20th Australasian Fluid Mechanics Conference
Perth, Australia
5-8 December 2016

Effects on Natural-Convection Heat Transfer of a Partial Partition in a Cubic Enclosure

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Abstract

Experiment and computation have been conducted in this work to investigate the effects of a partial partition on the natural-convection heat transfer in a cubic enclosure. The enclosure is similar to one that houses electronic equipment; and the partial partition whose orientation with respect to the horizontal can vary is attached to a vertical wall of the enclosure. Such partial partition could in reality be used to shield a region of the enclosure from excessive heat, or represent an object that is different to the housed equipment. The enclosure's floor has parallel bands that are heated to a constant, high temperature, to mimic rows of heated equipment; and the bands are separated by gaps that are kept at a lower temperature that is also constant. The enclosure's ceiling has a constant temperature which is lowest. All vertical walls of the enclosure and the partial partition are assumed to be adiabatic. Rayleigh number is fixed in this work. The software package Fluent-ANSYS15 has been used for a 2-D computation; and the standard k-epsilon turbulence model has been employed. Depending on its orientation, the partial partition has been found to alter significantly the flow field which in turn affects the heat transfer from the hot bands on the floor. Also, fairly good agreement has been obtained between experimental measurements and computational results.

Introduction

Natural-convection flow inside enclosures has many applications ranging from solar energy collectors, heat exchanger systems, ventilation and heating of buildings, temperature control of electronic equipment, etc. In recent years, many studies have been conducted on fluid flow and heat transfer inside enclosures with partitions of different configuration. One incentive is to improve the insulation properties of fluid layers, and controlling the heat transfer through an enclosure using partitions located and oriented appropriately. Lakhal et al [4] studied numerically natural-convection heat transfer in inclined rectangular enclosures with conducting fins attached to the heated walls over a range of Rayleigh-number (Ra) value and aspect ratio of the enclosure. The results indicate that the heat transfer through the cover is considerably affected by the fins. Sarris et al [8] investigated numerically natural convection in a glass-melting tank heated locally from below. Effects of geometry and position of the heated strip and tank's geometry on the flow patterns and heat transfer were investigated for Ra in the range 10^2 - 10^7. A numerical study was carried out by Bilgen [1] in differentially heated square cavities, which are formed by horizontal adiabatic walls and vertical isothermal walls with a thin fin attached. It was found that Nusselt number (Nu) is an increasing function of Ra, but a decreasing function of fin length and conductivity ratio. Rouger and Penot [7] experimentally investigated natural convection of air in a differentially heated cavity, containing an obstacle at the ceiling (lintel). It was found the lintel affects significantly the thermal field in the upper part of cavity, especially the stratification parameter. Costa [2] numerically studied natural convection in partitioned square enclosures filled with air. Two partitions of finite thickness were used. The overall Nu was obtained and compared from different placements and lengths of the partitions, varying Ra and different thermal boundary conditions imposed to the enclosure. Onyango et al [5] studied numerically natural convection in a three dimensional square cavity with a heat source positioned in the middle of the bottom surface. Two opposite vertical walls and the top wall had a low, constant temperature, while the other two vertical walls and the non-heated part of the bottom surface were insulated. The result showed that the position of the source of heat is one of the most important parameter on flow and the temperature field.

In this work, experiment and computation are conducted to investigate the effects of a partial partition on the natural-convection heat transfer in a cubic enclosure. The enclosure is similar to one that houses electronic equipment; and the partial partition whose orientation with respect to the horizontal can vary is attached to a vertical wall of the enclosure. The enclosure is heated on the floor with separated, parallel high-temperature bands to mimic rows of heated equipment. The objective is to determine the influence of the inclination angles of the partial partition on the flow and heat transfer characteristics of the air inside the enclosure.

Computation

Computation has been conducted on a 2D model of a cubic enclosure as per Figure 1. An adiabatic partial partition could pivot through 180^o about the mid-point of the left vertical wall of the enclosure. Both vertical walls are adiabatic, whereas the bottom side exposed to step function heating with alternating temperatures of 358K and 381K respectively, in 12 steps as shown in Figure 1. The top wall (roof) is isothermal at 275K. The enclosure filled with a fluid of Prandtl number Pr = 0.725. Ra value is 10^9 indicating that the buoyancy-induced flow inside the enclosure is turbulent. Other fluid properties correspond to those of air and assumed to be constant; but Boussinesq approximation applies for the temperature-induced change in density giving rise to buoyancy. A steady-flow RANS scheme is used with the standard k-ε turbulence model and standard wall functions; and governing equations are those associated with turbulent natural convection. The other boundary condition is thus: Non-slip condition on all surfaces (U=0, V=0)

The CFD software Fluent-ANSYS15 has been used, with 8,200 finite-volume-method cells. This 8,200-cell grid has been seen to be adequate. Good numerical convergence has also been obtained. Local and average Nu values are numerically obtained; those are then compared with experimental results to validate the computational work. Also the velocity field and temperature distribution are obtained and shown.

