

Research Paper

Experimental investigation and general correlation of passive heat transfer in enclosures at different operating, orientations and venting configurations

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HIGHLIGHTS

- Comprehensive heat transfer experiments in enclosures for different operating and geometric conditions.
- Effects of enclosure inclination, venting configurations, opening ratios and Rayleigh numbers are investigated.
- General correlation of Nusselt No. in terms of Rayleigh No., opening ratio, inclination angle and venting arrangements.

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ABSTRACT

Free convection heat transfer in enclosures was previously studied for specific operating, geometric and orientations conditions. A lot of correlations are presented in the literature for specific conditions. The current work aims to do inclusive experiments of heat transfer in heated enclosures for different inclination angles, venting configurations, venting opening ratios and Rayleigh numbers. Three various configurations of enclosure openings for air venting were examined: (i) top openings, (ii) side openings, and (iii) top and side openings. The experiments were conducted for wide ranges of venting opening ratios, inclination angles and Rayleigh numbers for each configuration of air venting. Developed correlation was presented for the studied venting configurations to expect the enclosure Nusselt number in terms of Rayleigh number, opening ratio, enclosure inclination angle and openings arrangements. It was found that the developed correlations can predicts the present experimental and previous results within $\pm 8\%$ error. The results showed that for all the studied venting configurations, the Nusselt number enhances with increasing the Rayleigh number, vents opening ratio and in range from 0° to 90° enclosure inclination angle. Furthermore, the top and side openings configuration showed the greatest Nusselt number.

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1. Introduction

Passive heat transfer in fluid layers of enclosures and cavities has a wide range of engineering applications for example solar collectors, heat transfer in buildings materials and cooling of electronic equipment including portable computers and mobile phones, power supplies, transformers and telecommunications enclosures. Although many numerical and experimental works and heat transfer correlations are available in the literature for passive heat transfer in enclosures, most of these studies and correlations were conducted for specific cases and conditions. General correlation of passive heat transfer in enclosure that can be applied under a wide variety of geometric, operating (Rayleigh numbers),

venting configurations and orientations conditions are not available in the literature. In the present study comprehensive heat transfer experiments are conducted and the results are investigated and correlated in dimensionless heat transfer correlation that can be applied for a wide-ranging enclosures geometric, orientation and operating conditions.

Literature review reveals that several experimental and numerical works have been achieved to investigate free convection in enclosures fluid layers [1–10]. Some of these studies were conducted for fully closed enclosures and others for vented enclosures. The location of the heat source, power density, enclosure geometric dimensions, enclosure inclination angle and opening ratio in case of vented enclosures were the main investigated parameters of the different studies. In these studies, to study certain parameters the other parameters were kept constant at fixed values, therefore the results of each study are valid only under the conditions of

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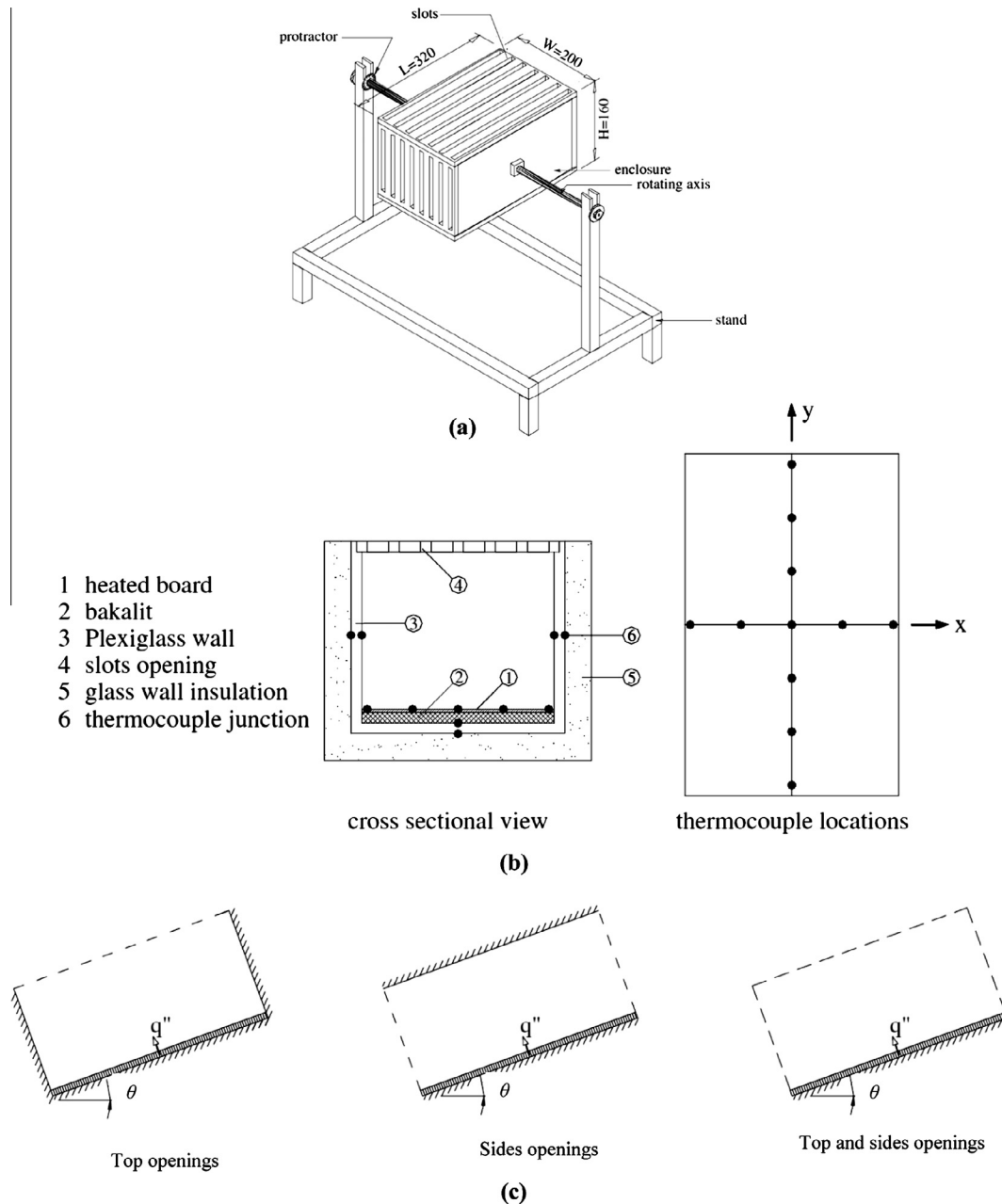


Fig. 1. Experimental facility: (a) schematic diagram of experimental facility, (b) thermocouples distribution along the heated surface, (c) schematic of different venting openings configurations.

this study. Lasance and Joshi [1] conducted a review and summary of passive heat transfer correlations and results in enclosures. Several numerical investigations were conducted to study free convection in fully open inclined cavities [2–5] under isothermal side walls and at various ranges of Grashof number and aspect ratio. The studies showed the dependence of the heat transfer coefficient and flow field in the cavity on the inclination angle and aspect ratio. Other similar numerical works were conducted under conditions of adiabatic side walls and isothermal bottom wall of fully open cavity at different ranges of Grashof numbers [6–11]. Miyamoto et al. [12] investigated numerically 2D heat transfer in fully and half partially open inclined enclosures under isothermal walls boundary conditions. Nada [13] experimentally investigated free convection heat transfer of annular layer of fluid in annular cavity at a wide-ranging Rayleigh number, annulus aspect ratio, and different annulus inclination. The results displayed the augment of

the heat transfer with the increase of annulus gap width, lowering the annulus inclination with the horizontal, and rising Rayleigh number.

