An Investigation into Forced Convection Heat Transfer through Porous Media

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ABSTRACT
Theoretical and experimental investigations of forced convection heat transfer from a heated flat plate embedded in porous media with a constant heat flux had been carried out in the present work. The experimental investigation included a set of experiments carried out to study the effect of Reynolds number and heat flux on the temperature profile and local Nusselt number. The investigation covered values of heat flux of (1000, 2000, 3000, 4000 and 5000 W/m²) and Reynolds number values of (24118, 44545, 739832 and 82208). Fluent program has been used to simulate all cases of the experimental work. The numerical investigation covers all values of heat flux and velocities in the experimental work. The initial values and boundary conditions are similar for both theoretical and experimental investigation. It is observed that the local wall temperature gradually increases with the flow direction, decreases with the Reynolds number and increases with heat flux, but the fluid temperature progressively decreases in the porous medium with the vertical direction away from the heated wall, and the results show an increase in local Nusselt number when Reynolds number and heat flux increase.

Keywords
Forced convection, Porous Media, Experimental and Numerical Study

1. INTRODUCTION
Forced convection heat transfer in a confined porous medium has been a subject of intensive studies during the last decades because of its wide applications, including: Chemical, Environmental, Mechanical, Geological and Petroleum. The porous medium has a great ability to transfer heat and the importance of porous medium in many applications have been extended widely in the study of this type of heat transfer problems to improve heat transfer rates by increasing the amount of heat exchange between the surface carrier and the external fluid. The problem had been investigated by many researchers; Aydm and Kaya. [1] studied the laminar boundary layer flow over a flat plate embedded in a fluid saturated porous medium in the presence of viscous dissipation; Inertia effect and suction/injection are included by using the finite difference method. Beckerman and Viskanata [2] studied the forced convection boundary layer flow and heat transfer along a flat plate embedded in porous medium by including both, inertia and boundary effects. Bejan and Nield [3] described the time evolution of the temperature and heat transfer in the vicinity of a flat wall embedded in a parallel flow through a saturated porous medium. The flow is uniform and steady, while the wall is suddenly subjected to heating or cooling. Cheng and Hsiaom [4] studied the unsteady forced convection on flat plate embedded in the fluid-saturated porous medium with inertia effect and thermal dispersion. Hady and Ibrahim [5] studied the effect of the presence of an isotropic solid matrix on the forced convection heat transfer rate from a flat plate to power-law-Newtonian fluid saturated porous medium. Kaviany [6] studied the boundary –layer treatment of forced convection heat transfer from a semi-infinite flat plate embedded in porous media and the effect of the presence of an isotropic solid matrix on the forced convection heat transfer rate from a flat plate, using the integral method. Nakayama et.al [7] studied the non-Darcian boundary layer flow and forced convection heat transfer over a flat plate in a fluid-saturated porous medium. The momentum equation, which includes the convective inertia term, the Forchheimer term, and the Brinkman term in addition to the Darcy term, was solved by means of the local similarity solution using novel transformed variables deduced from a scale analysis. Pantokratoras and Magyari [8] studied the steady forced convection flow of a power-law fluid over a horizontal plate embedded in a saturated Darcy-Brinkman porous medium. The pertinent boundary value problem was investigated analytically, as well as numerically by a Finite Difference method. Qahtan [9] studied the non-Darcian flow and forced convection heat transfer characteristics through and over porous layer on a heated wall at constant temperature. Governing equations were solved numerically by using Finite Difference Approximation. Vorticity–stream function method has been used in this study. The effects of Reynolds number, Darcy number, and inertia parameter for porous media and Prandtl number are considered on the time stability. Vafai and Huang [10] studied the heat transfer pregnancy forced through the layer from the center of follicular placed on the plate with a constant temperature, they studied transport phenomena through the interface between the layer of porous media and the fluid, the flow through the porous media has been expressed by the formulation of Brinkman _ Forschmaar derived from Darcy's law, but the energy equation has been formulated depending on the model of a single equation **(LTE)**. Zhao and Song [11] studied the forced convection in a saturated porous medium subjected to heating with a permeable wall perpendicular to flow direction and showed that the heat transfer rate from the permeable wall to the fluid can be described by a simple equation: **Nu=Pe**. The present work investigates the forced convection heat transfer from a flat plate embedded in porous media numerically (Fluent Program) and experimentally with wide range of heat flux and velocities.

2. DEFINITION OF THE PROBLEM
The geometry of the problem is shown in figure (1). The geometry consists of a rectangular box contains the heated flat plate and also contains the porous medium (glass beads). A two dimensional Cartesian coordinate system was considered in this study, where the ***X***-direction represents the flow direction., while the y-direction represents the vertical direction of heat supplied from the flat plate to the adjacent porous media.

