Building Energy Efficiency Program

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Research Update Report of the Simplified IR Box Modeling

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1. BACKGROUND

1.1 INTRODUCTION OF THE EXPERIMENT

A set of experiment instrumentation, which are shown in Figure 1, were constructed in laboratory, to analyze the different specimens' performance. The instrument is mainly divided in to Climate Chamber and Measurement Chamber, to simulate the outdoor and indoor environments. The specimen to be tested was fixed between these two chambers. With a traversing measurement system, the parameters near the specimen's indoor surface were obtained. Infrared imager was also used in the measurement chamber, to obtain the temperature of the specimen's surface. So the chamber was also called IR Box. As a natural convection process, a downward flow of low temperature was generated along the surface of specimen. The study of this phenomenon is not only for occupants' comfort, it is also a important component of building envelope thermal performance , especially for determine surface temperatures, heat transfer through the specimen and the likelihood of moisture condensation on the interior surfaces. As the specimen sample between the chambers, this experiment plat form can test different sample with same environment condition.

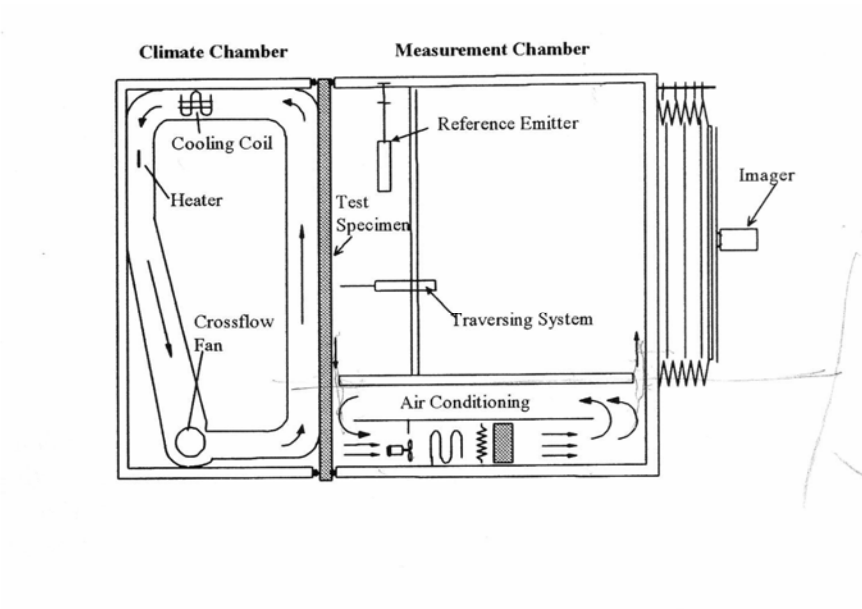

Fig. 1.1: Laboratory Chamber

1.2 GOALS OF NUMERICAL MODELING

This project describes the computer modeling results of the heat transfer characteristics of projecting products. The finite element mesh was generated and the numerical modeling was performed. The calculations using turbulent flow mathematical models were performed to the IR Box in order to ascertain how the window affects the flow patter and natural convection heat transfer in the

cavity.
The results of this numerical simulation will be compared to experimental data. Specifically, surface temperature distributions and local heat transfer coefficients along the indoor side of the projecting product will be compared. To obtain the performance of different specimen, this study focused on the modeling the measurement chamber. As the first stage of the research, simplification was made to the geometry of the chamber. The local heat transfer coefficient

2. MATHEMATIC MODEL

2.1 GOVERNING EQUATIONS

To model the natural convection in control volume, several governing equations are applied: mass conservation, momentum equation, and energy conservation. With the boundary condition, these equations can describe the natural convection air flow behavior in the cavity.

