

# Experimental Investigation of Natural Convection Heat Transfer from Square Cross Section Cylinder in a Vented Enclosure

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## ABSTRACT

In the present work, the natural convection heat transfer from horizontal cylinder with square cross section situated in a square enclosure, vented symmetrically from the top and the bottom is investigated experimentally. The experimental work includes temperature measurements of the cylinder surface and the environment during transient state to determine Nu for the unbound and the bound cylinder. The studied variable ranges were:  $10^7 \leq Ra \leq 6.6 \times 10^7$ ,  $2 \leq W/D \leq 4$ ,  $0.25 \leq O/D \leq 4$ . The results indicated an increase in Nu with increasing Ra. Furthermore, Nu increased proportionally with the vent opening size at low enclosure widths. It showed inverse proportionality with opening size for high enclosure widths. The maximum percentage of the enhancement is more than 20% for bounded square cylinder as compared with unbounded square cylinder.

**Keywords:** Heat Transfer, Square Cylinder, Enclosure, Experimental

## الملخص

يتناول البحث دراسة عملية لانتقال الحرارة بالحمل الطبيعي من اسطوانة افقية ذات مقطع مربع موضوعة في تجويف مهوى بشكل متناظر من الاسفل والاعلى. ويتضمن الشغل العملي قياس درجات الحرارة لسطح الاسطوانة والمحيط الخارجي خلال الحالة العابرة لحساب عدد نسلت للاسطوانة الحرة والمحاطة بغلاف. مدى الدراسة لعدد رايلي يتراوح بين  $10^7$  و  $6.6 \times 10^7$  ونسبة عرض الغلاف الى قطر الاسطوانة يتراوح بين 2 و 4، بينما نسبة فتحة الغلاف الى قطر الاسطوانة فيتراوح بين 0.25 و 4. ويتبين من النتائج ان عدد نسلت يزداد بزيادة عدد رايلي. بينما يتناسب عدد نسلت بشكل طردي مع حجم فتحة التهوية عند عرض التجويف ذات الحجم القليل. ويتناسب عدد نسلت بشكل عكسي مع حجم فتحة التهوية عند عرض التجويف ذات الحجم الكبير.

## NOMENCLATURE

Symbol	Definition	Unit
$\bar{Nu}$	Average Nusselt number, $(h.D/k)$ .	
D	Cylinder diameter.	M
h	Convective heat transfer coefficient.	$W/m^2 \cdot ^\circ C$
K	Thermal conductivity of the air.	$W/m \cdot ^\circ C$
O	Enclosure opening size.	M
H	Characteristic length for noncircular cylinder.	M
$Pr$	Prandtl number, $(\nu/\alpha)$ .	
Ra	Raylieh number, $(g\beta\Delta TD^3/\nu\alpha)$ .	
$Ra_H$	Raylieh number based on H, $(g\beta\Delta TH^3/\nu\alpha)$ .	
t	Time.	Seconds
T	Temperature.	$^\circ C$
W	Enclosure Width.	Cm
<b>Greek Symbols</b>		
$\Delta T$	Difference between cylinder surface temperature and environmental temperature.	$^\circ C$
$\mu$	Viscosity of the air.	kg/m.s
$\dot{\beta}$	Coefficient of thermal expansion.	$1/^\circ C$
$\nu$	Kinematic viscosity.	$m^2/s$
$\theta$	Dimensionless temperature.	
<b>Subscript</b>		

S	Cylinder surface.	
$\infty$	Environment.	
D	Circular cylinder diameter.	

## INTRODUCTION

Natural convection heat transfer from relatively long horizontal cylinders of noncircular section has received only limited attention in the literature even though there are a number of practical applications where such heat transfer needs to be predicted. Furthermore, there is even less information relating to the natural convection heat transfer for the case of cylinders enclosed in vented enclosures. The natural convection heat transfer from unbounded horizontal cylinder has been of interest to several investigators because of its importance in technology. The earliest measurements were those of Nusselt [1]. McAdams [2], and Churchill and Chu [3] gave a correlation for Nu in laminar and turbulent regions. Kuehn and Goldstein, [4] presented a correlating equation for the natural convection heat transfer from horizontal circular cylinders to a cylindrical enclosure using the conduction layer model. The results approached the correlation for heat transfer from a free horizontal cylinder as the outer cylinder diameter became infinite and approached quasi steady as the inner cylinder diameter approached zero.

Hollands et al. [5] experimentally investigated the natural convection heat transfer from long isothermal horizontal cylinders of noncircular cross sections at various orientations, covering the Ra range from  $10^3$  to about  $10^9$ . The resulting correlation only requires the geometric specifications of the body height and perimeter with agreement with previous data of about  $\pm 10$  percent.

When it comes to enclosed cylinders, the paucity of information is surprising considering the many applications of such configuration. Sparrow and Pfeil [6] performed a comprehensive study to determine the natural convection heat transfer from heated cylinder situated in a vertical channel in air. The height ranging from 5 to 15 times the cylinder diameters, inter wall spacing ranging from 1.5 to 10 diameters and Ra ranging from  $1.5 \times 10^4$  to  $2 \times 10^5$ . It was found that a cylinder situated in a vertical channel experiences enhanced heat transfer as compared with an unshrouded cylinder. Enhancements of up to 40% were encountered for the investigated conditions. Farouk et al. [7] experimentally studied natural convection heat transfer from a horizontal isothermal cylinder placed between two vertical adiabatic walls. The experiment covered  $2 \times 10^3 < Ra < 3 \times 10^5$  and wall spacing to cylinder diameter ratios of 1.5 to 12. The enhancement of heat transfer from the cylinder decreased with increased spacing between the walls for the entire range of Ra with maximum enhancement of 45 percent obtained at the smallest Ra and W/D studied.

