ISSN-L: 2223-9553, ISSN: 2223-9944
Vol. 4 No. 2 March 2013

THE INFLUENCE OF INCLINATION ANGLE ON NATURAL CONVECTION IN A RECTANGULAR ENCLOSURE

Thamer Khalif Salem

Mechanical Engineering, College of Engineering,
Tikrit University,
IRAQ.

thamer_khalif@yahoo.com

ABSTRACT

An experimental work was conducted to investigate the influence of inclination angle on natural heat transfer in a rectangular enclosure. The enclosure length was (100 cm) with a square cross section (10×10 cm). All the enclosure surfaces were insulated except the upper surface, where half of it was cooled and the other half was heated. The tests were done for three inclination angles (15^o, 45^o and 75^o), three mass flow rates (0, 0.0275 and 0.062 kg/sec) and five heat fluxes (4.92, 19.2, 39.96, 69.6 and 108 W/m2) for ($1.95\times10^5 \leq Ra \leq 4.96\times10^6$).

It was found that the highest Nu_x was at (x/L=0.33) where the highest heat transfer occurred between the hot air and the cold surface (high temperature difference). It was also found that as the inclination angle increased, the Nu_x and Nu increased. This is because the falling cold air disrupted the rising hot air which led to an increase in the heat transfer between the hot air and the cold surface. Increasing the mass flow rate and the heat fluxes had increased the heat transfer through the enclosure. A correlation was proposed to relate Nu with Ra and the inclination angle. A good agreement was found when comparing the experimental data and the present correlation with a previous work.

Keywords*:* Natural Convection, Inclined Enclosure, Heating and Cooling

INTRODUCTION

Natural convection has a considerable importance by many researchers for the past few decades because of its many engineering and physical applications. These applications include heating and cooling spaces, cooling of electronic components and solar collectors. Various thermal conditions, orientation and dimensions have been studied theoretically and experimentally for rectangular enclosures.

Experimental data were presented by Tso *et al.* [1] for laminar natural convection cooling of water in a rectangular cavity with a 3×3 array of heaters on one wall and the cavity at various inclined angles. Generally, heaters in the same row had close temperatures due to weak velocity in the horizontal direction. Numerical simulations in 2-D and 3-D were also reported. The main conclusions obtained were heaters in the same row that had almost the same temperature. This is due to weak velocity in the horizontal direction, Complex flow pattern evolution, and flow field. Temperature fields were observed due to interactions of buoyancy forces tangential and normal to the heater surfaces, in the horizontal orientation, with an increase in Ra*. The flow pattern evolutions were complex and identified with three main characteristics, which are toroidal convection, bimodal convection, and R-B convection. As for the increase in heat transfer, the horizontal orientation was best because the lowest Nu among the three rows was higher than those of other orientations. As for inclining the enclosure towards the horizontal orientation (heated from below), the flow and temperature fields became complex and distorted. Three-dimensional effects such as fluid flow in horizontal direction and distorted temperature field began to appear and intensify. This greatly enhanced the heat transfer for the top row, edge effect increased heat transfer greatly, and they that for most liquids (except metal liquids), whose Pr is greater than 1, the effects of Pr are negligible, and for most gases whose Pr are about 0.7, the effects of Pr are small.

Aminossadati and Ghasemi [2] had investigated natural convective laminar flow numerically in a two-dimensional and square enclosure at various angles of inclination respect to horizontal. Two adjacent walls of the enclosure are insulated and the other two were kept at different temperatures. The influence of Ra representing the effects due to the differential heating of the enclosure walls as well as the effect of inclination angle on natural convection flow were studied. The results of flow and temperature fields showed different behaviors at various inclination angles and Ra. At low *Ra*, the flow field did not show significant changes with the inclination angle. They found that as the inclination angle rose, the average temperature in the enclosure started to increase until it reached its maximum and then decreased. The rate of decrease was more appreciable at high Ra. Basically, at all Ra, as the inclination angle rose the average temperature in the enclosure started to increase until it reached its maximum and then decreased. They also found that average Nusselt number remained unchanged at low Ra.

The natural convection of water in inclined side-heated rectangular box was investigated by Dekajlo and Kowalewski [3] both experimentally and numerically. The cavity had aspect ratio L/H = 3 and the two opposite isothermal walls. It was kept at different temperatures and four other adiabatic walls. The working fluid was water. The enclosure inclination varied from 0° to 90°. The modified Ra varied in the range of $1\times10^6 < Ra_y < 7.5\times10^6$. The parametric numerical study designates three distinct flow regimes: the steady state flow regime was for high cavity inclinations 60°-90°, the oscillatory flow regime was for intermediate inclinations 30°-50° and the transition to Rayleigh-Bénard flow regime was for small inclinations 0°-20°. Their experimental results and numerical simulations performed for the first flow regime indicated strong three-dimensionality of the flow structure, excluding the possibility for simplified, two-dimensional numerical modeling. The two remaining flow regimes appeared to be strongly time-dependent and only qualitative comparison with the experiments was presently available.

Tasnim and Mahmud [4] had numerically reported the buoyancy induced flow and heat transfer characteristics inside an inclined L-shaped enclosure. A control volume based finite-volume method was applied to discretize the governing equations with collocated variable arrangement. SIMPLE algorithm was used and the system of equations was solved by Stone's SIP solver with full multigrid acceleration. At particular range of Ra (conduction-dominated zone), they found heat transfer rate to be equal when Ra distribution is independent. Average heat transfer varied linearly in the convection-dominated zone, which was determined by another range of Ra. Influence of angle of orientation on the average heat transfer rate in the conduction dominated zone was insignificant. However, in the convection-dominated zone, the Nu-angle of orientation relationship was characterized by one angular location for maximum and two angular locations for minimum values of the Nu and two symmetrical parts of the profile: one with respect to angle of orientation 90° and another with respect to angle of orientation 225°. Angle of orientation did not influence significantly the Nu-aspect ratio relationship. But a significant influence of *Ra* was observed on the Nu-aspect ratio relationship which showed a complicated behavior, especially in the convection-dominated zone.

