Numerical analysis of heat transfer in a double glass window

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Abstract: This paper presents a numerical approach to study conductive and convective heat transfer in a double glass window. A tow-dimensional steady state model developed, based on fundamental equations of mass, momentum, and energy conservation and it was solved by finite volume method. Boundary conditions were assumed to be convection heat transfer on two sides, and zero heat flux at bottom and top. The problem was solved for four different window heights, each one with ten different glass spacing. Two different gap filling gas (Air and Argon), and two different frame materials (UPVC and Aluminum) were used in problem. To compare the results, a similar problem was solved for a single glass window too. Optimum glass spacing is obtained for different window's heights which is 14mm for 0.5m height and 17mm for other heights. Maximum heat transfer reduction for optimum glass spacing was about 82% and average of decrease in heat transfer for optimum cases was 79% in comparison with single glass.

Keywords: Double glass window, Natural convection, Numerical modeling

Nomenclature:

A: aspect ratio

g: gravitational acceleration (m/s^2)

h: heat transfer coefficient (W/m^2k)

H: window's height (m)

K: thermal conductivity of air (W/mk)

L: window's width (m)

Nu: Nusselt number

P: Pressure (pa)

u: x-direction velocity (m/s)

v: y-direction velocity (m/s)

Greek Symbols

α: thermal diffusivity (m^2/s)

β: thermal expansion coefficient (K^{-1})

ν: kinematic viscosity (m^2/s)

Subscripts

∞: reference ambient

Pr:	Prandtl number	c:	cold ambient
q:	heat transferred from window (W)	h:	heated ambient
Ra:	Rayleigh number	m:	average of heated and cold ambient
T:	temperature (˚C)	s.g.:	single glass

1 Introduction

Windows are weak obstacles against heat transfer. In cold climate they are responsible for 10-25% of heat loss from the heated ambient to the cold surrounding. A double glass window can be modeled as a rectangular cavity, filled with a gas. Convection heat transfer in gas layers within rectangular cavities with side walls each placed in different ambient (Figure 1), appears in many engineering applications and hence a great deal of investigations was dedicated to better understanding of the problem [1–4]. There are so many articles available, including great amount of experimental data. Wright [2] compiles and discusses available experimental data and correlations.

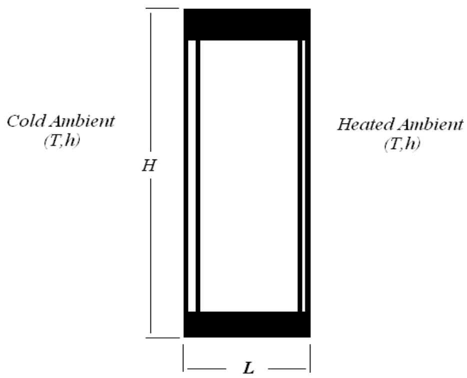

Fig.1 Rectangular cavity

The main goal, from air-conditioning point of view, is to compute the total heat transfer through the cavity, i.e. the average Nusselt number. The objective of the work presented here is to obtain optimum glass spacing for different window's height, and also study the role of frame and filling gas properties on heat transfer from window. The problem was solved for four different window heights; 0.5m, 1m, 1.5m and 2m. For each window height ten different glass spacing; 5mm, 6mm, 7mm, 8mm, 10mm, 12mm, 13mm, 14mm, 17mm, and 19mm was studied to achieve minimum heat transfer for each case. Two different filling gases (Air and Argon) and also tow different frame materials (UPVC and Aluminum) was checked for one of the cases above to study their effect on conductive and convective heat transfer.

2 Natural Convection Analysis

In most practical problems of convective transport to or from a rigid surface, the flow in the vicinity of the body is in turbulent motion. However, at the solid–fluid interface itself, the no-slip boundary condition ensures that turbulent velocity fluctuations vanish. Thus, at the wall, diffusive transport of heat and momentum in the fluid is precisely expressible by the laws applicable to laminar flow. There is

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a thin but very important sub-layer immediately adjacent to the solid surface where the transport of heat and momentum is predominant by molecular diffusion.

