2012 University Turbine Systems Research Workshop
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Steady and Unsteady Conjugate Heat Transfer in Turbine Cooling

Tom Shih,[1,2] Mark Bryden,[2] Robin Ames,[3] and Rich Dennis[3]

[1]School of Aeronautics and Astronautics, Purdue University

[2]Ames National Laboratory, U.S. Dept. of Energy

[3]National Energy Technology Laboratory, U.S. Dept. of Energy

Students: *Kyle Chi,* Surya Muthukannan Chinnamani, *Kenny Hu, C.-S. Lee, Jason Liu, S.K. Sathyanarayanan, Christelle Wanko, Adam Weaver*

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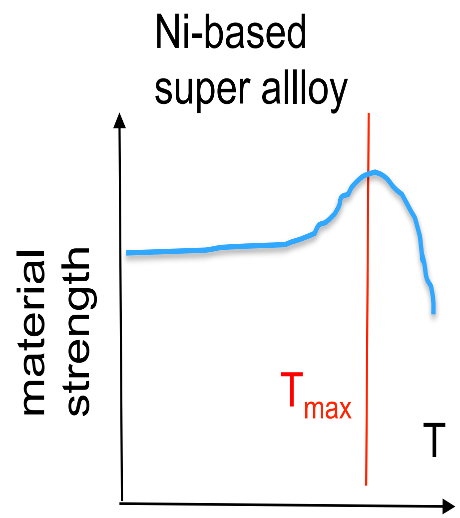
Background

Challenges:

- Modern turbines are designed to operate close to the material's maximum allowable temperature and temperature gradient. *So, there is little room for mistakes in designs.*
- Burning coal-derived fuels affect heat transfer through increased flow rates, erosion/deposition, and high-temperature oxidation. *Adds burden & uncertainty to cooling designs.*
- New designs that greatly reduce cooling flow are outside of the **empirical** design experience. *Thus, need* ***physics/ math-based*** *design tools.*



Bottom line: need better design tools and better understanding of the flow and heat transfer as a function of design and operating parameters.

On design tools, the issues are

- verification, validation, and uncertainty quantification
- **Quality of** and **info on** benchmark data used for validation
- turbulence modelling, conjugate vs non-conjugate CFD analysis

Outline of Talk

Show challenges from two validation studies.

Assess the accuracy of a measurement technique widely used to generate benchmark data to validate CFD.

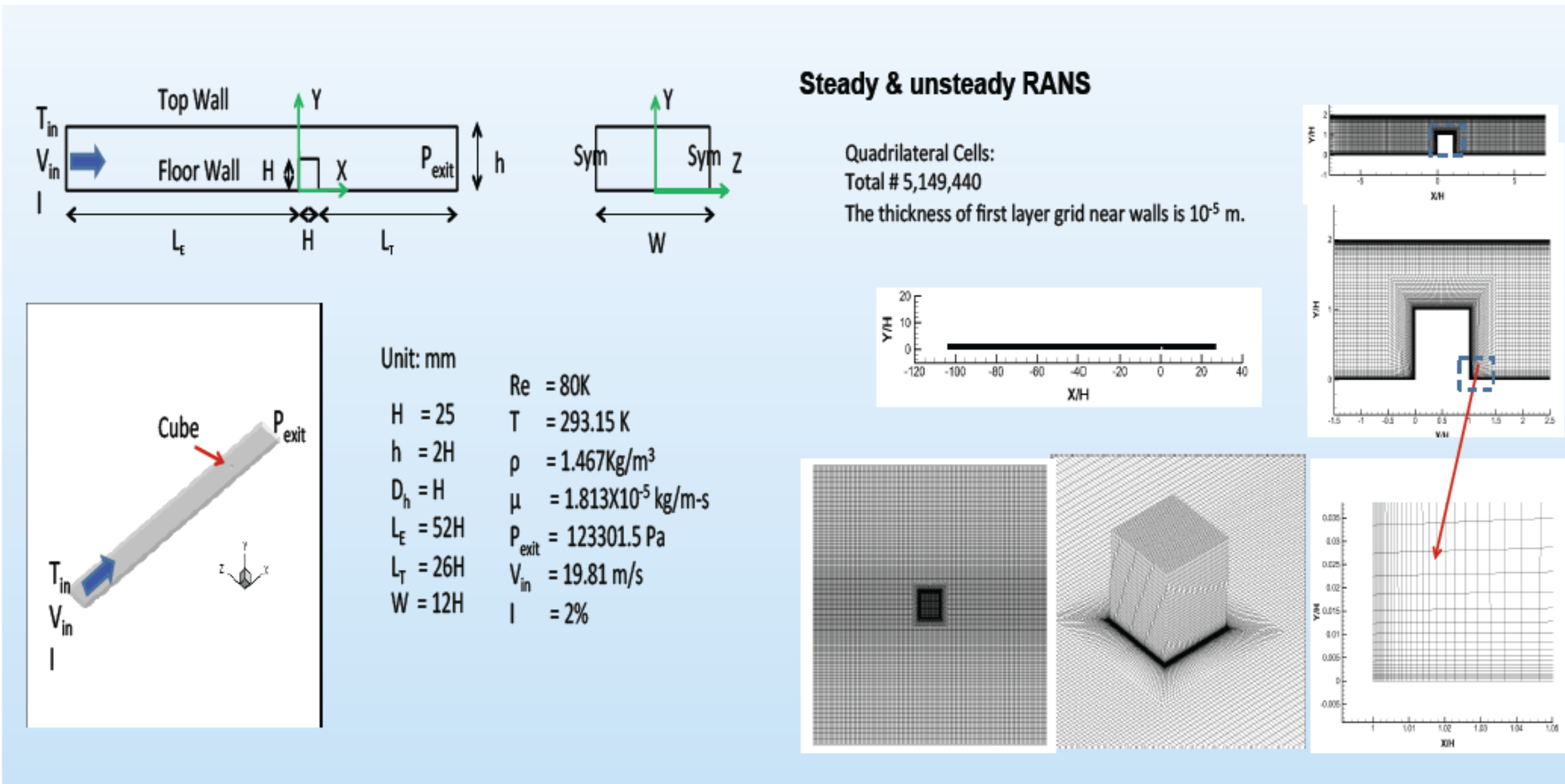
Contrast conjugate vs. non-conjugate CFD analysis in the predicted heat transfer.

Show complications induced by unsteady heating and cooling.

Summarize.

Assess Turbulence Models: Flow over Wall-Mounted Cube

Objective: Assess steady & unsteady RANS: SST, RSM-τω and LES (lattice Boltzmann) for a more complicated problem.
Why? Must know when unsteady RANS and LES are needed and why.



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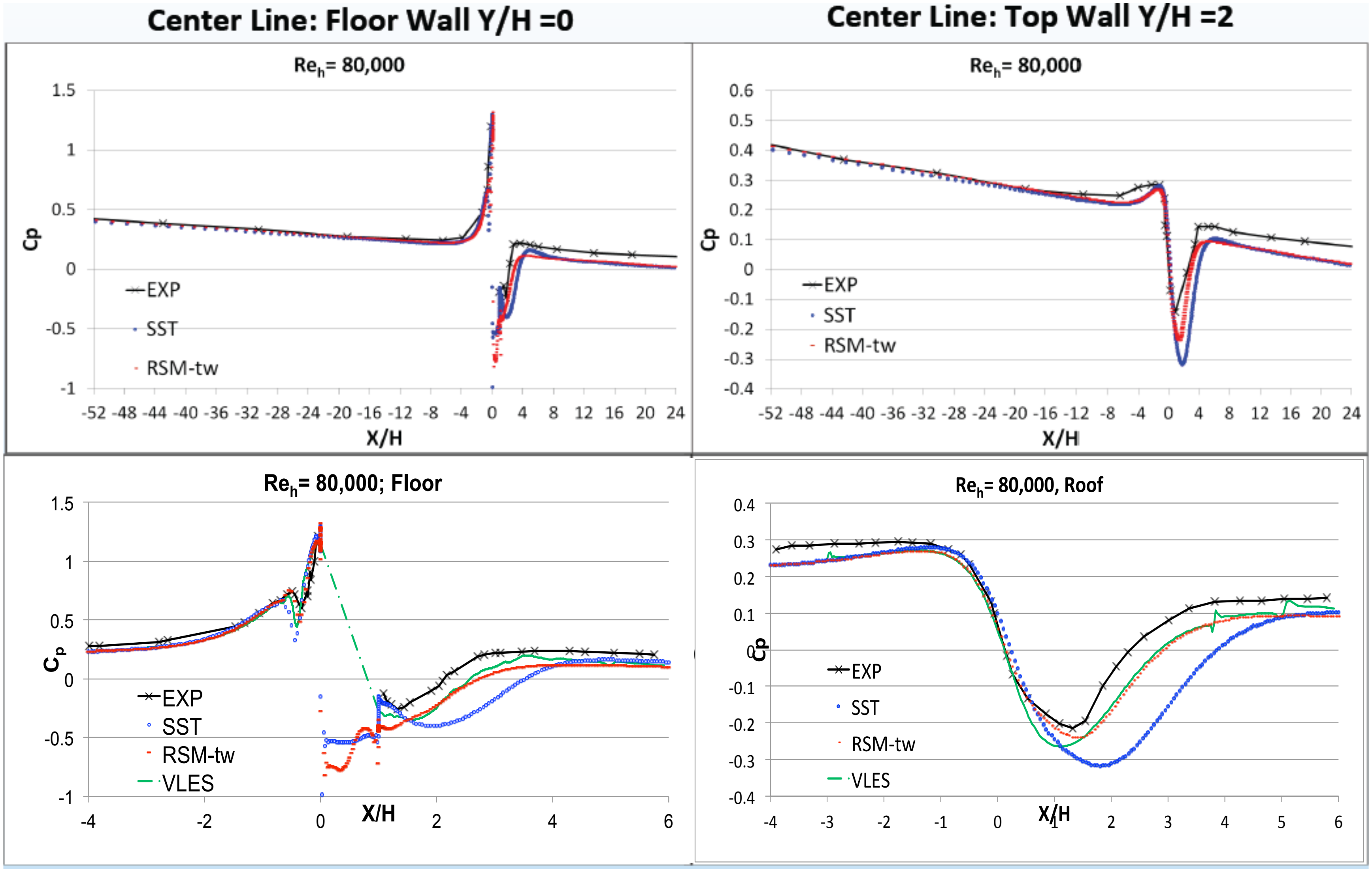
Assess Turbulence Models: Flow over Wall-Mounted Cube



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Assess Turbulence Models: Flow over Wall-Mounted Cube

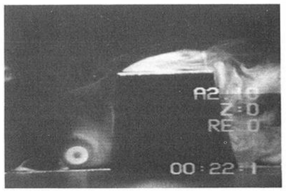


Fig. 7(a)

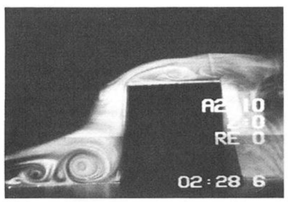
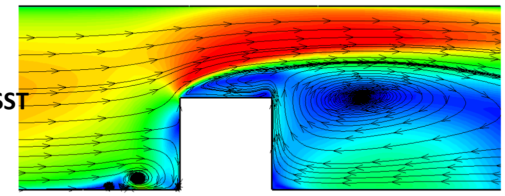


Fig. 7(b)

Fig. 7 Laser-sheet visualization of the flow in front of a cube (plane $z/H = 0$)

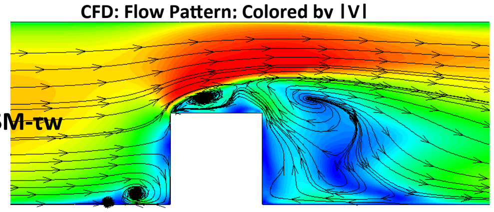
Martinuzzi 1993 Fig 7
1 or 2 possible models

SST



CFD: Flow Pattern: Colored by |V|

RSM-τw



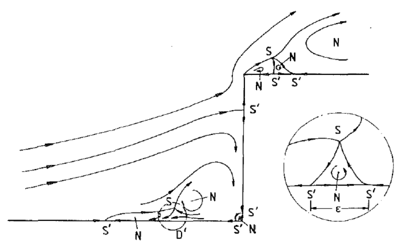
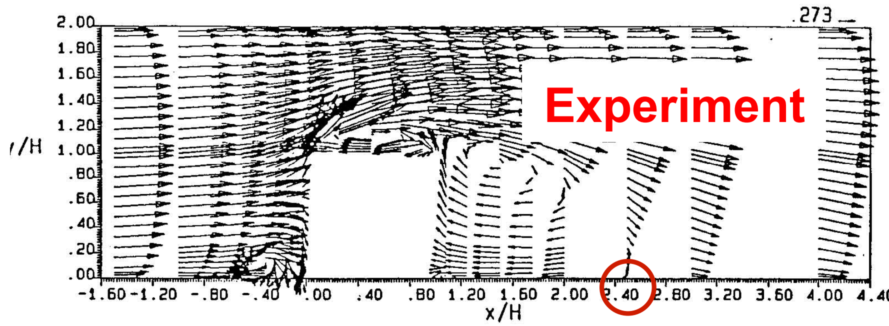
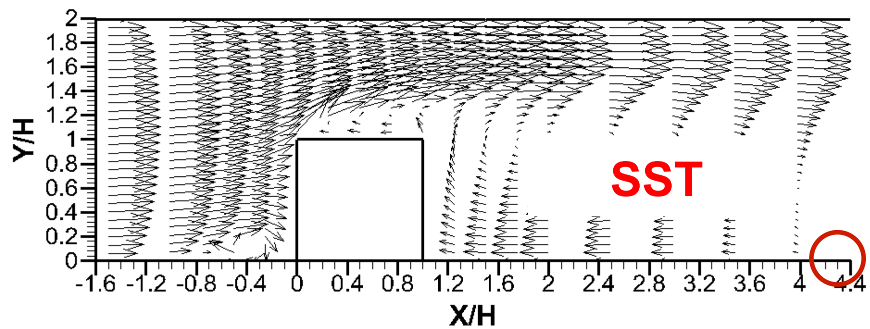
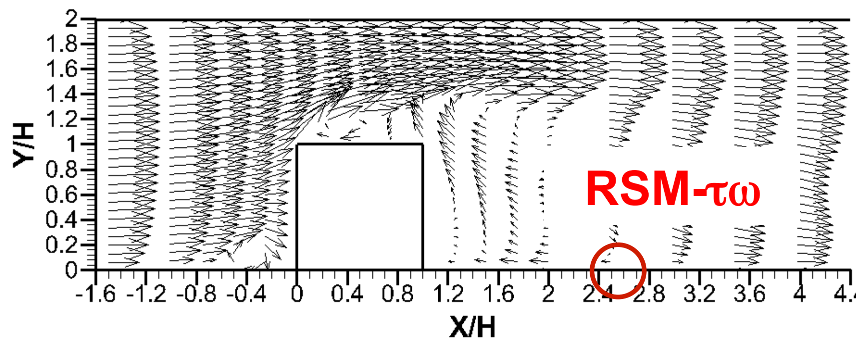


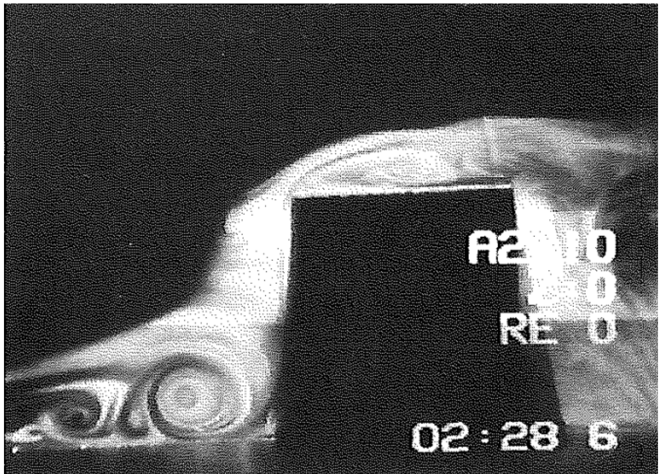
Fig. 8 Schematic representation of the mean-streamline model upstream of a cube in the $z/H = 0$ plane



