# NATURAL CONVECTION IN A PARTIALLY OPENED BOX FILLED WITH A POROUS MEDIUM

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## ABSTRACT

This research studies the heat transfer in porous medium experimentally. Experimental investigation is carried out of free convection heat transfer for three dimensional in a box. The box filled with saturated porous medium, manufactured from glass with dimensions (30 × 30 × 30) cm. The porous media of plastic balls is used with homogeneous in diameter (11.7 mm). The lower wall of the box is heated by an electrical heater, while the other walls are thermally isolated. The effect of a porous medium on free convection heat transfer is studied for five values of heat fluxes (348, 576, 839, 1147, and 1384W/m$^2$).Rayleigh number is ranging between ( 1537.64 and 5260.62 ) and for average Nusselt number between ( 52.44 and 61.24). The results showed that the temperature inside the space increases as Rayleigh number increases. Also, the average Nusselt number increases with increase of Rayleigh number for the first three values of heat flux, but it decreases after heat flux because of domination of the conduction heat transfer on convection heat transfer. Also an empirical correlations were obtained in this study.

**KEYWORDS: Free convection, Porous media, Enclosure.**

# انتقال الحرارة بالحمل الحر في تجويف مفتوح جزئياً مملوء بوسط مسامي

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## الخلاصة

يدرس هذا البحث انتقال الحرارة في الاوساط المسامية عمليا. العمل التجريبي تم لانتقال الحرارة بالحمل الحر لصندوق مكعب مملوء بوسط مسامي مشبع بالهواء مصنوع من الزجاج بابعاد (30*30*30 سم) . وان الحشوة المسامية المستخدمة تتكون من كرات بلاستيكية متجانسة القطر بمقدار (11.7 ملم) حيث يكون الجدار الاسفل من الصندوق مسخن عن طريق مسخن كهربائي، بينما الجدران الاخرى معزولة حرارياً ، تم دراسة تأثير الوسط المسامي على انتقال الحرارة بالحمل الحر لخمسة قيم

مختارة للفيض الحراري وهي (348,576,839,1147,1384 واط/متر مربع) لمدى رقم رايلي (1537.64 الى 5260.62) ولمدى رقم نسلت (52.44 الى 61.24) وقد اظهرت النتائج العملية ان درجة الحرارة داخل الحيز تزداد بزيادة رقم رايلي . كما بينت ايضا ان معدل رقم نسلت يزداد مع زيادة رقم رايلي في القيم الثلاثة الاولى للفيض الحراري ، لكنه يتناقص بعد القيمة الرابعة للفيض الحراري وذلك بسبب سيطرة التوصيل الحراري على انتقال الحرارة بالحمل. وكذلك فقد تم في هذا البحث استنتاج علاقات ارتباطية (تجريبية).

## NOMENCLATURE

| | | |
|---|---|---|
| $D_h$ | Hydraulic diameter | m |
| $d_s$ | Diameter of solid beads | m |
| $g$ | Gravitational acceleration | m/s$^2$ |
| $H$ | Height of the box | m |
| $\bar{h}$ | The average heat transfer coefficient | W/m$^2$.$^o$C |
| $h_y$ | The local heat transfer coefficient | W/m$^2$.$^o$C |
| $k$ | Permeability | m$^2$ |
| $k_f$ | Thermal conductivity of the fluid | W/m.K |
| $k_m$ | Effective thermal conductivity of the porous medium | W/m.K |
| $k_s$ | Thermal conductivity of the spheres (beads) | W/m.K |
| $L$ | Length of the box | m |
| $m_S$ | weight of spheres. | kg |
| $v_{dis}$ | the volume of the water displaced | m$^3$ |
| $v_{total}$ | Box volume | m$^3$ |
| $v_{solid}$ | Plastic beads volume | m$^3$ |
| $q_w$ | Heat flux | W/m$^2$ |
| $ti$ | The local temperature | $^o$C |
| $t_S$ | The lower surface temperature | $^o$C |

## GREEK SYMBOLS

| | | |
|---|---|---|
| $\alpha_m$ | Thermal diffusivity of porous medium | m$^2$/s |
| ε | Porosity | |
| ρ | density | kg/m$^3$ |
| μ | Dynamic viscosity of fluid | kg/m.s |
| υ | Kinematic viscosity | m$^2$/s |

## SUBSCRIPTS

| | |
|---|---|
| $f$ | Fluid (air) |
| $i$ | Local |
| $m$ | Effective |
| $s$ | Solid |

