

## Experimental study of free convective heat transfer in a direction-sensitive open cavity

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### ABSTRACT

The aim of the present study was to propose a panel sensitive to the direction of heat transfer. For this purpose, a vertical rectangular cavity with prescribed dimensions was prepared and filled with water as the working fluid. A vertical mid-plane solid partition was installed within the cavity. Two relatively wide openings were created at the top and bottom of the partition, and they were equipped to operate as a pair of one-way flow controllers. The cavity was then fixed between two thick aluminum blocks by which the contact surfaces of the cavity were maintained at almost constant but different temperatures. The heat transfer rate through the cavity was evaluated for the same temperature difference applied in two opposite directions. Based on the results, the heat transfer rate in one direction was about 30% higher than that in the reverse direction. The difference in the heat transfer rate was clearly due to the individual flow patterns developed within the modified cavity. As a result, the proposed cavity is capable of restricting the heat transfer rate in one direction compared to the other, when applying the same temperature difference across the cavity.

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### 1. Introduction

Convective heat transfer in a sealed or open cavity with hot and cold walls has gained considerable research interest in recent decades due to its various applications. Reactor insulation, room ventilation, solar energy collection, and double-skin facades are examples of such practical and industrial applications. The flow field within a cavity is mainly characterized by a large re-circulating loop, which is formed by ascending and descending flows established parallel to the hot and cold walls, respectively. It has been demonstrated that heat transfer between the vertical hot and cold walls is mainly achieved by conduction when the Rayleigh

number is low, but the contribution of the convection becomes more predominant as this number is increased [1]. A large number of studies have recently been published concerning the effects of the geometry, working fluid properties, and thermal boundary conditions on convective heat transfer in an enclosure [2-4]. Heat transfer in a partitioned cavity is clearly different from that which occurs in a simple one due to modifications made inside a simple cavity. Partitions or obstacles within a cavity are generally used to simulate the type of convective heat transfer that occurs in more practical situations.

In an earlier study concerning heat transfer in a partitioned cavity, Zhang et al. [5] numerically investigated the thermal insulation effect of a

permeable screen installed inside a vertical rectangular enclosure. In a recent similar study, Avedissian and Naylor [6] considered a louvered metal blind located centrally within a double-glazed window. Using validated numerical results, a correlation was given for the average Nusselt number. Khalifa and Khudheyr [7] conducted an experimental investigation of the effects of 14 different configurations of partitions on natural convective heat transfer in an enclosure. Correlations for the tested configurations were reported and the percentage reduction in heat transfer for each case was compared to the case of a single cavity. Tasnim and Collins [8] analyzed the problem of laminar natural convective heat transfer in a square cavity with an adiabatic arc-shaped baffle. It was shown that the established flow and thermal fields were modified by the blockage effect of the baffle, and the degree of flow modification was enhanced by increasing the shape parameter of the baffle. Coman et al. [9] experimentally studied laminar free convection in a box with differentially heated end-walls. The cavity was divided into two connected chambers using a partial-height barrier. They found that the temperature fields in the two chambers were very different and that a complex flow structure had developed. Heat transfer was found to have little dependence on the barrier height or barrier position but decreased rapidly when the gap above the barrier became smaller. Rezaei et al. [10] employed an inclined barrier as a partition inside a two-dimensional cavity. Based on an experimental investigation, they concluded that the overall heat transfer rate through the cavity depends strongly on the inclination angle of the insulated barrier. Garoosi et al. [11] investigated natural convection heat transfer of nanofluid in a two-dimensional square cavity containing several pairs of heater and coolers. The walls of the cavity were insulated, whereas a constant temperature was assumed for the heaters and coolers. The simulation results indicated that the location of the isotherm obstacles has the most significant influence on the heat transfer rate. Moreover, it was found that there was an optimal volume fraction for the nanoparticles at each Rayleigh number in which the maximum heat transfer rate could be obtained. The common outcome of the above-mentioned studies was that the partition degraded overall heat transfer by altering the developed flow pattern within the cavity.