Al-Mahroom, [8] experimentally investigated the natural convection heat transfer from a horizontal circular cylinder placed in a vented square enclosure with symmetrical openings at the bottom and the top of the enclosure. The study included temperature measurements to obtain Nu and flow visualization to display the flow patterns. The study included the effect of enclosure width (W) and opening size (O). A maximum enhancement of 25% in Nu was reported at  $Ra=10^5$  and minimum values of W/D and O/D of 1.5 and 0.5 respectively.

There are some numerical investigations deals with natural convection heat transfer from cylinders in closed enclosure. Kim, et. al.[9] investigated numerically the characteristics of two dimensional natural convection problems in a cooled square enclosure with a heated circular cylinder. The finite volume method was used to simulate the flow and heat transfer over an inner circular cylinder. A detailed analysis for the distribution of streamlines, isothermals and Nusselt numbers were carried out to investigate the effect of the locations of the heated inner cylinder on the fluid flow and heat transfer in the cooled square enclosure for different Rayleigh number in the rang  $10^3 \leq Ra \leq 10^6$ .

Habeeb S. J.[10] study the effect of hot square cylinder placed on a cooled elliptical enclosure of a laminar natural convection numerically. The governing equations in the vorticity-stream

function formulation which were transformed into body fitted coordinate system. An elliptical function is used, which makes the coordinate transformation from the physical domain to the computational domain be set up by an analytical expression. About 48 tests are performed for different ratios of the geometry and Rayleigh number range from 103 to 106. The results showed that, the increase of the major axis of the enclosure leads to increase the average Nusselt number and decrease the flow strength for all Rayleigh numbers.

The present work deals with experimental investigation of natural convection heat transfer from a horizontal cylinder of square cross section situated in a vented square enclosure. The vented enclosure shall have two symmetrical openings on its upper and lower sides. The experimental work includes temperature measurements of the cylinder surface and its environment to determine the Nu from the unbounded cylinder and cylinder situated in a vented enclosure. The effect of Ra, enclosure width and opening size on Nu shall be evaluated.

## EXPERIMENTAL WORK

### Theory of the Experiment:-

Implicit in the transient cooling method adopted in this work was the assumption that the temperature decay is slow enough that the measured results are close enough to the steady state condition. Hollands et al. [5] checked this assumption, the results agreed with the transient results, well within experimental error.

In the present work, each run included heating the cylinder to the specified temperature then shutting off the heater and allowing it to freely cool down while recording the data over the designed intervals. This allows the coverage of a wide range of Ra over a relatively short time, approximately three hours for each experiment. The cylinder was assumed as isothermal (constant temperature). Assuming a lumped capacitance model, (Biot number = 0.002 on the average for this experiment), as well as: negligible radiation and end heat losses, the heat transfer model reduces to:

$$\frac{(T - T_{\infty})}{(T_i - T_{\infty})} = e^{-\frac{hA}{\rho c V} t} \quad \dots (1)$$

Where  $T_i$  is the initial temperature of the cylinder at the beginning of the cooling state in °C,  $T$  is the temperature at each time step in °C,  $T_{\infty}$  is the environment air temperature at each time step in °C,  $\rho$  is the density of the material in kg/m<sup>3</sup>,  $c$  is the specific heat of the material in kJ/kg°k,  $V$  is the volume of the cylinder in m<sup>3</sup>, and  $A$  is the surface area of the cylinder in m<sup>2</sup>.

Once the heat transfer coefficient is found, Nu and Ra are calculated as follows:

$$Nu_D = \frac{hD}{k}, \quad Ra_D = \frac{g\beta\Delta T D^3}{\nu^2} Pr, \quad \text{for the circular cylinder and}$$

$$Nu_H = \frac{hH}{k}, \quad Ra_H = \frac{g\beta\Delta T H^3}{\nu^2} Pr, \quad \text{for the square cylinder.}$$

In the above,  $D$  is the circular cylinder diameter,  $H = (4z_f \bar{P}^2)^{1/3}$  is characteristic length of non circular cylinders, and the air properties are calculated at the film temperature  $T_f$ .  $P$  and  $Z_f$  in the equation for  $H$  are the cylinder perimeter and the largest vertical height of the square section, Hollands et al. [5].

### The Experimental Setup

Two cylinders of different cross sectional shapes (the circle, and the square) were used as shown in figure (1). Each model (or cylinder), as shown in figure (2), was nominally 590 mm long including the length of its two ends. The circular cylinder, has a diameter of 50 mm and the square

cylinder has a side length of 50 mm. Each model was machined from aluminum with a 20 mm diameter longitudinal hole drilled along its full length to accommodate the electrical heating element. The heater assembly is enclosed in a 7 mm diameter brass tube and centrally located within the test specimen using fine-grade sand fill. The cylinders were highly polished to minimize radiation losses. End plugs were fabricated of mica and attached to each end of the cylinder to centrally hold the heating elements. Additional caps, made of cork were used to further control the heat losses from the ends.