$y$ $y$ – axis

**NON-DIMENSIONAL NUMBERS**

$Nu$ The local Nusselt number

$\overline{Nu}$ The average of Nusselt number

$Ra$ The Rayleigh Number

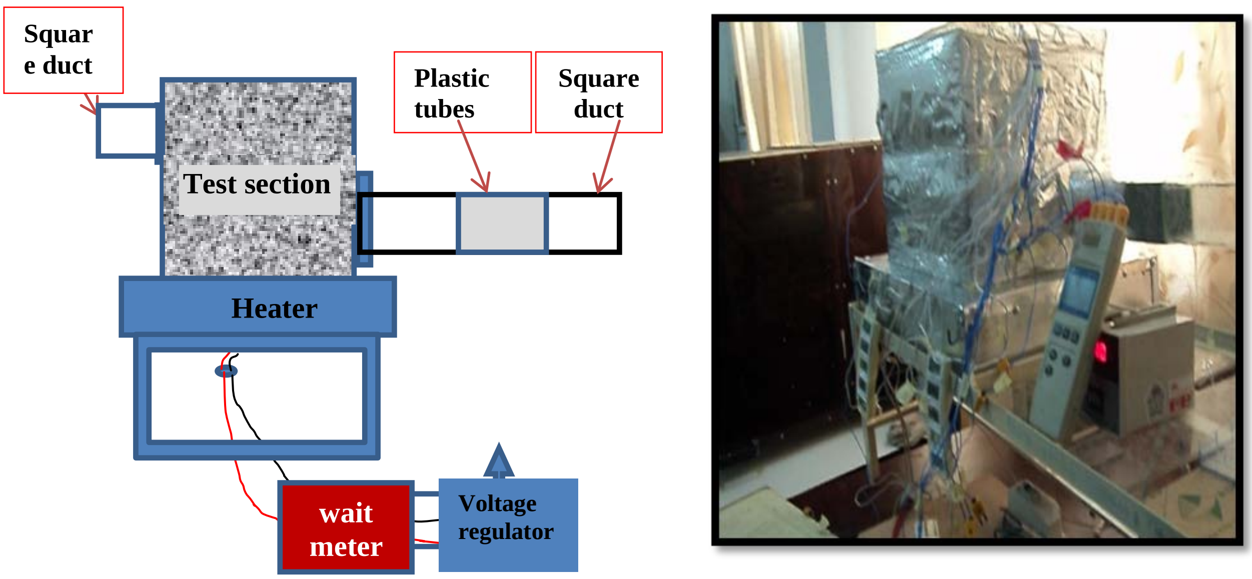
## 1. INTRODUCTION

The topic of heat transfer in porous media is one of the important field in the engineering study in nowadays. The theme of free convective heat transfer in square box filled with saturated porous media is important in many practical fields. For example, petroleum technology and mechanical and chemical engineering, civil engineering, environment engineering and the field of farming engineering. Heat transfer problems through porous media have become an intensive research topic for last few decades because of their possible applications in many members of science and technology. The most important research on this topic can be reviewed as following: **(Prasad, and Kulaki, 1984)** studied numerically the two-dimensional steady natural convection in a porous rectangular cavity heated from the side for different values of (width/height) ratio and Rayleigh numbers. The average Nusselt number was observed to be a maximum in a restricted range of aspect ratio (width/height), depending on the Rayleigh number ($Ra$*). They established a criteria in terms of aspect ratio and ($Ra$*) for the existence flow regimes **[1]. (Misirlioglu et al, 2005)** investigated numerically the steady-state free convection inside a cavity made of two horizontal straight walls and two vertical bent wavy walls and filled with a fluid-saturated porous medium.The wavy walls were assumed to follow a profile of cosine curve. The horizontal walls were kept adiabatic, while the bent-wavy walls were kept isothermal at different temperatures. He concluded the heat generated in the porous medium cannot be transferred through the porous medium from the right (hot) wall to the left (cold) wall **[2]. (Watit and Phadungsak, 2006)** conducted a study on the transient natural convection flow through a fluid-saturated porous medium in a square enclosure with a convection surface condition. The cavity was insulated except the top wall was partially exposed to an outside ambient. The exposed surface allows the convective transport through the porous medium, generating a thermal stratification and flow circulations. It was found that the heat transfer coefficient, Rayleigh number and Darcy number greatly influenced the characteristics of flow and heat transfer mechanisms. Furthermore, the flow pattern was found to have a local effect on the heat convection rate **[3]. (Hakan Oztop, 2006)** investigated numerically the natural convection heat transfer in a partially cooled and inclined rectangular enclosure filled with a saturated porous medium. One of the side walls had a constant hot temperature and one of the adjacent walls was partially cooled, while the remaining ones were adiabatic. He found that the inclination angle was the dominant parameter on the heat transfer and fluid flow as well as aspect ratio **[4]. (Tanmay et al, 2006)** investigated numerically the natural convection flow in a square cavity filled with a porous matrix. They used Darcy–Forchheimer model to simulate the momentum transfer in the porous medium. They concluded that the non-uniform heating exhibits greater heat transfer rates at the center of the bottom wall than that with the uniform heating case for all Rayleigh number regimes **[5]. (Saleh et al, 2010)** investigated the unsteady natural convection in a square region filled with a fluid-saturated porous medium having non-uniform internal heating and heated laterally. The heated wall surface temperature varied sinusoidally with the time about fixed mean temperature. The opposite cold wall was maintained at a constant temperature. The top and bottom horizontal walls were kept adiabatic. The flow field was modeled with the Darcy model and was numerically solved using a finite difference method. It was found that strong internal heating can generate significant maximum fluid temperatures above the heated wall. The location of the maximum fluid temperature moved with time according to the periodically changing heated wall temperature. The augmentation of the space-averaged temperature in the cavity is strongly depended on the heating amplitude and rather insensitive to the oscillating frequency **[6]. (Muyassar Ismaeel, 2011)** investigated the natural convection heat transfer in a square porous cavity with partial cooled vertical walls. The left vertical side wall was partially heated, and the right side wall was partially cooled. Depending on the positions of the hot and cold parts, nine cases were considered in his investigation. Flow and heat transfer characteristics for all cases were studied for a range of Rayleigh number ($50 \leq Ra \leq 500$). He solved the governing equations numerically with the aid of the finite difference technique and Gauss–Siedel method. The numerical results showed that there are significant changes in the flow and temperature fields and the rate of heat transfer due to the change of the positions of hot-cold parts. Also, he showed that the maximum heat transfer occurs for the Lower-Upper arrangement, while the minimum heat transfer occurs for the Upper-Lower arrangement. It was observed that the arrangements of active portions played an important role on flow, temperature fields and heat transfer **[7].**