Steady-state natural convection taking place in rectangular cavities filled with air was studied by Bairi *et al.* [5] both experimental and numerically. Both hot and cold active walls of the cavity were maintained isothermal at temperatures T_H and T_C, respectively and the other walls (channel) that close the cavity were adiabatic. Different angles of inclination of the cavity from 0° to 360° were considered. This included the analysis of several significant situations corresponding to inclinations of 0° vertical active walls, 90° hot wall down, Rayleigh–Benard convection and 270° hot wall up, and pure conductive mode. Two aspect ratios A= L/H = 0.75 and 1.5 were treated. The numerical study was carried out by means of the finite volume method. They concluded that the temperature and velocity profiles obtained were congruent with the expected heat convection exchanged characterizing this kind of cavities. In addition, the comparison between their experimental data and the numerical model outputs, it was showed that the edge effects were small as far as the convection heat transfer was concerned. Although they affected the local flows in the vicinity of the side walls, their overall effect was limited and the two-dimensional model was suitable enough to describe the problem. This had the advantage of limiting the computing time compared to 3-D modeling.

Bairi *et al.* [6] had presented work related to the thermal control by natural convection of the electronic assemblies contained in confined spaces. They had performed a numerical and experimental study to determine the thermal behavior in a cavity where the electronic assembly was a wall made of discrete hot sources under dynamic operation. The treated cavity was an air-filled cube that consists of two active opposing walls connected by a channel enclosure. The channel was adiabatic and the two active walls were the responsible of the natural convection flow inside the cavity. The cold wall was maintained isothermal at temperature T_C. The second active wall consists of 5 bands of which 3 are heated and maintained at T_H, separated by 2 other adiabatic bands. Calculations in steady-state regime were carried out by means of the finite volumes method. The numerical results had been compared with experimental measurements and deviations encountered were slight. Differences between the calculated and measured Nu were of 5% on average, which was within the margin of the experimental and numerical uncertainties. The temperature distribution on the passive walls originated by the natural convection flow had the same aspect in both cases and the computed values differed from the calculated ones only in about 0.4°C on average. In general terms, heat exchanges occurred with different intensity on the three heated bands, but this difference tended to disappear when the angle of inclination increased and for high Ra values. They found out that heat exchanges were 10% lower on average than those corresponding to cavities with an entirely isothermal hot plate.

Effects of inclination angle on natural convection heat transfer and fluid flow in a two-dimensional enclosure filled with Cu-nanofluid had been analyzed numerically by Abu-Nada and Oztop [7]. The performance of nanofluids was tested inside an enclosure by taking into account the solid particle dispersion. The angle of inclination was used as a control parameter for flow and heat transfer. It was varied from 0° to 120°. The governing equations were solved with finite-volume technique for the range of Ra as $10^3 \leq Ra \leq 10^5$. Results had clearly indicated that the addition of copper nanoparticles had produced a remarkable enhancement on heat transfer with respect to that of the pure fluid. Heat transfer was enhanced with an increase in Ra almost linearly but the effect of nanoparticles concentration on Nusselt number was more pronounced at low Ra than at high Ra. It was found that lower heat transfer was formed for 90°. But higher values of volume fraction became insignificant from the fluid flow point of view at this inclination angle. Effects of inclination angle on percentage of heat transfer enhancement became insignificant at low Ra but it decreased the enhancement of heat transfer with Nano fluid. Finally, the inclination angle was a good control parameter for both pure and Nano fluid filled enclosures.

The present work investigates experimentally laminar natural convection in an inclined rectangular enclosure. All its surfaces were insulated except the upper surface, where its lower half was heated and the upper half was cooled.

EXPERIMENTAL WORK

Figure 1 show that the experimental rig (a) a photograph and (b) a schematic drawing are used to investigate the effect of heating the lower half and cooling the upper half of the left surface of an inclined enclosure on natural convection inside the enclosure.

A rectangular enclosure with a square cross section area, which is made of galvanized iron oriented inclined, is used. The enclosure thickness (δ) and width (B) are (10 cm), its length is (100 cm).

The lower half of the left surface test section was heated electrically using an electrical resistance of (5 m) in length with a resistance ($1\Omega/\mathrm{m}$). The heater was supplied with an alternative electrical power monitored via a variable voltage transformer. An ammeter was used to measure the current passes through the heater and a voltmeter to measure the voltage

While the upper half was cooled by water using a cooling tank (10 × 10 × 50 cm). The cooling water flow rate can be controlled by two valves one for inlet and another for outlet water from the cooling tank. The water mass flow rate is measured by using a (1000 ml) glass container and stop watch.

In order to reduce the heat losses, the test section was thermally insulated with fiber glass of (1 cm) in thickness. The two ends of the apparatus section and other surfaces were insulated electrically and thermally using fiber glass.

Eighteen calibrated thermocouples (T-type) were used to measure the temperature: on the left heated and cooled surface (5 locations along the surface), the opposite inclined surface (5 locations along the surface), the temperature of the air along the center of the enclosure (5 locations), two thermocouples to measure the temperature of the inlet and outlet water from the cooling tank and a thermocouple to measure the laboratory ambient temperature.

