CCBST 15TH CANADIAN CONFERENCE ON BUILDING SCIENCE AND TECHNOLOGY

Numerical study of the effect of window configuration on the convective heat transfer rate of a window

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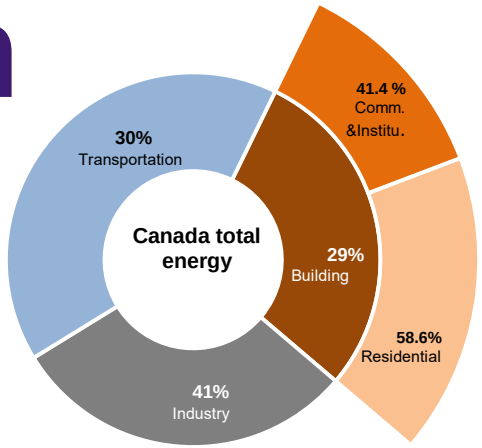
Western Engineering

Vancouver

Outline

- Introduction
- Objective and methodology
- Validation with experiment
- Numerical analysis
- Conclusions

Introduction



(NRCan Energy facts, 2015)

- Globally, Canada is one of the largest energy consumers and space heating accounts for a remarkable 63% of the energy used in the average home, while in commercial and institutional settings, space heating accounts for 56% of all energy use.

  (nrcan.gc.ca)

- This in large part is attributed to the thermal bridging caused by façade elements such as windows.

- In fact, 20 – 40% of energy in buildings is wasted through windows. (Lee et al., 2013)

Covered with low R-value and high solar gain (Straube, 2012)

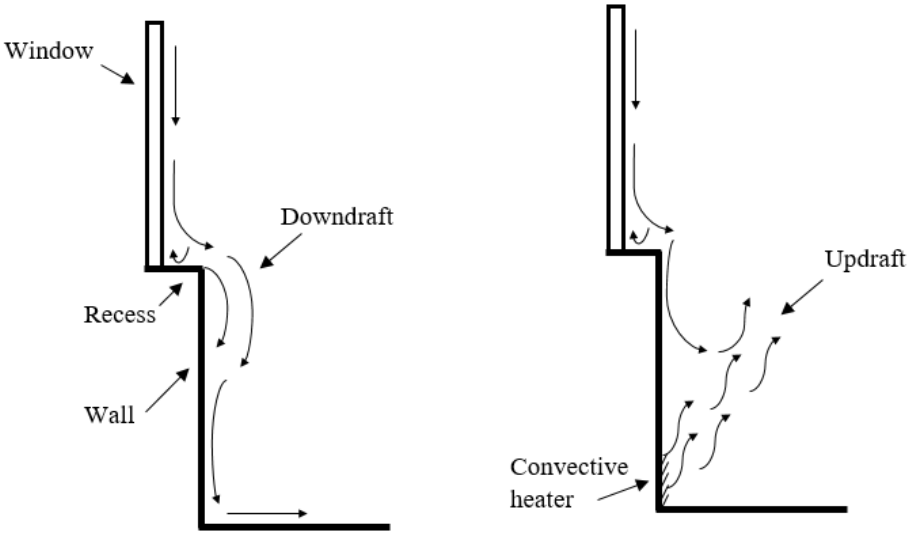
Window configuration

- Windows are configured in different forms.
  - Psychological
  - Aesthetic
  - Daylighting
  - Heating & cooling

- Proper design and selection of window system is regarded as one of the most important passive strategies for saving energy in buildings.

Window configuration

- In winter, outdoor conditions primarily influence the indoor surface temperature of windows.
- This leads to a temperature gradient in the indoor environment which induces a <u>downdraft</u> affecting the thermal comfort of occupants.
- This phenomenon is sensitive to the configuration and location of windows.



Window configuration

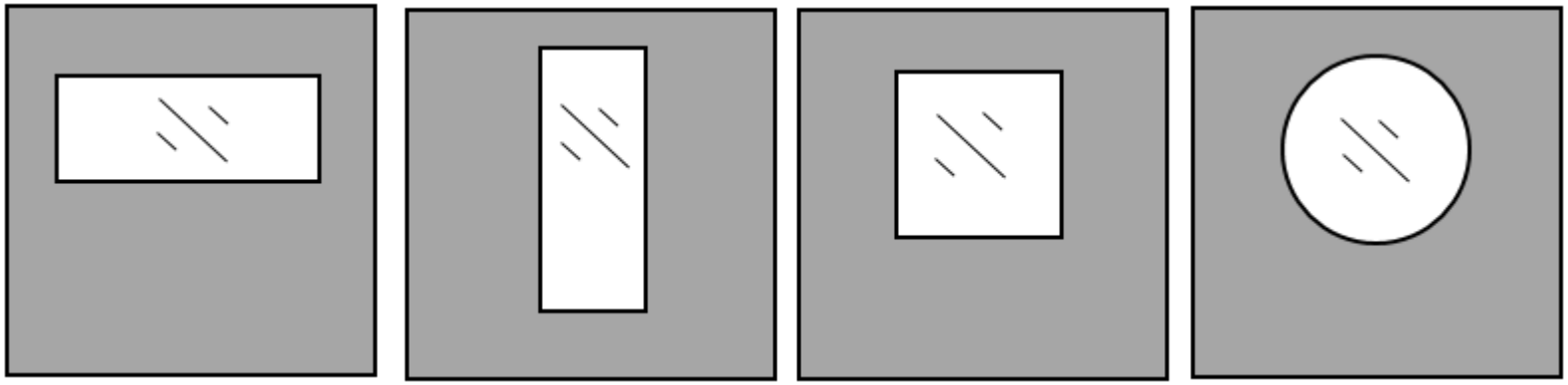
- ASHRAE standards 90.1:2010 and ASHRAE 189.2014 provide:

"The total vertical fenestration area shall be less than 40% of the gross wall area"

- Although this is useful information, it does not account for differences in window configurations as well as the thermal and lighting performance **for the same WWR**.

Different forms of fenestration configuration
http://inhabitat.com

Window configuration

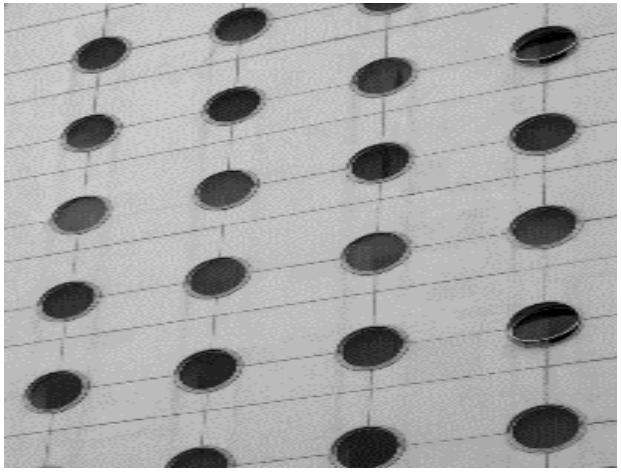


30% WWR but with different window configurations

- It is important for the guideline to accommodate as the question of which of the four window configurations is more energy efficient and thermally comfortable?

