# An Investigation of Natural convection Heat Transfer in a cubic Enclosure with Oriented Partial Partitions at Different Angles

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## Abstract:

Natural convection heat transfer in an enclosure provided with inclined partitions to the two adiabatic sides, heated from the bottom with uniform heat flux and cooled from the top at constant temperature is studied experimentally and numerically in this work. The inclined partitions is well covered with an insulated material, so that, it can be assumed as parts of the adiabatic walls that places on. The governing parameter, Rayleigh number, is fixed in this work within $2.6 \times 10^{11}$, so that the effect of inclination angles of the two side's partitions can be investigated. The inclination angles of the two baffles range as ($0^o \leq$ and $\geq 150^o$). In numerical solution the effect of turbulence is modelled using (k-ε) model. Some application need to use the enclosed fluid layers as insulation, so that one purposes of this work deals with improve the insulating properties of fluid layers. The experimental and numerical works are done in 36 runs, grouped into 6 collections. Each collection with 6 runs done under a fixed inclination angle of one baffle and change the second baffle inclination angle to investigate the enclosure flow field and heat transfer. The result shows that a multi cells forms when the two baffles aboard to each other's, which is a reason to make a separation between a cold, and hot circulation cells that forms in the enclosure and act as insulator. It is also conclude that for all cases, the long insulated baffle of any inclination angle causes a reduction to the heat exchange inside the enclosure due to the damping cause to the flow field. The less average Nusselt number occurs when the two angles are equals, and the worst case is ($\theta=\beta=90^o$).

Keywords: natural convection, turbulence, enclosure, inclined partial partitions.

# التحقق في الحمل الحراري الحر داخل حيز مكعب يتضمن حواجز تميل بزوايا مختلفة

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الكلية التقنية / بغداد

## الخلاصة

الحمل الحراري الحر داخل مغلف مسخن من الاسفل بفيض حراري ثابت ومبرد من الاعلى بدرجة حرارة ثابتة في حين عزلت الجوانب الاخرى حراريا وثبت حواجز مائلة بزوايا مختلفة على جانبين متقابلين تمت دراستها عمليا ونظريا. تم تغليف الحواجز بمادة عازلة بشكل جيد لتكون وكأنها جزء من الجدران المعزولة حراريا. تم

تثبيت عدد رالي للدراسة على $2.6x10^{11}$ لغرض ايجاد تأثير زوايا ميلان الحاجزين. مدى زوايا ميلان الحاجزين ( اكبر من او يساوي$0^o$ و اصغر من او يساوي $150^o$ ). تم تمثيل تأثير الجريان الاضطرابي في الحل العددي باستخدام نمذجة (k-ε). بعض التطبيقات تحتاج الى خصائص العزل الحراري لطبقات المائع قرب السطح الساخن، لذلك فان احدى اهداف هذه الدراسة تعنى بتطوير هذه الخصائص. الدراسة العملية والنظرية اجريت بعمل 36 تجربة على شكل ستة مجاميع، وكل مجموعة دراسية تضمنت ستة تجارب تحت تأثير تثبيت زاوية ميلان الحاجز بإحدى الجهات وتغير زاوية ميلان الحاجز في الجانب الاخر لغرض معرفة تأثير اختلاف الميلان على حقل الجريان والتوزيع الحراري داخل المغلف. النتائج اظهرت حصول عدد كبير من الخلايا الدوامية عندما يكون الحاجزين متقاربين ، حيث تعمل هذه الخلايا على احداث منطقة عازلة بين السطحين الساخن والبارد. كما تم استخلاص ان الحواجز المعزولة حراريا والمائلة باي زاوية والمثبتة على جانبي المغلف تعمل على احباط عملية انتقال الحرارة داخل المغلف بسبب احباط حركة حقل الجريان. ان اقل معدل لعدد نسلت في جميع التجارب ال36 حدث عندما كانت ($\theta=\beta=90^o$).

## Nomenclatures:

| | | | | | |
|---|---|---|---|---|---|
| B | Baffle length | (m) | Ra | Raylieh number | |
| g | Gravitational acceleration | $(m/s^2)$ | t | Thickness | (m) |
| $h_{fg}$ | Heat of fusion of water | (kJ/kg) | T | Temperature | $(^oC)$ |
| H | Cubic enclosure length | (m) | $T_c$ | Cold temperature | $(^oC)$ |
| k | Thermal conductivity | (kg/s) | U | Velocity component in x-direction | (m/s) |
| $\dot{m}$ | Ice melt mass flow rate | $(W/m^2.K)$ | V | Velocity component in y-direction | (m/s) |
| n | Normal direction | | α | Thermal diffusivity | $(m^2/s)$ |
| $Nu_x$ | Local Nusselt number | | ν | Kinematic viscosity | $(m^2/s)$ |
| $\overline{Nu}$ | Average Nusselt number | | θ | Left baffle inclination angle | (degree) |
| $q''$ | Constant heat flux | $(W/m^2)$ | β | Right baffle inclination angle | (degree) |

## Introduction

The studies of natural convection heat transfer and fluid flow inside enclosures of complex geometry as in those of multi partitions of different configuration, has attracted the interest of wide researches in recent years. Some studies interested in the effect of these partitions in the applications range from nuclear reactors design, heat exchangers design, and enhancement of room air and cooling of electronic devices. Others is interested in find means of improve the insulating properties of fluid layers, which has a wide range of applications related to the reduction of heat losses from flat plate solar collector, so that the heat transfer rate through in an enclosure may be controlled by the use of conductive or adiabatic partitions located and oriented in a controlled manner. The case of natural convection in an enclosure heated and cooled through the vertical walls while the horizontal walls are kept adiabatic has received a great consideration in studies, that is due to many industrial application employ this concepts. Some others concerned with the case of natural convection inside enclosure motivated by heating and cooling the horizontal walls, while the vertical walls are insulated. For both cases, partitions may be added to improve and control the heat transfer and fluid flow characteristics.

Yucel and Ozdem [1] investigated numerically fluid flow and heat transfer in partially divided square enclosures, with the horizontal end walls are adiabatic or perfectly conducting and the side walls are maintained at uniform but different temperature. It is observed that, mean Nusselt number increases with increasing Rayliegh number and decreases with increasing number of partition, however, decrease in mean Nusselt number is less at low Rayliegh number, and increasing partition height decreases mean Nusselt Number.

Ambarita H. et al. [2] studied numerically a differentially heated square cavity, which is formed by horizontal adiabatic walls and vertical isothermal walls, with two perfectly insulated baffles attached to its horizontal walls at symmetric position. A parametric study was carried out using Rayliegh number from $10^4$ to $10^8$, non-dimensional thin baffles length 0.6, 0.7, and 0.8, and non-dimensional baffle position from 0.2 to 0.8. It was observed that the two baffles trap some fluid in the cavity and affected the flow fields, also found that Nusselt Number is an increase function of Rayliegh number, a decreasing one of baffle length, and strongly depends on baffle position.

Frederick R. L. [3] numerically studied natural convection in an air-filled, differentially heated, inclined square cavity, with diathermal partition on its cold wall. The partition cause convection suppression and heat transfer reduction up to 47% relative to the undivided cavity at the same Rayliegh number, partition length, and inclination. For long partitions, transition to bi-cellular flow occurs. At high Rayliegh numbers the heat transfer reduction is affected by secondary buoyancy forces generated by the partition.