It is customary to quantify the non-radiative heat flux (q) which includes convective as well as conductive heat transfer, in a non-dimensional format. The average Nusselt number for non-radiative heat transfer across the air layer (Nu) is defined as

$$Nu = \frac{q\,L}{k\,(Th-Tc)} \qquad (1)$$

where k (W/m K) is the thermal conductivity of air at T_m; L (m) the width of air layer (see Fig. 1); q (W/m^2) the average non-radiative heat flow across air layer; T_c (˚C) the temperature of cold ambient (outside); T_h (˚C) the temperature of heated ambient (inside); and T_m is the temperature at which air properties are evaluated $(T_c+T_h)/2$.

Theory shows that Nu is a function of the Rayleigh number (Ra_L), Prandtl number (Pr), and aspect ratio (A). Therefore, the functional dependence of the correlations is of the form

$$Nu = f(Pr, Ra_L, A) \qquad (2)$$

where Ra, Pr, and A are defined as

$$Ra_L = \frac{gL^3\beta\,(Th-Tc)}{\alpha\nu} \qquad (3)$$

$$Pr = \frac{\nu}{\alpha} \qquad (4)$$

$$A = \frac{H}{L} \qquad (5)$$

α (m^2/s) is the thermal diffusivity of air at T_m; g (m/s^2) the acceleration of gravity; H (m) the height of air layer (see Fig. 1); β (K^{-1}) the thermal expansion coefficient of air; and ν (m^2/s) is the kinematic viscosity of air at T_m.

The value for Pr of air at 20 ˚C is 0.71. Gases used for window applications have only slightly different values for Pr at this temperature (argon: 0.68, krypton: 0.65, xenon: 0.66 and sulfur hexafluoride 0.68). These differences in Pr are of minor importance for convective heat transfer only. Hence, for the previously mentioned applications, Eq. (2) can be written as

$$Nu = f(Ra_L, A) \qquad (6)$$

All thermo physical properties were assumed to be constant except for density, which is treated with the Boussinesq approximation. The flow was assumed to be two-dimensional and steady state.

3 Numerical Method

In this work, to quantify the heat transfer due to natural convection, numerical simulations were used to solve the equation of energy and the equation of momentum using the CFD code. Governing equations are mass, momentum, and energy. The mass conservation equation can be written ass

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad (7)$$

The equation of motion in x and y direction can be written as

$$\frac{\partial(uu)}{\partial x} + \frac{\partial(uv)}{\partial x} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{\mu}{\rho}\frac{\partial^2 u}{\partial x^2} + \frac{\mu}{\rho}\frac{\partial^2 u}{\partial y^2} - g \qquad (8)$$

$$\frac{\partial(uv)}{\partial x} + \frac{\partial(vv)}{\partial x} = -\frac{1}{\rho}\frac{\partial p}{\partial y} + \frac{\mu}{\rho}\frac{\partial^2 v}{\partial x^2} + \frac{\mu}{\rho}\frac{\partial^2 u}{\partial y^2} \qquad (9)$$

The Boussinesq approximation can be written in the form

$$\rho - \rho_\infty \cong -\rho\beta\,(T - T_\infty) + \cdots \qquad (10)$$

Substituting in equation (8) we have

$$\frac{\partial(uu)}{\partial x}+\frac{\partial(uv)}{\partial x}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\frac{\partial^2 u}{\partial x^2}+\frac{\mu}{\rho}\frac{\partial^2 u}{\partial y^2}-g\beta(T-T_\infty) \quad 11)$$

The two dimensional energy equation, without the viscous dissipation term can be written as

$$\frac{\partial(uT)}{\partial x}+\frac{\partial(vT)}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \quad (12)$$

The boundary conditions are stated as follows

Temperature field:

at $x=\frac{L}{2}$: $q=-k\left(\frac{\partial T}{\partial x}\right)=h_h\ (T_{(L/2,y)}-T_h)$

at $x=-\frac{L}{2}$: $q=-k\left(\frac{\partial T}{\partial x}\right)=h_c\,(T_{(-L/2,y)}-T_c)$

at $y=\frac{H}{2}$: $\frac{\partial T}{\partial y}=0$

at $y=-\frac{H}{2}$: $\frac{\partial T}{\partial y}=0$ (13)

