AN EXPERIMENTAL INVESTIGATION OF FORCED CONVECTION IN A PACKED PIPE WITH A POROUS MEDIUM

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ABSTRACT

Forced convection heat transfer in a horizontal packed pipe is studied experimentally. The objective of the present work is to study the effects of Reynolds number (based on the particle diameter $61.8 \leq Re_d \leq 1210.6$ and on the pipe diameter $701 \leq Re_p \leq 3414$) and the ratio of pipe diameter to particle diameter on the forced convection and to obtain the experimental correlations. An experimental apparatus is designed and constructed. The outer surface of the pipe is heated under constant heat flux. Experiments are carried out for water and sand grains of 2.67, 4.175, 5.67, 8.15 and 12 mm average particle diameters. The results show that the average Nusselt number increases with the increase of Reynolds number. This result is verified when Reynolds number increases as a result of increasing the mass flow rate. Also, the average Nusselt number slightly increases with decreasing particle size. Good agreement between the present and the previously published results is observed.

KEY WORDS: Heat transfer, Forced convection, porous media, packed pipe

1. INTRODUCTION

The number of investigations on convective heat transfer through a saturated porous medium has been on the rise during the past decade. This is due to the broad range of many engineering applications such as chemical catalytic reactors, compact thermal collector-storage systems, building thermal insulation, solid matrix heat exchangers, petroleum reservoirs geothermal appertains, packed spheres ground water
hydrology and the manufacturing numerous products in chemical industry.

Flow through a packed pipe with a porous medium has three types of regimes. Fand and Thinakaran, 1990, [1] have studied the flow through infinite porous media whose matrices are composed of spheres in these regimes. The types of regimes, namely, Darcy regime ($10^{-5} \leq \text{Re}_d \leq 2.3$), Forchheimer regime ($5 \leq \text{Re}_d \leq 80$) and turbulent regime ($\text{Re}_d \geq 120$) plus two regimes of transition between these three regimes. They studied experimentally the influence of the wall on the flow through the pipes packed with the porous medium for $(D/d)$ from 1.4 to 40. The results showed that for $(D/d)$ less than or equal to 40, the wall effect becomes significant and the flow parameters are functionally dependent on this ratio.

Most of the existing studies deal primarily with the mathematical simplification for porous media based on Darcy law which neglects the effect of solid boundary and viscous and inertia forces. The measurements of Benenati and Brosilow, 1962, [2], show a distinct porosity variation in packed beds. From these measurements, it is found that the porosity is a function of the distance from the boundary where the high porosity region is close to the external boundary. Kaviany, 1985, [3] studied theoretically the laminar flow through a channel bounded by isothermal parallel plates. In this study the inertia effect and the axial conduction were neglected. The results show that the entrance length decreases rapidly as the porous media shape parameter increases; and is proportional to $\gamma^{-2}$. Also, the average Nusselt number increases with increasing this parameter.

The effects of the solid boundary and inertia on forced convective heat transfer in porous media over flat plate were analyzed for a constant porosity by Vafai and Tien, 1980, [4]. These effects were shown to play a significant role in highly permeable media, high Prandtl number fluids and large pressure gradients. Vafai, 1984, [5] studied numerically the effects of variable porosity, channeling and inertia on the flow and the heat transfer characteristics in porous media where an impermeable surface is found. The results of these studies show that the maximum velocity occurs near the wall due to the maximum porosity at the wall. Also, the large pressure gradient causes
a higher peak velocity near the wall which decreases the thermal boundary layer. As a result of decreasing the thermal boundary layer, the average Nusselt number increases. El-Kady, 1994, [6] studied numerically forced convection heat transfer and flow in an annular channel filled with a saturated porous medium under constant wall temperature. The results show that the non-Darcian effects have a significant influence on velocity profiles. Also, the cooling effect at the wall propagates faster in the duct with decreasing the particle size. This leads to that the values of Nusselt number at the inner and the outer walls of the channel increase. Hunt and Tien, 1988, [7] studied numerically velocity and temperature profiles for non-Darcian flow. They indicated that flow inertia, near wall porosity and thermal dispersion significantly alter the velocity and temperature profiles. The velocity profiles indicated that the maximum velocity occurs at a distance of half particle diameter, d/2, from the wall. The average Nusselt number based on the bed diameter nearly increases linearly with the particle Reynolds number. Poulikakos and Renken, 1987, [8] studied numerically the forced convection in a channel filled with a saturated porous medium where effects of flow inertia, variable porosity and Brinkman friction were considered. They showed that including these effects decrease the peak velocity near the wall. Also, the above mentioned effects increase the values of the average Nusselt number more than that of the Darcy model. Poulikakos and Kazmierczak, 1987, [9] studied theoretically the fully developed forced convection in a channel partially filled with a porous matrix for constant heat flux or constant wall temperature. Brinkman model is used to analyse the porous region where the inertia effect was neglected. They showed that as the porous medium thickness increases, both the flow velocity and the average Nusselt number decrease. Also, the average Nusselt number increases with increasing the ratio of the effective thermal conductivity to the fluid thermal conductivity (k_e / k_f).