Ghassemi M. ET. al. [4] investigated numerically the effect of two insulated horizontal baffles placed at the walls of a differentially heated square cavity. The vertical walls are at different temperatures while the horizontal walls are adiabatic. The result presented for Rayliegh numbers from $10^4$ up to $10^7$ observed that the two baffles trap some fluid in the cavity and affect the flow. Also, Nusselt number increase and decrease with baffle length, and changes with baffles position.

Ben-Nakhi A. And Chamkha A.J. [5] investigated numerically steady, laminar, natural convection fluid flow in a square enclosure attached with an inclined high conductive heated thin fin of arbitrary length equal to 20%, 35%, and 50% of the side length, positioned in the middle of the hot wall of the enclosure. A transverse temperature gradient is applied on two opposing walls of the enclosure, while the other two walls are adiabatic. It is found that the Rayliegh numbers and the thin fin inclination angle and

Length have significant effects on the average Nusselt number of the heated wall including the fin of the enclosure.

Varol Y. ET. al. [6] studied experimentally and numerically natural convection heat transfer in an adiabatic inclined one-fin attached one sides of the square enclosure. Bottom wall of the enclosure has higher temperature than that of the top wall while vertical walls are adiabatic. The governing parameters are Rayliegh number ($8x10^5 \leq Ra \leq 4x10^6$) and inclination angle ($30^o \leq$ and $\leq 120^o$). It was observed that multiple cells were formed at the studied Rayliegh number for each inclination angles, the inclination angle affects the flow strength and temperature distribution, and heat transfer can be controlled by attaching an inclined fin onto wall.

Nansteel M.W. [7] investigated experimentally natural convection in a two dimensional rectangular enclosure fitted with partial vertical divisions. Partial divisions were found to have a significant effect on the heat transfer, especially when the division are adiabatic.

Nansteel M. W. And Greif R. [8] investigated experimentally the heat transfer and fluid flow in a rectangular enclosure fitted with a vertical adiabatic partition. The partition was oriented parallel to the two vertical isothermal walls one of which was heated and the other cooled. Water was used as the working fluid and two cases of partitions studied. The dependence of the flow and the cross-cavity heat transfer on the Rayliegh number and on the size of the opening in the partition is studied.

Beymark J. ET. Al. [9] describes an experimental study aimed on determining the effect of internal partitions on the natural convection heat transfer across an enclosure using cubic geometry heated from the side, with an internal partial vertical partition. Two test cells with different working fluids are used. Nusselt - Rayleigh aperture width correlation curves were developed for both air and water.

In the present work, the effect of an inclination angles of an adiabatic partitions which are fixed to the two adiabatic sides of an cubic enclosure heated from the bottom with constant heat flux and cooled from the top at constant temperature, on natural convection at specified Rayleigh number in a turbulent range, is studied experimentally and numerically. The objective is to determine the influence of couples of inclination angles of the two baffles on the flow and heat transfer characteristics of the air inside the enclosure under uniform heat flux. Based on author knowledge, there is no experimental and numerical study for natural convection inside an enclosure with twin baffles fixed in the vertical facing sides oriented with different angles.

## Physical Modelling:

The schematic of the Physical situation of the problem, which is a middle section of cubic enclosure, is shown in figure (1). The bottom hot wall keeps at uniform heat flux and the top cold wall is at constant temperature. The vertical walls are perfectly insulated. A system of baffles with length (B) and thickness of (t) located in the middle of the vertical walls and inclined with different couple of inclination angles denoted by ($\theta$), and ($\beta$) ranged ($0^o \leq$ Angle $\leq 150^o$) are used.

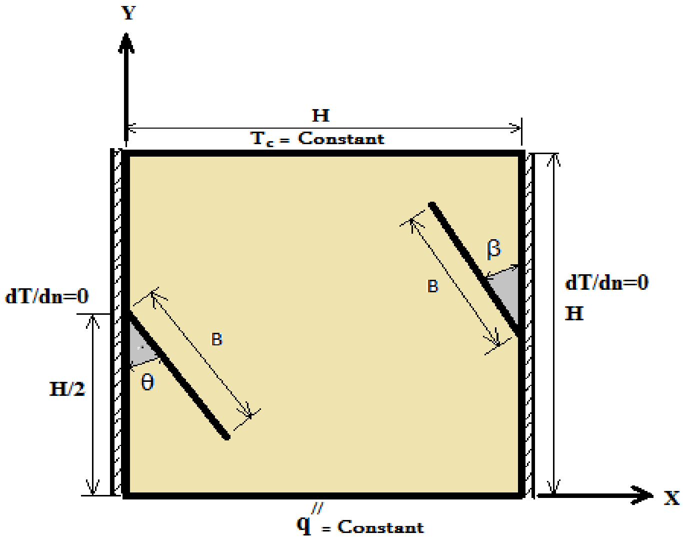



**Figure (1) Schematic of the square cavity with the attached baffles**

Due to the high Rayleigh number (Ra=2.6x$10^{11}$) of the case under investigation, the behaviour of the air inside the enclosure studied as turbulent flow. The thermo-physical properties of the air inside the enclosure are assumed constant in the bulk temperature of the average hot wall temperature and the constant cold wall temperature, except the density in the gravity force which is assumed to be temperature dependent, and the Bossiness approximation is used. Viscous model- standard k-ε with standard wall function model is used to solve the governing equation associated with turbulent natural convection problem.

The following boundary conditions are used to solve the governing equations:

1- On the bottom wall (Y=0), U=0, V=0, $q'' = con.$
2- On the top wall (Y=1), U=0, V=0, $T = T_c$
3- On the left wall (X=0), and right wall (X=1), U=0, V=0, and $\frac{\partial T}{\partial n} = 0$ where, n is the normal to the walls.
4- On the two baffles sides, U=0, V=0, $\frac{\partial T}{\partial n} = 0$

With the above assumption and boundary condition, Fluent under Ansys 14.5 release is used to solve the problem. C1-epsilon used in the solution was (1.44), whereas C2-epsilon was (1.92). Under relaxation factors used for the solution control are: pressure (0.2), density (0.95), momentum (0.5), turbulent kinetic energy (0.5), turbulent dissipation rate (0.5), turbulent viscosity (1), and energy (0.8). The numerical results which are get represent the local and average Nusselt numbers which then compared with the experimental results to validate the work. Also the flow velocity field and temperature distribution counters for each case under investigation are got and shown.