Momentum equations:

at $x=\frac{L}{2}$: $u\left(\frac{L}{2},y\right)=v\left(\frac{L}{2},y\right)=0$

at $x=-\frac{L}{2}$: $u\left(-\frac{L}{2},y\right)=v\left(-\frac{L}{2},y\right)=0$

at $y=\frac{H}{2}$: $u\left(x,\frac{H}{2}\right)=v\left(x,\frac{H}{2}\right)=0$

at $y=-\frac{H}{2}$: $u\left(x,-\frac{H}{2}\right)=v\left(x,-\frac{H}{2}\right)=0$ (14)

This problem has been analyzed by a laminar tow-dimensional steady state finite volume method. Conservation equations are solved by second order upwind discritisation scheme and a variant of SIMPLE solving algorithm, described by Patankar [5]. To be able to have precise results near the glass surface in the spacing, grids are chosen to be much smaller in this area. Figure 2 shows an example of a non-uniform, structured, Cartesian grid used in simulations. Depending on the size of the cavity, grids were chosen in different sizes and quantity. Grid independence studies showed that across quite wide range grid spacing exerted a very minor influence on the calculated heat transfer only.

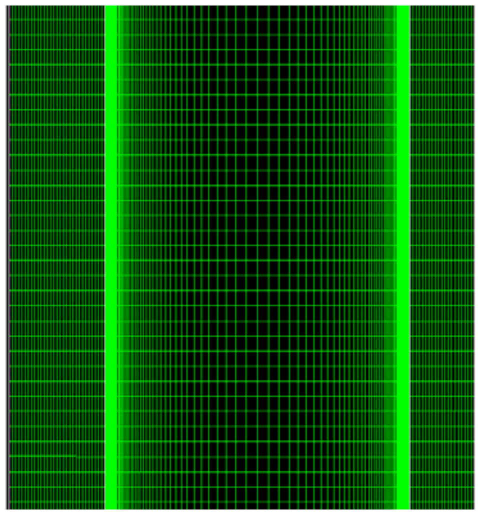

Fig.2 Grid, including glasses on left and right sides and spacing in the middle

Heated ambient and cold ambient temperature was assumed to be 20˚C and -10˚C respectively. Also convection heat transfer coefficients were assumed to be constant and equal to 20 W/m^2k for hot side and 10 W/m^2k for cold side. Other than the gas density which is assumed to follow the Boussinesq approximation, the other thermo physical properties are assumed to be constant. At first, the frame material and the filling gas were chosen to be UPVC and air respectively. Glasses' thickness for inside and outside was 4mm and 6mm respectively. Frame thickness was 100mm and it is assumed that there is no heat transfer between the frame and surrounding wall. It is obvious that heat transfer mechanism is convection for glass spacing and tow ambient-faced sides, and it is conduction through glasses and frame.

Heat transfers between frame and glasses by conduction and no contact resistance was assumed. A schematic of double glass window discussed in this paper is brought in figure 3.

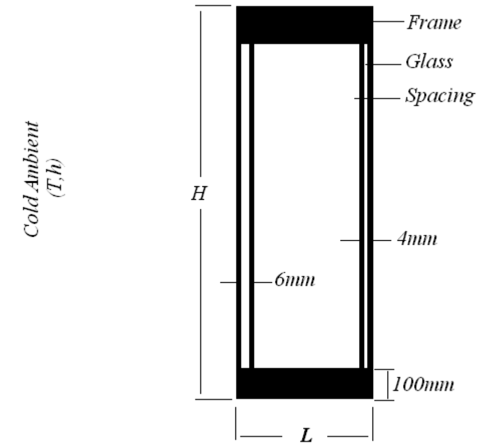

Fig.3 Schematic of the double glass window

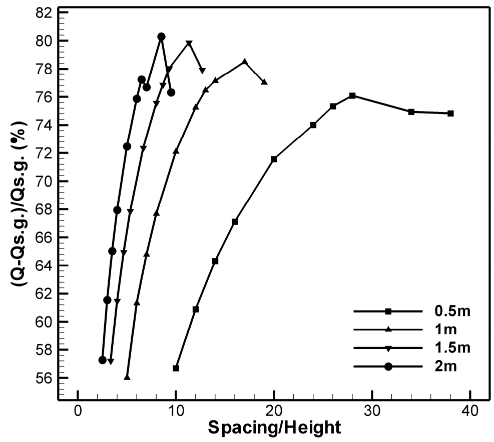

Fig.4 Percentage of decrease in heat transfer in a double glass window in comparison with single glass for different window's heights