Chakroun et al. [14] and Elsayed and Chakroun [15] studied experimentally the heat transfer in an inclined partially opened enclosure for different opening ratios. Adiabatic side walls and constant heat flux bottom wall conditions were maintained in this studies. The cavity opening was positioned at different location in the upper surface. The study revealed Nusselt number improves with increasing the opening ratio. Nada and Moawed [16] experimentally investigated heat transfer in inclined slots open enclosures for different venting surfaces; sides, top and side and tops. The study was conducted at fixed value of heat flux. The study showed improving Nusselt number with increasing the openings area. Yu and Joshi [17] carried out experimental work to study passive heat transfer from a separate heat source of flush in horizontal

enclosure which is compact and partially open. The effect of conduction and radiation in enclosures heat transfer were investigated numerically and experimentally by Adams et al. [18] and Yu and Joshi [19] for discretely heated enclosures. The results revealed that conduction and radiation contributes in heat transfer in enclosure and cannot be neglected.

Nada [20] studied experimentally heat transfer and fluid flow characteristics of natural convection over heated rectangular finned base plate in narrow enclosures which arranged horizontally and vertically. The study was conducted at a wide-ranging Rayleigh number (Ra), fin spaces and fin lengths. It was found that using of fins permanently improves the rate of heat transfer if it was compared with enclosure of a bare base plate. Moreover, fin-array geometry optimization of and correlations of Nusselt numbers have been developed. More recently, Tamayol et al. [21] implemented experimental and numerical investigations for free convection heat transfer from enclosures with and without rectangular fins. It was found that the radiation heat transfer has represents about 50% of enclosure heat transfer. Morshed et al. [22] investigated the laminar mixed convection in a lid-driven square cavity with two isothermally heated square internal blockages. More recently, natural convection in double-diffuse natural convection in enclosures and cavities were investigated [23–26]. The effects of the aspect ratio, heating source orientation, segments heat source and buoyancy ratio were studied in these works.

As revealed above, most of the earlier studies of enclosures or vented cavities had been conducted with varying some of the controlling parameters and keeping the other parameters constant. This makes the results and deduced correlations are restricted for these conditions. Comprehensive experimental studies that varying all the controlling parameters to deduce general correlation of heat transfer aren't presented in the current literature. Consequently, the current work targets to fill this gap by conducting a comprehensive experiments to study the effects of the controlling parameters (Rayleigh number, enclosure inclination angle, venting opening ratio and venting distribution along the enclosure surface) on heat transfer and develop a more general heat transfer correlation that can be applied for a wide ranges of operating and geometric conditions.

2. Experimental facility and procedure

2.1. Experimental facility

The schematic of the experimental setup is illustrated in Fig. 1a. It mainly consists of a cuboid enclosure with interior dimensions of 20, 16 and 32 cm (L , H and W). The heat source was applied on the enclosure bottom side. The bottom side was fabricated from two attached layers. The inner-side layer was fabricated from Bakelite plate 1 cm thick while the outer layer was fabricated from Plexiglass plate having a thickness of 8 mm. An electric heating board of 20×32 cm ($L \times W$) dimensions was attached internally on the Bakelite plate. The electric heating board was connected to a power supply (DC) to regulate the heated input power. Others enclosure walls were fabricated from a Plexiglass sheet of 8 mm thickness. Enclosure walls, except walls of venting openings, were insulated thermally by glass wool of 5 cm thick to reduce the enclosure heat losses. The enclosure was fixed on a rotating mechanism to change the enclosure inclination. The inclination angle was adjusted using attached frame protractor. To attain complete natural convection environment by avoiding any other effects (e.g. wind, etc.) on free convection throughout the experiments, the enclosure was protected by 1-m side length plastic sheets from the four sides and opened top and bottom. The electrical voltage and current supplied to the heating board were determined by a digital ammeter and

voltmeter with accuracy of 0.025%. The temperature distribution along the heating surface was measured by using eleven k-type thermocouples equally distributed along the heating surface as illustrated in Fig. 1b. Two other k-type thermocouples were attached to the inner and outer surfaces of the enclosure walls (see Fig. 1b) to calculate the heat losses from walls. Another thermocouple was used to measure the ambient outside temperature of the enclosure. The thermocouples were calibrated using a constant temperature path and ± 0.2 °C measuring accuracy was attained. A data acquisition system and a PC computer were used to acquire and record the temperature signals.

2.2. Experimental procedure and parameters ranges

The ranges of the studied parameters are given below:

Rayleigh number	2×10^8 to 2×10^9
Enclosure inclination angle	0–180° at increments of 30°
Opening ratio OR (ratio between areas of wall openings to the total area of the wall)	0.25–1 at increments of 0.25
Openings configuration	Top opening, side opening and top and side openings (see Fig. 1c)

The experimental program and procedure were as follows:

1. Prepare the enclosure with the required venting surface.
2. Prepare the venting surface with the required opening ratio.
3. Adjust the enclosure inclination angle to the required angle.
4. Supply the input power to the heated board and adjust it to the required power.
5. Wait for attaining the condition of steady state, about four run hours were needed to achieve steady state conditions. The achievement of steady state conditions was assured when the measured temperatures were with variation of 0.2 °C for 10 min.
6. Record all measurements (current, voltage, thermocouples readings).
7. Repeat steps 4–6 with various Rayleigh numbers (about seven values).
8. Repeat steps 3–7 with various enclosures inclination angles (0, 30, 60, 90, 120, 150 180).
9. Repeat steps 2–8 with various venting opening ratio (0.25, 0.5, 0.75 and 1).
10. Repeat steps 1–9 with various enclosure venting surfaces (top surface, sides surfaces and top and sides surfaces).

2.3. Data reduction

Temperature measurements were used to calculate the Rayleigh number (Ra) of the free convection from:

$$Ra = Gr Pr = \frac{g\beta(T_s - T_\infty)H^3}{\alpha\nu} \quad (1)$$

where g is the acceleration of gravity, T_s is the heated surface average temperature, T_∞ is the ambient temperature, H is the enclosure height, and β , α and ν are the volume expansion coefficient, thermal diffusivity and kinematic viscosity of air, respectively. All properties were determined at $(T_s + T_\infty)/2$.

The radiation and conduction heat losses ($q_c + q_r$) across the enclosure walls were estimated. The heat transfer rate by free convection (q) across the enclosure openings is calculated from the heat balance to the entire enclosure as follows:

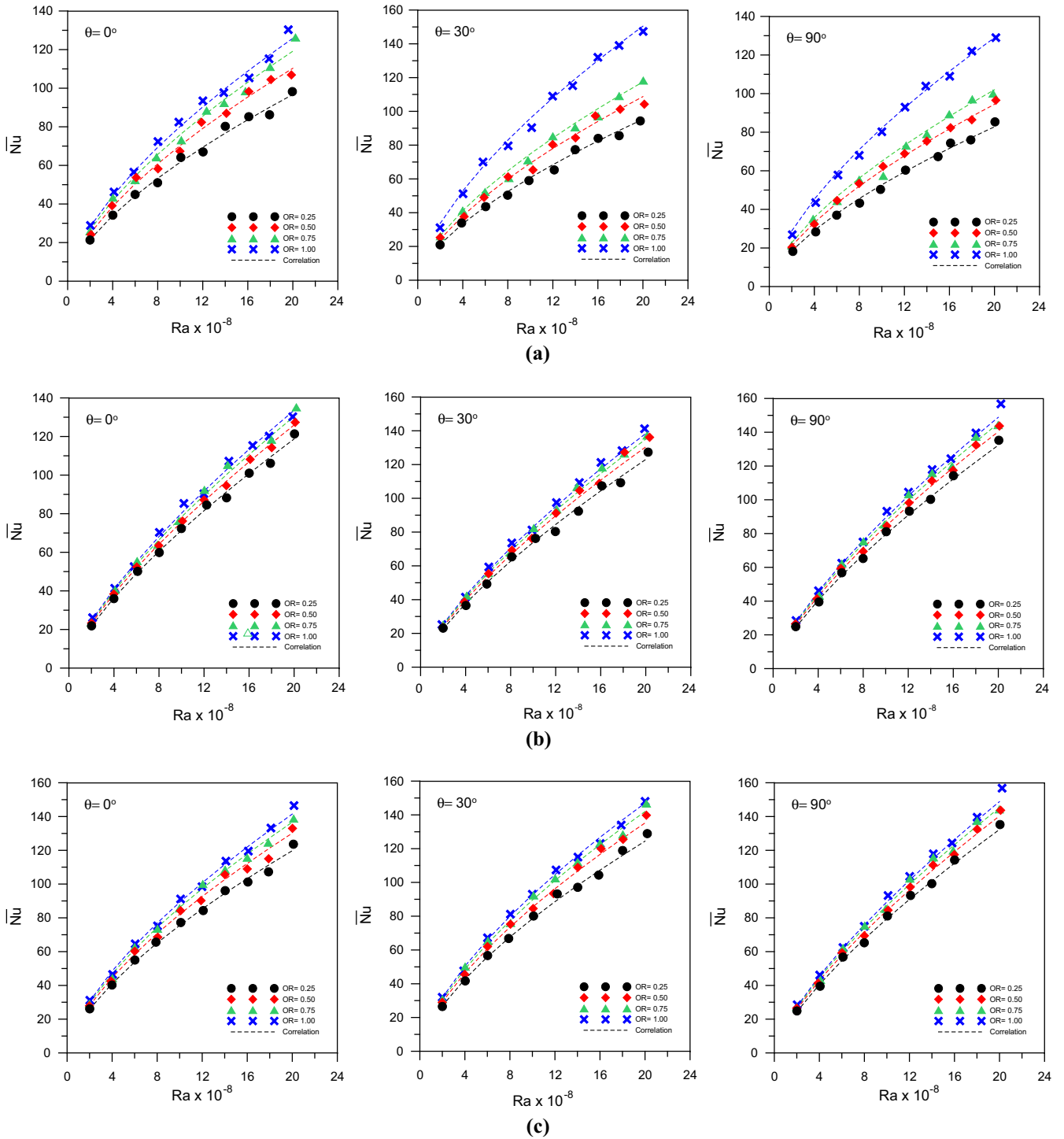


Fig. 2. Effect of opening ratio on \overline{Nu} : (a) top openings enclosure, (b) sides openings enclosure, (c) top and sides openings enclosure at different θ .

$$q = VI - (q_c + q_r) \tag{2}$$

where V and I are the input voltage and current to the heated surface, q is the heat transfer from the heated surface by free convection, q_c is the heat losses by conduction across the enclosure walls and q_r is the radiation heat transfer from the heated surface to the ambient over the enclosure openings. The conduction heat losses are calculated from the conduction heat transfer through the blind walls of the enclosure using the following equation

$$q_c = \sum kA_j \Delta T_j / t \tag{3}$$

where j is the blind wall number, k_w is the wall thermal conductivity, A is the area of side wall, t is the wall thickness and ΔT_j is the temperature difference across the enclosure blind walls. Heat transfer by conduction from the vented wall is zero as the vented wall is considered open wall at the ambient temperature.

The heat loss rate by radiation from the heating board across the wall venting openings of the enclosure was calculated as follows:

$$q_r = \frac{A\sigma[(T_s + 273)^4 - (T_\infty + 273)^4]}{(1 - \epsilon)/\epsilon + 1/F_s} \tag{4}$$

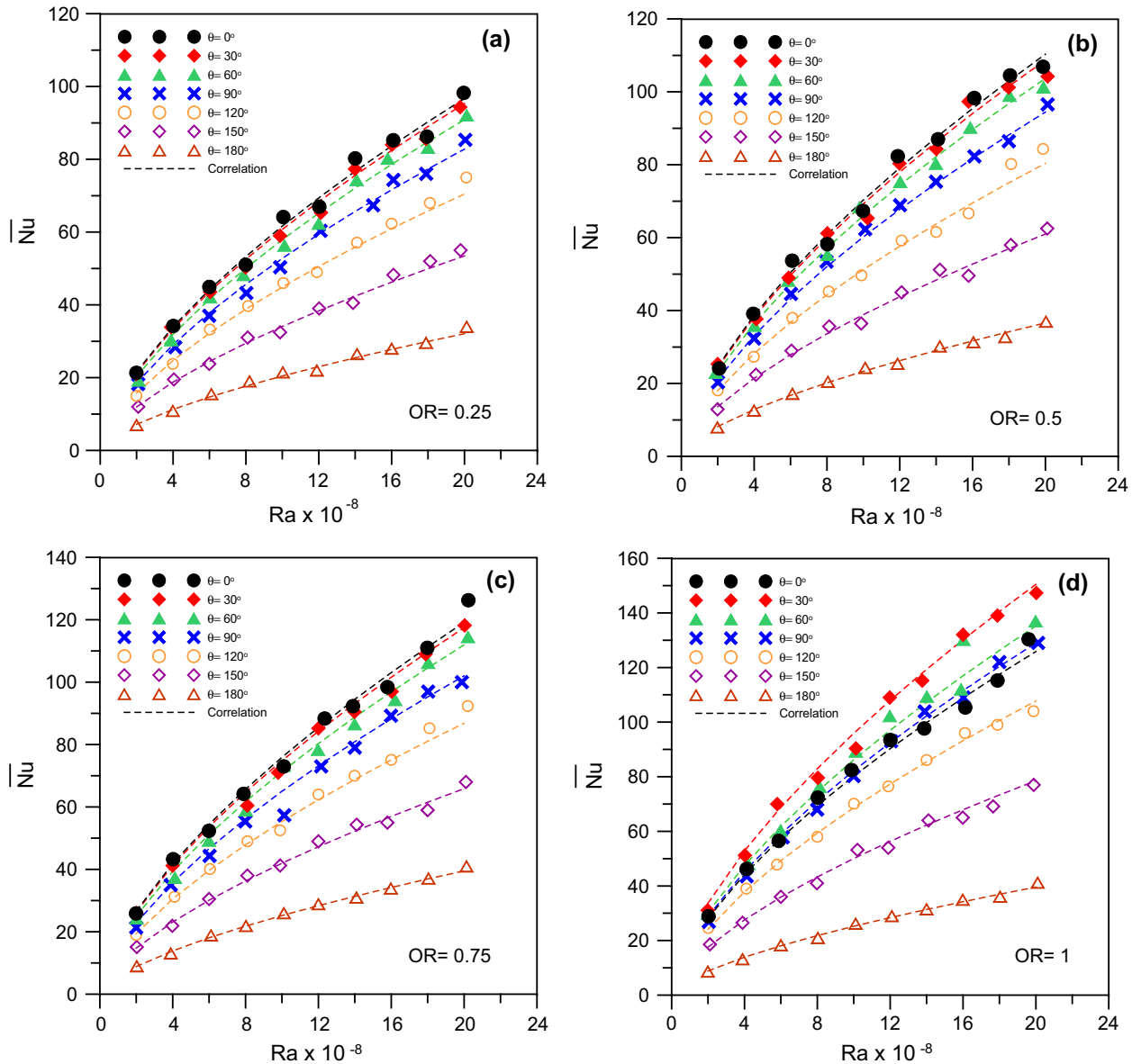


Fig. 3. Effect of enclosure inclination on \overline{Nu} of top openings enclosure at different OR and Ra.

