Natural Convection Heat Transfer in Cylindrical Enclosure of Porous Media with Periodic Boundary Conditions

Ihsan Y. Hussain
Professor, Department of Mechanical Engineering
University of Baghdad

Nasr A.Jabbar
Scientist Department of Mechanical Engineering
University of Baghdad

ABSTRACT
Natural convection heat transfer through a fluid-saturated porous media in inclined cylindrical enclosure was investigated experimentally and numerically in the present work. Numerical solution of the governing equations was made with the appropriate boundary conditions by using a CFD code FLUENT (fluid flow) 6.3.26. The experimental tests were carried out for inclination angles (0^o, 25^o, 50^o, 70^o, and 90^o) at period ranges (10, 22, 50, and 90 min) and amplitude (500, 1000, 1500, and 2000 W/m^2). The Results indicate an increase in the heat transfer with increasing of Rayleigh number, time, amplitude, period and angle of inclination from the horizontal position. Theoretical predications agree well with experimental results.

Keywords
Heat transfer Natural Convection; Cylindrical Enclosure; Porous Media; Periodic Heat flux Variation

1. INTRODUCTION
Free convection heat transfer from inclined wavy surface has received attention because of its vast applications. Some of these applications include ground water flows, oil recovery processes, thermal insulation engineering food processing etc. Extensive literature on the topic is availed for porous media, Slimi et. al. 1998[1] studied two – dimensional and transient fluid flow and heat transfer by natural convection in a vertical cylinder opened at both ends filled with a saturated porous medium and heated with a uniform lateral heat flux. The study was carried out using the forchheimer – extended Darcy flow model. Taofik et.al. 1999[2] studied unsteady natural convection which occurs in a vertical cylindrical enclosure opened at both ends, filled with a fluid saturated porous medium with a periodic lateral heat flux density. The study was carried out by the use of the Darcy law and it takes in to account heat conduction in the wall. The set of equations was solved numerically by the standard finite volume method. Saaed 2000[3] proposed a simple numerical expression for average Nusselt numbers over isothermal horizontal cylinder for all Rayleigh by using Darcy flow model. And also Khalid Abd.Al-hussein 2001[4] obtained a simple relation for Nusselt number which is a strong function of modified Rayleigh number, time, radius ratio, and aspect ratio. AL-Najar 2004[5] used the finite difference method to investigate the steady free convection from a two separated horizontal cylinders embedded in saturated porous media bounded by rectangular cavity. The cylinders kept isothermally hot while the bounded cavity is isothermally. It found that the large heat gained to the cavity been at the upper horizontal wall above cylinders. Saleh 2008[6] studied numerically unsteady natural convection heat transfer through a fluid–saturated porous media in inclined pipe enclosure. The temperature at cylindrical sidewall Tw was sinusoidal variation in z-direction. The problem was analyzed for modified Rayleigh number with range of (50- 300), angles of inclination (0°, 25°, 60°, 90°), amplitude of sinusoidal temperature (0.2, 0.4, 0.8) and period (0.005, 0.01, 0.02). The numerical results show Nusselt number variation increases with increase of amplitude and decreases with increase of period and is constant or increase very little with increase of angle. Sankar el.at. 2011[7] studied influence the heat source length, modified of Rayleigh number and Darcy number, size and location of the heater affects. The numerical results show the heat transfer rate decreases with an increases in the heater length, while an increase in the radius ratio, modified Rayleigh number and Darcy number increases the heat transfer. They found that the size and location of the heater affects the flow intensity heat transfer rate in the annular cavity. The present work investigates the natural convection heat transfer from a cylindrical enclosure filled with saturated porous medium numerically (Fluent Program) and experimentally with wide range of parameter, heat fluxes, inclination angles and times.

2. EXPERIMENTAL WORK
2.1 Test Section
The test section used in this study is depicted in the schematic diagram of figure (1). An experimental rig was designed and constructed in the Heat Transfer Lab, at the Mechanical Engineering Department, University of Baghdad. The test section was Aluminum tube with (48 mm inside diameter), (50 mm outer diameter) and a length of (250 mm). An electric heaters consisting of thin strips manufactured from Nickrom (nickel-chromium alloy) with (3 mm width) and (0.42 mm thickness) were used. These strips were longitudinally wrapped on a layer of mica. The upper and lower faces of the heater were covered with a double layer of mica of (0.5 mm thickness) and placed between two thin plates, which were formed in cylindrical shape. The cylindrical heater was fixed around the outer surface of test tube by three screws. The entire test rig was fixed in cylindrical holder (85mm diameter), (25mm length), which was connected with frame to allow inclining the test cylindrical enclosure with the desired angle. The frame was provided with a protractor with its range indicator from (0^o to 180^o). The test section part was filled with glass bead. The average glass sphere diameter (12mm). A (26) thermocouples Alumel-Chromel (type K) was divided into four groups in four levels by fixing on rake, the levels were separated by uniform vertical distance of (50mm) along the test section. The horizontal distance between the thermocouple was (7mm), which was chosen to cover up the change in temperature profile. In one group, the thermocouples were distributed in three horizontal rakes, separated by (90°) to measure the temperature with different angles in the same level, and two thermocouples were used to check symmetry.