3. NUMERICAL SIMULATION

In order to analyze the flow field and heat transfer over heated plate embedded in porous media, a solution of the Navier-stokes and energy equations is required. In the present work, the problem was solved numerically using a CFD Code FLUENT 6.3.26 after describing the mesh model using the Gambit 2.2.30 after describing the mesh model using the Workbench 14.0. The system geometry in the present work basically consists of a box (which represents the flow tunnel) and this box contains heated plate topped with porous media, see figure (2). The geometry is generated by using Workbench 14.0 as a two cuboid with specific location, interconnecting them by some interrelationships prepared for meshing and boundary conditions specifications. A very smooth mesh, a higher order element type tetrahedral / hybrid in 3D is used for mesh generation to approximate precisely the geometry interfaces

4. EXPERIMENTAL WORK

The experimental rig was designed and constructed in the Heat Transfer Lab, at the Mechanical Engineering Department University of Baghdad, were the experiments were carried out. A suction type, low speed wind tunnel with solid wood walls was used in the present work, the fan: driven by a one phase (AEI) A.C motor with a speed of 2850 rpm, the valve by which the flow rate of air can be controlled through opening from 0 % to 100 %. The Reynolds number values are (24118 (1.86 m/s), 44545 (3.45m/s), 739832 (5.73 m/s) and 82208 (6.37 m/s)). The test section consists of a rectangular box with (length 240 mm x width 100 mm x height 70 mm), both sides of the box are made of wood, the front and back side are made of iron wires mesh of size (10 mm x 10 mm). The bottom side is made of Glass fiber matrix with 10 mm thickness where the heated flat plate were fixed with dimensions (240 mm x 100 mm), upper side is open, see figure (3). The porous material is glass beads, **see table (1)** for properties. The heater that used was (KANTHAC) with 9.919 Ω/m resistance, 1.4 mm width and 0.11 mm thickness and made from aluminum strip wounded around a piece of Mica in a way to give the heater equal temperature in all regions of the flat plate. The number of thermocouples used in the present work was thirty Alumel-Chromel (type K) thermocouples, the thermocouples measure the temperature through the boundary layer. The following parameters were recorded during the test: heat flux and velocity. The ranges of measured variables are shown in **table (2).**

Table1. Properties of Porous Media

Parameters	Values
Porosity (ε)	0.343
Permeability (K)	3.3 x 10^{-5}(m^2)
Density (ρ)	2563 (kg/m^3)
Thermal Conductivity (k)	0.87(w/m.C)
Specific Capacity (C_p)	670 (J/kg .C)

Table 2. Ranges of Measured Variables

Parameters	Values
Velocity Of Air Through Wind Tunnel	(6.367, 5.73, 3.45, 1.868 m/s) 20% , 40% , 70% , 100%
Voltage	49 to 108 Volt
Current	0.48 to 1.07 Amp
Power	23.52 w to 115.56 w
Heat Flux	1031.579 to 4964.984 w/ m^2
Number of Tests	20 Test
Time of testing	7:30 A.M to 12 A.M in April Month

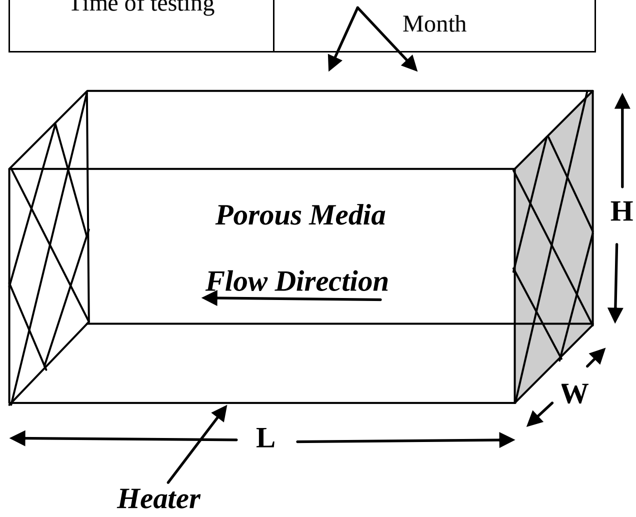


Fig.1: Model Geometry

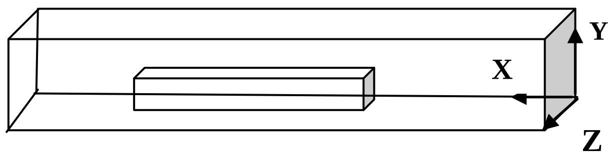
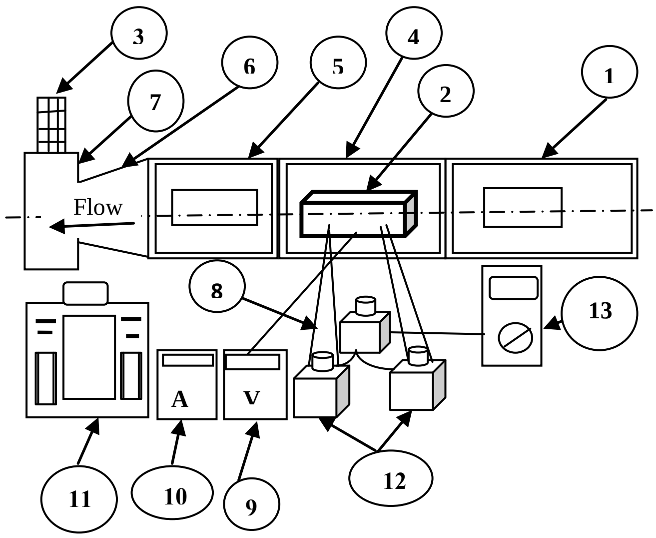