Ebrahimi et al. [12] experimented with a flat-plate, closed-loop, pulsating heat pipe in which total heat transfer was increased using interconnecting

channels. These channels affected the flow regime and enhanced the flow circulation, and therefore heat transfer, in one direction. A numerical procedure was followed on a single-phase liquid to show the role of interconnecting channels in achieving one-way flow. A modified heat pipe was also proposed by Ernst and Toth, which is available in the literature as a registered invention [13]. When using this heat pipe, heat transfers in one direction in a normal fashion and in the opposite direction as well, but only up to a prescribed point beyond which the reverse heat flow cuts off. This occurs because of the use of a limited liquid filling and at least one artery that is closed at the normal evaporator end and opened at the normal condenser end.

A review of the available literature on heat transfer in a partitioned cavity indicates that the problem of direction-sensitive heat transfer has rarely been studied thus far despite its applications. This includes thermal energy storage and thermal protection for electronic devices in which heat transfer is desired in one direction however, it should be restricted in the opposite direction. With this in mind, the present study was carried out with the aim of proposing a partitioned cavity in which heat is freely convected by flow recirculation in one direction but is restricted by the suppression of flow recirculation in the opposite direction. Flow circulation is clearly caused by the temperature difference existed within the fluid inside the cavity. This newly proposed feature of a partitioned cavity could be useful in the design and application of a unidirectional thermal layer.

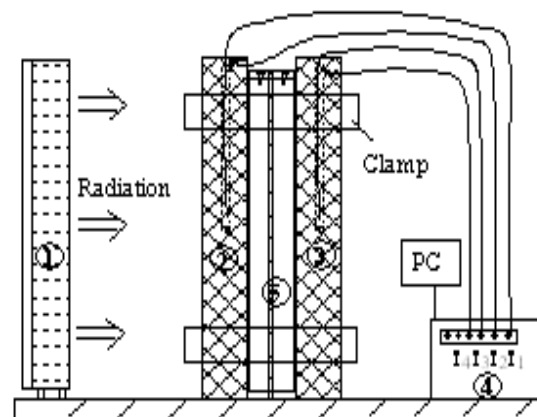


Fig. 1. Schematic diagram of the experimental apparatus.

- 1 - Electrical heater
- 2 - Hot aluminum block
- 3 - Cold aluminum block
- 4 - Temperature monitoring and recording system
- 5 - Cavity with a dividing plate and filled with water

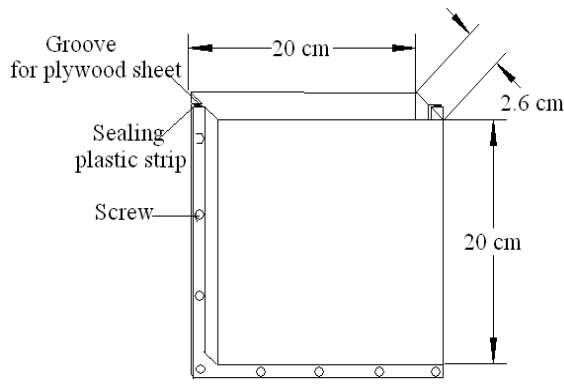


Fig. 2. Sketch of the cavity.

## 2. Experimental rig and procedure

As Fig. 1 indicates, the test rig was mainly composed of a cavity containing a dividing plate, a pair of aluminum blocks, an electrical heater, and temperature recording-controlling devices. The cavity was built using two symmetrical pieces made of steel plate 1 mm in thickness. A sealing plastic strip 2 mm in thickness was placed between the two half-pieces, which were screwed together to form the cavity. The overall height and length of the cavity were both 20 cm, with 2.6 cm of internal width (see Fig. 2).

Three plywood sheets 2 mm in thickness with a surface area of  $20.2 \times 20.2 \text{ cm}^2$  and coated with anti-water paint were prepared. The first plywood sheet was a simple plate, but two openings with the same dimensions ( $14 \times 3 \text{ cm}^2$ ) were made at the bottom and top sections of the other sheets. The surfaces of the openings were then covered using a very thin, rigid plastic layer (about 0.1 mm in thickness) in the third plywood sheet. These plastic sheets, which were slightly larger than the openings, were hung from the plywood surface at the top of the openings using anti-water adhesive tape. These plastic sheets could swing open under even the very weak air flow made by the movement of the sheets within the stagnant surroundings (see Fig. 3). Each of the plywood sheets was fixed in turn within the grooves at the middle of the cavity to divide the inside into two symmetrical parts.