Experimental Setup

The experimental study was carried out in a cubic enclosure (30x30x30cm) figure (2). The three sides of the enclosure were constructed by the use of low thermal conductivity block wood and insulated by (25mm thickness) glass wood insulation. The front side is constructed by the use of double pan glass window to allow visualize. The top side is constructed used pure aluminium sheet (0.1mm) fabricated as fully closed container with gate use as crashing ice-vessel, and covered from all outsides by (25mm thickness) glass wood insulation to insure constant temperature of the top wall at about 3^{o}C $\pm 0.5^{o}$C. A drainage hole at the bottom of the crashing ice-vessel join to a drain pipe for the purpose of drain molten ice water continually and prevents the forming of high temperature film of water beneath the crashing ice. After steady state condition has 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 amount of heat transfer from the bottom side of the enclosure to the top side can be evaluated. This can be used as an indicator to the accuracy of the insulation. The bottom wall of the enclosure equipped with a heating system constructed:

1- The heating surface produce from pure Aluminium plate 8mm thickness to insure uniform temperature distribution throughout the heating surface. To produce a step function of temperatures distribution, an uncoated and coated strips formed by the used of low conductivity coating material which allows to get the step temperatures distribution in 12 steps of 381K and 358K respectively as a result of increasing the thermal resistance in the coated strips

2- An electrical resistance heater fixed under the aluminium sheet, constructed from strips of heating element (1mm) width made of chrome-nickel alloy with resistance (10 ohm/m) and wrapped with (5 mm) pitch around a three (100x300 mm) mica sheets of (0.5 mm) thickness, to ensure the electrical insulation. The overall resistance of each one of the three heaters is (32 ohm).The heaters is carefully mounted on a (320x320 mm2) mica sheet and covered with other (320x320 mm2) mica sheet to prevent the electrical contact. The main heater separated in three parts to insure the control on the temperature to be constant throughout the heating plate.

3- Two guard heaters produced in the same manner of the main heater (ring heater 55 ohm, and base heater resistance 65ohm) used to prevent heat transfer in the lateral and downward directions.

4- The heaters are covered from the sides and from bellow by a glass wool of (80 mm) thickness to reduce the heat loss. All heaters system is bounded with wood box (25mm thickness) to prevent any heat loss from bellow and from the lateral sides.

Nine thermocouples were interplant in the main Aluminium plate to insure a constant temperature distribution in the plate, and eight were interplant in the two guard heaters four in each one for the purpose of controlling the electrical current that must be flow throw each heater to insure a constant temperature in the heating plate. The thermocouples were calibrated to measure a temperature difference error of about ($\pm 0.3^{o}$C). The three heaters were supplied with AC-current via five voltage regulators. The voltage and current supplied to each heater was measured with a calibrated volt meters and ampere meters at an accuracy of about (±1.2%). The net power supplied to the main heater as a source of heat enters to the enclosure was (380 W/m^2). This power represented the electrical power measured from the reading of the volt and ampere meters.

An adiabatic partial partition with length B=0.135m located in the middle of the left vertical wall and inclined with different inclination angles denoted by θ, ($0^{o} \leq \theta \leq 180^{o}$) are used. The non-heat conductive partial partition was fabricated from thin metal coated from the two sides with rubber with overall thickness (t=0.005m).The enclosure is filled with air.

For the purpose of measuring the local heat transfer coefficient above each strip in the direction perpendicular to the partial partition orientation, the temperatures inside the enclosure in three locations (strip surface, 0.2mm & 0.4mm above the hot wall surface) of the 12 strips were measured using a sheathed thermocouple probe type(TD745) with accuracy of ($\pm 0.2^{o}$C). The sheathed thermocouple probe could be move parallel to the hot strips from a small hole in the side of the enclosure. These measured temperatures allows find the temperature gradient above each strip, and then find the local heat transfer coefficients and local Nusselt number according to the following equations [3]:

$$Nu_x = -\frac{\left.\frac{\delta T}{\delta y}\right|_{y=0} .H}{T_h - T_c} \tag{1}$$