CALCULATION PROCEDURES

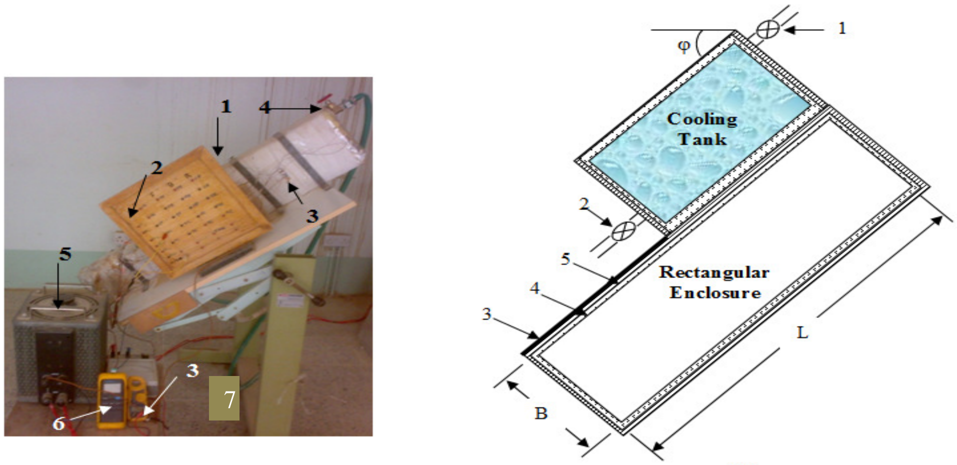
The test procedure can be listed as follow:

1. Control the water mass flow rate by the inlet valve.
2. The test section is set to the proper voltage and this is achieved using a variable transformer.
3. After (1.5-2 hr.) the system reaches the steady state condition. The enclosure surfaces temperatures, the inlet and outlet water temperatures, the water mass flow rate through the cooling tank, the laboratory temperature and the heater voltage and current have been registered.

The heat generated Qg was dissipating from the enclosure surface by convection and radiation.

4. $$Q_g = \frac{IV}{A_s} \qquad (1)$$
5. Where $A_s = 0.5 \times BL$
6. $$Q_g = q_c + q_r \qquad (2)$$

ISSN-L: 2223-9553, ISSN: 2223-9944
Vol. 4 No. 2 March 2013



(A)
1- Test section, 2- Meter panel. 3- Cooling tank. 4- Valve. 5- Voltage regular, 6- Ammeter, 7- Voltmeter.

(B)
1- Inlet water valve, 2- Outlet water valve, 3- Thermal resistance, 4- Thermal insulator, 5- Fibber glass.

Figure 1. (a) Photo graphs (b) Schematic drawing of the test section (B)

Where qc and qr are the fraction of the heat flux dissipating from the enclosure surface by convection and radiation, respectively which can be calculated as [8]:

$$q_r = \frac{\sigma\left(T_H{}^4 - T_C^4\right)}{\frac{2}{\varepsilon} - 1} \tag{3}$$

Where ε is the surface emissivity of the enclosure and it is estimated as 0.3 for galvanized iron [9].

Where the quantity of the heat absorbed by water Qw is calculated as follow [10]:

$$Q_w = \dot{m}_w C_{pw} (T_{wo} - T_{wi}) \tag{4}$$

Local convection heat transfer coefficient and local Nusselt numbers have been calculated indirectly measured respectively using the following equation [8]:

$$h_x = \frac{q_c}{(T_{Hx} - T_{Cx})} \tag{5}$$

$$Nu_x = \frac{h_x . \delta}{k} \tag{6}$$

Then the overall longitudinal average $\bar{h}$ can calculated as flow [8]:

$$\bar{h} = \sum_{x=1}^{5} h_x / 5 \tag{7}$$

The non-dimensional overall average Nusselt number and Rayleigh numbers are calculated as follow [8]:

$$Nu = \frac{\bar{h} . \delta}{k} \tag{8}$$

$$Ra = \frac{g\beta(T_{Hx} - T_{Cx}).\delta^3}{\nu\alpha} \tag{9}$$

Where all the air properties were indicated at the film temperature Tf = (TH-TC)/2.

RESULTS AND DISCUSSION

An experimental work was conducted to investigate natural convection heat transfer in an inclined rectangular enclosure, half of its upper surface was uniformly heated and the other half was cooled. Forty five tests were conducted for three inclination angles (15^{o}, 45^{o} and 75^{o}), three different mass flow rates (0, 0.0275 and 0.062 kg/sec) and five heat fluxes (4.92, 19.2, 39.96, 69.6 and 108 W/m^{2}).

In order to validate the results, a comparison was done with Kothandaraman and Subramanyan [11] for (0 kg/sec) case as shown in fig. 2. The recent results were (34%) at (φ=45º) differs from Kothandaraman and Subramanyan [11] since the latter case was for heated upper surface and cooled lower surface which can be acceptable.

As shown in fig. 3 the Nu_x distributes along the enclosure at (0 kg/sec) for the three inclination angles for different heat fluxes. The heavy cold air (x/L =0.17 →0.5) moves downward while the light hot air moves upward along the upper surface causing increasing in heat transfer between the hot air and the cold surface. The effect of heat losses is appeared in the terminal sides, at (x/L=0.17) and (x/L=0.83). As the inclination angle increases Nu_x is increasing because the falling cold air disrupted the rising hot air which increases the heat transfer between the hot air and the cold surface.

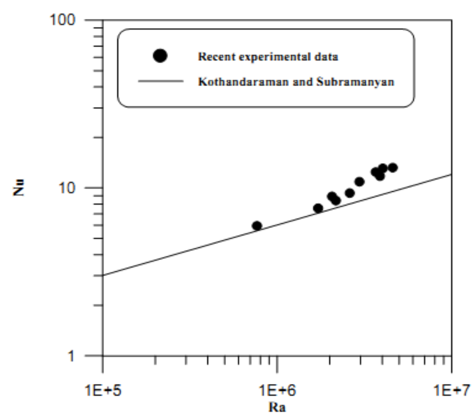


Figure 2. Comprision between precent experimental work with Kothandaraman and Subramanyan [11] at $\dot{m} = 0.0$ kg/s

Figure 4 shows the effect of the inclination angle on Nu_x at (0.062 kg/s). The same general behavior was observed. The Nu_x also increased as the inclination angle increased.