where A is the heated surface area, T_s is the heated surface temperature, ε is the surface emissivity and F_s is the shape factor between the heated surface and the walls with openings. In Eq. (4) the venting enclosure surface (surrounding) was assumed as an ideal body at T_∞ . F_s was determined from the expressions and the graphs given by that presented by Incropera and De Witt [27] and Suryanarayana [28]. The conduction and radiation heat rates were estimated for all the experiments and were within 13% and 6% from the input power, respectively. Taken the ambient temperature as the reference temperature, the heat transfer by free convection from the heated surface to the convected air and the local heat transfer coefficient by free convection from the heated surface is calculated by:

$$q = hA(T_s - T_\infty) \quad \text{Or} \quad h = \frac{q}{A(T_s - T_\infty)} \quad (5)$$

where T_s and T_∞ are the local heated surface and ambient temperatures, respectively. Measurements revealed that the temperature variation along the heated surface was negligible along the x -axis and varies insignificantly along the y -axis. Consequently, to calculate the average convective heat transfer coefficient, the heating

surface was divided into six equal sections as illustrated in Fig. 2. The average convective heat transfer coefficient can be given by:

$$\bar{h} = \frac{1}{6} \sum_{i=1}^6 \frac{(q/A)}{(T_{s,i} - T_\infty)} \quad (6)$$

where i is the part number of the heated board. The Nusselt number can be expressed as:

$$\overline{Nu} = \frac{\bar{h}H}{k} \quad (7)$$

Eqs. (1)–(7) can be combined together to be in the form $\overline{Nu} = f(V, I, T_{s,i}, T_\infty, H, t, k, k_a, \varepsilon)$. The uncertainty ($\Delta \overline{Nu}$) in \overline{Nu} was calculated from [29]:

$$(\Delta \overline{Nu})^2 = \sum_{i=1}^n \left(\frac{\partial \overline{Nu}}{\partial x_i} \Delta x_i \right)^2 \quad (8)$$

where $\overline{\Delta x_i}$ is the uncertainty in the x_i variable. The maximum uncertainty in the different measured variables were: 0.25%, for V and I ,

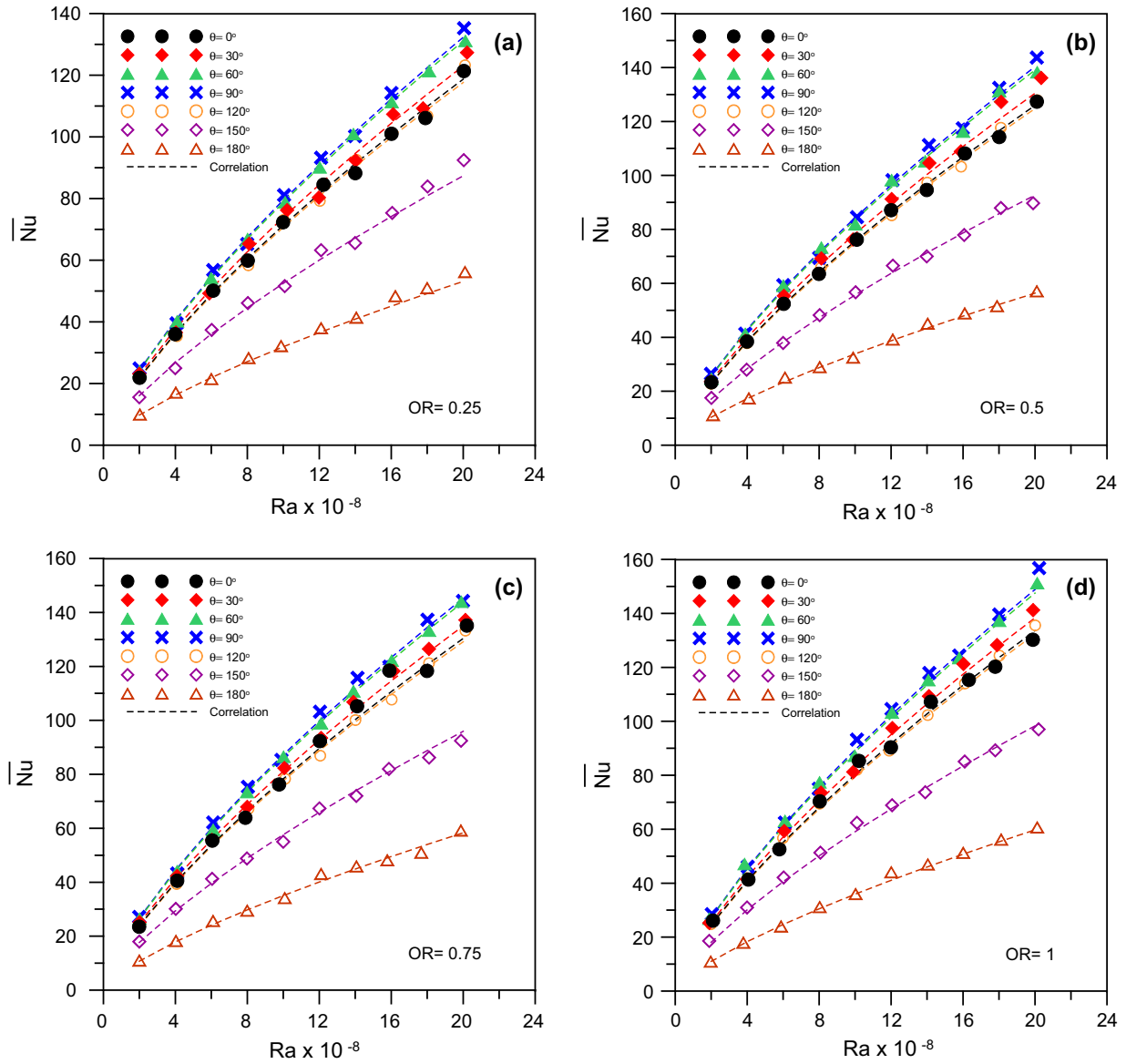


Fig. 4. Effect of enclosure inclination on \overline{Nu} of sides openings enclosure at different OR and Ra.

0.8% for any temperature measurements, 0.55% for any length measurements, 0.5% for k_{air} , 2% for $k_{plexiglass}$, and 5% for $\epsilon_{heated\ surface}$. It was obtained that the uncertainty in Ra and \overline{Nu} varies from 3% to 5% and from 4% to 8%, respectively.

3. Results and discussions

The present study was conducted to correlate and investigate the effects of Rayleigh number, venting opening ratio, enclosure inclination angle and the venting surface configurations on Nusselt numbers. Figs. 2–9 display the variation of the average Nusselt number, \overline{Nu} against Rayleigh number, Ra for three different venting configurations and at different opening ratios and enclosure inclination angles. The figures illustrate increasing of \overline{Nu} with the increase of Rayleigh numbers for the three venting configurations and at any opening ratio, OR, and enclosure inclination angle, θ . The increase of the \overline{Nu} with Rayleigh is due to the increase of the buoyant force acting on the air. Increasing the buoyant force increases the convection current of air flow which results to higher heat transfer rate.