1 Mass Conservation
The conservation of mass means the change of the mass in a control volume is equal to the net mass flow rate through the control volume.

$$\partial_0 \rho + \partial_i (\rho u_i) = 0$$

In which:

$$\partial_0 = \frac{\partial (\ )}{\partial t}$$

$$\partial_i = \frac{\partial (\ )}{\partial x_i}$$

For 2-D problem, i=1, 2; so u1=u, u2=v, x1=x, x2=y

2 Momentum equation (Newton's Second Law)

According to the Newton's Second Law, the force acting on a volume is equal to the momentum change of the control volume.

$$\partial_0 (\rho u_i) + \partial_j (\rho u_j u_i) = \rho g_i + \partial_j \Re_{ji} + \rho F_j$$

In this formula:

$\rho g_i + \partial_j \Re_{ji} + \rho$ the sum of pressure, viscous and body forces

For the pressure and viscous, the stress tensor $\Re$ is defined with Stokes assumption (μ +(3/2) λ =0）.

$$\Re_{ji} = -\delta_{ij} P - \frac{2}{3} \mu \delta_{ij} \partial_k u_k + 2\mu \partial_{(i} u_{j)}$$

where

$$\partial_{(i} u_{j)} = \frac{1}{2} (\partial_i u_j + \partial_j u_i)$$

3. **Energy Conservation (First law of Thermodynamics)**

In the energy conservation, the internal and kinetic energy increscent of the control volume is equal to the net heat flow and output work of the control volume.

$$\partial_0(\rho e + \tfrac{1}{2}u_i u_i) + \partial_j(\rho u_j(e + \tfrac{1}{2}u_i u_i)) = -\partial_i q_i + H$$

Where:

$$H = q_s + q_D + q_{ch} + q_j + q_p$$

H is the sum of the heat generation;
q_s Heat generation due to applied sources or sinks per unit volume
q_D: Heat generation due to viscous dissipation
q_R: Heat generation due to chemical reactions
q_J: Heat generation due to electrical (Joule) heating
q_P: Heat generation due to participating-media radiation

2.2 MODEL ASSUMPTION

The assumption of this model is shown below.

1 The problem is a 2-D case.
2 The flow in the cavity is steady
3 No-slip exists for all wall surfaces, the air velocityes of the cavity on all surfaces of the chamber are set to zero.
4 The air in the cavity is incompressible, the density of the air is constant, and the viscous dissipation is negligible.
5 The Boussinesq approximation has been applied, which means the gravity term is the onlu significant body focre
6 As a natural convection, the velocity and gradients are small, so the viscous dissipation can be neglected.
7 The fluid properties are isentropic and will not change with the temperature.
8 No heat generation in the air cavity.

According to these assumptions, the conservation equations can be simplified as:
Mass equation:

$$\frac{\partial u_i}{\partial x_i} = 0$$

Momentum equation:

$$\rho\left(u_j \frac{\partial u_i}{\partial x_j}\right) = -p,_i + \left[\mu\left(\frac{\partial u_i}{\partial x_i}\right)\right]_{,j} + \rho_0 \beta g_i (T - T_0)$$

Energy equation:

$$\rho c_P \left(u_i \frac{\partial T}{\partial x_i}\right) = -\lambda \frac{\partial^2 T}{\partial x_i^2}$$

2.3 TURBULENCE MODELING

As the governing equations are inadequate for turbulence numerical modeling, Reynolds decomposition is employed to the equations. The fluctuation components of the decomposition represent the turbulent behavior in the flow.

Then, with Boussinesq Eddy-Viscosity Approximation, the turbulence kinetic energy can be represented by:

$$\rho\partial_0 k + \rho U_j \partial_j k = \mu_T(\partial_j U_i + \partial_i U_j)\partial_j U_i - \rho k\omega + \partial_j\left[\left(\mu + \frac{\mu_T}{\sigma_k}\right)\partial_j k\right] + \beta\frac{\mu_T}{\sigma_T}\partial_i T\, g_i$$

