An Experimental and 3d Numerical Study for Natural Convection with Radiation Inside Rectangular Enclosures

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Abstract:

The effects of turbulent natural convection with the interaction of surface radiation in a rectangular enclosure have previously been numerically and theoretically studied by the authors. This work extends these results by comparing the 3D numerical calculations with the experimental results.

In this research an experimental study was performed for a hot wall temperatures ranging from 50C° to 75C° with an increment of 5C° with almost constant cold wall temperature at about 9°C and for three enclosure

sizes of aspect ratios 2.0, 1.0 and 0.5. This allows the calculation of total heat transfer from the hot and cold walls using six thermocouples in each side. 3D numerical calculations were performed using the ANSYS 13 workbench software and for the same range of parameters of the experimental study.

The comparison between the thermal calculations of the experimental results and the 3D numerical results were within an average deviation of less than 13% for the three aspect ratios.

Keywords: Natural Convection, Radiation Interaction, Rectangular Enclosures, Heat Transfer, Turbulence, Aspect ratio

Nomenclature:

C_p	*Specific heat capacity*	$(Jkg^{-1}K^{-1})$
$\dot{m}$	*Mass flow rate*	(kgs^{-1})
$\dot{Q}_{tot}$	*Total heat transfer*	*(W)*
$\dot{Q}_{loss}$	*Heat losses*	*(W)*
T_{in}	*Inlet temperature*	*(K)*
T_{out}	*Outlet temperature*	*(K)*
T_h	*Hot temperature*	*(K)*
T_c	*Cold temperature*	*(K)*

1.Introduction:

Enclosed spaces represent one of the most popular heat transfer problems and have a practical interest in many engineering applications, such as design of buildings for thermal comfort, nuclear reactors, solar collectors, and the cooling of electronic equipment.

Many researchers have studied the importance of natural convection with radiation interaction in square and rectangular enclosures. Balaji and Venkateshan [1], 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 emissivities and provide some reasons for the discrepancies noted between the experimental and theoretical correlations. They [2] provided correlation equations to calculate convection and radiation Nusselt numbers in square enclosures.

Colomer et al. [3], looked at the three-dimensional numerical simulation of the interaction between the laminar natural convection and the radiation in a differentially heated cavity for both transparent and participating media. Their work reveals that in a transparent fluid, the radiation significantly increases the total heat flux across the enclosure. Akiyama and Chong [4], investigated the numerical analysis of natural convection with surface radiation in a square enclosure, they provided a correlation equation to calculate Nusselt number. Shati et al. [5] studied the natural convection with radiation interaction in square and rectangular enclosures. Their work focused on the effect of aspect ratio and realistic conditions (which include the effect of different enclosure size, changing the cold and hot wall temperatures, and using different fluids, for all properties varied with temperature) on the turbulent natural convection with and without surface radiation. Then they reported a correlation equation for the natural convection for both types of enclosures (square and rectangular). Their analysis used a new dimensionless group which demonstrates the relation between the convection and radiation heat transfer inside the rectangular and square enclosures. Shati et al. [6] then provided an empirical solution to turbulent natural convection heat transfer with radiation interaction in square and rectangular enclosures to calculate the total Nusselt number. The empirical solution allows the simple calculation of either the total heat transfer rate, given the fluid, the geometry and the temperatures of the hot and cold walls, or via a straightforward iterative technique, the temperature of one wall, given the other wall temperature and the total heat transfer.

In this research an experimental study was performed for square and rectangular enclosures for three aspect ratios of 2.0, 1.0 and 0.5. Tests were carried out for hot wall temperatures ranging from 50°C to 75°C. This allows the calculation of total heat transfer from the hot and cold walls using six thermocouples in each side.

Experimental setup and procedure:

During the course of the experiments many modifications were made to the cavity to reduce the heat losses through the walls and to the hot and cold side fluid systems to improve the control of the wall temperatures.