The influence of opening ratio on the \overline{Nu} for the different venting configurations and some enclosure inclination angle (0° , 30° and 90°) for example is shown in Fig. 2. The figure illustrates that the average Nusselt number increases with increasing the venting opening ratio and this trend is the same for any venting configurations and enclosures inclination angles. This is owing to the decrease of the resistance to the air circulation motion in the enclosure with increasing the opening ratio due the increase of area of openings. This increases the rate of the hot air exchange by cold air which leads to high heat transfer from the heated surface. The results and figures also show that the influence of the opening ratio on \overline{Nu} decreases and tends to vanishes with increasing the inclination angle to approach 180° . This is due to the presence of still air zone in the enclosure as θ approach to 180° . This stagnant zone occupies the entire enclosure volume when the inclination angle reaches 180° .

Figs. 3–5 present the variation of the \overline{Nu} against the enclosure inclination angle for the entire range of Rayleigh number at different opening ratios for the three states of venting configurations, respectively. In case of top-venting configuration, Fig. 3

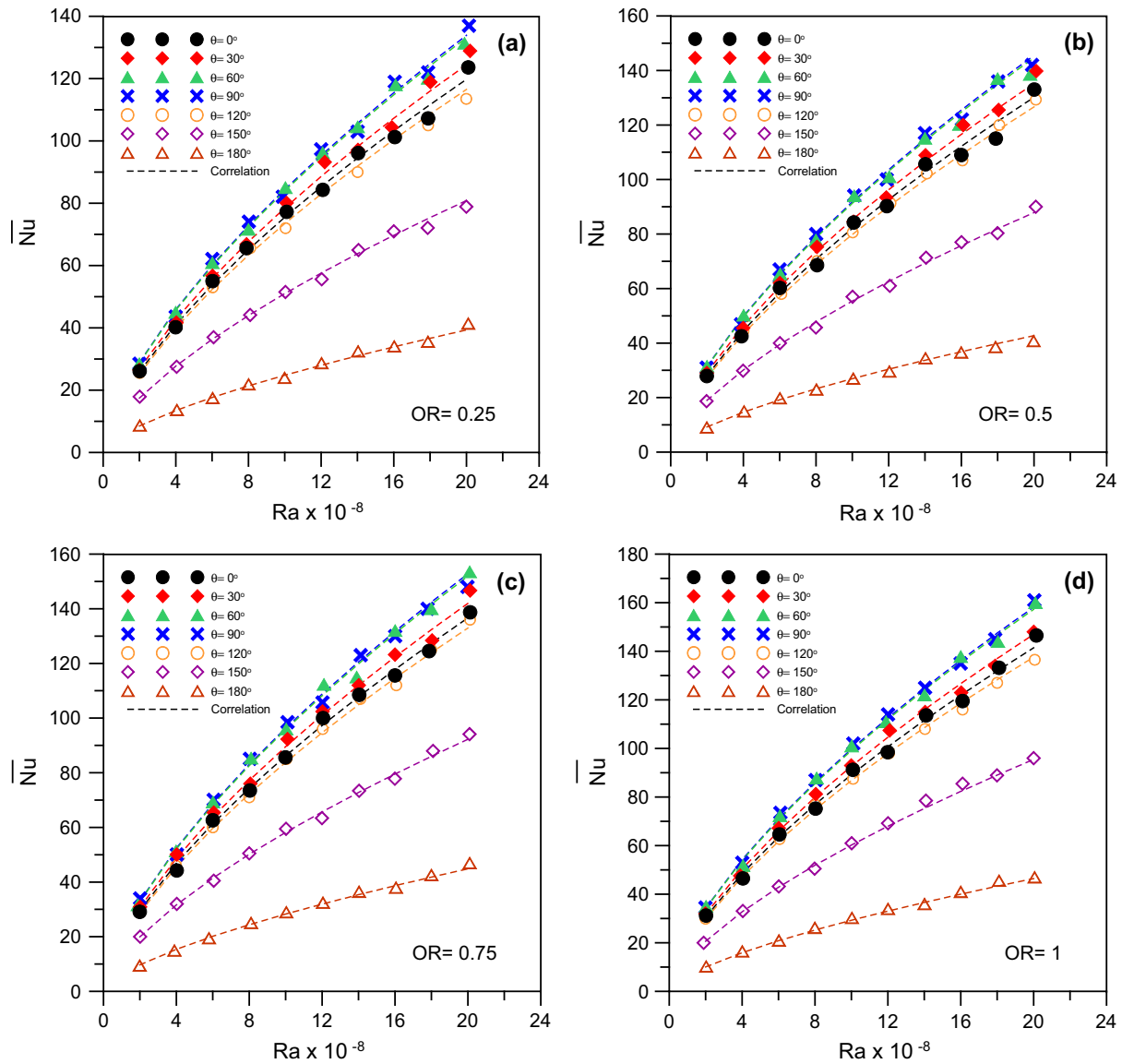


Fig. 5. Effect of enclosure inclination on \overline{Nu} of top and sides openings enclosure at different OR and Ra.

displays that at $\theta < 60^\circ$ and $2 \times 10^8 < Ra < 20 \times 10^8$, the variation in the θ has approximately no effect on the \overline{Nu} , however at $\theta > 90^\circ$, the average Nusselt number decreases with increasing θ . These discrepancies are due to the buoyant driven force that helps heat to transfer outside the enclosure. In case of small inclination angle ($\theta < 60^\circ$), natural convection happens upward the heated surface and multiple thermal plumes form and rise at different locations of the heated surface and the warm air ascending from the heated surface is replaced by descending cooler air from the ambient. As θ approaches 90° , air was drawn horizontally into the cavity toward the heated wall. As the air is heated it rose, turned around 90° and accelerated slightly as it moved toward the openings. For $\theta > 90^\circ$, free convection starts at the bottom of the heated surface and the tendency of the air to move up is obstructed by the surface and a still air zone appears at the enclosure top side. This still air zone increases as θ increases. This stagnant zone lessens the area fraction of the heated surface that drives the flow throughout the enclosure and this directs to higher walls temperature and lower average Nusselt number. In case of side-opening and top and side opening configurations, Figs. 4 and 5 illustrates the increase of the

\overline{Nu} with increasing θ from 0° to 90° followed by the decrease of \overline{Nu} with increasing θ from 90° to 180° . The average surface temperature increases and \overline{Nu} decreases with increasing θ from 90° to 180° . This behavior of the \overline{Nu} with θ is owing to the behavior of the buoyant driven flow with the inclination angle. At $\theta = 0^\circ$, cold air was drawn from the side openings and multiple thermal plumes are created at different locations of the heated walls and rise up. In top and side openings the hot air exits from the top wall openings. Nevertheless, in side openings configuration, this convected hot air motion is stopped by the enclosure top blind wall and this drives the air to vary its direction horizontally to exit from the openings of the top part of the sides walls. As θ increases to near 90° , the orientation of the heated surface approaches the buoyant force direction and this improves the convection flows from the heated surface and consequently increases the heat transfer rate. Cold air was drawn at the bottom of the vertical heated surface and a thermal boundary layer is formed on its surface. Cold air is pulled into the boundary layer from the bottom in the case of side openings and from the bottom and sides in the case of sides and top-openings, and the hot air is ejected at the top. Increasing