2.2 Periodic Electric Circuit

To oscillate the side wall heat flux, the electric circuit shown in fig. (2) was used. This circuit was designed to periodically vary the power of the main heater. The major components of the circuit are twin timer (SUNCHO, SHT-T1E), resistance (220V, H550/27R±5%), and primary main heater. The timer was basically a switch that opens and closes cyclically. The period is adjustable, using the top tuning knobs to limit time on and time off, and can vary from (1.0) s to (3.0) h. The period of the voltage oscillation was adjusted by changing the timer setting, whereas the amplitude, or change in voltage drop across the main heater, was adjusted by changing the supply voltage by the variac. The purpose of adding the resistor was to reduce the value of the heat flux to 40% from main heat flux (without resistor), therefore amplitude wave was made.

2. 3 Experimental Procedure

The steady state and transient experiments followed essentially the same basic procedure expect that for steady state experiments the timer and external resistor in the main electric circuit were removed. The heater circuit was turned on, and the timer period and power by variac was set. The system runs for several cycles until the initial transient died and the time-average wall temperature became nearly constant where after three hours, the system reached to the steady state condition. The digital thermometer type (12 Channels) which was used for recording the temperatures.

3. NUMERICAL SIMULATION

In order to analyze the flow field and natural convection heat transfer in cylinder filled with porous media, a solution of Navier-stokes equations and energy equation is required. The mathematical model of the fixed and oscillating boundary conditions are solved numerically using a CFD Code FLUENT (fluid flow) 6.3.26 after describing the mesh model using the Gambit 2.2.30. The geometry of the problem shown in fig. (3) Is generated by using Gambit 2.2.30. The geometry consists of a cylindrical enclosure, porous medium, with dimensions of the radius of the tube (R=24mm), the length of the cylinder (L=250mm), and filled with porous media (glass beads with diameter 12mm). The user defined function wallprof.c (UDFs) is used to specify a sinusoidal heat flux variation on the wall. A UDF is a function (programmed by the user) written in C language which can be dynamically linked with FlUENT solver. The computations were carried out for only one half of the computational domain as in fig. (4) due to symmetry about central vertical line.