## 2. EXPERIMENTAL APPARATUS AND DATA REDUCTION

### 2.1 TEST RIG DESCRIPTION

The test rig is designed and manufactured to fulfill the requirements of the test program for a free convection heat transfer. The experimental apparatus consist basically of the test section, the constant heat flux heater and the measuring devices. Most parts of the test rig are manufactured, and a care is take to prevent any air leakage during operation between the test rig parts connection **(Fig 1-A)** illustrates schematic diagram of experimental apparatus, and **(Fig 1-B)** shows the photograph of the test rig.




**(A)** **(B)**

**Figure (1):** (A) schematic diagram of experimental apparatus,(B) the photograph of the test rig.

## 2.2 THE TEST SECTION

The experimental model used in the present study is a square box filled with spherical plastic beads. The main dimensions of the box are; length (L=30 cm), width (W=30 cm) and height (H=30 cm) as shown in **(Fig. 2)**. All the walls of the box are constructed from a glass of thickness (6 mm). A glass wool insulation of (50 mm) thickness is applied to insulate all walls except the lower wall in order to reduce the heat loss. On the bottom wall of the square box a uniform heat flux is provided by means of electric tape heater. The partially opened opposite sides of the box are punctured with square holes (6×6) cm at the right lower and the left upper squares to the entry and exit air respectively. Then, a galvanized steel mesh is cited in order to prevent the beads from going out of the box. One hole was drilled in the center of the upper wall for entry (25) thermocouples in order to measure the temperatures inside the box. The thermocouples are arranged to form a tree the distribution in order to measure the temperature at different locations according to the grid distribution. The grids are distributed horizontally and vertically in order to take into account all the temperature variation in the model, as shown in **(Fig. 2)**. The square box is filled with plastic (polypropylene) beads saturated with air. The plastic beads constant diameter is (11.7 mm) and having the physical properties as shown at **[8]**. The square holes are existed into two square ducts (6 cm×6 cm) with (70 cm) and (15 cm) length. They are manufactured from limpid plastic of (4 mm) thickness. The box is stand at an iron frame to prevent any air leakage from corners, silicon is used.

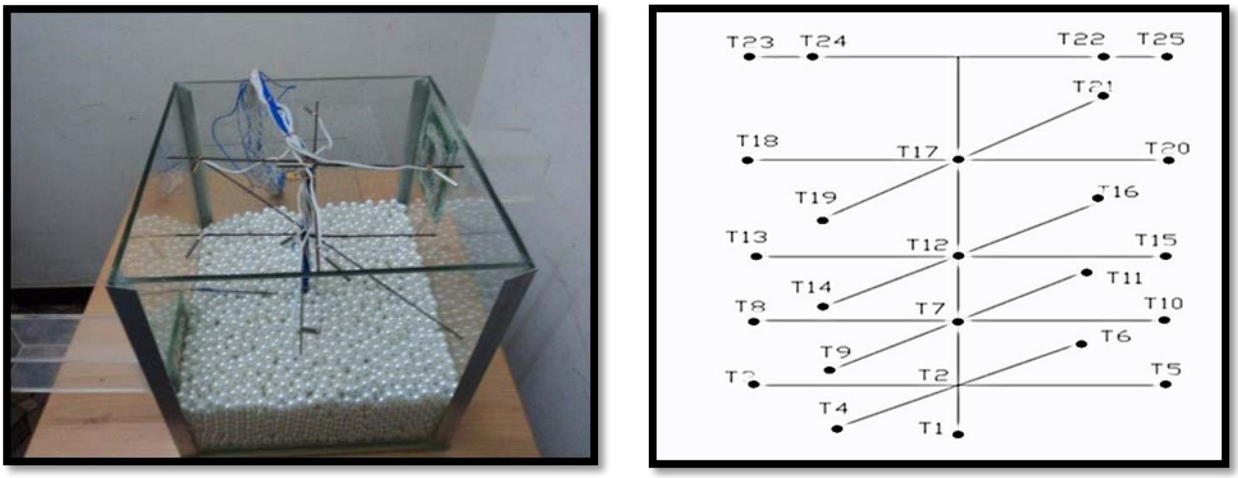



**Figure (2)** thermocouples distribution on grid.

## 3. LAYOUT AND MEASURED PARAMETERS

During the experimental investigation, the main parameters measured are:-

1. The surface temperature of the box.
2. The temperature distribution within the test section.
3. The heat flux (current and voltage)

Experiments were carried out to study the effect of influence of heat flux range, and to determine the temperature distribution in the porous media. These parameters can be summarized in **(Table 1)**.

## 4. EXPERIMENTAL PROCEDURES

Experiments were conducted to measure the temperatures and heat fluxes. The experimental work was done in a specially designed cubic box filled with a porous medium where the thermocouples distributed in three dimensions. The general steps were followed in this experimental investigation for free convection heat transfer is given below:

1. Arranging the required outlet heat flux from the heating system at the bottom of the test section is arranged manually by adjusting the supplied voltage accordance to the first value of the heat flux required which is (348W/m$^2$).

