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Effect of Window Location and Surface Absorptivity on Temperature inside an Enclosure-Experimental and Numerical Study

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Abstract

This paper presents the effect of outlet window location and surface absorptivity inside an enclosure on temperature, external surfaces subjected to variable heat flux boundary condition. For analysis, scaled down size of a typical room has been considered, the enclosure has inner dimension 50cm x 40cm x 30cm (LxBxH), with the longer side oriented along east-west direction. The walls and roof of the enclosure are made of 10mm thick asbestos sheet. A door of 10 cm height from the floor is considered as inlet. Experiments were conducted for outlet window of 10 cm height located at 10cm, 15 cm, 20 c measured from the floor. The external surfaces roof, east wall and west wals were heated using electrical heating coil strips. For each window configuration, temperatures of the air inside the room were recorded using data acquisition system at fifteen locations for every five minutes. From the results, it was observed that for higher surface absorptivity, lower temperature index was observed when outlet window is at mid height of the west wall. It was also observed that as the surface absorptivity at the external surfaces decreases, window located at 10 cm from the floor provides lower temperature index. Numerical simulations conducted showed lesser deviation from the experimental values.

Keywords: Passive cooling, variable heat flux, outlet window location.

Introduction

In India, major consumers of the energy are space conditioning of buildings (35%), transportation (25%) and agriculture and industry (40%)[1]. Humans in an enclosure feel uncomfortable because of the heat which is being generated by the body due to metabolism and external heat entering the enclosure. Uncomfortable environment also occurs because of the heat generated by equipments like refrigerator, electronic gadgets, fans and others. Human body feels comfortable when it is in thermal equilibrium with the surroundings. In HVAC systems, DBT of 25^0C, 50% relative humidity and 0.1 – 0.2 m/s air velocity are taken as design conditions for maximum comfort[2]. Heating or cooling systems for human occupancy are designed keeping the above conditions into account. This thermal equilibrium or thermal comfort can be obtained by many methods which either uses active elements or passive elements. In naturally ventilated enclosure, air moves as a result of pressure differences generated by wind or buoyancy or both.

Many researchers have widely investigated the natural ventilation aspects in buildings. Few investigators considered natural ventilation due to buoyancy only, few others considered only wind forces and few more considered both wind and buoyancy forces for study.

Dubovsky et al[3] conducted experiments to study the feasibility of natural ventilation on a small sized box, with external walls being insulated, to study the feasibility of natural ventilation. Flow was induced by using a hot plate which was maintained at constant temperature by using two heater coils dipped in a water tank. Various inlet openings of constant gap but different locations were used as variable parameter. It was concluded that natural ventilation by downward facing plate was feasible. In addition to experimental work, they also carried out numerical simulations. Ziskind et al.[4] has reported experimental and numerical investigations on small scale models and then on real size detached one story building. Various inlet opening location and exit location for heated plate were discussed, for steady and transient conditions. The flow of air was induced by a hot element of the building heated by solar energy. It was shown that the exit opening should be located near the ceiling in order to obtain uniform distribution of temperature inside the building. Khedari et al.[5] investigated the impact of solar chimney on indoor temperature fluctuation and air change in a school building. They used southern wall chimney of 2 m^2 and two solar chimneys on roof of 1.5 m^2 each, made of common construction materials, for a room of 25 m^3 in volume. For this condition, they reported an air change rate of 8-15 volumes/hr. Duration of the experiment was limited from 9 am to 5 pm. Effect of openings such as door and windows were also considered.

Prianto and Depecker[6] reported the effect of opening design, balcony configuration and internal division on inducing air speed inside the building. Sinha et al.[2] studied numerical analysis of temperature distribution inside an enclosure, for various inlet and exit locations under steady state conditions and for various values of Grashoff number and Reynolds number. Walls were maintained at constant temperature for analysis. From the analysis, it was concluded that depending upon the inlet and outlet locations, primary hot air and stratified air portions changes. Raman et al.[7] designed passive solar systems for thermal comfort conditioning of buildings in composite climate. Solar chimneys were used to supply hot and cold air for ventilation during winter and summer season respectively, while solar heat flux entering the building was reduced by the use of cavity in the wall and insulation. Jiang et al.[8] conducted measurement in a wind tunnel and numerical simulation with large eddy simulation on a model of dimension 250 mm x 250 mm x 250 mm for natural ventilation without considering solar heat flux. Single-sided windward, single-sided leeward and cross ventilations were considered for study. Detailed airflow filed such as mean and fluctuating velocity and pressure distribution inside and around building like models were measured by wind tunnel tests and compared with Large Scale Eddy model results for model validation. From these studies, good agreements between most of the measured and numerical results were obtained. Vedavyasa et al.[9] conducted experimental study on effect of window location with and without ventilator on temperature inside an enclosure. When the exit window was located at mid-height of the west wall, temperature inside the enclosure was found lower compared to other configurations.