Nusselt number can also find according to the local heat flux in each location according to:

$$Nu_x = \frac{q''H}{k(T_{h_x} - T_c)} \tag{2}$$

And the average Nusselt number can be calculated according to:

$$\overline{Nu} = \int_0^H Nu_x dx \tag{3}$$

Rayliegh number can be calculated according to:

$$Ra = \frac{g\beta(T_h - T_c)H^3}{\upsilon^2}\Pr \tag{4}$$

Error Analysis and Repeatability check:

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,

$$Nu = f(\delta y, \delta T, T_h, T_c, H) \tag{5}$$

According to the uncertainty analysis given by Doebelin [3], the uncertainty in the *Nu* can be given by,

$$\Delta Nu = \left|\Delta(\delta y)\frac{\partial Nu}{\partial(\delta y)}\right| + \left|\Delta(\delta T)\frac{\partial Nu}{\partial(\delta T)}\right| + \left|\Delta T_h\frac{\partial Nu}{\partial T_h}\right| + \left|\Delta T_c\frac{\partial Nu}{\partial T_c}\right| + \left|\Delta H\frac{\partial Nu}{\partial H}\right| \tag{6}$$

Nusselt number uncertainty analysis shows that the maximum error is (±0.352). Repeatability check details of the temperature distribution measured for three times repeated tests for each run tests, show that the percentage difference in the readings is not exceed 5% for all readings.

Result and Discussion:

In this section the experimental and numerical results introduce. Average Rayleigh number is calculated using equation (4) ($\overline{Ra}$ =1.5x10^9). This $\overline{Ra}$ refers that the natural heat convection is in its turbulent mode [6]. The effect of the turbulence numerical solution is modelled using (k-ε) model. The change in the flow field inside the enclosure due to the inclined partial partition with different angles is the main concern to understand the behaviour of the natural heat convection, which perse produce due to the fluid density change as a result of heat exchange with the heating wall. This kind of heating (step function temperature change) represent as a simulation to the electronic equipment's fixed in the bottom of the cubic box for the purpose of controlling the heat dissipate from it.

Figure (3) show numerically the contours of velocity magnitude (m/s) for the enclosure with and without partial partition with different inclination angled. Part (A) explains the air behaviour in the enclosure without partition which shows a circulation of air anticlockwise. By dividing the bottom of the enclosure to 12 strips and numbering each strip starting from the left, it seems that maximum velocity occurs in the strips number 5 and 6, that is explain the high exchange of heat at that strips. It is also shows that the strips of high temperature (381K) in the sequence make a jump in the velocity level which increase the heat exchange within it. Due to the circulation of flow anticlockwise the upstream of the flow in the first strips (1-6) causes a higher heat exchange compare with the strips (7-12). The space above the first strip number (1) shows a small revise circulation portion in the corner of the enclosure causes a reduction in the heat exchange, whereas the space above the last strip number (12) shows stagnation causes lower heat exchange in compare with all other strips. Part (B) represents the enclosure with a partition inclined by 45^o. The partition causes an obstruction between the hot wall and the cold wall of the enclosure to the strips (1-5) which causes reduce the heat exchange in these strips. Maximum velocity can be noticed over the strips (6). Strips (7-10) also show high level of heat transfer due to the effect of the anticlockwise circulation of the air flow which imping strip (6) on the hot surface assuming it as a leading edge after sliding through the partition. The left corner under the partition shows a revers flow cause reduce in the heat exchange. Part (C) represents the enclosure with a partition inclined by 90^o. This case shows that the enclosure is divided in to two flow field, the upper part circulate anticlockwise whereas the lower part circulate clockwise, means that there is no direct contact in the circulation of air in the lower part with the cold surface of the enclosure, results in reduce the heat exchange compare with the previous cases. Due to the anticlockwise circulation of the air near the hot wall, the flow leading edge relative to the hot wall starts from the right of the enclosure. The higher flow velocity noticed in strips (7-9). That is explained the reason for the high level of heat exchange shifted to the right side of the enclosure compare with previous cases. Part (D) represents the enclosure with a partition inclined by 135^o. In this case a film of Simi stagnation air formed in the upper portion of the enclosure cases a kind of insulation between the main flow field and the cooling surface, causes a reduction in heat exchange inside the enclosure. The main flow field circulate clockwise and the maximum flow velocity noticed above strips number (5-7). The revers circle formed in the right corner is bigger than the previous case.

