Center for Energy Efficiency and Renewable Energy
C E E R E
Building Energy Efficiency Program

University of Massachusetts
Department of Mechanical and Industrial Engineering
160 Governor's Dr.
Amherst, MA 01003-9265

Laboratory Report

Numerical Study of Mixed Convection Heat Transfer and Flow in Enclosure Modeling Steady-State Conditions in Laboratory Test Chamber

Part 1: Modeling Glazing System in LBL Laboratory Chamber

May, 2002

The representation of the thermal boundary conditions of a fenestration surfaces has a major influence on the accuracy of predicting the thermal properties of a fenestration products. To represent internal surface heat-transfer rates of glazing are usually used average values of film coefficient for the entire surfaces. Because these values are averages, they are often not accurate for specific locations on the surface. Local surface heat-transfer film coefficients can be derived from data of numerical modeling of heat-mass transfer in testing chambers with fenestration specimens. Computational fluid dynamics (CFD) is being extensively used for this modeling.

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. The main aim of this study to assess the ability of a low-Reynolds-number (LRN) k-ε turbulence model to predict heat transfer and fluid flow along vertical cooled flat plate which thermal properties and boundary conditions simulated glazing system. The existence of non-uniformity of glazing thermal properties and flow separation due to a sudden change in fenestration geometry plays an important role in distribution of heat transfer coefficient along fenestration surfaces.

In this part of the work we describe mixed convection simulation of conditions of LBL laboratory thermal chamber designed for testing thermal performance of fenestration products [1]. Numerical calculations have been performed for cooled flat plate with uniform and non-uniform surface thermal resistance along the plate and are compared with experimental measurements in the laboratory thermal chamber [1] used fan and heater.

1. Numerical model and method

In our study we used LRN k-ε turbulence model with variable coefficients described in details in [2] as VC LRN k-ε turbulence model. Therefore we do not give here the model equations that are available in ref. [2] and our report [3].

A computer code named FLU2TURB has been developed to solve the two-dimensional steady turbulent problem. The numerical discrete method is based on the upwind and fully implicit transient differencing control volume scheme used respectively for the convective, diffusive and time-dependent terms in the governing equations where the velocity control volumes are staggered with respect to the main control volumes. The resulting algebraic equations are solved iteratively using a line-by-line TDMA solution procedure and the SIMPLE algorithm formulated by Patankar [4]. Steady-state solutions are obtained using under-relaxation techniques.

To model radiation heat exchange in the enclosure with glazing we used simple radiation model based on the approximate formula [5]

$$q = 5{*}10^{-8}\,(\,T_a^4 - T_s^4), \qquad (1)$$

where q - heat flux on a glazing (or insulation wall) surface, W/m^2;

T_a - average temperature of a chamber surfaces surrounding wall with glazing, K;

T_g - temperature of a glazing (or wall) surface, K.

2 Glazing system as cooled flat plate with uniform and non-uniform thermal resistance

2.1 The geometry and boundary conditions

To compare results of numerical modeling with experimental data [1] we defined the geometric dimensions and temperature and inlet flow conditions close to the conditions used in the experiment. Figure 1 shows geometry and boundary conditions of the enclosure used for modeling flow along cooled vertical flat plate. We consider a two-dimensional rectangle cavity with a cold left wall and with inlet and outlet slots at the bottom. The height of the cavity is *H* (1.4 m) and width is *W* (0.9 m). The geometry and boundary conditions are given in Figure 1.

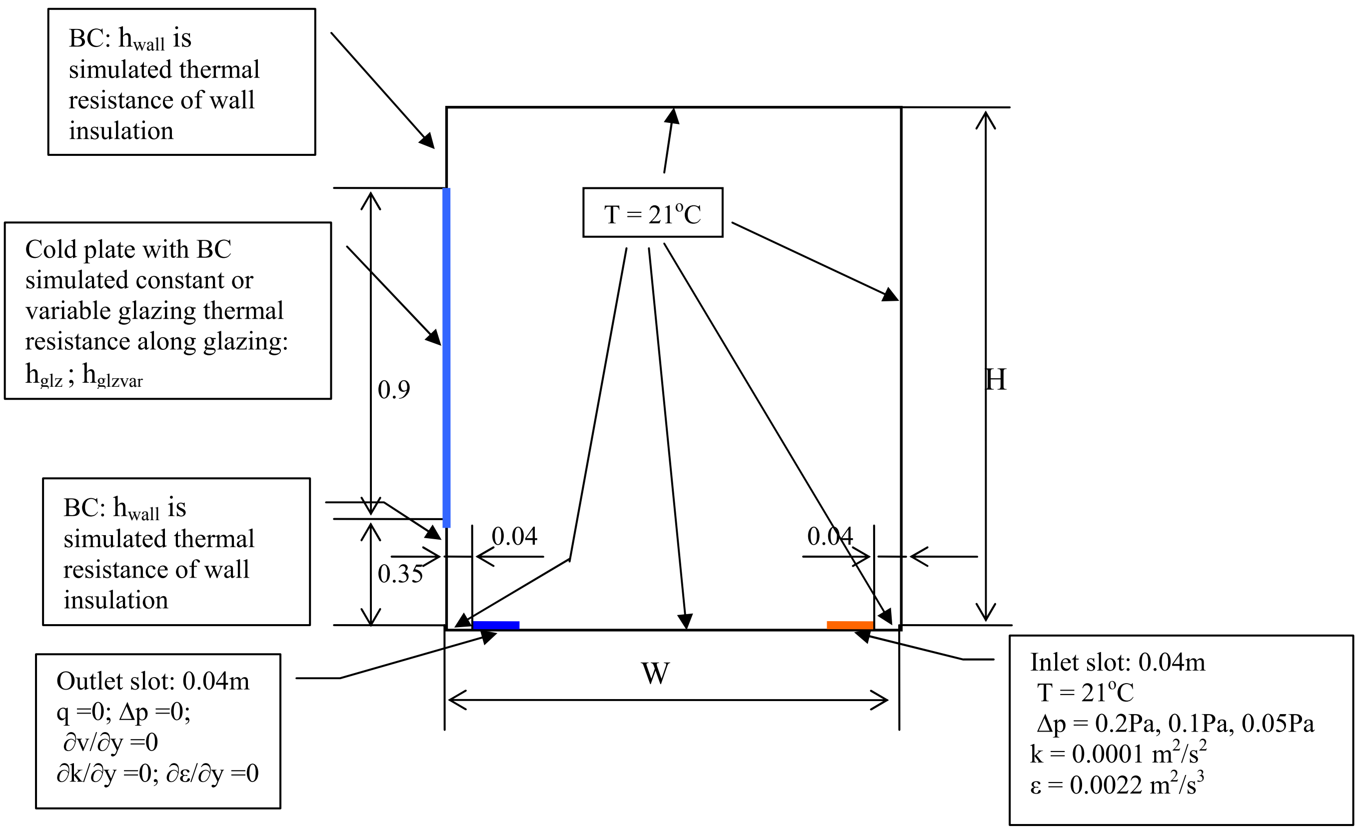

Figure 1. Geometry and boundary conditions of the modeled enclosure.