Fig.2: Geometry of the Problem



1. **Contraction No.1**	**8. Thermocouple**
2. **Model**	**9. Volte meter**
3. **Valve**	**10. Ameter**
4. **Test Section**	**11. Variac**
5. **Contraction No.2**	**12. Selector Switches**
6. **Conical Section**	**13. Reader**
7. **Fan**	

Fig. 3: Diagram of Experimental Apparatus

5. RESULTS AND DISCUSSION

5.1 Experimental Results

The temperature profile has been tested with the different coordinates at different velocities for different heat flux. The direct reading of temperature profile with normal distance for different heat flux values of ($q^{''}$=1000, 2000, 3000, 4000 and 5000 W/m^2) and different velocities values of (V=1.87(20% opening), 3.45 (40% opening), 5.73 (70% opening) and 6.37 (100% opening) m/s) is plotted in figure (4). It is seen that, the local wall temperature gradually increases with the increase in the axial position and the decrease in the Reynolds number, Increasing (Re) yields faster flow through the porous media over heated wall and therefore thinner thermal boundary layers that require a longer distance to develop and don't allow the heat to transfer into the porous media. At constant heat flux for different velocity values of (V=1.87(20% opening), 3.45 (40% opening), 5.73 (70% opening) and 6.34 (100% opening) m/s), figure (5) shows that the temperature progressively decreases in the porous medium with the vertical direction away from the heated wall. It is clear also that the increase of the Reynolds number leads to decrease in the temperature at constant heat flux. As the Reynolds number decreases, the temperature variation extends from the heated wall into the porous media. Figure (6) shows the temperature profile with normal distance at constant velocity for different heat flux values of ($q^{''}$=1000, 2000, 3000, 4000 and 5000 W/m^2). The results show that the fluid temperature increases with increasing heat flux. The local Nusselt number depends on the temperature profile along downstream coordinate. The figure (7) represents the local Nusselt number distribution at any point on the surface of the plate and comparison of local Nusselt number with different velocities at constant heat flux. As seen from these figures, the local Nusselt number increases with the axial position along the flow direction and the increase in the Reynolds number. At constant heat flux, the heat transfer coefficient decreases with the axial position along the flow direction and increases with increase in the Reynolds number because of the difference between the bulk air and surface temperature, and the thickness of thermal boundary layer increase ($h_x = \frac{q^{''}}{T_w - T_\infty}$) . the local Nusselt number increases with the axial position along the flow direction because the heat transfer coefficient is multiplied by the local distance (X) to find the local Nusselt number at any location, The figure (8) represents comparison of local Nusselt number with different heat fluxes at constant velocity. It is clear that the local Nusselt number increases with the heat flux because the Nusselt number equals ($Nu = \frac{h_x x}{k_m}$) and ($h_x = \frac{q^{''}}{T_w - T_\infty}$), so when heat flux increases Nusselt number increases too. The average Nusselt number was calculated from an integration of the local Nusselt number (Nu_x) and has been plotted with the Reynolds number for different values of heat fluxes. The average Nusselt number was calculated from an integration of the local Nusselt number (Nu_x) and has been plotted with the Reynolds number for different values of heat fluxes. The figure (9) shows that the average Nusselt number increases with increasing Reynolds number at constant heat flux. The high value of average Nusselt number due to the low difference between the air and heated plate temperature and the thin thermal boundary layer formed. Figure (10) shows the correlation of average Nusselt number (Nu) with Reynolds number (Re) of experimental results in the form:

$$Nu = c\, Re^{\,n} \qquad \text{Eq. [1]}$$

Where c and n are empirical constants presented in **Table (3)** for the heat flux range (1000, 2000, 3000, 4000, and 5000 W/m^2).