In order to measure heat transfer through the cavity, a method similar to Lee's Disc Experiment [14] was employed. With the purpose of providing almost constant temperatures on both surfaces of the cavity, two aluminum blocks with approximately the same dimensions as the cavity ( $22 \times 22 \times 2.5 \text{ cm}^3$ ) were

prepared. As Fig. 1 illustrates, four K-type thermocouples were embedded within the aluminum blocks, two at the geometrical centers, and two at the top of the aluminum blocks. Another K-type thermocouple was used to measure the temperature of the surroundings, which was fairly constant. The thermocouples were connected to a data logging system (USB-TEMP series from MC Corporation) to monitor and record the local temperatures. The discrepancy in the thermocouples' readings was less than  $\pm 0.1^\circ\text{C}$  in the range of  $20\text{--}60^\circ\text{C}$  when they were placed at the same point within the aluminum block in a test procedure. The cavity was then squeezed between the aluminum blocks, which were pressed together using two adjustable clamps to reduce thermal contact resistance between the surfaces (see Fig. 1). In each stage of the experiment, the apparatus was first assembled and then the recording system was activated to measure and record the temperatures of the aluminum blocks. The electrical heater was turned on to increase the temperature of the adjacent aluminum block,  $T1$ , up to a prescribed value; then, its operation mode was changed to on-off mode to keep the block at a constant temperature. Meanwhile, heat was transferred throughout the cavity to the next aluminum block, which was exposed to the surroundings, until steady state heat transfer was established. In order to reduce heat transfer from the side surfaces of the cavity, these surfaces were covered with a 2.5-cm thick fiberglass layer. When a steady state was reached (temperatures  $T3$  and  $T4$ , measured at the top and center of the cold aluminum block, were almost constant,  $dT/dt < 0.0004^\circ\text{C}/\text{Sec}$ ), the experiment was terminated.

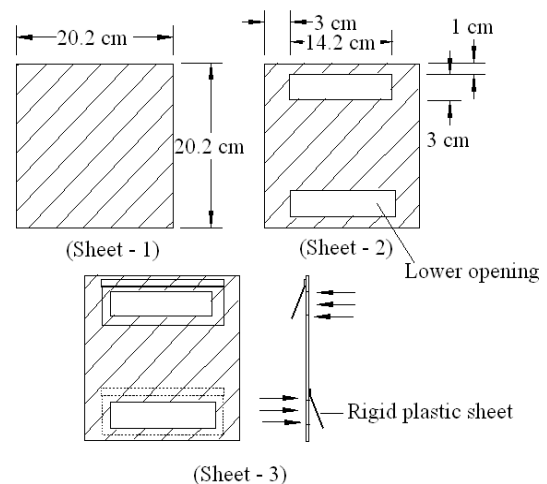


Fig. 3. Plywood sheets installed inside the cavity as the dividing plate.

### 3. Heat transfer to the surroundings

To obtain the heat transfer rate from the cold aluminum block to the surroundings maintained at  $T_\infty = 19.5^\circ\text{C}$ , a simple test was carried out in which one of the aluminum blocks was first heated to about  $60^\circ\text{C}$  with an electrical heater. The heater was then removed and an insulator plate made of a foam layer (5 cm in thickness) was placed at the back of the block. The other surfaces were exposed to the surroundings (like the main experiments). The temperature variations of the block were then recorded at two distinguished points with a time interval of one second. The recorded temperature variations are presented in Fig. 4 (fewer points are used for the sake of concise representation).

It can be seen that the temperatures measured at the center and top of the block were very close because of the higher thermal conductivity of the block. The maximum temperature difference in the two curves was only  $0.34^\circ\text{C}$  for the block temperature between  $35^\circ\text{C}$  and  $55^\circ\text{C}$ . Therefore, a constant temperature assumption was used for the aluminum blocks in the major stages of the experiment, and the effect of this assumption on the final results was taken into account in the uncertainty analysis.