2. Waiting the steady state condition is reached about (180-270) minute.

3. Recording the temperatures distribution through the porous media within the enclosure using the (25) thermocouples at (0, 0.5, 9.5, 19, 28.5, 30, 30.6) cm in the y-direction till reaching the upper surface.

4. Repeating the experimental procedure step (1) to step (3) for all heat fluxes considered.

## 5. CALCULATIONS OF POROUS MEDIA PARAMETERS

### 5.1 POROSITY ($\varepsilon$)

The average porosity of the porous medium is compute for each particle with constant diameter of (11.7 mm). The porosity or void fraction of a porous medium is defined by **[9]:**

$$\varepsilon = \frac{v_{total} - v_{soilds}}{v_{total}} \tag{1}$$

The plastic beads volume is measured by multiply the volume of one bead by the total number of the plastic spheres, which gives $\varepsilon = 0.422$.

### 5.2 DENSITY ($\rho$)

$$\rho_s = \frac{m_s}{v_{dis}} \tag{2}$$

The density of the plastic spheres is found to be (989.16 kg/m$^3$).

### 5.3 PERMEABILITY (*k*)

The permeability depends on the porosity and plastic sphere diameter as given by **[10 & 11]**.

$$k = \frac{d_s^2 \ \varepsilon^3}{150\,(1-\varepsilon)^2} \tag{3}$$

The permeability of the plastic spheres is found to be ($2.0529*10^{-7}$ m²).

## 6. CALCULATION OF THE EFFECTIVE THERMAL CONDUCTIVITY ($k_m$)

An important property of the porous medium is the medium thermal conductivity ($k_m$). As reviewed in the previous investigations, **by [8, 10,** and **12]**, a typical mixing rule based on the volume fraction is usually being applieder.

$$k_m = \varepsilon k_f + (1-\varepsilon)k_s \qquad (4)$$

In which ($k_f$) calculated at the average working temperature of 60˚C and its values is ($28.65*10^{-3}$). The parameter ($k_s$) used in the present work is taken as given by **[8]**. The prediction values made by equation (4) is found to be accepted only if ($k_f \approx k_s$ ), i.e. the conductivity ratio, $\lambda = k_f / k_s \cong 1$.

Since the conductivity ratio is less than unity, ( $\lambda$=0.1155), the better choice is by using Krupiczka's correlation according to (Prasad 1989) Krupiczka's correlation based on a two dimensional heat transfer model that predicts more accurate values of ($k_m$ ) **[13].**Therefore correlation takes the form:

$$k_m = k_f * \lambda^{-n} \qquad (5)$$

where:

$$n = 0.28 - 0.757 \log_{10} \varepsilon + 0.057 \log_{10} \lambda \qquad (6)$$

Thus ($k_m$) is found to be (0.08617 W/m.K).

## 7. THE HEAT CAPACITY

The heat capacity of the porous medium is estimated by using the mixing rule based on the volume fraction such as:

$$(\rho\, c_P)_m = \varepsilon(\rho\, c_P)_f + (1-\varepsilon)(\rho c_P)_s \qquad (7)$$

Where:

$(\rho\, c_P)_m$ is the heat capacity of the porous medium.

$(\rho\, c_P)_f$ is the heat capacity of the air.

$(\rho\, c_P)_s$ is the heat capacity of plastic spheres.

The specific heat of the solid sphere is obtained from **[8]**, and it is equal to ($c_{ps}$ = 1.939 KJ/kg .K). The constant pressure specific heat of the air is obtained from the references according to the average working temperature of 60˚C, and it is equal to ($c_{pf}$ = 1.008 KJ/kg .K).

## 8. THE RAYLEIGH NUMBER (*Ra*):-

For the natural convection, Rayleigh number can be defined in equation (8) as following:

$$Ra = \frac{g\beta\ k\ q_w H^2}{k_m\ \ v_f\ \alpha_m} \tag{8}$$

*(β)* is the thermal coefficient of volume expansion and ($\alpha_m$) is the thermal diffusivity Where of the porous medium as given by **[13]**.In the following relation.

$$\alpha_m = \frac{k_m}{(\rho\ c_P)_m} \tag{9}$$

All the main parameters in the equation (8) are either calculated from the tables or from the previous calculations.

## 9. HYDRAULIC DIAMETER CALCULATION ($D_h$)

The hydraulic diameter for a medium composed of uniform spherical particles requires an appropriate definition of the hydraulic diameter as follows according to **[9]**.

$$D_h = \frac{2d_s\ \varepsilon}{3(1-\varepsilon)} \tag{10}$$

Where ($D_h$) is equal to (5.695 mm).

## 10. CALCULATION OF HEAT TRANSFER COEFFICIENT(h)

The local heat transfer coefficient by free convection can be calculated as:

$$h_y = \frac{q_w}{t_s - \ (t_i)_y} \tag{11}$$

The local Nusselt number ($Nu$) can be determined by: -

$$Nu = \frac{h_y\ \ D_h}{K_m} \tag{12}$$

Also the average of Nusselt number ($\overline{Nu}$) can be calculated as:

$$\overline{Nu} = \frac{1}{H}\int_{y=0}^{y=H} Nu\,.dy \tag{13}$$

The above integration can be done by using the trapezoidal rule as follows:

$$\int_a^b f(x).dy = \frac{h}{2}\left[f(y_o) + 2\sum_{i=1}^{n-1} f(y_i) + f(y_n)\right] \tag{14}$$

Sample calculation of is presented in **(Table 2)**.

