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The Estimation of the Natural Convection Heat Transfer and Flow in Enclosure Modeling Steady-State Conditions in Laboratory Thermal Chambers

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For fenestration systems it is very important to know distribution of a film coefficient along fenestration surfaces to improve testing and analysis results of evaluation fenestration thermal properties. In this work we tried to answer the question whether pure natural convection can predict heat transfer and airflow in laboratory thermal chambers usually using fan and heater for creation standard testing conditions.

To estimate heat transfer and airflow in laboratory chamber we used the numerical modeling of 2-dimensional natural convection in rectangle enclosure with thermal conditions close to laboratory chambers testing thermal properties of windows. We consider a two-dimensional rectangle cavity with a cold left wall and a hot right wall. The horizontal walls are adiabatic. The height of the cavity is *H* and width is *W*. The geometry and boundary conditions are given in Figure 1.

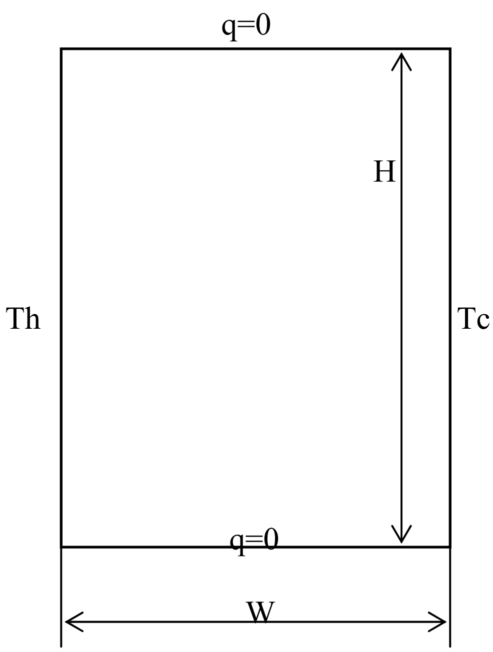

Figure 1. Geometry and boundary conditions of the modeled enclosure.

To compare results of numerical modeling with available experimental data [1] we defined the next geometric and temperature values:
H = 1.3m, W = 0.9m, Tc = 5.1C^o, Th = 35C^o.

For these values mean temperature Tm = (Tc + Th)/2 = 20.1C^o and Rayleigh number scaling by height is 6.7×10^9 that means presence of turbulence in the enclosure.

For modeling turbulent natural convection in the two-dimensional enclosure (Figure 1) we used Low-Reynolds-Number (LRN) k-ε turbulence model [1,2]. Numerical solution to the enclosure flows reported below were obtained using code FLU2TURB [2]. The results are compared with experimental measurements in the laboratory thermal chamber [3] used fan and heater.

Figure 2 displays the predicted vertical velocity along cold wall (modeling inside glazing surface at an enclosure) at the locations: Y = 0.25, Y = 0.5, Y = 0.75. The maximum value vertical velocity is 0.315

m/sec close to the location of transition from laminar to turbulent natural convection (Y = 0.45). The maximum vertical velocity observed in [3] is 0.28 m/sec at location Y ≈ 0.2 from the glazing bottom.