Table 3. Correlation Equation Parameters for Experimental Results

$q^{''}$(W/m^2)	c	n	Correlation Factor (R)
1000	0.389	0.348	0.962
2000	0.240	0.386	0.952
3000	0.301	0.358	0.971
4000	0.301	0.344	0.946
5000	0.008	0.652	0.976

5.2 Numerical Results

The temperature distributions with the vertical direction in the various positions of heated flat plate are presented in figures (11) to (14) for various heat flux and various velocities. The outlet bulk temperature increases with the increase in the value of heat flux but decreases with the increase in Reynolds number. It can be seen that for a given axial distance of heated plate and Reynolds number the temperature progressively decreases in the porous medium and increases with the direction of flow where the largest temperature of fluids is at the end of the plate. It is clear also that the increase of the Reynolds number leads to decrease in the temperature. As the Reynolds number decreases, the temperature variation extends from the heated wall and it is clear that the local Nusselt number increases with increasing the heat flux. The temperature distribution, which has been found by Fluent, agrees with behavior of experimental results. In the ANSYS (fluent) program, Nusselt number has been investigated to express the heat transfer rate, at constant heat flux, the figures (15) and (16) show that the local Nusselt number increases with increasing the velocity and decreases with the axial distance of the heated plate, The local Nu begins with high value at the inlet of entry length region due to the low difference between the air and heated wall temperature and thin thermal boundary layer formed, then progressively the difference between air and heated wall increases and the thickness of thermal boundary layer increases until reach the end of entry length region. In the ANSYS (fluent) program the heat transfer coefficient is multiplied by the total length (L) to find the local Nusselt number at any location.

5.3 Verification

The experimental and numerical temperature profile results for ($q^{''}$=2000 W/m^2, Opening 20%=2.15m/s) has been compared with each other, as shown in figure (17). It reveals that the numerical temperature profile follows the same behavior as the present experimental results but is approximately with a mean difference 7.1%. The experimental and theoretical Nusselt number results for ($q^{"}$=2000 W/m^2, Opening 20%=2.15m/s) has been compared with each other in figure (18). The Figure reveals that the numerical simulation Nusselt number follows the same behavior as the present experimental results, with mean difference 10.3%.The present result for temperature profile with axial direction in the porous medium with the vertical direction away from the heated wall shown in figures (4) agrees with the results of (Cui et.at 2001) [12] (experimental study) shown in figure (19) and (Kifah 2004) [13] (experimental and numerical study) shown in figure (20). The present result for temperature profile with vertical position shown in figures (5) agrees with the results of (Zhao and Song 2001) (experimental and numerical study) shown in figure (21). The present results for local Nusselt number with the length of heated plate shown in figure (7) agrees with results of (Calmidi1 et.at. 2000) [14] (experimental and numerical study) shown in figure (22). The present results for the average Nusselt number shown in figures (9) agrees with results of (Kifah 2004) (experimental and numerical study) shown in figure (23).

6. CONCLUSIONS

The results of the present work indicates that the temperature in the axial direction increases as Reynolds number decreases and heat flux increases. The temperature decreases with the vertical direction as the Reynolds number increases and the heat flux decreases. The heat transfer rate increase as the Reynolds number increases. The local Nusselt number increases with increasing the velocity and decreases with increasing the length of the heated plate. Comparison gives a good agreement between the present and previous works.

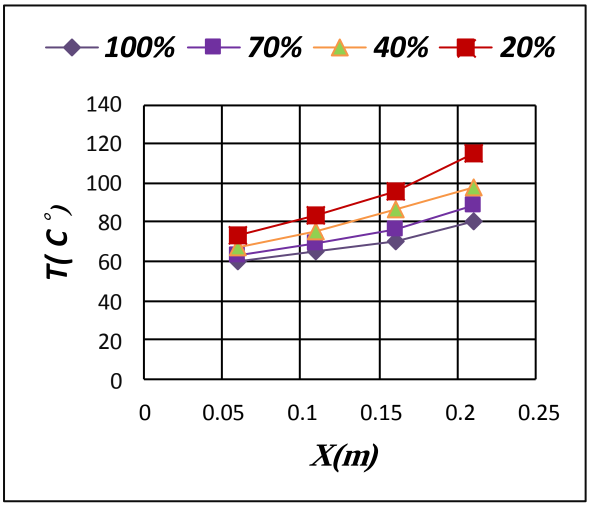


Fig. 4: Temperature Profile with Axial Position for ($q^{"}$=1000 W/m^2)

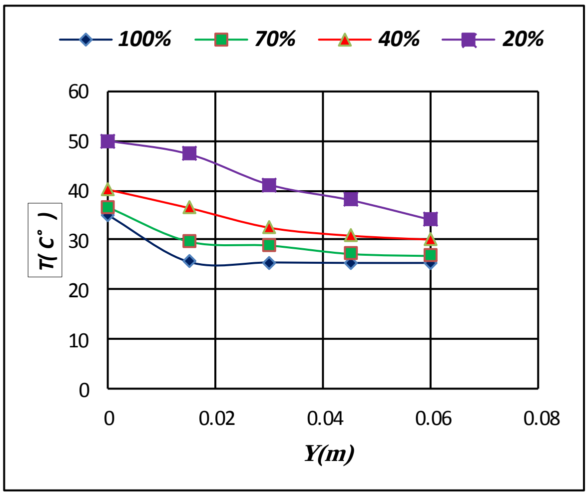


Fig.5: Temperature Profile with Normal Distance for ($q^{"}$=1000 W/m^2)