From the literature, it is observed that few investigators considered only buoyancy driven natural ventilation, in which inlet opening location and exit location[3,4] as parameter with insulated/isothermal boundary conditions. Few researchers[7] considered natural ventilation under wind forces only. Few researchers[5, 6,8] considered both wind driven and thermal buoyancy for natural ventilation. Few researchers considered ventilation with single sided windward and leeward configurations. Even though[9] studied the effect of outlet window location with and without ventilator, depth of ventilator and surface absorptivity were maintained constant.

Most of the constructed buildings/enclosures for experimental purposes are located in undisturbed location. In such cases, wind forces can be considered. In actual cases, most of the houses are located adjacent to other houses and orientation along east-west direction is preferred. This means, solar heat flux falling on north and south walls can be neglected. Considering one story building/enclosure, wind velocity can generally be neglected at the ground level. Paints used on the external walls and roof have different absorptivity. Windows are essential part of any residential building/enclosure used for human beings living and are economical. Therefore, in the present study, effort has been made to study the effect of window location and ventilator, located in the west wall, for fixed inlet window height and location, for different values of absorptivity, on temperature inside an enclosure. Further, actual solar heat flux data are considered instead of all isothermal/insulated boundary condition. North and south walls were assumed to be insulated. For analysis purpose scaled down model having dimension of 50cm x 40c m x 30cm height is considered.

Material and Method

Experimental set-up: The sectional view of the experimental set up is shown in figure-1

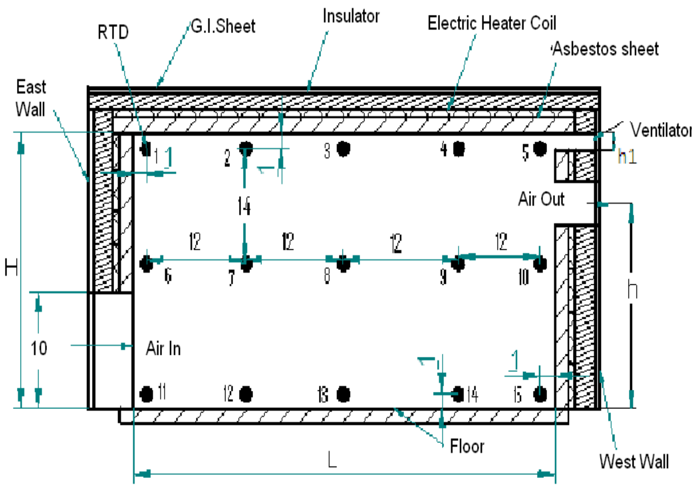


Figure-1
Sectional view of the experimental set-up (Dimensions: cm)

The experimental setup consists of rectangular box of inner dimension 50cm x 40cm x 30cm. The inner dimension selected is the scaled down model of the room configuration used in[7]. The walls of the enclosure are made of asbestos of 10 mm thickness. The external surfaces of north and south faces are insulated by using glass wool. Three electric strips are used for heating the external surface and four strips are used for heating the east and west external surfaces. On the external surfaces of the heater coils, 25mm thickness glass wool is placed. In order to facilitate or the accurate measurement of temperature, thin galvanized iron sheets are used for all configurations.

Temperature measurements: Temperature measurements were conducted inside the enclosure by using calibrated RTD sensor. The ambient air temperature and air temperature inside the enclosure were recorded for every five minutes using data acquisition system. The RTD sensors were simultaneously exposed to ambient for about 20 minutes to verify the readings.

Least count of the temperature indicator is 0.1^{0}C. It was observed that the maximum difference between the RTD sensors was found to be 0.4^{0}C. Maximum and minimum temperature uncertainity calculated is found to be 2.8% and 0.78% respectively. Configurations and variable parameters used for the experimental study are given in the table-1

Table-1
Configurations used for experimental study

Configuration	H (cm)	H (cm)	h_1 (cm)	Non-dimensional opening height (h*)	Surface Absorptivity (α)
Case A	30	20	1.0	0.67	0.6, 0.4, 0.2
CASE B	30	15	1.0	0.67	0.6, 0.4, 0.2
Case C	30	10	1.0	0.50	0.6, 0.4, 0.2

Table 1 shows the parameters considered for experimental study. Height of the enclosure 'H' maintained constant and is equal to 30 cm. Location of the outlet window is varied from 15 cm to 20 cm and is measured from the floor of the enclosure. Different values of the surface absorptivitiy represent the surface absorptivity of the color used on the external surfaces of the enclosure. For simplicity, it is assumed same on all the external surfaces.