The enclosure was constructed of Perspex sheets to form a box of 50 cm inside length with variable dimensions of the square cross section. Widths of 20, 12.5, 10 cm can be achieved as shown in figure (3). Four moveable black Perspex sheets of 3 mm thickness situated at the bottom and the top of the enclosure in order to control the size of the opening. Copper-Constantan thermocouples were fixed at the center of each Perspex sheet to measure the temperature of the enclosure. The entire setup was housed within a rigid frame made of brass supports, of dimensions 0.5 m by 0.5 m by 0.8 m. The cylinder was situated at the center of the enclosure in a position (0.75m) above the floor. The whole rig was placed in a closed isolated room. A schematic configuration of the enclosure is shown in figure (3). Twelve copper-constantan thermocouples were embedded in putty made of Aluminum papers (4 mm below the surface) in each body at locations on opposite faces 15 mm, 170 mm, 330 mm, and 485 mm from one end, figure (3). Measurements confirmed the uniformity of the surface temperature distribution of the cylinder. The thermocouples were placed at various locations axially and circumferentially along the cylinder. Six copper-constantan thermocouples were distributed equally at location 0.5 m above and beneath the cylinder and in axial direction along the cylinder to measure the temperature of the environment air. A Comark type display unit was used for temperature measurements in conjunction with a 12-channel selector switch.

Preliminary tests were conducted before running the main experimental works. These experiments included the calibration of the measuring devices, air stratification, uniformity of surface temperature, and radiation and convection heat losses. The temperature-measuring device was calibrated using mercury in glass thermometer. The difference between the measurements of the thermometer and the thermocouple were correlated and corrected. The temperature gradient along the height of the test rig, externally, was about 0.25°C/m that ensures negligibility of the air stratification. The maximum difference between temperature measurements along the cylinder surface was less than 2% of maximum temperature in longitudinal and circumferential directions. The value of heat losses, as compared with convection heat transfer from the cylinder, was about 2-3% for radiation and 2.5% for conduction end losses.

Uncertainty analysis was performed to establish the reliability of the results. The analysis method is based on a careful specification of the sources of uncertainties in the various primary experimental measurements. In the present work, the final result is Nu which is a function of the independent variables  $T, T_s, T_\infty, k, D, t, \rho, \dots$ , etc. thus,

$$Nu_D = Nu_D(T, T_s, T_\infty, k, \rho, D, t) \quad \dots (3)$$

Let  $w_{Nu}$  be the uncertainty in Nu and  $w_T, w_{T_s}, w_{T_\infty}, w_t, w_D, w_\rho, w_k$  be the uncertainties in the independent variables.

Assume that the uncertainties of the density, thermal conductivity, and specific heat are negligible and  $w_T = w_{T_s} = w_{T_\infty}$ . Then the uncertainty of Nu can be calculated from, [13]:

$$\omega_{Nu_D} = \left\{ \left( \frac{\partial Nu_D}{\partial D} \omega_D \right)^2 + \left( \frac{\partial Nu_D}{\partial t} \omega_t \right)^2 + \left( \frac{\partial Nu_D}{\partial T} \omega_T \right)^2 + \left( \frac{\partial Nu_D}{\partial T_s} \omega_T \right)^2 + \left( \frac{\partial Nu_D}{\partial T_\infty} \omega_T \right)^2 \right\} \quad \dots (4)$$

Assume  $\omega_T \approx \omega_{T_s} \approx \omega_{T_\infty} \approx 1.^\circ C$ , including cold junction uncertainty,  $w_t=1$  second/total time of the cooling (in seconds), and  $w_D=10^{-6}$  m. The variation of uncertainties of square cylinder are 0.5% as minimum and 2% as maximum.

## RESULTS AND DISCUSSION

The conducted experiments covered the following ranges of variables:

$$10^7 \leq Ra \leq 6.6 \times 10^7$$

$$2 \leq W/D \leq 4$$

$$0.25 \leq O/D \leq 4$$

The experimental work covered the natural convection heat transfer from unbounded (free) cylinder (circular and square cross sections) as well, for comparison purposes. Good agreement with Holland's data, [5], for square cylinder was indicated as well with maximum deviations of about 8%, as shown in figure (4). The comparisons between the present work and those of Sparrow [7], Al-Mahroom [8] and Kuehn [9] for the circular cylinder, shown in figure (5), indicated excellent agreement with a maximum deviation of 3.5%. The effects of the Ra, enclosure width, and enclosure opening size on Nu for all cylinder cross sections are discussed in the following paragraphs:-