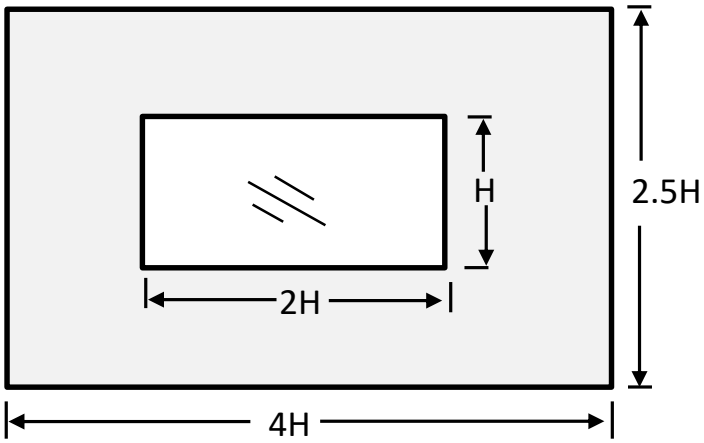
- Does not provide any explicit way to evaluate thermal and lighting performance of the windows.

Sample configurations

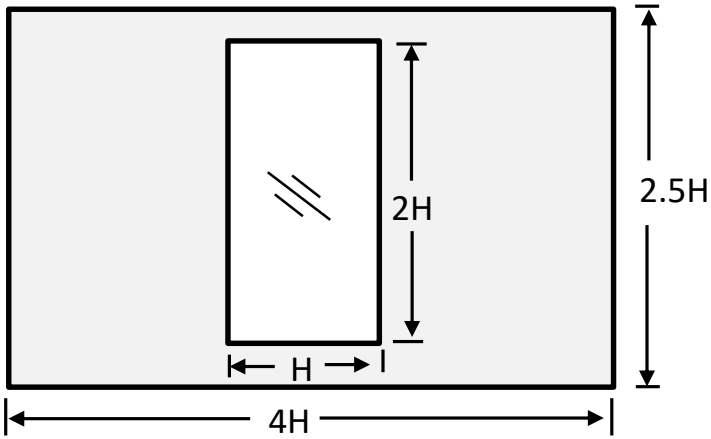


Sample configurations

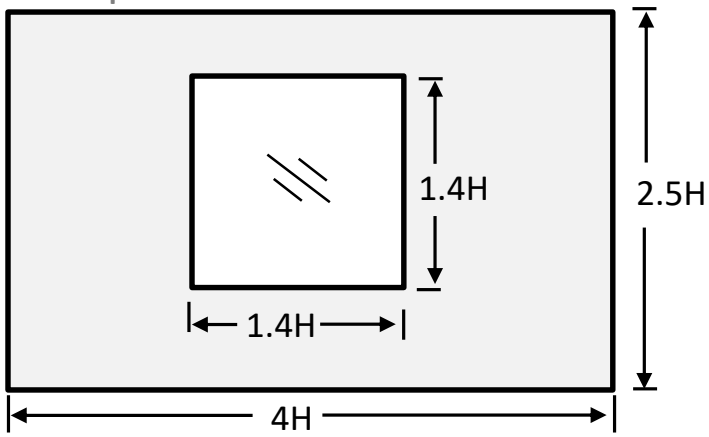
Case 1: Horizontal



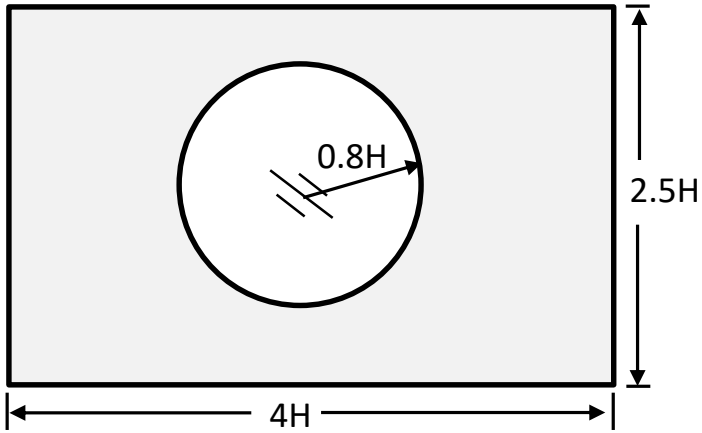
Case 2: Vertical



Case 3: Square



Case 4: Circular



20% WWR

Research objective

- To investigates the influence of different window configurations on the convective heat transfer rate of a window.

Methodology

- Experimental data validation
  - Vertical heated plate
- Numerical model for validation
  - Computational model
  - Sensitivity analysis
  - Comparing with experimental results
- CFD based *CHT* evaluation
  - Convective heat transfer computation
  - Boundary conditions
  - Evaluation window *CHT*

CFD based *Nu* evaluation

Experimental data for validation

- Experimental data by Churchill and Chu (1975) for a flow over a vertical heated isothermal plate is used.

- In this experimental analysis, the Nusselt number for the entire Rayleigh number (*Ra* = 10^7-10^{12}) from laminar, transition to turbulent natural convective regimes were evaluated.

$$\overline{Nu_L} = \left\{0.825 + \frac{0.387 Ra_L^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}}\right\}^2$$

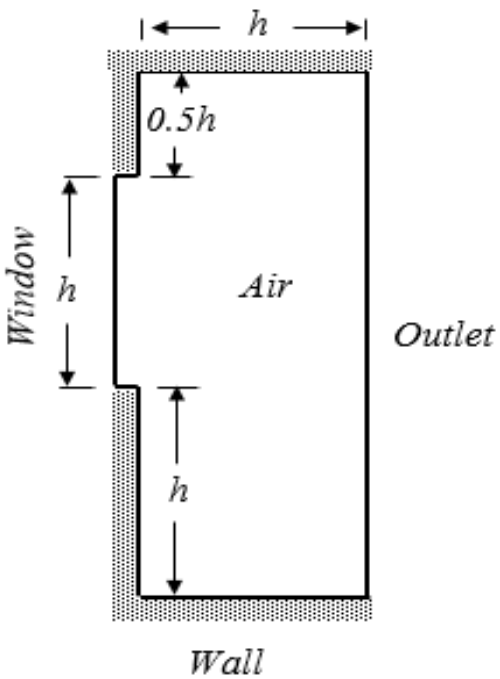
$$Ra = Pr.Gr = \frac{\vartheta}{\alpha}.Gr = \frac{\mu C_p}{k}\left(\frac{\beta g (T_w - T_f) H^3}{\vartheta^2}\right)$$

$$Nu = \frac{hL}{k} = \frac{\bar{q}_w' H}{k(T_f - T_w)}$$

CFD based *Nu* evaluation

Numerical model for validation

– A 2-D heat transfer in a vertical isothermal window is used.