Figure 5 shows the effect of the mass flow rate on the Nu_x along the enclosure at (4.92 W/m^{2}) for the three inclination angles and different mass flow rates. At (x/L =0.33) the largest value is observed where the rising hot air meets the falling cold air, and at (x/L =0.67) the lowest value is obtained, where the hot air is dominating this part of the enclosure, and less heat transfer is obtained. Increasing the inclination angle increased the heat transfer rate because of increasing the disrupting of the cold air for the hot air.

As shown in fig. 6, compared to fig. 5, the Nu_x increased as the heat flux increased for the three inclination angle and for the three mass flow rates, since the amount of the convected heat is increased with increasing the heat flux.

As can be observed from fig. 7, the temperature ratio in the cold region was low because the temperature of the air in the center of the enclosure was low, so is the upper surface temperature. However, the surface temperature was high in the hot regions, which lead to an increase the temperature ratio. Increasing the inclination angle will increase the disrupting of

ISSN-L: 2223-9553, ISSN: 2223-9944
Vol. 4 No. 2 March 2013

the cold air to the hot air which will increase the temperature ratio. The same is happening when the mass flow rate is increased to (0.062 kg/sec) as shown in fig. 8.

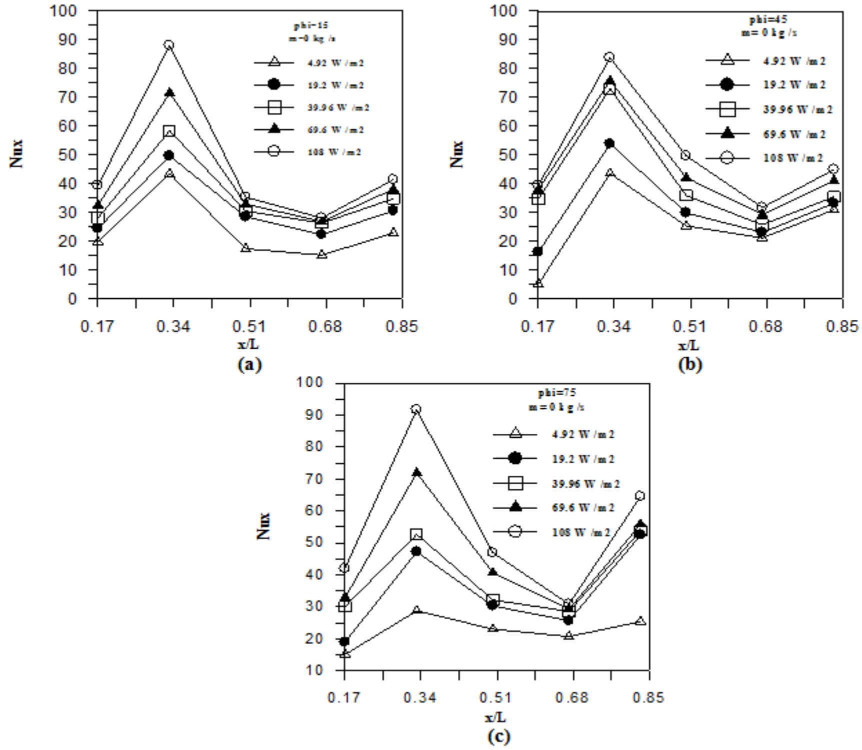


Figure 3. Effect of heat flux on local Nusselt numbers distribution along the enclosure at $\dot{m} = 0\,kg/s$: (a) 15 deg., (b) 45 deg. and (c) 75 deg

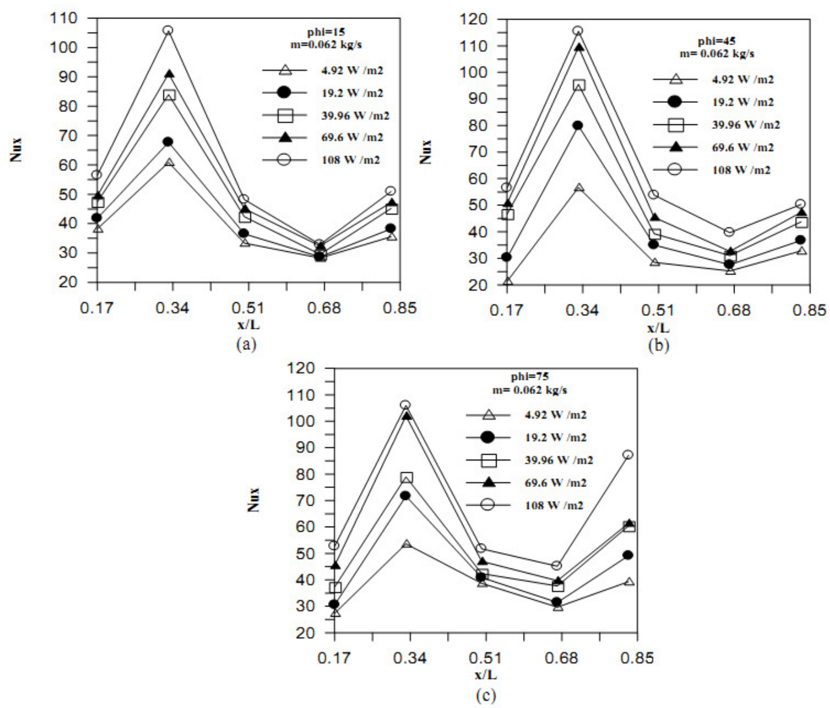


Figure 4. Effect of heat flux on local Nusselt number distribution along the enclosure at $\dot{m}$ =0.062 kg/s: (a) 15 deg., (b) 45 deg. and (c) 75 deg.