Input Data: For the present study, solar radiation heat flux on horizontal surface, for the city of Bangalore, in the month of May[10] is taken as input. In order to get intensity of radiation on east and west wall surfaces, the following correlation is used. If θ is the angle between an incident beam of flux I_{bn} and the normal to a plane surface, then the equivalent flux falling normal to the surface is given by I_{bn} * Cos (θ)[11].

The angle θ is related by a general equation to Φ the latitude, δ the declination, γ the surface azimuth angle, ω the hour angle, and β the slope and is given below.

Cos (θ) = sin (Φ) (sin (δ) cos (β) + cos (δ) cos (γ) cos (ω) sin (β)) + cos (Φ) (cos (δ) cos (ω) cos (β) –
Sin (δ) cos (γ) sin (β)) + cos (δ) sin (γ) sin (ω) sin (β) (1)
For east and west surfaces $\beta = 90^0$,
For horizontal surface $\beta = 0^0$.
Total radiation reaching a surface is given by[11]
$I_g = I_b + I_d$ (2)
As the breakup of diffuse and beam component for Bangalore city is not available, the same available, the ratio of I_b and I_d available for Delhi[12] has been assumed to be hold good for Bangalore city also.
Voltage input to the coils is calculated by using the below expression.
$V = \sqrt{(W/R)}$ (3)
The variation of average resistance of the heater coils used for heating the roof external surface is +/-3Ω and that of the west surfaces is +/-2Ω.

Experimental Procedure: The experimental procedure is given below. Initially, the internal air, surface wall and roof temperatures were measured. The heating coils provided on the east wall, west wall and roof were given required power supply which is maintained at constant value for one hour, using auto-transformer. Even though solar flux variation is sinusoidal, here between two consecutive hours, it is assumed as linear. The average heat flux between the two consecutive hours is given as heat flux and is assumed to remain same for that hour. The heat input was changed after every hour based on the solar flux calculations and subsequently the power was switched off. Temperatures were recorded every five minutes. The same procedure was followed for the next few days. When the difference between air temperature of any channel, at any time, for the present and previous day is less than 1^0C, it is assumed that the system has reached a steady state. The experiment is continued for that day and at the end of the day, experiment was stopped. Error analysis for the RTD sensors have been carried out and the maximum error is found to be 0.7^0C.

Numerical Study: Numerical study has been carried out for most of the configurations studied when the system has attained steady state. The basic conservation equations were solved numerically by using FLUENT 13.0/14.0 CFD software. The form of the equations is given below.

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i u_j)}{\partial xi} = 0$$

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i u_{ij})}{\partial xi} = -\frac{\partial P}{\partial xi} + \frac{\partial \tau}{\partial xj} + \rho gi$$

$$\frac{\partial(\rho h)}{\partial t} + \frac{\partial(\rho u_i u_{ij})}{\partial xi} = -\frac{\partial P}{\partial xi} + \frac{\partial \tau}{\partial xj} + \rho gi$$

where **ρ** is the density, **u** is the velocity component in i direction, **p** is the static pressure, **t** is the time, $\mathbf{u_i}$ is the Cartesian co-ordinate in x direction, $\boldsymbol{\tau_{ij}}$ is the stress tensor, $\mathbf{g_i}$ is the gravitational acceleration in i-direction, h is the static enthalpy, **k** is the thermal conductivity and **T** is the temperature. Large viscous stresses and compressibility effects are not present in this problem; the viscous term was not activated. For buoyancy term, either the temperature-density relation can be given or boussenesq approximation can be activated. Here, Bousenesque approximation is activated.

Boundary Conditions: i. Boundary conditions for the wall are no penetration, no slip condition. ii. A small strip of 0.5 cm galvanized iron was considered to be acting as heat generating element equivalent to solar heat flux. When open state is considered, the temperature, the temperature of the surrounding is imposed at the inlet opening. At the exit opening, measured temperature of the air or approximate value is given. With this, FLUENT adjusts the boundary condition, by extrapolating the values from interior grid cells adjacent to the exit. iii. Floor of the enclosure is treated as insulated boundary. iv. At the inlet and outlet, an extended boundary is taken to make inlet and outlet as atmospheric pressure condition. v. A thin strip of galvanized iron sheet of thickness 0.5 cm is assumed to generate heat per unit volume equal to the heat flux at that time. Amount of heat entering the enclosure is calculated based on the resistance above and below the heater coil strip.

Results of comparison of the experimental and numerical simulations are expressed interms of percentage of deviation. It is defined as the ratio of difference in the value of experimental and numerical simulation to that of experimental value. Percentage of deviation at each measured location is given in tables from x to y. Percentage of deviation interms of average value at each layer, average of all the sensors and average below the living height is also given in table z for three locations and for CASE B at absorptivity values of 0.2, 0.4 and 0.6.

Figure-5 shows the mesh pattern used for numerical simulation. For every 2 mm, elements are taken and biased with a value of 1.03. This value is sufficient to capture boundary layer.