By the definition of the specific rate of turbulence with turbulent kinetic energy and characteristic length:

$$\omega = constant\,\frac{k^{1/2}}{\ell_T}$$

the Specific Dissipation Rate of Turbulence Kinetic Energy can be shown below:

$$\rho\partial_0\omega + \rho U_j\partial_j\omega = \alpha'\frac{\omega}{k}\mu_T(\partial_j U_i + \partial_i U_j)\partial_j U_i - \beta'\rho\omega^2 + \partial_j\left[\left(\mu + \frac{\mu_T}{\sigma_\omega}\right)\partial_j\omega\right] + \alpha'(1-c_3)\beta\frac{\mu_T}{\sigma_T}\partial_i T g_i\frac{\omega}{k}$$

and the turbulent dissipation can shown by the kinetic energy and dissipation rate:

$$\varepsilon \equiv k\omega$$

3. ANALYSIS OF THE SIMPLIFIED IR BOX MODEL

3.1 GEOMETRY DESCRIPTION

In the first stage of modeling, some simplifications of geometry are applied to the IR Box model.

1 All the planes surround the box are set as wood.
2 The geometry of the window's vertical section is simplified as a rectangle, and named "panel"
3 The shape of the box is simplified as a rectangle, the infrared imager is also neglected
4 At the bottom of the box, air inlet is on the right side and the outlet is on the left side.

The geometry of the model is shown in Fig. 3.1 blow:

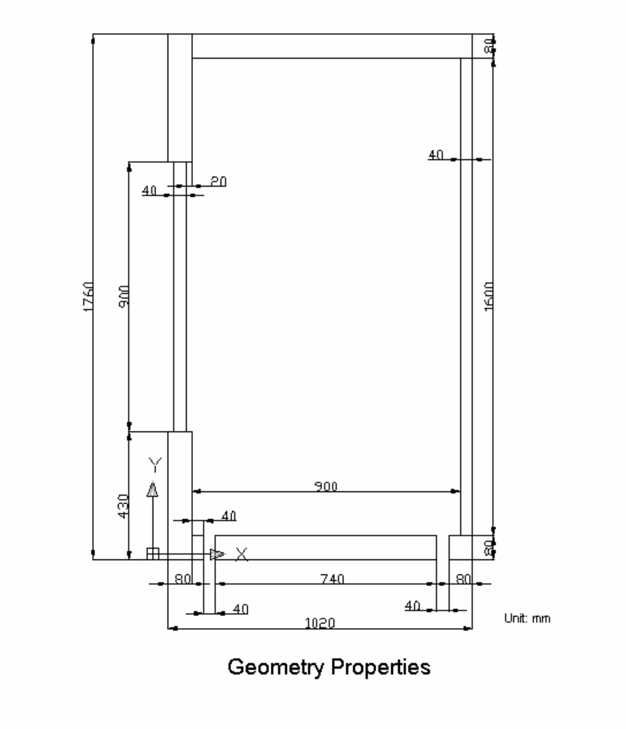

Fig. 3.1 Geometry properties of the IR Box model

3.2 BOUNDARY CONDITION AND INITIAL CONDITION

3.21 ESTIMATION OF THE TYPE OF FLOW

Rayleigh Number and Reynolds Number estimation

The formula below is used to calculate Rayleigh Number

$$Ra = \frac{\rho\beta(T_2 - T_1)L^3}{\mu\beta}$$

with the air properties, which are shown bellow:

Density	ρ=1.2067 (kg/m^3)
Volume expansion coefficient	β= 0.00342 (K^{-1})
Gravity	g=9.81 (m/s^2)
Length from cold side to warm side	L= 0.9 (m)
Viscosity	μ= 1.8088x10^{-5} (Ns/m^2)
Conductivity	k=0.02567 (W/mK)
Specific heat	Cp=1006.052 (J/kgK)
Environment temperature	Ti=21(^{o}C)
	To=-18(^{o}C)

The value of T_2, T_1 and Ra are based on the calculation in Tab.3.1

Tab. 3.1 Calculation of Ra number:

Loop	T_1 (^oC)	T_2 (^oC)	T_2- T_1 (^oC)	Ra
1	-17.9681	20.9808	38.9490	$3.0X10^9$
2	-0.0406	5.9020	5.9427	$4.5\ X10^8$
3	-0.0094	5.8757	5.8851	$4.54\ X10^8$
4	-0.0106	5.8768	5.8875	$4.54\ X10^8$
5	-0.0106	5.8768	5.8875	$4.54\ X10^8$

For the Reynolds number, the formula below is used:

$$\mathrm{Re} = \frac{\rho UL}{\mu}$$

ρ: Density ρ=1.2067(kg/m^3)
U: Velocity U=0.5(m/s)
L: Characteristic L=1.6(m)
μ: Viscosity $\mu = 1.8088x10^{-5}$ (Ns/m^2)

So, $Re=5.34x10^4$

With specific aspect ratio A, the ratio of the height and width of the cavity, the transition limit from laminar flow to turbulence is shown in Fig.3.1

Batchelor (1954) Laminar to Turbulent: $Ra = 10^9 (A)^{-3}$ for A < 42

$Ra = 1.37 \cdot 10^4$ for A > 42

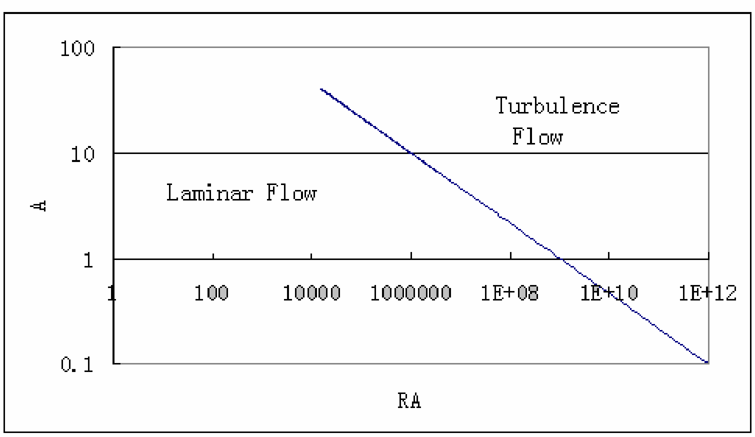

Fig. 3.1: The transition limit from laminar flow to turbulence flow

With the aspect ratio A=1.77 and Ra=4.54 $X10^8$, the air flow in the cavity is near the line. So the flow type in the cavity is transition flow.