It is necessary to explain how we model thermal resistance and outer film coefficient of glazing system and surround insulation wall using various boundary conditions. Total thermal resistance of a glazing system (or wall) is given by formulae

$$Ro = 1/U_{factor} = 1/h_{out} + R_g + 1/h_{in}, \qquad (2)$$

where h_{out} - outside film coefficient,
R_g – thermal resistance of glazing (central part),
h_{in} - inside film coefficient,
U_{factor} - total thermal transmittance of glazing (central part).

Inside film coefficient is equal the sum of two components: convection heat-transfer coefficient h_c and radiation coefficient h_r that are unknown values and need to determine them using corresponding convection and radiation models. From (2) we derive expression for boundary heat-transfer coefficients that are simulated thermal properties of glazing (or wall) for steady-state conditions

$$h_{glz} = 1/(1/ h_{out} + R_g), \qquad (3)$$

or for known value of glazing U-factor

$$h_{glz} = 1/(1/U_{factor} - 1/h_{instand}), \qquad (4)$$

where $h_{instand} = 7.0$ W/(m^{2o}C) - standard value of inside film coefficient.

In the present study we used the next values: total thermal transmittance of glazing (central part) $U_{factor} = 1.37$ W/(m^{2o}C); standard value of inside film coefficient $h_{instand} = 7.0$ W/(m^{2o}C); outside film coefficient $h_{out} = 25$ W/(m^{2o}C); boundary film coefficient h_{glz} that is modeling glazing thermal resistance and outer film coefficient on glazing surface is 1.7 W/(m^{2o}C); boundary film coefficient h_{wall} = 0.6 W/(m^{2o}C).

In order to determine temperature and heat flux distribution on glazing surface more accurate without calculation convective motion in IGU cavity we used simple method developed by Curcija and et al [6] for representation heat transfer rate through glazing as a set of the constant radiation heat transfer coefficient and variable convective heat transfer coefficient along vertical sides of glazing based on correlations derived from the experimental and numerical works. To transform the variable transfer coefficient to boundary film coefficient we used formulae (4). Figure 2 shows distribution of defined boundary film coefficient along plate that is modeled glazing.

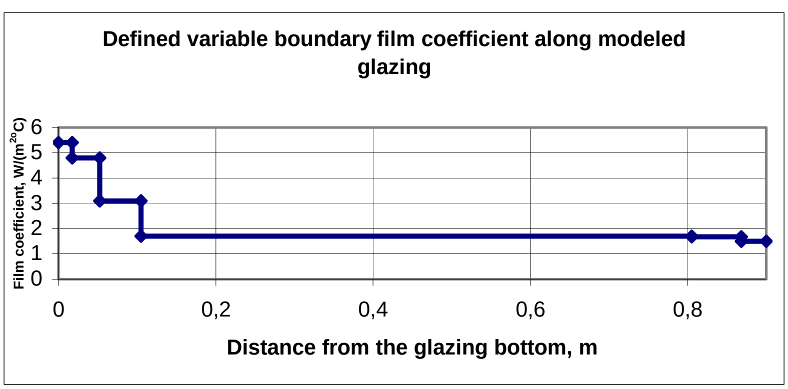

Figure 2. Chart of defined film coefficient along cold plate that is modeling glazing.

2.2 Results and discussion

The fluid flow vector plot in the top and bottom parts of testing box is given in Figure 3.

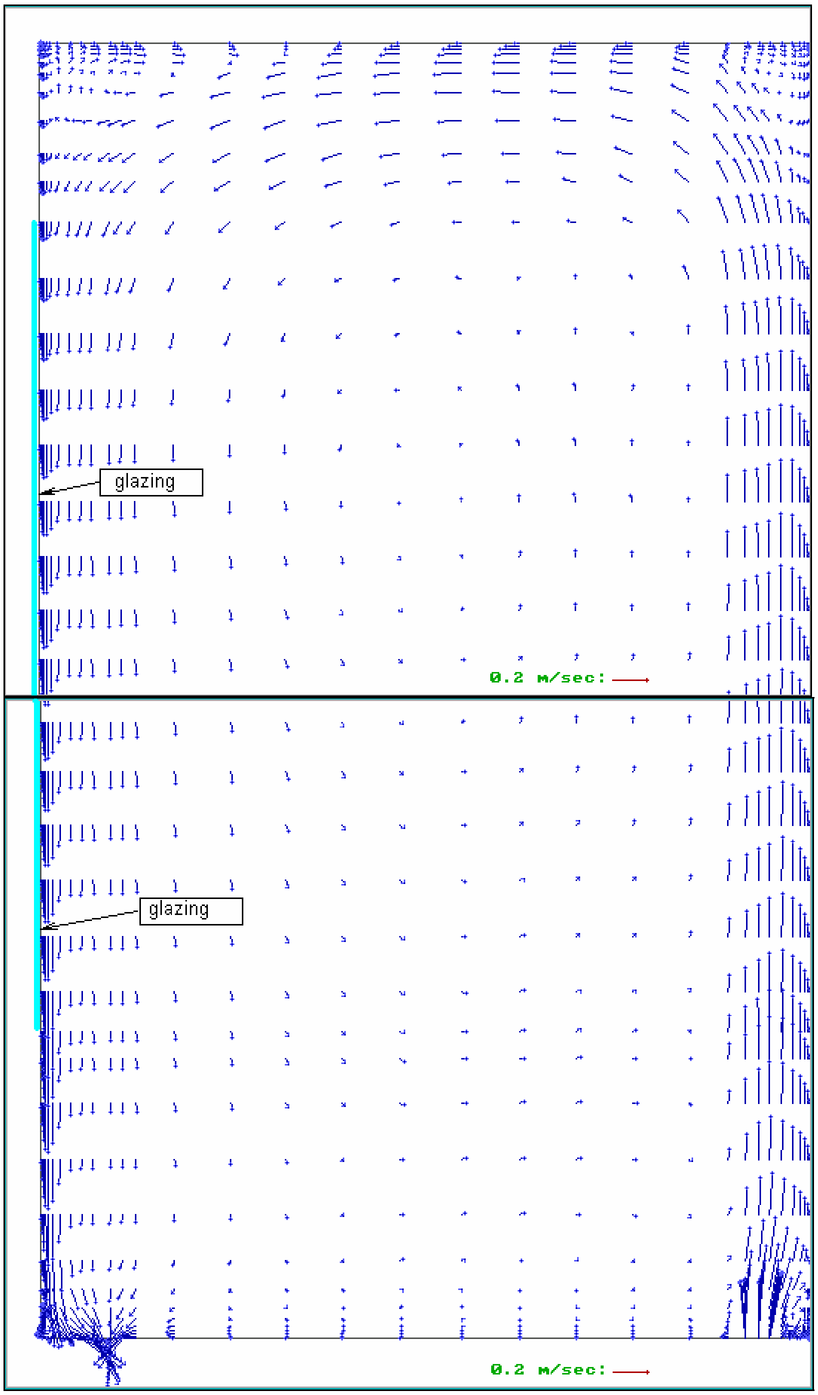

Figure 3. The fluid flow vector plot in the testing box.