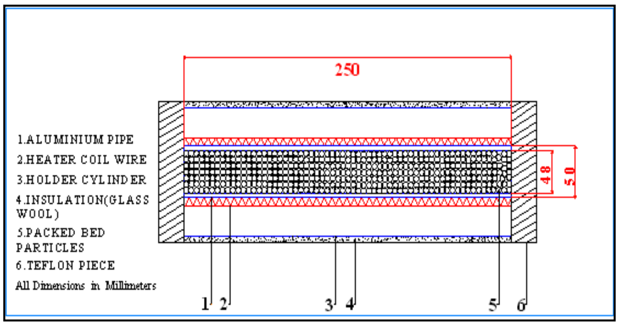


Fig. 1: Test Section

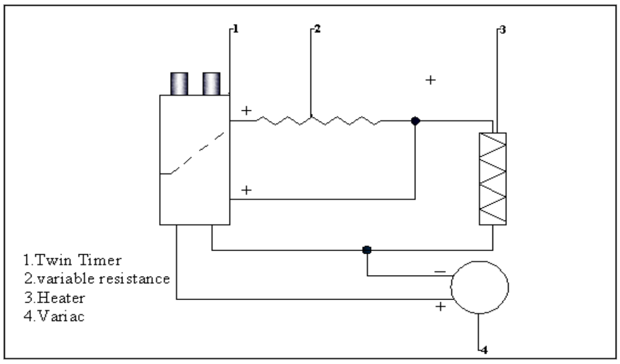


Fig. 2: Schematic Wiring Diagram of Periodec Electric Circuit

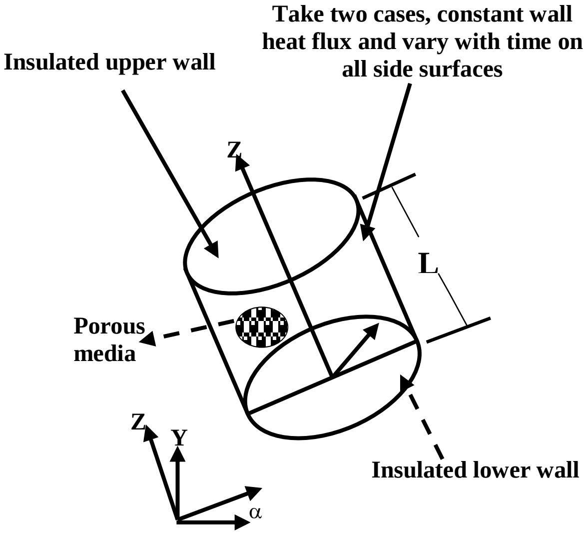


Fig. 3: Geometry and Coordinate System

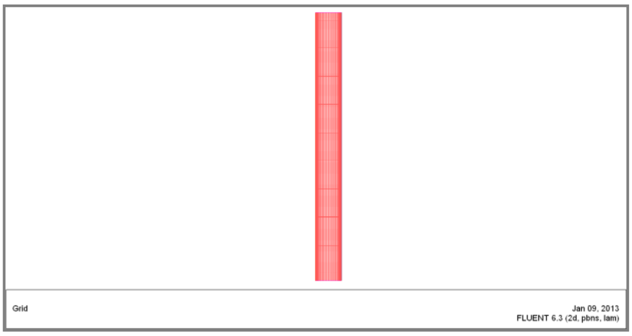


Fig. 4: Model with Meshes

4. RESULTS AND DISCUSSION

4.1 Numerical Results

Figs. (5 and 6) show the steady state temperature distribution in a form of contours maps. The effect of angle of inclination is introduced in the figures via selecting two angles values α= 0° and 90°. A high temperature values are on the wall of the cylinder, where more heat is transferred to the porous medium because of the low heat transfer resistance while the cold fluid occupies the center of cylinder. In case of horizontal positions in all heat fluxes the center of the contour lines coincides with the center of computational domain for all sections due to symmetry. In vertical positions, the hot layer occupies the upper right side and cold layer moves to lower left side due to difference in density, all this owing to a strong buoyancy force. Figures (7-8) show the isothermal lines of temperature distribution due to buoyancy- forces in the cylindrical enclosure in form contour maps. With increase of time more heat will transfer to computational domain. This effect can be see in figures with same period but different heat fluxes, the thickness of thermal boundary is very large and the isothermal lines move to left side and cold boundary confined in the left side near the lower part indicating a vigorous convection flow, the lines began to deform and becomes curvilinear with increase of (α). At upper part of cylinder the hot layer at right side is greater than at left side and the deformation of isothermal lines in right side is greater than that at left side, all this owing to ascending and descending flow. At fixed period heat transferred to enclosure is very high and increase with increasing amplitude and angle of inclination also the isothermal lines began to deform with increase of amplitude and angle, this deformation in the right side is greater than that in the left side. Hot regions formed in the right corners and this hot regions increase with increasing amplitude, Ra, and angle of inclination from horizontal position.

4.2 Experimental Results

For steady state the temperature variation in horizontal and vertical positions is plotted for selected runs in fig. (9). It is obvious that the rate of surface temperature rise is directly proportional to the wall heat flux. This can be attributed to the increase in growth of the thermal boundary layer due to buoyancy effect as the heat flux increases. At periodic boundary condition case; figure (10) shows the effect of amplitude on the convective fluid inside the enclosure. It is clear from these figures that for fixed period, the wall and temperatures time history looks qualitatively the same over the rang of amplitudes tested and takes on a sawtooth appearance in shape. Increasing the amplitude of the temperature oscillation "stretches" the scale of y-axis and sawtooth profile become more distinctive. Figure (11) exhibit the wall temperatures time dependence for experiments having the same amplitude but different period. It can be noticed that as the period increases, the chance for more heat to the enclosure increases with increasing the heat fluxes. The curvature of the temperatures profiles increased with period of oscillation, and this means that the convection heat transfer increases with period. The instantaneous average Nusselt number is integrated experimentally at the hot wall and a plot of Nusselt number verses time is shown in figure (12) and verses amplitude in figure (13). The average Nusselt number along the hot wall varies sinusoidally according to the variation of the hot wall temperature and it increases with increase of amplitude and decreases with increase of period. Figure (14) shows the variation of local Nusselt number with Z at period = 10 min and different angles of inclination, for this figure it is found that the local Nusselt number decreases with increasing α owing to the effect of boundary layer except at $\alpha = 90^o$, its value is constant due to the absence of body force in Z-direction, also Nu number decreases with increasing Z for the same reason. This figure depicts Nu number increase with increasing amplitude and this behavior is the same for all runs. In order to describe the relation between the dependent variable (Nusselt number) and the independent variables (Raylieh number and angle of inclination), a correlation have been made to describe the heat transfer data as in the form of ;

Nu= (-289.095) + (65.6721)*$\log_{10}$(Ra Sin (α))   Eq. [1]

And the following correlation work for α=0°, the values of constants are;

Nu = (0.0381565)* Ra** (1.372186)   Eq. [2]

4.3 Verification

The observed values of Nusselt number from the theoretical work have been compared with the experimental results for the same boundary condition (periodic boundary condition). The experimental and theoretical local Nusselt number results for (q=500 W/m^2 and α=0) at four time intervals (10, 22, 50, and 90 min) has been compared with each other, as shown in figure (15). The figure reveals that the experimental Nusselt number follows the same behavior as the present numerical results; there is a difference between the theoretical and experimental results as can be seen from these figures. These differences may be due to the heat loss, errors in measurements. The relative error is tabulated in table (1). To verify the results obtained of the present study, a comparison was made with the results achieved by previous studies. The present results for temperature profile agrees with behavior for different periods or different amplitude with the results of Kazmierczak and Muley 1994[8] (experimental study) shown in figure (16). The present results for Nusselt number agrees with the result of Antohe and Lage 1996 [9] shown in figure (17).