3.22 DEFINITION OF BOUNDARY CONDITION

The material types and boundary conditions are shown in Figure 3.2

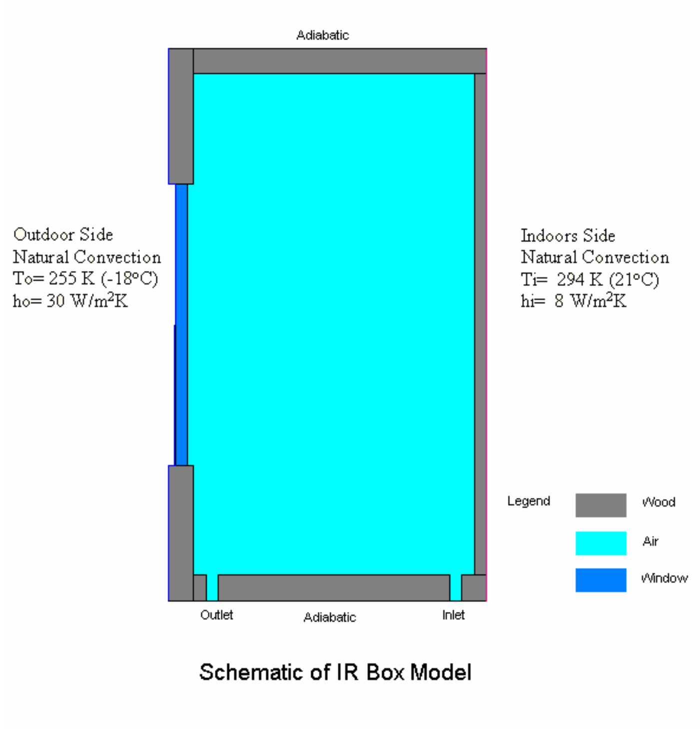

Fig. 3.2: Schematic of IR Boxe Model

The properties of the materials are shown in Tab. 3.2

Tab. 3.2 Material properties of the model

Material	Conductivity (W/mK)	Emissivity
Wood	0.11	0.90
Window	0.073	0.84

The Climate chamber's condition of the experiment platform is defined as boundary condition "Out door side". The boundary condition on the right side of the IR Box is defined as "Indoor side". The other were set as adiabatic boundary, which means no heat and mass exchange through these edges. The detail parameters of these boundary conditions are shown in Tab. 3.3.

Tab. 3.3 shows the indoor and outdoor side boundary conditions:

	Reference temperature	Convection coefficient
Indoor side	294 K (21^{o}C)	8 W/m^{2}K
Outdoor side	255 K (-18^{o}C)	30 W/m^{2}K
Others	Adiabatic	Adiabatic

Similar with the experiment, the simplified IR Box has air inlet and outlet. In this case, the inlet air speed is defined as 0.5 m/s, and the temperature of the air is 294 K (21^{o}C).

3.23 DEFINITION OF INITIAL CONDITION

The modeling of turbulence demands the initialization of the k and ε, especially the fluid flow with inlet and out let. With the recommendation of Fidap document, the formulas followed are used to specify the initial value of k and ε.

To estimate the initial kinetic energy k, the formula below is used:

$$k = \alpha u_{\infty}^{2}$$

α: Turbulence intensity, for this case, α=0.1.
u_∞: here, the velocity of the inlet air flow (0.5 m/s) is set to u_∞.
So, kinetic energy k≅ 0.002 m^2/s^2
To estimate the initial dissipation ε:

$$\varepsilon = \frac{k^{3/2}}{0.1\delta}$$

δ: Characteristic width of the shear layer at the inlet plane, here is the half with of the inlet, 0.02m.
Then ε≅ 0.044 m^2/s^3.

3.3 MESH DEVELOPMENT

In Fidap modeling, low-Re effects on the turbulence field in the near-wall region are predicted by means of a single layer of specialized near-wall elements. The k-ω model can be used in conjunction with a fine near-wall mesh to model both the mean flow variable and the turbulence variables near the wall and to resolve any geometric feature that may be present in the viscous sub-layers.

The software GAMBIT is used as mesh tool in this case. For the turbulence modeling, boundary layers are applied to the boundary of the air cavity. As the research purpose is focused on the heat transfer properties near the inside surface of the window, the mesh near the window is generated very fine. Fig. 3.4 shows the mesh of the IR Box model.

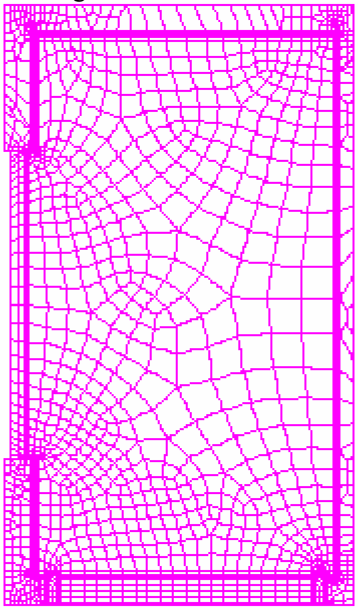

Fig. 3.4 Mesh of the IR Box model

Eight-node isoparametric quadrilateral elements were used in the finite element modeling of the IR Box. The eight-node quadrilateral elements use biquadratic shape functions to approximate the velocity and temperature. An eight-node quadrilateral element is shown in Fig. 3.5.