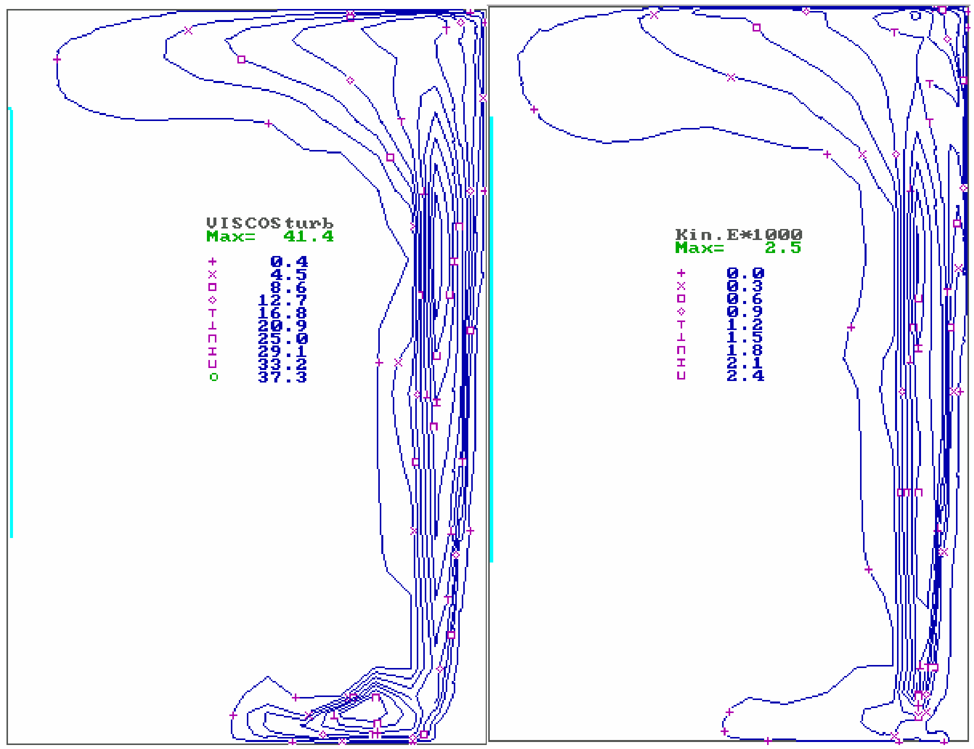

Figure 4. Distribution of no dimensional turbulent viscosity and turbulent kinetic energy, $k{*}10^3$ m^2/s^2.

The predicted no dimensional turbulent viscosity and turbulent kinetic energy distribution are shown at Figure 4. LRN k-ε model predicted maximum turbulent kinetic energy to $2.5\text{x}10^{-3}$ m^2/s^2 and maximum no dimensional turbulent viscosity $\nu_t/\nu = 41.4$. From distribution of turbulent quantities it is clear that the glazing does not undergo influence of turbulence disturbing and we have not fully developed turbulence regime in modeled box. Character of heat transfer on glazing surface in this case is more close to laminar than to turbulent regime.

The distribution (see Fig. 4) does not depend practically from inlet velocity and inlet values of turbulent kinetic energy and dissipation of turbulent kinetic energy but is very sensitive (as can see below) to geometry of the test box.

Figure 5 displays the predicted vertical velocity along cold flat plate (modeling inside glazing surface at an enclosure) at the locations from the bottom of plate: Y = 0.542 m and Y = 0.772 m. The maximum value of vertical velocity is 0.27 m/s for inlet velocity 0.45 m/s and 0.25 m/s for inlet velocity 0.33 m/s close at the location (H = 0.542 m). The maximum vertical velocity observed in [1] is 0.22 m/s at location H ≈ 0.62 m from the glazing bottom (size of glazing area tested in [1] is 0.83 m).

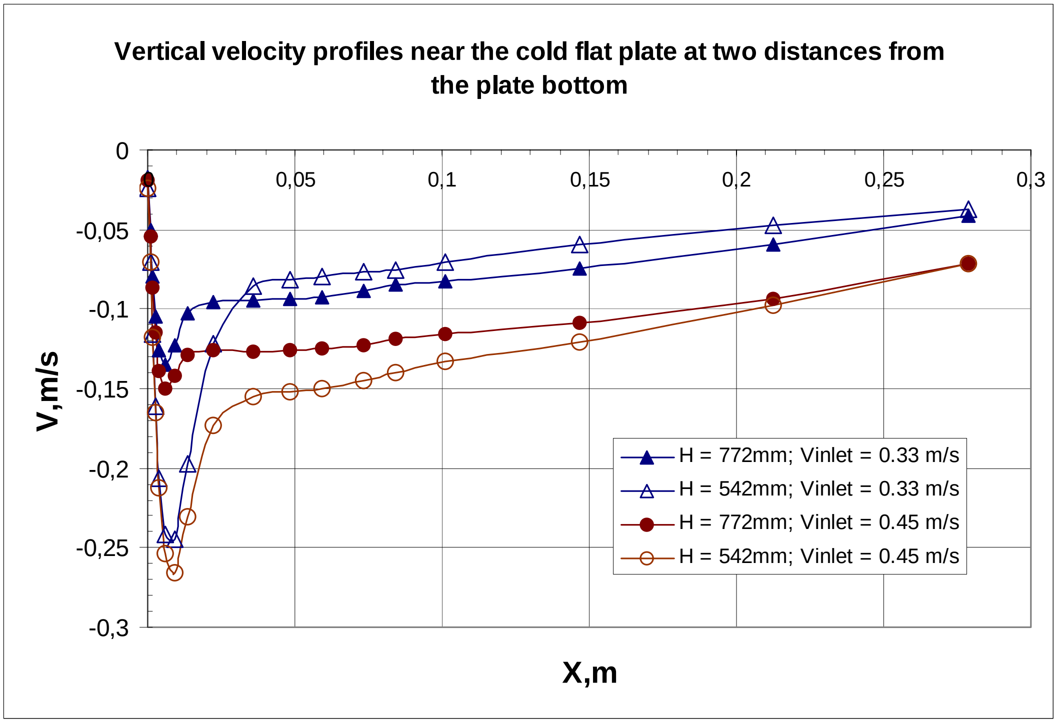

Figure 5. Predicted vertical velocity profiles near the flat plate modeled glazing.

Figure 6 and Figure 7 are shown temperature and heat flux distributions along cold wall with modeled glazing for two inlet velocities: 0.45 m/s and 0.33 m/s. To obtain various velocities we defined pressure differences: 0.2 Pa, 0.1 Pa and 0.05 Pa, and inlet velocity was correspondingly 0.45 m/s, 0.33 m/s and 0.29 m/s. From Figure 6 and Figure 7 it is seen that temperature and heat flux distributions do not depend on inlet velocity until value of 0.29 m/s (here is not shown).