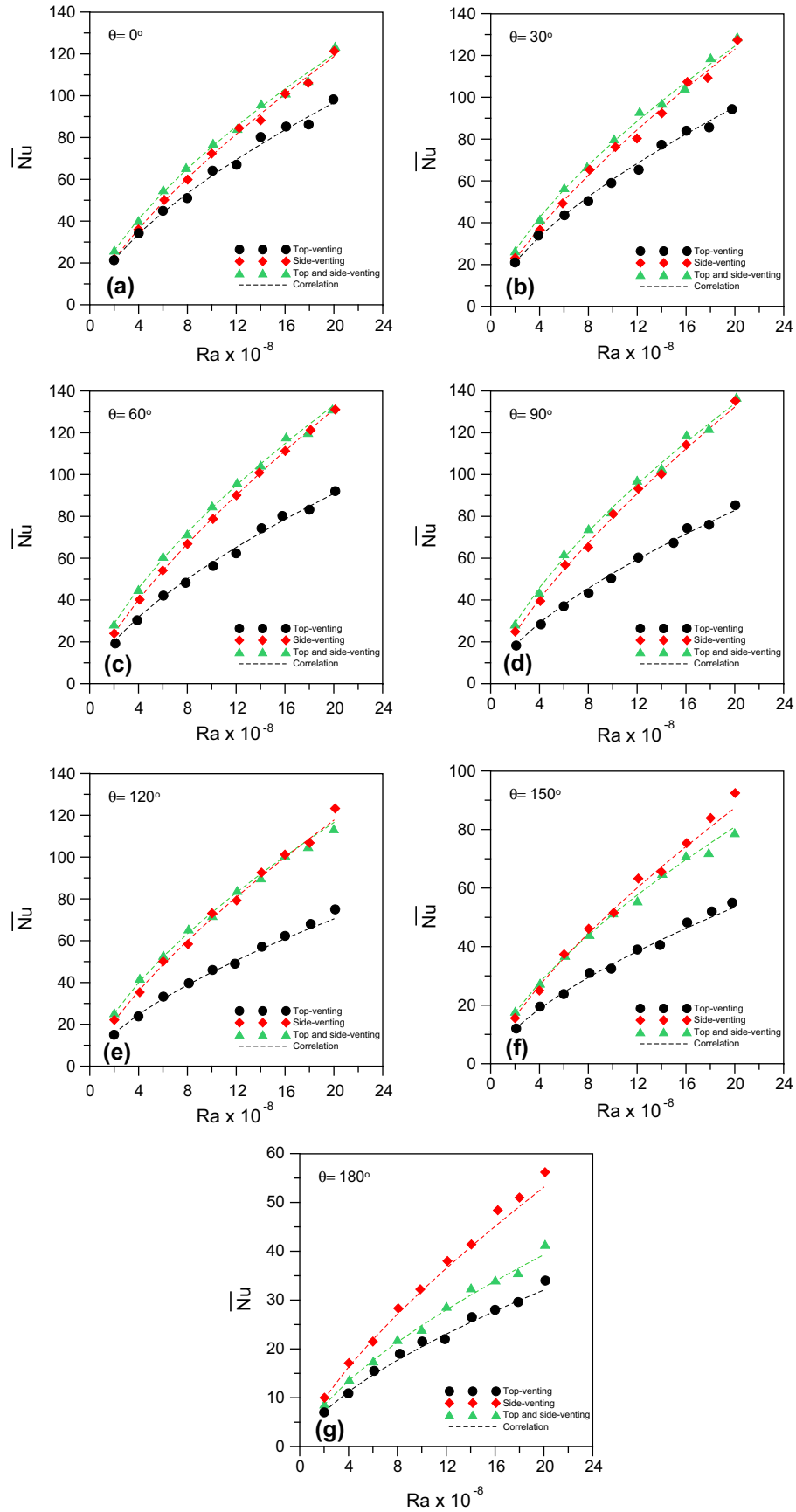


Fig. 6. Effect of venting openings configurations on \overline{Nu} at $OR = 0.25$ and different θ and Ra .

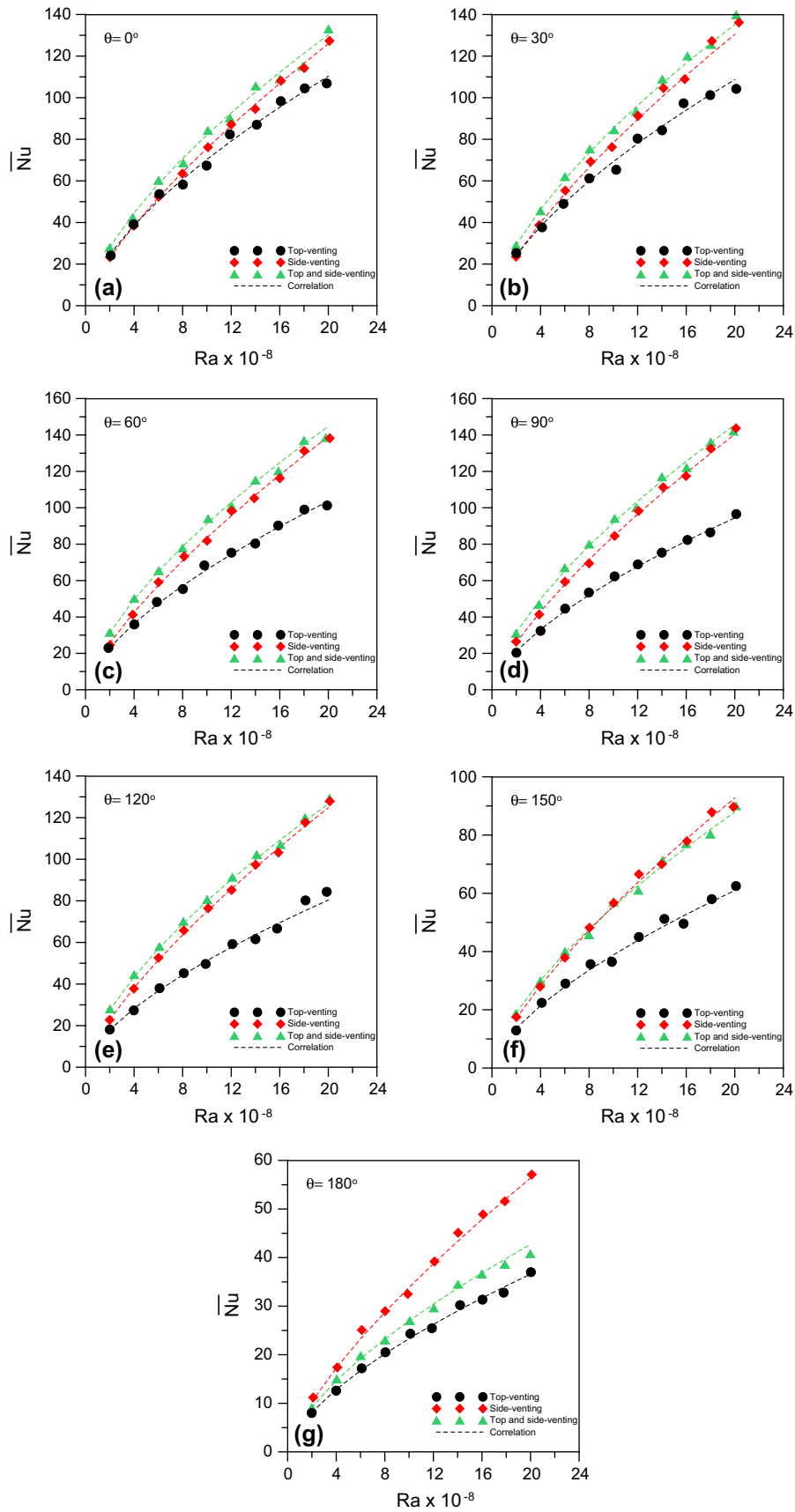


Fig. 7. Effect of venting openings configurations on \overline{Nu} at $OR = 0.5$ and different θ and Ra .