Figure (2) Experimental Apparatus

## Experimental Apparatus:

A cubic enclosure with (H=0.3m) shown in the photograph of figure (2). The bottom hot wall is under uniform heat flux of (350 W/m$^2$), and the top cold wall is at constant temperature $T_c$ of (2±0.5$^o$C). The vertical walls are perfectly insulated. A system of non-conductive baffles with length B=0.135m located in the middle of the vertical walls and inclined with different couples of inclination angles denoted by θ, and β ranged (0$^o$ ≤ Angle ≤ 150$^o$) are used. The non-conductive baffles were fabricated from thin metal coated from the two sides with rubber with overall thickness (t=0.005m). The enclosure is filled with air. The enclosure is constructed from three sides by the use of low conductivity block wood, and from the front side by the use of double pan glass window to allow visualize. The top side is constructed from pure aluminium sheet (0.1mm) fabricated as fully closed container with gate to use as crashing ice-vessel, and covered from all outsides by (25mm thickness) sterol-board to insure constant temperature of the top wall at about 2$^o$C ±0.5$^o$C. The crashing ice-vessel is connected with drain pipe at its bottom to drain molten ice water continually and prevents the forming of high temperature film of water beneath the crashing ice. After steady state condition reached in the enclosure, the amount of molten ice water is collected in a scaled beaker for 10 minutes, to calculate the amount of heat received from the enclosure by the ice to melt in this period of time using thermodynamic heat balance equation ( $Q_{ice\,melt} = \dot{m} h_{fg(of\ water\ at\ 1^o C)}$ ) so that the exact heat transfer from the top side can be evaluated. The bottom wall of the enclosure is constructed from sheet of pure aluminium (8mm) thickness to insure better uniform heat flux. An electrical resistance heater fixed under the aluminium sheet, constructed from strips of (1 mm) width made of chrome-nickel alloy with resistance (10 ohm/m) and wrapped with (5 mm) pitch around a (300x300 mm$^2$) mica sheet of (0.5 mm) thickness, to ensure the electrical insulation. The overall resistance of the heater used is (96 ohm). The heater is carefully mounted on a (320x320 mm$^2$) mica sheet and covered with other (320x320 mm$^2$) mica sheet to prevent the electrical contact. The heater is covered from bellow by a glass wool of (80 mm) thickness to reduce the heat loss and then covered with a wood cover plate. All heater system is bounded with wood box (25mm thickness) to prevent any heat loss from the transverse side. Eight thermocouples (0.3mm) copper-constantan type-T distributed in two horizontal plane level parallel to the plane of the heater (four in each plane) are plant in equal distances from all sides inside the glass wool. The first thermocouples plane located (5mm) from the heater plane whereas the second one Located (60 mm) apart, for the purpose of heat loss measurement from the bottom side of the heater. The thermocouples were calibrated to measure a temperature difference error of about (±0.5$^o$C). The heater is supplied with AC-current by voltage regulators. The voltage and current supplied to the heater was measured with a calibrated volt meter and ampere meter at an accuracy of about (±1.2%). The net power supplied to the enclosure was (350 W/m$^2$). This power represented the electrical power measured from the reading of the volt and ampere meters, after subtracting the heat loss from the bottom of the heater by conduction through the glass wool insulation. This losses was calculated applying Fourier's law of heat conduction in finite difference form: $Q_{losses} = -k_{insulate} A(\Delta T / \Delta y)\ Watt$, where $\Delta T$ represents the temperature difference between the average readings of the two plane level thermocouples, ($\Delta y$ =60mm), and ($k_{insulate}$=0.036 W/m$^2$.$^o$C). The $\Delta T$ reading was in the order of 50$^o$C, and that means ($Q_{losses}$) in the order of (30.6 W/m$^2$) which represent (8.5%) of the total heat generated by the electrical heater and can be recovered by increase the power generated by the electrical heater to maintained power in put to the enclosure at (350 W/m$^2$), and compared with the power out to the ice for accuracy.

The heat loss by conduction through the block wood surfaces of the enclosure were estimated by applying Fourier's law of heat conduction in finite difference form mentioned above, and it was found that the total amount of heat loss within (15.5 W/m$^2$), which represent only (4.4%) of the total heat supplied to the hot wall. Surfaces temperatures of the two sides of the block wood walls of the enclosure were got by 12 type-T thermocouples. In the same way, the estimating heat loss through the front wall double pan glass window, and the top wall ice vessel equal to (1.2%) and (5.2%) of the total heat supplied to the hot wall. That is mean; the percentage of the total heat loss through the walls of the enclosure to the environment to that enter to the enclosure is (10.8%).

To find the local heat transfer coefficient and then the local Nusselt number in the middle line of the hot wall plane, perpendicular to the baffles orientation, the hot wall surface divided into 12 part, and a 12 thermocouples type-T were plant in the hot wall only 0.2mm from the hot surface, insert from the bottom of the hot wall. The air temperatures inside the enclosure in a location (0.2& 0.4mm) above the hot wall surface in the centers of the 12 parts were measured using a sheathed thermocouple probe type(TD745) with accuracy of (±0.2$^o$C) which could be moved parallel to the hot wall enter from a small holes in the side of the enclosure. This air temperatures measurements allows find he temperature gradient in each one of the 12 parts of the hot wall and then can find the local and average Nusselt number, according to the following equations **[10]**:

$$Nu_x = -\frac{\left.\frac{dT}{dy}\right|_{y=0} .H}{T_h - T_c} = \frac{q''H}{k(T_{h_x} - T_c)} \quad (1)$$

And the average Nusselt number can be calculated according to:

$$\overline{Nu} = \int_0^H Nu_x dx \quad (2)$$

Rayleigh number can be calculated according to:

$$Ra = \frac{g\beta H^4 q''}{k\upsilon\alpha} \quad (3)$$

## Error Analysis:

Uncertainties and error experimentally can be arising from calibrations, measurements, kind of instruments and readings. Nusselt number as in equation (2) is a function of (5) independents variable as shown,

$$Nu = f(q'', T_h, T_c, H, k) \quad (4)$$

According to the uncertainty analysis given by Doebelin **[11]**, the uncertainty in the *Nu* can be given by,

$$\Delta Nu = \left|\Delta q'' \frac{\partial Nu}{\partial q''}\right| + \left|\Delta T_h \frac{\partial Nu}{\partial T_h}\right| + \left|\Delta T_c \frac{\partial Nu}{\partial T_c}\right| + \left|\Delta H \frac{\partial Nu}{\partial H}\right| + \left|\Delta k \frac{\partial Nu}{\partial k}\right| \quad (5)$$

Nusselt number uncertainty analysis shows that the maximum error is (±1.918).

## Results and Discussion:

In this investigation an experimental and numerical solutions is done to the problem represented by the effect of change the inclination angles (θ) and (β) of a couple of adiabatic baffles attached to the vertical adiabatic walls. The results in 6 collections shown in figures (3) to (8) represent the local Nussult number with respect to x-position, the velocity field and temperature distribution contours for the cases at which (θ) or (β) is fixed and change the other. From these figures it is clear that there is a good agreement shown between the experimental and numerical solutions.

Figure (3) shows the collection of (β=0) and (θ) changes from $0^o$ to $150^o$. The left column shows the local Nusselt number with (x=0→0.3m) experimentally and numerically. It shows that local Nusselt number increase with (x) to some value and then decrease depends on the baffle inclination angles, and upon the direction of the cell circulating. Generally, Nusselt number decrease as (θ) increase up to $90^o$ and then it will increase slightly up to (θ =$150^o$). The flow field contour at the middle column and the temperature distribution column right column explain this situation. When (θ =0), the whole of the enclosure act to transfer the heat from the bottom wall to the top wall. But,

When (θ) increase up to $30^o$, part of the enclosure act as heat trap, and this trap increase as (θ = $60^o$ and $90^o$), and two circulating cells start to form, one of a hot air near the hot wall and other cold near the cold wall with little mixing between them at the centre of the enclosure. As (θ) increase farther to $120^o$, main circulating cell will form with another small one which results increasing the local Nusselt number.