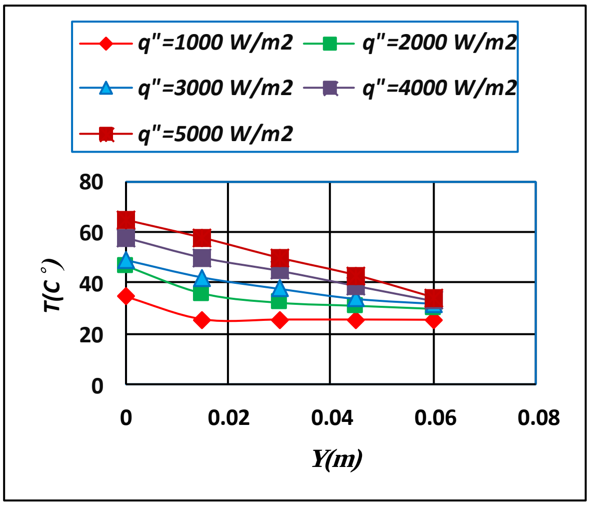


Fig.6: Temperature Profile with Normal Distance for Opening (100%=6.4m/s)

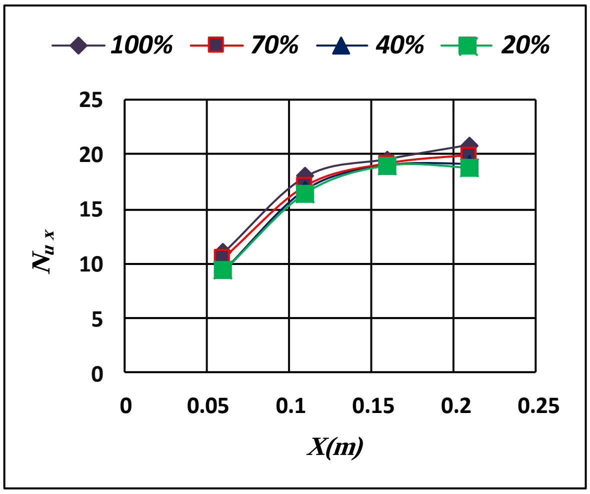


Fig. 7: Local Nusselt Number for ($q^{"}$=1000 W/m^2)

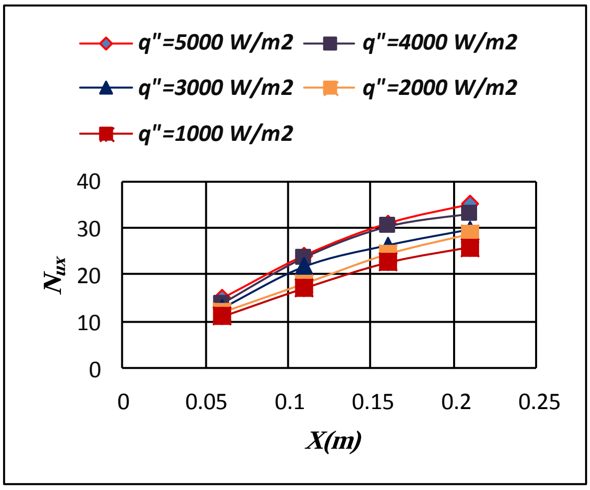


Fig. 8: Local Nusselt Number for Opening (100%=6.4m/s)

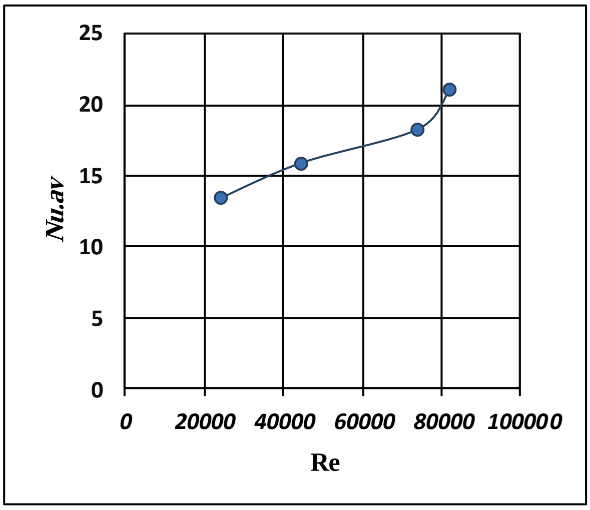


Fig. 9: Average Nusselt Number for (q''=1000 W/m^2)

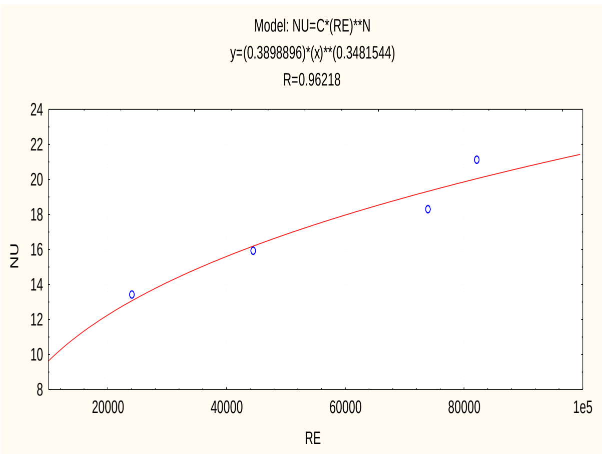


Fig.10: Nusselt Number Correlation of Experimental Result for (q''=1000 W/m^2)