- a- **Rayleigh number effects:** The effects of Ra on Nu are displayed in figures (6-7). Nu increases with increasing Ra for all enclosure widths and opening sizes. The reason of this behavior is inherent in the definition of Ra. Ra is the ratio between buoyancy and viscous forces; therefore, the increase in Ra increases the fluid velocity around the cylinder, hence the heat transfer and Nu. It is noted that the Nu increased by 15% when Ra value is increased with same enclosure width and opening size that display the noticeable effect of Ra on Nu. However, the ratio is nonlinear at 5 to 100. The correlation between Nu and Ra are exponential with exponent value of 0.23.
- b- **Enclosure Width Effects:** the effects of changing the enclosure width on Nu are presented in figure 6. Nu variation with enclosure width depended on enclosure opening sizes. For all Ra, higher Nu was obtained for the enclosed cylinder in comparison with the free cylinder which ensures the importance of situating the cylinder in the vented enclosure. The results also indicated a strong dependence of Nu on the enclosure geometric dimensions. These effects are highlighted by comparing the enclosed cylinder Nu results for different enclosure widths with those of the free cylinder. As shown in figure (6), Nu varies proportionally with enclosure width at  $O=2.5\text{cm}$  and  $O/D=0.5$  especially at high Ra. The strength of these effects decrease at the low Ra end of the results. Reduced effects are seen for larger opening sizes due to the reduced jet velocities impinging on the cylinders. For the full opening case, Nu decreases with increasing the enclosure width, again due to moving away from confined, guided flow to free unbound convection. The maximum Nu number increase for the square cylinder occurs at  $W/D=20\text{cm}$ ,  $O=2.5$  with a value 13.5%.
- c- **Opening Size Effects:** the reduction in the opening size increases the inlet jet velocity of the air which led to an increase in Nusselt number. At high enclosure widths, the Nusselt number is inversely proportional with enclosure opening size. A maximum Nusselt number is obtained at  $W=20\text{cm}$ ,  $O=2.5\text{cm}$  and  $Ra \approx 5.6 \times 10^7$  with value 20%, figure (7). The influence of opening size reduces with increasing its value to reach Nusselt numbers these values from unbounded (free) square cylinder. The behavior of the Nusselt numbers from square cylinder situated in low enclosure width is different, the Nusselt number increase is small and the maximum value of the increase is about 4-4.5%.

Nusselt number results for free horizontal cylinders of different cross sections are correlated with Rayleigh number in the following manner:-

$$\begin{aligned} Nu &= 0.7929 Ra^{0.2106} && \text{For circular cylinder} \\ Nu &= 0.8135 Ra^{0.2295} && \text{For square cylinder} \end{aligned} \quad \dots (5)$$

The constants are obtained from a least square fit using Microsoft Excel 2003.

Nusselt number results from bounded horizontal cylinders of square cross section is correlated with Rayleigh number, enclosure width, and opening size. The correlation equation is

$$Nu = 1.1184 Ra^{0.21063} (W)^{0.07006} (O)^{-0.05289} \dots (6)$$

The constants are determined from the software program "Statistica" using Non-Linear Multiple regression analysis with forward stepwise method.

Figures (8) display the observed values, from the experimental work, versus the predicted values, from equation (5), for circular cylinder. Both the correlation coefficients and residual values indicate a good strength of the correlations with an average residual percentage of about 2%.

## CONCLUSIONS

Natural convection heat transfer from horizontal cylinder with square cross section shape in a square vented enclosure was investigated experimentally over a fairly wide range of Rayleigh number considering the effects of enclosure widths and opening sizes on Nu as well. The results yielded the following Nu-Ra correlation for unbounded cylinders:

$$Nu = 0.7929 Ra^{0.2106} \quad \text{For circular cylinder}$$

$$Nu = 0.8135 Ra^{0.2295} \quad \text{For square cylinder}$$

$$\text{For } 10^5 \leq Ra_H \leq 6.6 \times 10^7$$

while the results yielded the following Nu- Ra correlation for the given ranges of enclosure width, and opening size:

$$Nu = 1.1184 Ra^{0.21063} (W)^{0.07006} (O)^{-0.05289}$$

The ranges of the study are:

$$10^7 \leq Ra \leq 6.6 \times 10^7$$

$$2 \leq W/D \leq 4$$

$$0.5 \leq O/D \leq 4$$

The work indicated that enclosing the cylinder in the vented enclosure enhanced the heat transfer by 20% for the square shape. The Nu increased proportionally with enclosure width for low opening size. The heat transfer was enhanced by 20% for O=2.5cm and W=20cm as compared with the unbounded cylinder and 16% for O=2.5cm and W=10cm. The Nu is inversely proportional with enclosure width for full opening size. The enhancement is very poor because the W/D is relatively large. The Nu increased with decreasing opening size for large enclosure width. The heat transfer enhanced by 20% for O=2.5cm and W=20cm as compared with the unbounded cylinder and 11% for O=10cm and W=20cm.

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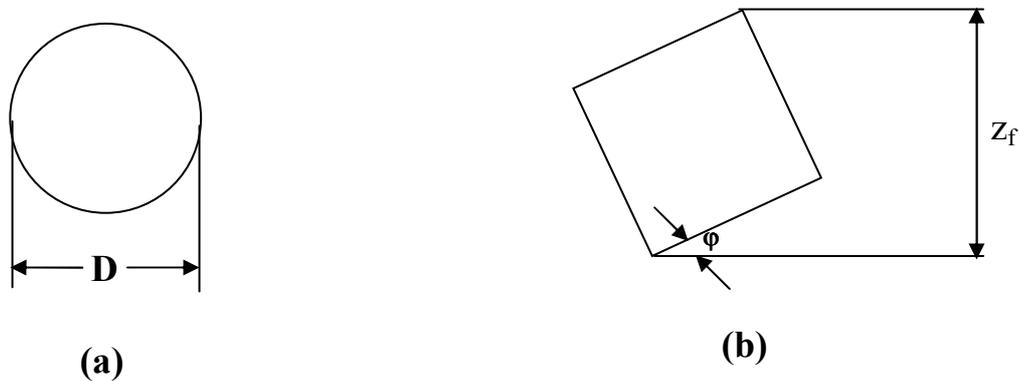


Figure (1) Different cross sectional shapes of the cylinders (a) circle, (b) square