Figure (4) shows the collection of (β=30) and (θ) changes from $0^o$ to $150^o$ in three column as in figure (3). In this collection the left baffle act as reducer to the size of the bloom in the enclosure and there is a trap space on the right corner of the enclosure in all cases in addition to the traps forms from the left size due to the small (θ). Local Nusselt number reduces in the same manner as in figure (3) but its overall level is lower compare with previous collection, and its maximum value shift to the left. A two cells form as (θ) increase to (θ =$60^o$) but its size is smaller compare with the same case of figure (3), and vanish as (θ =$150^o$) to make one cell with higher maximum value of local Nusselt number in this case.

Figure (5) shows the collection of (β=60) and (θ) changes from $0^o$ to $150^o$ in the same manner as before. The maximum local Nusselt number shift more to the left due to the effect of right baffle, and a two cells forms earlier as (θ =$0^o$). When (θ =$30^o$) the lower cell shown to be contracted and divided into two cells in the vertical direction, and then the two cells divided into six cells with small sub-cells as (θ =$60^o$) in a symmetry shape. When (θ =$90^o$) the upper half forms one main cell but the lower half of the enclosure forms three main cells with many sub-cells in the two half's. The cells start to form inverse S-shape for (θ =$120^o$) with some sub-cells and the number of cells reduces as (θ =$150^o$). The multi cells forms reduce the heat exchange between the hot & cold walls, and that is explain the reason of reduce the level of local Nusselt number in this case. The maximum local Nusselt number noticed when (θ =$0^o$) and then when (θ =$150^o$).

Figure (6) shows the collection of ($\theta$ =90$^o$) and ($\beta$) changes from 0$^o$ to 150$^o$. Local Nusselt number seems to be lower than the previous collection especially for ($\beta$=30$^o$ to 120$^o$). The multi cells forms especially when the two baffles aboard to each other, so that case (d) of ($\theta$ =$\beta$=90$^o$) can be assumed the worst case in the heat exchange performance of all cases.

Figure (7) shows the collection of ($\theta$ =120$^o$) and ($\beta$) changes from 0$^o$ to 150$^o$. The size of the cells is larger than that of the previous collection results higher local Nusselt number, although there are a lot of sub-cells which is reduce the local Nusselt number compare with the first three collection cases. The worst case here is when ($\theta$ =120$^o$ and $\beta$ =120$^o$) due to the totally separation between the hot and cold portion of the enclosure as a result of the blockage done by the baffles.

Figure (8) shows the collection of ($\theta$ =150$^o$) and ($\beta$) changes from 0$^o$ to 150$^o$. The large cells in this collection is the reason of relative high local Nusselt number, but the difficulty of the fluid flow in the cases of ($\beta$=60$^o$, 90$^o$, 120$^o$) cases the reduction of it.

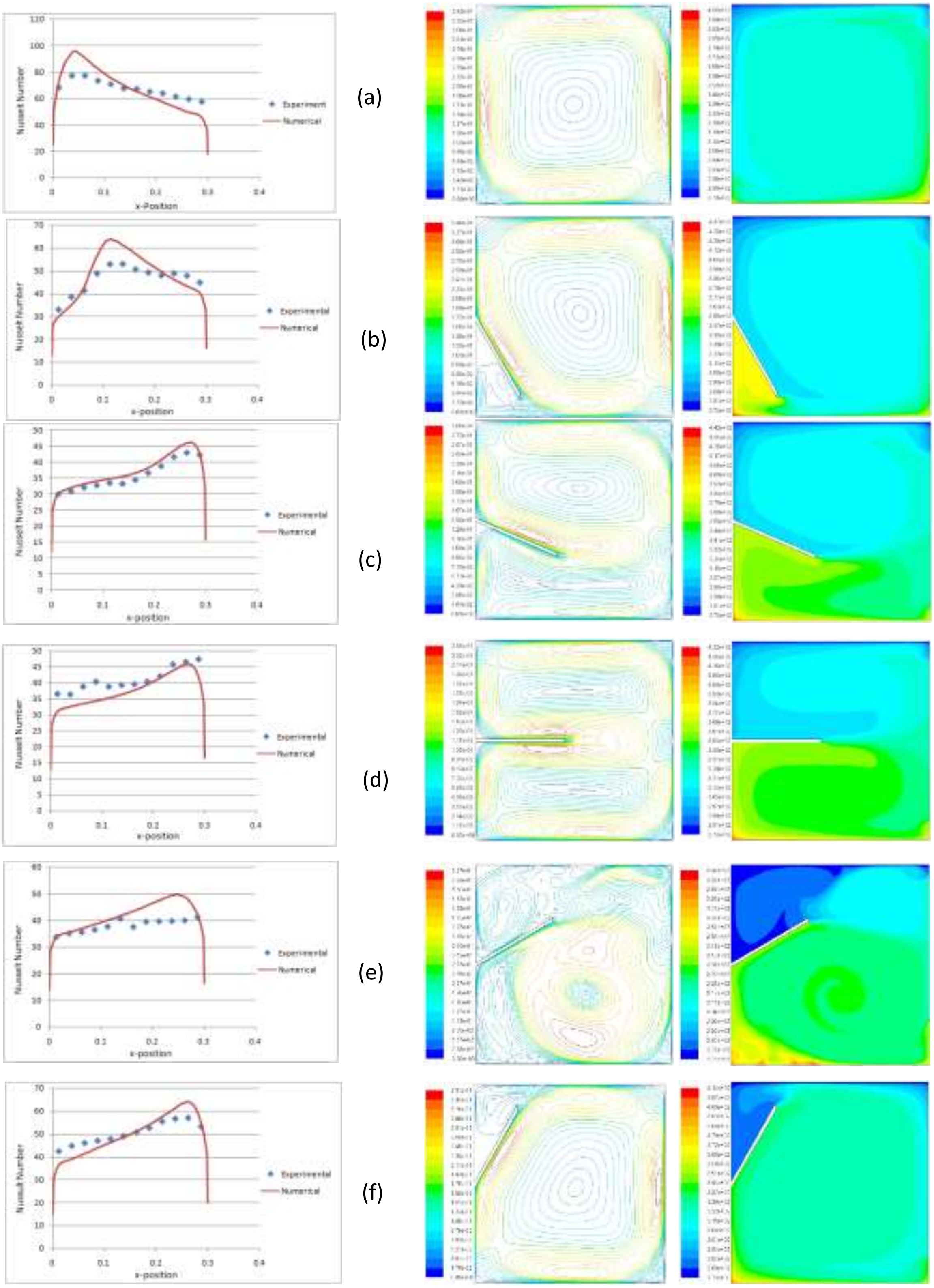
Figure (9) shows the local Nusselt number with x-position on the left of each sub-figure and average Nusselt number with inclination angles on the right of each sub-figure from (a) to (f) as a compact of each collection which are discussed. The right figures show clearly that the average Nusselt number decrease as the inclination angles of the baffles increase for all collections and then increase. For all cases, the long insulated baffle of any inclination angle causes a reduction to the heat exchange inside the enclosure due to the damping cause to the flow field. Figure (10) shows the variation of average Nusselt number for the cases of $\theta$=$\beta$. This figure shows that the less average Nusselt number occurs when the two angles are equals, and the worst case is ($\theta$=$\beta$=90$^o$). Figure (11) shows the variation of average Nusselt number with all cases under investigation. It is shown that the highest average Nusselt number is at the collection of $\beta$=0$^o$, and the lowest is at the collection of $\theta$=90$^o$.