## 11. RESULTS AND DISCUSSION

### 11.1 TEMPERATURE DISTRIBUTIONS

The steady state temperature distributions along the vertical direction for different *x* - *z* planes are illustrated. **(Figs.1.2.and3.)** show the temperature distributions (y). As it is seen from **(Figs. 1-A, B & C)**, the temperature distributions decreases along the *y*-direction indicate a linear temperature distribution pattern. Upon moving upward, the heat resistant increases causing a reduction in the temperature values. **(Fig. 1-A)** is gave similar behavior of **(Fig. 1-C)**, because (x-y) planes (*z* = 0 cm to *z* = 29 cm) lies in the entrance and exit vents planes, which cause a temperature difference. However, figure (1-B) for plane (*z* =14.5 cm) lies in the center of the box and does not lose a lot of temperature, since the vent is far enough from this plane. So, the temperature distributions according to (*x*) values are logical at (*x* = 29 cm), the temperature will be less, because this plane is near to upper exit vent and for all cases. The above mentioned is coincident with the contour drawing for planes (*x*-*y*-*T*) for the behavior of temperatures at each point for each plane. The ascending and descending convective flow within the box gives the curvilinear isothermal lines in these figures. However, the isothermal contour maps still give some of asymmetry about the mid vertical line. The hot air is occupied the bottom regions, and the cold air is occupied the upper regions. Also, one can conclude from **(Figs. 1-D, E& F)** that the convective flow is stronger, making isothermal linear lines, especially at the lower corners of the plane. Therefore, the isothermal lines near these corners seem to be more smoothly.

### 11.2 NUSSELT AND RAYLEIGH NUMBERS

Heat transfer rates in terms of Nusselt number are presented in **(Fig. 4)**. **(Fig. 4-A)** manifests the effect of Rayleigh number (*Ra*) on the variation of the average Nusselt number ($\overline{Nu}$). As seen from this figure, the average Nusselt number increases with Rayleigh number as the heat flux ($q_w$) values increase until ($q_{w3}$ = 839W/m$^2$), and then the average Nusselt number value decreases, because of the temperature of the air reaches almost a constant values, and so that the value of heat transfer by bead conduction is indicated a dominant element on heat transfer by convection in air. The thermal capacity for beads ($c_{ps}$ =1939 J/kg. K) is higher than the thermal capacity of air ($c_{pf}$ =1008 J/kg. K), so the air reaches the thermal stability faster than the beads. Nusselt number ($\overline{Nu}$) depends on the value of porous media thermal conductivity factor. So the increase in the heat flux ($q_w$) will cause increase in porous media conductivity, this cause drop the average Nusselt number ($\overline{Nu}$) value. Since **(Fig. 4-B)** shows that the Nusselt number decreases along the *y*-axis, and it is similar in behavior to that of all heat flux ($q_w$) values. These results may verify the physical interpretation with the results of **[5]**, **[6].**

The statistical analysis (using STATISTICA software) based on the minimum sum of square error (SSE) led to the following relationship given in **(Table 3)**.

## 12. CONCLUSIONS

The following points can be concluded from the present experimental work:

1.The values of temperature and local Nusselt number remain stable approximately after (*y*=12.5 cm), above this region it is regards as the cooling plane.

2.The temperature values in the box vary inversely with the box height and directly with the heat flux.

3.A rapid increase in the temperature values within the box is deduced as the heat flux is increased due to a strong convective flow that increases the heat transfer within the box.

4.A wavy shape of the distribution temperature is obtained along a line parallel to the horizontal *x*-direction. The isothermal lines become more wavier in shape as the heat flux increases.

5.The rate of free convection heat transfer increases with the Rayleigh number increases at low values of heat flux applied on the heating surface.

6.The rate of free convection heat transfer is inversely proportional to the Rayleigh number for high values of heat flux, because the effect of conduction heat transfers is more than the convection of heat transfer.

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**Table (1):** Parameters range of the experimental work.

| Atmospheric temp. ($^oC$) | Heat flux ($W/m^2$) | Period to reach steady state (min) |
|---|---|---|
| 19 - 22 | 348 - 1384 | 180 - 360 |

**Table (2):** sample of the Nusselt number calculations.

| $q_w$ ($W/m^2$) | $q_{w1}$ =348 | $q_{w2}$ =576 | $q_{w3}$ =839 | $q_{w4}$ =1147 | $q_{w5}$ =1384 |
|---|---|---|---|---|---|
| $\overline{N_u}$ | 52.44 | 53.53 | 61.24 | 58.2 | 56.4 |
| ***Ra*** | 1537.64 | 2446.46 | 3487.13 | 4488.86 | 5260.62 |

**Table (3):** Specific empirical equation for free convection.

| $\overline{Nu} = C1\ Ra^{C2}$ | | |
|---|---|---|
| $C_1$ | $C_2$ | $R^2$ |
| 28.552 | 0.08434 | 81.6 % |