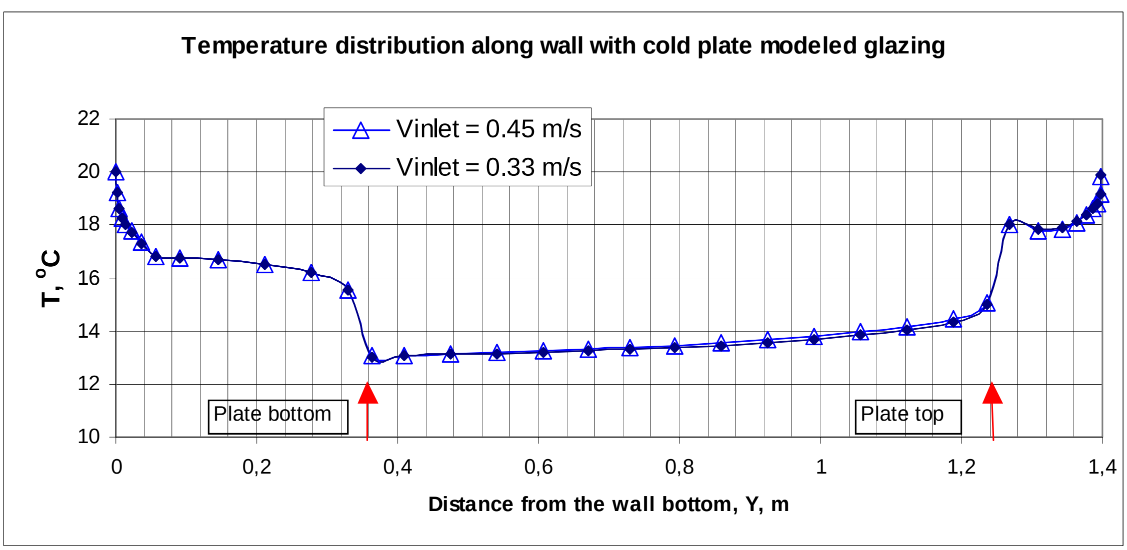

Figure 6. Predicted temperature distribution along cold wall with flat plate that is modeling glazing.

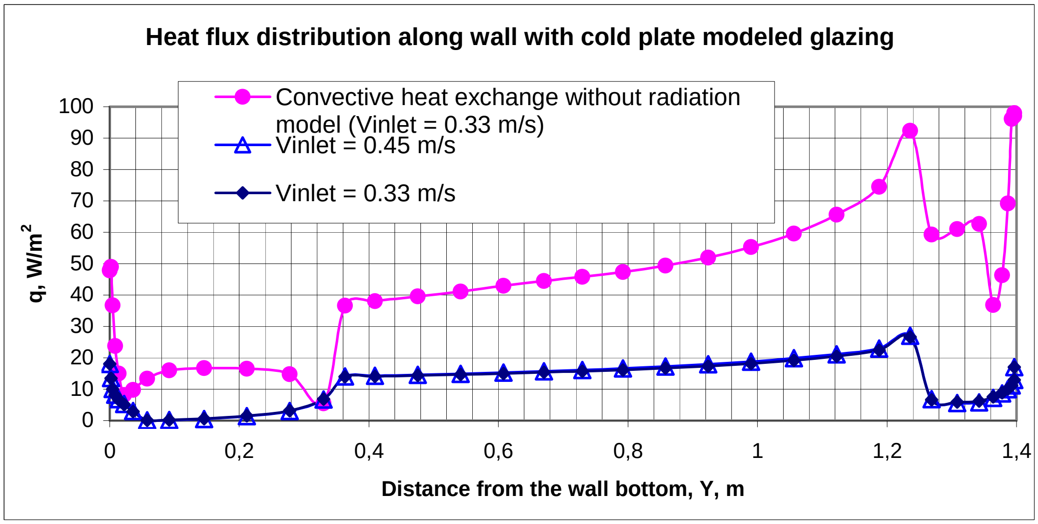

Figure 7. Comparison between heat flux distributions along cold wall with radiation heat exchange and without one.

Figure 8, Figure 9 and Figure 10 are shown temperature, heat flux and convection heat transfer coefficient distributions along modeled glazing with uniform and non-uniform thermal resistance along its length. Convection heat transfer coefficient was determined from equation

$$h_c = q_c/(T_{ref} - T_s), \tag{5}$$

where q_c - convection heat flux;
T_s - surface temperature;
T_{ref} - reference (inlet) temperature (21°C).

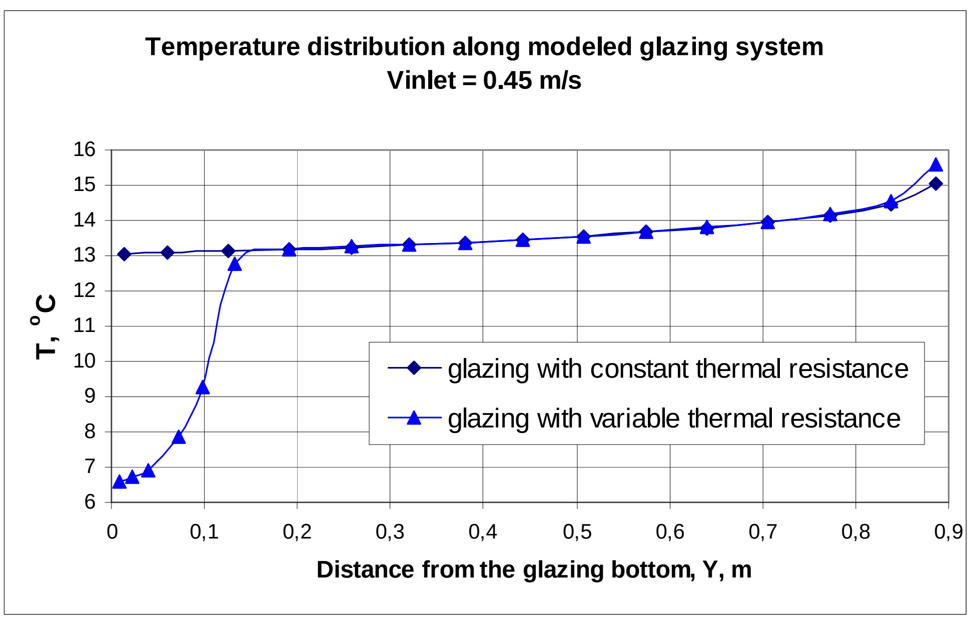

Figure 8. Comparison between temperature distributions along modeled glazing with uniform thermal resistance and variable one along its length.

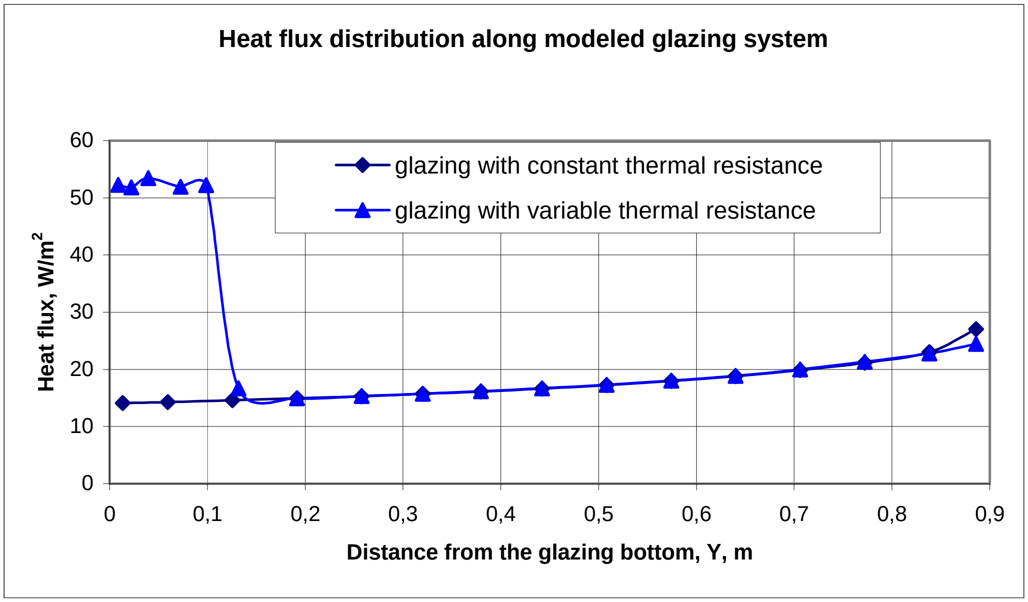

Figure 9. Comparison between heat flux distributions along modeled glazing with uniform thermal resistance and variable one along its length.