## Conclusion:

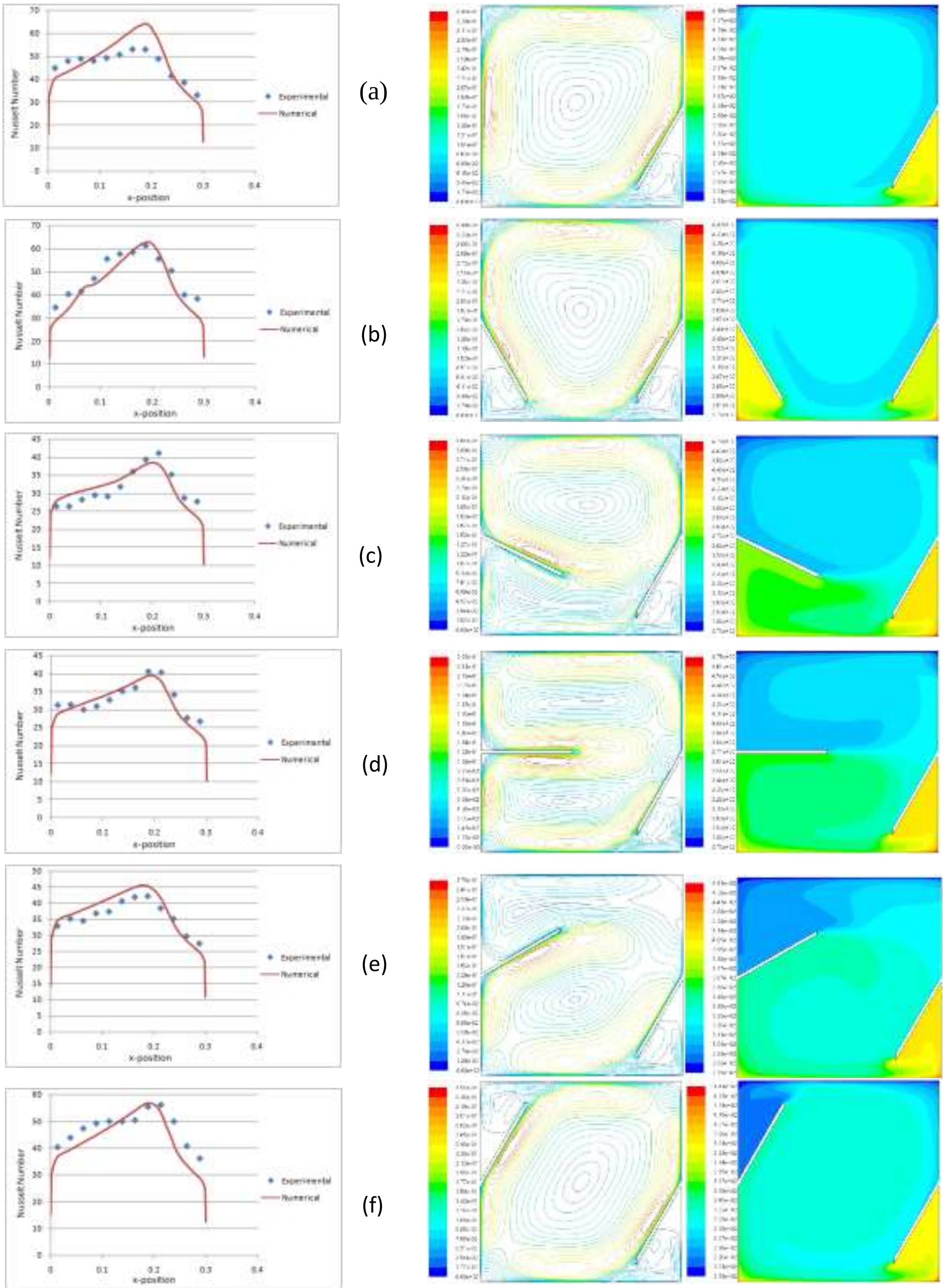
In this investigation, the effect of attached a couple of insulated baffles, oriented with couples of inclination angles, in a facing sides of an cubic enclosure heated from the bottom with uniform heat flux and cold from the top by constant temperature, whereas other sides kept insulated, which has done experimentally and numerically, gave a good answers to the problems of improving the insulating properties of fluid layers. This result allow to use a long insulated baffle in controlling the heat exchange inside the enclosure, especially to reduce the heat losses from flat plate solar collector, and other industrial applications. Following is the important high marks in this investigation:

1- The experimental solution gets a good agreement with the numerical solution to the problem.
2- The long insulated baffle of any inclination angle causes a reduction to the heat exchange inside the enclosure due to the damping cause to the flow field.
3- The multi cells forms inside the enclosure reduce the heat exchange between the hot & cold walls, and that is explain the reason of reduce the level of local Nusselt number.
4- The average Nusselt number decrease as the inclination angles of the baffles increase for all collections and then increase.
5- The highest average Nusselt number is at the collection of $\beta=0^o$, and the lowest is at the collection of $\theta=90^o$.