Figure (4) shows numerically the contours of Static Temperature (K) for the enclosure with and without partial partition with different inclination angles. Part (A) represent the enclosure without partition, shows clearly the cooling of the equipment is very effective, the green colour which dominant the enclosure clarified that situation. The green colour cold flow field is come very close to the bottom surface of the enclosure causes effective cooling to the hot wall. Part (B) of the partition inclined by 45^o shows an ineffective cooling field of yellow colour in the left side of the enclosure, whereas in part (C) of the 90^o inclination angle and part (D) of the 135^o inclination angle show that the ineffective field of yellow colour includes whole the bottom side of the enclosure which reduce the heat exchange.

Figure (5) shows numerically the heat flux on the heating wall under different inclination angles of the partial partition. What was explained previously on the behaviour of the flow field clarify these graphs. Line (0.0^o) shows high level of heat exchange between the heating surface and the air flow in the enclosure. The highest amount of heat flux occurs in the left strips which represents the leading edge of the flow with respect to the hot wall and reduce gradually at the trial edge. Maximum heat flux was about 1200 W/m^2. Line (45^o) with partition inclined by 45^o shows reduction in the heat flux to maximum of 900 W/m^2 reduces to about 700 as a second maximum value. The left side shows minimum value of heat flux. Line (90^o) shows maximum heat flux of 800 W/m^2 near the right side, whereas part (D) shows maximum heat flux of 700 W/m^2 at about the middle of the enclosure. Line (135^o) collected part (A-D) to shows a comparison between each case.

To validate this work, figure (6) shows a comparison between the experimental results depends on the experimental tests and numerical result given by Ansis 15. The experimental local Nusselt number calculated according to equation (1) after getting the temperature gradient on the heating surface on each 12 slips. The experimental test repeated three times for each case of partition inclination angle. Each experimental text required at least 90 minutes to reach study state condition. The steady state condition materializes when the 9 thermocouples interplant in the heating Aluminium plate measured the same temperature measured in the 4 thermocouples interplant in the ring guard heater and the 4 thermocouples interplant in the base guard heater. The thermal physical properties of the air are measured according to the bulk mean temperature of the average temperature of the heating wall and the cooling wall temperature. The figure shows that the change of local Nusselt number with x-position experimentally and numerically have good match. This gives full idea about how the equipment's can be arranged inside a cubic enclosure according to the amount of heat flux have to remove from each, and what inclination angle of the partial partition must be fixed for that purpose.

Conclusion

This investigation represents a try to perform a method to arrange number of equipment inside an enclosure in a manner depend on the amount of heat flux produce by each one so that finally can keep all in the same design temperature for these equipment's. The non-conductive partial partition which attached to the non-conductive left side of the enclosure can be rotate to different inclination angles to get different thermal situation inside the enclosure. All vertical sides of the enclosure assumed non-conductive, whereas the top is cooled to a specified low temperature, and a step function of high temperatures applied on the bottom side. The important high marks in this work are:

1-The results show that there is a good match between the experimental and numerical results.

2- The inclined non-conductive partial partition has a significant effect on the thermal condition of the enclosure.

3- Increasing the inclination angle of the partition allow increasing the cells inside the enclosure.

4- The inclined partial partition can be assumed as damping mean to the flow field velocity in a control manner help to keep the temperature of different power out equipment's in the same temperature.

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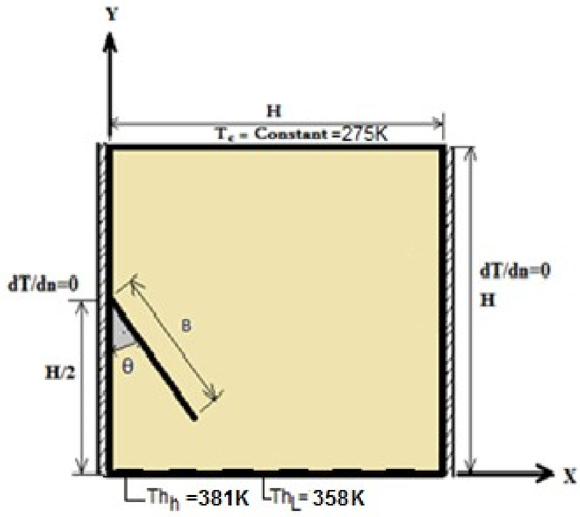
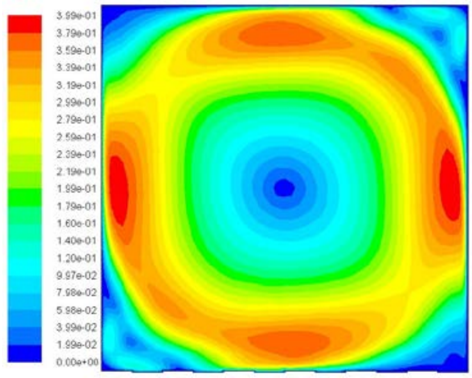
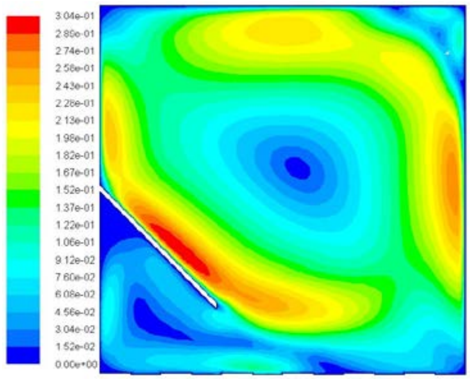