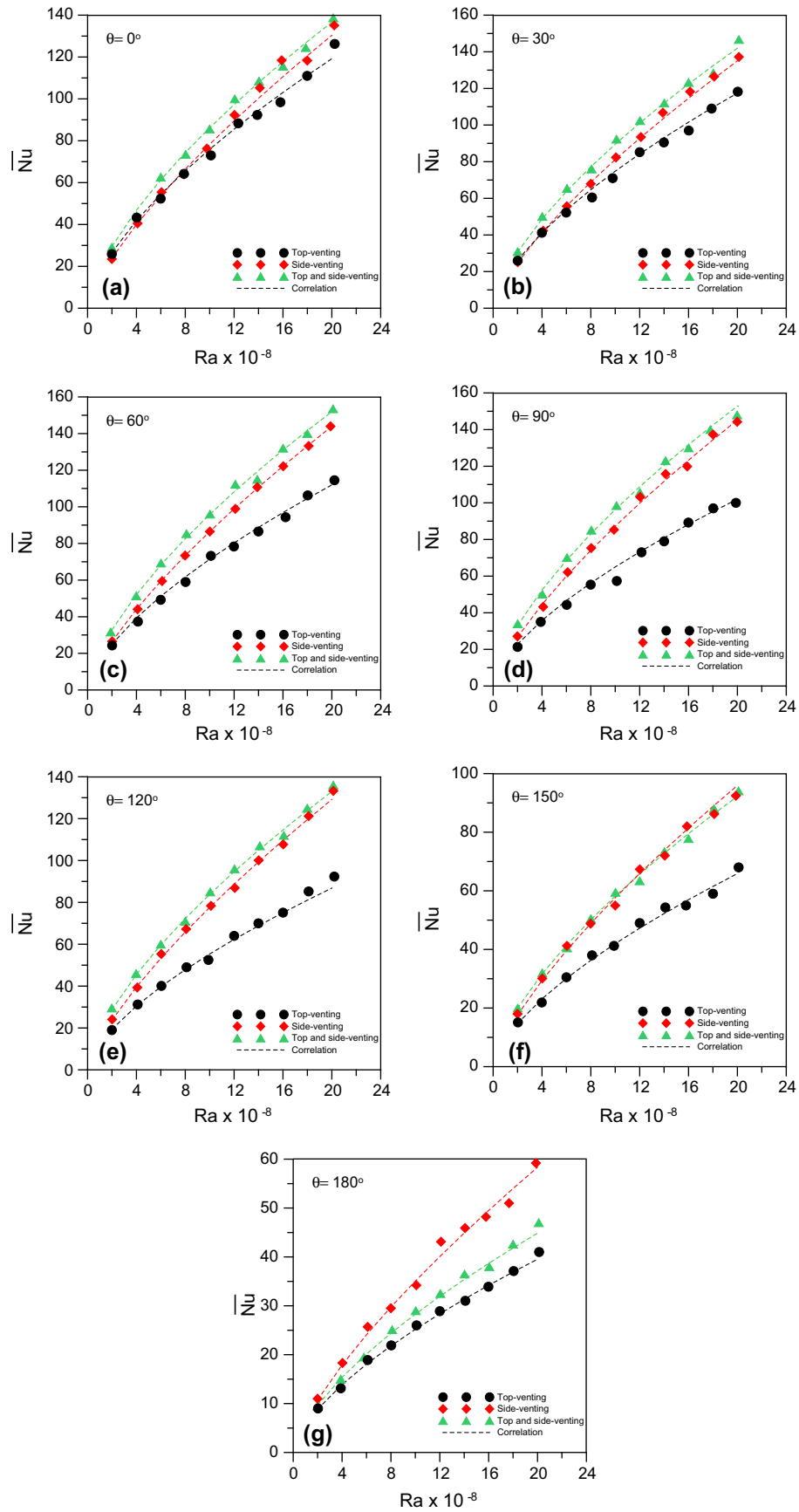


Fig. 8. Effect of venting openings configurations on \overline{Nu} at $OR = 0.75$ and different θ and Ra .

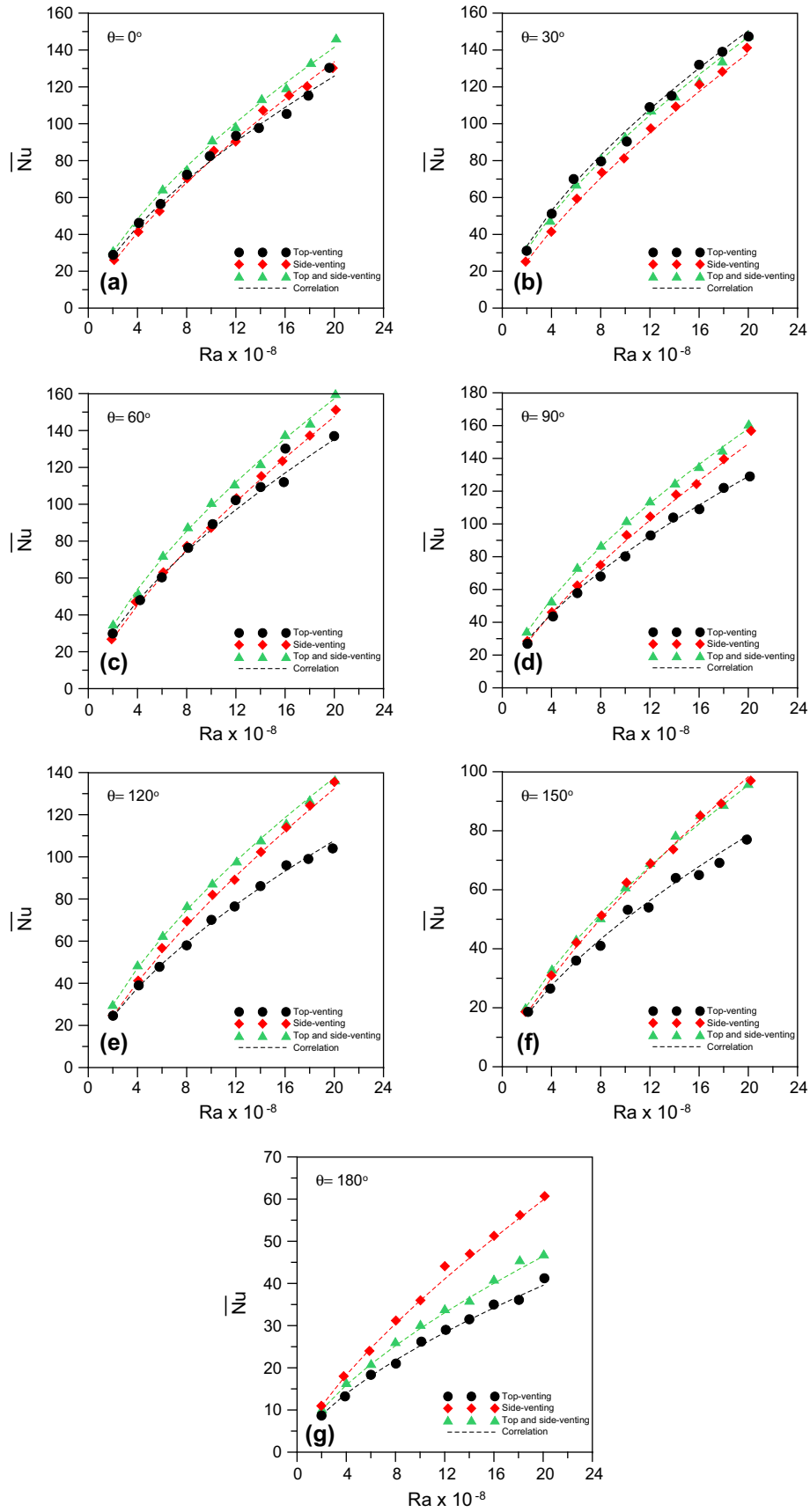


Fig. 9. Effect of venting openings configurations on \overline{Nu} at $OR = 1$ and different θ and Ra .