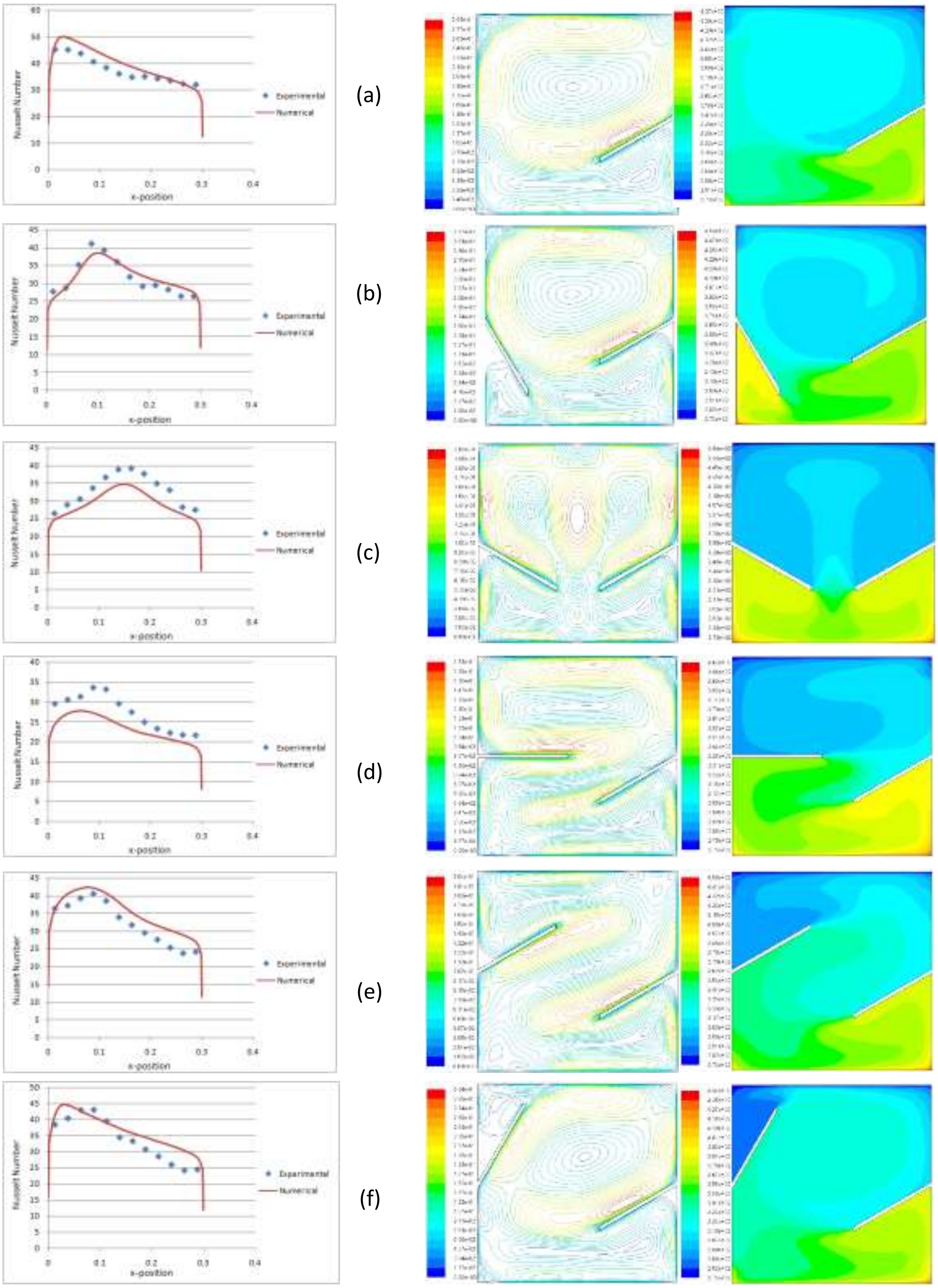
It is recommended that further study to be directed toward find the effect of change of Rayleigh number on the behaviour of the same problem. This will allows to find an empirical equation to find average and local Nusselt number as a function of Rayliegh number, ($\theta$), and ($\beta$) angles.



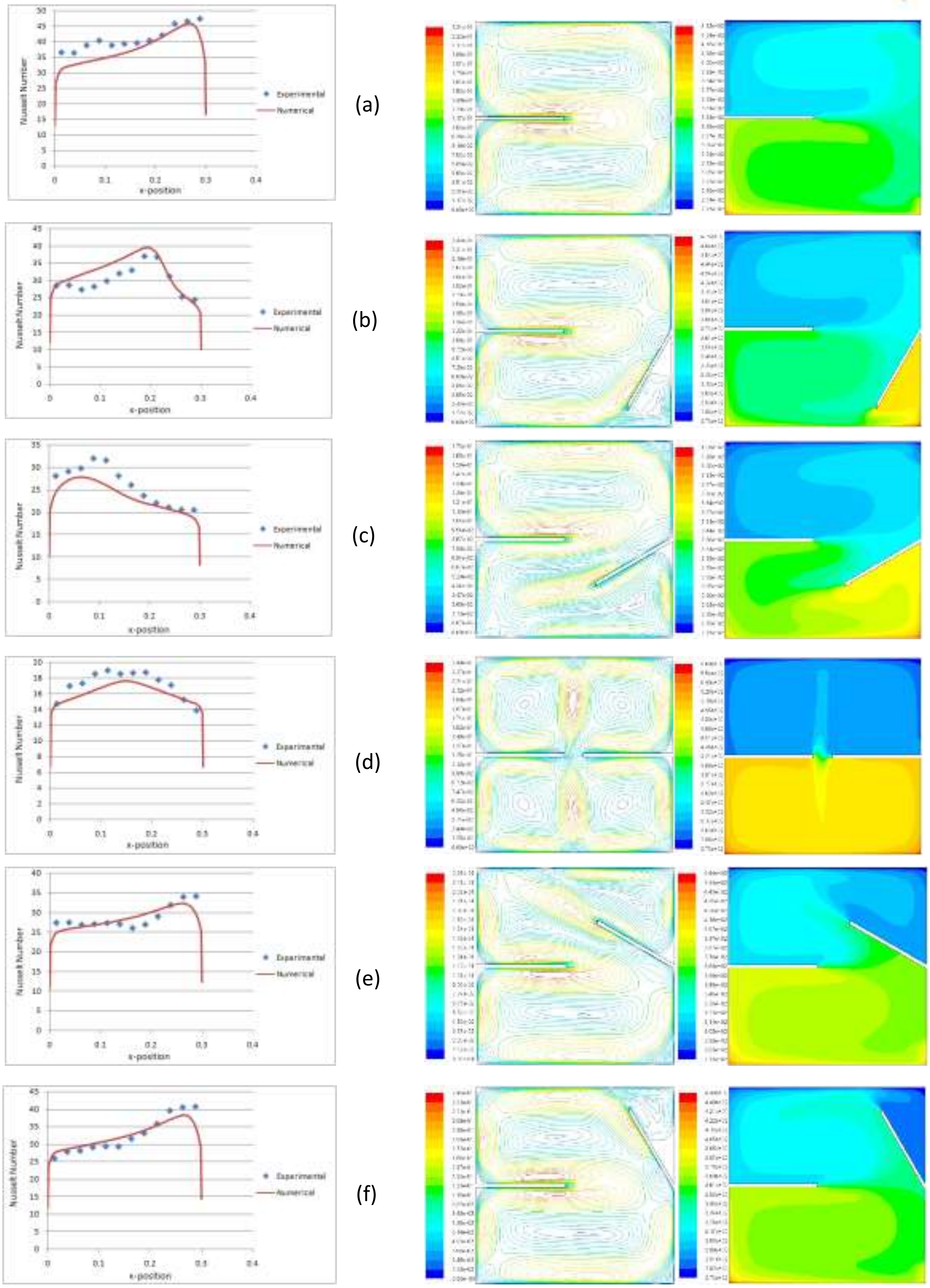

Figure(3) Local Nusselt on the left column, contours of the velocity magnitude(m/s)on the middle column, and contours of the temperature (K) on the right column, (Row-a) θ=0, β=0, (Row-b) θ=30, β=0, (Row-c) θ=60, β=0, (Row-d) θ=90, β=0, (Row-e) θ=120, β=0, (Row-f) θ=150, β=0