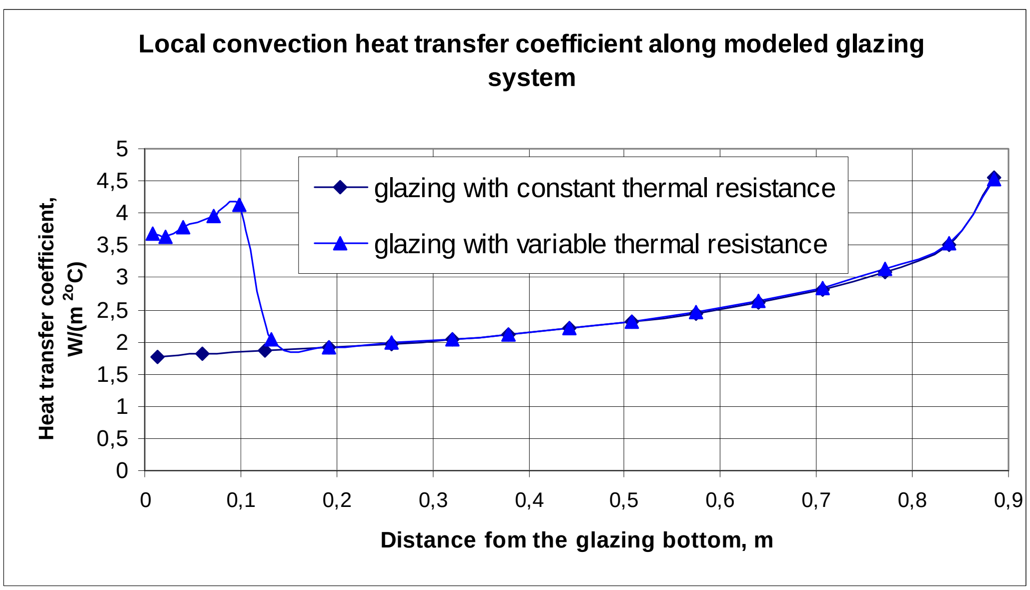

Figure 10. Comparison between convection heat transfer coefficient distributions along modeled glazing with uniform thermal resistance and variable one along its length.

The results of this stage of the study show us that temperature distribution on modeled glazing surface and heat transfer rate and turbulent quantities do not depend on inlet velocity in the range from 0.29 m/s to 0.45 m/s and initial values of inlet values of turbulent kinetic energy and dissipation of turbulent kinetic energy. Used modeling of glazing with variable thermal resistance along its length gives temperature distribution more close to experimental data. Predicted average value of radiation heat transfer coefficient on glazing surface was 4.94 W/(m^{2o}C). For mean temperature 20.5C^o and obtained average glazing temperature 13.0C^o Rayleigh number scaling by glazing height 0.9 m is 6.1×10^8.

3 Glazing system as cooled plate with glass and non-uniform thermal resistance and surrounded wall steps

3.1 The geometry and boundary conditions

In this part of the work we studied how wall steps (or fenestration steps) have an effect on flows and heat transfer rate for glazing surface. To make temperature distribution along glazing more smooth we incorporated glass (3 mm thickness) in the model of glazing. The geometry and boundary conditions are given in Figure 11.

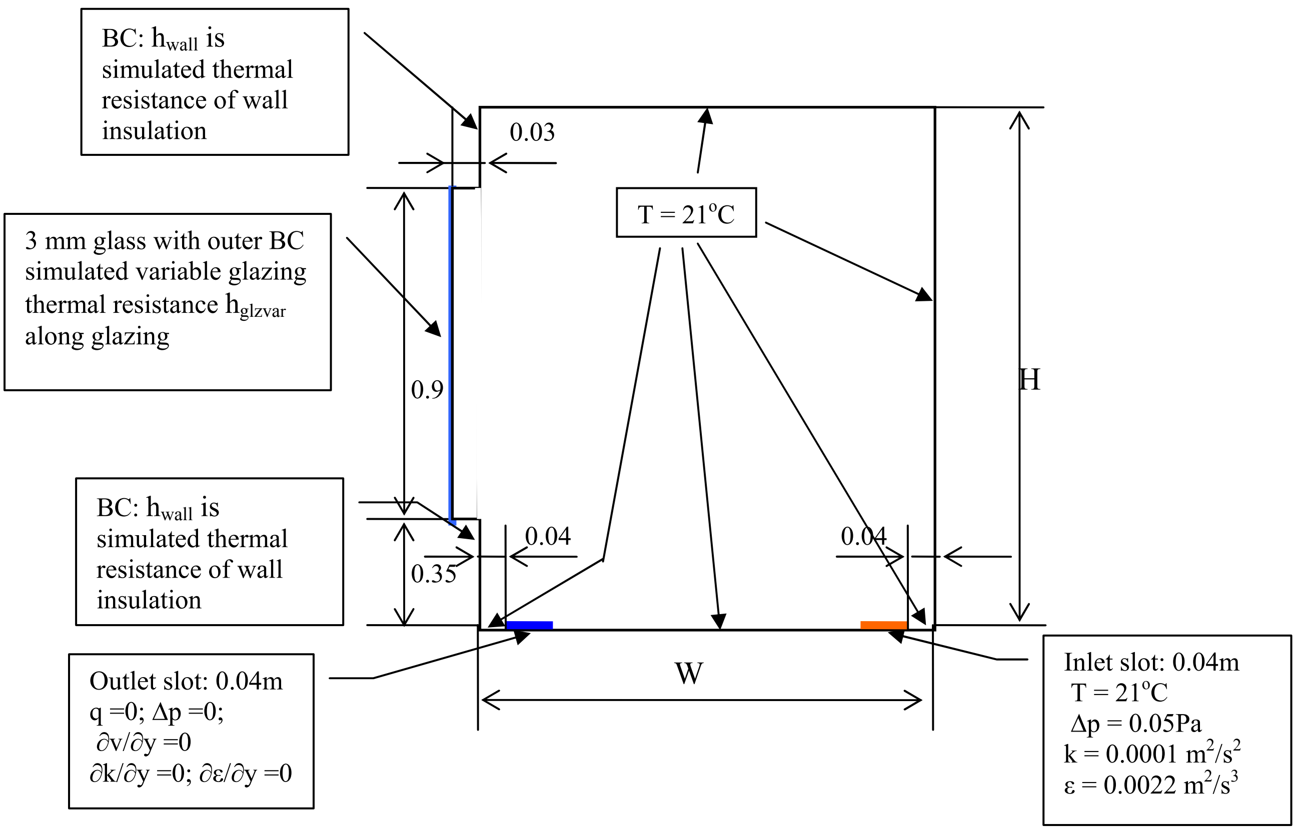

Figure 11. Geometry and boundary conditions of the modeled enclosure.