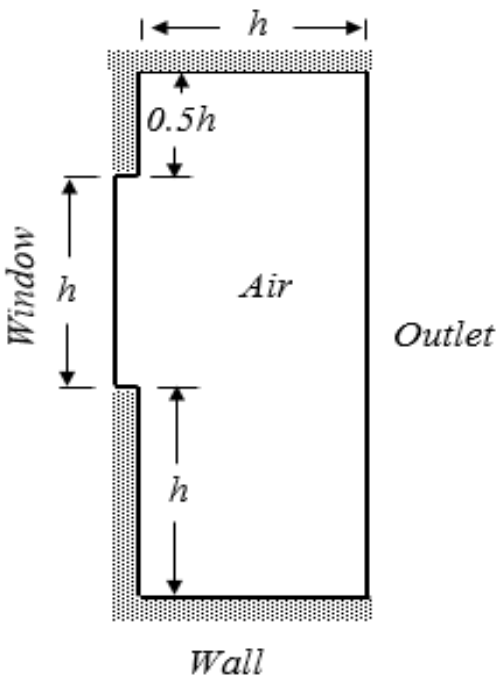


Name	Boundary conditions
Wall	Adiabatic (u=v=0, q=0)
Window	T_w=310 K (isothermal)
Outlet	Pressure outlet, T_a =290 K

CFD based *Nu* evaluation

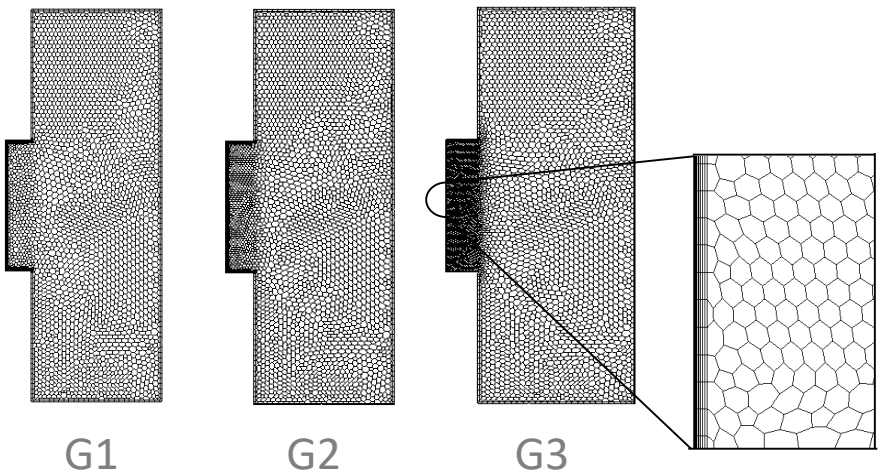
Numerical model for validation

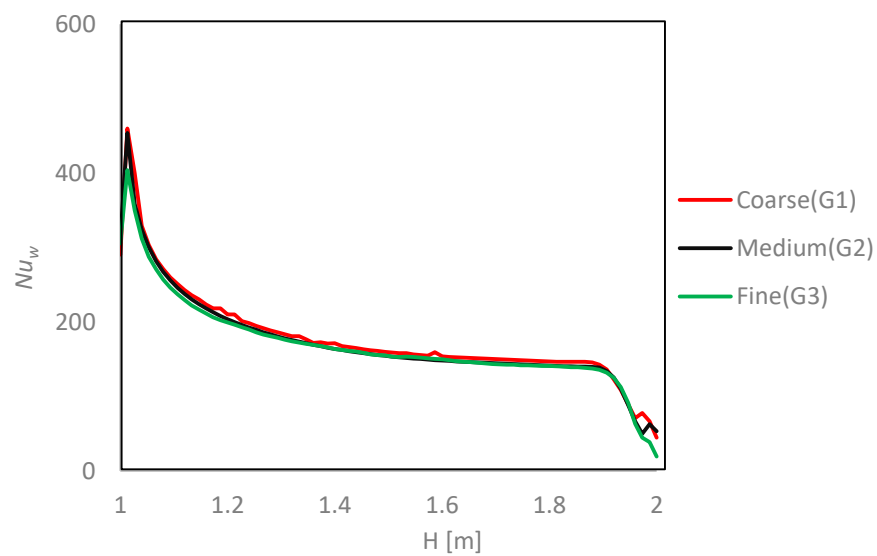
– A 2-D heat transfer in a vertical isothermal window is used.



- High-resolution (RANS) CFD simulations using $SST\ k-\omega$ *LRNM* approach has been used to resolve the near-wall heat transfer in conjunction with the buoyancy force.
- In addition, fluid properties are treated as constant values, except when changes in temperature lead to changes in density and the development of a buoyancy force.
- The Boussinesq approach.
- Only **natural convection**

