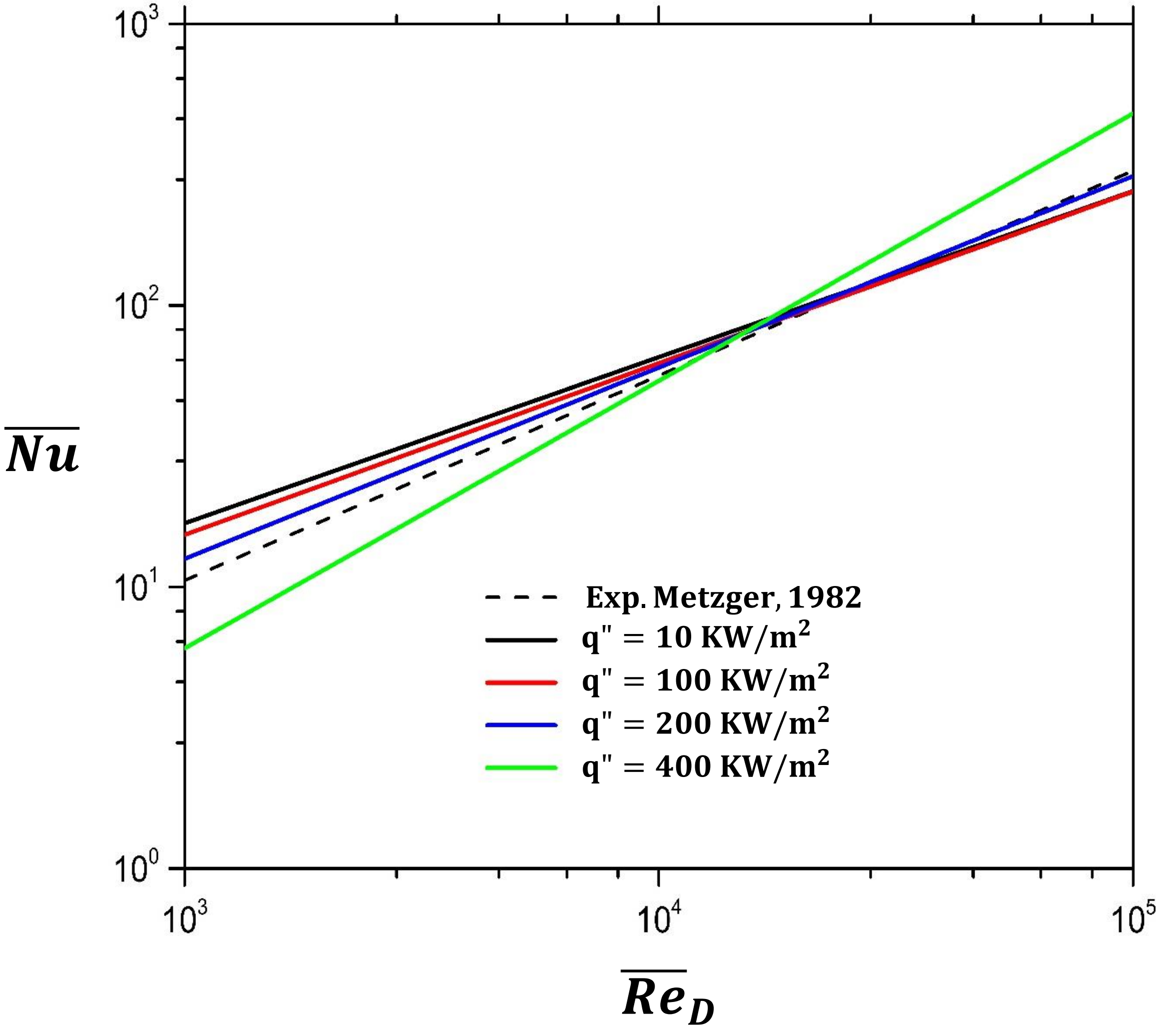
regional average about each row (i) of pins in streamwise direction

$i = 1$ $i = 10$



Correlations: $\overline{Nu}$ v.s. $\overline{Re}_D$

Mean Nusselt numbers ($\overline{Nu}$) for the entire ten row array as functions of the mean Reynolds numbers ($\overline{Re}_D$)



Correlations: $\overline{Nu} = C\overline{Re}_D^M$	
isothermal	
Exp. Metzger, 1982	$\overline{Nu} = 0.069\overline{Re}_D^{0.73}$
const heat flux (CFD)	
q" = 10KW/m²	$\overline{Nu} = 0.286\overline{Re}_D^{0.59}$
q" = 100KW/m²	$\overline{Nu} = 0.226\overline{Re}_D^{0.61}$
q" = 200KW/m²	$\overline{Nu} = 0.115\overline{Re}_D^{0.68}$
q" = 400KW/m²	$\overline{Nu} = 0.009\overline{Re}_D^{0.95}$

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Summary

Measuring HTC by transient methods could have some issues.

Measuring HTC by steady-state methods is relatively independent of the BC imposed on the wall surface.

The Re in an internal cooling passage may vary appreciably along the passage because of the rise in coolant temperature from 400 ^{o}C to 700 or 800 ^{o}C.

HTC is currently obtained under conditions, where T_w/T_b is near unity. Thus, scaling of HTC is needed. The question is how?