3.2 Results and discussion

The fluid flow vector plot in the bottom part of testing box is given in Figure 12. The predicted no dimensional turbulent viscosity and turbulent kinetic energy distribution are shown at Figure 13. For the glazing with wall steps LRN k-ε model predicted maximum turbulent kinetic energy to 7.4×10^{-3} m^2/s^2 and maximum no dimensional turbulent viscosity $\nu_t/\nu = 66.5$. In this case the glazing is within turbulence disturbing limits and heat transfer on glazing surface has turbulent character.

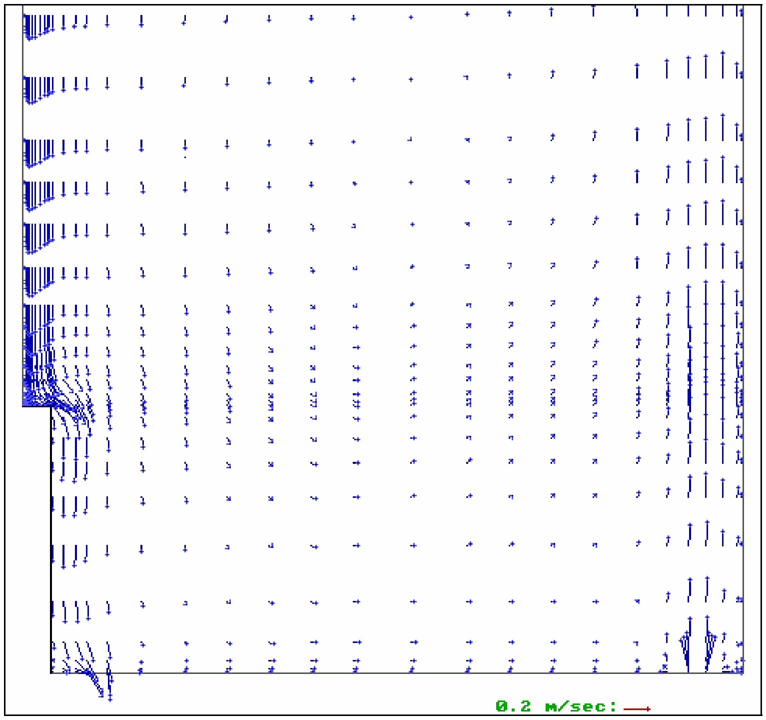

Figure 12. The fluid flow vector plot in the bottom part of testing box.

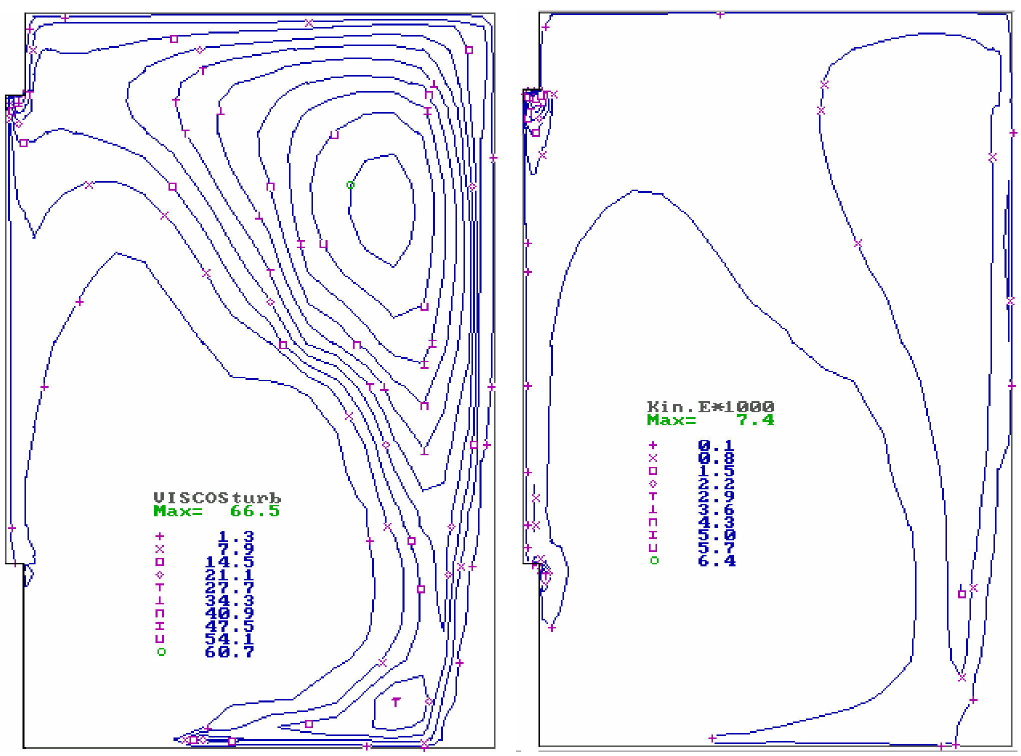

Figure 13. Distribution of no dimensional turbulent viscosity and turbulent kinetic energy, $k*10^3$ m^2/s^2.

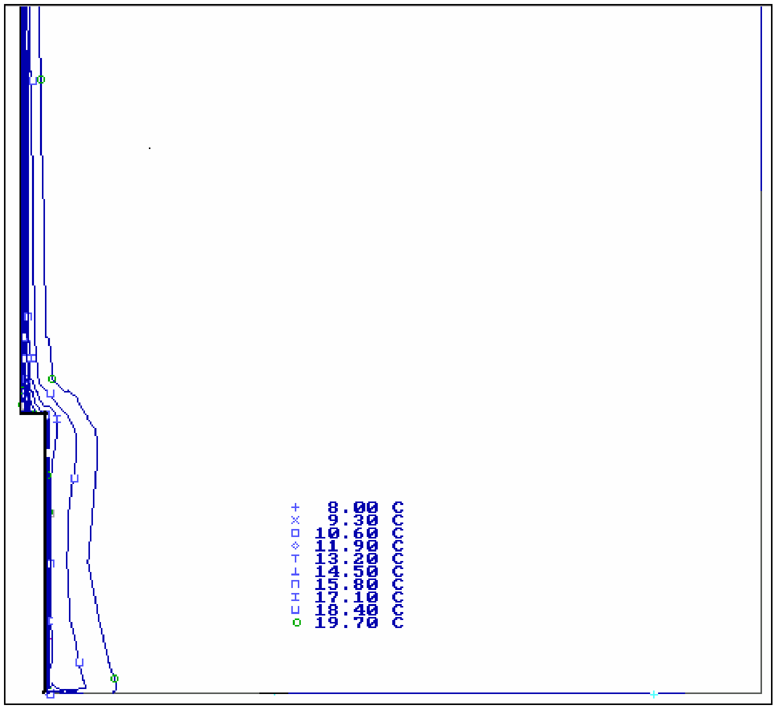

Figure 14. Contour plot of temperature distribution in the bottom part of modeled test box.