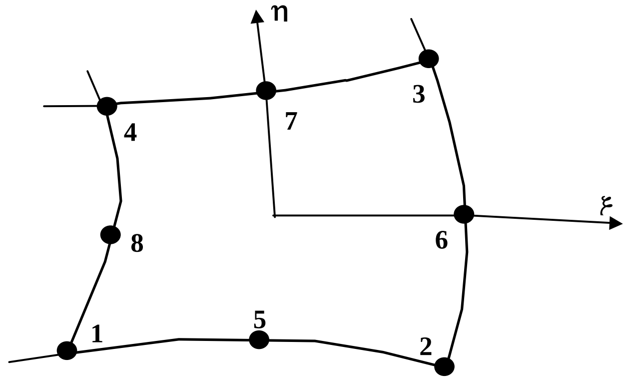

Fig. 3.5: Eight-node quadrilateral element

For these elements, linear shape functions are used to interpolate for the pressure after obtaining the velocity and temperature solutions. Normalized coordinates r, s, and t varying from –1 to 1 are used to express the shape functions N_u, N_P, N_T. Coordinate transformation using the Jacobian Matrix from the physical coordinates x, y, and z (global coordinates) to normalized coordinates r, s, and t (local coordinates) is:

$$J = \begin{pmatrix} \frac{\partial N^T}{\partial r}\mathbf{x} & \frac{\partial N^T}{\partial r}\mathbf{y} & \frac{\partial N^T}{\partial r}\mathbf{z} \\ \frac{\partial N^T}{\partial s}\mathbf{x} & \frac{\partial N^T}{\partial s}\mathbf{y} & \frac{\partial N^T}{\partial s}\mathbf{z} \\ \frac{\partial N^T}{\partial t}\mathbf{x} & \frac{\partial N^T}{\partial t}\mathbf{y} & \frac{\partial N^T}{\partial t}\mathbf{z} \end{pmatrix}$$

4. INTERMEDIATE RESULTS

1 Temperature contour of the simplified IR Box is shown in Fig. 4.1

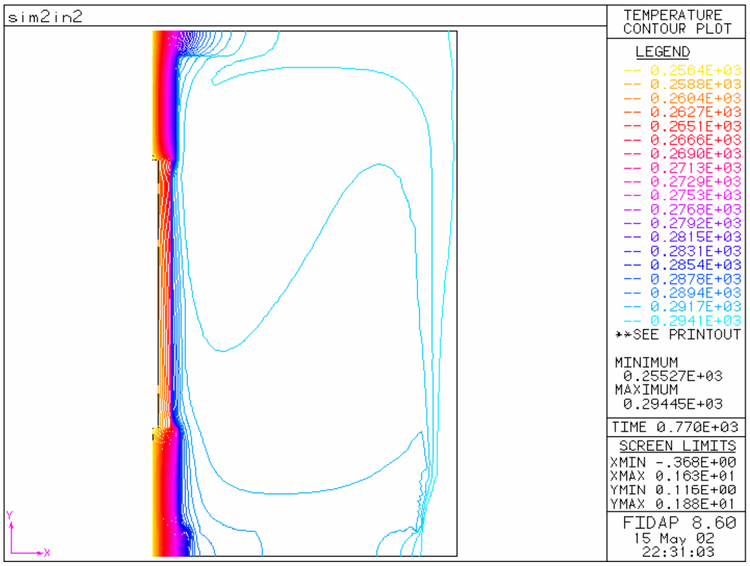

Fig. 4.1: The temperature contour of the whole simplified IR Box

2 Local heat flux along the surface of window is shown in Fig. 4.2

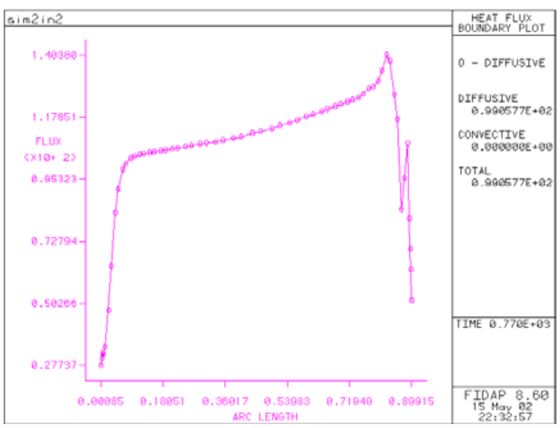

Fig.4.2: Local heat flux along surface of window

The average heat flow q_{ave}=110.64 (W/m^2)