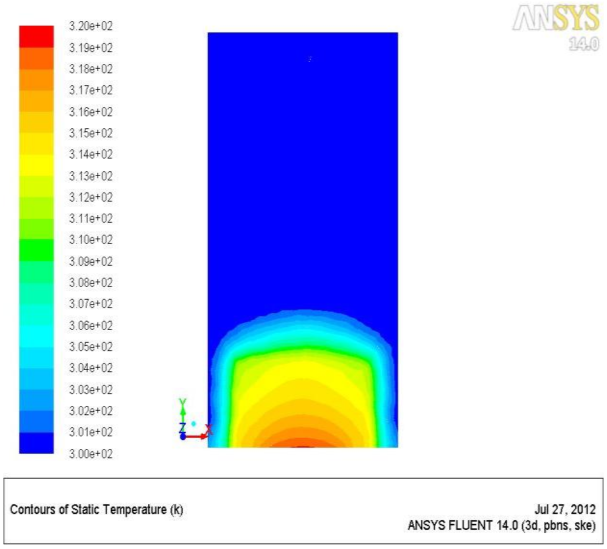


Fig. 11: Temperature Profile with Normal Distance for (q''=2000 W/m^2, Opening 100%=6.4 m/s, X= 0.11m)

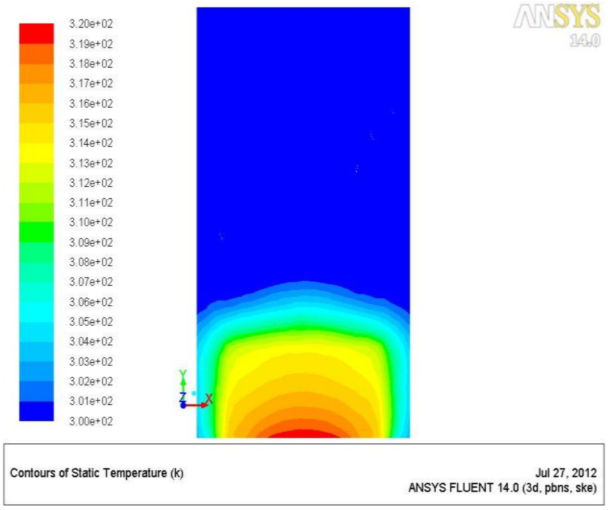


Fig. 12: Temperature Profile with Normal Distance for (q''=2000 W/m^2, Opening 100%=6.4 m/s, X= 0.21m)

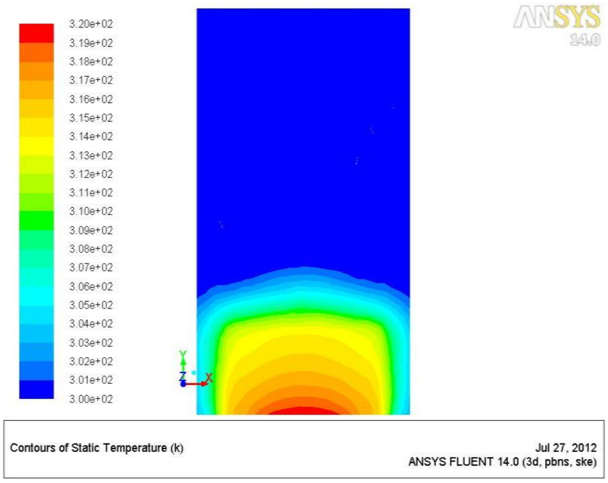


Fig. 13: Temperature Profile with Normal Distance for (q''=3000 W/m^2, Opening 100%=6.5 m/s, X=0.11)

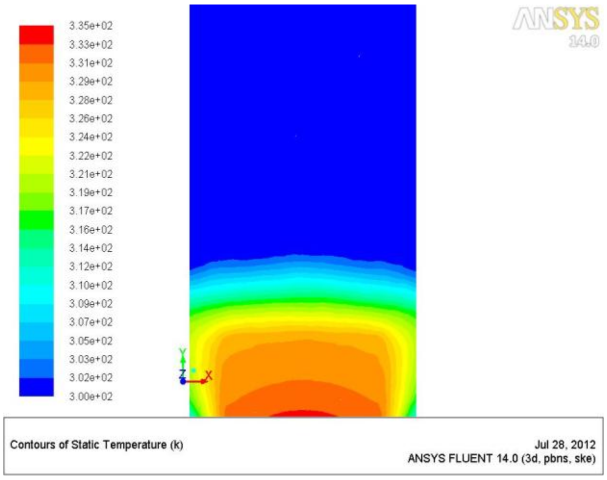


Fig. 14: Temperature Profile with Normal Distance for ($q^{''}$=3000 W/m^2, Opening 100%=6.5 m/s, X=0.21)