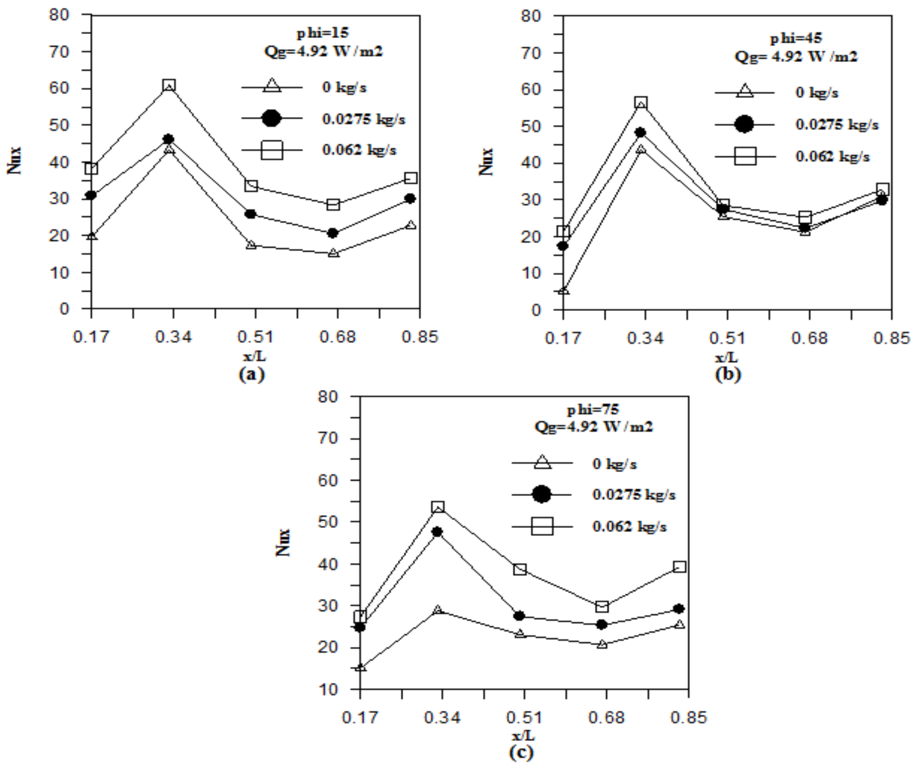


Figure 5. Effect of mass flow rate on local Nusselt number distribution along the enclosure at Qg=4.92 W/m^2: (a) 15 deg., (b) 45 deg. and (c) 75 deg.

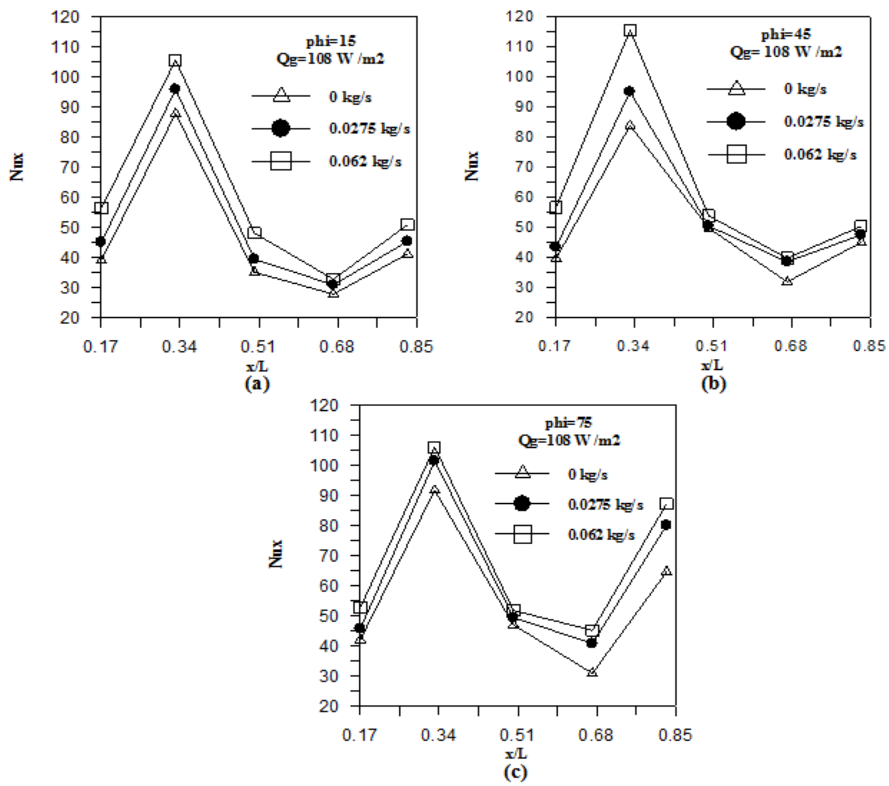


Figure 6. Effect of mass flow rate on local Nusselt number distribution along the enclosure at Q_g=108 W/m^2: (a) 15 deg., (b) 45 deg. and (c) 75 deg.