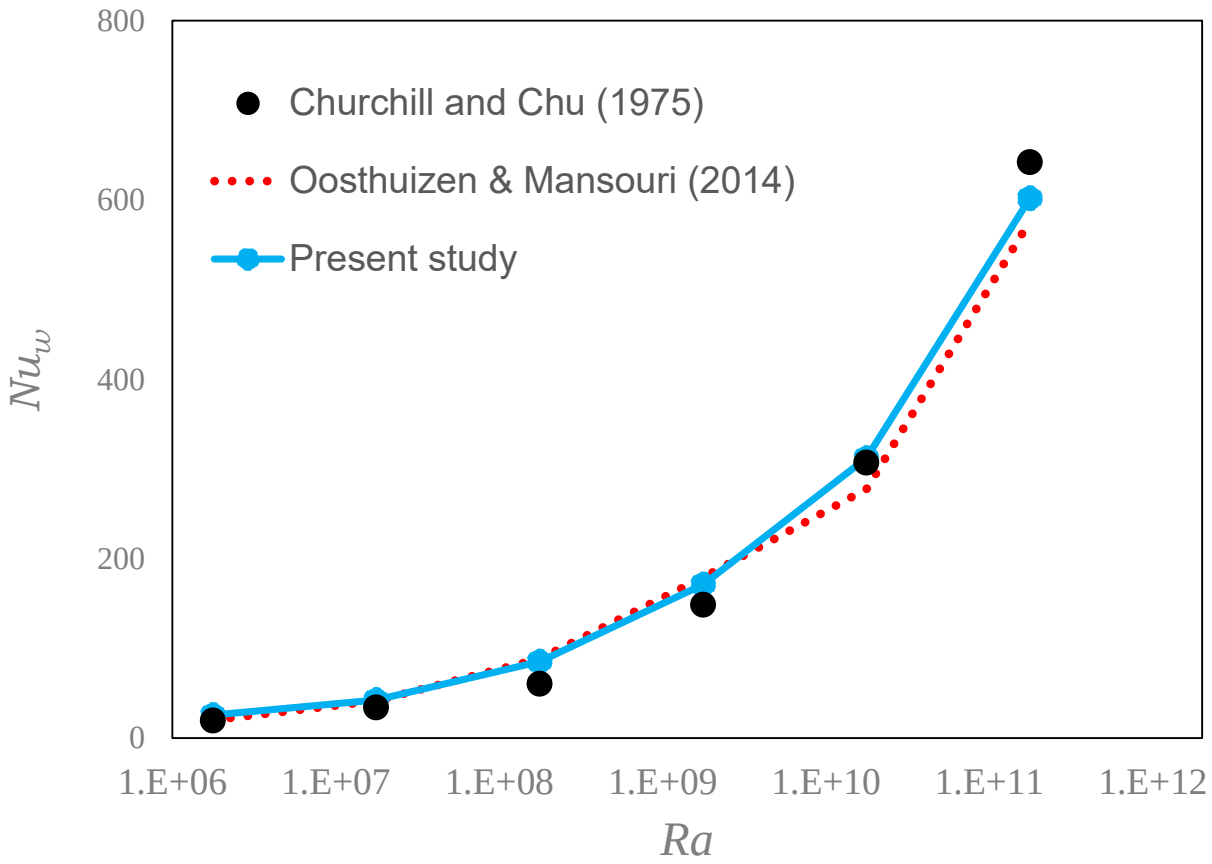
Grid sensitivity





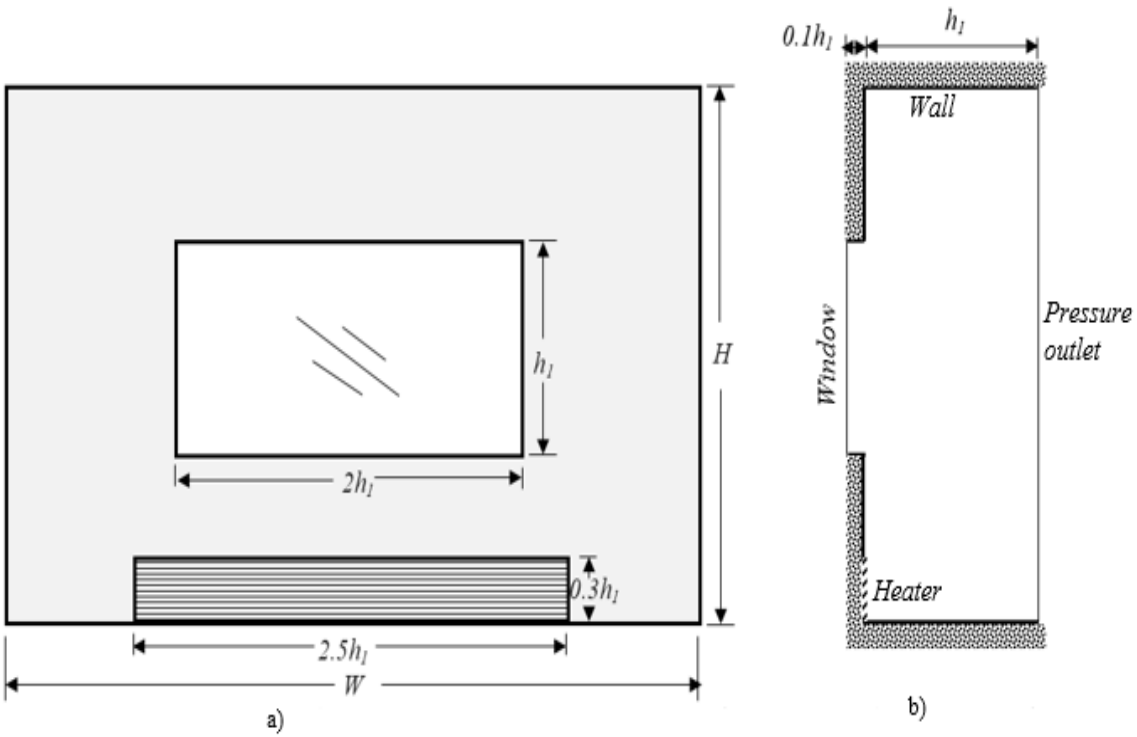
G1(Coarse)	G2(medium)	G3(Fine)
24638	34898	52350

Comparison of experimental and CFD results



CFD based evaluation of Convective heat transfer rate

- Computational domain:



- Only the convective heat transfer considered
- SST k-w closure
- Gravity
- Boussinesq

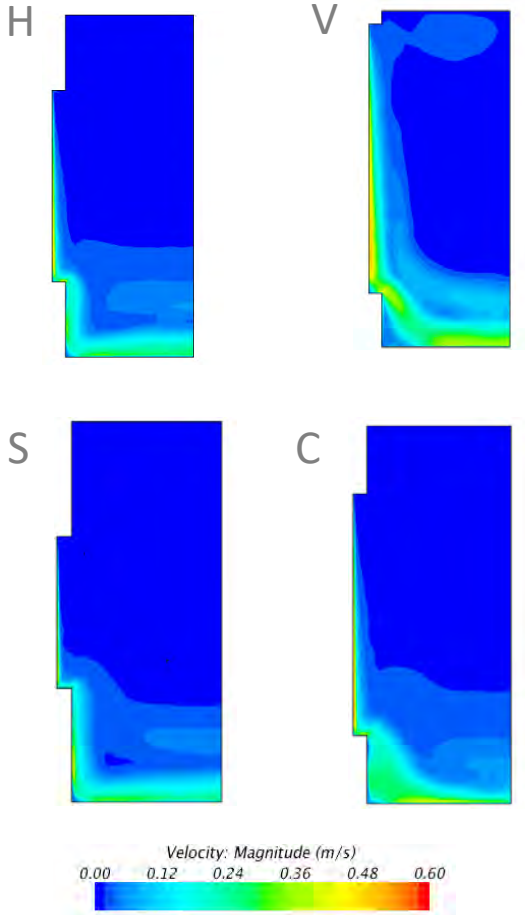
$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = \vartheta\frac{\partial^2 v}{\partial x^2} + g\beta(T - T_\infty)$$