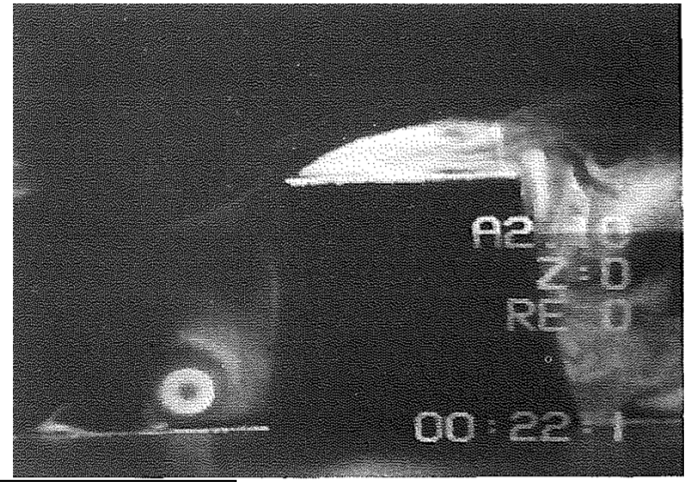




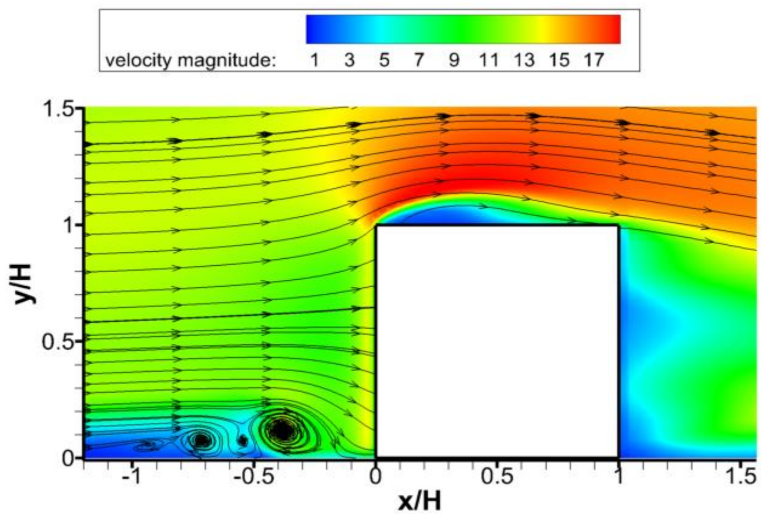
Assess Turbulence Models: Flow over Wall-Mounted Cube

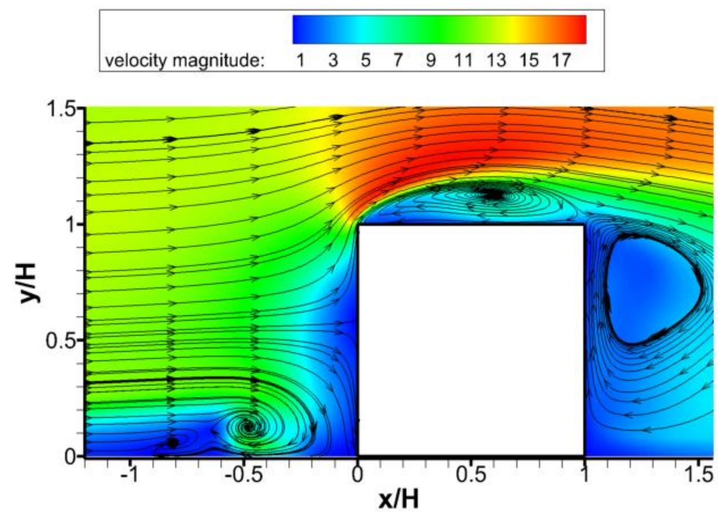


Experiment (Martinuzzi & Tropea, 1993)



VLES via LBM with 900 nodes (Present study)





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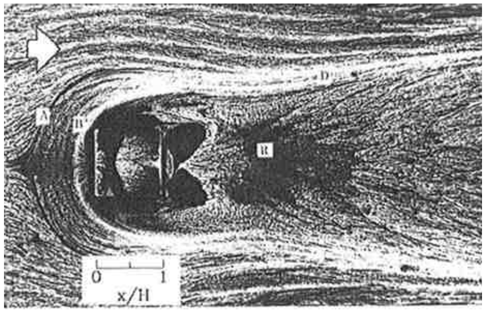
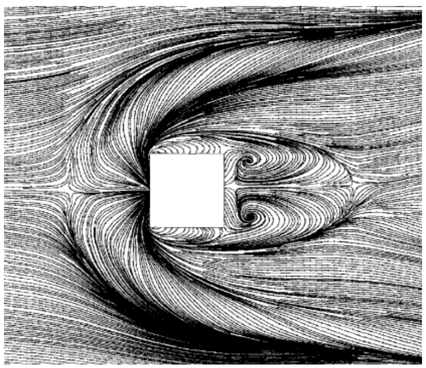
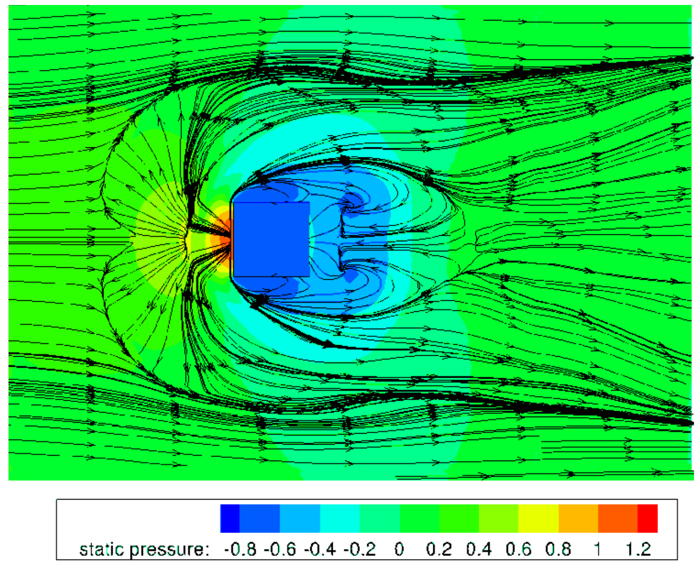
Assess Turbulence Models: Flow over Wall-Mounted Cube

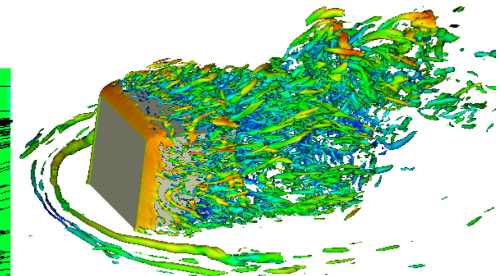


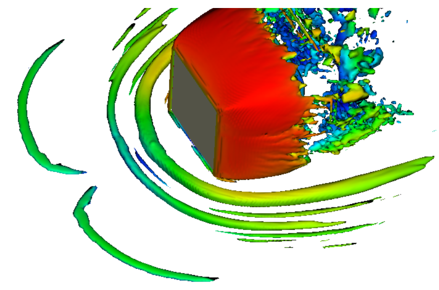
Martinuzzi & Tropea, 1993 (Experiment)



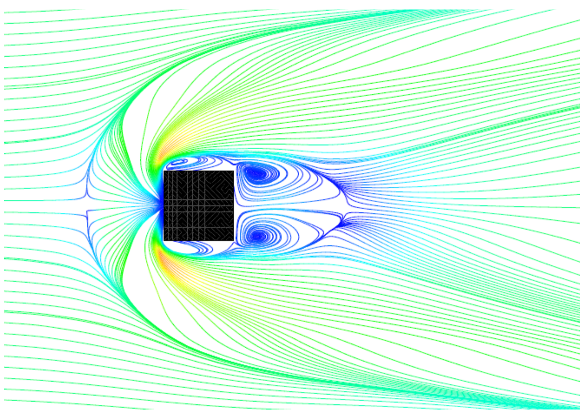
Shah & Ferziger, 1997







Present study: VLES simulation



Krajnovic and Davidson, AIAA 2002

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Assess Turbulence Models: Flow over Wall-Mounted Cube

Summary:

- Steady RANS cannot yield correct solutions for this configuration – not even stress-omega full Reynolds stress model that integrates to the wall.

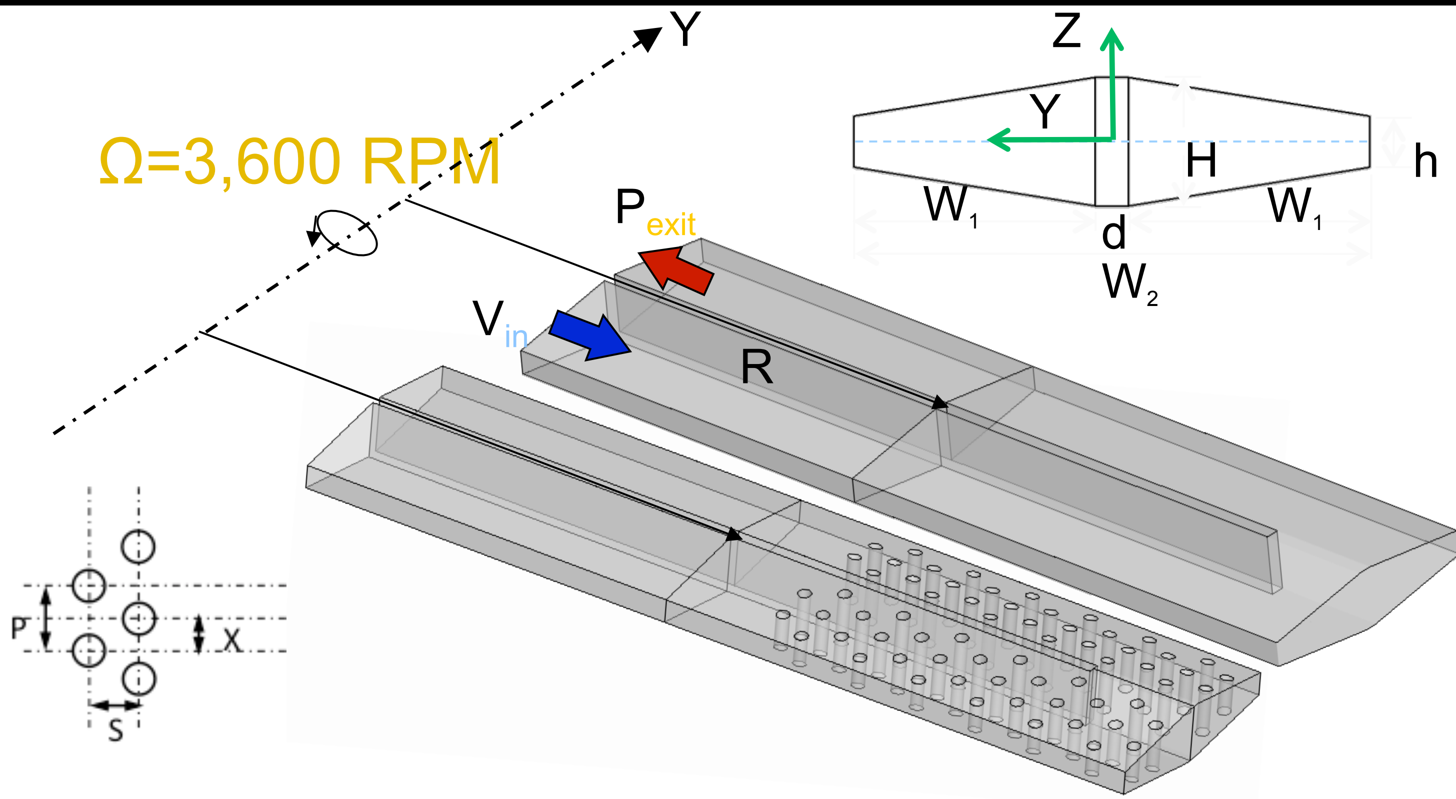
- Unsteady RANS produced reasonable results.

- Exploring difference between time-averaged results from large-eddy simulation (LES) and unsteady RANS based on stress-omega.

- Goal of studies on the cube and other configurations is to understand what flow and heat-transfer mechanisms could or could not be predicted by steady RANS, unsteady RANS, and LES.

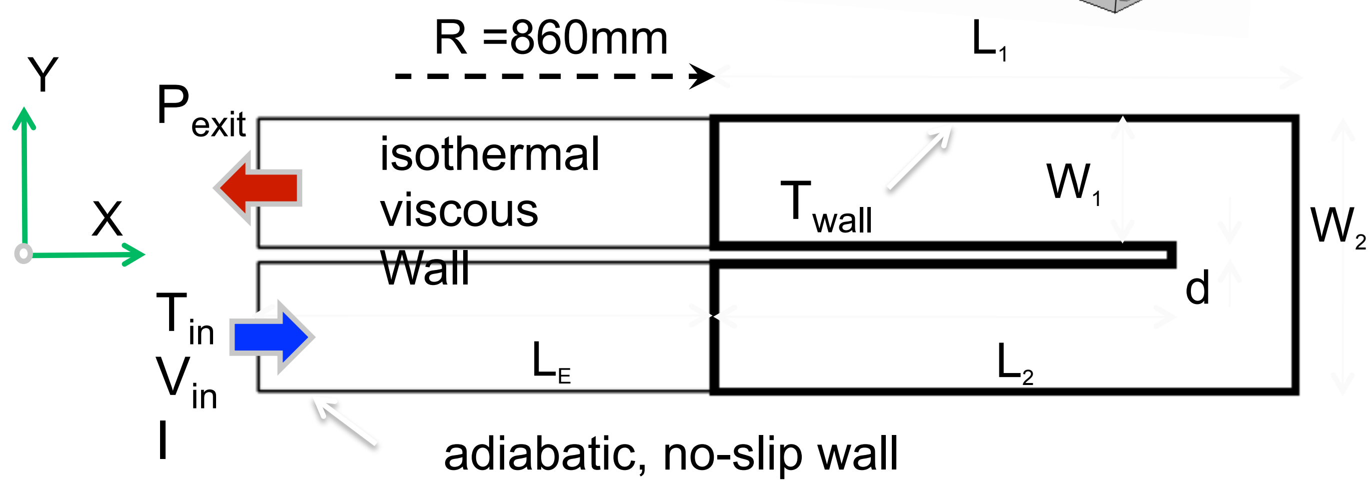
Flow in a U-Duct with a Trapezoidal Cross Section

(Siemens: Crawford, Marra, Prakash, Brown, Lee)



Unit: mm

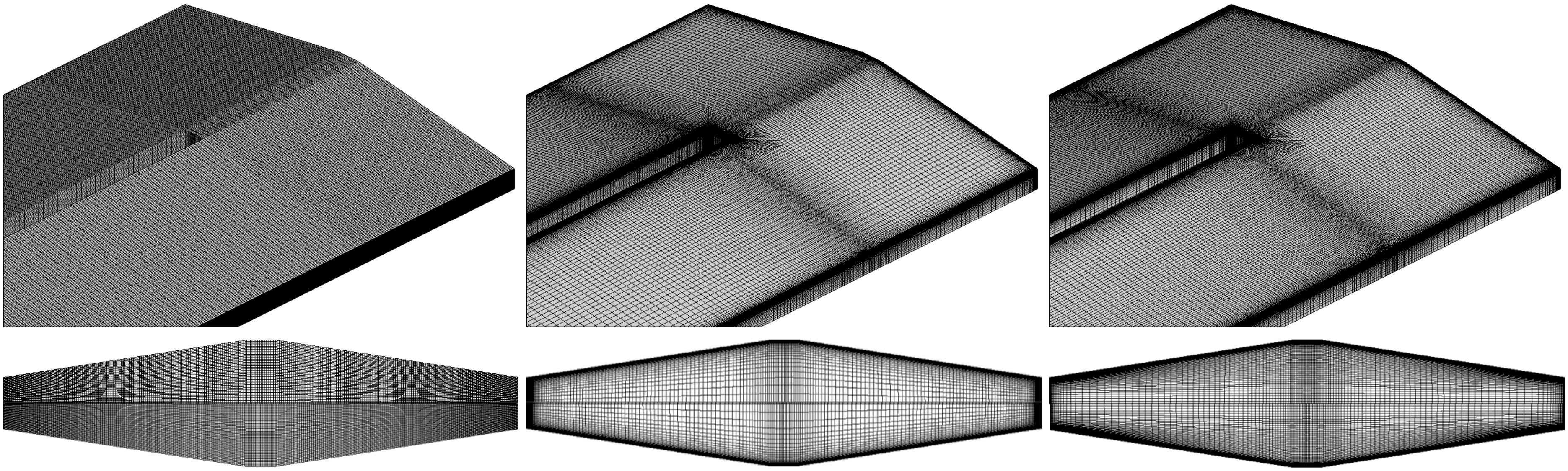
R	= 860
D_h	= 29.04
d	= 6.35
L_1	= 246
L_2	= 192
W_1	= 54.24
W_2	= 114.83
H	= 28.48
h	= 11.72
L_E	= 192