Table1. Relative Error of Comparison

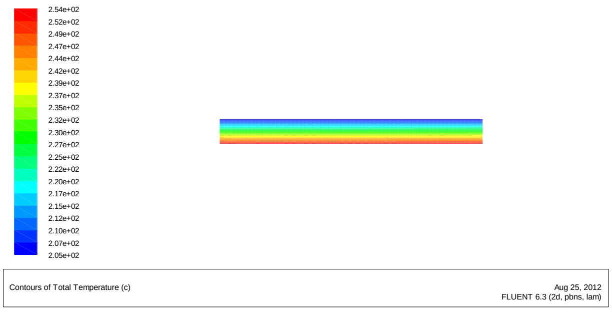
Period (min)	Err %
10	17.6
22	5.3
50	5.17
90	4.07

5. CONCLUSIONS

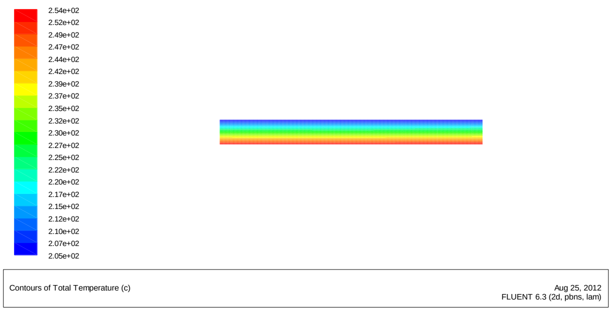
It was concluded that the pattern of the isothermal lines change periodically in the same manner as the hot surface and inside temperatures cycle. Hot regions formed with increased Ra, time and angles of inclination from horizontal to vertical positions. The total heat transfer rate to the enclosure is dependent on the amplitude and period of the pulsating sidewall hear flux. The heat transfer rate is increasing as the pulsating amplitude increases. Further, the heat transfer enhanced with increasing period. Average Nusselt number was shown to be an increasing function of the amplitude, decreases with increased period of the wall temperature oscillation and increases little with increased Ra, and decreased α .The fluid and heat transfer parameters have the same behavior in the numerical and experimental analysis for all parameters used. Experiments have been carried out to check the numerical results. The measured values for average Nusselt numbers agree with the numerical values quite well for the entire range studied.

6. NOMENCALTURE

D	Diameter (mm)
Nu	Nusselt Number
L	Length of the Cylinder (mm)
R	Radius of the Pipe (mm)
Ra	Raylieh number
q	Heat Flux (W/m^2)
t	Time (min.)
T	Temperature (°c)
Z	Axial coordinate
α	Inclination Angles (degree)