Results and Discussion

Experimental Results: Experimental study on reduction of temperature inside an enclosure is carried out for five configurations, namely Case A, CASE B and Case C as shown in table-1. The performance of the outlet window location is expressed by the relative index known as temperature index (dT)[5]. This is also necessary as the experiments were conducted on different days. Temperature index is defined as the temperature difference between average room temperature below the living height and ambient. Living height is taken as 1.6 m above the floor. This index gives the value by which the average room temperature below the hiving height is above the ambient. Temperature Index below the living height is the difference between the temperatures of average of thermocouples numbered from 6 to 15 and the ambient temperature. Lower value of temperature index is preferred. Since the experiments were conducted in laboratory, with laboratory windows in closed position, the flow is by buoyancy. Because of this the velocity was very negligible and hence not measured.

Figure 2 shows the variation of temperature index for surface absorptivity of 0.2, when the outlet window is at different location. From the figure, it is observed that the temperature index for the Case A varies from a value of -0.3^{0}C at 7 am to a value of 3.4^{0}C at 4 pm. For the CASE B, temperature index varies from value of -0.7^{0}C at 8 am and to a value of 5.1^{0}C at 2 pm. For the CASE B, temperature index varies from value of -0.7^{0}C at 8 am and to a value of 5.1^{0}C at 2 pm. For Case C, temperature index varies from value of -1.1^{0}C at 6 am and to a value of 3.6^{0}C at 2 pm. It is also observed that for the Case C, the temperature index is lower compared to Case A and CASE B almost at all the time. Temperature Index of Case A is lower compared to CASE B upto 11 pm and afterwards, CASE B has lower temperature index till 6 am.

Figure- 3 shows variation of temperature index for surface absorptivity of 0.4, when the outlet window is at different location. From the figure, it is also observed that temperature index for the Case A varies from a value of -2.1^{0}C at 6 am to value of 5.5 ^{0}C at 3 pm. For the CASE B, temperature index varies from value of -0.6^{0}C at 6 am and to value of 6.7^{0}C at 3 pm. For the CASE B, temperature index varies from value of -0.9^{0}C at 5 am and to value of 7.2^{0}C at 3 pm. It is also observed that the temperature index for the Case A is lower compared to CASE B and Case C, at almost all times. Temperature index variation is negligible between CASE B and CASE B from 5 pm to 10 pm. After 10 pm, the temperature index for the Case C is lower. From 6 am to 4 pm, there is a fluctuation between the CASE B and Case C.

Figure-4 shows the variation of temperature index for surface absorptivity of 0.6, when the outlet window is at different location. From the figure, it is observed that the temperature index for the Case A varies from a value of 0.3^{0}C at 6 am to value of 9.3 ^{0}C at 2 pm. For the CASE B, temperature index varies from value of 0.6^{0}C at 6 am and to value of 9.3^{0}C at 2 pm. For the CASE B, temperature index varies from value of -0.7^{0}C at 5 am and to value of 10.6^{0}C at 2 pm. It is also observed that for Case C, temperature index is lower from 6 am to 10 am compared to Case A and CASE B, again from 9 pm to 6 am. When compared between Case A and CASE B, temperature index is lower for Case A between 6 am to 10 am. From 11 am to 8 pm, almost remains same till 3 am. After that the temperature index is lower for CASE B compared to Case A till 6 am.

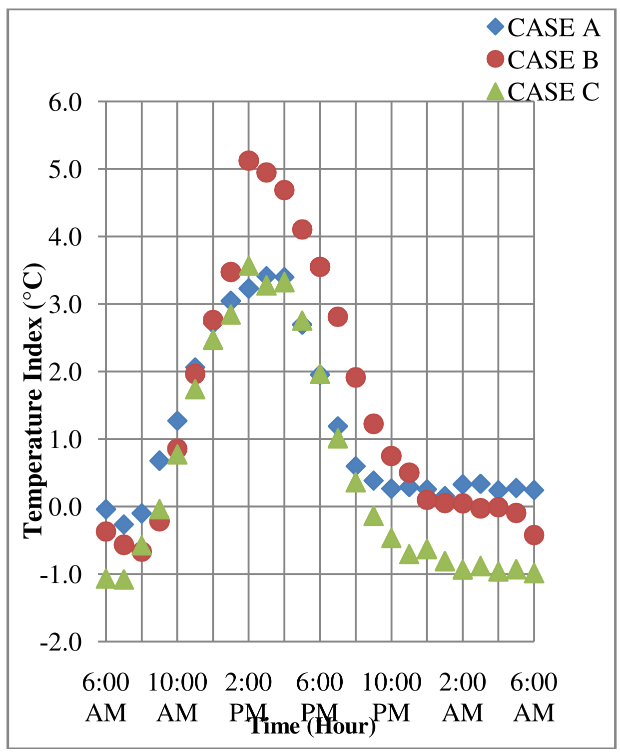


Figure-2

Effect of exit/ourtlet window location on temperature index for surface absoprtivity of 0.2