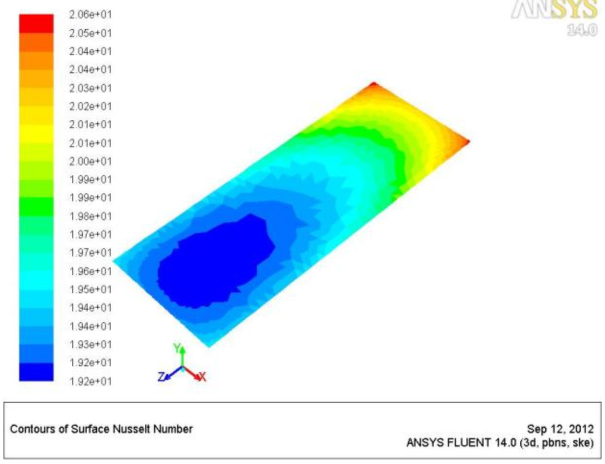


Fig. 15: Local Nusselt Number for ($q^{''}$=2000 W/m^2, Opening 70%=5.73m/s)

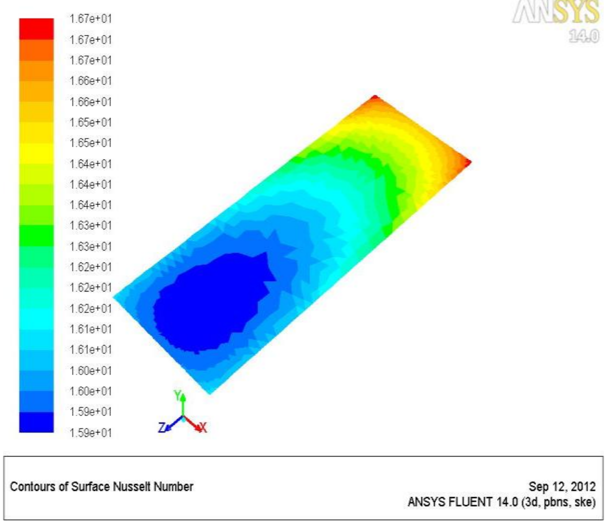


Fig. 16: Local Nusselt Number for ($q^{''}$=2000 W/m^2, Opening 20%=2.15m/s)

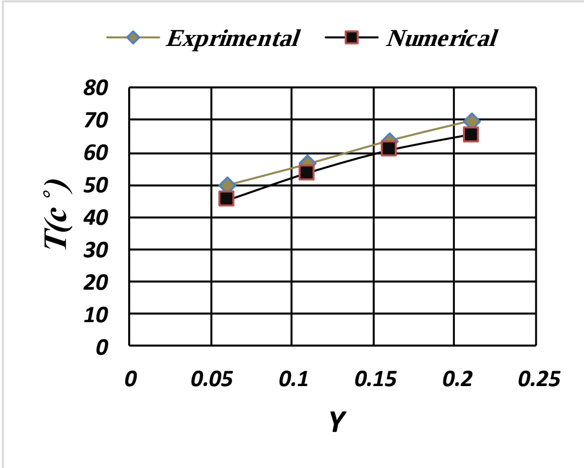


Fig. 17: Comparison of Experimental Temperature Profile with Numerical Results ($q^{''}$=2000 W/m^2, Opening 20%=2.15 m/s)

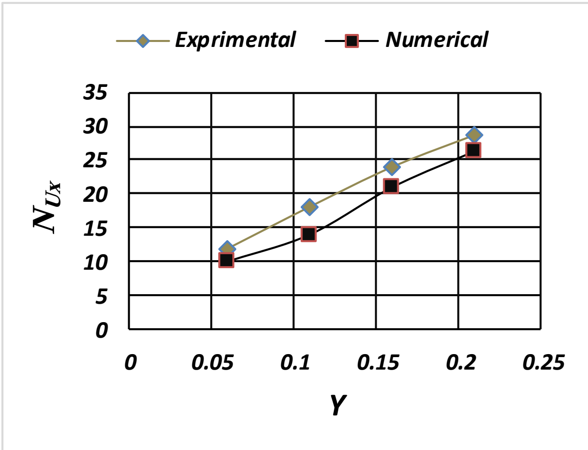


Fig. 18: Comparison of Experimental Nusselt Number with Numerical Results ($q^{''}$=2000 W/m^2, Opening 20%=2.15 m/s)