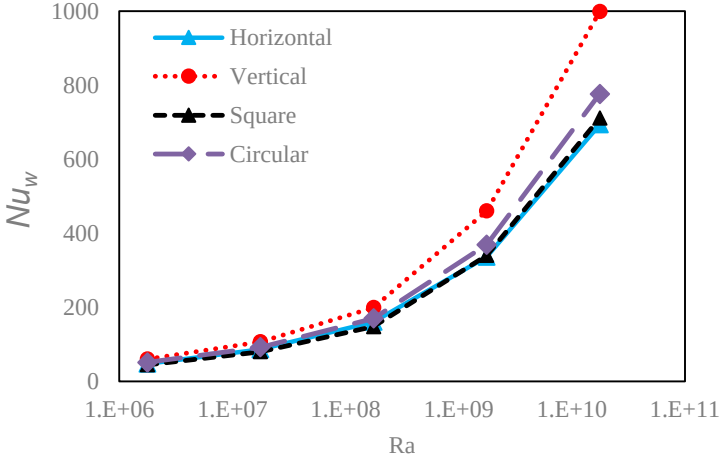
$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \frac{k}{\rho C_p}\left(\frac{\partial^2 T}{\partial x^2}\right)$$

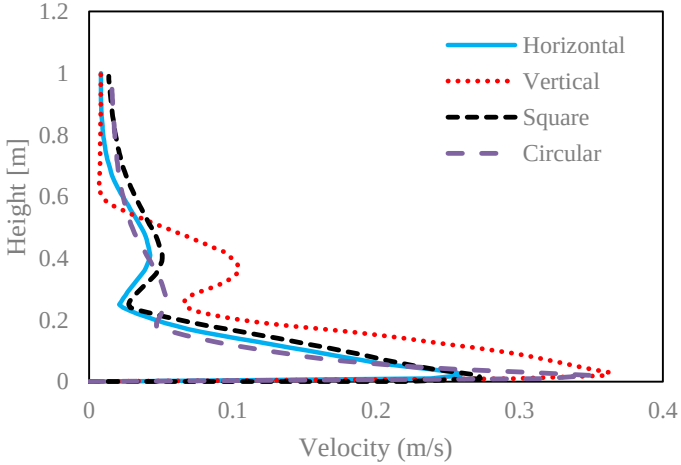
Name	Boundary conditions
Wall	Adiabatic (u=v=0, q=0)
Window	Isothermal, T_w=283 K (isothermal)
Heater	Isothermal, $T_h > T_a$
Outlet	Pressure outlet, T_a =294 K

Results

No convective heater below window



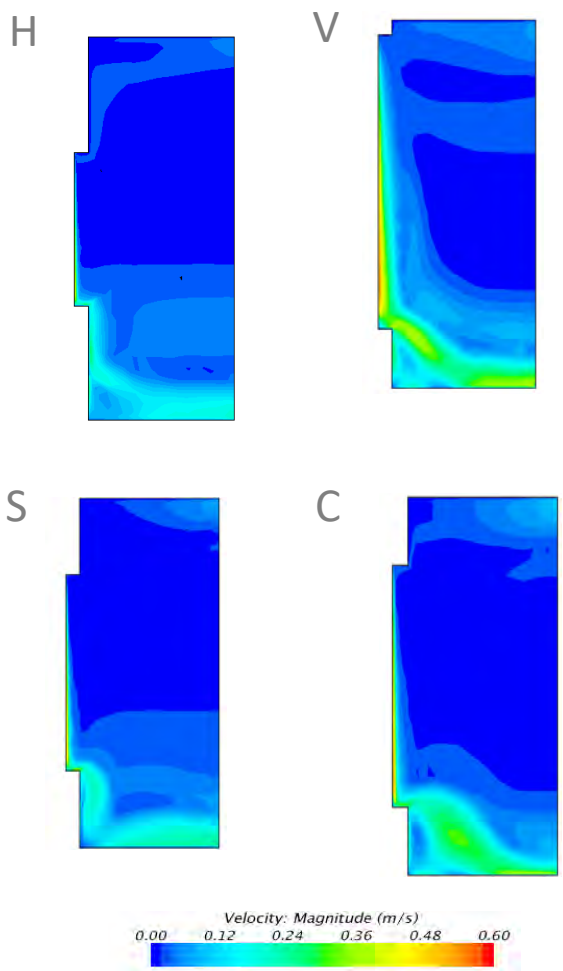




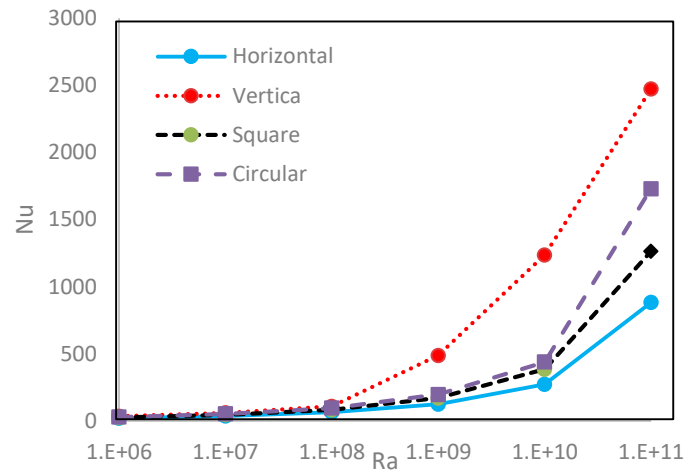
H : Horizontal S : Square
V : Vertical C : Circular