4 Results and Discussion

Solving the problem shows a noticeable reduction in convection heat transfer, in comparison with a single glass window. Minimum and maximum of these energy saving is 56% and 82% respectively. The reason can be the thermal resistance of the gas, which the spacing is filled with. Figure 4 shows the percentage of heat transfer reduction in different window's height and spacing, in a non-dimensional form.

The best glass spacing to minimize the heat transfer was 14mm for window's heigth of 0.5m and 17mm for other window's heights, i.e. 1m, 1.5m, and 2m. The optimum glass spacing for different window's height can be seen in figure 5. The amount of heat transfer for each spacing and height is also brought in figure 5.

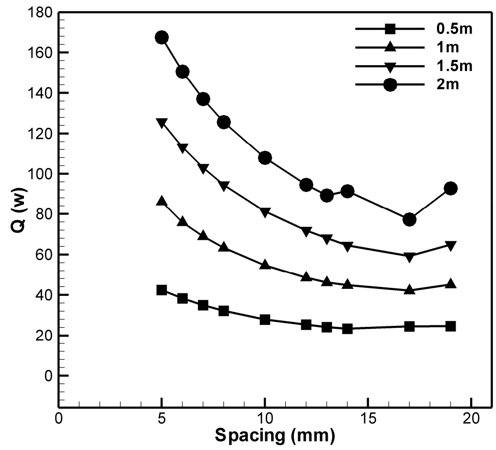

Fig.5 Heat transfer versus spacing magnitude for different window's heights

To investigate effects of filling gas and frame material properties on heat transfer, one of the cases, i.e. 1.5m height and 17mm spacing, is solved for an Aluminum frame and Argon filling gas separately. Result shows a 1.7% of increase in overall heat transfer for Aluminum frame in comparison with UPVC frame. In the other hand we had the noticeable amount of 25% of reduction in overall heat transfer for spacing filled with Argon in comparison with Air-filled spacing. Other than showing a suitable resistance against convection, because of having radiation-absorbing properties, Argon and other similar gases are convenient choices for double glass windows as filling gas [6].

Figure 6 shows the local heat transfer along the right side glass for maximum and minimum heat transfer, in the spacing. As it can be seen, heat transfer reduces as we move upper and upper. It can be reason of the effect of thermal stratification [7].

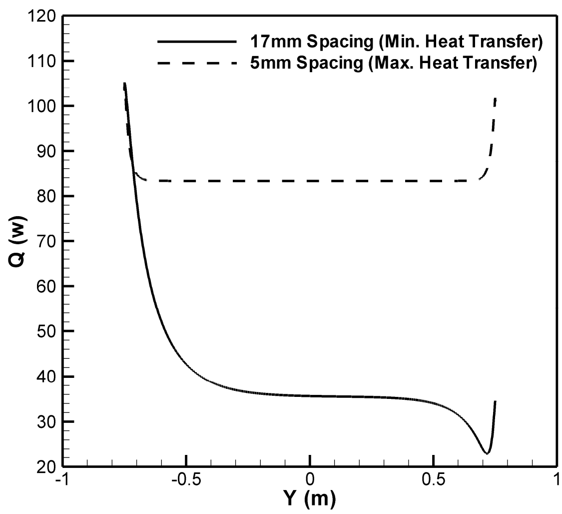

Fig.6 Local heat transfer along the right side glass in the spacing

To investigate the effect of thermal stratification, by noticing the figure 7, it can be seen that as we move upward along the glass, the slope of temperature curves decrease and the temperature field moves toward stratification. Because of the effect of thermal stratification, we have a reduction on heat transfer rate. This matter can also be considered in term of velocity. Figure 8 shows the y-direction velocity profiles in three different heights in the spacing. As it can be seen, the velocity magnitude is much less in the top of the cavity in comparison with middle. The decrease in velocity magnitude causes the decrease in Nusselt number and consequently, the heat transfer. But at the bottom of the cavity as it can be seen in figure 8, the thickness of boundary layer is small yet and because of that, we have a large amount of heat transfer in comparison with middle and top.