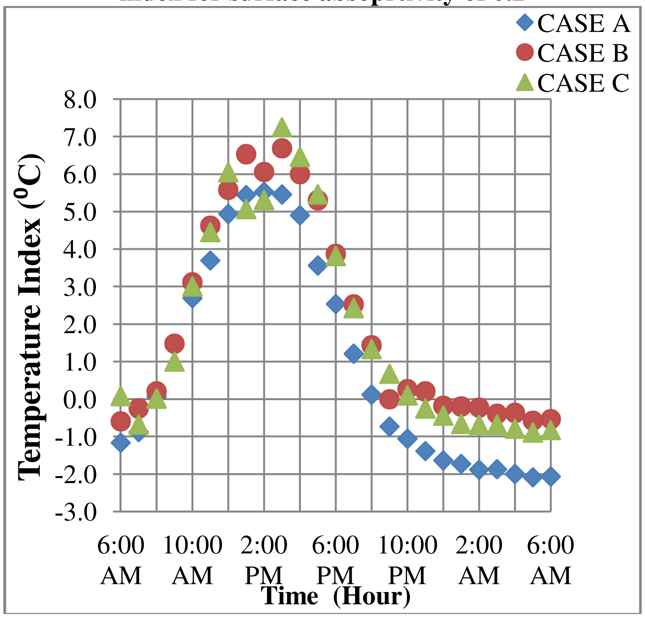


Figure-3

Effect of exit/outlet window location on temperature index for surface absoprtivity of 0.4

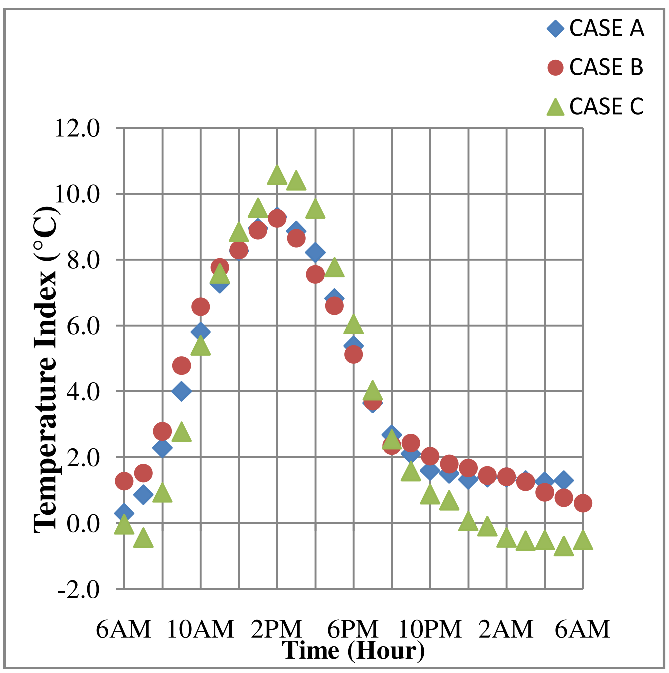


Figure-4

Effect of exit/outlet window location on temperature index for surface absoprtivity of 0.6

From the above graphs, it is observed that for the given outlet window location h*, as the solar heat flux varies from zero to maximum and again to zero, the temperature index also increases and then decreases, the magnitude of temperature index differes depending upon the exit window location. When the surface absorptivity at the surfaces decreases, the temperature index also decreases. When configurations at different outlet opening locations are compared at same absorptivity, it is observed that temperature index is higher for higher absorptivity and lower for lower surface absorptivity. It is also observed that, as the surface absorptivity increases, the temperature index for the CASE B decreases. The change in the tempeature inside an ecnclosure depends upon the magnitude of cold air entering and hot air leaving. Again as the location of the outlet window changes, the amount of hot air leaving and cold air entering also changes. This inturn depends upon the buoyancy effect inside the enclosure. As the buoyancy effect increases due to increase in the surface absorptivity, when outlet window is located at mid-height (CASE B) of the west wall, this may provide better escape of hot air and entry of cold air into the enclosure and hence provides lower temperature index.

Numerical Results: For two dimensional numerical studies, results are expressed in terms of percentage of deviation. It is defined as the difference in experimental value and numerical value to that of experimental value. Fig.6. (a) and (b) shows the two-dimensional transient temperature field and velocity vector field at 7 am of the first day of the experiment for the CASE B at 7 am. It is observed that hot air is accumulated near the roof of the enclosure. It is because higher heat flows from the roof of the enclosure and stratification takes place in this region. Experimental results also show higher temperature near the roof. Table shows average percentage of deviation. More or less equal percentage of deviation (-2.7%, -2.6% and -2.6%) is observed as average of all the sensors, average below living height and average at top layer respectively. At living height and bottom layer, average percentage of deviation is observed to be -3.2%. Minus sign indicates that the numerical simulation values are higher than the experimental values. Highest percentage of deviation of -4.8% is observed at location number 7.