Test cavity:

The cavity (or enclosure) consists of six walls: a hot side, a cold side and four side walls. The cavity has three configurations; these three configurations allowed the testing of three different aspect ratios, and in each of these the only change in the enclosure design was be the length of the top, sides and bottom walls (the hot and cold sides were kept fixed in size). The enclosure is designed using polystyrene boards of 120mm thickness and two layers of cavity wall double insulation, one from inside and another layer from outside this is shown in figure 1. The temperature of the hot and cold side walls were controlled using PID controllers. All the six walls were smooth and the top, sides and bottom walls were smooth shiny and the hot and cold walls were smooth black.

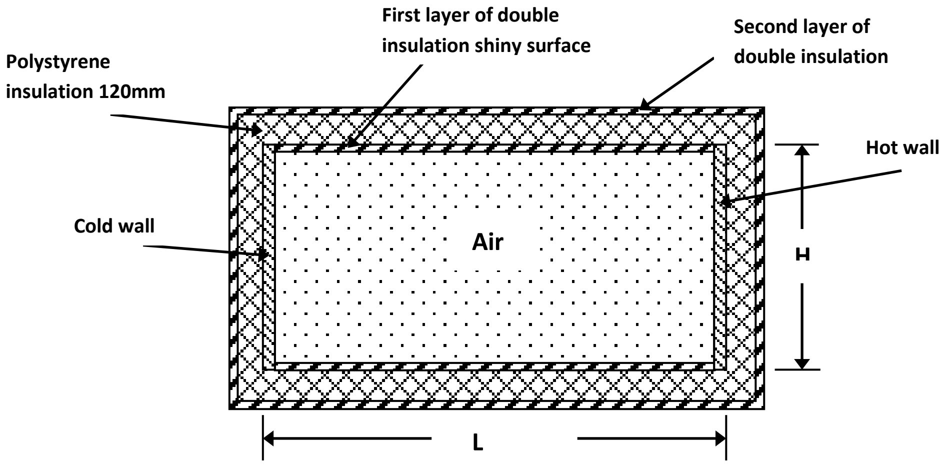


Figure 1 Schematic diagram of the enclosure with the hot and cold sides and all the insulation layers.

Hot side loop:

The hot side of the cavity is a flat plate single panel radiator with standard dimensions, being 600mm high by 600mm wide arranged and supported by a steel frame. The radiator was insulated to a thickness of 120mm on the back side using polystyrene boards.

A simplified hot water central heating system was constructed and connected to the radiator. A diagram of the hot side circuit setup is shown in figure 2. The heat input device was a 3kW immersion heater. To keep the temperature steady (to within ±0.1°C) in accordance with the European standard EN 442-2 [7], a PID temperature controller was fitted to the heater to match the energy input to the heater with the output from the system. The PID controller provides proportional, integral and derivative control, and has been adjusted to automatically compensate for temperature changes in the system.

The flow rate was measured using a glass tube rotameter which was connected to the outlet of the radiator. The water was circulated by a standard central heating pump. The water in the hot side enters the radiator from the top side and exits from the bottom side according to the British standard BS-3528-1977 [8].