Re	= 37K, 100K
T_{in}	= 673.15 K
T_{wall}	= 1273.15 K
P_{exit}	= 2,300,000 Pa
V_{in}	= 3.48, 9.41 m/s
I	= 5%

Flow in a U-Duct with a Trapezoidal Cross Section

(Siemens: Crawford, Marra, Prakash, Brown, Lee)

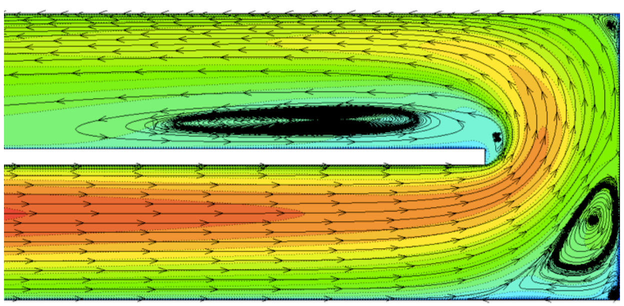


2,530,560 cells
for half domain

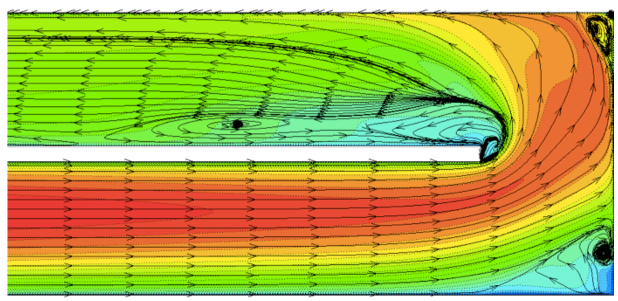
3,694,080 cells
for half domain

4,129,920 cells
for half domain

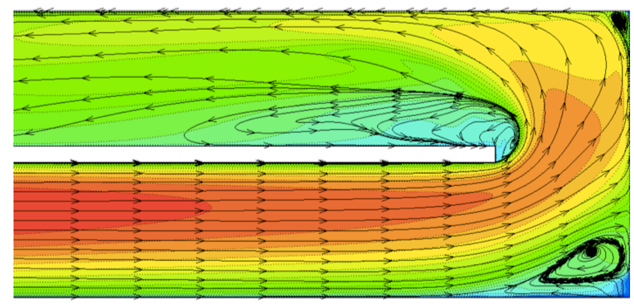
k-ε



SST



RSM-τw



Flow in a U-Duct with a Trapezoidal Cross Section

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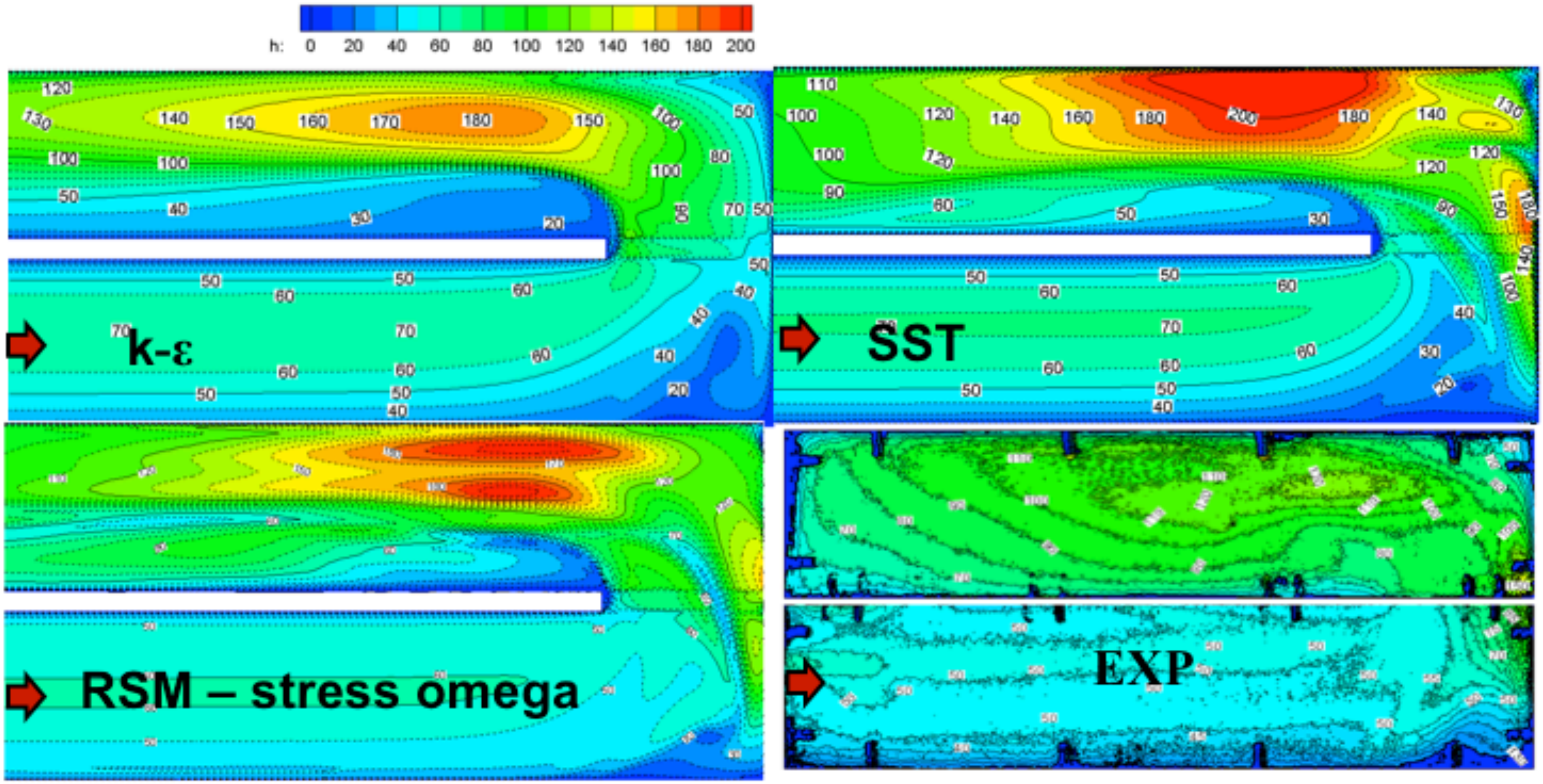

Experiments from Mingking Chyu, Pitt

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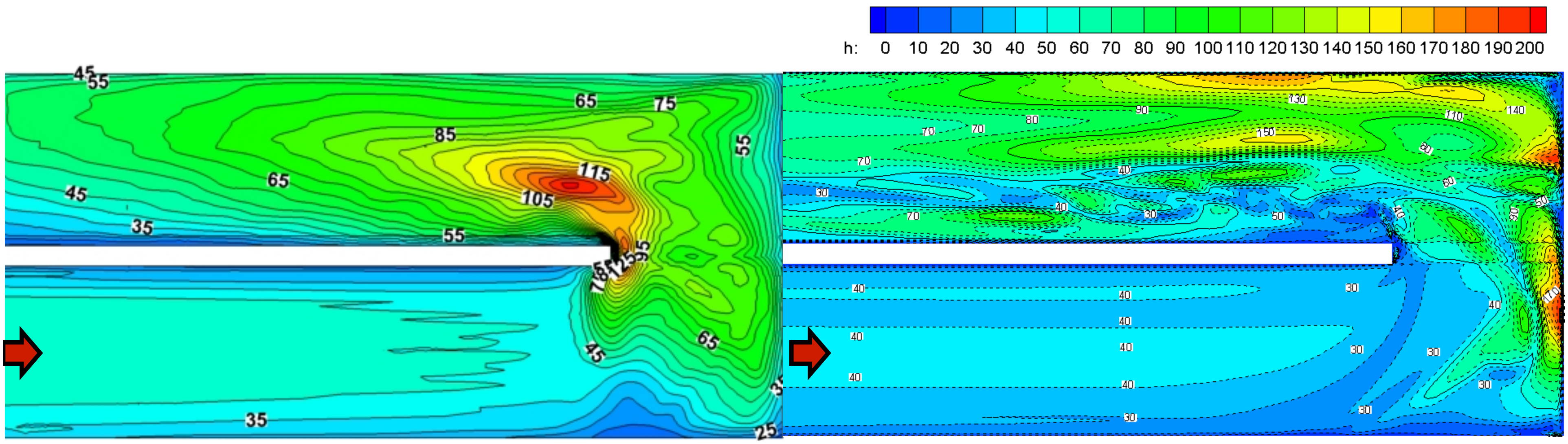
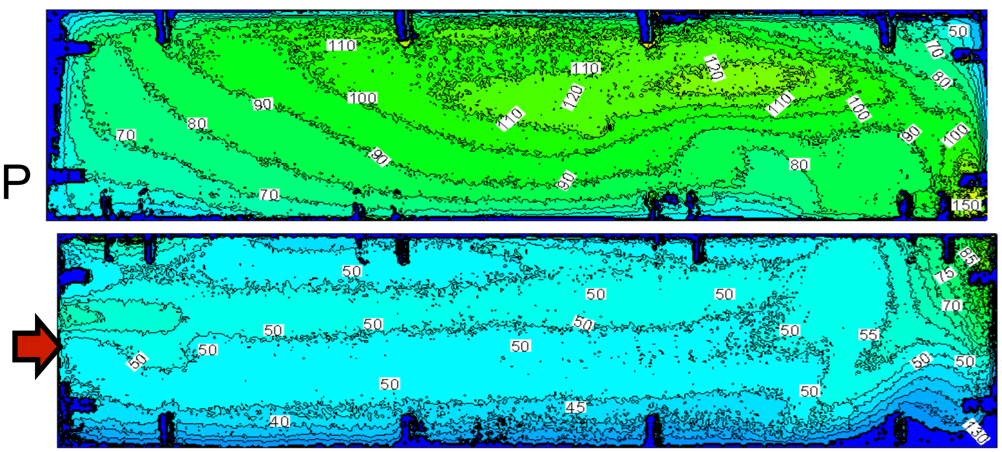
Flow in a U-Duct with a Trapezoidal Cross Section
(Siemens: Crawford, Marra, Prakash, Brown, Lee)



VLES

Unsteady RANS (RSM –stress omega)

EXP



Experiments from Mingking Chyu, Pitt

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Outline of Talk

Show challenges from two validation studies.

Assess the accuracy of a measurement technique widely used to generate benchmark data to validate CFD.

Contrast conjugate vs. non-conjugate CFD analysis in the predicted heat transfer.

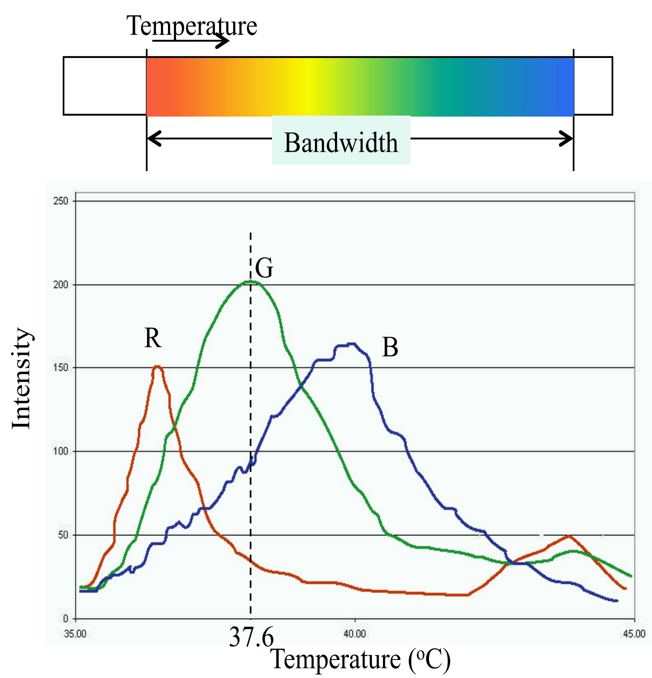
Show complications induced by unsteady heating and cooling.

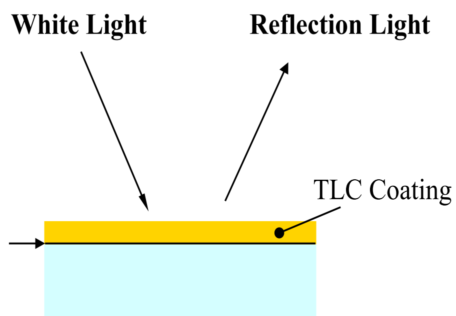
Summarize.

Assess Accuracy of Measurements of Heat Transfer

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

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





Thermochromic Liquid Crystal (TLC)

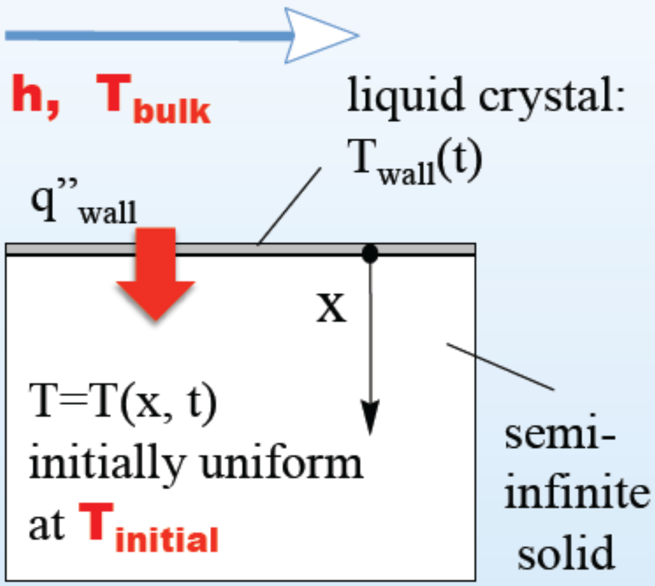
green light reflection gives the best intensity.

Surface temperature is measured as a function of time.

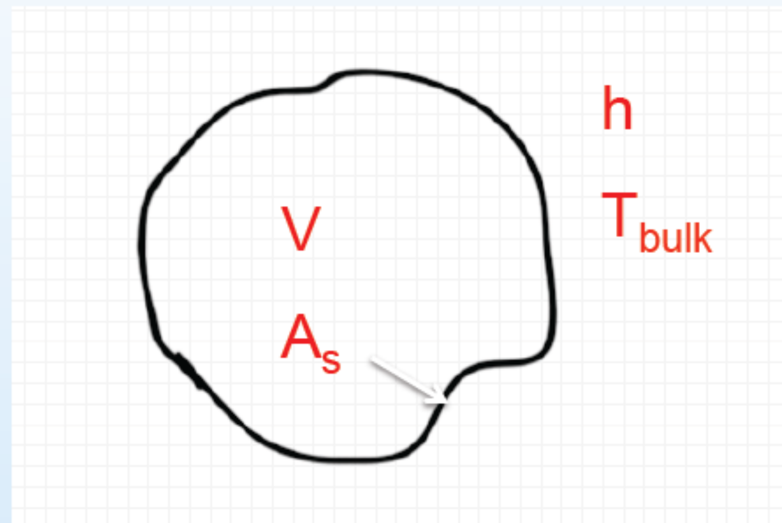
The heat transfer coefficient is then inferred from either 1-D or 0-D exact solutions.

Assess Accuracy of Measurements of Heat Transfer

1-D Model



0-D Model



Assumptions:

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

Experimental Technique:

- Measures T = f(time)
- Calculate h at time when T = 37.6 ^{o}C

Assess Accuracy of Measurements of Heat Transfer

Objectives are to determine:

- How good are the assumptions for practical problems with highly variable h and T_{bulk}?
- Does the measured h under transient conditions match h under steady-state conditions?