Using the block temperature variations, the rate of temperature drop at each point,  $dT/dt$ , was calculated and presented in Fig. 5. In order to avoid the effect of small disturbances occurring in the temperature measurement, a smoothed variation curve was obtained using Microsoft Excel datasheet facilities. To accomplish this, a second-order polynomial as a trendline was fitted to the experimental data and its equation is also presented in Fig. 5. The difference between the experimental data and the smoothed values was at most 7.2%, which was considered in the uncertainty analysis.

$$Q = -mc \frac{dT}{dt} \quad (1)$$

in which  $m = 3.27 \text{ kg}$  and  $c = 896 \text{ J/kg-K}$  represented the mass and the specific heat capacity of the aluminum block, respectively. Fig. 6 presents the calculated heat rate transferred from the aluminum block maintained at the experimental condition to the surroundings. This figure indicates that the heat transfer rate to the surroundings reduces as the block temperature decreases. To be confident about the accuracy of the heat transfer rate presented in this figure, the average convective heat transfer

coefficient in dimensionless form as the Nusselt number was calculated using the following correlation recommended by Churchill and Chu [15]:

$$Nu_H^{\frac{1}{4}} = 0.825 + \frac{0.387 Ra_H^{\frac{1}{4}}}{\left[ 1 + \left( \frac{0.492}{Pr} \right)^{\frac{9}{16}} \right]^{\frac{1}{4}}} \quad (2)$$

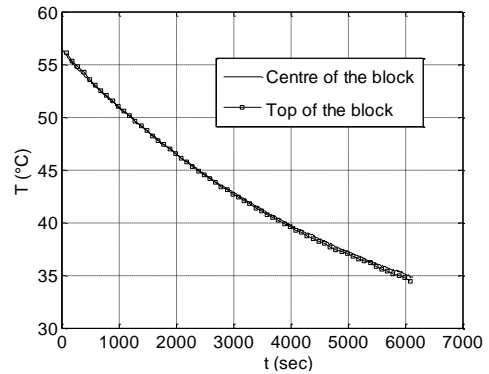


Fig. 4. Temperature variations of the aluminum block ( $T_\infty = 19.5^\circ\text{C}$ ).

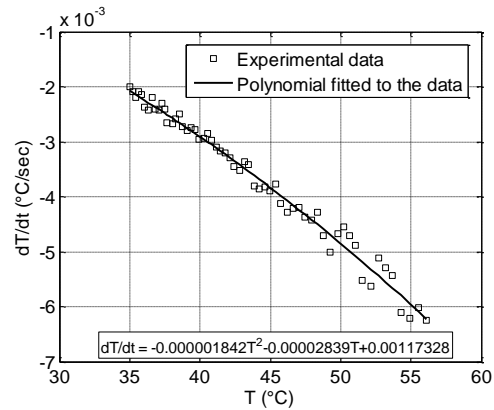


Fig. 5. Temperature reduction slope evaluated for different block temperatures ( $T_\infty = 19.5^\circ\text{C}$ ).

Next, the heat transfer rate from the block to the surroundings was calculated.

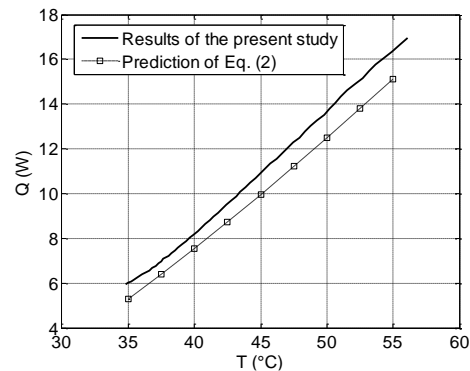


Fig. 6. Heat transfer rate from the aluminum block to the surroundings ( $T_\infty = 19.5^\circ\text{C}$ ).

This correlation is valid for free convective heat transfer from a vertical surface over the entire range of the Rayleigh number. Using physical properties of the air at mean temperature,  $T_m = (T_b + T_\infty)/2$ , the average convective heat transfer coefficient was calculated, based on which the heat transfer rate to the surroundings at  $T_\infty = 19.5^\circ\text{C}$  was specified. Also, the height of the aluminum block,  $H = 22$  cm, as quoted in this equation, was the length scale in dimensionless parameters. It is evident from Fig. 6 that the data predicted by Eq. (2) represent almost a linear variation, being slightly lower than the data obtained in the present study. Considering the uncertainties that exist in the results of the present study on the one hand and the approximation involved in Eq. (2) on the other, the results of this step in the present study could be considered reliable. In addition, the transient feature within the results of the present study could give rise to the discrepancy observed in Fig. 6.