Occupied zone 0.6 m from window

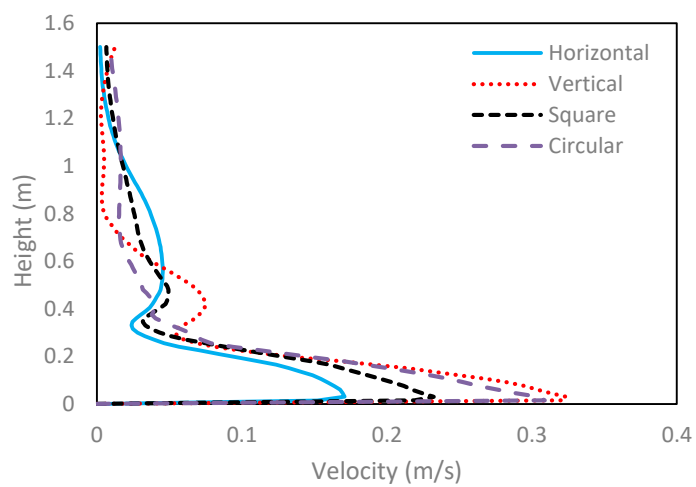
Convective heater below window



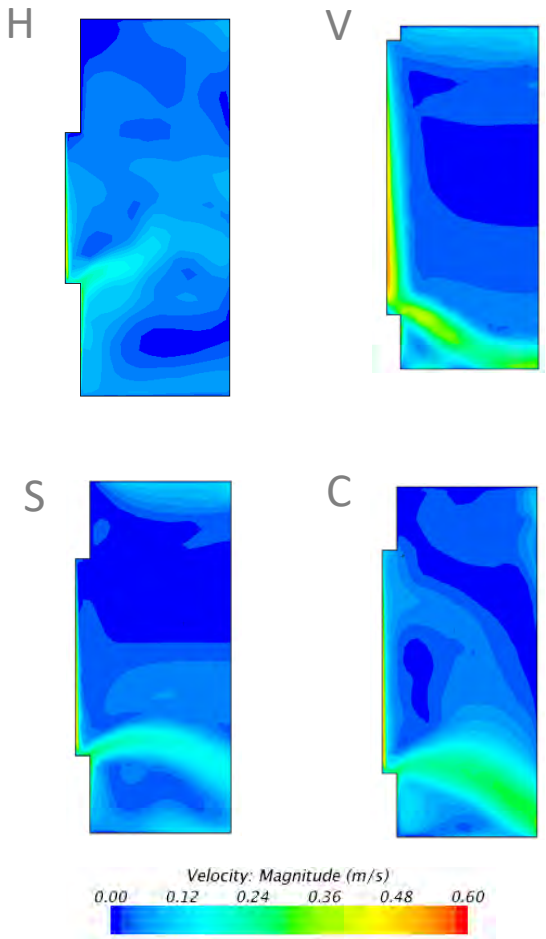
$For\ Ra = 1.77 * 10^9$



$$\boldsymbol{\theta} = \frac{(T_h - T_f)}{(T_f - T_w)} = \mathbf{1}$$



Convective hearer below window



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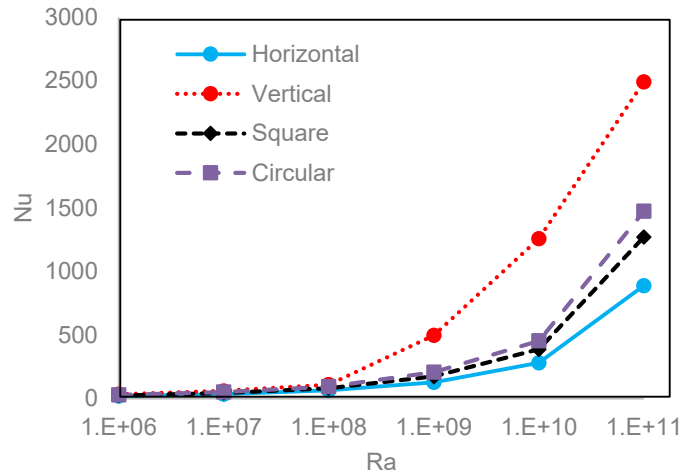
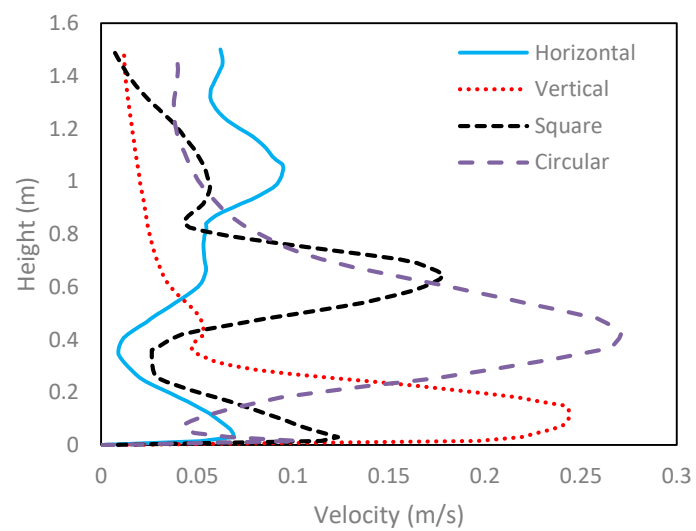
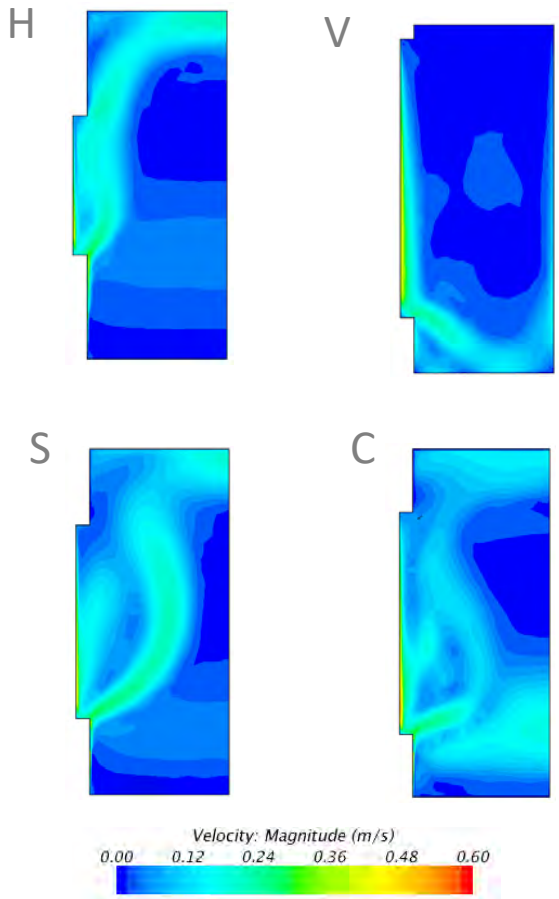


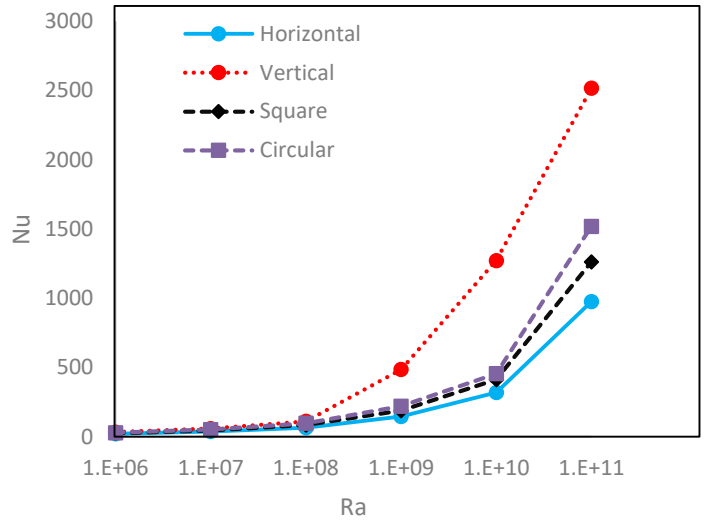
$$\boldsymbol{\theta} = \frac{(T_h - T_f)}{(T_f - T_w)} = \mathbf{2}$$