Approach

- Choose problems where h & T_{bulk} vary appreciably: pin fins
- Perform 3-D unsteady RANS to get $T_{wall} = f(t)$.
- Use 1-D & 0-D exact solution to get h_{EFD}.
- Assess h_{EFD} with $h_{CFD} = q''_{wall} / [T_{wall}(t) - T_{bulk}]$ & with h_{CFD} from steady RANS.

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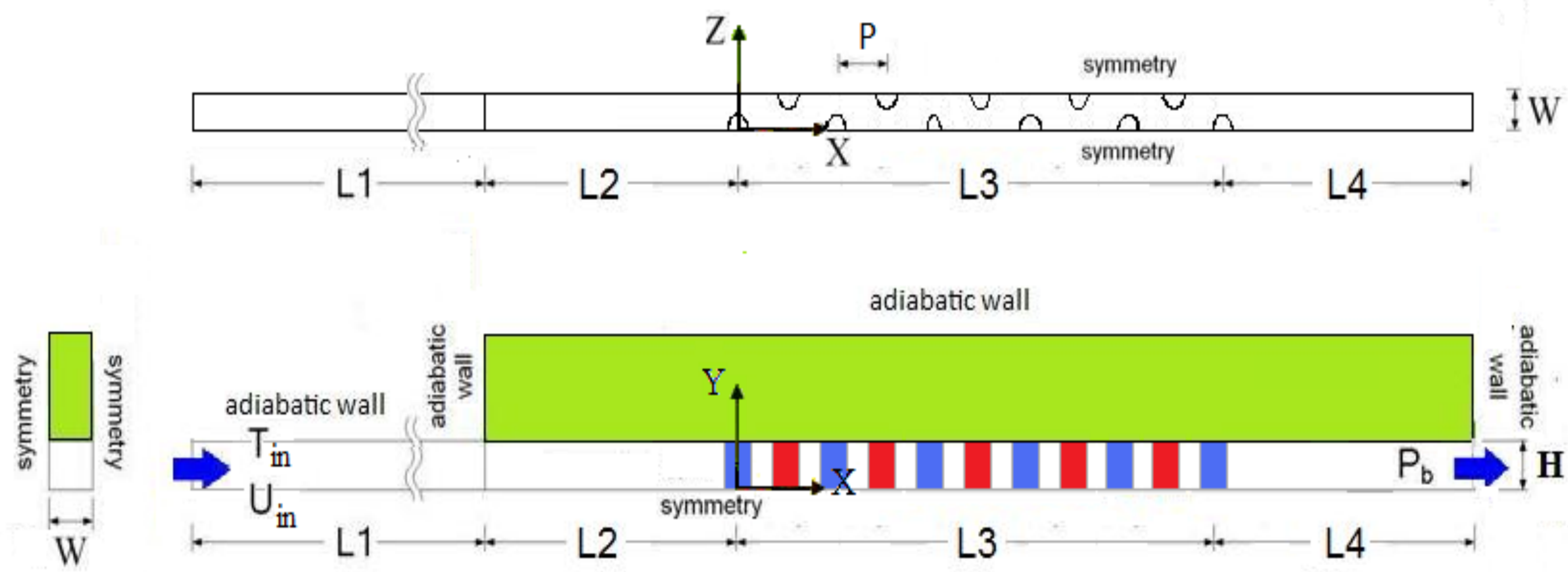
Assess Accuracy of Measurements of Heat Transfer

Test Problem: staggered array of pin fins in a duct
(not drawn to scale)



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Assess Accuracy of Measurements of Heat Transfer

Formulation

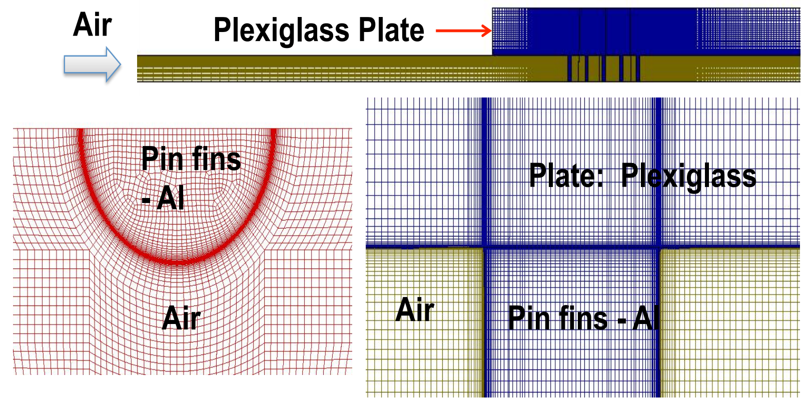
- **Gas Phase:** ensemble-averaged continuity, momentum (compressible Navier Stokes) and energy with thermally perfect gas, C_p k = f(T), Sutherland for m, and SST turbulence model (without low Re correction)
- **Solid Phase:** Fourier law with constant k_{solid}

Code : Fluent 13.0

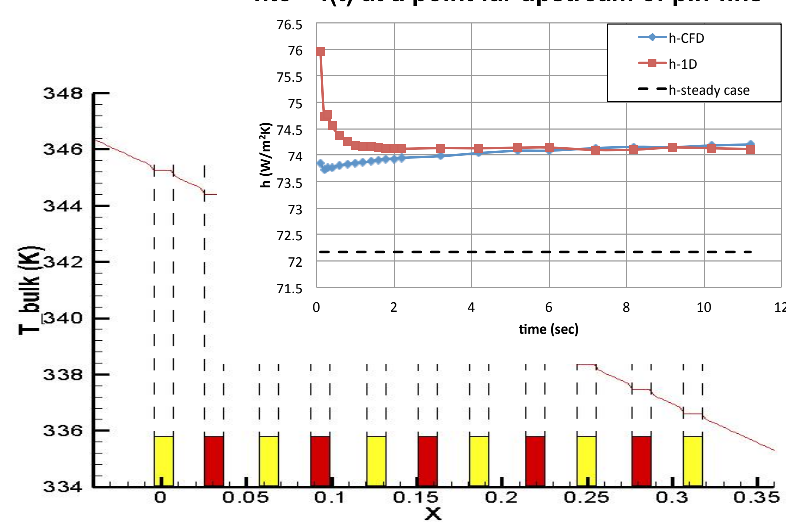
Algorithm

- **Steady RANS:** COUPLED with 2nd order for pressure, 2nd order upwind for all other equations.
- **Unsteady RANS:** 2nd order implicit with 30 iterations per time step.

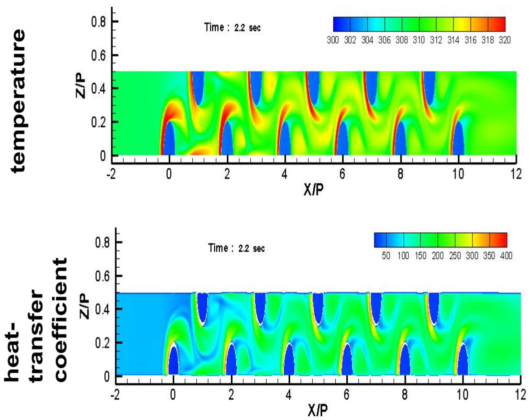
NOTE: PISO required excessively small time step sizes to be stable (blows up even with $\Delta t = 10^{-5}$ seconds). With the implicit coupled scheme, you can use 10^{-3} seconds and higher.

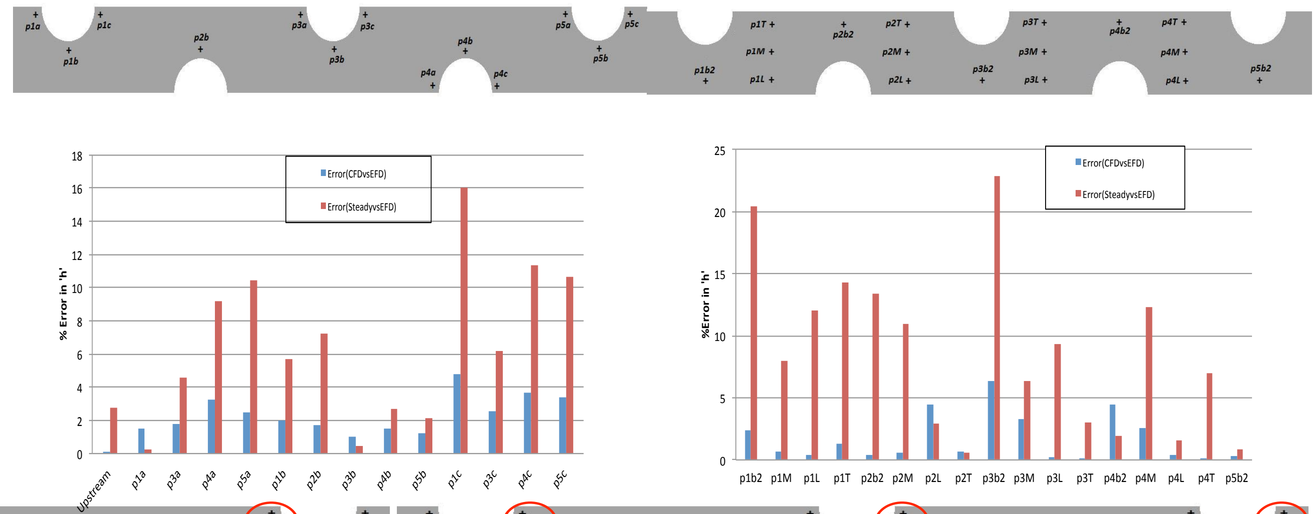


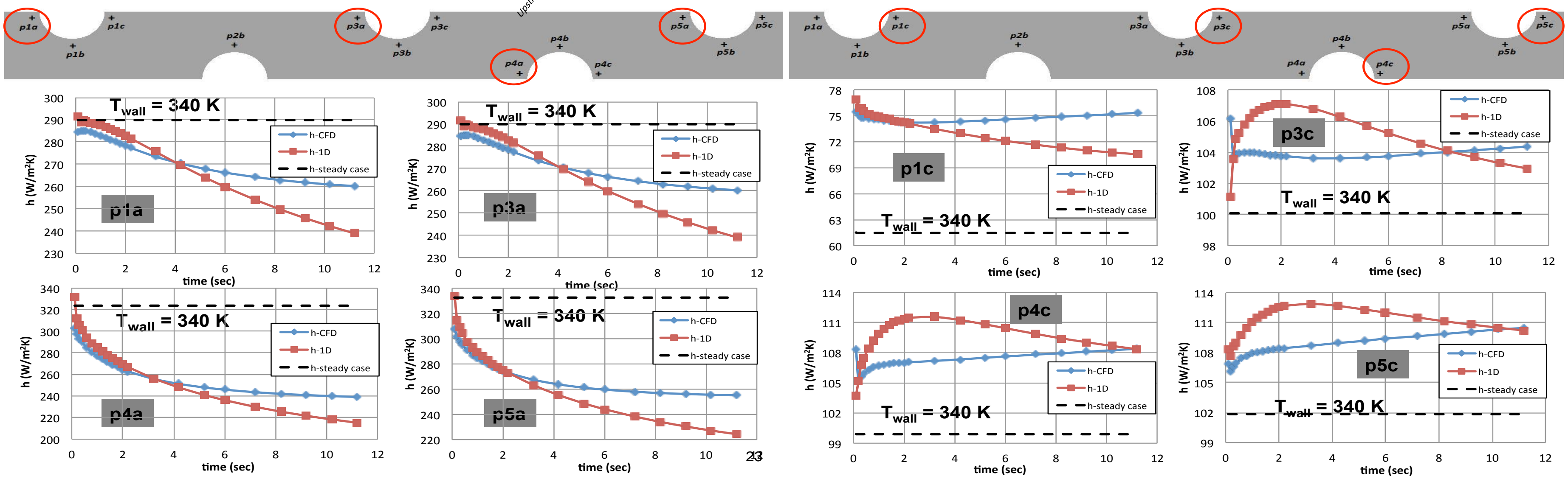
Total Cells = 4,139,030 (**channel:** 1501 along X, 35 along Y, 38 along Z from X = – L2 onwards and 16 cells in Z upstream of X = – L2; **pin fin**: 111 in the azimuthal direction with 54,460 cell in each half pin; **plate:** total = 1,893,430 with 1,422 along X, 35 along Y, 38 along Z.



Assess Accuracy of Measurements of Heat Transfer







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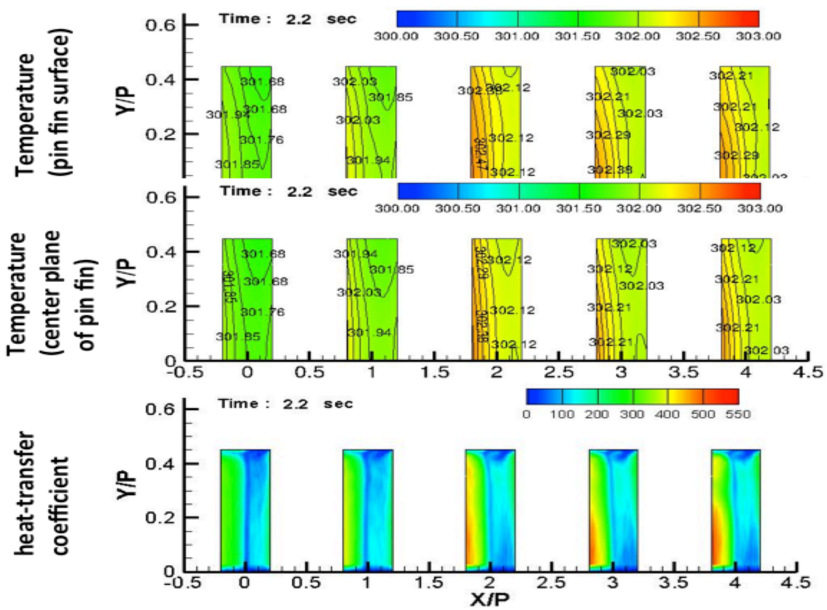
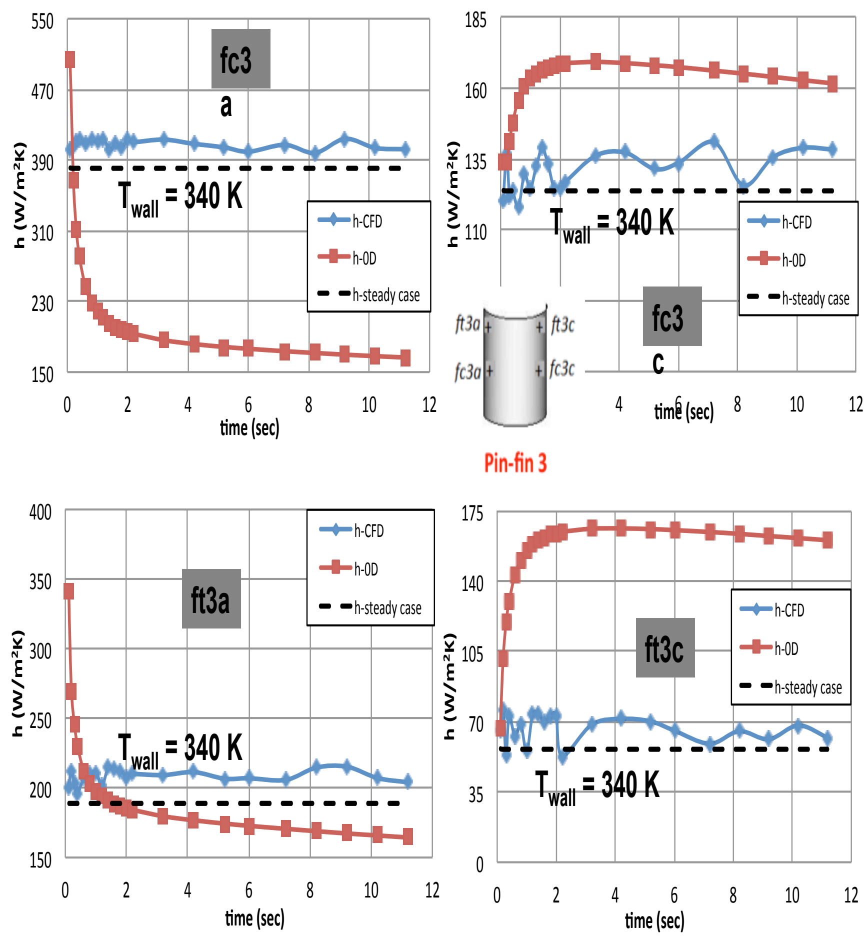
Assess Accuracy of Measurements of Heat Transfer



T is fairly uniform (Bi < 0.1)

h is highly non-uniform!
Thus, though Bi << 1, 0-D assumption is terrible.