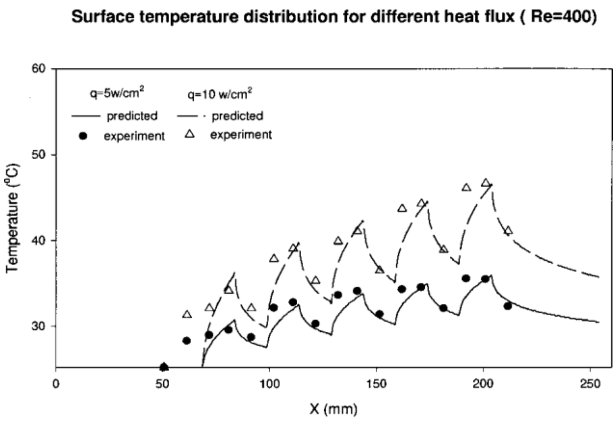


Fig. 19: The Surface Temperature Distributions on the Surface of a Heated Wall at The Reynolds Number Re=400(Cui et.at 2000)

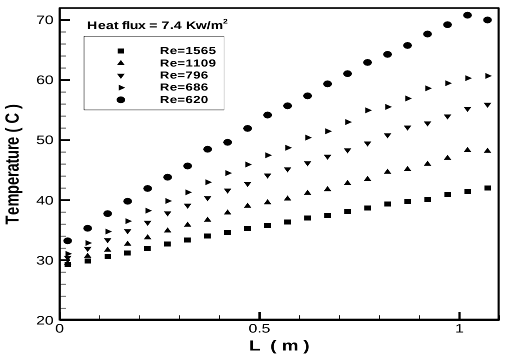


Fig. 20: Channel Wall Temperature Distributions (Kifah 2004)

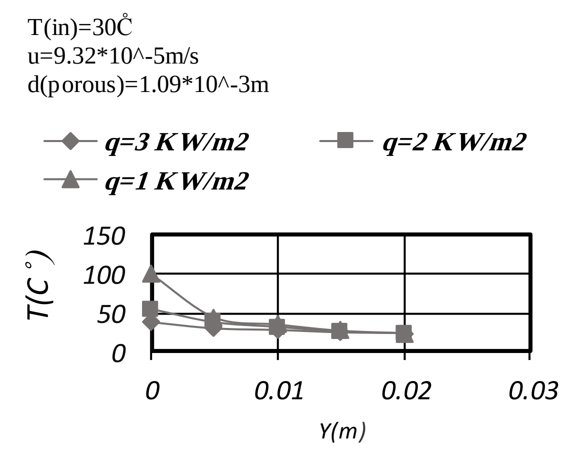


Fig. 21: Temperature Profile in Porous Media (Zhao and Song 2001)

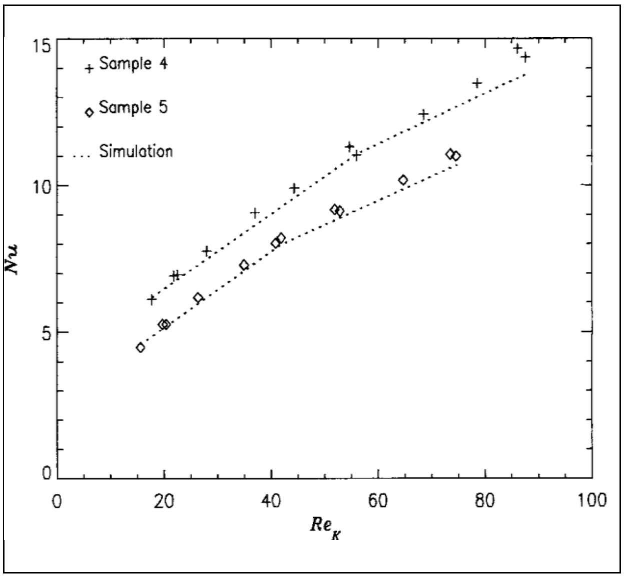


Fig. 22: Nusselt Number as a Function of Reynolds Number (Calmidi1 et.at. 2000)

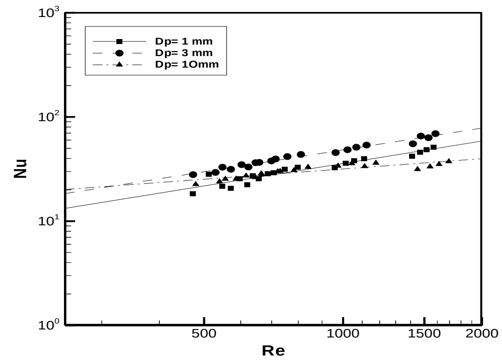


Fig. 23: The variation of The Experimental Average Nusselt Number with the Reynolds Number (Kifah 2004)

7. NOMENCALTURE

h Heat Transfer Coefficient
K Permeability
L Characteristic Length of the Plane Wall
Nu Nusselt Number
q'' Heat Flux
T Temperature
V Velocity in X-direction
X Flow Direction
Y Vertical Space Coordinate

7.1 Subscripts

∞ Location away from the wall outside the boundary layer.
f Fluid.
m Medium.
w Wall.
x Location in the x-Direction.

8. ACKNOELEDGMENTS

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