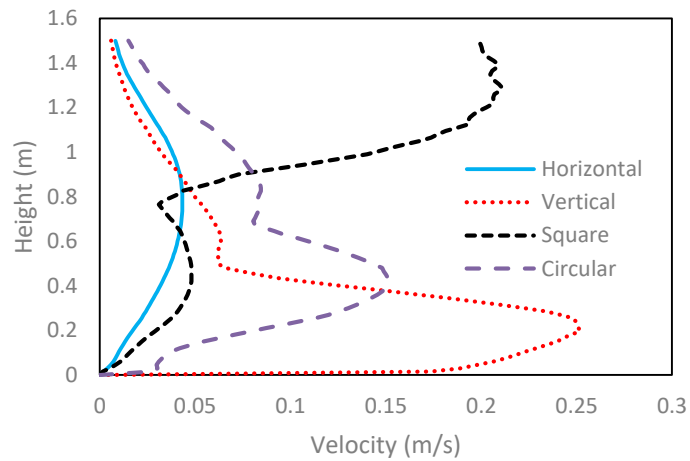
Convective hearer below window



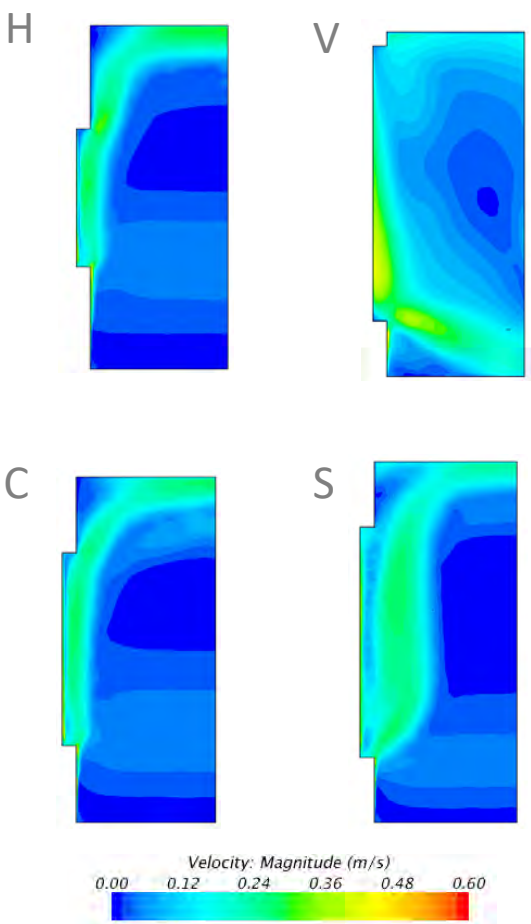
$For\ Ra = 1.77 * 10^9$



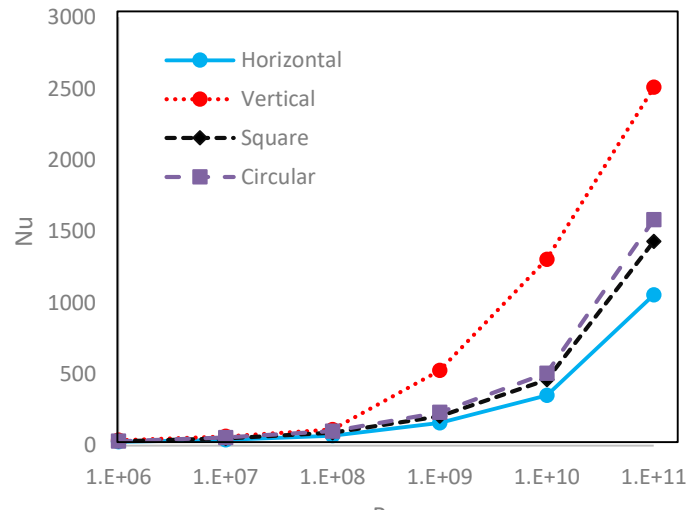
$$\boldsymbol{\theta} = \frac{(T_h - T_f)}{(T_f - T_w)} = \mathbf{3}$$