Error: up to 200%

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Assess Accuracy of Measurements of Heat Transfer

Summary:

1-D exact solution: plexiglass plate

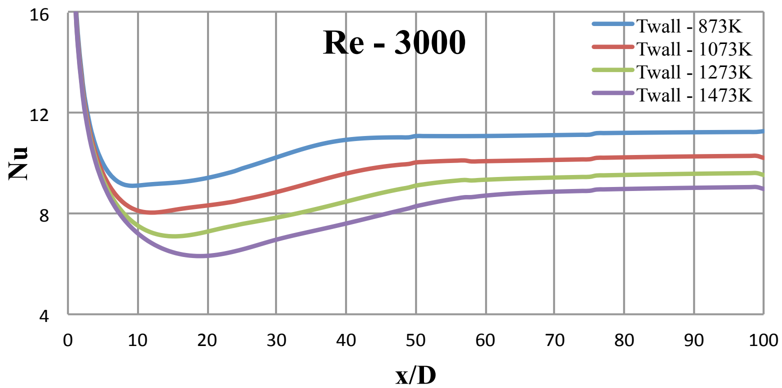
- |h-1D – h-CFD|/|h-CFD| < 5%
- |h-1D – h-steady CFD|/|h-steady FD| as high as 23% at locations checked

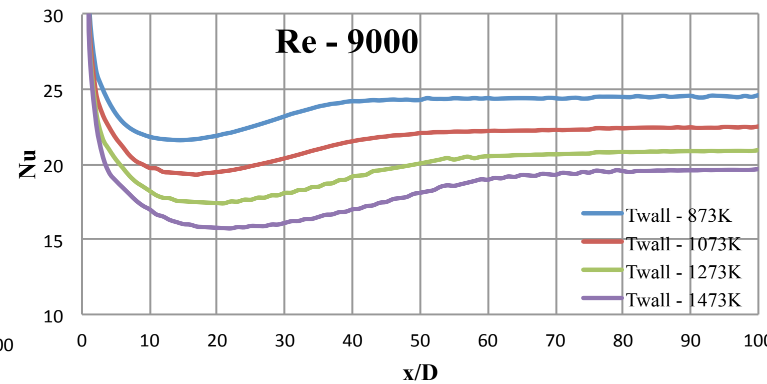
0-D exact solution: aluminum pin fins

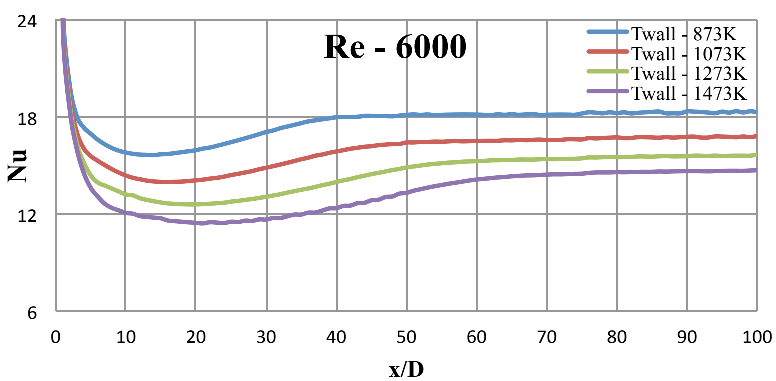
- |h-1D – h-CFD|/|h-CFD| as high as 200%
- |h-1D – h-steady CFD|/|h-steady FD| as high as 200%
- 0-D exact is good only if h around the object is nearly the same, which is untrue for pin fins.

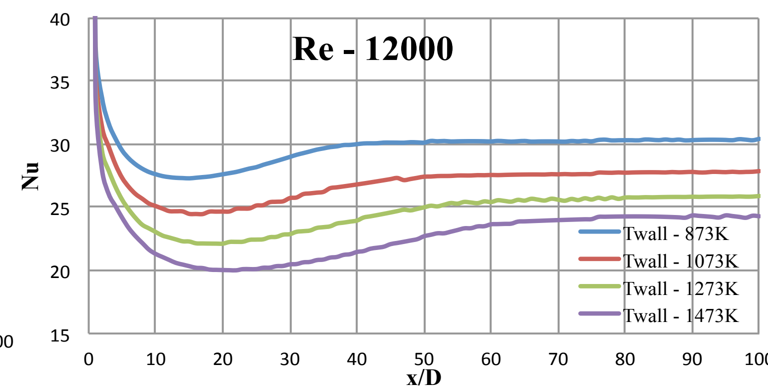
Assess Accuracy of Measurements of Heat Transfer

Summary: $Nu_x = hx/k = f(T\_wall, Re\_inlet, Pr\_inlet, x/D)$









The minimum in Nu shifts downstream with higher wall temperature.
Re = $f(T_b)$ varies by by up to 40% from inlet to x/D = 100.

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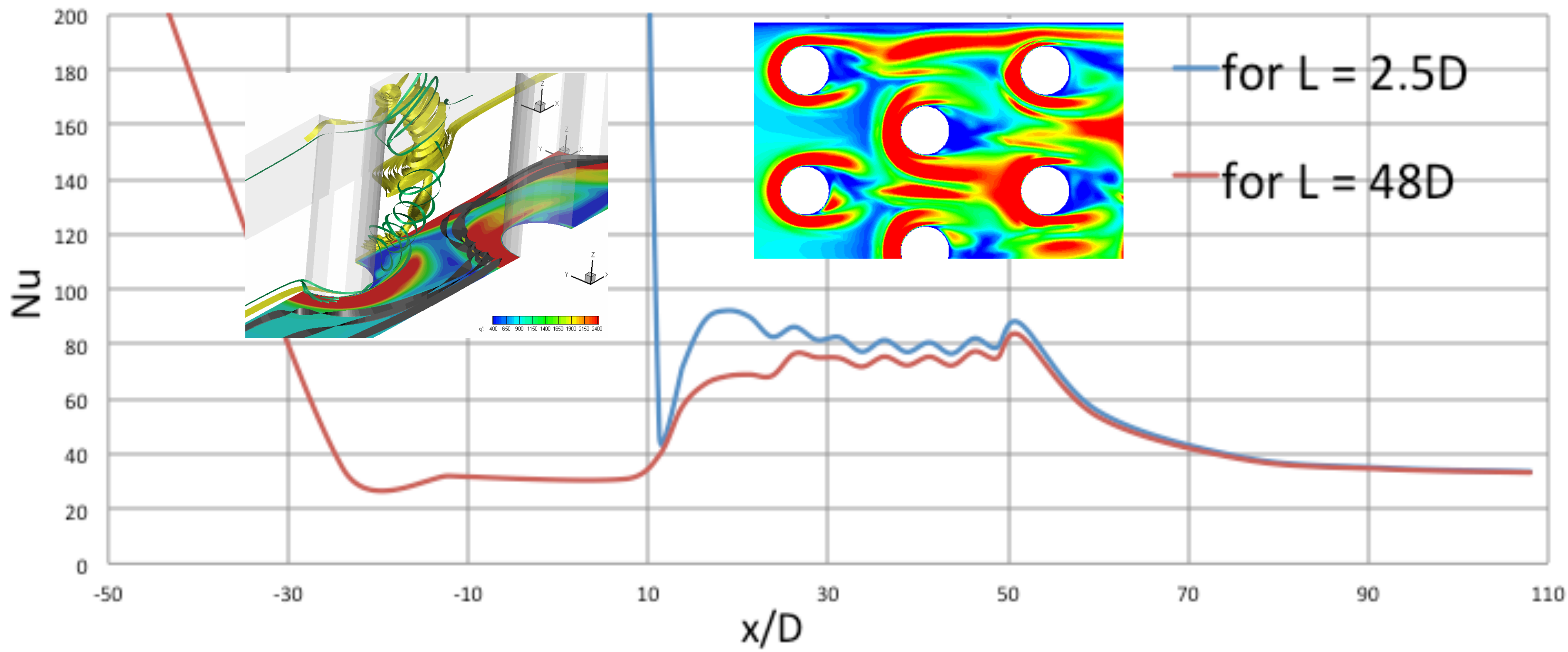
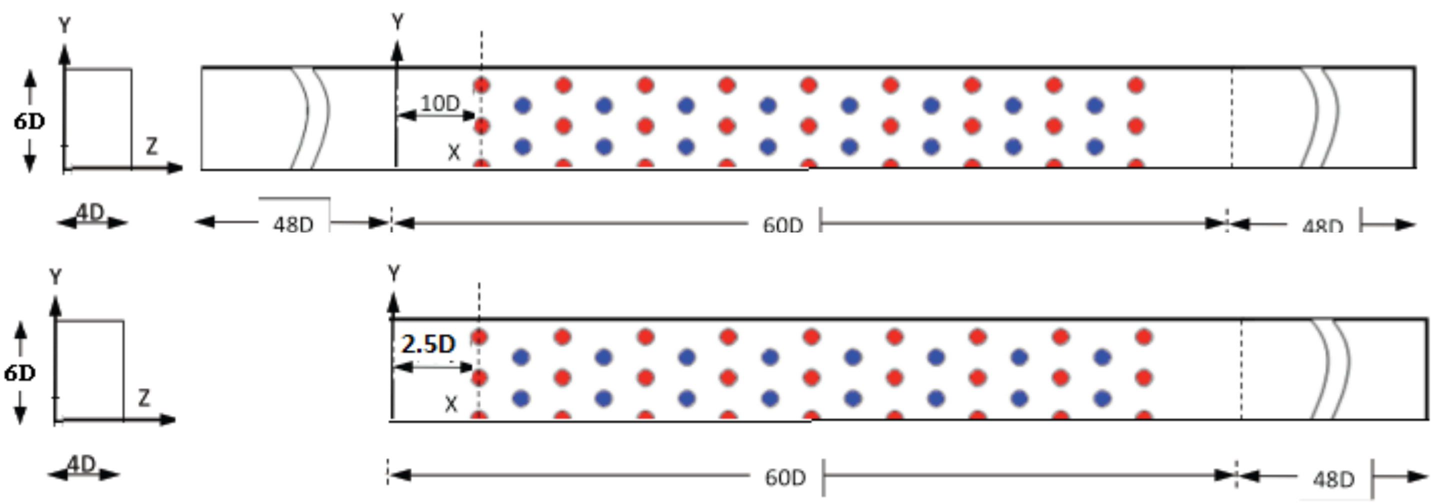
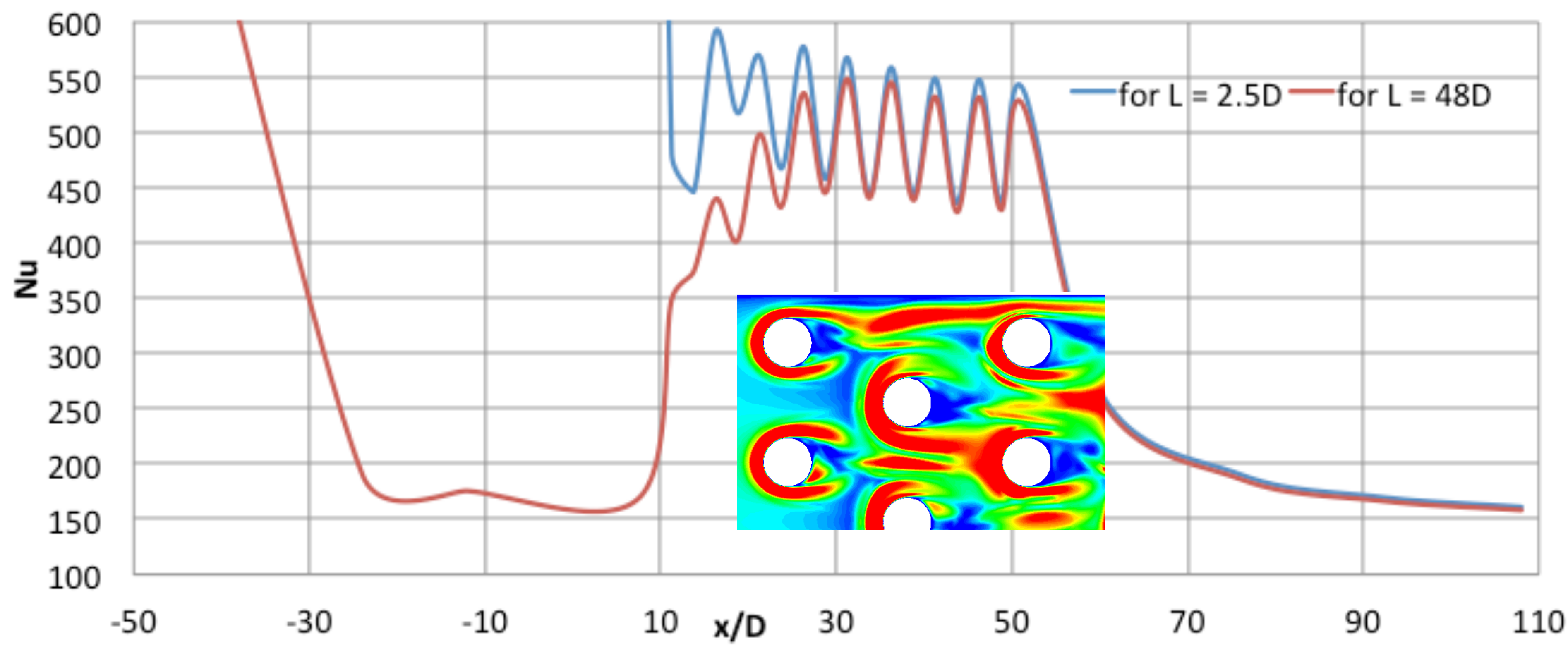
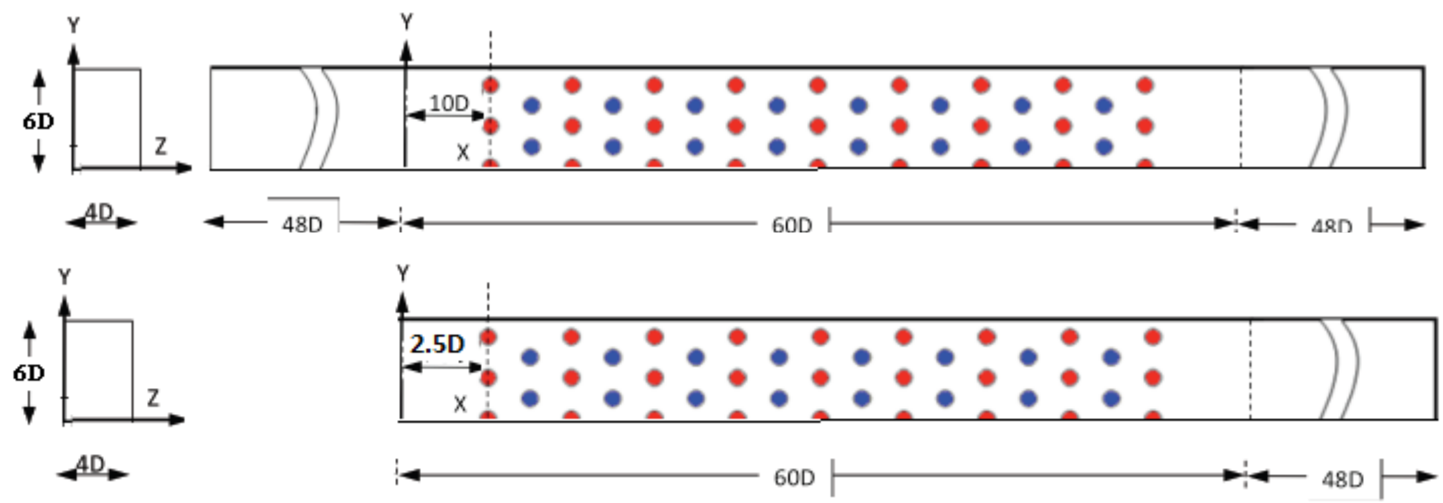


$Re_{Inlet} = 15{,}000 \quad T_{inlet} = 70^{0}C \quad T_{wall} = 40^{0}C \quad P_b = 1bar$





Re_{Inlet} = 150,000 T_{inlet} = 400^{0}C T_{wall} = 900^{0}C P_b =



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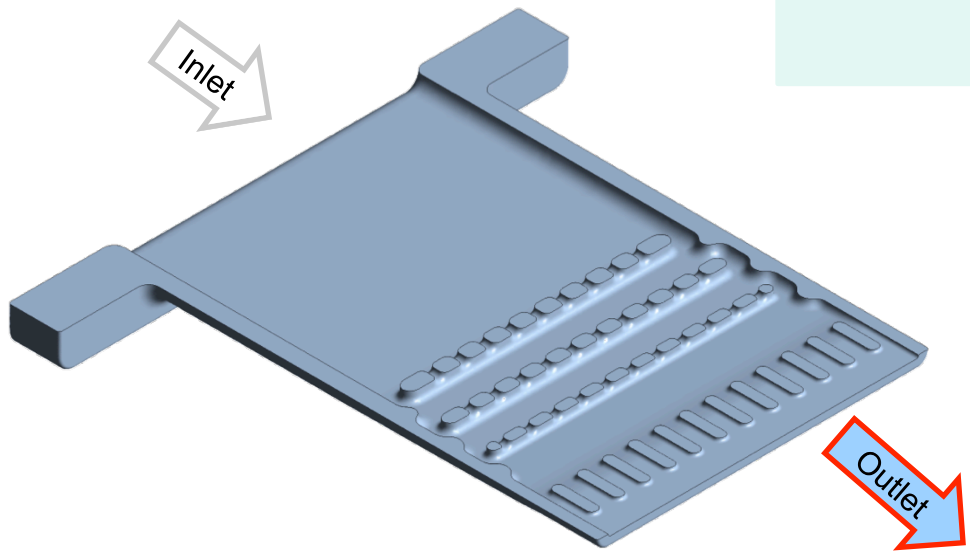
Assess the accuracy of a measurement technique widely used to generate benchmark data to validate CFD.