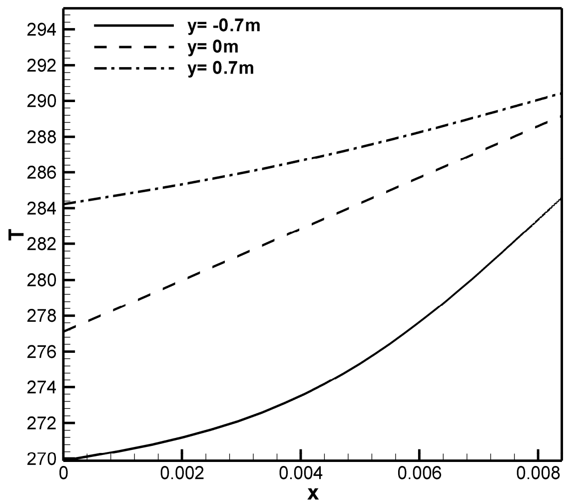

Fig.7 Temperature field in three different heights of the glass spacing

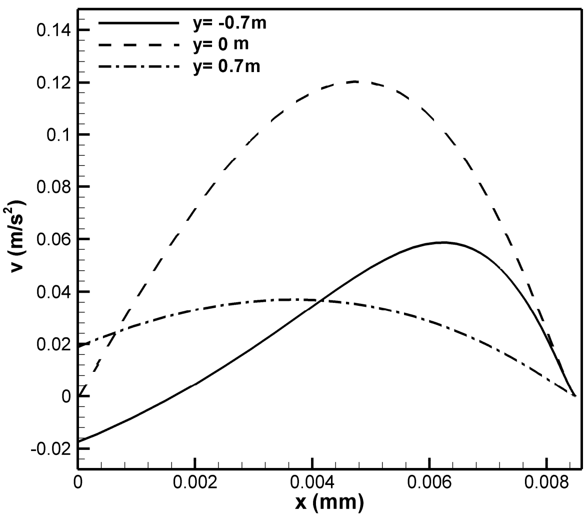

Fig.8 Magnitude of velocity in three different heights of the glass spacing

5 Conclusion

A two-dimensional steady state finite volume numerical method was developed for natural convection and conduction heat transfer in a double glass window, in order to achieve the best glass spacing for different window's height. Both convection and conduction mechanisms of heat transfer was taken into consideration. Results show a great amount of reduction in heat transfer in comparison with single glass. The best glass spacing was calculated for four different window's height, to minimize heat loss.

References:

[1] H. Manz, Numerical simulation of heat transfer by natural convection in cavities of facade elements, Energy and Buildings 35 (2003) 305–311.

[2] J. Wright, A correlation to quantify convective heat transfer between vertical window glazings, ASHRAE Trans. 106 (1996) 940–946.

[3] S. Yin, T. Wung, K. Chen, Natural convection in an air layer enclosed within rectangular cavities, Int. J. Heat Mass Transfer 21 (1978) 307–315.

[4] EM 673, Glass in building: determination of thermal transmittance (U value), calculation method, European Committee for Standardization, Brussels, 1997.

[5] S.V. Patankar, Numerical Heat Transfer and Fluid Flow, Hemisphere Publishing Corporation, Washington, 1980.

[6] Kamal A.R. Ismail, Carlos T. Salinas, Jorge R. Henriquez, Comparison between PCM filled glass windows and absorbing gas filled windows, Energy and Buildings 40 (2008) 710–719.

[7] A. Bejan, Convection Heat Transfer, Wiley Interscience Publication, New York, 1984.