Experimental and Numerical Study on Forced Convection Through Porous Medium Under Non-Darcian flow

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Abstract— **An experimental and numerical work investigates the effect of metallic porous materials, inserted in a pipe, on the rate of heat transfer under non-Darcian flow condition. The pipe is subjected to a constant and uniform heat flux. The permeability and form coefficient of the porous medium inserted is found without applying heat flux and then the heat transfer experiment is conducted. The results obtained was lead to the conclusion that higher heat transfer rates can be achieved using porous inserts at the expense of a reasonable pressure drop. Experiment is repeated using water in different velocity and different porosity . From the result also concluded that the heat transfer from wall to water is increases with increase in Reynolds number and heat transfer decrease with increase in porosity .**

Keywords— Non-Darcian flow, Porous media, Reynolds number, Porosity.

NOMENCLATURE

$\in$-porosity
u- velocity[m/s]
K-permeability of porous medium[m^2]
C-form coefficient[1/m]
$\frac{\Delta p}{L}$-pressure gradient[kpa/m]
C_F-Forcheimers coefficient
μ-dynamic viscosity[Ns/ m^2]
ρ- density of fluid[kg/m^3]
K_{eff}-effective thermal conductivity[w/mk]
k-thermal conductivity of fluid[w/mk]
Re-reynolds number
Nu-nusselt number
Pe- peclet number

I. INTRODUCTION

Convective heat transfer in a porous medium has been a subject of intensive studies during the past two decades because of its wide applications including geothermal energy engineering, groundwater pollution transport, nuclear waste disposal, chemical reactors engineering, insulation of buildings and pipes, and storage of grain and

coal, electronic cooling, drying processes, solid matrix heat exchangers, heat pipe, enhanced recovery of petroleum reservoirs, etc. Cheng [1] provides an extensive review of the literature on convection heat transfer in fluid saturated porous media with regard to applications in geothermal systems. The state of art concerning porous media models has been summarized in the book by Nield and Bejan [2] as well as the book by Kaviany [3]. For the case of boundary layer flow over a flat plate embedded in a porous medium, the overall Nusselt number over the plate heated at a constant heat flux is given by [2]

$$Nu = 1.329Pe^{0.5} \quad (1)$$

where the Nusselt number is defined as Nu=(hdp/ ke) and the Peclet number as Pe=(udp/ke/ρcp) with h being the heat transfer coefficient, dp the diameter of the porous medium, U the velocity, and ρ, Cp, as well as ke representing the fluid density, the fluid specific heat, and the effective thermal conductivity of the porous medium, respectively.

Recently, increased demands for dissipating high heat fluxes from electronic devices, high power lasers, and X-ray medical devices have created the need for new cooling technologies as well as improvements in existing technologies. To meet such demands, different cooling schemes have been proposed. One major category of heat exchangers for such applications is referred to as porous media heat exchangers. The basic idea of the porous media heat exchangers is that enhanced cooling can be achieved because (i) larger surface areas available in porous particles as extended surfaces for heat transfer and (ii) mixing of fluids due to the presence of particles. A pumped single-phase porous media heat exchanger has recently demonstrated the capability for removing high heat fluxes. One of the major disadvantages using a porous heat exchanger is the large pressure drop across the heat exchanger. In order to overcome this disadvantage, one of the best methods is to reduce the flow velocity while keeping a higher heat transfer coefficient. This implies that the study of heat transfer enhancement in porous media is important.

Al-Nimr and Alkam [4] numerically investigated the problem of transient forced convection flow in a concentric annuli partially filled with porous substrates located either on the inner or the outer cylinder. An increase of up to 12 times in the Nu number was reported in comparison with the clear annuli case and the superiority in thermal performance of the case when the porous substrate was emplaced to the inner cylinder was outlined. Based on the results obtained, Alkam and Al-Nimr [5] further investigated the thermal performance of a conventional concentric tube heat exchanger by emplacing porous substrates on both sides of the inner cylinder. Numerical results obtained showed that porous substrates of optimum thicknesses yield the maximum improvement in the heat exchanger performance with moderate increase in the pumping power.