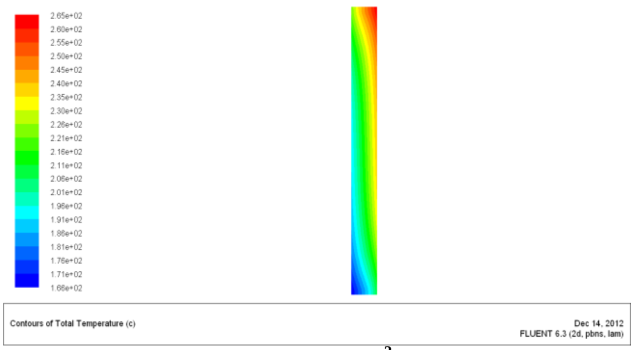


(a) q=2000w/m^2

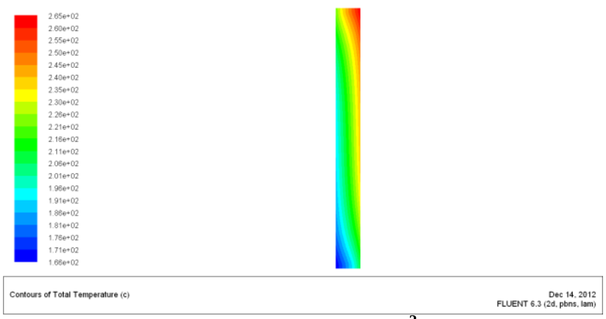


(b) q=1500 w/m^2

Fig .5: Temperature Distribution for Horizontal Position

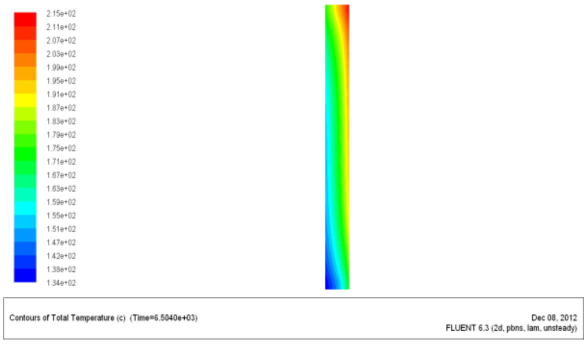


(a) q=2000w/m^2

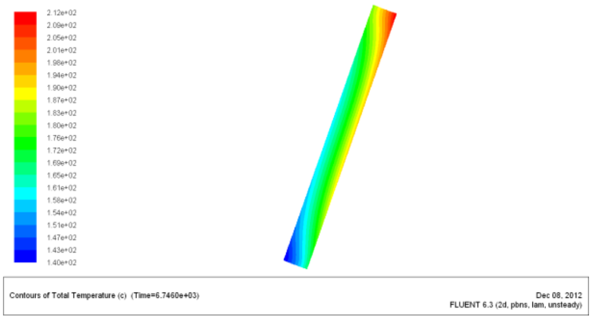


(b) q=1500 w/m^2

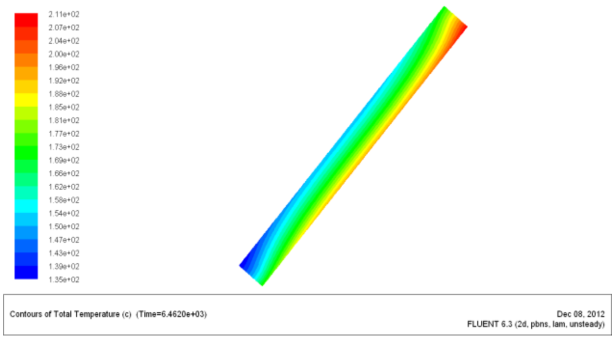
Fig. 6: Temperature Distribution for vertical Position



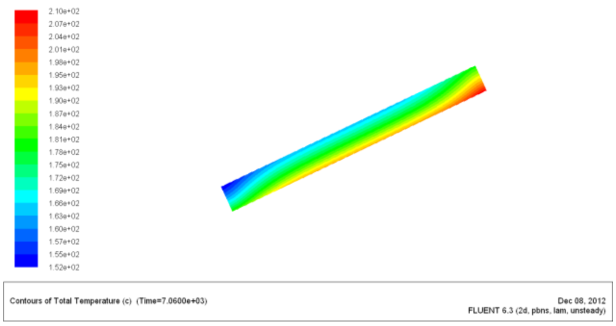
(a) α= 0°



(b) α= 25°

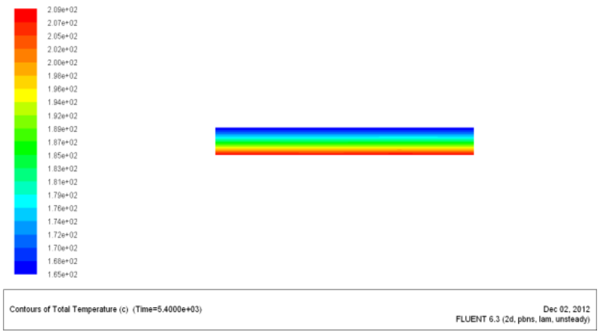


(c) α= 50°

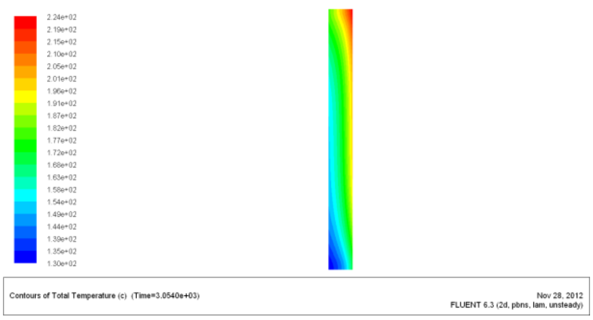


(d) α=70°

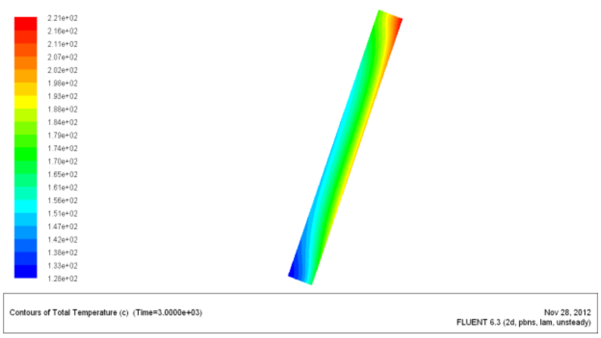
Fig. 7: Temperature Distribution for q=1500 w/m^2, t= 90min



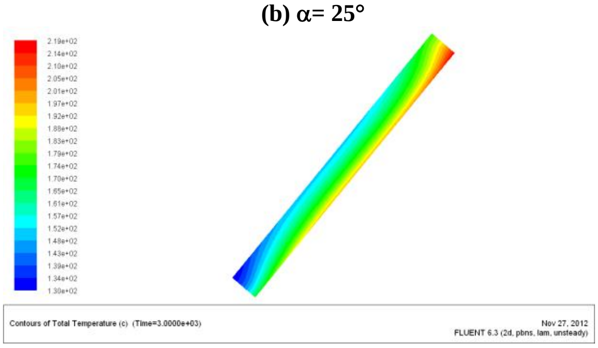
(e) α=90°
Fig. 7: Contd.