Contrast conjugate vs. non-conjugate CFD analysis in the predicted heat transfer.

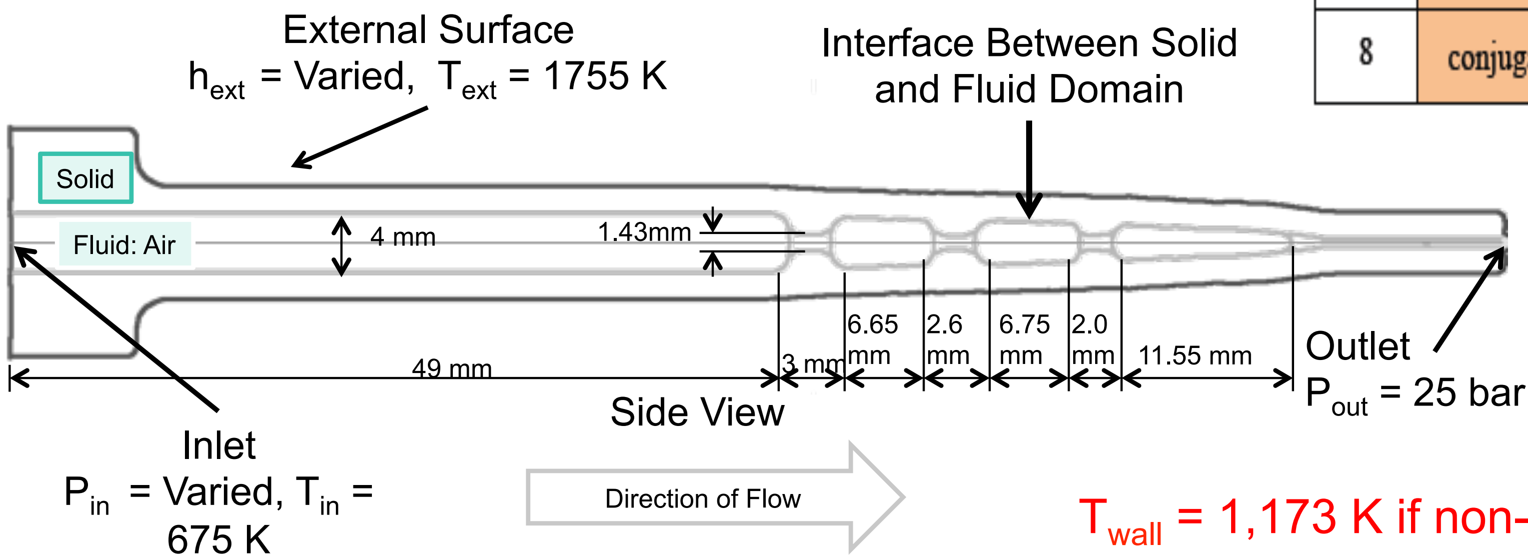
Show complications induced by unsteady heating and cooling.

Summarize.

Conjugate vs Non-Conjugate CFD (Mikro)

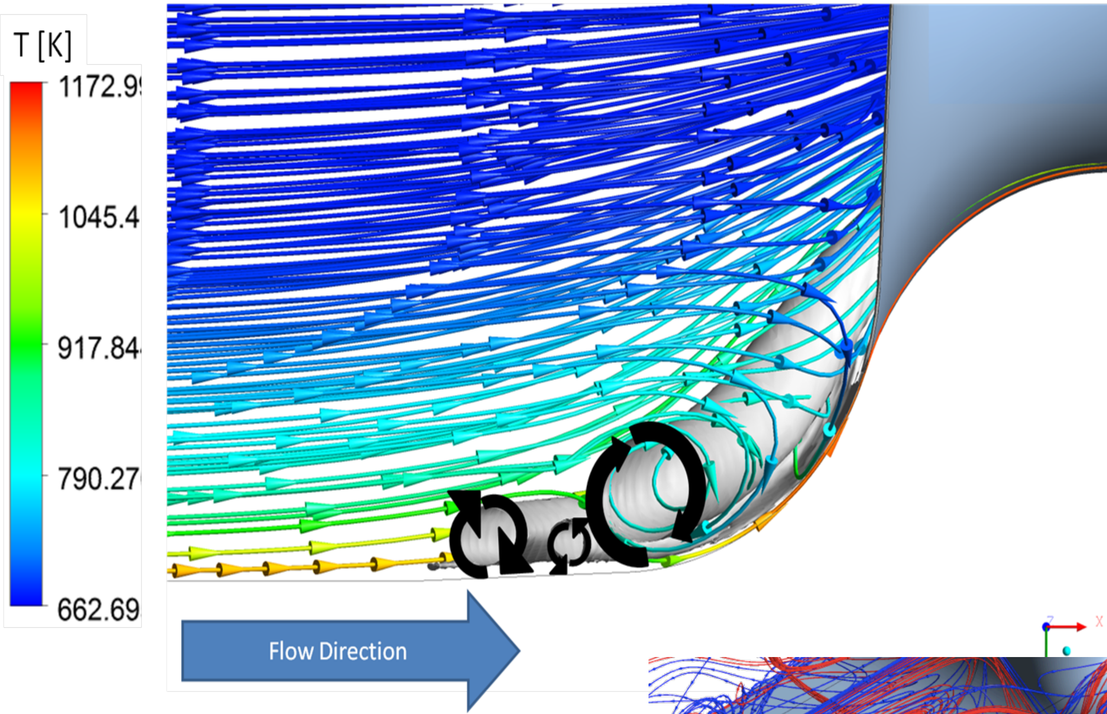


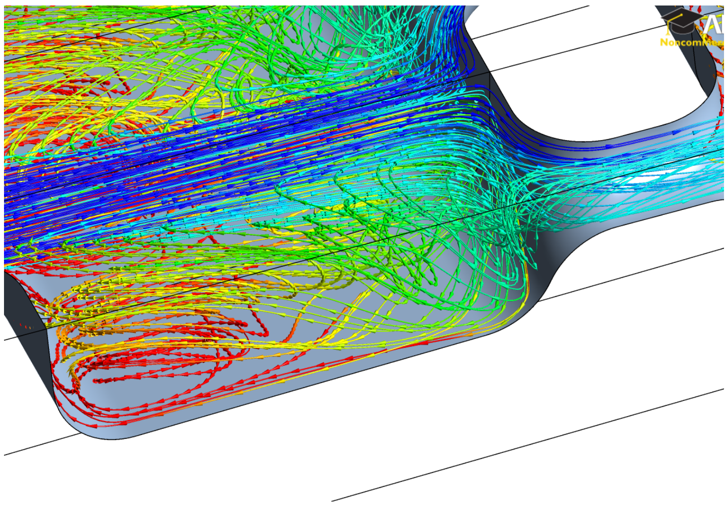
Case No.	Isothermal/ conjugate	ΔP (bars)	$h_{external}$ (W/m^2-K)
1	isothermal	1	N.A.
2	isolthermal	2	N.A.
3	isothermal	3	N.A.
4	isothermal	4	N.A.
5	isothermal	5	N.A.
6	conjugate	4	2,000
7	conjugate	4	4,000
8	conjugate	4	6,000

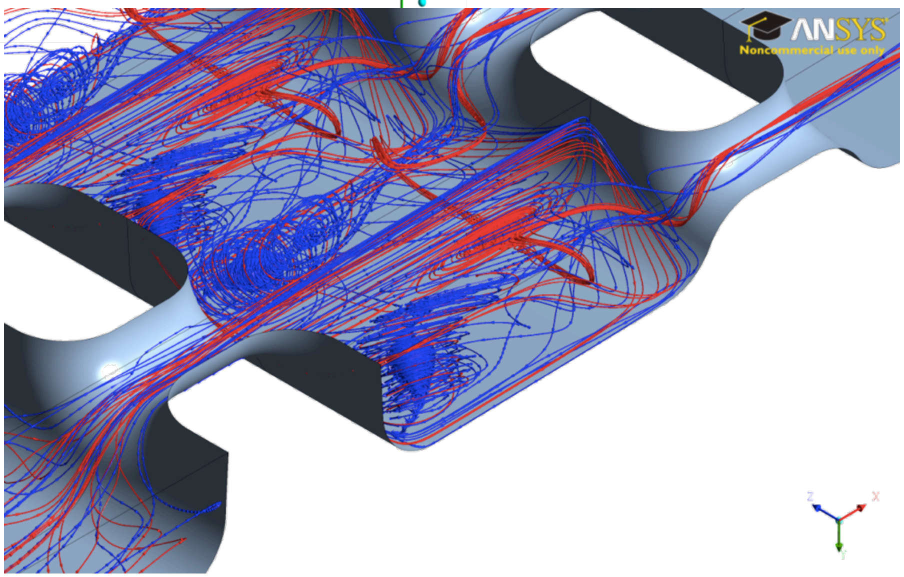


T_{wall} = 1,173 K if non-conjugate

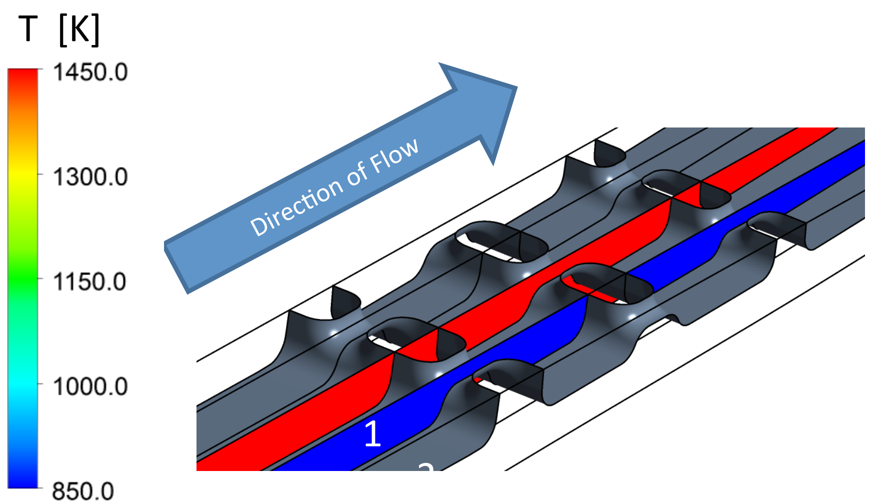
Conjugate vs Non-Conjugate CFD (Mikro)