#### 4. Experimental results and discussion

In order to evaluate the thermal resistances involved in the experiment, the cavity itself (with no dividing plate and no water inside) was first tested. The cavity was filled with foam with a density and thermal conductivity of  $16.6 \text{ kg/m}^3$  and  $0.037 \text{ W/m-K}$ , respectively, to prevent air intervention from the surroundings. To reach a steady heat transfer situation, the hot aluminum block was heated to  $53^\circ\text{C}$  and maintained at that temperature. The temperature variations of both the hot and cold aluminum blocks obtained from this step in the experiment are given in Fig. 7. This figure illustrates that under a steady condition, which is reached in about 1.6 hours, the temperature of the cold aluminum block,  $T_c$ , would be about  $35.73^\circ\text{C}$ .

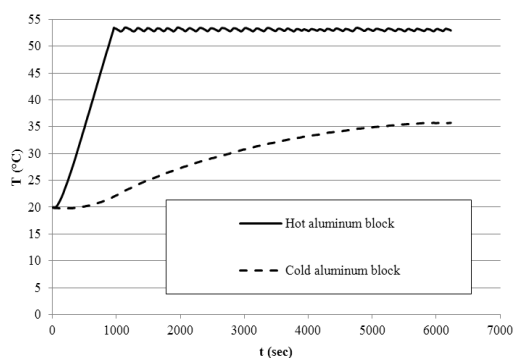


Fig. 7. Temperature variations of the hot and cold blocks (cavity filled with foam).

Using this temperature value and in reference to Fig. 6, the heat transfer rate from the cold block to the surroundings was evaluated to be  $6.26 \text{ W}$ . This value would be very close to the heat rate passing through the cavity, neglecting the heat loss from the side surfaces of the cavity. The steady heat transfer rate throughout the cavity was then used to calculate the equivalent thermal conductivity of the cavity,  $k_e$ , defined by

$$K_e = Q \frac{L}{A(T_h - T_c)} \quad (3)$$

where  $A$  is the cross-sectional area of the cavity perpendicular to the heat transfer direction and  $L$  is the cavity's external width. Using Eq. (3), the equivalent thermal conductivity of the cavity,  $k_e = 0.2537 \text{ W/m-}^\circ\text{C}$ , was calculated from this step of the experiment. This poor level of thermal conductivity was mainly due to the higher thermal resistance of the cavity structure itself. Contact thermal resistances existing between the aluminum blocks and the cavity surfaces also contributed to this equivalent thermal conductivity. The data obtained from the experiment were insufficient to specify these two thermal resistances individually.

At the next step of the experiment, having assembled the apparatus, the cavity was filled with water and the heater was turned on. Similar to the previous step, temperature variations of the aluminum blocks were recorded until a steady heat transfer state was reached. Fig. 8 presents these temperature variations, in which the temperature of the cold aluminum block is  $47.13^\circ\text{C}$  under the steady condition. Using this temperature value, the heat transfer rate from the cold block to the surroundings would be  $12.06 \text{ W}$ , as evaluated from Fig. 6. When the thermal resistance of the water contained within the cavity becomes parallel to the cavity structure's resistance, a considerable decline results in the overall thermal resistance of the cavity. Using Eq. (3), the equivalent thermal conductivity of the cavity was calculated to be  $1.426 \text{ W/m-}^\circ\text{C}$  in this situation. It is worth noting that this value is more than two times greater than the thermal conductivity of the stagnant water itself ( $0.6351 \text{ W/m-}^\circ\text{C}$  at  $40^\circ\text{C}$ ). Additionally, the contact thermal resistances were also compensated for, by the convection of the water contained within the cavity.

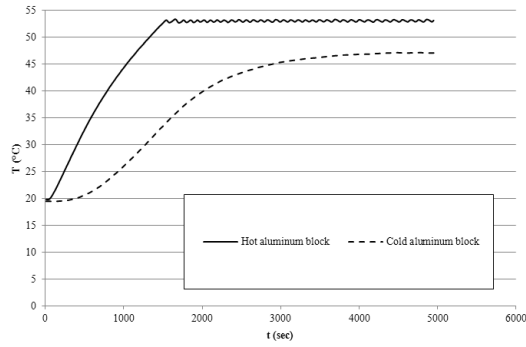


Fig. 8. Temperature variations of the hot and cold blocks (cavity filled with water).

