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Natural convection in a room with two opposite heated vertical walls

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Abstract
In this study, investigation of radiation and natural convection in cubic enclosure has been carried out. A model of an enclosure representing a room was constructed from polystyrene boards. Two vertical walls are supplied with constant heat flux in the range of 9.4-47.8 W/m^2. Temperatures of walls, ceiling, floor and air inside enclosure were measured using a 26 K-type thermocouples under steady state condition. Heat transfer was investigated for Rayleigh numbers in the range $4.4x10^7 \leq Ra \leq 1.2x10^8$ with Prandtl number of 0.71. Detailed results including temperature profiles and correlation equations for convection heat transfer coefficient in terms of temperature difference between the heated surface temperature and the temperature of the air have been obtained for the walls of the enclosure.

Keywords: Natural convection; Room; Heat transfer; Heated wall.

1. Introduction
Natural convection as one of heat transfer mechanisms is encountered in many areas of human activity, for example, in electronics for optimization of cooling systems, in chemical industry for improvement of manufacturing methods of bulk semiconductor monocrystals, in space technologies for creation of reliable cooling system for airborne electronics and heat pipes, and in mechanical engineering for designing of nuclear reactors operator bodies [1]. Numerous studies have been conducted in the past, and most of them are concentrated in rectangular cavities because it represents one of the simplest geometries with many applications in industry.
Khalifa [2] investigated convective heat transfer over the interior surfaces of a real-sized test cell. The study was conducted under controlled steady-state conditions to cover nine of the most widely used heating configurations in buildings. Rahimi and Sabernaeemi [3] studied radiation and free convection in the heat transferred from the ceiling surface of a room to other internal surfaces. They also studied free convection and radiation in the heat transferred from the heated floor of a room to the other internal surfaces [4]. Henekes et al. [5] studied numerically the laminar and turbulent natural convection flow in a two dimensional square cavity using three different turbulence models. Balaji and Venkateshan [6] numerically investigated the interaction of surface radiation with laminar free convection in a square cavity. They elucidated the importance of surface radiation even at low emissivity and provide some reasons for the discrepancies noted between the experimental and theoretical correlations. They also derived correlation equations to calculate convection and radiation Nusselt numbers in square enclosures [7]. Shati [8] studied numerically, theoretically and experimentally the effects of natural convection with and without the interaction of surface radiation in square and rectangular enclosures. The analyses were

carried out over a wide range of enclosure aspect ratios. Experimental and numerical study was performed by Salat et al. [9] on turbulent natural convection in a cavity. Two guard cavities were used to ensure adiabatic conditions for the test cavity. Sanders [10] studied convective heat transfer in buildings by using a removable panel scheme to study different geometries. The film coefficient was compared to film coefficients determined from published natural convection correlations. Cholewa et al. [11] presented the results of experimental research on heated/cooled radiant floor conducted in the laboratory room in the climatic chamber to estimate the heat transfer coefficients for the surface of cooled/heated radiant floor. Edward et al. [14] studied natural convection in a three dimensional rectangular enclosure in the form of a room with heaters placed on opposite walls and two windows each on the adjacent opposite walls. Rachid et al. [15] studied numerically natural convection and surface radiation within a square cavity filled with air and submitted to discrete heating and cooling from all its walls. Two heating modes were considered: in the first mode, the cavity was heated discreetly on the left vertical and bottom horizontal walls, while in the second mode, the left vertical and top horizontal walls of the cavity were heated.

Present study will focus on the radiation and free convection in enclosed space (enclosure). The cubic enclosure has been constructed from polystyrene boards and covered with glass wool insulation. Two electrical heaters were placed in enclosure of interior dimensions 60 x 60 x 60 cm and walls thickness of 9.5 cm. The heaters simulated two vertical heated walls. The enclosure was placed inside a large steady environment.

2. Experimental work and test rig description

A model enclosure represent a room was constructed from polystyrene boards with interior dimensions of 60cm x 60cm x 60cm and wall thickness of 9.5 cm. Two vertical walls are supplied with constant heat flux of 9.4-47.8 W/m^2 using heaters to simulate the heating effect which may result, for example, from a sun patch striking the wall. Solid-state relay (SSR) is used as an autotransformer that regulates the voltage for the heaters during the tests.