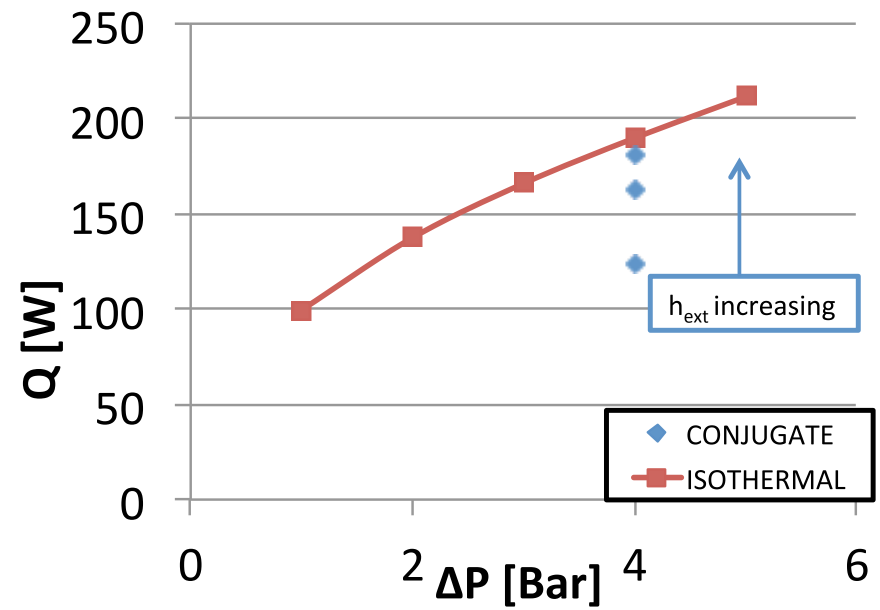


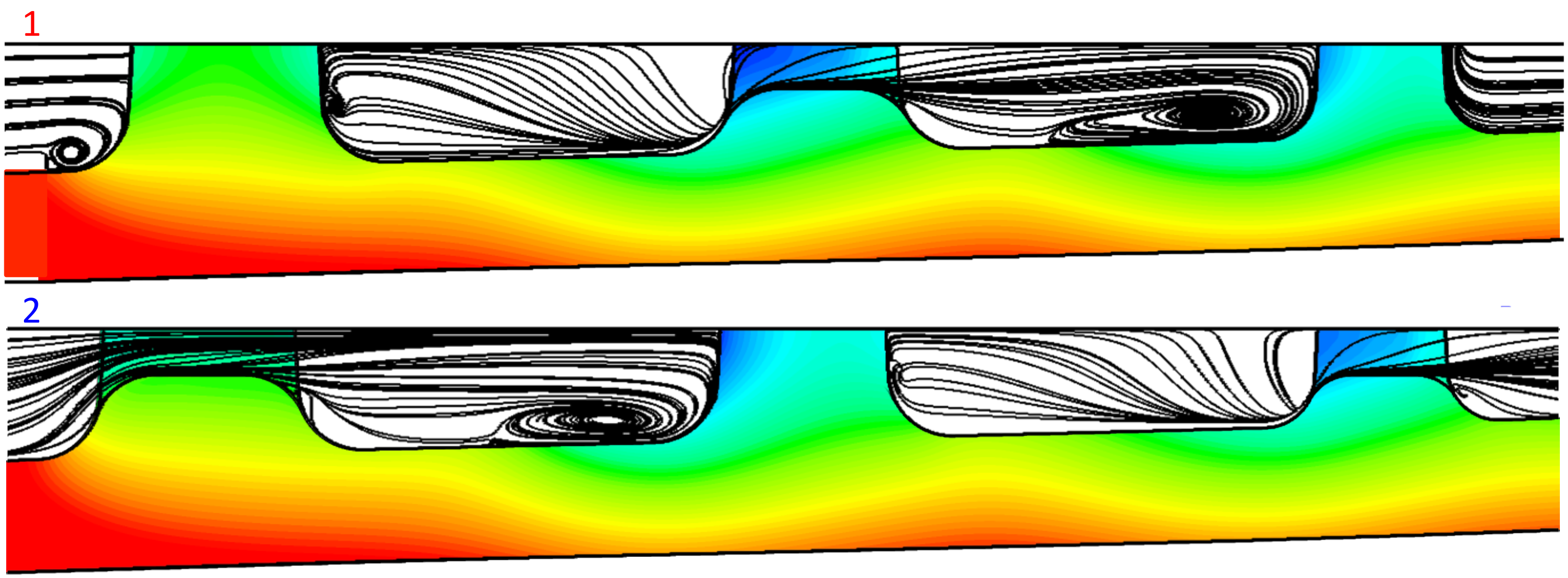




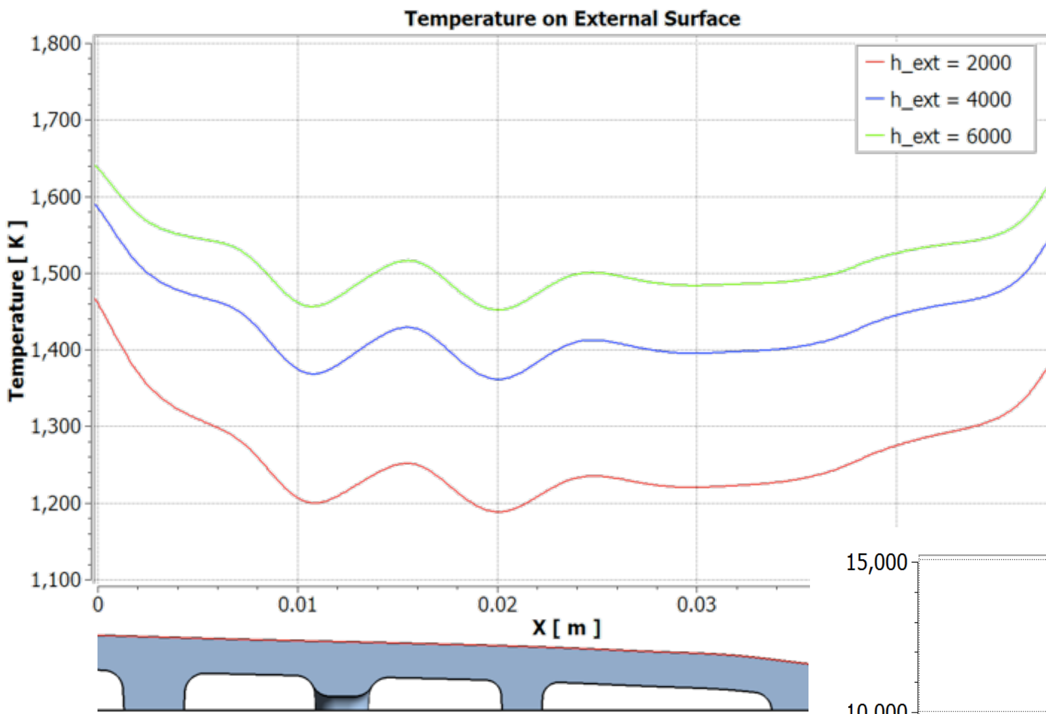
Conjugate vs Non-Conjugate CFD (Mikro)

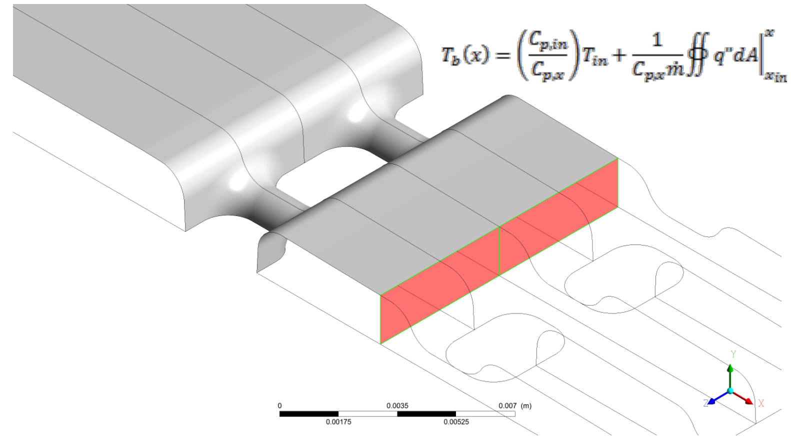


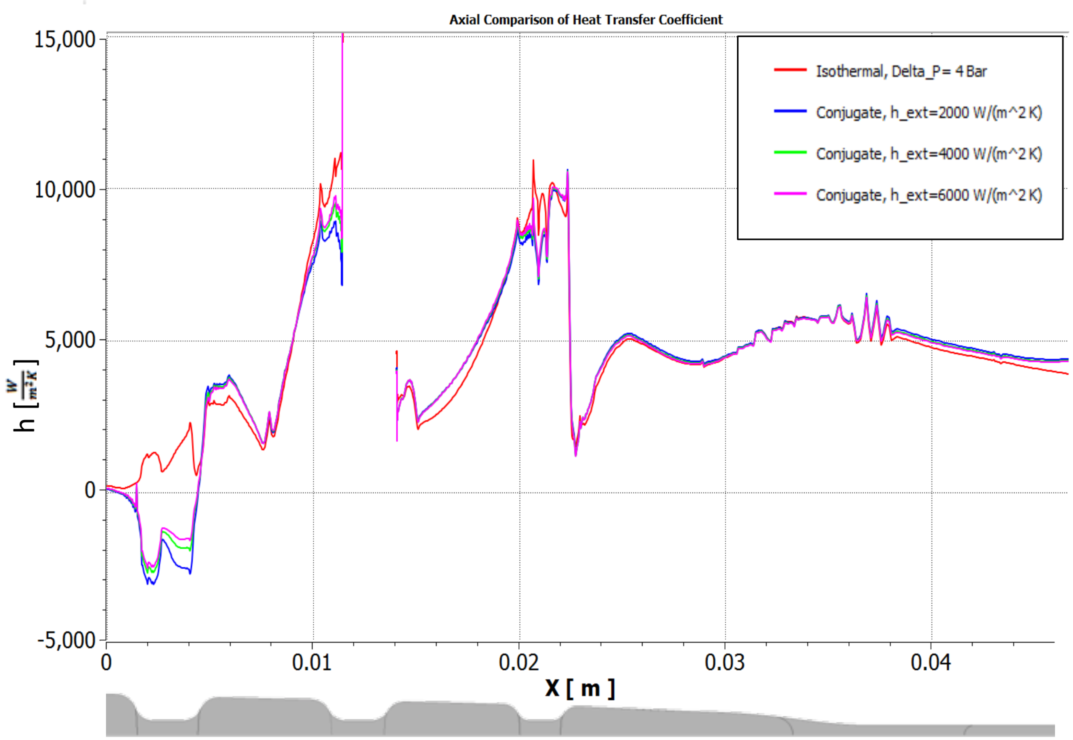




Conjugate vs Non-Conjugate CFD (Mikro)

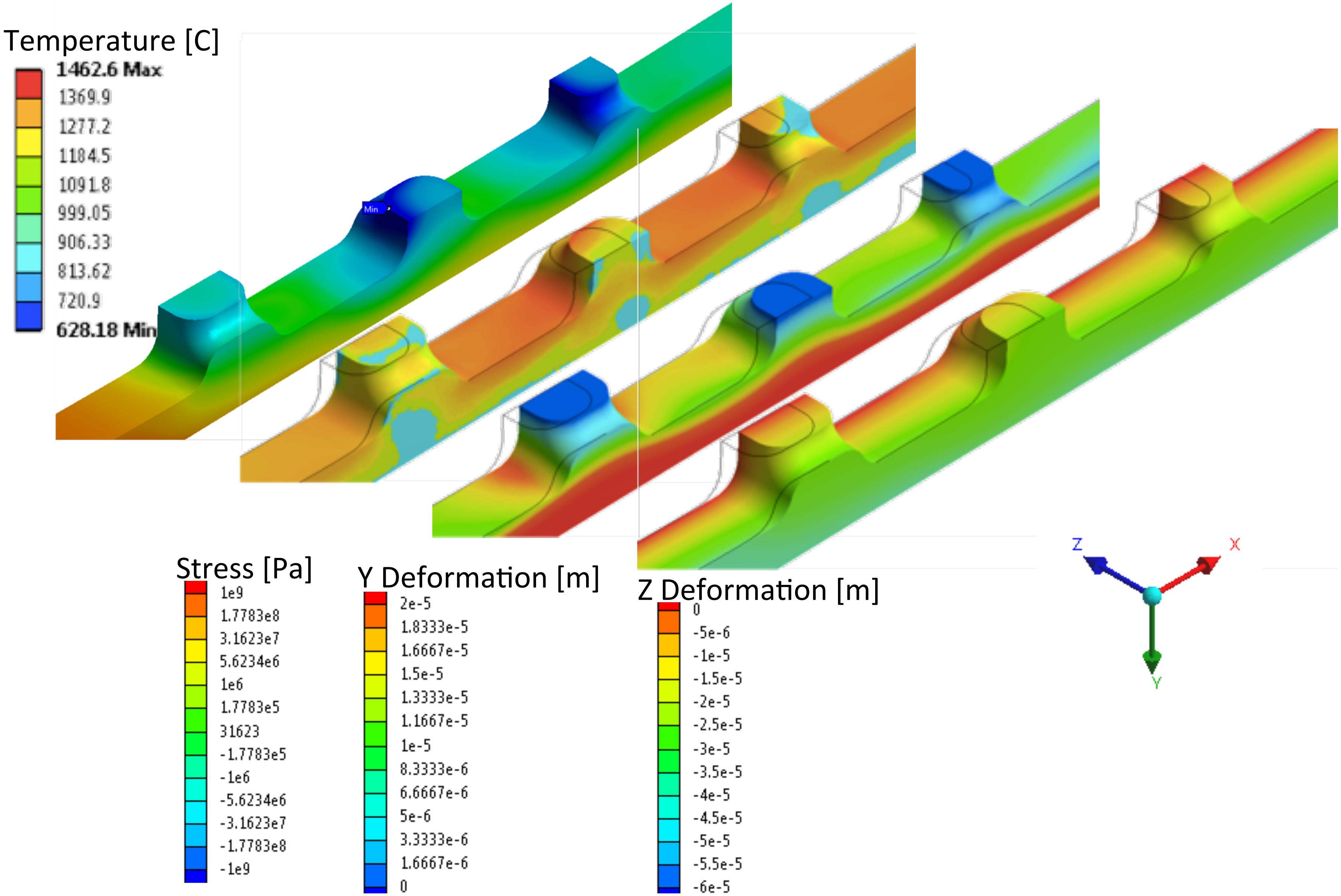




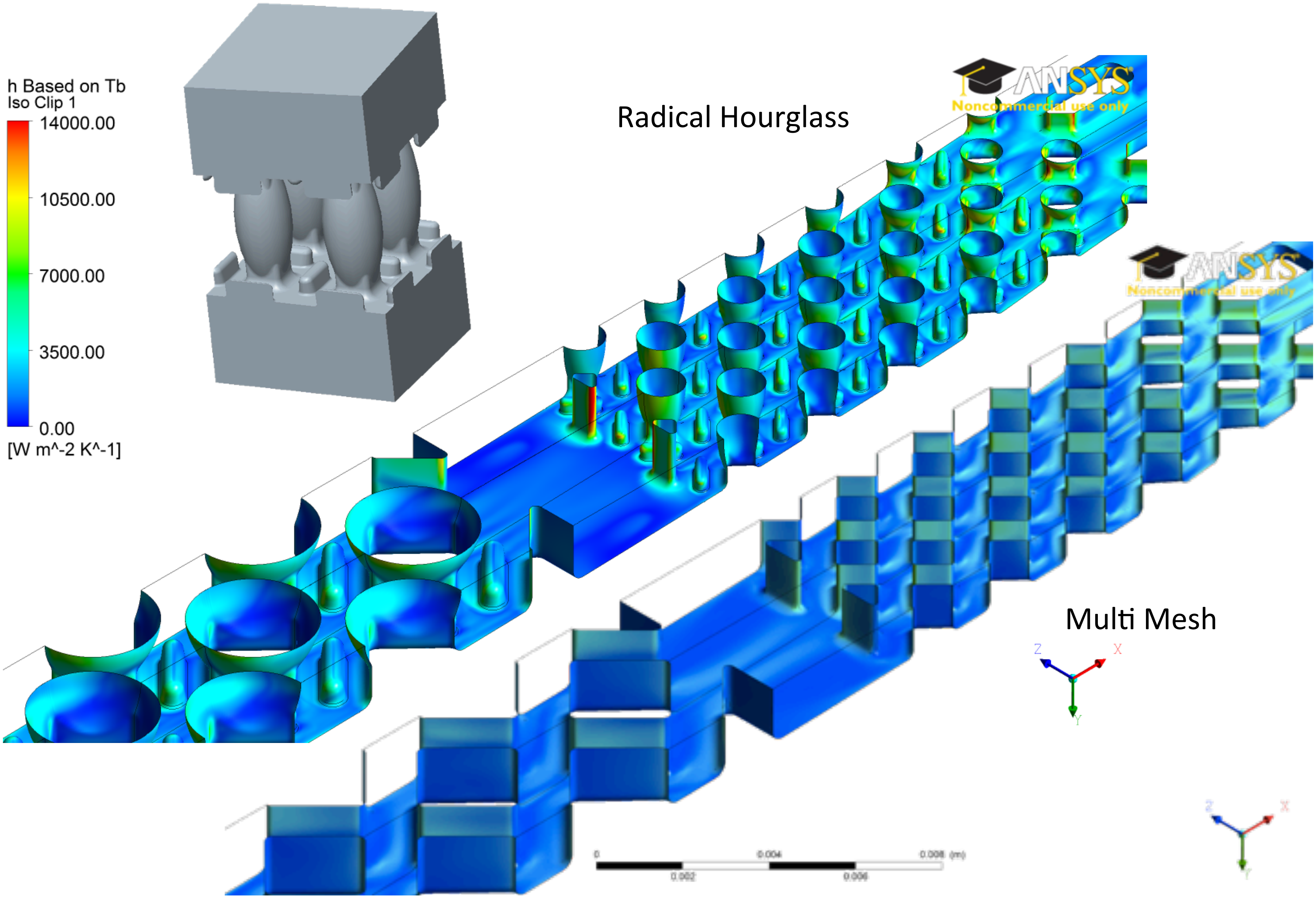


With conjugate CFD, T_b can be higher than the local surface temperature so that h can be huge or negative! Is h the right way to look at problems with complicated geometries?

Conjugate vs Non-Conjugate CFD (Mikro)



Conjugate vs Non-Conjugate CFD (Mikro)



Outline of Talk

Show challenges from two validation studies.

Assess the accuracy of a measurement technique widely used to generate benchmark data to validate CFD.

Contrast conjugate vs. non-conjugate CFD analysis in the predicted heat transfer.

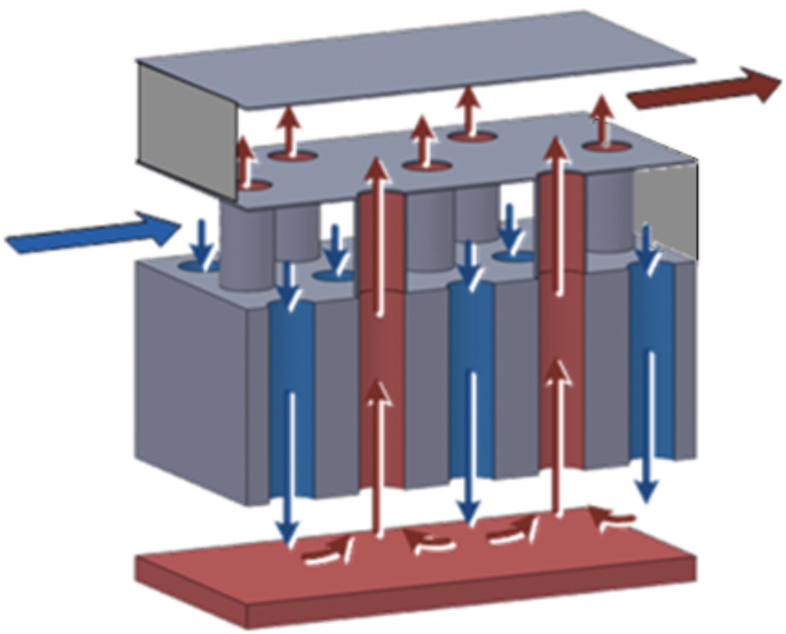
Show complications induced by unsteady heating and cooling.