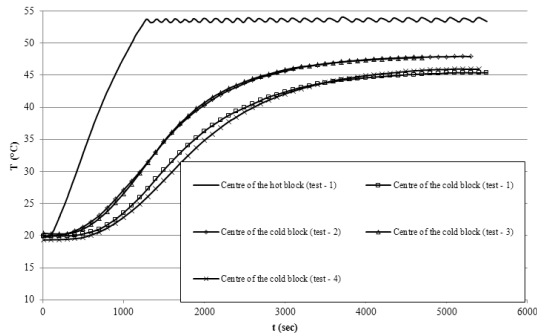


Fig. 9. Temperature variations of the hot and cold blocks (test-1, 2, 3, and 4).

Table 1. Quantitative results of the experiments

case	Q (W)	$T_h$ (°C)	$T_c$ (°C)	$K_e$ (W/m <sup>2</sup> °C)
Empty cavity	6.26	53.08	35.81	0.2525
Water only	12.06	53.05	47.13	1.4260
Test - 1	11.48	53.63	45.54	0.9933
Test - 2	12.35	53.63	47.93	1.5166
Test - 3	12.29	53.63	47.87	1.4935
Test - 4	11.67	53.63	46.06	1.0791

As the main part of the experiment, three plywood sheets were fixed in turn within the cavity and the temperature variations were recorded at the center of both the hot and cold blocks until a steady heat transfer situation was established. In these three cases, denoted by test-1, test-2, and test-3, sheet-1, sheet-2, and sheet-3 were fixed inside the cavity accordingly. One excessive test, test-4, was also carried out in which the direction of sheet-3 was reversed so that the gates were pushed to close by the flow convection. The temperature variations obtained from all of the experiments are presented in Fig. 9, in which one temperature profile has only been plotted for the hot aluminum block because of similar temperature variations. This figure indicates that the results of the two cases denoted by test-1 (simple

sheet within the cavity) and test-4 (sheet with unfavorable openings) were quite similar. Also, comparable temperature variations were seen for the test-2 and test-3 cases, in which the plastic covers on the openings were not present or were forced to open by the flow circulation.

The rate of heat loss from the cold block was then determined from Fig. 6 having the steady temperature of this block, and the equivalent thermal conductivity of the cavity was also calculated. To make a better comparison between the results, all the evaluated parameters are summarized in Table 1. Based on the results tabulated in Table 1, the heat transfer rate throughout the empty cavity was very poor compared to the cavity filled with water. The equivalent thermal conductivity of the cavity decreased by about 30% when installing a simple plywood sheet at the mid-plane submerged in water. When two openings were created on the dividing plate, the equivalent thermal conductivity improved even more than it did in the last case due to a better water recirculation inside the cavity. A slight difference in the conductivity was detected when these openings were covered by rigid plastic sheets, but they were nonetheless in a favorable flow direction. Finally, when the direction of the gates was reversed, the conductivity of the cavity again reduced to a lower value closer to that of a simple plywood sheet.

These findings indicate that an open cavity filled with water and equipped with a dividing plate, as described in the present study, has potential use as a panel sensitive to the direction of heat transfer. The gates were opened by flow convection, and hence heat was transferred freely in one direction while being restricted in the opposite direction when the gates were closed. The heat transfer rate in one direction was about 30% higher than that of the opposite direction, using the present experimental apparatus.

### 5. Uncertainty analysis

Uncertainty analysis based on the method suggested by Kline and McClintock [16] was performed. This method assumes that the total uncertainty in a main variable,  $W_R$ , is related to the uncertainty of the independent variables,  $W_1, W_2, \dots, W_N$ , by the following relation

$$W_R = \left[ \left( \frac{\partial R}{\partial X_1} W_1 \right)^2 + \dots + \left( \frac{\partial R}{\partial X_N} W_N \right)^2 \right]^{1/2} \quad (4)$$

where  $R$  is a function of the independent variables  $X_1, X_2, \dots, X_N$ . According to Eq. (3), the equivalent thermal conductivity of the cavity is a function of the heat loss rate from the cold aluminum block, the temperatures of the cold and hot blocks, and geometrical parameters. Hence, Eq. (4) was transformed to the following equation to evaluate the uncertainty in the equivalent thermal conductivity:

$$W_{k_e} = \left[ \left( \frac{L}{A(T_h - T_c)} W_Q \right)^2 + \left( \frac{Q}{A(T_h - T_c)} W_L \right)^2 + 2 \left( \frac{QL}{A(T_h - T_c)^2} W_T \right)^2 + \left( \frac{QL}{A^2(T_h - T_c)} W_A \right)^2 \right]^{\frac{1}{2}} \quad (5)$$