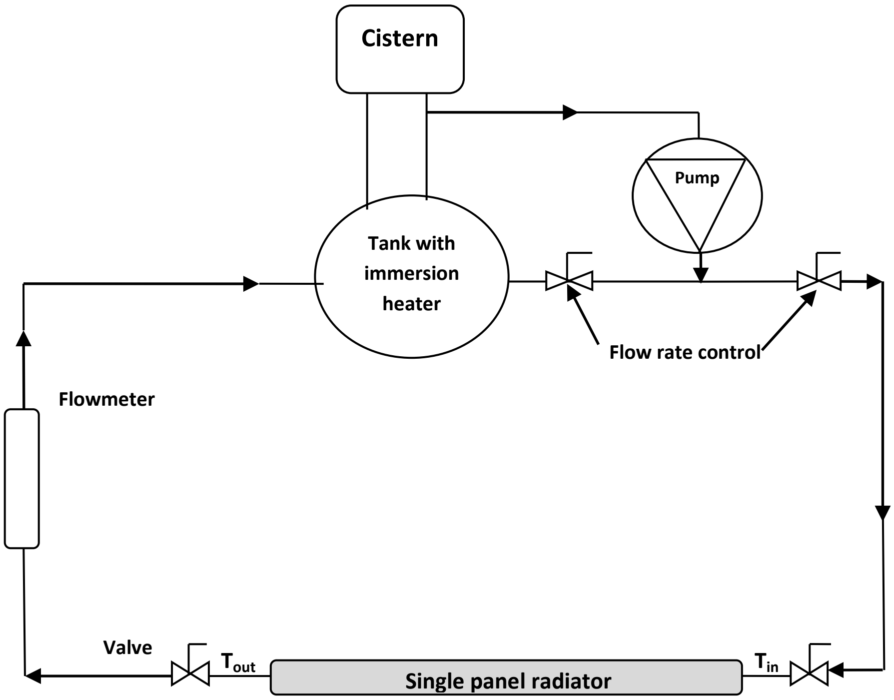


Figure 2 Schematic diagram of the radiator hot water side.

The water temperature at each measuring point just before and after the radiator was measured using three T-type thermocouples attached to the surface of the pipes, which were insulated with 25mm thick foam. The pipes were assumed to be at the same temperature as the hot water. The six thermocouples were attached to a PC through a USB based data acquisition system, so it was possible to monitor and record the temperatures at a suitable sampling rate, which was selected to be every five seconds.

To measure and compare the total heat transfer from the hot side to that from the cold wall side inside the enclosure, the total heat transfer from the hot wall side of the cavity (radiator) can be calculated using the following equation:

$$\dot{Q}_{tot(h)} = \dot{m}_h Cp_h (T_{hin} - T_{hout}) \quad (1)$$

Where $\dot{Q}_{tot(h)}$ is the total heat transfer rate from the hot side of the enclosure, Cp_h is the specific heat of the hot water side and $T_{h_{in}}$ is the water inlet hot wall temperature and $T_{h_{out}}$ is the water outlet hot wall temperature and $\dot{m}_h$ is the hot water mass flow rate.

Cold side loop:

The cold side of the cavity is also a flat plate single panel radiator with standard dimensions of 600mm high by 600mm wide, supported using a steel frame. The radiator was also insulated from the back to a thickness of 120mm using polystyrene boards and the gap between the radiator and the polystyrene board was filled using expanding foam insulation. The inlet and outlet of the radiator was connected to a recirculating chiller, (NESLAB CFC-FREE CFT-75), to maintain the water inlet of the cold wall temperature fixed at about 9°C. The CFT recirculating chiller is designed to provide a continuous supply of cooling fluid at a constant temperature and volume [9]. The unit consists of an air-cooled refrigeration system, a sealable reservoir, recirculating pump, and a temperature controller. The cooling capacity of the unit is 2500W with a temperature range of +5°C to +30°C [9].

The water temperature before and after the radiator was again measured using three T-type thermocouples attached to the surface of the pipes, which were insulated with 25mm thick foam. The pipes were assumed to be at the same temperature as the water. The six thermocouples were attached to a PC through a USB based data acquisition system to monitor and record the temperatures at the selected sampling rate, which was every five seconds.

Tests were carried out for only one inlet cold temperature to the radiator which was around 9C°. The radiator connected to the lab fixed base using a steel frame by a way in which it can be simply moved towards or away from the wall to get a range of different aspect ratios. The total heat transfer from the cold wall side of the cavity can be calculated using equation by just replacing the hot side temperatures with the cold side temperatures.