It is mentioning that the literature contains a large number of numerical investigations and a very small number of experimental studies on the use of porous materials for natural and forced flow applications. In the present work water is flowed flow through a porous media made by filling small carbon steel balls of 6.35 mm diameter inside a brass tube of 72.8mm.Water flow rate is varied and for each particular flow rate predetermined radial heat flux is given. The variation in temperature and pressure for different flow rates is plotted against the different flow velocities. The present work is done for porosity (∈) of 0.4

II. Problem Definition

The schematic diagram of the problem is presented in Fig. 1. Flow in a pipe fully filled with porous material and heated from the exterior with a constant and uniform heat flux, is considered for both numerical and experimental investigation. In the numerical study, where the flow is assumed to be laminar, a stream of water with uniform velocity and temperature is considered to enter the pipe. Its thermo physical properties are considered to be constant for all numerical simulations. The experimental study investigates the heat transfer enhancement over a range of pore based Re 0 to 40

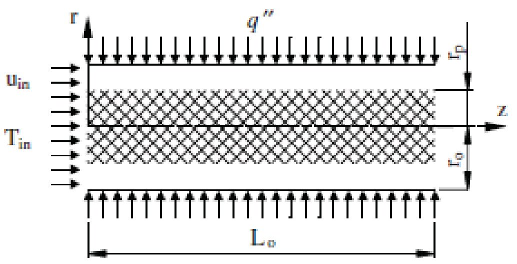

Figure1.Schematic diagram of the problem

III. Methodology

A. Numerical analysis

The numerical model has been developed for the problem defined. The numerical procedure comprises of a two dimensional computational model. Numerical modeling and analysis of the problem was done by using commercially available computational fluid dynamic software fluent in ansys workbench 14

B. Governing equations

Continuity equation

$$\frac{\partial(\rho u)}{\partial z}+\frac{1}{r}\frac{\partial(\rho v)}{\partial r} = 0 \qquad (2)$$

Equation (2) implies that total mass flux in z and r direction is constant. First term means total mass flux z direction and second term means total mass flux r direction.

z-momentum equation

$$\frac{\partial(\rho uu)}{\partial z}+\frac{1}{r}\frac{\partial(\rho uv)}{\partial r} = -\frac{\partial(p)}{\partial z}+\frac{\partial}{\partial z}\left(\mu\frac{\partial u}{\partial z}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\mu\frac{\partial u}{\partial r}\right)-f\left(\frac{\mu u}{k}\right)-f\left(\frac{\rho C_F}{\sqrt{k}}\right)|u|u \qquad (3)$$

Equation (3) implies that total momentum in z direction is constant. Left hand side indicates the momentum force in z direction. While the right hand side specifies the viscous and pressure forces in z direction

r- momentum equation

$$\frac{\partial(\rho uv)}{\partial z}+\frac{1}{r}\frac{\partial(\rho vv)}{\partial r} = -\frac{\partial(p)}{\partial r}+\frac{\partial}{\partial z}\left(r\mu\frac{\partial u}{\partial z}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\mu\frac{\partial v}{\partial r}\right)-f\left(\frac{\mu v}{k}\right)-f\left(\frac{\rho C_F}{\sqrt{k}}\right)\times|u|v\frac{\mu v}{r^2} \qquad (4)$$

Equation (4) implies that total momentum in r direction is constant. Left hand side indicates the momentum force in r direction. While the right hand side specifies the viscous and pressure forces in r direction

energy equation

$$\frac{\partial(\rho cuT)}{\partial z}+\frac{1}{r}\frac{\partial(\rho crvT)}{\partial r} = \frac{\partial}{\partial z}\left(k_e\frac{\partial T}{\partial z}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(rk_e\frac{\partial T}{\partial r}\right) \qquad (5)$$

Equation (5) implies that total energy is conserved. Left hand side indicates the energy stored z and r direction. While the right hand side specifies the thermal energy change in z and r direction. The parameter f is set to 1 for flow in porous medium and zero for clear flow region

C. Experimental setup

Tests were carried out using brass pipe of diameter 72.8mm.All section of pipes were joint together by flanges and screws. The test section was heated at the exterior over its entire length with a ceramic band heater. The magnitude of the heat flux was adjusted by varying the intensity of the current measured with the ammeter and supplied by the direct current power supply. A honeycomb rectifier of 50 mm length installed at the entrance of section in order to remove eddies and provide a more uniform velocity profile.