3 Heat transfer coefficient distribution along the surface of window is shown in Fig. 4.3:

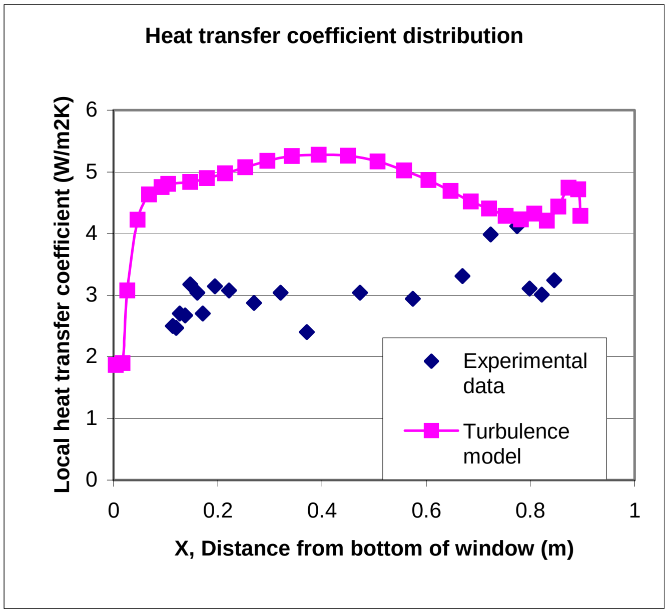

Fig. 4.3: Local heat transfer coefficient along the surface of window

$$h = \frac{q_s}{T_{ref} - T_s}$$

q_s: Local heat flux;
T_s: Local temperature;
T_{ref}: Reference temperature, here, the average temperature of the warm side of the IR Box is used as reference temperature, T_{ref}=12.7(^oC).

4 Vertical velocity (UY) profile

The velocity profile is obtained according to the different height along the Y-axis, which starts from the bottom of the window.

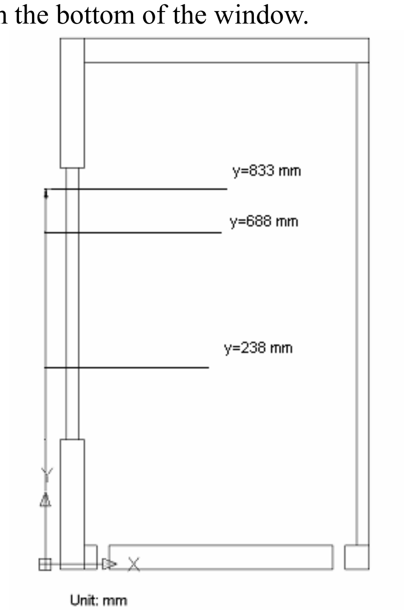

Fig.4.4: Obtain velocity profiles from different coordinate Y

The velocity profile from experimental data and turbulence model with radiation are shown below in different coordinate Y.

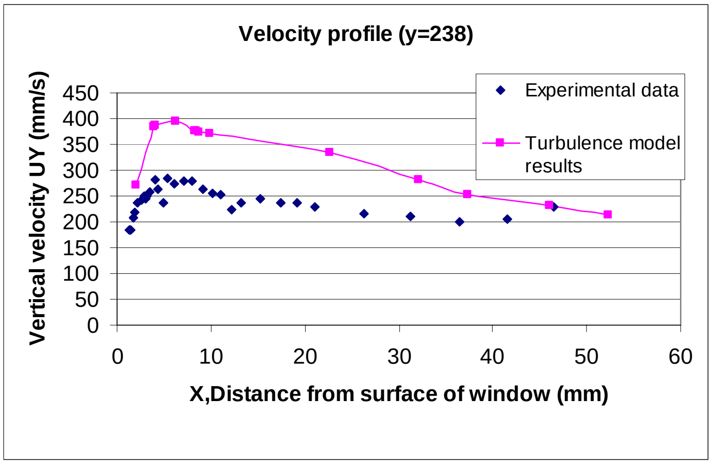

Fig. 4.5: Velocity profile of experimental data and turbulence model at 238mm from bottom of window.