All the twelve thermocouples used for this work were calibrated using an electronic reference thermometer which has a resolution of 0.01°C. All the readings of the thermocouples had an error of ±0.2°C.

Thermal calculations and results:

The thermal calculations were performed for a range of hot wall temperatures ranging from 50C° to 75C° and for three enclosure sizes of aspect ratios 2.0, 1.0 and 0.5. The thermal calculations are include the collecting of the thermocouples readings for both the hot and cold sides and save them then taking the average values of each three thermocouples. The thermal calculations also include the calculation of the total heat transfer from the hot side and from the cold side. Moreover the thermal calculations include the calculation of the heat losses through the enclosure side walls which is the difference between the hot and cold heat transfer.

Collecting the thermocouples readings starts just after checking the thermocouples, the flow rate readings and the adjustment of the temperature controllers for both hot and cold sides. To this end it was possible to monitor and record the temperatures every five seconds, and save them to files. Collecting the experimental data took from 24 hours to 48 hours depending on the stability of the data collected during the day. The repeatability of the data collected using the thermocouples and in some cases show the stability of the thermocouple collected data for more than two days for both the hot and cold sides. The flow rate was monitored during the experimental time and collected manually. After getting enough collected data from the experiment for each hot wall temperature the experiment was deemed to be completed.

Calculations of the thermal results for each hot wall temperature (which ranged from 50C° to 75C°) were performed using equation (1) to calculate the total heat transfer for hot and cold walls. Then the heat losses were calculated using the following equation:

$$\dot{Q}_{loss} = \dot{Q}_{tot(h)} - \dot{Q}_{tot(c)} \qquad (2)$$

The final results for each aspect ratio and at each hot wall temperatures are shown in figure 3. The figure shows the relation between the heat transfer from the hot or the cold walls as a function of temperature ratio between hot and cold walls.

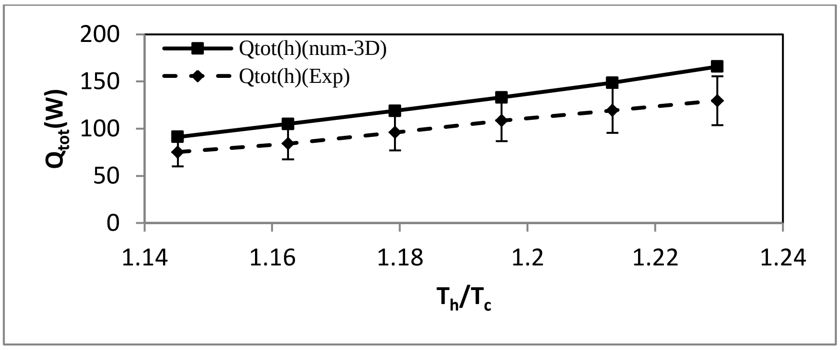
The calculated results involves the propagation of the uncertainty in the region, which is approximately ±20% for $\dot{Q}_{tot(h)}$ and ±20% for $\dot{Q}_{tot(c)}$. The principle source of this error is due to the uncertainty in the thermocouples and flow rate readings.

Numerical procedure and results:

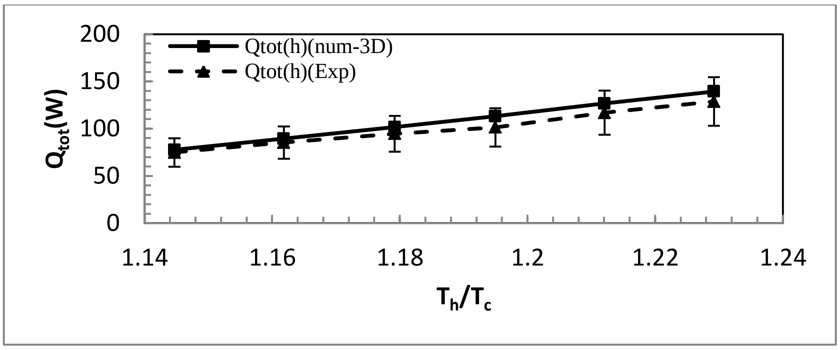
As the experimental results are 3D, the models that will be compared to the experimental ones must be 3D models. These calculations were performed for square and rectangular enclosures with three different aspect ratios 0.5, 1.0 and 2.0. Also the calculations were performed for different hot wall temperatures ranging from 50°C to 75°C with almost constant cold wall temperature at about 9°C. All the boundary conditions for the numerical simulation were selected to be almost the same as the experimental boundary conditions. The turbulent model used and its conditions were the same as that explained in detail by the authors in [5].