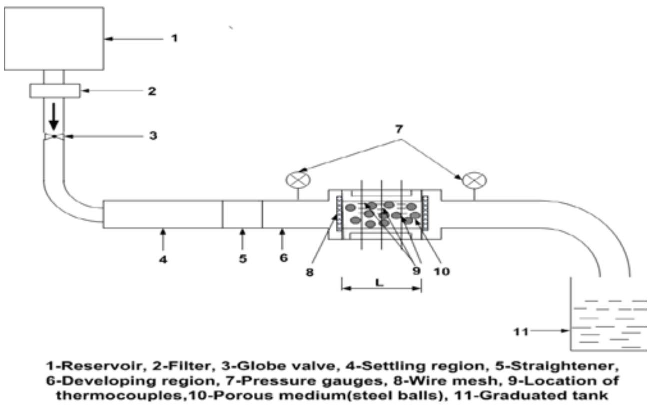

Figure 2.Schematic diagram of experimental setup

The pressure drop caused by inserting porous media inside the heated section was measured with two pressure gauges. Temperature of fluid inside the heated section is measured using a bundle of 5K-type thermocouples at four axial positions. The mass flow rate of fluid flowing inside the rig is adjusted with the help of valves.

Figure 3.Photograph of experimental setup

IV. RESULT AND DISCUSSIONS

The experiment is conducted initially without applying heat flux and the variation of velocity and pressure gradient is plotted as shown in figure 4.After plotting this curve the equations for the curves were compared to the HDD equation and the permeability(K) and form coefficient(C) is found.

Were HDD equation is

$$\frac{\Delta P}{L} = \frac{\mu}{K}U + C\rho U^2$$

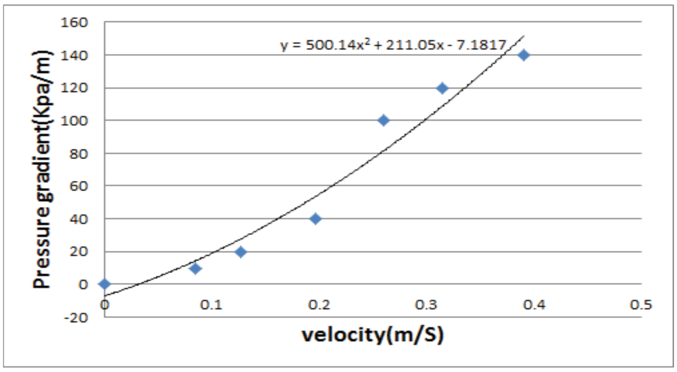

Figure 4.Variation of pressure gradient for different velocity (experimental) ∈ =0.4

Tabulated value in table 1 for permeability and form coefficient was used for the further calculations in both experimental and numerical simulations.

Table 1 Characteristics of the porous medium

Experimental Result	
Permeability	3.92×10^{-6}
Form coefficient	0.5041

Heat transfer experiment is conducted using a heat flux of 310000 w/m^2.and velocity varied from 2 mm/sec to 5 mm/sec, which is the non Darcian region used in this heat transfer experiment. porosity varied from 0.4 to 0.8

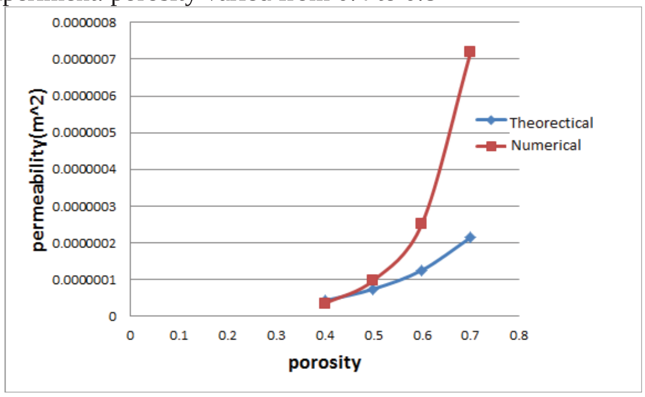

Figure 5 Variation of permeability in theoretical and numerical

Figure 5 indicate the variation in permeability in numerical and theoretical for different porosity. Theoretical value of permeability is found using an equation given by

$$K = \frac{D_p{}^2 \text{Ø}^2}{180(1-\text{Ø})^2} \qquad (6)$$

Where Ø is the porosity

From the figure it is clear that at lower porosity the variation between theoretical and numerical is very small, but the variation increases considerably as the porosity increases.