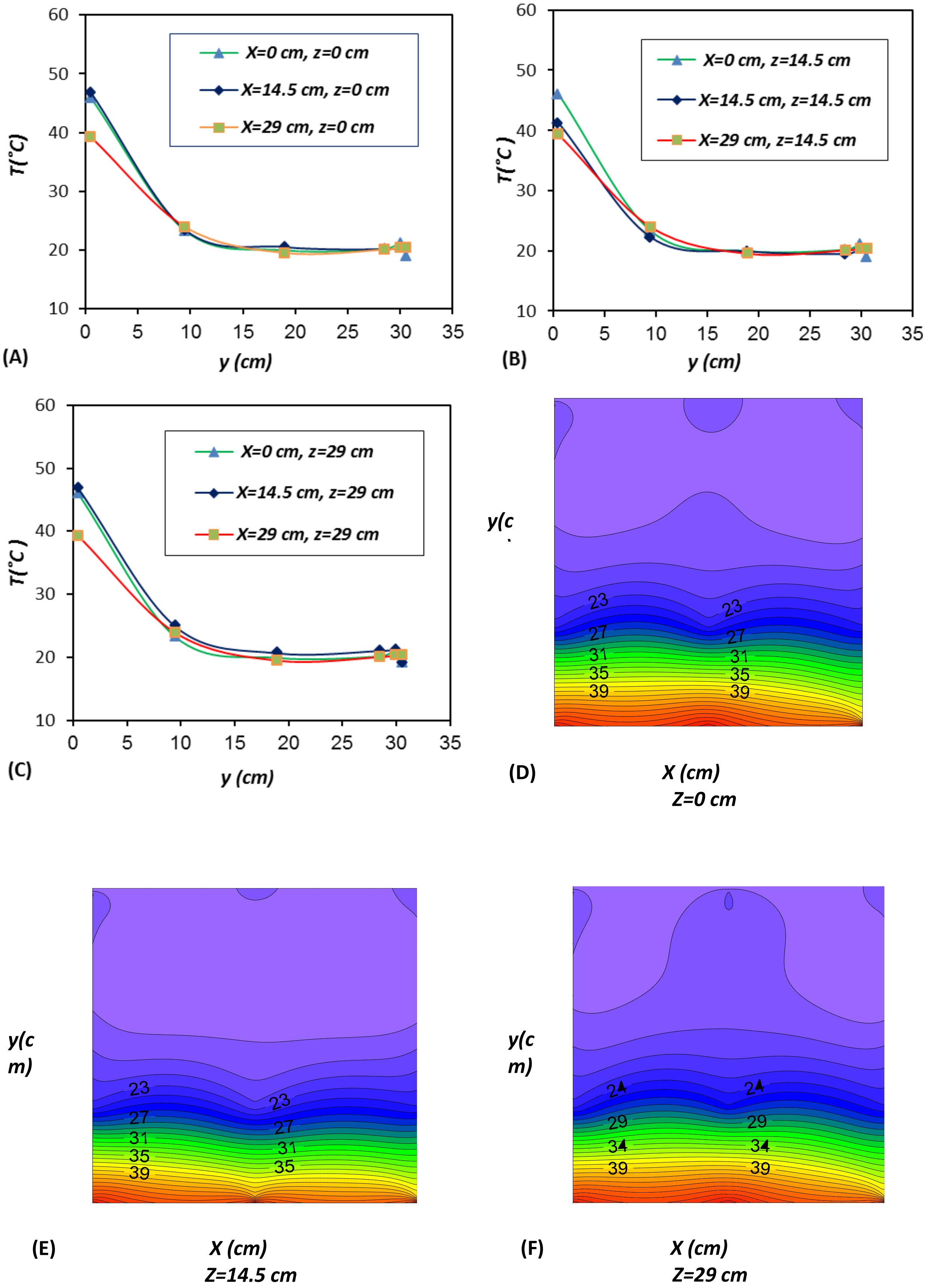


**Figure (1):** Temperature distribution (A, B and C) and contours (D, E and F) at ($q_{w1}$ =348 W/m²).

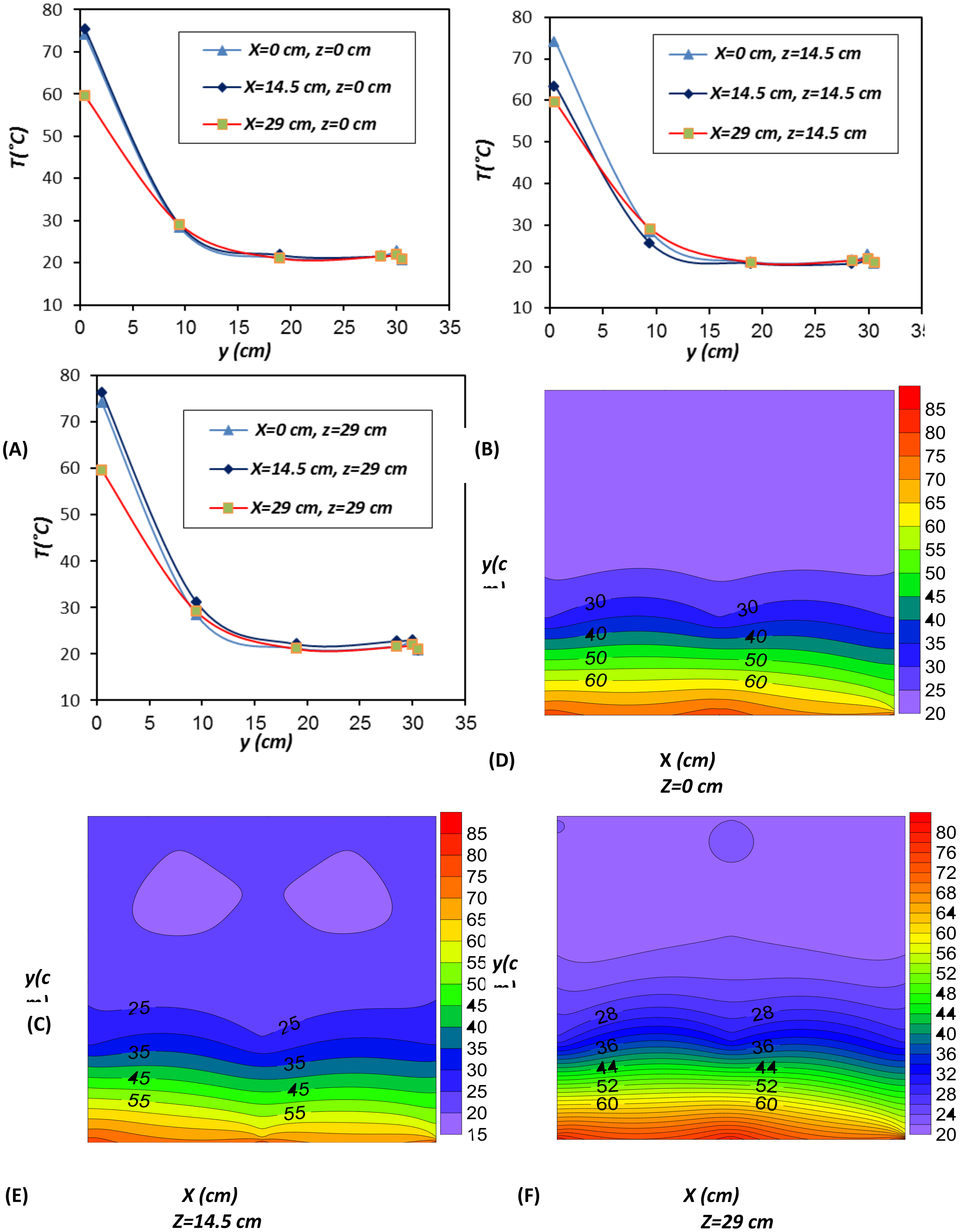