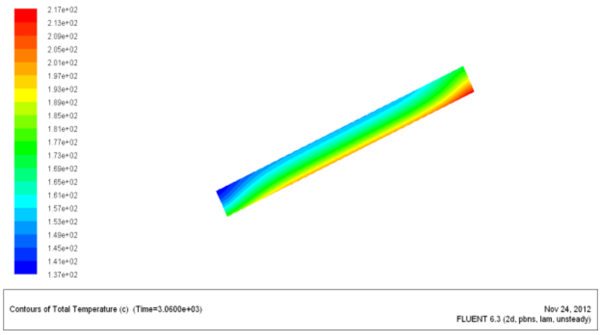
(a) α= 0°



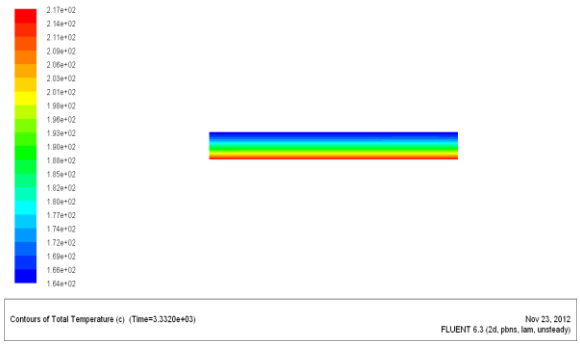
(b) α= 25°



(c) α= 50°
Fig. 8: Temperature Distribution for q =2000 w/m^2, t=50min



(d) α=70°



(e) α=90°
Fig. 8: Contd.

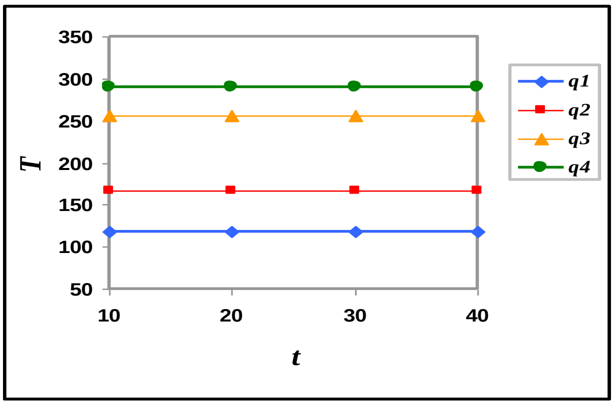


Fig. 9: Steady State Surface Temperature variation with Different Heat Fluxes: (q_1=500,q_2=1000,q_3=1500, q_4=2000W/m^2)

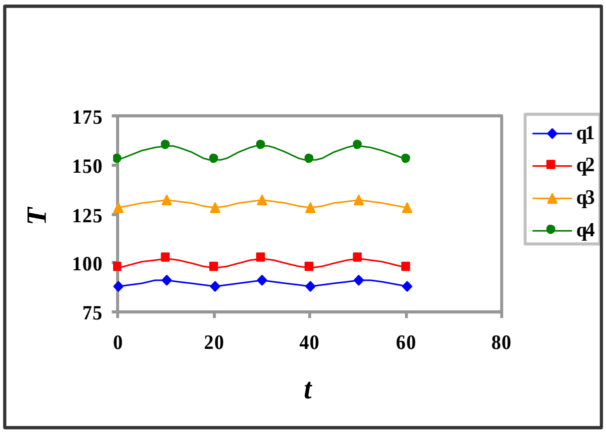
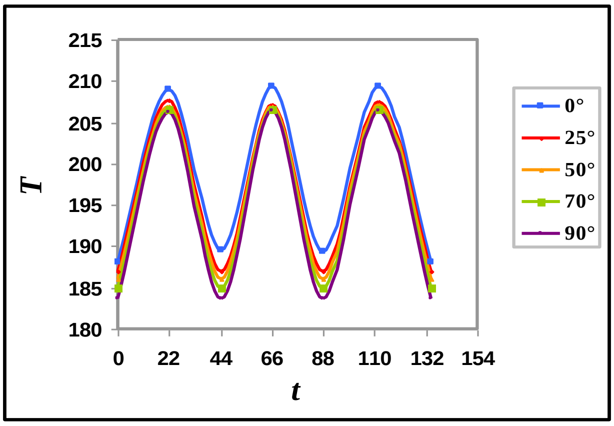
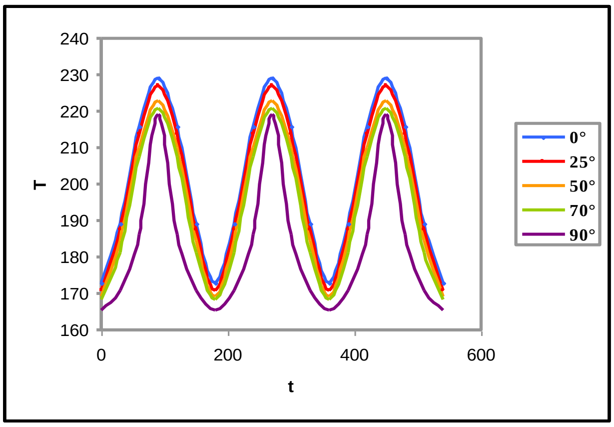