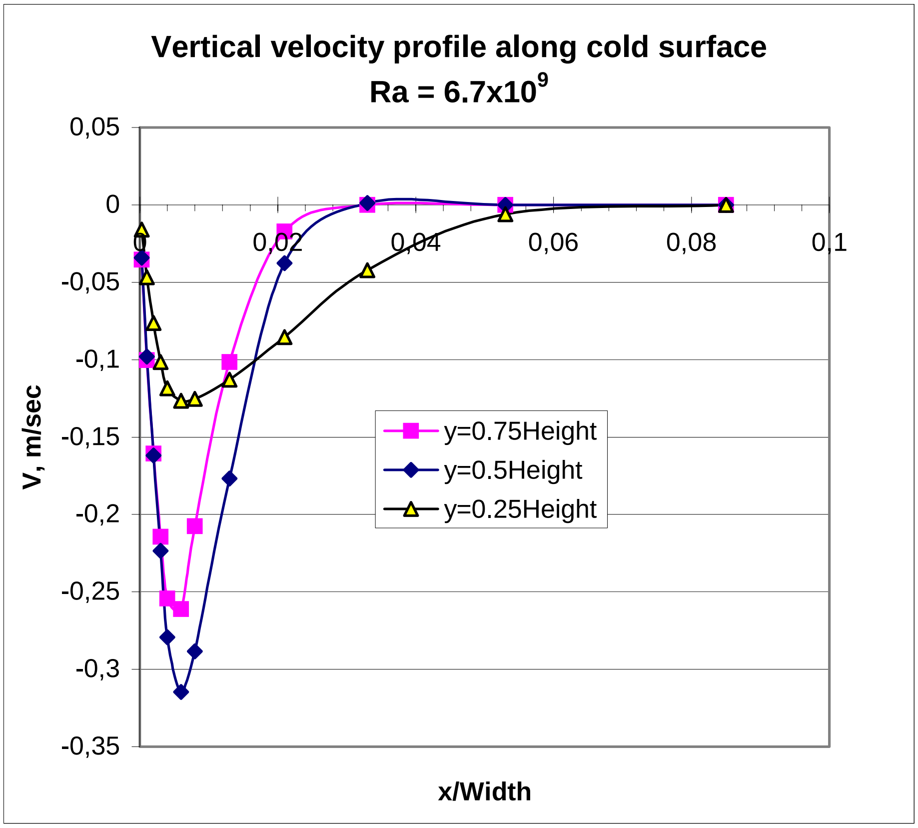

Figure 2. Predicted vertical velocity profiles along cold wall of the enclosure.

Predicted by LRN k-ε model average Nusselt number (Nu_{av}) is 95.0. The average convection film coefficient on cold wall surface can be determine from equation

$$h_c = Nu_{av}\, k_m / H, \qquad (1)$$

where k_m - air conductivity at mean temperature air in an enclosure.
From (1) we obtain for our conditions

$$h_c = 95 * 0.0263 / 1.3 = 1.92 \text{ W/(m2 C}^o).$$

The average convection film coefficient on window glazing obtained in experimental work [3]:

$$h_c \approx 3 \text{ W/(m2 C}^o).$$

Figure 3 displays the predicted distribution of local Nusselt number along the cold wall and Figure 4 displays the predicted distribution of local convection film coefficient along the cold wall.

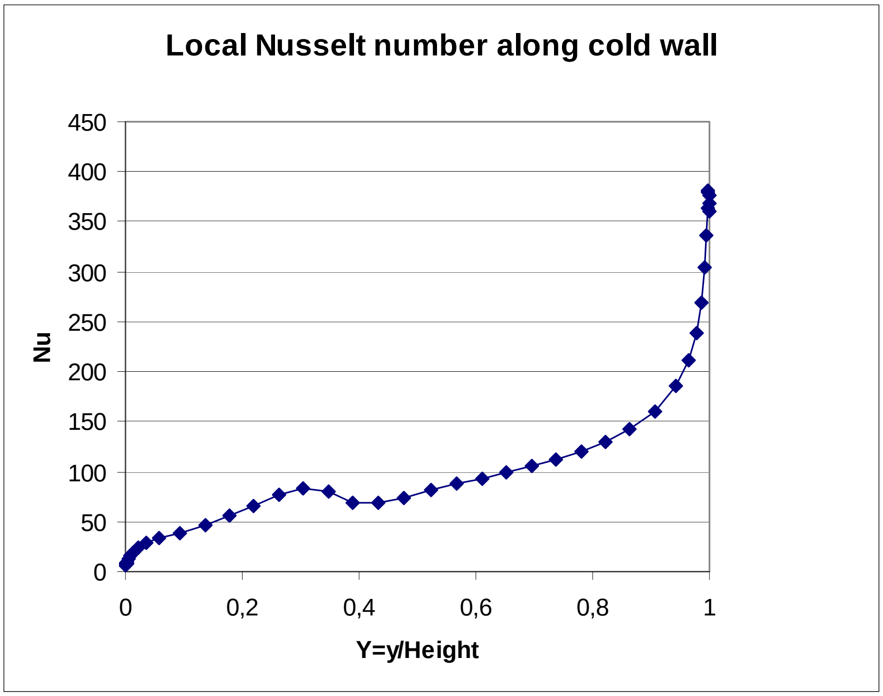

Figure 3. Distribution the local Nusselt number along the cold wall of the enclosure.