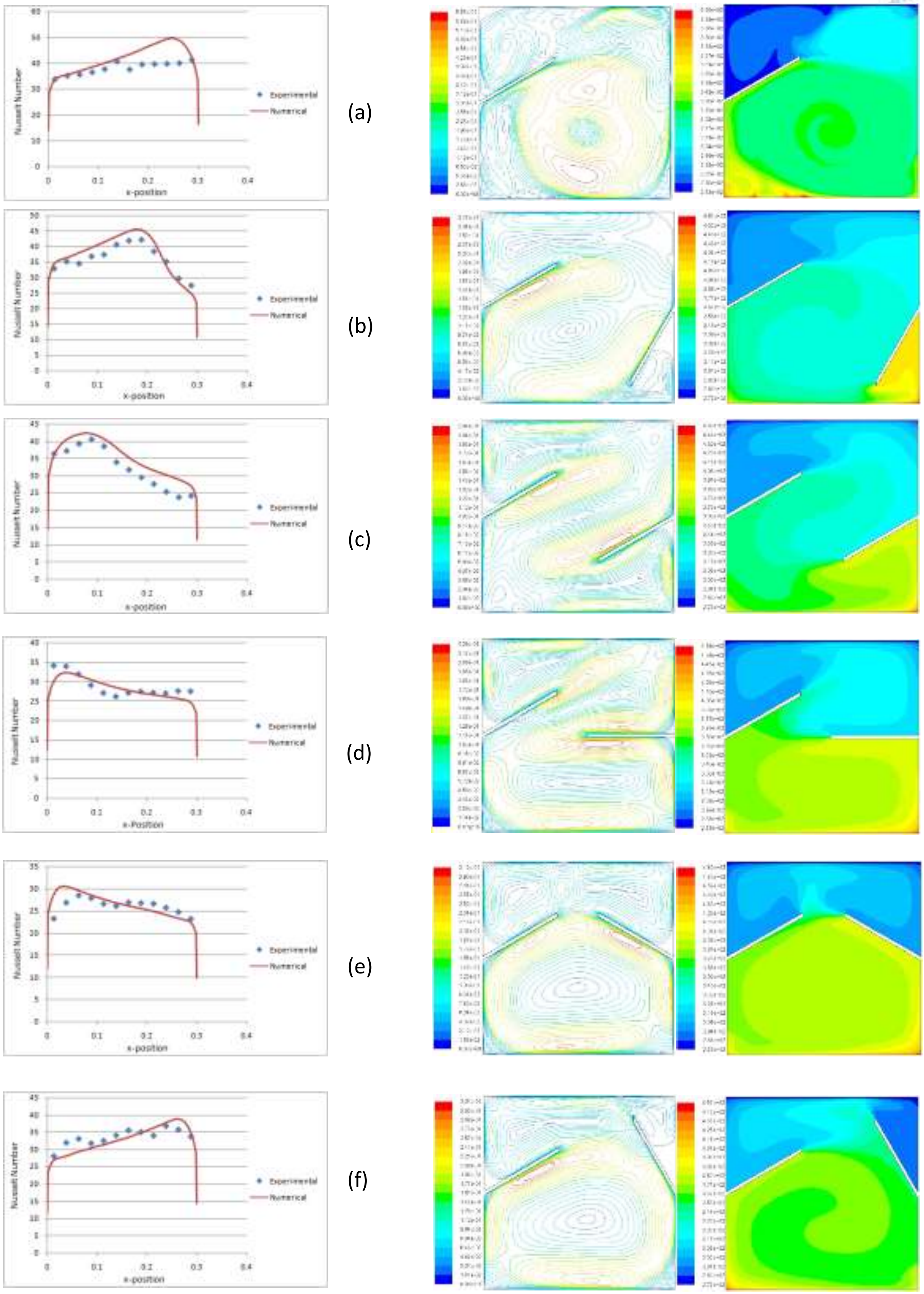

Figure(4) Local Nusselt on the left column, contours of the velocity magnitude(m/s)on the middle column, and contours of the temperature (K) on the right column, (Row-a) θ=0, β=30, (Row-b) θ=30, β=30, (Row-c) θ=60, β=30, (Row-d) θ=90, β=30, (Row-e) θ=120, β=30, (Row-f) θ=150, β=30



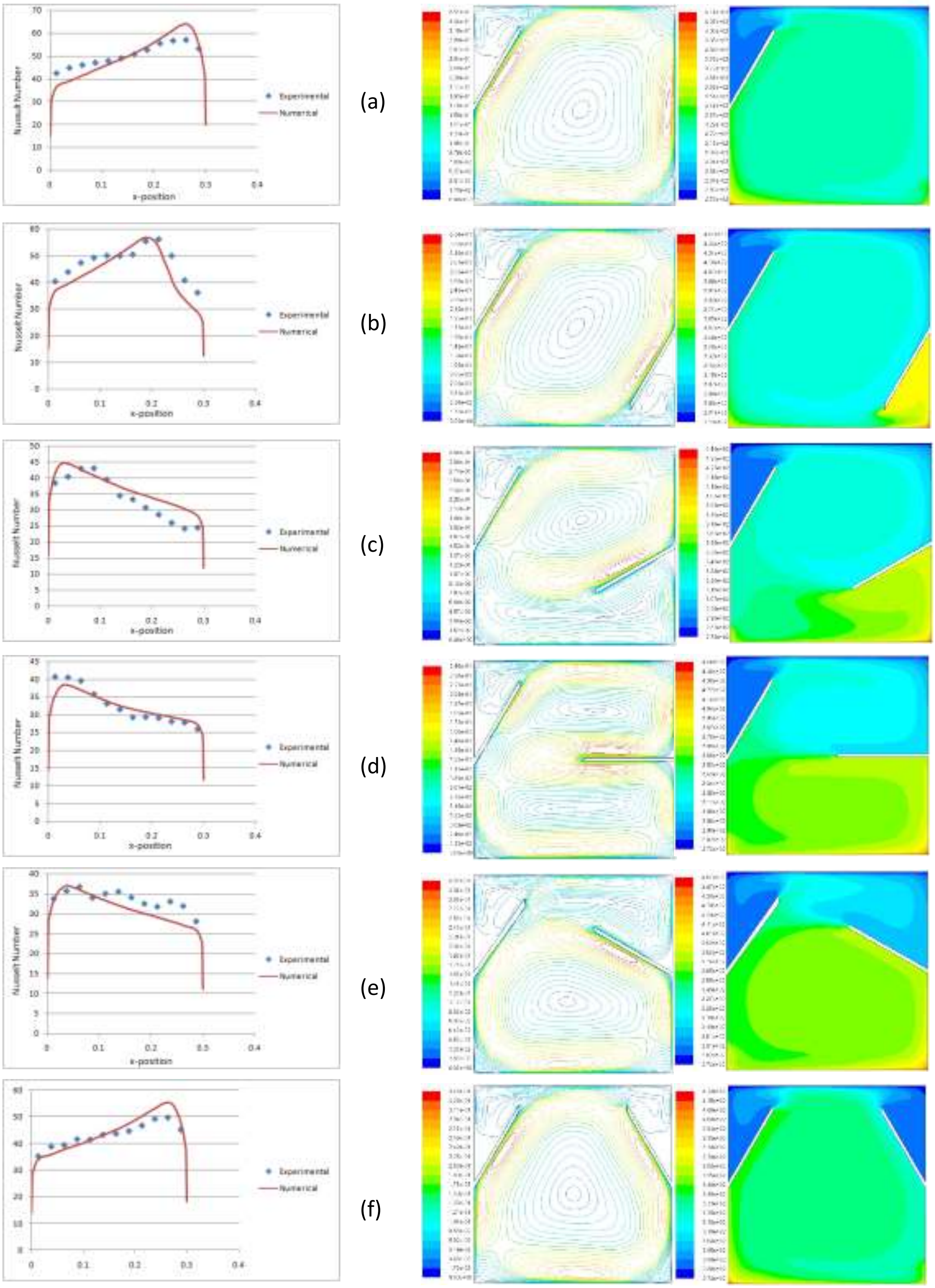

Figure(5) Local Nusselt on the left column, contours of the velocity magnitude(m/s)on the middle column, and contours of the temperature (K) on the right column, (Row-a) θ=0, β=60, (Row-b) θ=30, β=60, (Row-c) θ=60, β=60, (Row-d) θ=90, β=60, (Row-e) θ=120, β=60, (Row-f) θ=150, β=60