ISSN-L: 2223-9553, ISSN: 2223-9944
Vol. 4 No. 2 March 2013

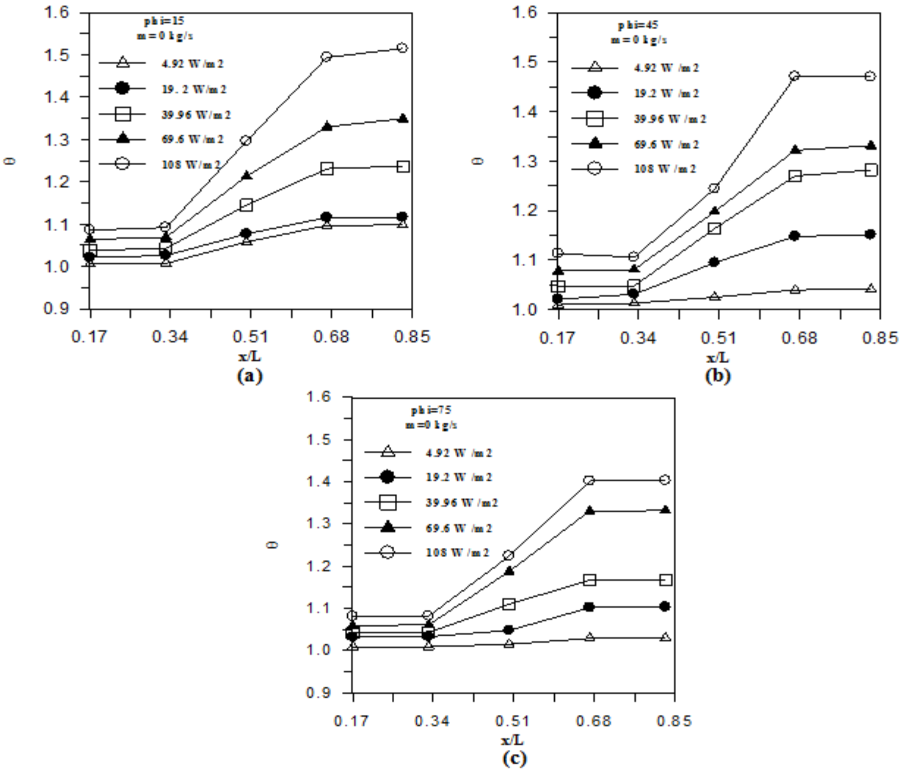


Figure 7. Effect of heat flux on the temperature ratio distribution along the enclosure at ṁ =0.0 kg/s: (a) 15 deg., (b) 45 deg. and (c) 75 deg.

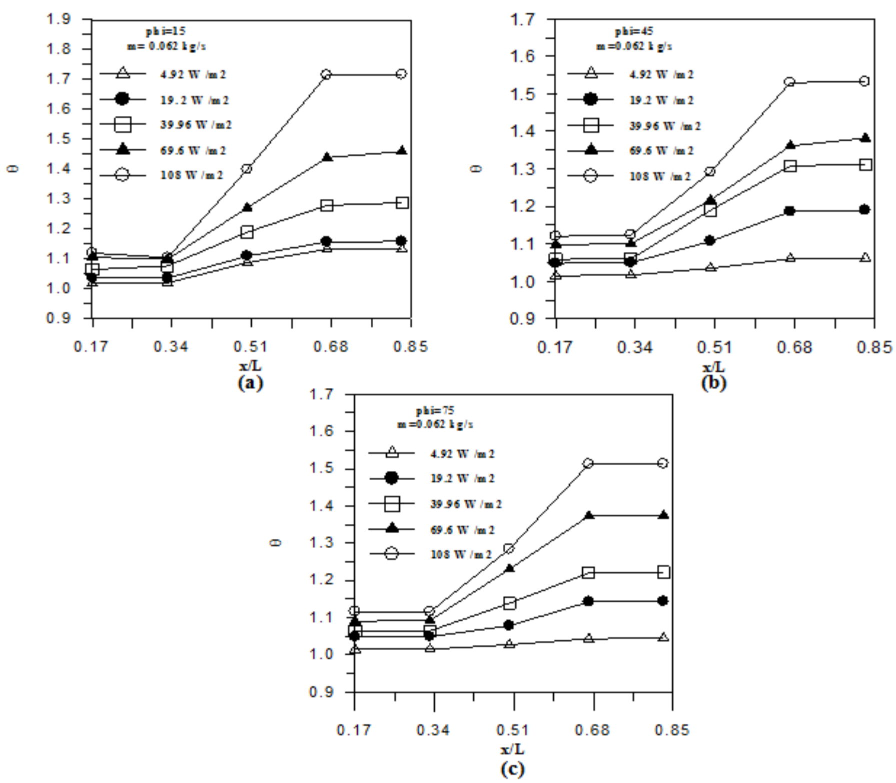


Figure 8. Effect of heat flux on the temperature ratio distribution along the enclosure at ṁ =0.062 kg/s: (a) 15 deg., (b) 45 deg. and (c) 75 deg.