The numerical calculations were performed using the ANSYS 13 workbench software. The different enclosure shapes were designed using the design model in the ANSYS 13 work bench and then a non-uniform mesh was created for each enclosure aspect ratio using the ANSYS mesh; after that the numerical calculations performed using FLUENT in the ANSYS 13 workbench.

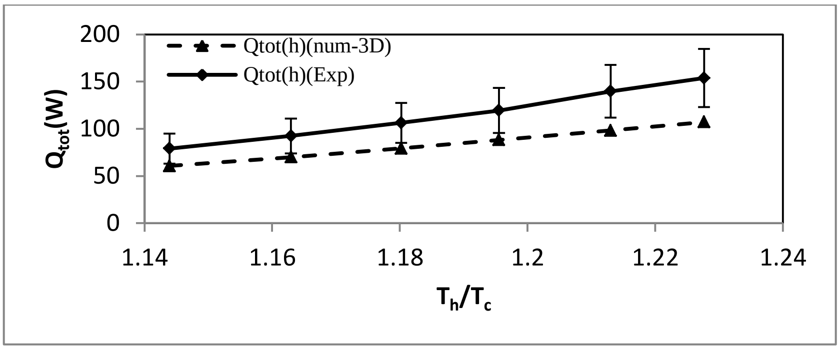
The thermal results show that the total heat transfer as a function of temperature ratio increases linearly for the three aspect ratios as the temperature ratio increase this can be seen clearly in figure 3. Also from this figure it can be seen that, as the aspect ratio increases, the overall trend of the total heat transfer increases as a function of temperature ratio.



(a)



(b)



(c)

Figure 3 Comparison between 3D numerical results and experimental results for aspect ratio a) 2.0, b) 1.0 and c) 0.5

Results comparison and discussion:

The 3D numerical calculations were provided so that they could be compared with the experimental results for three aspect ratios and for a hot temperature ranging from 50°C to 75°C. The 3D numerical calculations performed using the same boundary conditions as the experiments. The thermal results will be compared with the experimental results at different aspect ratios and different hot wall temperatures.

Figure 3 shows the comparison between these results for different aspect ratios and different hot wall temperatures. From this figure it can be seen that the 3D numerical results become closer to the experimental results as the aspect ratio approaches unity. Also from the figure it can be seen that the heat transfer increases as the aspect ratio decreases.

Compared to the experimental results, the 3D numerical results were within an average deviation of less than 13% for the three aspect ratios.

Conclusions:

The heat transfer inside a rectangular enclosure is studied experimentally for hot wall temperatures ranging from 50C° to 75C° for six temperature ratios and for three enclosure sizes of aspect ratios 2.0, 1.0 and 0.5.

From the experimental results it can be seen that the total heat transfer is increased as the temperature ratio increases. Also from the experimental results it can be seen that the total heat transfer from the hot side increases as the aspect ratio decreases. Conversely the total heat transfer from the cold side increases as the aspect ratio increase which means that the heat losses increase as the aspect ratio decreases

The 3D numerical results become closer to the experimental results as the aspect ratio getting closer to unity. The comparison between the experimental results and the 3D numerical results were within an average deviation of less than 13% for the three aspect ratios.

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