Temperatures were measured using a digital thermometer and 26 calibrated K-type thermocouples. High conductivity thermal paste (k=1.46 W/m.K) was used in the contact area between the thermocouple and the wall to minimize the thermal contact resistance. The temperatures of the heated walls were controlled to be within the range 30 to 50°C. Model was covered with glass wool Insulation. The enclosure was placed inside a large steady environment. The heating system was turned on to achieve a steady state condition before collecting the experimental data. Figure 1 shows the side view of the enclosure.

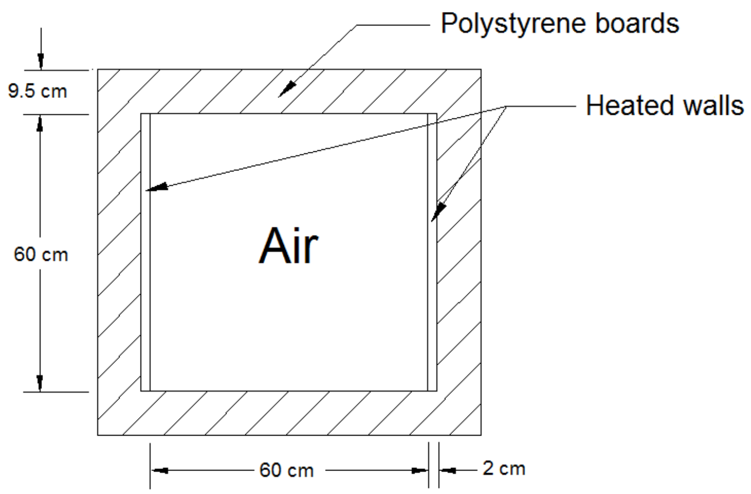

Figure 1. Schematic diagram of the enclosure

3. Methodology

The physical properties of air are calculated at the mean film temperature (average of surface and bulk temperature) [12].

$$T_f = \frac{T_s + T_b}{2} \tag{1}$$

The constant heat flux is calculated by:

$$Qf = V.I/A \tag{2}$$

The conduction heat loss is given by:

$$Qcond = A.\frac{T_s - T_{out}}{\Sigma(\frac{t}{k})} \tag{3}$$

Radiation heat transfer is calculated by:

$$Q_{rad} = A\,\sigma\,f\,\left(T_s^4 - T_{mr}^4\right) \tag{4}$$

Where f and T_{mr} are given as in Khalifa [1]:

$$f = \left[\frac{1-\varepsilon_1}{\varepsilon_1} + \frac{1}{F} + \frac{A_1}{A_2}\frac{1-\varepsilon_2}{\varepsilon_2}\right]^{-1} \tag{5}$$

$$T_{mr} = \frac{\sum_{i=2}^{n} T_i.A_i}{\sum_{i=2}^{n} A_i} \tag{6}$$

Where (n) is the number of the surfaces involved in the radiative exchange with surface 1 and (Ti) and (Ai) are the temperature and area of the individual surfaces respectively.
Total heat transfer for the heated walls is calculated by:

$$Q_T = Q_f - Q_{cond} - Q_{rad} \tag{7}$$

For cold walls the equation becomes:

$$Q_T = Q_{cond} + Q_{rad} \tag{8}$$

Heat transfer coefficient has been calculated by:

$$h = \frac{Q_T}{A.(T_s - T_{in})} \tag{9}$$

The Rayleigh number is given by:

$$Ra = Gr.Pr = \frac{g\,\beta\,(T_s - T_{in})\,L^3}{\nu^2}.Pr \tag{10}$$

4. Results and discussions

A constant heat flux of 9.4-47.8 W/m^2 was supplied to the heaters by keeping the voltage at a constant level by the Solid-State Relay (SSR). Two vertical heaters (60x60 cm) simulated the heat source on the internal surface of selected walls of the room. The experiments were performed for nine different constant heat flux values. The temperatures on the internal surfaces of the room were noticed to reach the steady state condition after around 1 hr as recorded by the digital temperature recorder.
Based on the temperature distribution on the surfaces of the heated walls, the other vertical walls, the floor and the ceiling of the enclosure, heat transfer rates by radiation, conduction and convection for all walls of the enclosure were calculated. The heat transfer coefficient (h) obtained on the walls were correlated against the temperature drop difference (ΔT).
Figure 2 shows the data and the correlations obtained for the vertical heated walls with deviation of about 4 %. The increase in the heat flux leads to increase the surface temperature of the hot wall(s) and thereby cause an increase the surface-air temperature difference (ΔT).

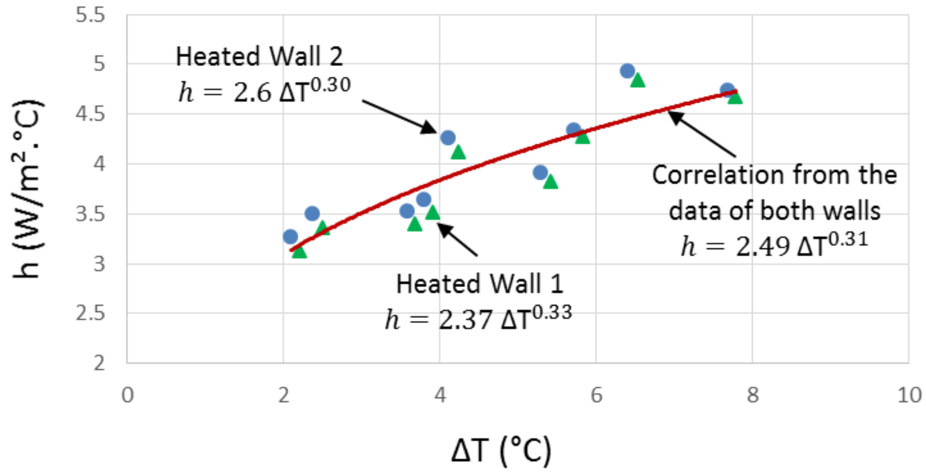

Figure 2. Correlation equations for heated walls (hot vertical walls)

The convection heat transfer coefficient (h) is very often expressed as a function of the temperature difference between the heated surface temperature (T_s) and the temperature of the air (T_{in}) [13].

$$h = C(\Delta T)^n \tag{11}$$

Where C and n are constants.
Figure 3 shows the data and the correlations obtained for the vertical cold walls with deviation of about 7%. The temperature variation on the internal surfaces is negligible from the middle part toward the floor. The temperature distribution on the external surfaces of the enclosure is affected by the surrounding conditions. The temperature difference between the cold walls temperature and that of air (ΔT) is less than that for the heated walls.

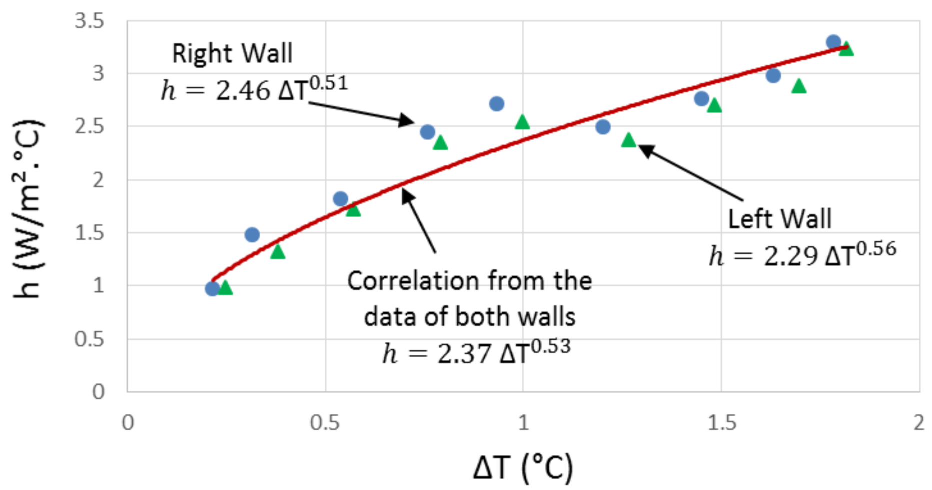

Figure 3. Correlation equations for right and left wall (cold vertical walls)