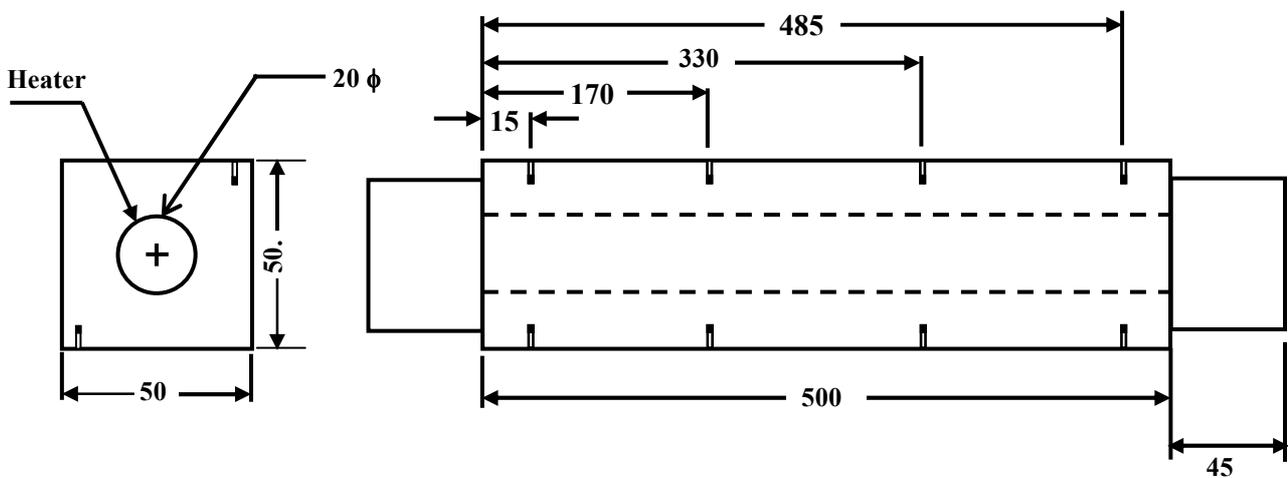


Figure (2) Drawing of square cross-section model showing holes for heater and thermocouple placements. (All dimensions in millimeters).

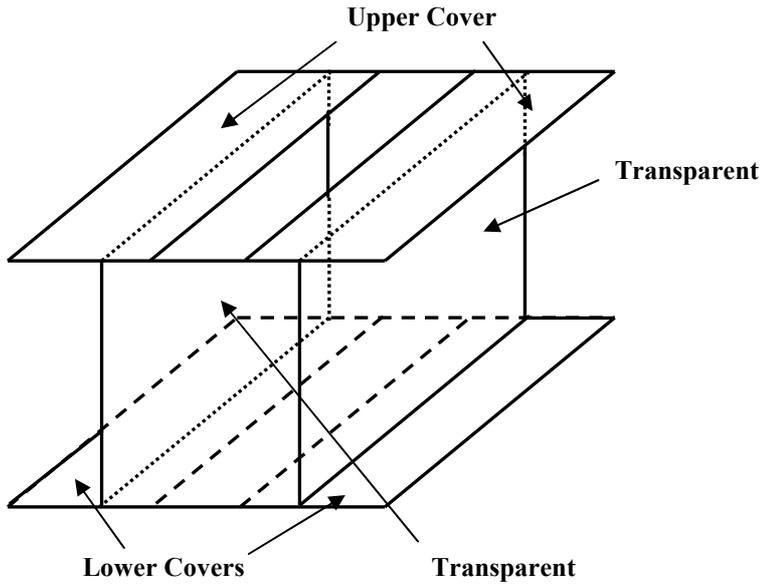
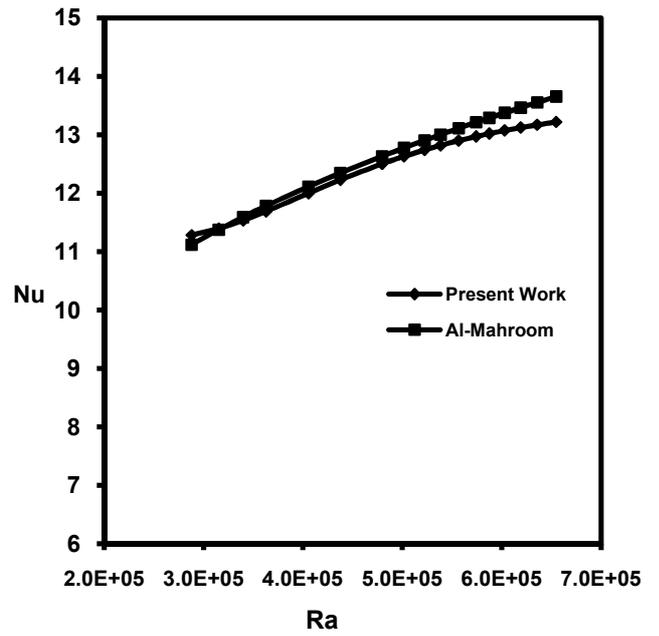


Figure (3) Schematic sketch of the enclosure.