**Figure (2):** Temperature distribution (A, B and C) and contours (D, E and F) at ($q_{w3}$ =839 W/m²).

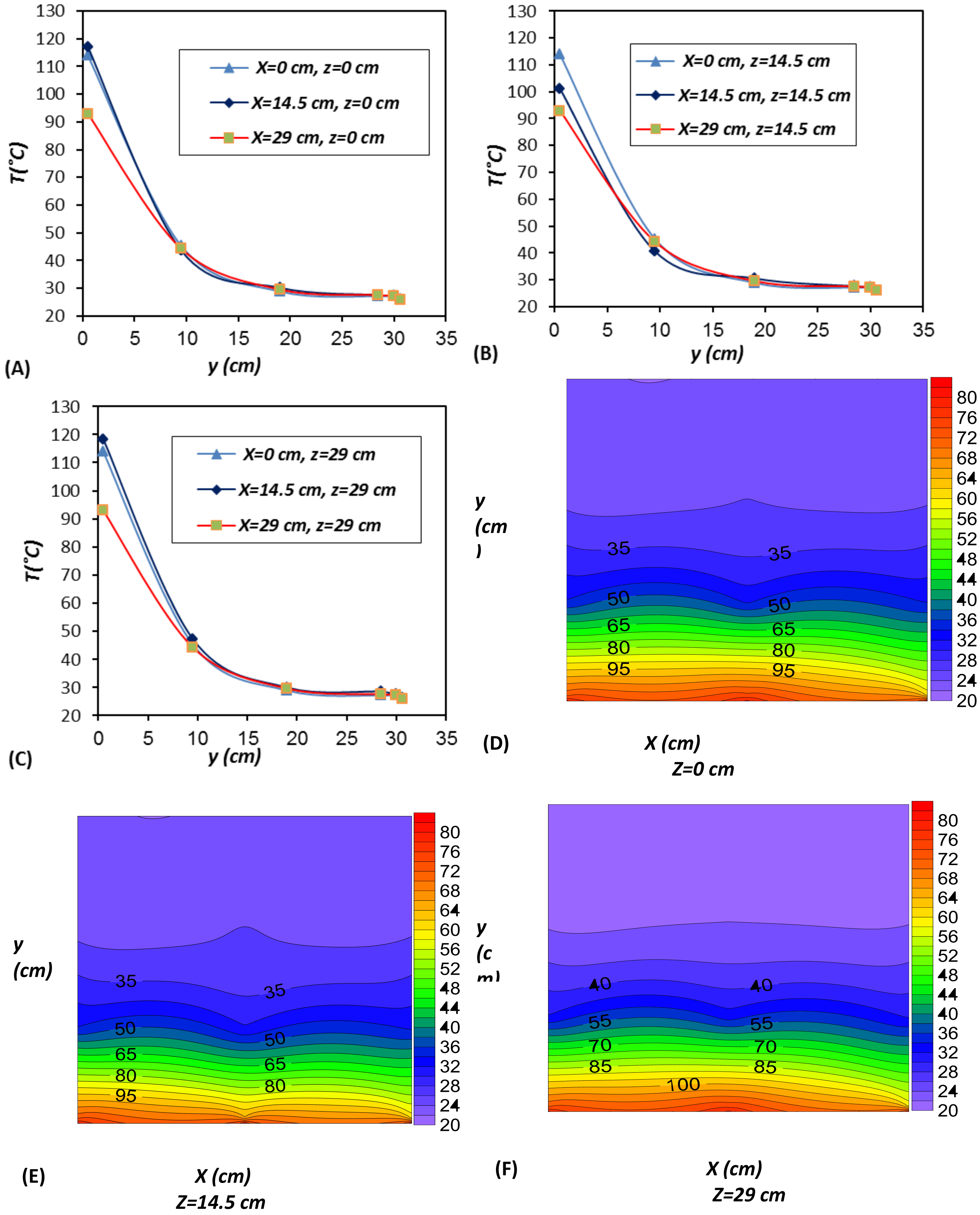


**Figure (3):** Temperature distribution (A, B and C) and contours (D, E and F) at ($q_{w3}$ =1384 W/m²).