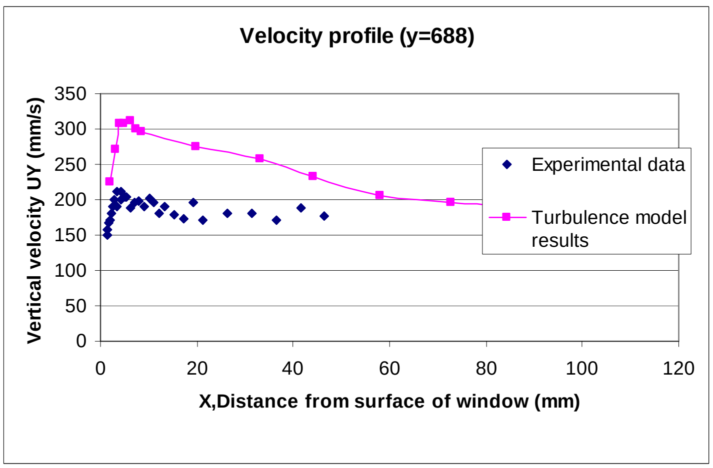

Fig. 4.6: Velocity profile of experimental data and turbulence model at 688mm from bottom of window.

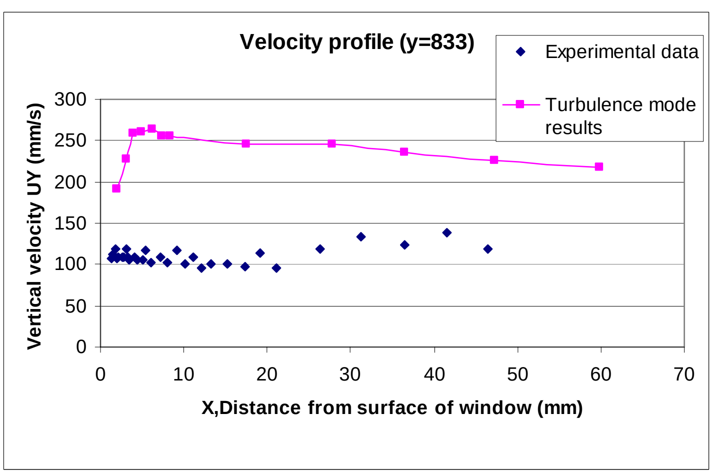

Fig. 4.7: Velocity profile of experimental data and turbulence model at 833mm from bottom of window.

6. RESULT DISCUSSION AND PLAN OF THE WORK

The curve shape of the simulation results matched the experimental date well, which means the simulation represent the fluid field's relative change very well. The value difference between the experiment data and simulation results mainly because the simulation did not include the radiation affect in the IR Box. The radiation will reduce the temperature difference, which will improve the simulation results of heat transfer coefficient and slower the velocity of the air flow.

For the next step of the IR Box modeling, radiation affect will be applied to the model, which will improve the result well. As some simplifications were applied, the detail will be added to the geometry, to get a closer model of the IR Box. As a transition flow, the natural convection flow in the cavity may have the characters of both turbulent flow and laminar flow. The result of laminar model is very necessary to be compared with the turbulent model and the experiment result.

7. REFERENCE.

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Power, J.P.. 1999. "Finite Element Model of Turbulent Flow and Heat Transfer in a Fenestration System". Ph.D. Thesis. Department of Mechanical Engineering, University of Massachusetts.

"Experimental Techniques for Measuring Temperature and Velocity Fields to Improve the Use and Validation of Building Heat Transfer Models"