Figure(6) Local Nusselt on the left column, contours of the velocity magnitude(m/s)on the middle column, and contours of the temperature (K) on the right column, (Row-a) θ=90, β=0, (Row-b) θ=90, β=30, (Row-c) θ=90, β=60, (Row-d) θ=90, β=90, (Row-e) θ=90, β=120, (Row-f) θ=90, β=150




Figure(7) Local Nusselt on the left column, contours of the velocity magnitude(m/s)on the middle column, and contours of the temperature (K) on the right column, (Row-a) θ=120, β=0, (Row-b) θ=120, β=30, (Row-c) θ=120, β=60, (Row-d) θ=120, β=90, (Row-e) θ=120, β=120, (Row-f) θ=120, β=150




Figure(8) Local Nusselt on the left column, contours of the velocity magnitude(m/s)on the middle column, and contours of the temperature (K) on the right column, (Row-a) θ=150, β=0, (Row-b) θ=150, β=30, (Row-c) θ=150, β=60, (Row-d) θ=150, β=90, (Row-e) θ=150, β=120, (Row-f) θ=150, β=150

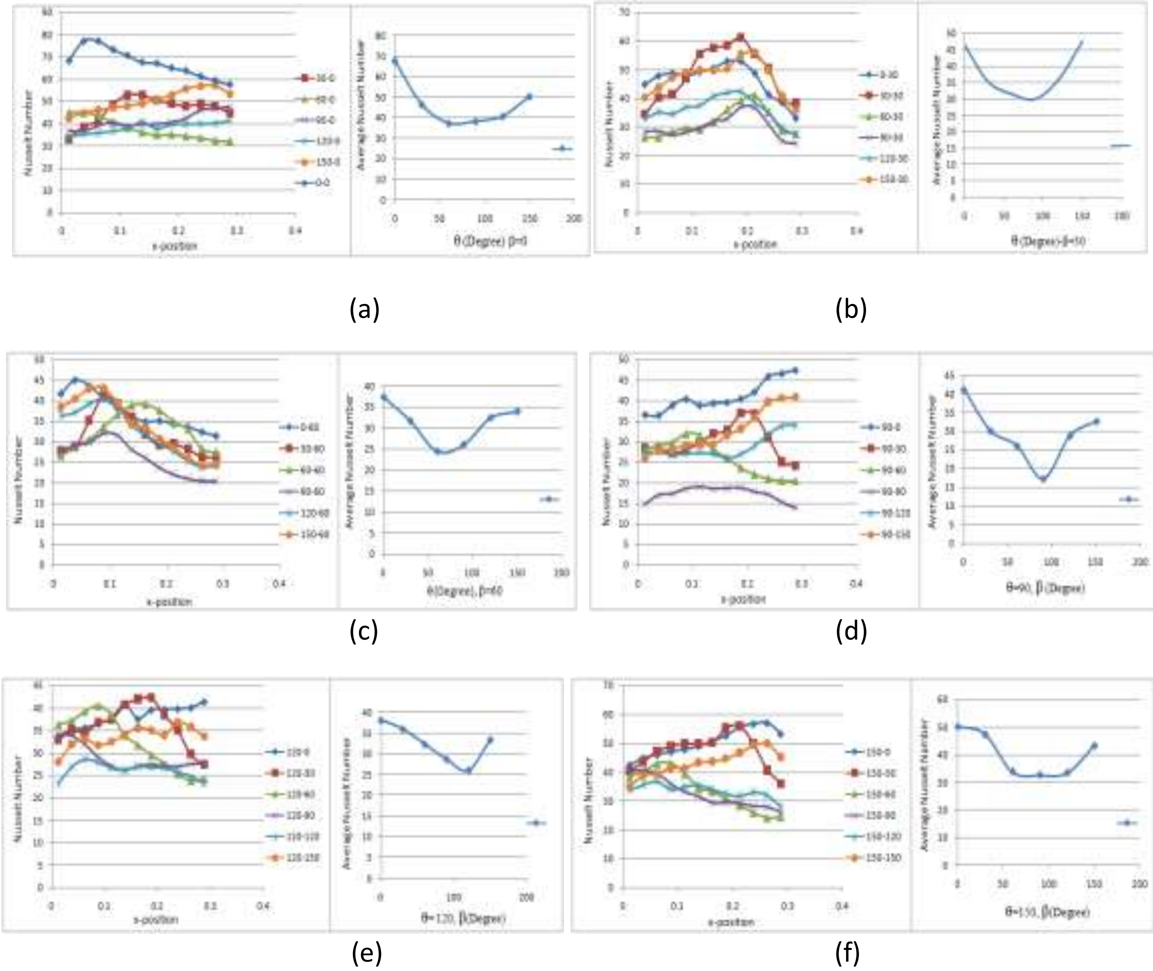



Figure (9): Variation of local Nusselt Number with the x-distance for different baffles orientation θ and β.

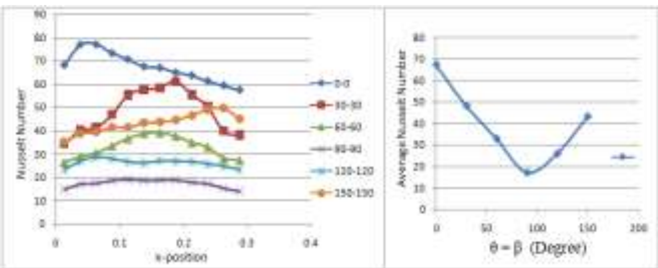



Figure (10): Variation of average Nusselt Number for the cases at which θ=β.

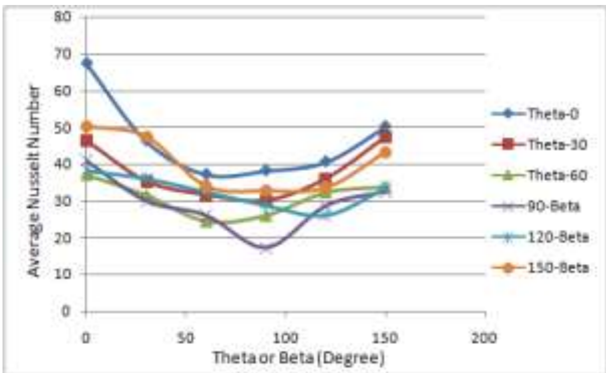



Figure (11): Variation of average Nusselt Number for the cases of variable (θ) or (β), and fixed the other.

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