The data and the correlations obtained for the ceiling and floor are shown in Figure 4. The temperature difference between surface temperature and temperature of the air (ΔT) for floor is higher than that for ceiling.
During the experiments, it was noted that the highest value of the internal surface temperature is near the enclosure ceiling and it decreases slowly in the downward direction towards the floor.

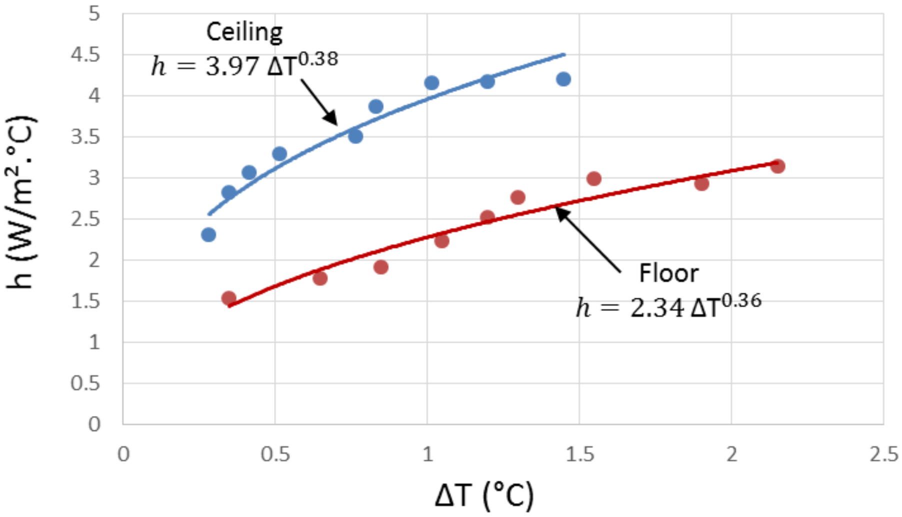

Figure 4. Correlation equations for ceiling and floor

5. Conclusions

Radiation and natural convection heat transfer in enclosure with opposite vertical heated walls configuration at different heat flux is investigated experimentally. The following conclusion may be drawn from the results:

1. The following empirical correlations for heat transfer coefficients against the surface-air temperature difference are obtained for the vertical heated walls, cold walls, ceiling and floor:
   a. For the vertical heated walls: $h = 2.49\ \Delta T^{0.31}$.
   b. For the vertical cold walls: $h = 2.37\ \Delta T^{0.53}$.
   c. For the ceiling: $h = 3.97\ \Delta T^{0.38}$.
   d. For the floor: $h = 2.34\ \Delta T^{0.36}$.
2. The experimental results were found to be in good agreement with those of previous studies.

Nomenclature

A	Area of the surface	m^2
Gr	Grashof number	----
g	Gravitational acceleration	m/s^2
h	Heat transfer coefficient	W/m².°C
k	Thermal conductivity	W/m. °C
L	Length of wall	m
Nu	Nusselt number	----
Pr	Prandtl number	----
Q_{cond}	Conduction heat transfer	W
Q_{conv}	Convection heat transfer	W
Q_f	Constant heat flux	W/m^2
Q_{rad}	Radiation heat transfer	W
Q_T	Total heat transfer	W
Ra	Rayleigh number	----
T_b	Bulk temperature of air	°C
T_f	Film temperature	°C
T_{in}	Air temperature inside enclosure	°C
T_{out}	Temperature out of the wall	°C
T_s	Surface temperature	°C
t	Thickness of walls	m
Greek symbol		
σ	Stefan–Boltzmann constant= 5.6697×10^{-8}	$W/m^2\ K^4$
β	Thermal expansion coefficient	1/K
ν	Kinematic viscosity of air	m^2/s
ε	Emissivity of the surface	----

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