(a)

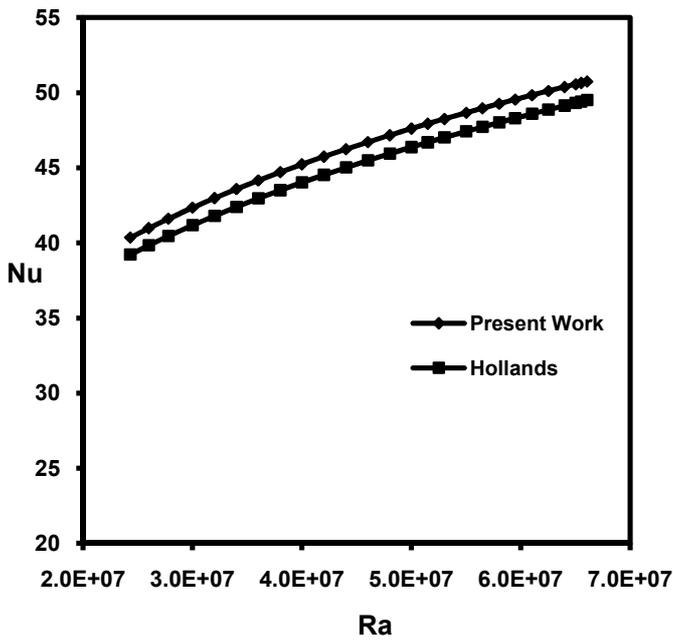
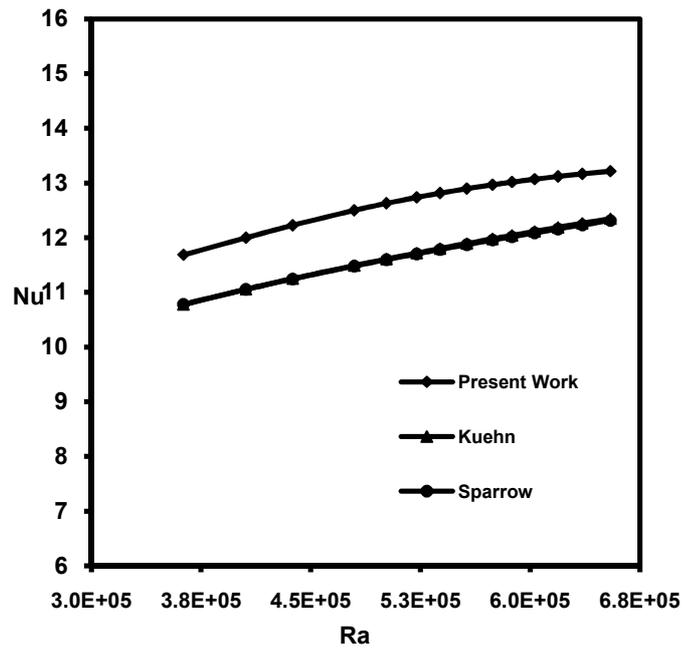


Figure (4) Comparisons of the Nusselt number results from free cylinder between present work and previous data [10] for square cylinder.



(b)

Figure (5) Comparisons of the Nusselt number results from free circular cylinder between present work and (a) Al-Mahroom [15] data, (b) Sparrow [12] and Kuehn [7] data.