Fig.10: Wall Temperature with Period 10min at α=0° and Different Amplitudes :(q_1=500,q_2=1000, q_3=1500,q_4=2000W/m^2)



(a) Period =10min



(b)Period=90min
Fig.11: The Wall Temperature with Time at given Amplitude (q=2000 w/m^2), for Different Periods

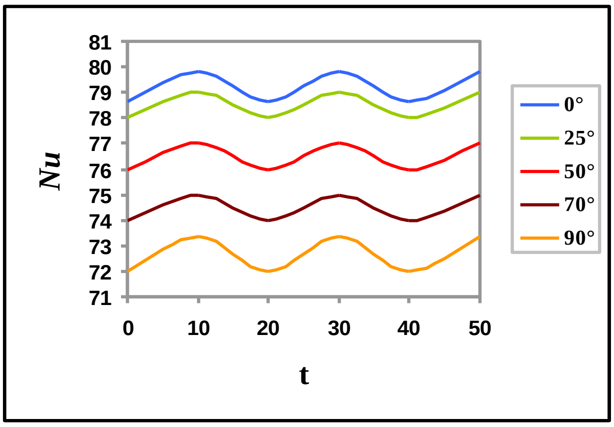


Fig.12: Experimental Result of Effect Period=10min on Average Nusselt Number at q= 2000w/m^2

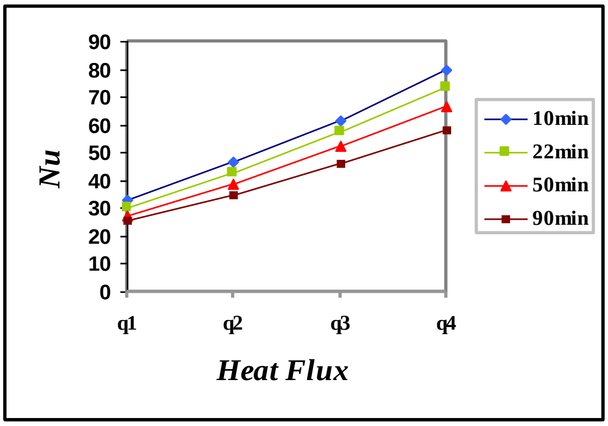


Fig.13: Effect of Amplitude on Average Nusselt Number

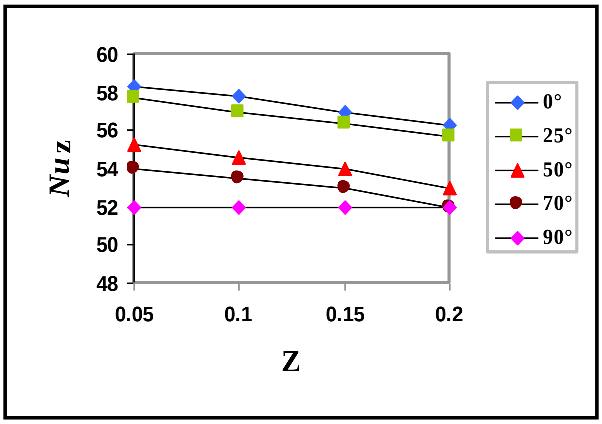


Fig.14: Variation of Local Nusselt Number with Z for q=2000 w/m^2 at Period =10 min.