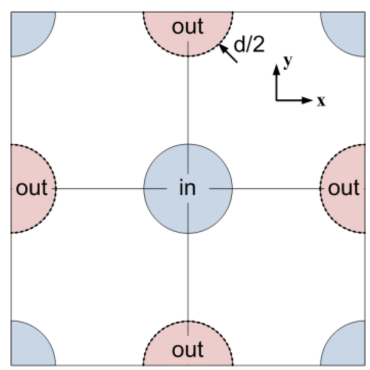
Summarize.

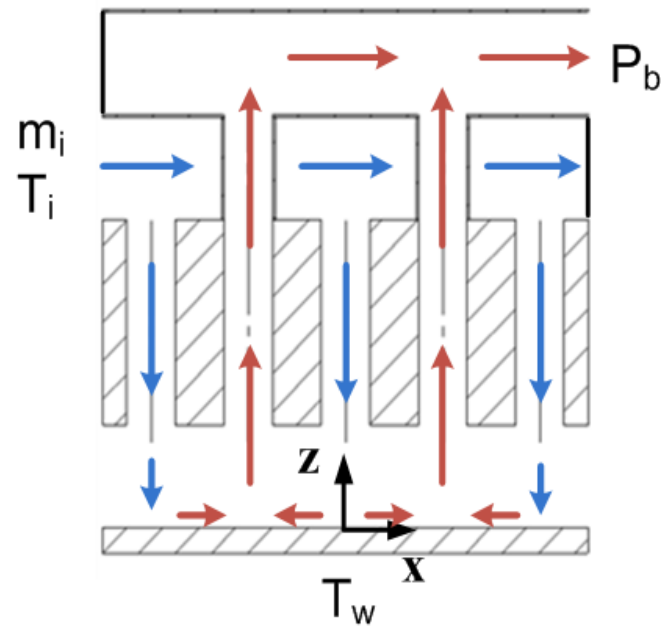
Cooling Configuration: Jet Impingement w/o Cross Flow

Objective: Explore, develop, and assess jet-impingement configurations with and without conjugate analysis under steady and unsteady heating and cooling.

Why? Jet impingement provides one of the highest heat-transfer rates.

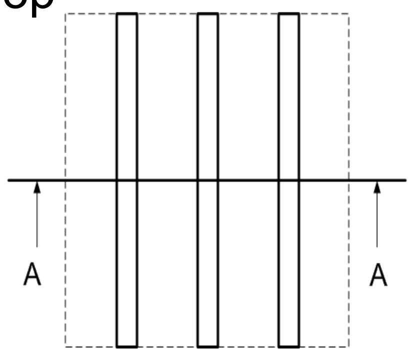






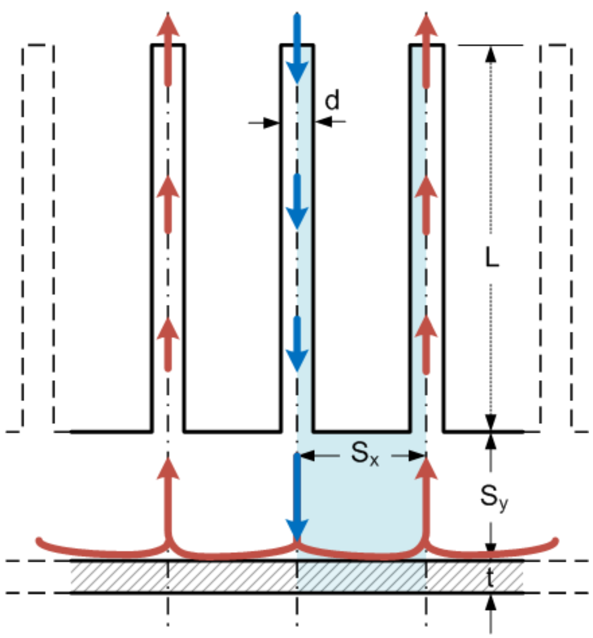
Cooling Configuration: Jet Impingement w/o Cross Flow

Top

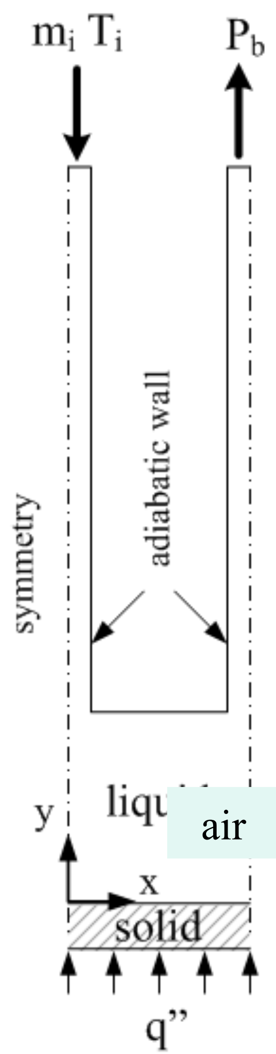


d = 5 mm
L = 30 d
S_x = 4d
S_y = 4d
t = 0.2d

A-A section



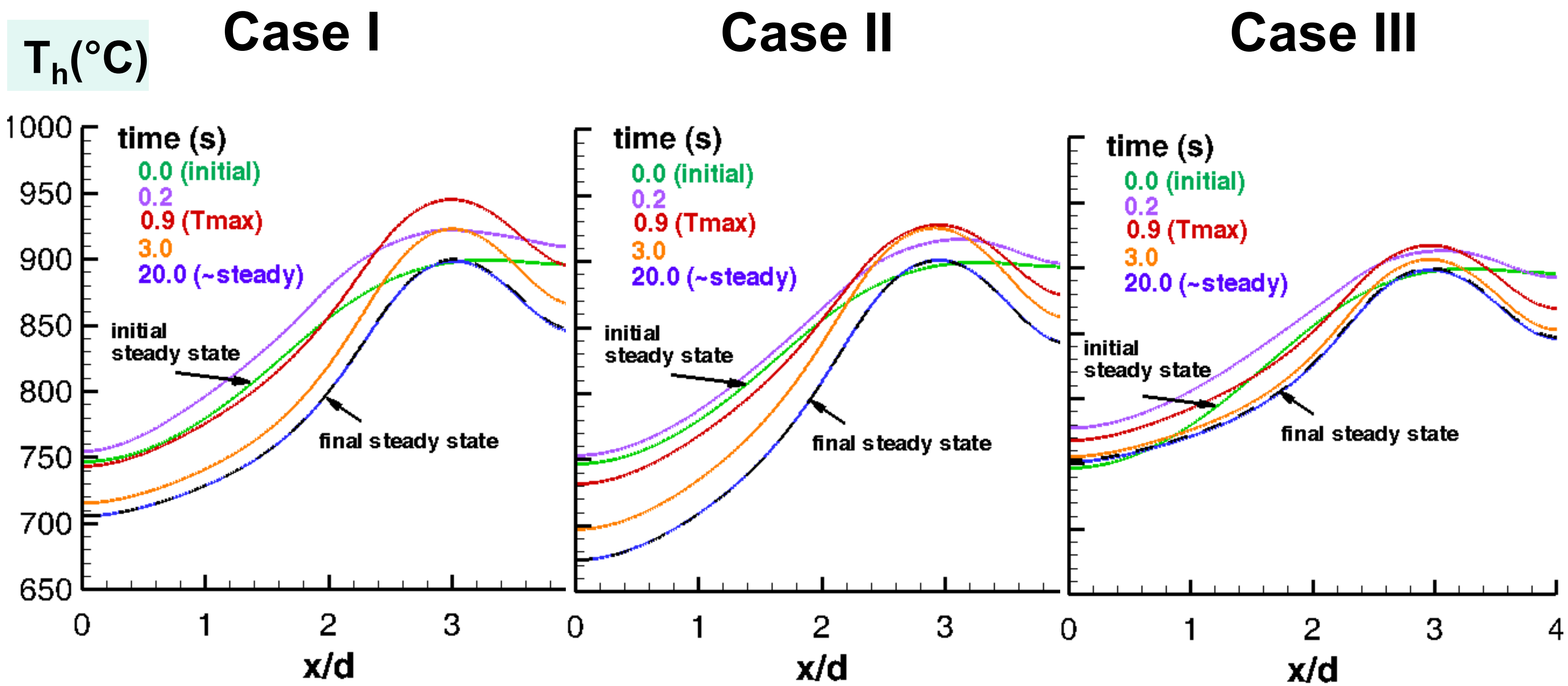
Computational domain



- Solid: super alloy (In 713C) – C_p, k = f(T)
- Fluid: air – ρ, C_p, k = f(T)
- Inlet: T_i = 673 K (400 °C)
- Outlet: P_b = 2,533,125 Pa (25 atm)
- Summary of Cases:

Case I		q'' (w/cm²)	Re_d	T_c (°C)
1	Steady	10	8K	400
2	Steady	68	150K	400
3	Transient	**Case I**-1 → **Case I**-2		
Case II		**q'' (w/cm²)**	**Re_d**	**T_c (°C)**
1	Steady	10	8K	400
2	Steady	68	125K	300
3	Transient	**Case II**-1 → **Case II**-2		
Case III		**h (w/m²-k)**	**Re_d**	**T_c (°C)**
1	Steady	161	8K	400
2	Steady	1090	150K	400
3	Transient	**Case III**-1 → **Case III**-2		

Transient T Profiles on Heated Surface



q'': 10 → 68 w/cm², Re_d : 8k → 150k

q'': 10 → 68 w/cm², T_c : 400 → 300 °C (Re_d : 125k)

q'': 10 → 68 w/cm², h_h: 161 → 1090 w/m²-k (Re_d : 155k)

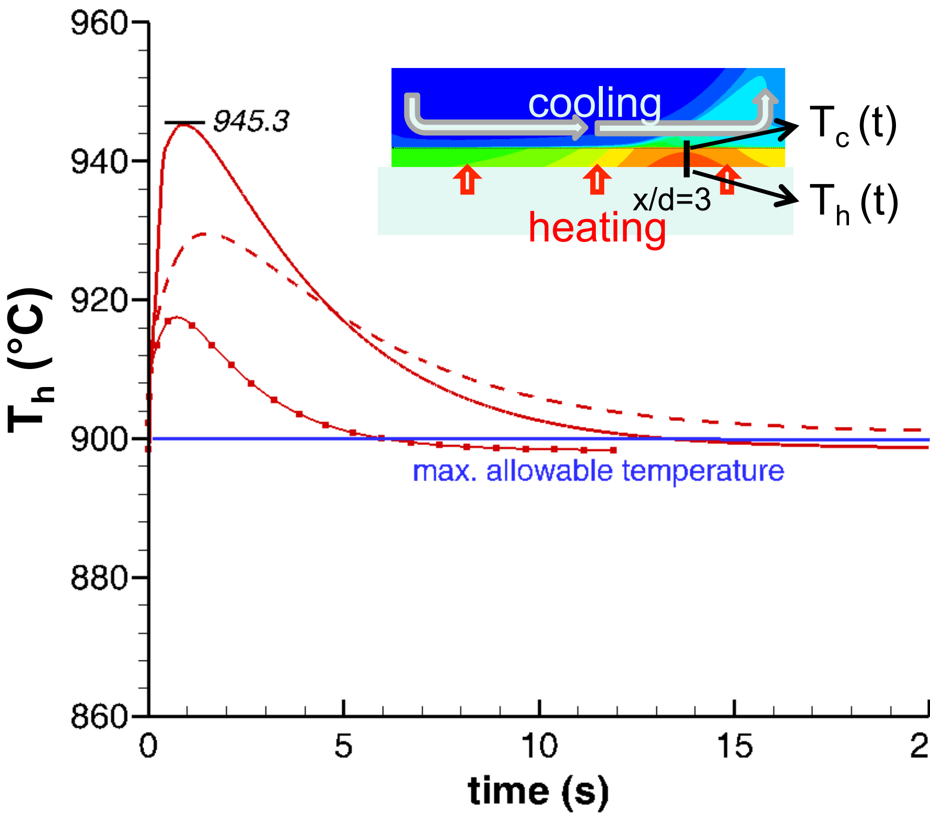
$T_h(t)$ and $T_c(t)$ at x/d = 3

Case I : **q”: 10 → 68 w/cm²**, Re_d : 8k → 150k

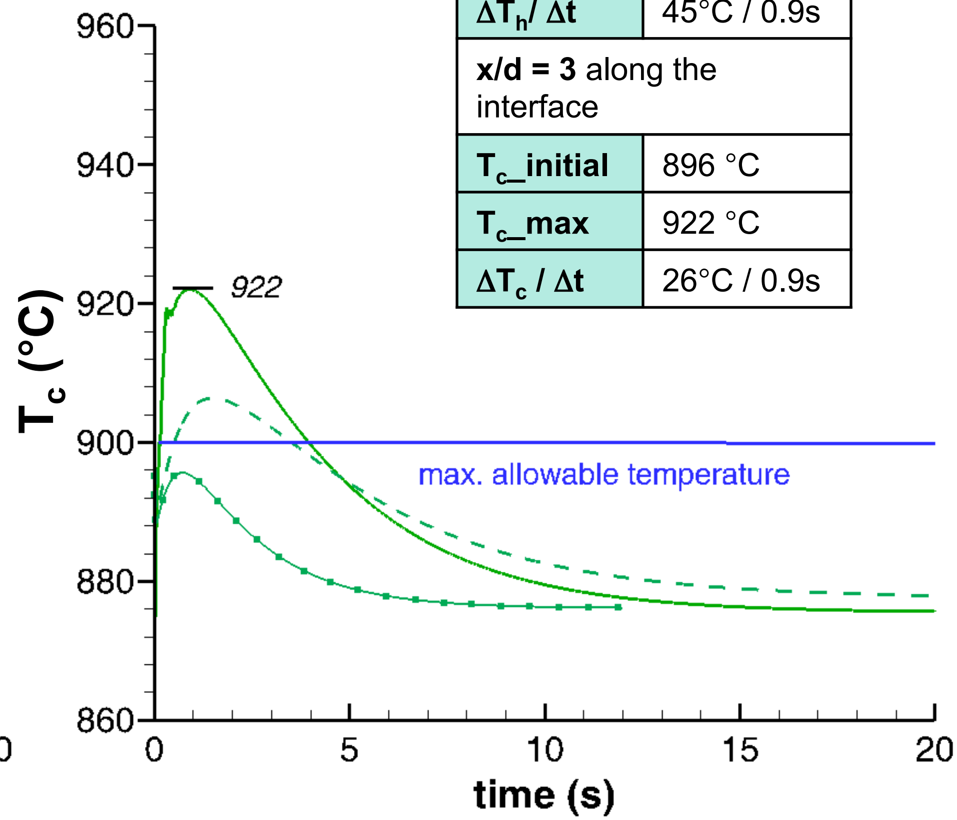
Case II: **q”: 10 → 68 w/cm², T_c : 400 → 300 °C**, Re_d : 8k → 125k

Case III: **h_h: 161 → 1090 w/m²-k**, Re_d : 8k → 155k

x/d = 3 on the bottom wall	
T_{h}_initial	900.3 °C
T_{h}_max	945.3 °C
$\Delta T_h / \Delta t$	45°C / 0.9s
x/d = 3 along the interface	
T_{c}_initial	896 °C
T_{c}_max	922 °C
$\Delta T_c / \Delta t$	26°C / 0.9s



heated side



cooled side

Summary

It is critical that CFD truly solves the experiments, including the measurement technique in the validation process.

$Re = f(T_b) = f(\text{position along duct})$; $Nu = f(Re_{inlet}, Pr_{inlet}$, heat-transfer enhancement, distance from inlet)

Need to rethink about T_b and h!

Conjugate CFD enables understanding temperature distribution within the material, which is a strong function of the coupling between the internal and the external heat transfer.

Unsteady heating and cooling require special attention to prevent over temperature.