The present work aimed at studying experimentally the effects of Reynolds number and the ratio of pipe diameter to particle diameter on forced convection. An experimental apparatus is designed and constructed. The outer surface of the pipe is heated under constant heat flux. Sand grains of different diameters are used.
2. EXPERIMENTAL APPARATUS AND PROCEDURES

2.1 Test Section

The experimental apparatus employed in this investigation is shown schematically in Fig.1a. It comprises test section, pumping circuit, instrumentation and power supply with associated controls.

The test section is composed of a horizontal copper pipe of 32 mm inside diameter and 3.2 mm thickness that is heated under constant heat flux as shown in Fig.1b. The copper pipe is packed with sand grains as a porous medium. The average diameters of the used sand grains are 2.65, 4.15, 5.67, 8.15 and 12 mm. The pipe-surface is heated using the main heater that is divided into two circuits and wounded uniformly around the outer surface of the pipe. The guard heater is wound between the two layers of insulation. Each layer is made of 12.7 mm asbestos rope insulation. The power supplied to the guard heater is controlled using a voltage regulator (1KW) to compensate the heat loss from the main heater. The power supplied to the main heater is calculated by measuring the voltage drop and the passing current in each circuit. The current through each circuit is measured using a multi-meter (division 0.2A, range 1-40 A) while the voltage drop is measured using a digital multi-meter (resolution 0.1 mV and accuracy ± 0.5 %). After adjusting the inlet and bypass valves to give the required flow rate, the pump is turned on for 30 min to achieve the uniform temperature along the test section. Then the heater is turned on and the reading of the thermocouples are recorded every 20 min. At steady state, reading of the thermocouples, the orifice-meter head, and the mercury manometer head are recorded. The inlet water temperature is nearly constant. Attached to the test section an entrance length, which is made of steel pipe of length 175 cm. In this section, the flow becomes uniform before it enters the heated section. Water flow is supplied to the packed pipe via a centrifugal pump of 3/4 HP. The mass flow is controlled by using two valves. The first is placed at the bypass line and the other is at the inlet to the orifice-meter to provide a range from 0.7 to 4 l/min. The water mass rate is measured by an orifice meter which is connected to the
Fig 1a: Schematic diagram of the experimental apparatus

1-The copper pipe (32 inside diameter, 3.2 mm thickness)
2-Electrical insulation
3-Main heater
4-Asbestos insulation (layer 1)
5-Guard heater
6-4-Asbestos insulation (layer 2)
7-Thermocouples

Fig. 1b: Details of the test section

Dimensions in mm

Fig. 1c: Thermocouples distribution
inverted manometer with minimum division 1 mm. Temperatures along the pipe surface and at both inlet and exit of the test section are measured using copper-constantan thermocouples of 0.3 mm diameter. Seventeen pairs of thermocouples are embedded into the pipe surface to measure the pipe surface temperatures as shown in Fig.1c. The readings of thermocouples are recorded using a digital thermometer with one decimal digit. Two pressure tabs are installed at both the inlet and the exit of the test section and connected to a U-tube mercury manometer with minimum division 1 mm.

2.2 Experimental Procedure

The pipe is filled with one type of a porous medium to obtain the desired ratio of pipe to particle diameters. The electrical power input to the main heater is determined by measuring both the voltage drop and the current for each circuit. The flow rate of the water is determined using the orifice meter. The head difference (in mm of water) is measured by the inverted manometer. The pressure drop across the test section is determined as a head difference in mm Hg using the mercury manometer. The temperatures of the pipe surface and flowing water are measured. All different readings are taken every 20 min until the readings are nearly constant. After this time, the system may be considered in a steady state. The above procedure is carried out for different flow rates for the same ratio of pipe diameter to particle diameter.