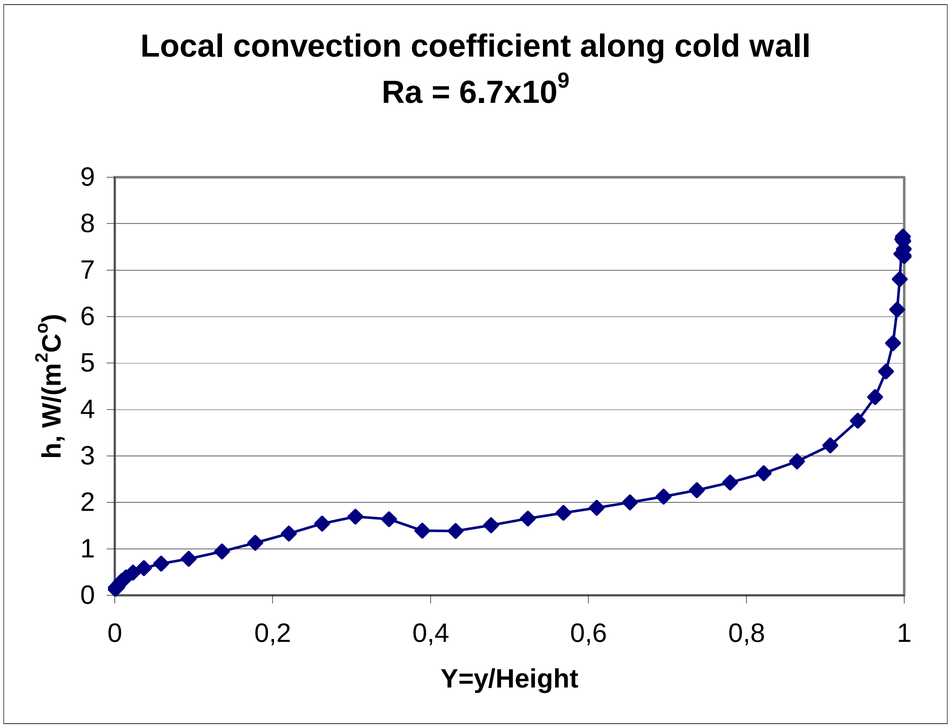

Figure 4. Distribution the local convection film coefficient along the cold wall of the enclosure.

To be sure in our conclusions we carried out modeling natural convection in the enclosure with dimensions H$*$W = 1.0m$*$0.9m at Ra = $3.1\text{x}10^9$. We obtained following results: Nu = 74.35, average convection film coefficient h_c = 1.93 W/(m2 C^o) and maximal vertical velocity V = 0.270 m/sec along cold wall of the enclosure.

CONCLUSIONS

Low-Reynolds-Number k-ε turbulence model predicts average Nusselt number Nu = 95.0, average convection film coefficient h_c = 1.92 W/(m2 C^o) and maximum value vertical velocity V = 0.315 m/sec along cold wall of the enclosure 1.3m$*$0.9m at Ra = $6.7\text{x}10^9$ and Nu = 74.35, average convection film coefficient h_c = 1.93 W/(m2 C^o) and maximum value vertical velocity V = 0.270 m/sec along cold wall of the enclosure 1.0m$*$0.9m at Ra = $3.1\text{x}10^9$.
These values were compared with experimental data obtained for a dual-glazed, low-emittance, wood-frame window in laboratory thermal chamber [3]: maximum value vertical velocity V = 0.280 m/sec along glazing and average convection film coefficient on window glazing h_c = 3 W/(m2 C^o).
From the foregoing results, it is concluded that the pure natural convection cannot predict heat transfer and airflow in laboratory thermal chambers using fan and heater for creation necessary testing conditions.

REFERENCES

1. T. J. Heindel, S. Ramadhyani, and F. P. Incropera, 1994, “Assessment of Turbulence Models for Natural Convection in an Enclosure”, Num. Heat Transfer, Part B, 26:147-172.
2. “Study of Turbulent Natural Convection Flow in Rectangular Enclosure”, report CEERE, UMASS, February 2002.
3. “Experimental Techniques for Measuring Temperature and Velocity Fields to Improve the Use and Validation of Building Heat Transfer Models”, LBL, CEERE UMASS, draft.