Figure (1) Schematic of the square cavity with the attached partial partition

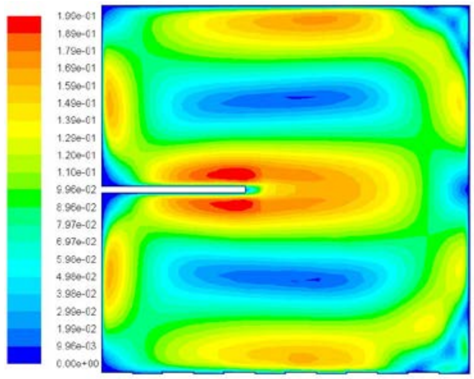
Figure (2) Experimental Apparatus



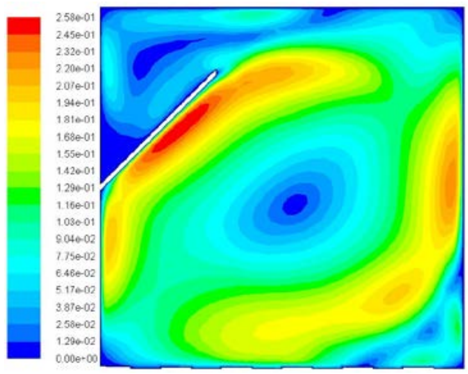
(A) Contours of Velocity Magnitude (m/s)
Enclosure without partial partition



(B) Contours of Velocity Magnitude (m/s)
Enclosure with partial partition inclined by -45^{o}-

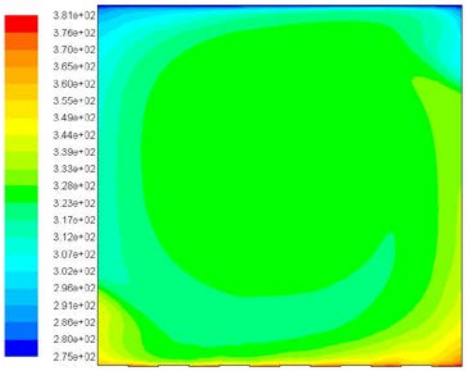


(C) Contours of Velocity Magnitude (m/s)
Enclosure with partial partition inclined by -90^{o}-

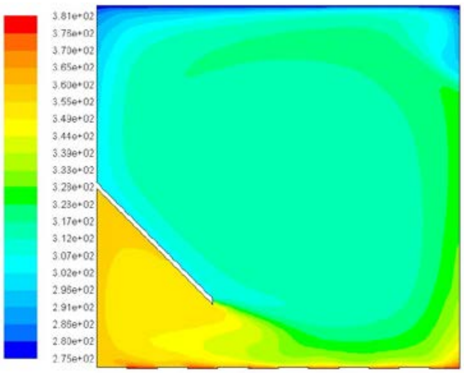


(D) Contours of Velocity Magnitude (m/s)
Enclosure with partial partition inclined by -135^{o}-

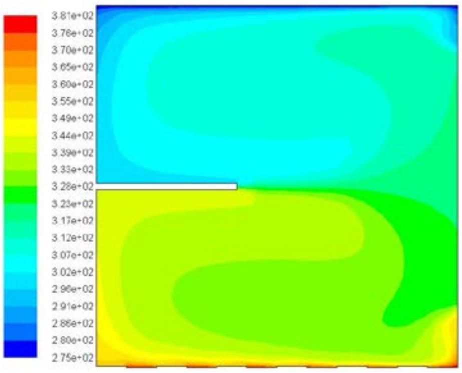
Figure (3) Contours of Velocity Magnitude (m/s) for the enclosure without partial partition and with partial partition with different inclination angled.