Table 1
Constants of Eq. (9).

Venting configurations	<i>m</i>	<i>n</i>	<i>C</i> ₁	<i>C</i> ₂	<i>C</i> ₃	<i>C</i> ₄
Top openings	0.65	0.19	3.75×10^{-5}	0	0.757	2.778
Side openings	0.735	0.085	8.7×10^{-6}	2.88	1.93	2.19
Top and side openings	0.665	0.12	3.022×10^{-5}	4.23	2.572	3.714

θ in the range $90^\circ < \theta < 180^\circ$, the problem comes to be like to free convection from inclined surface faces downwardly, and the heat transfer rate decreases with increasing θ (see Suryanarayana [28]). At $\theta = 180^\circ$, the surface becomes horizontal downward facing heated surface and the upward movements of the hot air to escape is obstructed by the top side and then the air moves horizontally to outflow from the openings of the side-openings.

To illustrate the effect of venting configurations, Figs. 2–5 were replotted in Figs. 6–9 with the venting configuration as a parameter. The figures show that \bar{Nu} for top and side openings

Table 2
Comparison between \bar{Nu} of present work and that of previous work at $Ra = 3.9 \times 10^8$.

θ	OR	W/H	\bar{Nu}	Reference
0	1	1	31	Chakroun et al. [14]
	1	1.25	36	Present
	0.5	1	32	Elsayed and Chakroun [15]
	0.25	1	25	
	0.5	1.25	31.78	Present
90	0.25	1.25	27.86	
	0.5	1	28	Elsayed and Chakroun [15]
	0.25	1	24	
	0.5	1.25	27.24	Present
	0.25	1.25	23.87	

configuration are always higher than those of top-openings and side-openings configurations. This results are valid for all opening ratios, at any enclosure inclination angles and at the studied range of Ra ($2 \times 10^8 < Ra < 20 \times 10^8$). This is because of the resistance of the top and side openings to the air circulation motion is smaller than those of top openings and side openings configurations. Moreover, the figures display that at small inclination angle the \bar{Nu} of top openings is slightly higher than that of side openings. Increasing θ , the \bar{Nu} of top openings decreases and that of side openings increases until the two values equalized at a certain θ in the range $0^\circ < \theta < 90^\circ$. Increasing θ beyond 90° , the \bar{Nu} for top openings configuration becomes smaller than that of side openings configuration. This also owing to the resistance of the air circulation in the cavity which is small for top openings configuration at small θ and high at high at bigger θ while the opposite is true for side openings configuration.

4. Empirical correlations and comparison with literature

Empirical correlation is developed from all the experimental data conducted for the entire ranges of the studied parameters. The correlation with less least square error was found to be in the form:

$$\bar{Nu} = C_1 Ra^m OR^n (1 - C_2 \cos^3 0.5\theta - C_3 \cos^2 0.5\theta - C_4 \cos 0.5\theta) \tag{9}$$

The constants m, n, C_1, C_2, C_3 and C_4 are given in Table 1 for the three venting configurations:

Comparison between the prediction of the correlation and the current experimental results are shown in Figs. 2–9. The deviation between the correlation estimation and the present experimental results are presented in Fig. 10. As seen in the figure, the developed correlation can predict the data of $7 \leq Nu \leq 80$ within $\pm 8\%$ error and the data of $80 < Nu \leq 161$ within $\pm 6\%$ error. Although the above correlation are developed for $W/H = 1.25$, it was possible however to use this correlation to predict average Nusselt number at different values of W/H by incorporating correlations of earlier work which relates Nusselt number with W/H .

The previous work found in the literatures were carried out at different boundary conditions of the enclosure walls, geometrical dimensions of the enclosure, venting configuration and ranges of Ra and θ than those of the present work. This make the comparison of the current work with those prior work is not a straight forward. However, it was possible to compare few cases of the present work with some previous works from the literature which were conducted at approximately similar conditions. Table 2 gives the comparison of average Nusselt number predicted by the current correlation with those obtained by previous work at $Ra = 3.9 \times 10^8$ at different values of W/H than that of the present study. The table shows that the prediction of the present correlation is close to the Nusselt numbers of the present study.

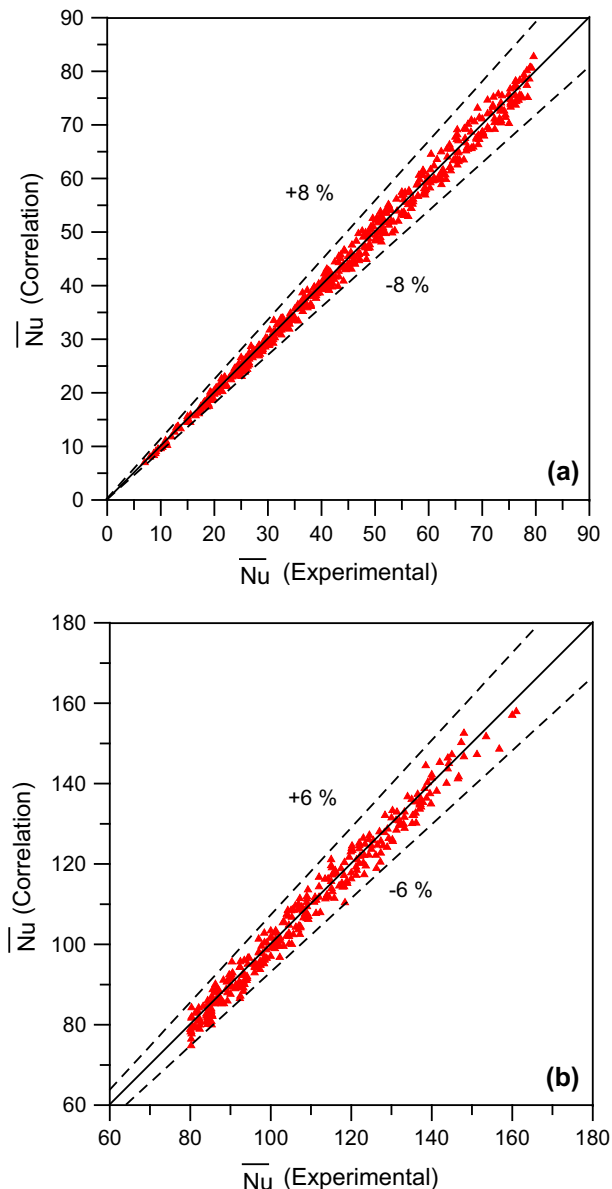


Fig. 10. Comparison between experimental data and correlation prediction.

5. Conclusions

Passive cooling of inclined rectangular enclosure has been studied under a wide ranges of Rayleigh number (2×10^8 – 2×10^9), inclination angle (0 – 180°), opening venting ratio (0.25 – 1), and venting surfaces configurations (top openings, side openings and top and side openings). Average Nusselt numbers were estimated from the experiments taking in account the conduction and radiation heat losses. The results showed that:

- For the different venting configurations, \overline{Nu} increases with increasing Rayleigh number and the venting opening ratio.
- For top openings configurations, \overline{Nu} decreases with increasing enclosure inclination angle. For side and side and top openings configurations, \overline{Nu} does not vary with θ until it reach 90° then it decreases with increasing θ .
- For any Rayleigh number, opening ratio and enclosure inclination angle, \overline{Nu} of top and side openings configurations is higher those of top and sides openings configuration.

General empirical correlation was developed to present the average Nusselt number in terms of Rayleigh number, opening ratio, inclination angle and venting configurations. The correlation prediction was compared with the current and earlier results and good agreements was obtained.

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