Figure 15, 16 and 17 are shown temperature, heat flux and convection heat transfer coefficient distributions along glazing for laminar and turbulent regimes.

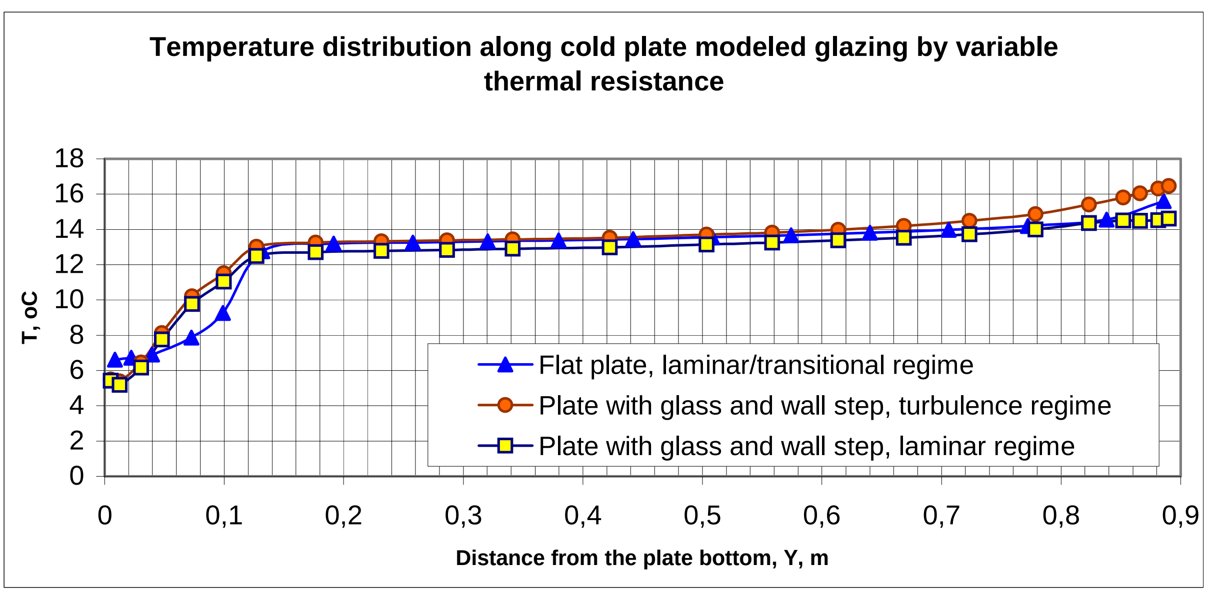

Figure 15. Comparison between temperature distributions along various glazing models.

It should be noted that the temperature distributions for laminar and turbulent regimes are very close to each other and that is why impossible to determine what regime exist in glazing area using only experimental information about temperature on glazing surface.

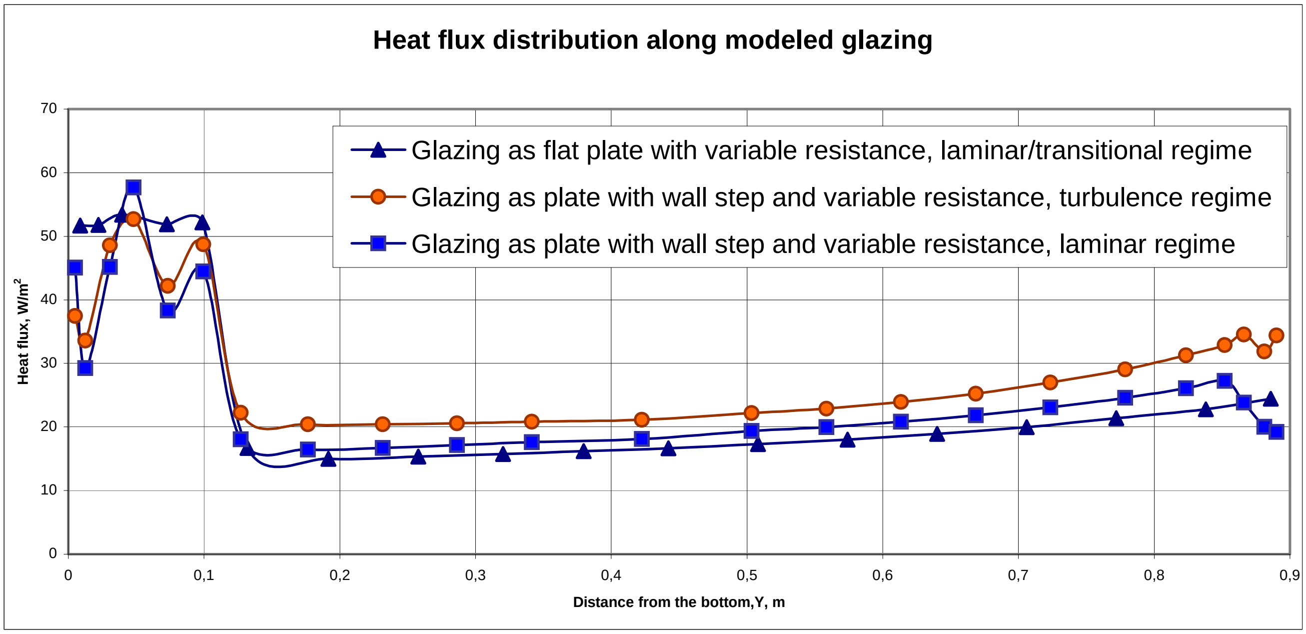

Figure 16. Comparison between heat flux distributions along modeled glazing.