According to the thermocouples' readings ( $\pm 0.1^\circ\text{C}$  discrepancy at the range of  $20\text{--}60^\circ\text{C}$ ), the maximum uncertainty in the temperature measurement,  $W_T$ , was  $\pm 0.4^\circ\text{C}$ , including the constant temperature assumption made for the aluminum blocks. At most, 7.2% uncertainty was created by replacing the experimental temperature gradient with the smoothed temperature gradient variation as indicated in Fig. 5. Hence, 8% uncertainty was considered in the evaluated heat transfer rate from the cold block to the surroundings. Finally, 2% uncertainty was assumed in the measurement of the geometrical parameters, based on which  $W_L = 0.00056$  m and  $W_A = 0.0008$  m<sup>2</sup> were obtained. Using Eq. (5) and the quantitative results obtained for the various cases of the experiment (see Table 2), total uncertainty in the equivalent thermal conductivity as the main variable was evaluated to be within the range of 9–13%, which was tabulated in Table 2 individually.

Table 2. Uncertainty analysis results for the equivalent thermal conductivity

case	$W_{k_e}$ (W/m <sup>2</sup> °C)	$(W_{k_e}/k_e) \times 100$
Empty cavity	6.26	9.090128
Water only	12.06	12.77918
Test - 1	11.48	10.99517
Test - 2	12.35	13.05725
Test - 3	12.29	12.97885
Test - 4	11.67	11.30671

## 6. conclusion

With the purpose of investigating heat transfer in a direction-sensitive panel, the study of heat transfer through an open vertical cavity with prescribed dimensions was conducted experimentally. The cavity was modified to reduce free convection by suppressing the clockwise recirculation of the flow, meanwhile having a negligible effect on the counterclockwise recirculating flow. The equivalent thermal conductivity of the cavity as a determinative

parameter in this study was first evaluated for a simple cavity. A reduction of about 30% in this parameter was observed using a dividing plate with poor thermal conductivity within the cavity. The conductivity of the cavity was then increased to that of a simple cavity, when two relatively wide openings were made at the top and bottom of the dividing plate. These openings were then adapted to operate as a pair of one-way flow gates restricting free convection within the cavity in one direction. In both cases of unfavorable recirculating flow and the cavity with simple dividing plate, the equivalent conductivity of the cavity was in the same range. Whereas, quite comparable amounts of the conductivity were obtained in both cases where no cover was present on the openings and there was a favorable recirculating flow.

As a result, the proposed cavity with a dividing plate and two one-way flow gates could be introduced as a direction-sensitive heat transfer panel so that heat can be freely transferred in one direction but restricted in the opposite direction. For better heat transfer reduction in one direction compared to another, further modifications should be made in the cavity and in the dividing plate.

## Nomenclature

$A$	surface area of the cavity, 0.04 m <sup>2</sup>
$c$	specific heat capacity of aluminum, 896 J/kg-°C
$H$	height of the aluminum block, 0.22 m
$k_e$	equivalent thermal conductivity of the cavity (W/m-°C)
$L$	cavity external width, 0.028 m
$m$	mass of the aluminum block, 3.27 kg
$Nu$	Nusselt number ( $hH/k$ )
$Pr$	Prandtl number ( $\mu c_p/k$ )
$Q$	heat transfer rate through the cavity (W)
$R$	an arbitrary function
$Ra$	Rayleigh number ( $g\hat{\alpha}(T_h - T_c)H^3/\nu\alpha$ )
$t$	time (sec)
$T$	temperature (K, °C)
$W$	uncertainty in an independent variable
$X$	independent variable, $R(X_1, \dots, X_N)$
Subscripts	
$b$	block
$h$	hot
$c$	cold
$\infty$	surroundings

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