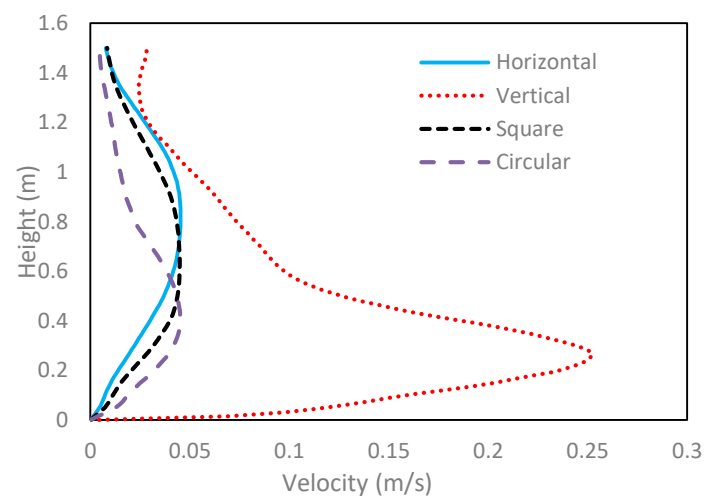


Convective hearer below window



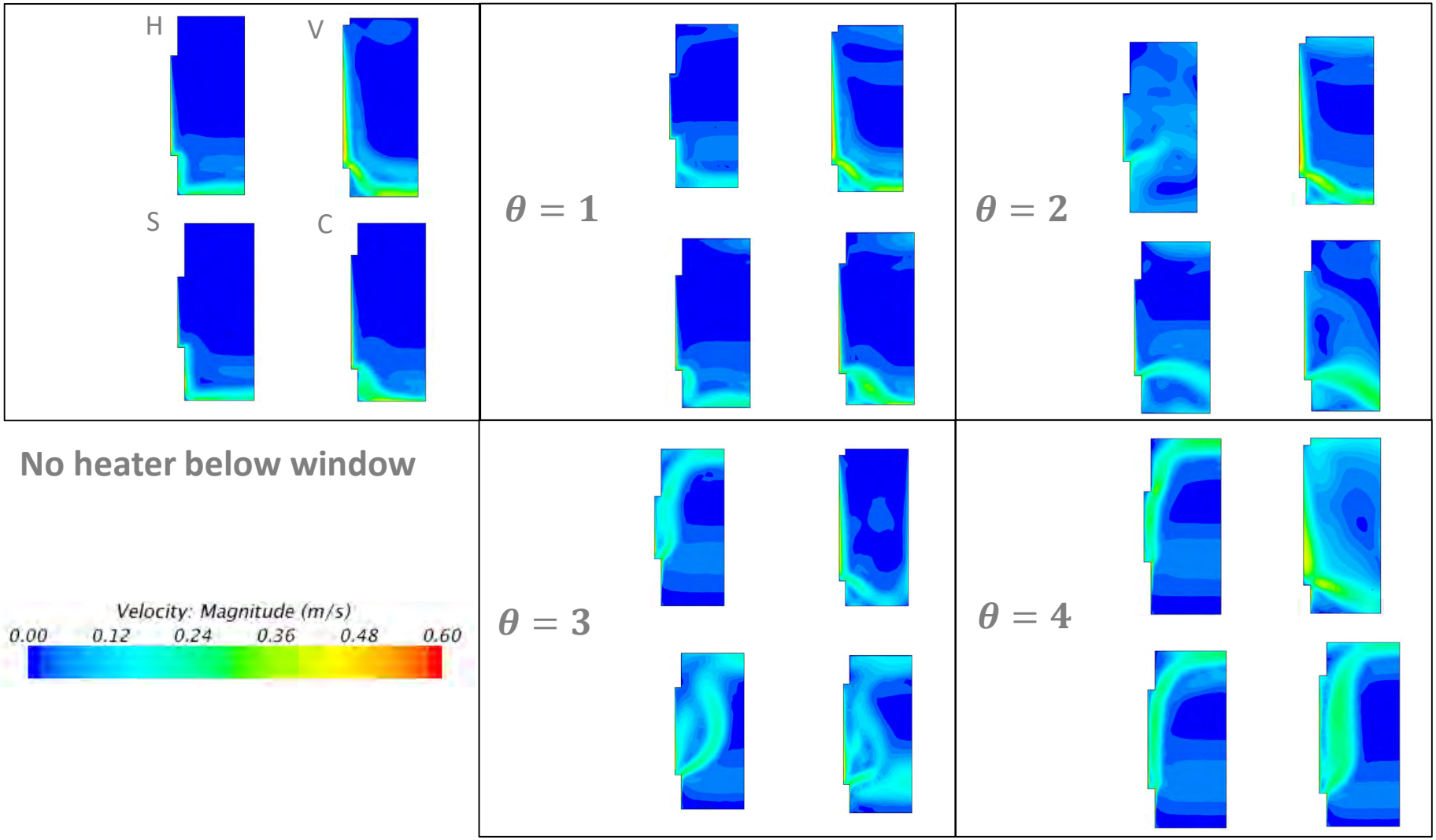
$For\ Ra = 1.77 * 10^9$





$$\boldsymbol{\theta} = \frac{(T_h - T_f)}{(T_f - T_w)} = \mathbf{4}$$

Velocity contour for $Ra = 1.77 * 10^9$

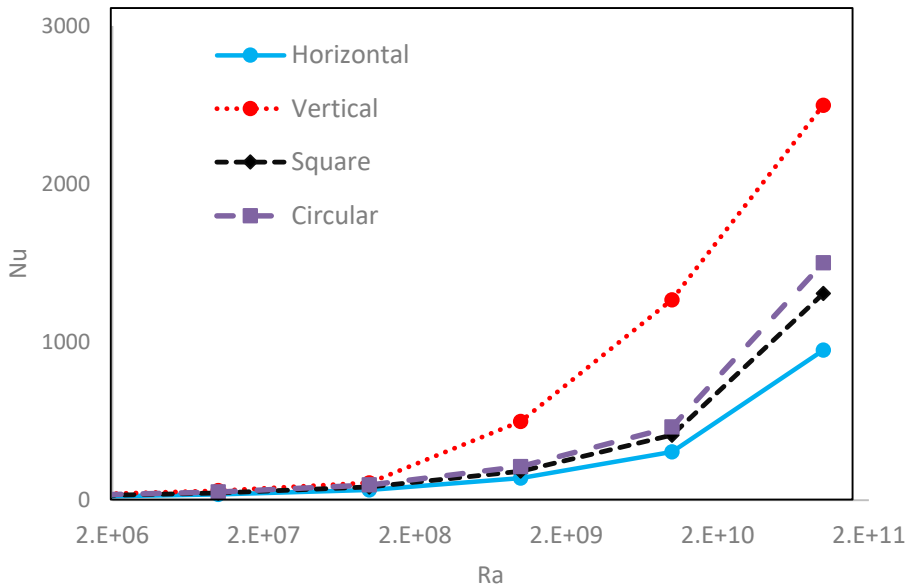


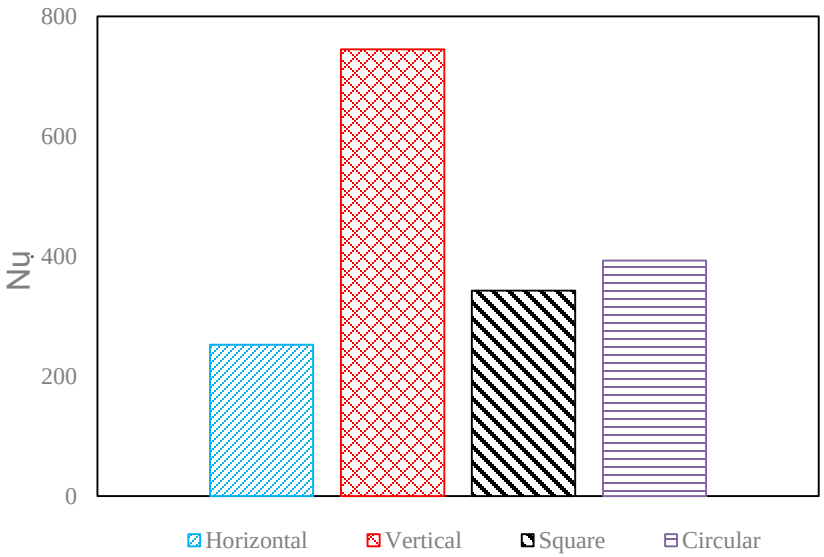
No heater below window

Convective heater below window

Conclusion

- Window average Nusselt number with Rayleigh number for four values of the dimensionless heater temperature, θ.
- Horizontal configuration shows least convective heat transfer rate value.





Acknowledgment

- Supervisors:
  Girma Bitsuamlak, PhD, PEng, F CSCE.
  Fitsum Tariku, PhD, PEng.

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