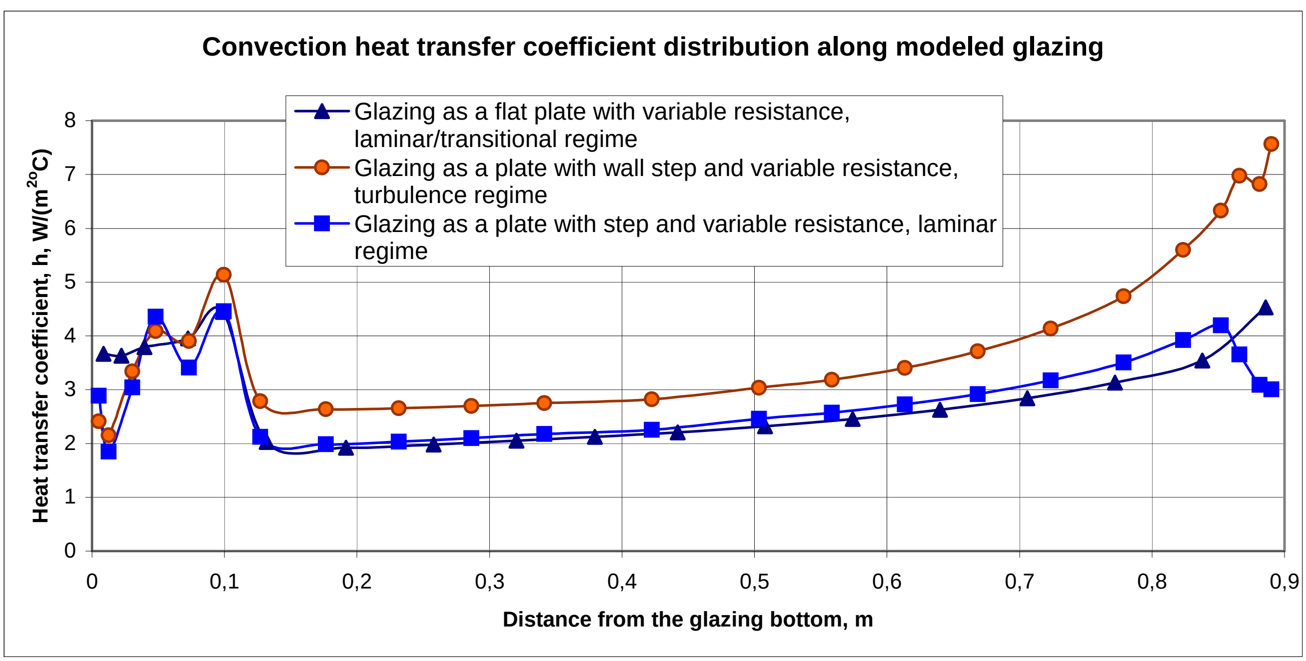

Figure 17. Comparison between convection heat transfer coefficient distributions along modeled glazing.

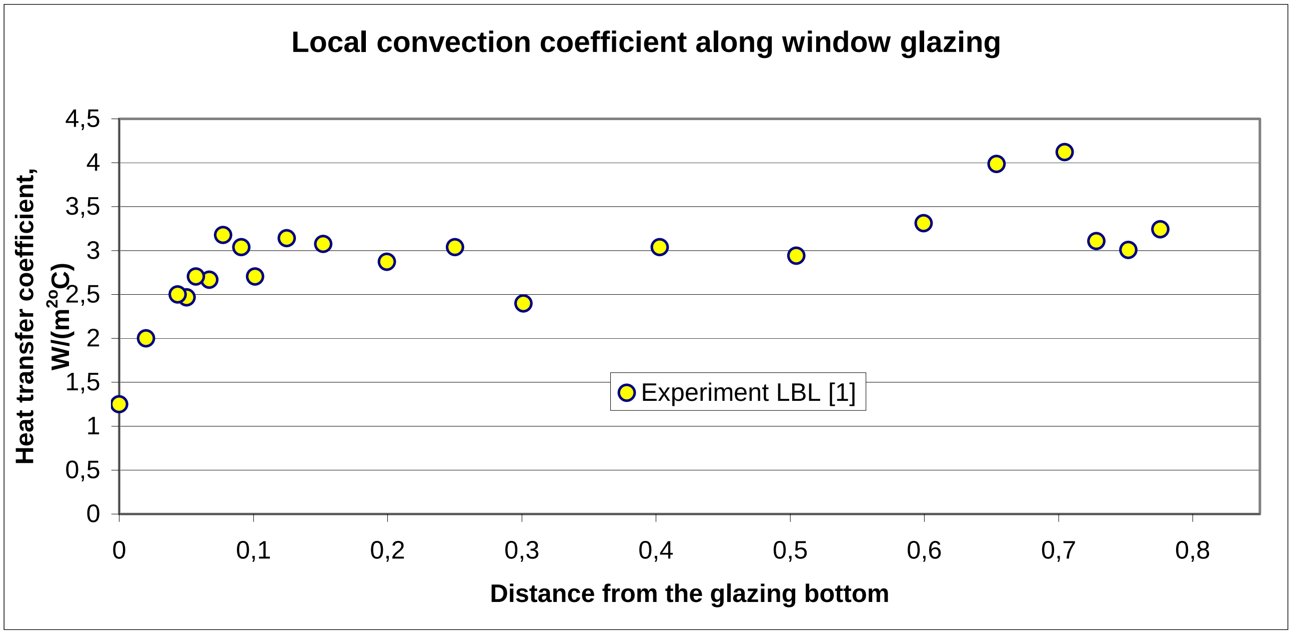

Figure 18. The distribution of convection heat transfer coefficient on window glazing obtained in experimental work [1].

The comparison between measured data [1] (see Figure 18) and predicted value of convection heat transfer coefficient on glazing surface (see Figure 17) shows that does exist turbulence regime in testing chamber in glazing area. Values of average heat transfer coefficient on glazing surface are given in table 1.

Table 1. Comparison between predicted and experimental values of average heat transfer coefficient on glazing surface.

Glazing model	Convection coefficient, $W/(m^{2o}C)$	Radiation coefficient, $W/(m^{2o}C)$	Total coefficient, $W/(m^{2o}C)$
Glazing: flat plate	2.7	4.94	7.64
Glazing with wall steps; laminar model	2.8	4.94	7.74
Glazing with wall steps; turbulent model	3.7	4.94	8.64
Experiment [1]	3.3	N/A	-

CONCLUSIONS

The low-Reynolds-number k-ε turbulence model and simple radiation model were used to predict heat transfer and fluid flow along vertical cooled flat plate with wall steps in enclosure with inlet and outlet slots modeled heat supplying in test chamber [1].

Temperature distribution on modeled glazing surface and heat transfer rate and turbulent quantities do not depend on inlet velocity in the range from 0.29 m/s to 0.45 m/s and initial values of inlet values of turbulent kinetic energy and dissipation of turbulent kinetic energy. Used modeling of glazing with variable thermal resistance along its length gives temperature distribution more close to experimental data. To increase accuracy of modeling it is necessary to model geometry of test box and fenestration products more exactly.

Low-Reynolds-Number k-ε turbulence model predicts average convection film coefficient h_c = 3.7 W/(m2 C^o) and maximum value of vertical velocity V = 0.27 m/s along modeled glazing with wall steps. These values were compared with experimental data obtained for a dual-glazed, low-emittance, wood-frame window in laboratory thermal chamber [1]: maximum value vertical velocity V = 0.28 m/sec along glazing and average convection film coefficient on window glazing h_c = 3.3 W/(m2 C^o).

From the foregoing results, it is concluded that the LRN k-ε model does a satisfactory job of predicting natural and mixed convection flow and heat transfer along vertical cooled/heated flat plate with wall (fenestration) steps and can be recommended for modeling of heat-mass transfer and air flows along glazing surfaces of fenestrations in test chambers.

REFERENCES

1. “Experimental Techniques for Measuring Temperature and Velocity Fields to Improve the Use and Validation of Building Heat Transfer Models”, laboratory report LBL, CEERE UMASS, draft, 199? .
2. T. J. Heindel, S. Ramadhyani, and F. P. Incropera, “Assessment of Turbulence Models for Natural Convection in an Enclosure”, Num. Heat Transfer, Part B, 26: pp 147-172, 1994.
3. “Study of Turbulent Natural Convection Flow in Rectangular Enclosure”, laboratory report CEERE, UMASS, February 2002.
4. S. V. Patankar, “Numerical Heat Transfer and Fluid Flow”, McGraw-Hill, New York, 1980.
5. Рекомендации по расчету светопрозрачных конструкций зданий с учетом светотехнических, теплотехнических, звукоизоляционных качеств и технико-экономических показателей. НИИСФ, Москва, 1985, 85 с.
6. D. C. Curcija, Yie Thao, J. Power, W. P. Goss, “Condensation Resistance Modeling”, BERL UMASS, 199?.

Measured velocity data near window glazing in LBL test chamber