2.3 Method of Calculations

2.3.1 Calculation of heat transfer coefficient

The local mean surface temperature of each segment, $T_{sm}(i)$, is considered to be the mean value between the top and the bottom measured temperatures as

$$T_{sm}(i) = \frac{T_{top} + T_{bottom}}{2.0}$$
The flow bulk temperature is calculated as
\[ T_{b(i)} = T_{b(i-1)} + \frac{q (\pi D \Delta z(i))}{m_t C_p} \]

The flow mean bulk temperature corresponding to each pair of thermocouples is calculated as the arithmetic mean value between the inlet and outlet temperatures of the segment as follow;
\[ T_{bm(i)} = \frac{T_b(i) + T_b(i-1)}{2.0} \]

The local heat transfer coefficient from the pipe inner surface to the flowing fluid is calculated as:
\[ h(i) = \frac{q}{T_{sm(i)} - T_{bm(i)}} \]

2.3.2. Dimensionless parameters

The porosity of the porous medium in the range of the present experimental study is calculated experimentally as:
\[ \varepsilon = \frac{V_v}{V_t} \]

where,
\[ V_v: \text{the volume of the water,} \]
\[ V_t: \text{the volume of the saturated porous medium(sand grains and water)} \]

Another important property of the porous medium is the permeability of the porous medium, \( K \), which can be calculated directly using the Kozeny-Carmen equation that is used in Fand and Thinakaran, 1990, [1];
\[ K = \frac{d^2 \varepsilon^3}{180 (1-\varepsilon)^2} \]

Darcy number is calculated from the definition;
\[ Da = \frac{K}{D^2} \]

Porous media shape parameter is calculated from the definition;

\[ \gamma = \sqrt{\frac{D^2 \varepsilon}{K}} \]

Reynolds number based on the pipe diameter, \( Re_D \), and on the particle diameter, \( Re_d \), is calculated respectively as;

\[ Re_p = \frac{V_o D}{\nu} \]
\[ Re_d = \frac{V_o d}{\nu} \]

The average heat transfer coefficient is calculated from;

\[ \bar{h} = \frac{1}{L} \int \bar{h} dz \]

The average Nusselt number based on the pipe diameter and on the particle size is calculated respectively as;

\[ \bar{Nu}_D = \frac{\bar{h} D}{k_f} \]
\[ \bar{Nu}_d = \frac{\bar{h} d}{k_f} \]

The fluid properties were taken at the flow mean temperature.

3. RESULTS AND DISCUSSIONS

Experimental runs were performed to obtain heat transfer characteristics including pipe-surface and flow bulk temperatures and the heat transfer results in terms of the average Nusselt number at the heated pipe surface.

The behavior of heat transfer rate from the pipe surface to the fluid was examined. Temperatures at the pipe surface and the inlet were measured. As the pipe-surface temperatures are measured, the flow bulk temperatures can be calculated from the heat balance method. The runs were carried out for flow rates ranged from 0.728 to
3.973 l/min for the ratios of pipe diameter to particle diameter 11.99, 7.67, 5.64, 3.93 and 2.67. The Reynolds number, $Re_d$, is ranged from 61.8 to 1210.6. The axial pipe-surface and the flow bulk temperature distributions are shown in Fig. 2. This Figure presents surface and bulk temperature distributions for sand grain of diameter 2.67 mm. As Reynolds number based on the particle diameter increases, the pipe surface and the flow bulk temperatures difference between inlet and outlet of the pipe decrease. This refers to the increase of heat transfer rate between pipe surface and the fluid flow through the porous medium as shown in Fig. 2. Good agreement between present experimental results and numerical results of Khalil, [10] is obtained. Figure 3 shows the decrease of the Reynolds number due to the decrease of the particle size for flow rate equal to 0.722 l/min, the pipe surface and flow bulk temperatures decrease. This can be attributed to the increase of the pore velocity and channeling effect at the small particle size. As a result of increasing the pore velocity and the channeling phenomena, the rate of heat transfer increases which leads to decrease the pipe surface temperature. In order to identify the thermally fully developed region for a packed pipe with a porous medium, it is useful to consider the corresponding case for unpacked pipe which is heated at constant heat flux. For the case of an unpacked pipe that is heated under constant heat flux on the pipe surface, it is well known that the bulk and the wall temperatures rise linearly in the fully developed region and at the same rate. It is obvious from Fig. 2 that the pipe-surface and flow bulk temperatures rise linearly and at the same rate where the difference between the pipe surface and the flow bulk temperature is nearly constant at a distance from the inlet of the heated section equal to (8-10)D.