Figure-7 (a) and (b) shows the two-dimensional transient temperature field and velocity vector field at 7 am of the first day of the experiment for the CASE B at 7 am. It is observed that hot air is accumulated near the roof of the enclosure. It is because higher heat flows from the roof of the enclosure and stratification takes place in this region. Experimental results also show higher temperature near the roof. Maximum percentage of deviation 9.2 is observed at channel number 7.

Figure-8 (a) and (b) shows the two-dimensional transient temperature and velocity vector field at 12 noon of the first day of the experiment for the CASE B. It is observed that hot air is accumulated near the roof of the enclosure. It is because higher heat flows from the roof of the enclosure and stratification takes place in this region. Percentage of deviation of channels from 5 to 12 is considerably higher. At other channels, the variation is within the acceptable level. Due to space limitations, percentage of deviation for each value of absotptivity is not shown.

Average percentage of deviation for different values of absorptivities for the CASE B is shown in table-2.

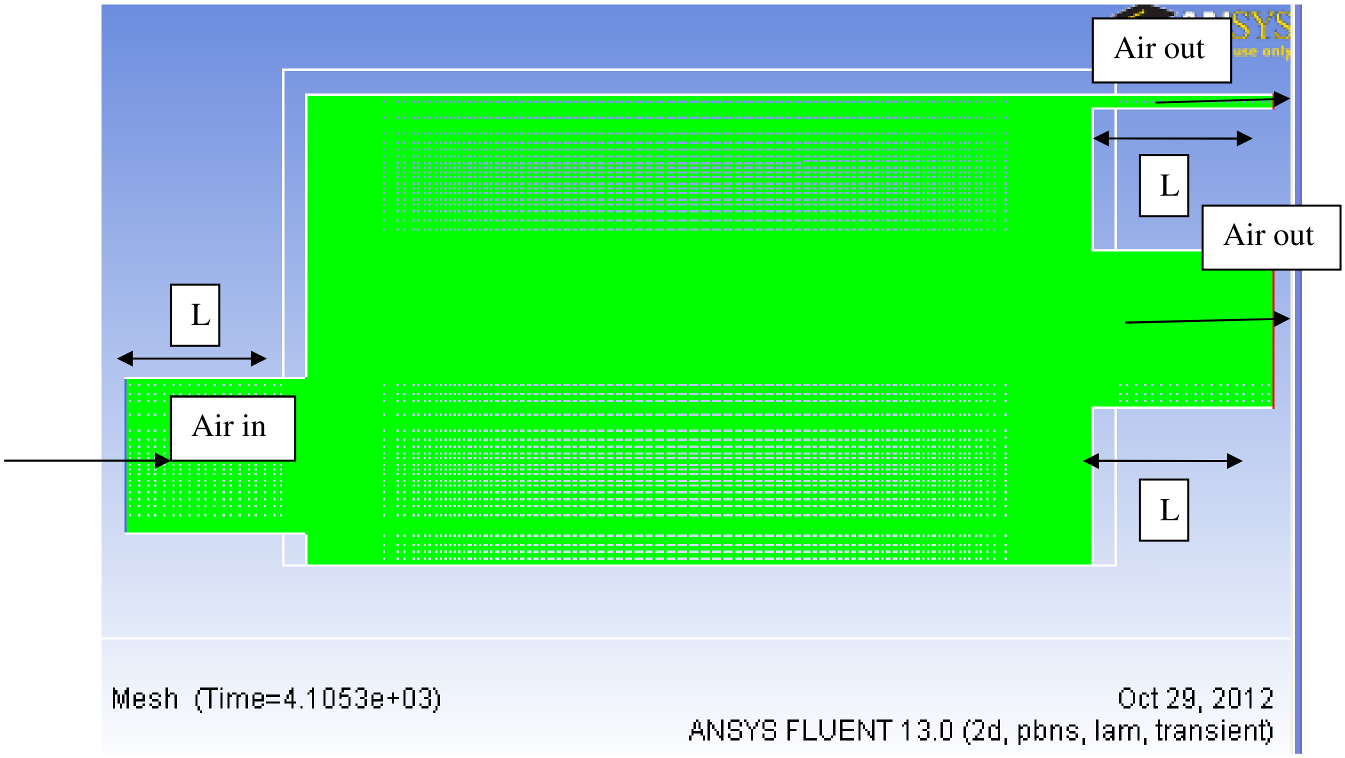
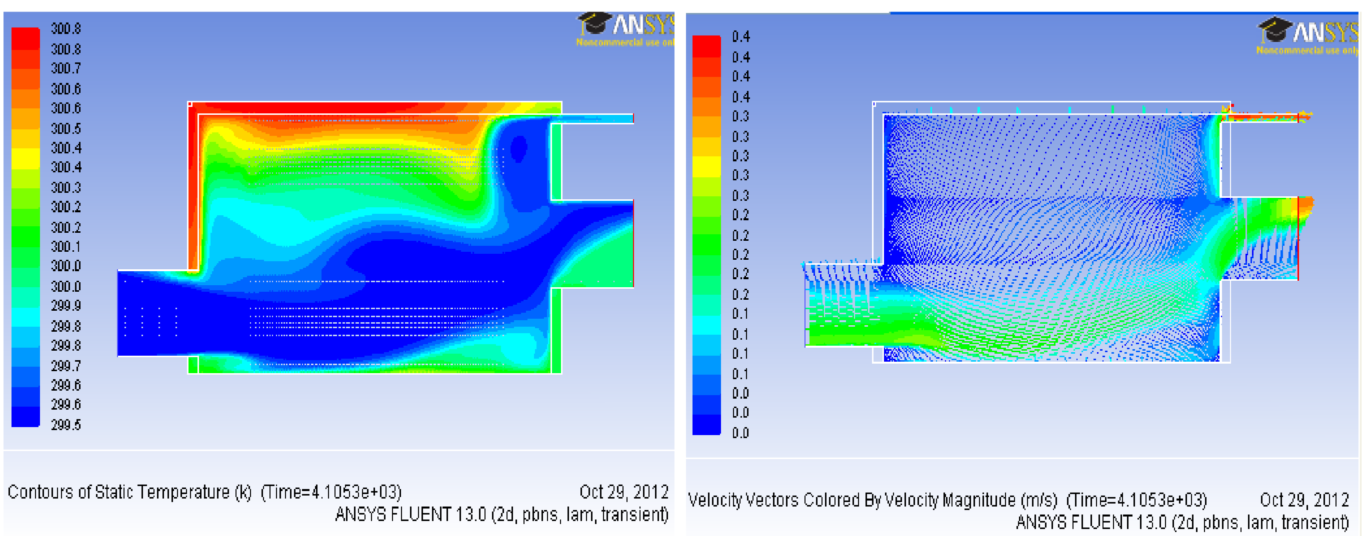
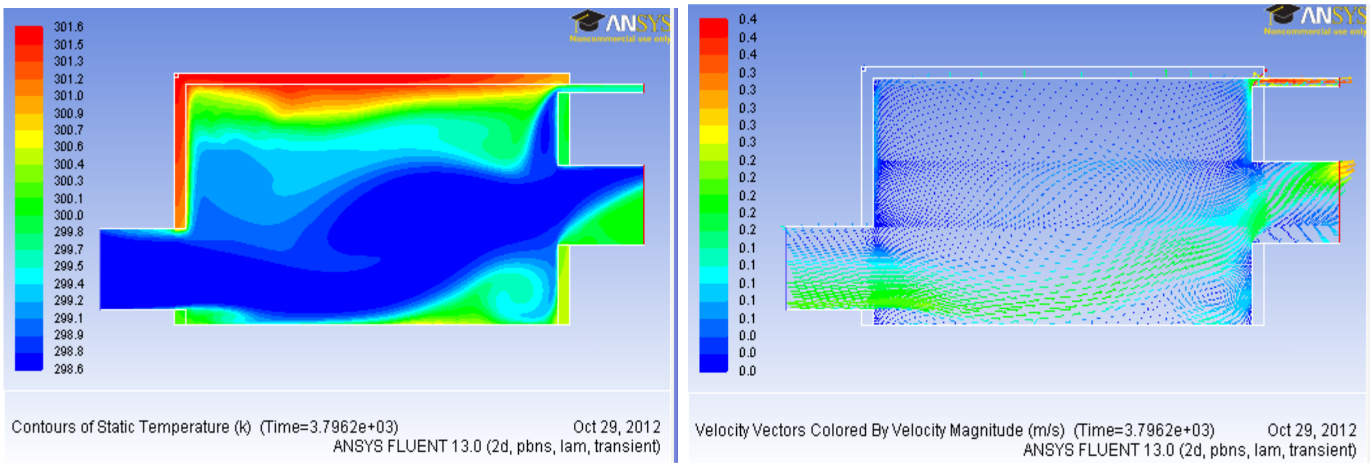