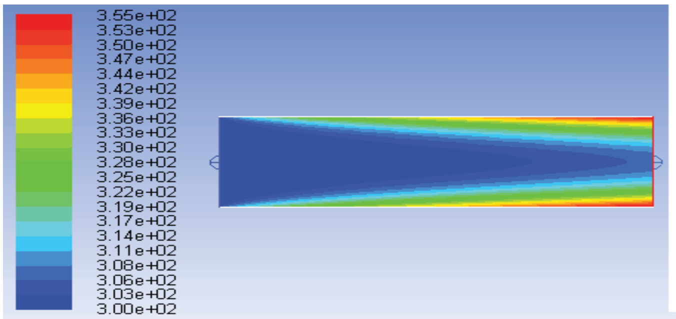

Figure 6 Contour temperature along test section
Re=17.83

Figure 6 shows the variation in temperature along the test section in numerical simulation with an inlet velocity of 2.4 mm/s. from figure it is clear that water is entering the test section at 300 K and gets heated while passing through the porous medium up to 355K.Temperature near to the wall is at a higher temperature than at the center.

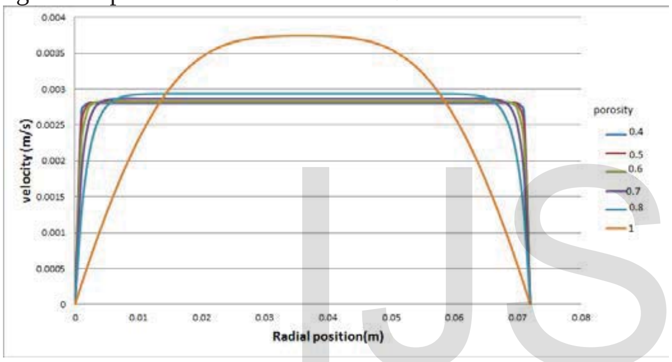

Figure 7 Velocity profile inside the porous medium for different porosity
Re=17.83

In figure 7 the velocity profile inside the porous medium for different porosity is plotted. From that curve we can say that the flow inside porous medium is a plug flow.as the porosity increases the drag force reduces so the velocity is increases and the velocity profile changes and finally it becomes parabolic increase of clear flow

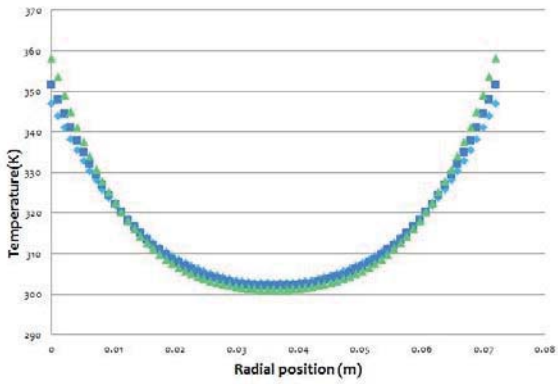

Figure 8 Temperature distribution in radial direction

The variation of temperature distribution along the radial direction is shown in figure 8. In this problem the local thermal equilibrium is assumed, that means inside the test section the temperature of water and steel ball is same everywhere. In figure 8 the temperature distribution is measured by varying the porosity and it shows that at lower porosity temperature at wall is lower than that of at higher porosity. This indicate the heat transferred from wall to porous medium is more effectively convected to water at lower porosity than at higher. This is because at lower porosity the heat transfer area available is more and this area reduces as the porosity increases.

The temperature distribution along the axial direction is also important in the heat transfer problem.

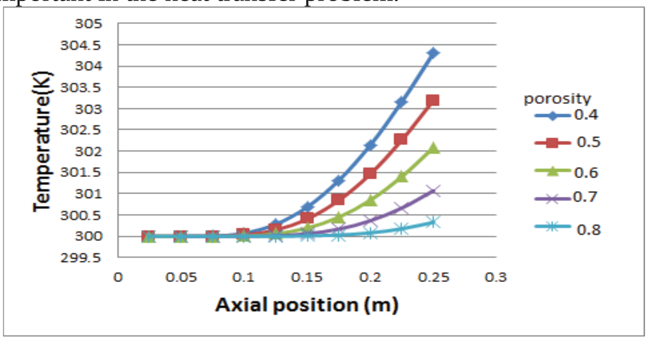

Figure 9.Variation axial temperature at center line of test section, Re=0.4

variation in axial temperature is shown in figure 9.The thermocouples fitted in center at different position reads the temperature for different velocity.