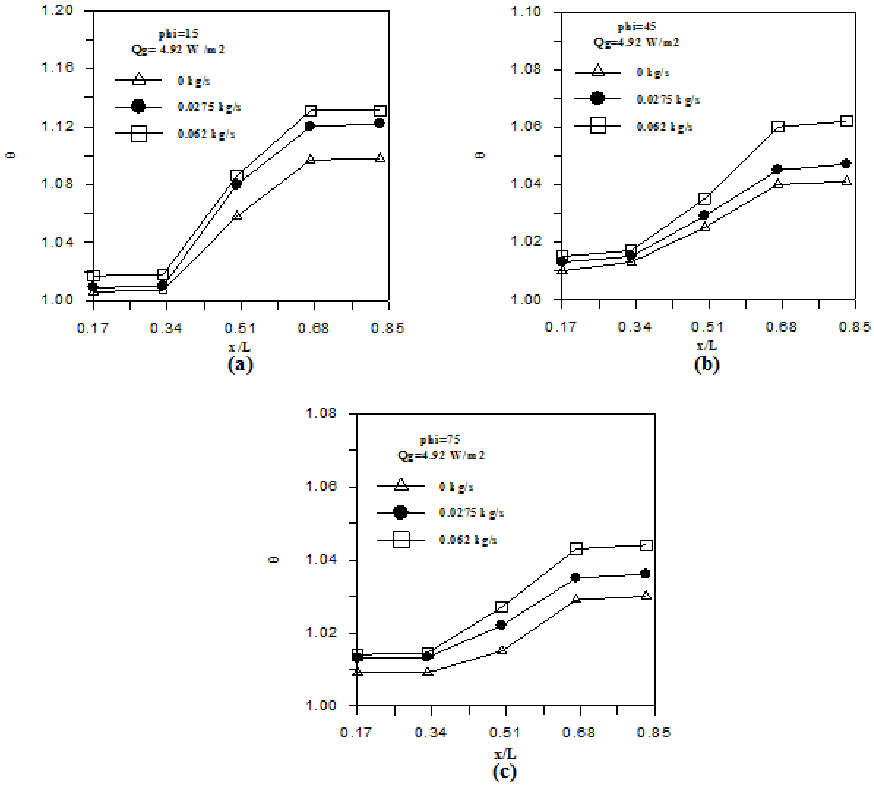


Figure 9. Effect of mass flow rate on the temperature ratio distribution along the enclosure at Q_g=4.92 W/m^2: (a) 15 deg., (b) 45 deg. and (c) 75 deg.

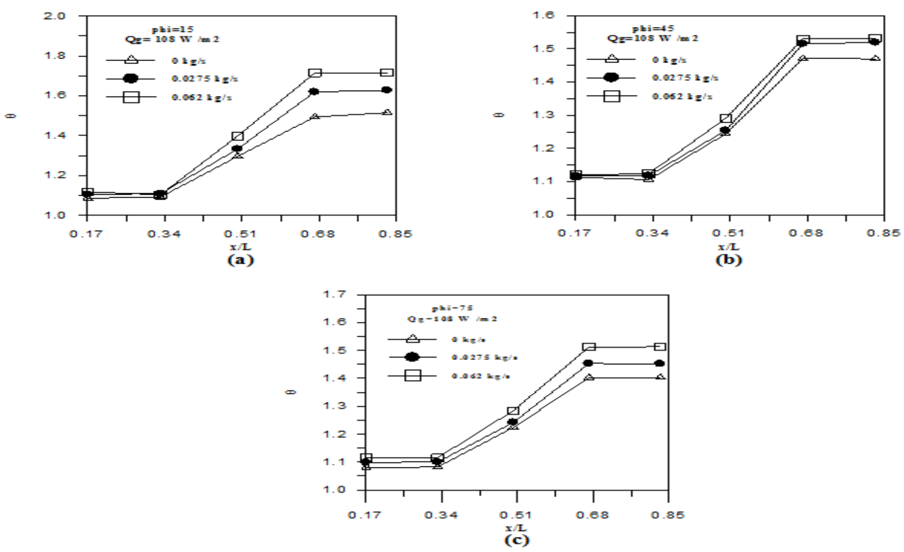


Figure 10. Effect of mass flow rate on the temperature ratio distribution along the enclosure at Q_g=108 W/m^2: (a) 15 deg., (b) 45 deg. and (c) 75 deg.

Figure 9 shows the effect of the mass flow rate on temperature ratio for the three inclination angles. As the mass flow rate increased, more heat was absorbed by the water, the air in the cold half became colder and fell down more rapidly and the hot air took its place which leads to an increase the temperature ratio. When the heat flux increased, more heat was supplied to the enclosure, and the temperature ratio increased as shown in fig. 10 for increasing heat transfer rate.

ISSN-L: 2223-9553, ISSN: 2223-9944
Vol. 4 No. 2 March 2013

As can be obtained from Fig. 11, as the inclination angle increases Nu also increases because the hot air spends more time moving near the surface –because of the disrupted caused by the cold air to the hot one- which increases the heat transfer.

Figure 12 supports this idea, since in this figure a comparison is done between the inclined case and the horizontal case Thamer [12]. This is happening because in the horizontal case the hot air can't go through the solid surface above it, so it hardly moves sideward to the cold half which leads to decrease the heat transfer rate.

The following correlation was proposed to relate Nu number with Ra number and the inclination angles:

$$Nu = \frac{0.094\, Ra^{0.318}}{(1+\sin\phi)^{-0.324}} \qquad (10)$$

For $1.95\times105 \le Ra \le 4.96\times106$ and $15° \le \varphi \le 75°$, with an error band of $\pm 30\%$.

Figure 13 compares the regression of the lines and experimental data with the correlation obtained eq. (10).

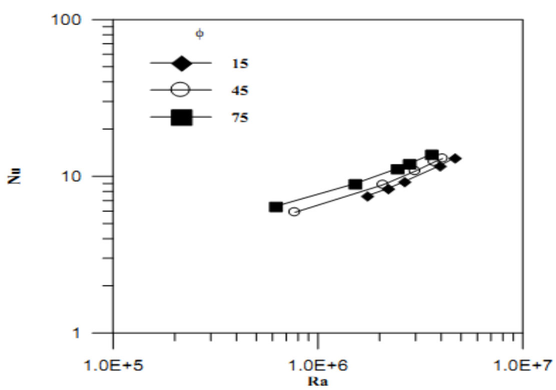


Figure 11. Influence of inclination angle on average Nusselt number versus Rayleigh numbers

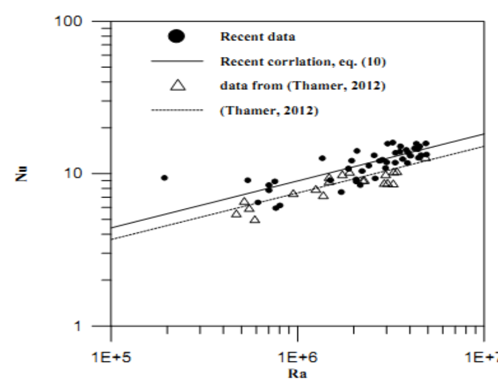


Figure 12. Comprision between inclined and horizontal experimental data Thamer [12] with the correlation obtained eq. (10)