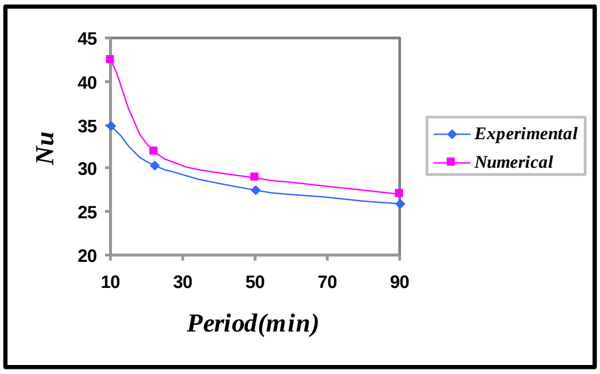


Fig.15: Comparison of Experimental and Numerical Results

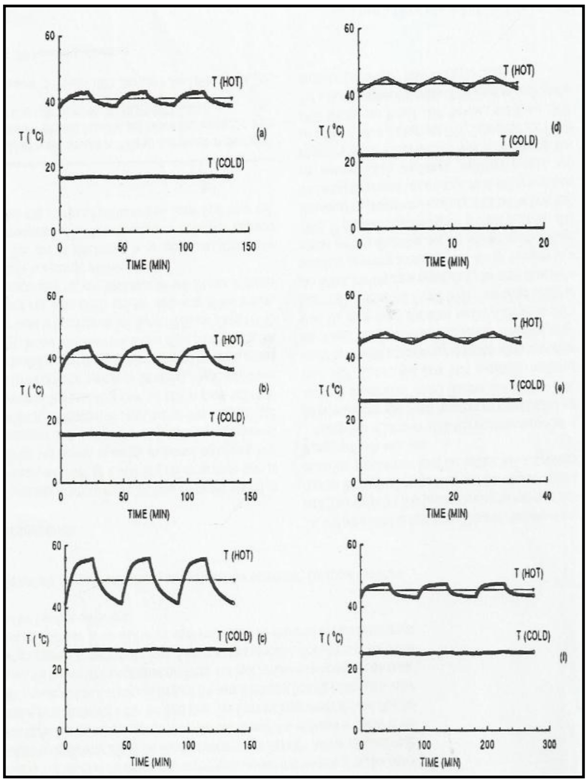


Fig.16: Temperature Profile of (Kazmierczak and Muley 1994)

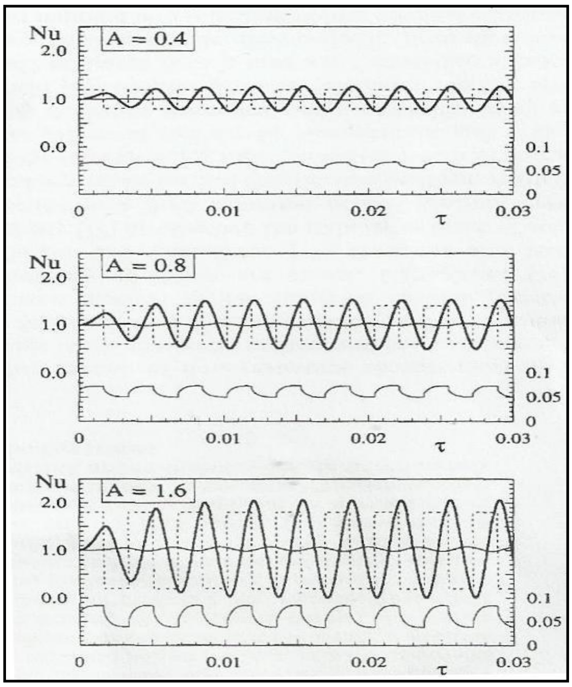


Fig.17: Nusselt Number Profile of (Antohe and Lage 1996)

7. ACKNOELEDGMENTS

First of all, it hardly seems possible that this long work is at end without the helpfulness and carefulness of Allah the creator Creation.

With all gratitude and respect, I wish to thank "Dr. Nabeh- Al Durobi"; the Mechanical Engineering Department of University of Technology, Thanks are due to "Assist.Lect. Thamer Mousa", the Electrical Engineering Department of University of Baghdad for their contribution to current work.

8. REFERENCES

1] Slimi, K., and Nasrallah, B. 1998; Transient Natural Convection in a vertical Cylinder Opened at The Extremities and Filled with a Fluid Saturated Porous Medium: Validity of Darcy Flow Model and Thermal Boundary Layer Approximations, Int. J. Heat mass Transfer, Vol. 41, Nos. 8-9, PP.1113-1125,.

[2] Taofik, A., Slimi, K., and Nasrallah, S.B. 1999; Free Convection in a Vertical Cylinder Enclosure, Int. J.Therm .Sci. Vol .39, PP. 616-634.

[3] Saaed. 2000; Numerical Study of Natural Convection Heat Transfer from a Horizontal Cylinder Embedded in a Rectangular Porous Medium, M.Sc. Thesis, Mosul University.

[4] Abdul-Hussein, K. J. 2001; Numerical Analysis of Unsteady Natural Convection in Porous Media Enclosed between two Concentric Vertical Cylinders, M.Sc. Thesis, Mosul University.

[5] Al-Najar, A.K. 2004; Numerical Steady of Natural Convection Heat Transfer from Two Horizontal Cylinders Embedded in Rectangular Porous Media, MSD. Thesis, University of Mosul.

[6] Saleh, M. M. 2008; Natural Convection in Confined Cylindrical Enclosure of Porous Media with Constant and Periodic Wall Temperature Boundary Conditions, MsD. Thesis, University of Baghdad.

[7] Sankar, M; Park, Y; Lopez, J.M. and Do, Y. 2011; Numerical Study of Natural Convection in a Vertical Porous Annulus with Discrete Heating , Int .J .Heat mass Transfer, Vol.54, PP.1493-1505.

[8] Kazmierczak, M., and Muley, A.1994; Steady and Transfer Natural Convection Experimental in A horizontal porous layer: The Effects of A thin Top Fluid Layer and Oscillating Bottom Wall Temperature, Int.J. Heat mass Transfer, .Vol.15, No.1.

[9] Antohe, B.V., and Lage, J.L. 1996; Amplitude Effect on Convection Induced by Time-Periodic horizontal heating, Int. J. Heat mass Transfer, .Vol.39, No.6, pp.1121-1133.