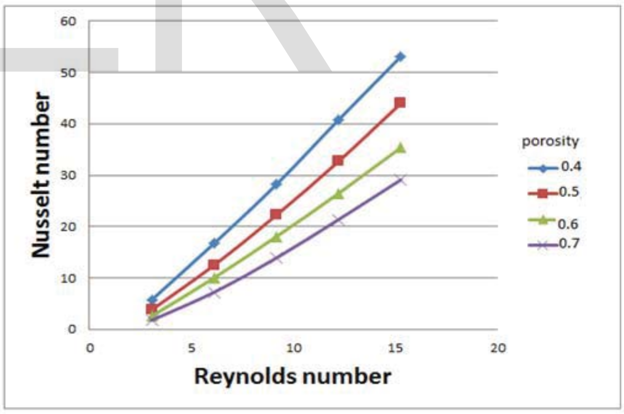

Figure 10.Variation of Nusselt number versus Rrynolds number-Darcian flow

By analyzing the temperature distribution we can say that at lower porosity the temperature difference between inlet and outlet is almost 5 K but at higher porosity this difference reduces to less than 1 K .This shows at lower porosity the heat transfer is much effective than at higher porosity in porous medium.

Figure 10 and Figure 11 represent the Nusselt number variation for different Reynolds number in case of Darcian and non Darcian flow respectively . As the Reynolds number increases the Nusselt number also increases. This indicate the heat transfer rate is higher in higher velocity. From this curve we can say that the heat transfer variation in non darcian flow is parabolic in nature and much more than that of darcian flow, which is almost linear. This variation in heat transfer is because the velocity is more in darcian flow and due to this high velocity thermal and momentum dispersion effect increases. The increase in mass flow rate is also a reason for the increased heat transfer

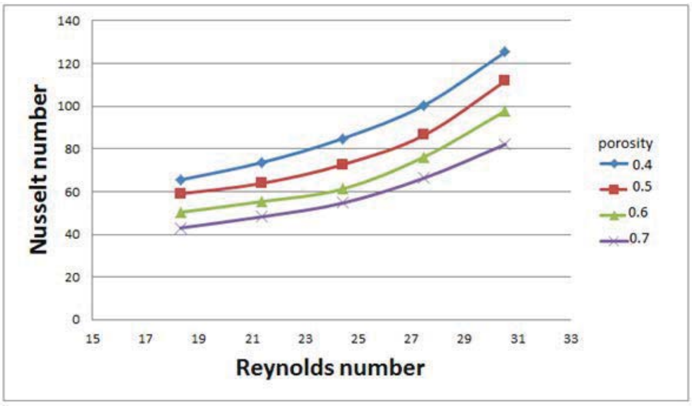

Figure 11Variation of Nusselt number versus Rrynolds number-non Darcian flow

V. CONCLUSION

This experimental and numerical study investigates the effect of porous inserts on the rate of heat transfer, for surface of a pipe subjected constant and uniform heat flux and the water flowing inside it. The following conclusions is obtained.

(1) The use of porous medium in heat transfer will always increase the rate of heat dissipation with the expense of pumping power. The reason for the heat transfer enhancement when using porous materials are as follows: Increase in effective thermal conductivity due to flow redistribution(dispersion), tortuosity. From Nusslet number and Reynolds number curve we can say the heat transfer increase with increase in Reynolds number and decrease with increase in porosity which is a validation of papper by Bogdan I. Pavel, Abdulmajeed A. Mohamad in 2004, “An experimental and numerical study on heat transfer enhancement for gas heat exchangers fitted with porous media,”[6]

(2) Also experimental determination of the characteristics of a porous material, permeability and form coefficient is very important because its proper determination will ensure a successful numerical simulation. The variation of pressure and velocity curve is plotted and the HDD equation is compared with the equation of the curve for getting the permeability and form coefficient. The result is in a good agreement with the paper of L. Wilson Arunn Narasimhan S. P. Venkateshan Permeability and Form Coefficient Measurement of Porous Inserts With Non-Darcy Model Using Non-Plug Flow Experiments[7]

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