Figure-5
Mesh configuration for CASE B



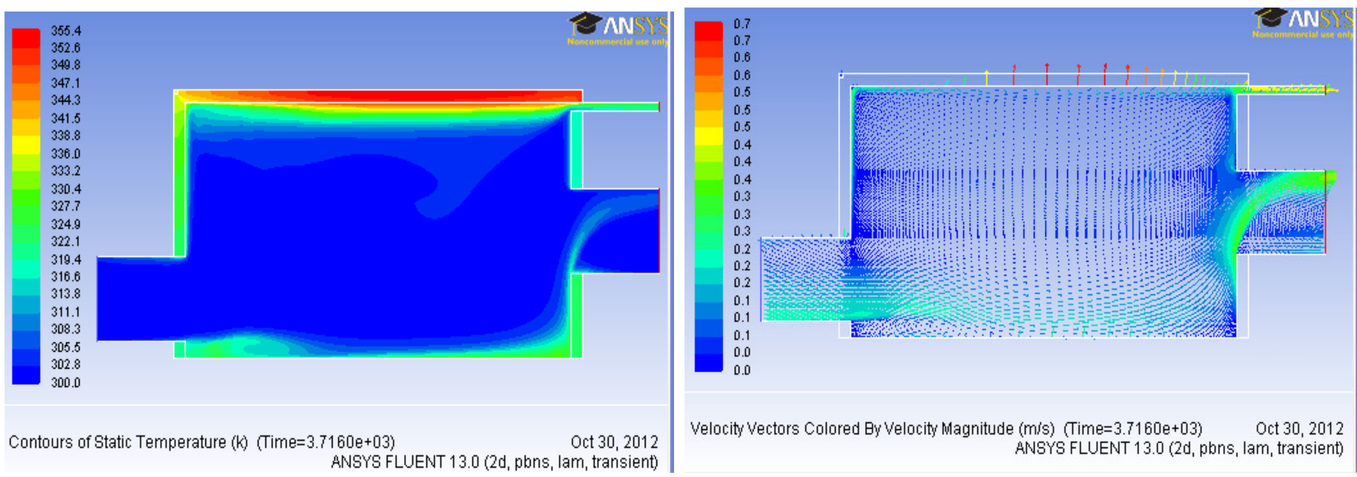
(a) (b)

Figure-6
Transient ventilation of the enclosure, for the case 10151at surface absorptivity of 0.2:
(a) 2-D simulated temperature field; (b) 2-D simulated vector field



(a) (b)

Figure-7
Transient ventilation of the enclosure, for the case 10151at surface absorptivity of 0.4:
(a) 2-D simulated temperature field; (b) 2-D simulated vector field



(b) (b)

Figure-8
Transient ventilation of the enclosure, for the case 10151at surface absorptivity of 0.6:
(a) 2-D simulated temperature field; (b) 2-D simulated vector field

Table-2
Percentage of deviation for CASE B at various values of absorptivities

% of Deviation	**CASE B**		
	α = 0.2	**α = 0.6**	**α = 0.6**
Average of all channels	-2.7	-0.9	4.7
Average below Living (middle) height channels	-2.6	-4.5	4.9
Average of top layer channels	-2.6	-4.5	4.4
Average at living (middle) height	-3.2	-1.9	18.2
Average at bottom layer channels	-3.2	-1.9	-17.8

Conclusion

Experiments were conducted for three configurations of surface absorptivities of 0.2, 0.4 and 0.6, continuously till steady state conditions were established, to study the ability of openings location to change temperature inside an enclosure. From the experimental results, it was observed that, at any given value of surface absorptivity, the temperature index for window located at any location, increases from 6 am, reaches maximum at 2 pm or 3 pm and again decreases. As the surface absorptivity decreases, Case C provides lower temperature index. At surface absorptivity of 0.4, Case A provided lower temperature index. Looking at this tendency, it can be said that, at higher absorptivity, CASE B provides lower temperature compared to Case A and Case C. From the numerical study, it was observed that the deviation of simulated results do not vary much. Hence numerical simulations can be used. This kind of study using passive methods for reducing temperature inside an enclosure, reduces the use of active elements such as fan, air cooler, etc, and hence reduces environmental pollution.

Nomenclature

θ	angle between an incident beam and normal to a plane surface,
Φ	latitude of a location (degree),
δ	declination (degree),
ω	hour angle, (deg/hour)
β	angle made by the plane surface with the horizontal (degree),
γ	surface azimuth angle (degree),
h	height of the opening (cm),
H	height of the wall (cm),
h*	non-dimensional opening height = h/H,
H*	non-dimensional opening centre height = h /H,
h_1^*	non-dimensional ventilator height = h_1/H,
I_g	hourly global radiation (W/m^2)
I_b	hourly beam radiation (W/m^2),
I_d	hourly diffuse radiation (W/m^2)
W	heat input (watt)
V	supply voltage (volts)
R	average coil resistance (ohms)
T	time (hour)
dT	temperature index (^{0}C)
α	surface absorptivity

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