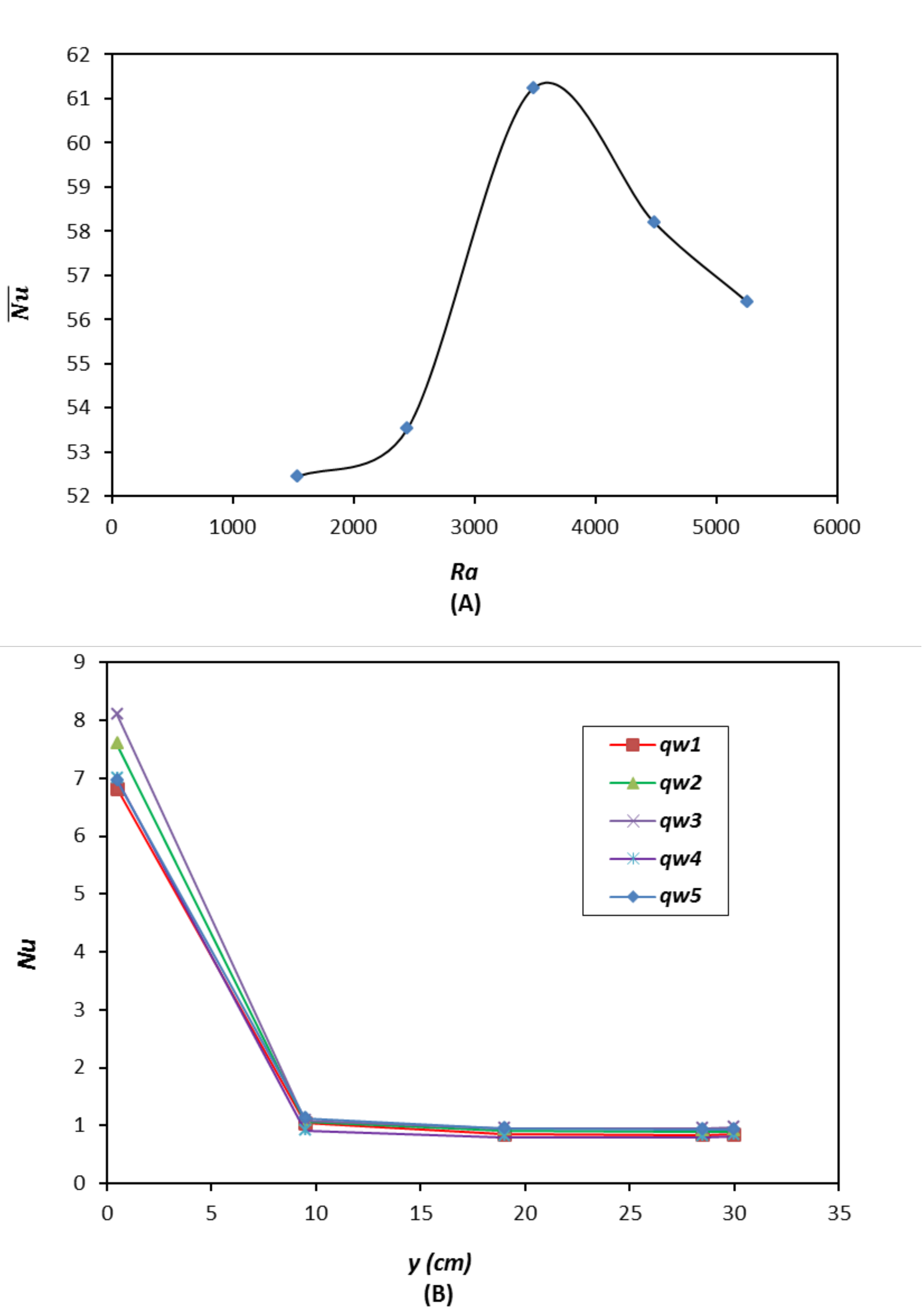


**Figure (4):** (A) Variation of the average Nusselt number with Rayleigh number, (B) Variation of local Nusselt number with y-axis, at $q_w$ = (348, 576, 839, 1147, 1384) W/m².