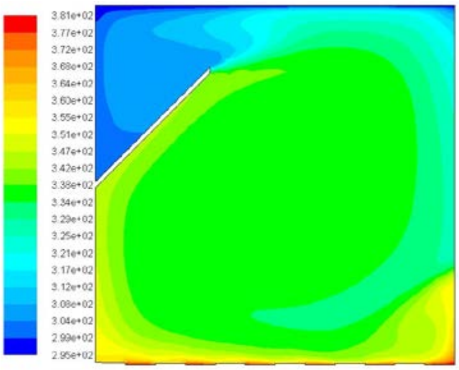
(A)Contours of Static Temperature (K)
Enclosure without partial partition



(B) Contours of Static Temperature (K)
Enclosure with partial partition inclined by -45°-



(C) Contours of Static Temperature (K)
Enclosure with partial partition inclined by -90°-



(D) Contours of Static Temperature (K)
Enclosure with partial partition inclined by -135°-

Figure (4) Contours of Static Temperature (K) for the enclosure without partial partition and with partial partition with different inclination angled.

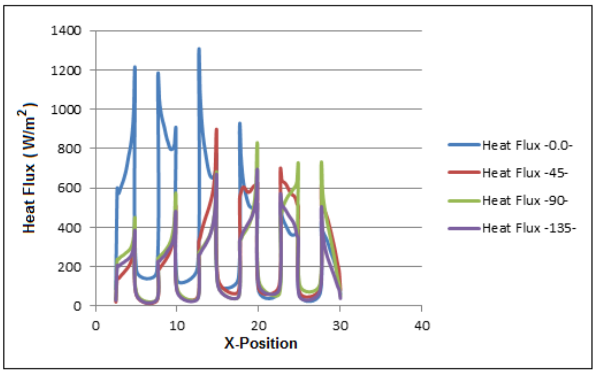
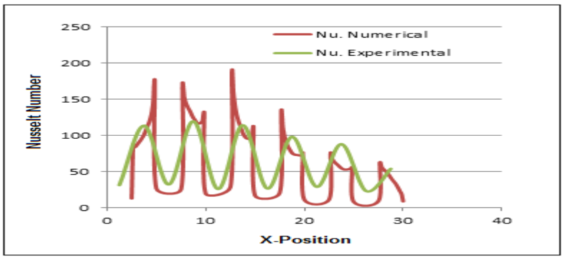
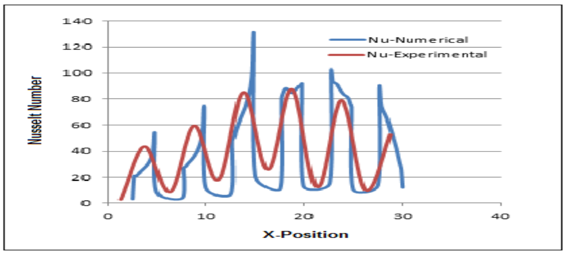


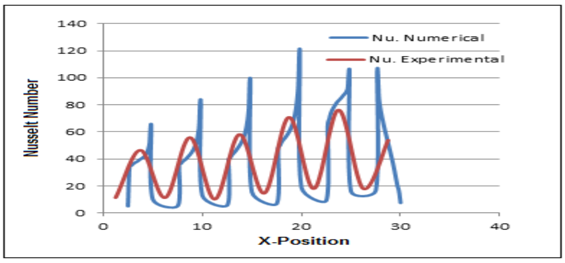
Figure (5) Heat Flux on the Heating Wall Under different inclination Angles of the partial partition (Numerical results).



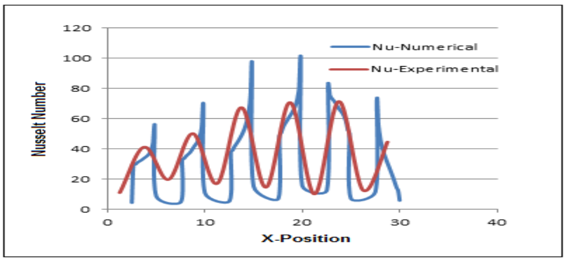
(A) Enclosure without partial partition



(B) Enclosure with partial partition inclined by 45°



C) Enclosure with partial partition inclined by 90°



(D) Enclosure with partial partition inclined by 135°

Figure (6) shows a comparison between the experimental and numerical results