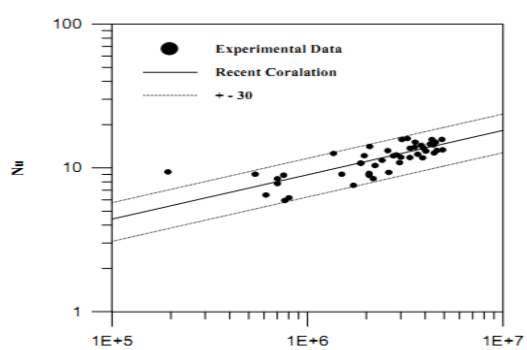


Figure 13. Comprision between precent experimental data with the correlation obtained eq. (10)

CONCLUSIONS

In this paper, natural convection in an inclined enclosure had been investigated experimentally. All the enclosure surfaces were thermally insulated except the upper surface, where half of it was cooled and the other half was heated. The following conclusions can be obtained:

1. At (x/L= 0.33) the highest Nu_x is found for all the tested inclination angles, heat fluxes and mass flow rates, because here the highest heat transfer occurs between the hot air and the cold surface. While at (x/L= 0.67) the lowest Nu_x is found, because heat transfer is the lowest in the enclosure for the low temperature difference between the warm air and the hot surface.
2. The highest temperature ratio was observed at (x/L= 0.67) while the lowest at (x/L= 0.17).
3. Increasing the heat flux and the mass flow rate increases the convicted heat through the enclosure leading to higher Nu.
4. Increasing the inclination angle had increased the disrupting between the cold and the hot air which increased the heat transfer rate and Nu.

NOMENCLATURE

A_s	surface area of test section, $[m^2]$	Ra	Rayleigh number, [-]
B	enclosure width, [m]	*Greek symbols*	
Cp	specific heat, $[J.kg^{-1}K^{-1}]$	□	thermal diffusivity, [m^2s□^{1}]
g	gravitational acceleration, $[ms^{-2}]$	□	volume expansion coefficient, $[K^{-1}]$
h_x	local convection heat transfer coefficient, $[Wm^{-2}K^{-1}]$	δ	enclosure thickness, [m]
I	electrical current, [A]	ε	emissivity, [-]
k	thermal conductivity, $[Wm^{-1}K^{-1}]$	θ	temperature ratio between the surface temperature and the temperature of the air in the center of the enclosure
L	enclosure length, [m]	σ	Stefan-Boltzmann constant $= 5.67 \times 10^{-8}$, $[Wm^{-2}K^{-4}]$
$\dot{m}_w$	mass flow rate of water, [kg/s]	ν	kinematic viscosity, $[m^2s^{-1}]$
Q_g	heat generated, $[Wm^{-2}]$	φ	inclination angle, [degree]
q_c	convection heat flux, $[Wm^{-2}]$	*Subscripts*	
q_r	radiation heat flux, $[Wm^{-2}]$	C	cold surface
Q_w	cooling heat, [Watt]	f	film
T	temperature, $[^oC]$	H	hot surface
V	heater voltage, [Volt]	i	inlet
x	axial distance, [m]	o	outlet
Dimensionless group		w	water
Nu	Nusselt number, [-]		

ISSN-L: 2223-9553, ISSN: 2223-9944
Vol. 4 No. 2 March 2013

REFERENCES

Tso, C. P., Jin, L. F., Tou, S. K. W. & Zhang, X. F. (2004). Flow Pattern Evolution in Natural Convection Cooling from an Array of Discrete Heat Sources in a Rectangular Cavity at Various Orientations. *International Journal of Heat and Mass Transfer, 47*, 4061-4073.

Aminossadati, S. M. & Ghasemi, B. (2005). The Effects of Orientation of an Inclined Enclosure on Laminar Natural Convection. *Heat and Technology, 23*, pp. 43-49.

Dekajlo, K. & Kowalewski, T. A. (2005). Experimental and Numerical Investigation of Natural Convection in Rectangular Inclined Enclosure. *EUROTHERM82, Numerical Heat Transfer, 82*.

Tasnim, S. H. & Mahmud, S. (2006). Laminar Free Convection inside an Inclined L-Shaped Enclosure. *International Communications in Heat and Mass Transfer, 33*, 936-942.

Bairi, A., Laraqi, N. & Garcia de Maria, J. M. (2007). Numerical and Experimental Study of Natural Convection in Tilted Parallelepipedic Cavities for Large Rayleigh Numbers. *Experimental Thermal and Fluid Science, 31*, 309-324.

Bairi, A., Garcia de Maria, J. M., Laraqi, N. & Alilat, N. (2008). Free Convection Generated in an Enclosure by Alternate Heated Bands, Experimental and Numerical Study Adapted to Electronics Thermal Control. *International Journal of Heat and Fluid Flow, 29*, 1337-1346.

Abu-Nada, E. & Oztop, H. F. (2009). *Effects of Inclination Angle on Natural Convection in Enclosures Filled With Cu–Water Nanofluid*, International Journal of Heat and Fluid Flow, 30, pp. 669-678.

Mohamed, E. A. (2007). *Experimental Study of Steady State Natural Convection Heat Transfer from Noncircular Metallic Ducts*. King Saud University, college of engineering research center, No. 33/426.

Siegel, R. & Howell, J. R. (1992). *Thermal Radiation Heat Transfer*, 3rd edition, McGraw-Hill, New York, USA.

Holman, J. P. (1977). *Experimental Method for Engineers*. (5th ed.). New York: McGraw-Hill Book Company.

Kothandaraman, C. P. & Subramanyan, S. (2008). *Heat and Mass Transfer Data Book* (6th ed.). New Delhi: New Age International Limited Publishers.

Thamer, K. S. (2012). An Experimental Study to Investigate the Upper Heating and Cooling Effect on Natural Convection Heat Transfer through a Horizontal Rectangular Enclosure. *Journal of Al-Qadisiya for Engineering, 5*.