Figure 4 shows that the local Nusselt number decreases with the axial distance in the flow direction. Local Nusselt number reaches an asymptotic value at about (8-10)times the pipe diameter where the flow in the packed pipe is considered thermally fully developed. It is clear from Fig. 4 that as Reynolds number increases due to increase of flow rate, the Nusselt number increases. Figures 5-6 show a comparison of the average Nusselt number between the present experimental results and numerical
1. \( \dot{m}=0.013 \text{ kg/s, } Re_d = 61.8 \text{ and } P^* = 1460.1 \)
2. \( \dot{m}=0.025 \text{ kg/s, } Re_d = 109.5 \text{ and } P^* = 1088.9 \)
3. \( \dot{m}=0.0359 \text{ kg/s, } Re_d = 150.4 \text{ and } P^* = 989.9 \)
4. \( \dot{m}=0.044 \text{ kg/s, } Re_d = 189.17 \text{ and } P^* = 882.3 \)

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**Fig. 2:** Pipe-surface and flow bulk temperature distributions at different Reynolds number (\( d=2.67 \text{ mm} \))

**Fig. 3:** Effect of particle size on the pipe-surface and the flow bulk temperature distributions
Fig. 4: Local Nusselt number distribution at different Reynolds number (d=2.67 mm)

Fig. 5: Comparison between the experimental and numerical [10] results for the average Nusselt number at different Reynolds number
results [10]. They show that Nusselt number increases with increasing Reynolds number. This can be attributed to the increase of the effective thermal conductivity of the saturated porous medium, pore velocity and the mixing levels through the packed pipe. Figure 7 shows a comparison of pressure drop between the present experimental results and the measurements of Fand and Thinkaran, 1990, [1] at different Reynolds numbers and Prandtl number equal to 5. A good agreement is obtained as shown in Fig. 7. Figure 8 shows the pressure drop per unit length of the packed pipe versus the Reynolds number based on the pipe diameter at different ratios of pipe-particle diameter. It is observed that the particle diameter decreases, the pressure drop increases.

4. EXPERIMENTAL CORRELATION

Based upon the experimental heat transfer results, the following correlations are presented for the average Nusselt number as a function of Reynolds number and the ratios of pipe diameter to particle diameter for water as:

\[
\overline{\text{Nu}}_D = 0.3 \text{Re}^{0.853}_D (D/d)^{0.0192}
\]

\[701 \leq \text{Re}_D \leq 3414\]

\[2.67 \leq D/d \leq 11.99\]

The maximum relative deviation between the computed and experimental values is 8.5%. This deviation is usually obtained at low Reynolds number and large ratios of pipe diameter to particle diameter.

5. CONCLUSIONS

An experimental correlation between Nusselt number as a dependent variable and Reynolds number besides the ratios of pipe diameter to particle diameter as independent variables is obtained. Local pipe-surface and flow bulk temperatures increase linearly in the axial direction and nearly at the same rate in the thermal fully
Fig. 6: Comparison between the experimental and numerical [10] results for the average Nusselt number at different Reynolds number (based on pipe diameter).

Fig. 7: Comparison of pressure drop between the present measurements and previous measurements [1] at Pr=5.
Fig. 8: pressure drop measurements across the test section

developed length. The local Nusselt number reaches an asymptotic value at axial distance of (8-10)D from the inlet of the pipe. The average Nusselt number increases with increasing Reynolds number. As the particle size decreases, the average Nusselt number slightly increases and the pressure drop increases. On the basis of the good agreement between the experimental and numerical results, it seems possible to determine the heat transfer parameters for forced convection in a horizontal packed pipe using either method. However, the numerical analysis gives more information, including the velocity field which is difficult to be obtained experimentally.

References


NOMENCLATURE

Symbols
C specific heat J/kg K
D particle diameter m
h pipe diameter m
h local heat transfer coefficient W / m²K
k thermal conductivity W / m K
K permeability m²
L pipe length m
m mass flow rate kg/s
P pressure drop Pa
q heat flux W / m²
r radial distance m
T temperature °C
v velocity m/s
z axial distance from inlet m
**Dimensionless Factors**

Da  Darcy number, \( K / D^2 \)
F  inertia coefficients
Nu  Nusselt number based particle diameter, \( \frac{h_d}{k_f} \)
Pr  Prandtl number, \( \frac{v}{\alpha} \)
P  dimensionless pressure drop
\( Re_d \)  particle Reynolds number, \( \frac{v_0 d}{v} \)
\( Re_D \)  Reynolds number, \( \frac{v_0 D}{v} \)

**Subscript**

b  flow bulk bottom surface
bm  mean flow bulk temperature
e  effective
f  fluid
o  based on the total pipe area
sm  mean surface
p  under constant pressure
top  top surface
t  total
v  void
w  wall

**Superscript**

---  average

**Greek letters**

\( \gamma \)  porous media shape parameter, \( \sqrt{\frac{D^2 \varepsilon}{K}} \)
\( \rho \)  density  \( \text{kg/m}^3 \)
\( \varepsilon \)  porosity
\( \nu \)  kinematic viscosity  \( \text{m}^2 / \text{s} \)