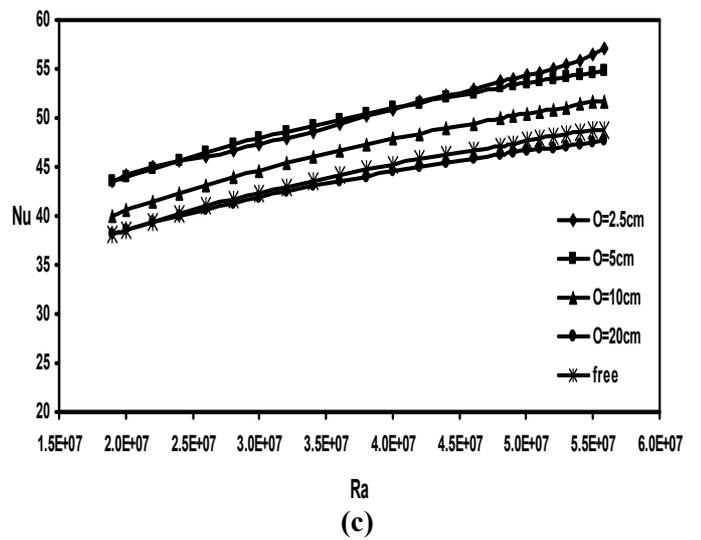
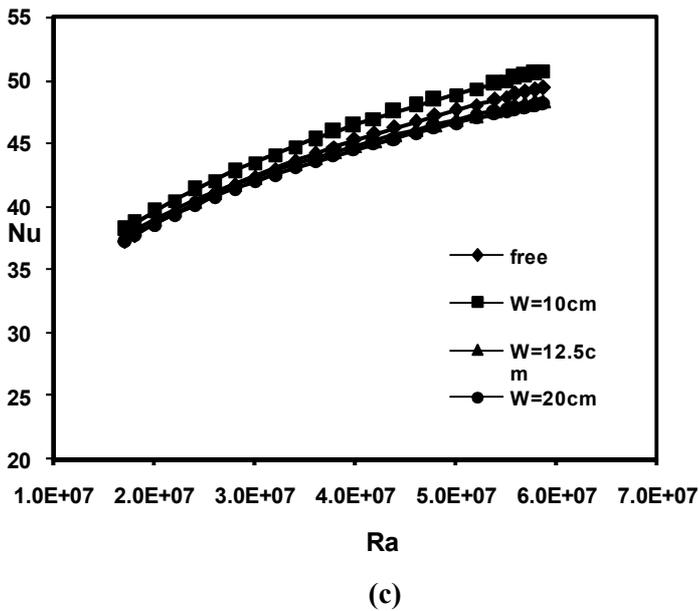
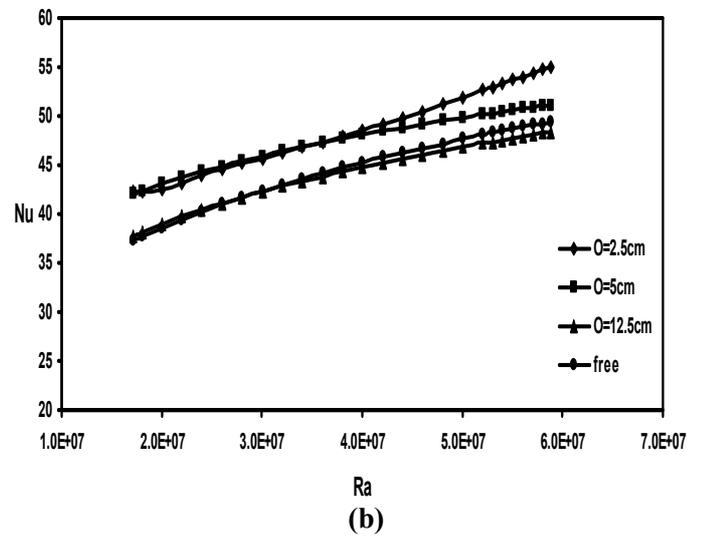
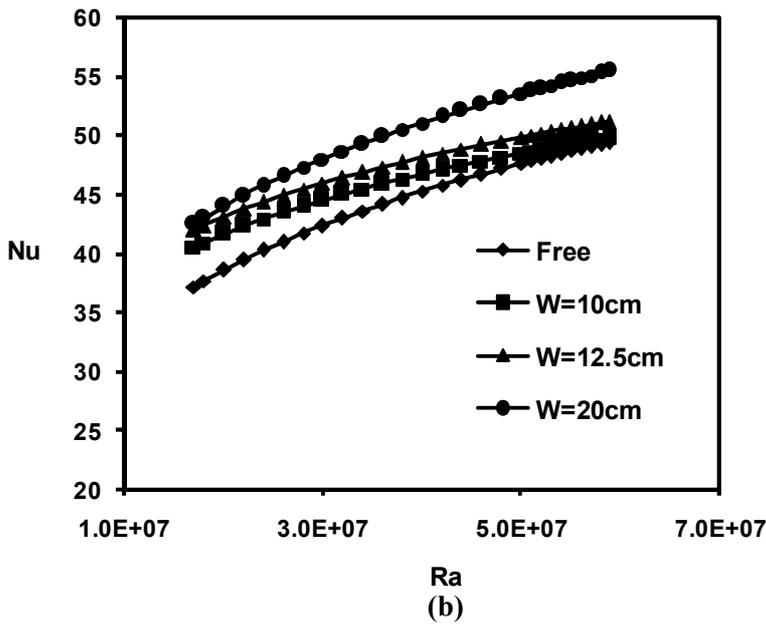
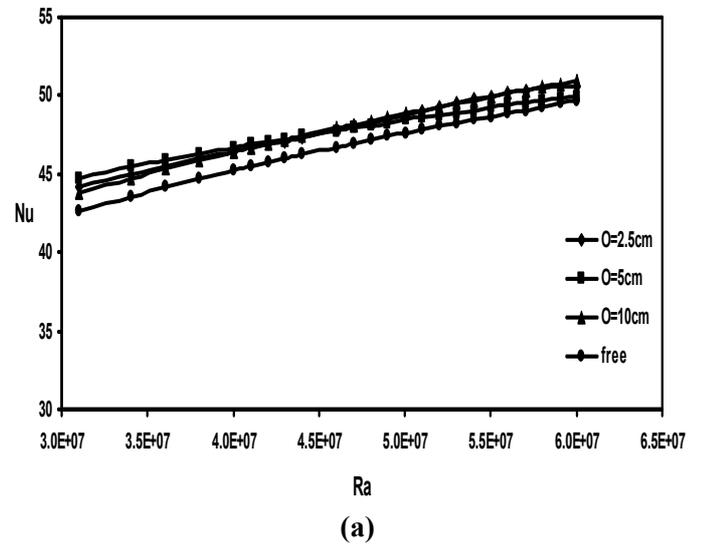
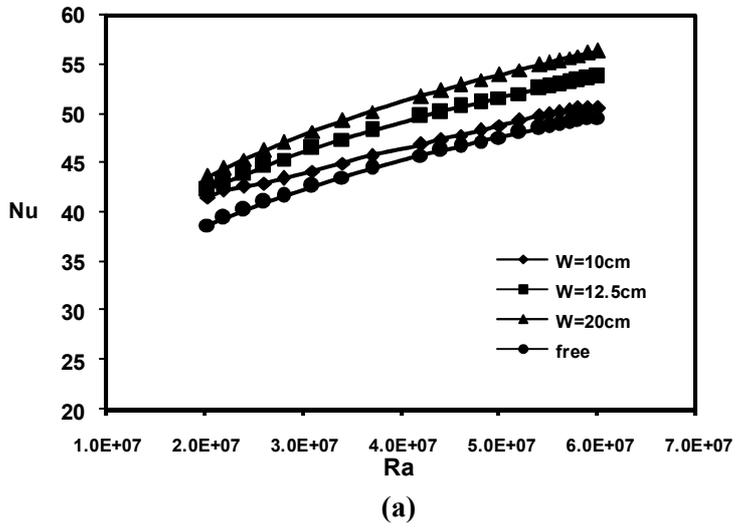
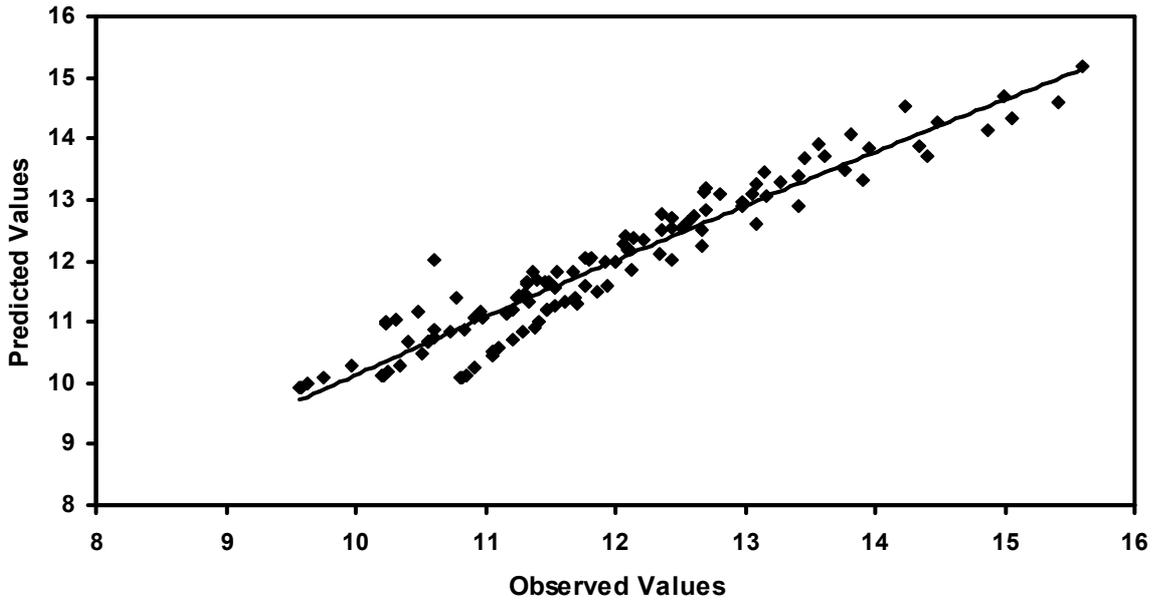
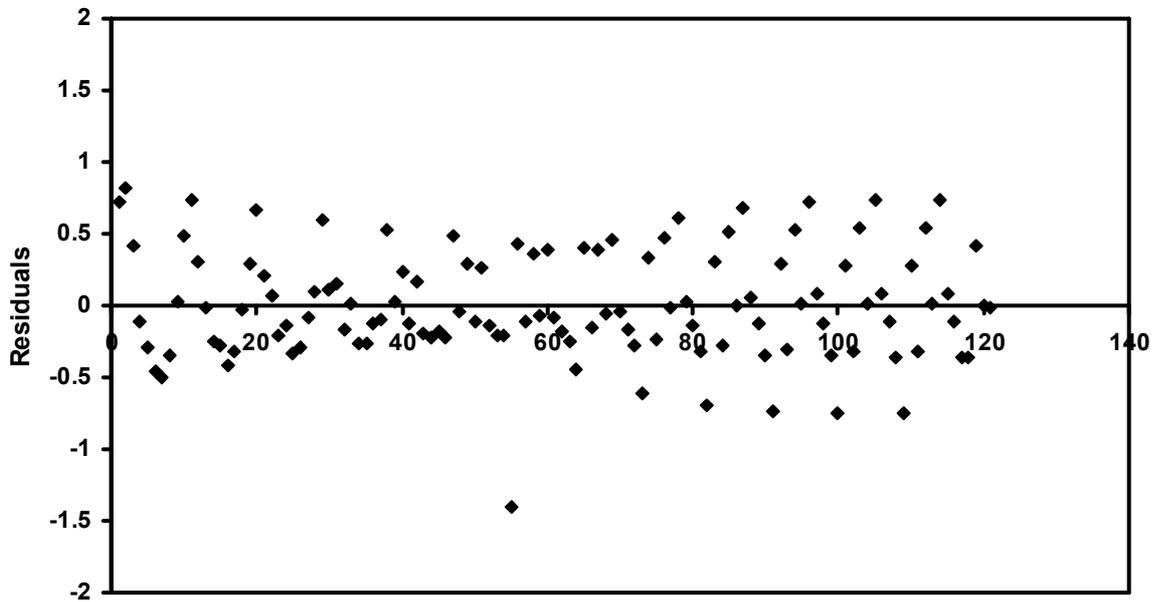


Figure (6) Nusselt number versus Rayleigh numbers for square cylinder at different enclosure width and opening sizes (a)  $O=2.5\text{cm}$ , (b)  $O=5\text{cm}$ , (c) full opening.

Figure (7) Nusselt number versus Rayleigh numbers for square cylinder at different opening sizes and enclosure width (a)  $W=10\text{cm}$ , (b)  $W=12.5\text{cm}$ , (c)  $W=20$ .



(a)



(b)

Figure (8) (a) Observed values from experimental results versus predicted values from equation (), (b) Residual of observed and predicted values, for circular cylinder.