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Convection Heat Transfer Analysis in A Vertical Porous Tube

Hikmat N. Abdulkareem[1] | Kifah H. Hilal[2]

Affiliations
[1] Al-Ma'moon university collage, Baghdad Iraq

[2] Technical engineering collage, Baghdad Iraq

Correspondence

Hikmat N. Abdul Kareem,
Al-Ma'moon university collage, Baghdad Iraq
E-mail:
hikmat.najeeb51@gmail.com

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Abstract

Forced convective heat transfer in a vertical channel symmetrically heated with a constant heat flux, and packed with saturated porous media, has been investigated experimentally in the present work. The channel was padded with spherical glass of three diameter (1, 3 and 10 mm) in a range 0.0416 < (particle diameter / inner channel radius) <0.416. The experimental setup, using a copper tube as a packed bed assembly with (48 mm) inside diameter and (1150 mm) heated length with a constant heat flux boundary condition. The test section was vertically oriented with water flowing against gravity and packed with glass spheres (1, 3 and 10 mm) diameter respectively. The results show that local Nusselt number increased at 34% with increasing Reynolds number at 65% while increased at 11% with increasing heat flux at 71%. Heat transfer rate increase as the particle diameter increase at the range of (1 – 3) mm but decrease with increasing particle diameter at the range (3 – 10) mm. Pressure drop through channel minimize at 97% as porosity increase at 23%.Many empirical relations, obtained experimentally.

Keywords: Forced convection, porous media, spherical particle, vertical tube, constant heat flux.

الخلاصة: تم إجراء بحث عملي لانتقال الحرارة بالحمل القسري خلال قناة عمودية مسخنة بفيض حراري ثابت و مملوءة بوسط مسامي مشبع. حشوة القناة المسامية و التي تتكون من كرات زجاجية بأقطار مختلفة (١, ٣, ١٠) ملم تستخدم على التوالي لمديات نسبة القطر (قطر الجزيئة / نصف قطر القناة) تتراوح بين (٠,٠٤١٦ –٠,٤١٦). الجهاز العملي المستخدم يتكون من اسطوانة من النحاس بقطر داخلي (٤٨ ملم) و بطول (١١٥٠ ملم) مسخنة بفيض حراري ثابت, جزء الاختبار يكون عمودي يمر خلاله الماء باتجاه معاكس للجاذبية و مملوء بكرات زجاجية بأقطار مختلفة هي (١, ٣, ١٠) ملم على التوالي. النتائج العملية بينت زيادة رقم نسلت الموضعي (Nux) بنسبة ٣٤% مع زيادة رقم رينولد (Re) بنسبة ٦٥% و الفيض الحراري المستخدم بنسبة ٧١%. أما معدل انتقال الحرارة فيزداد بزيادة قطر الجزيئة لمدى(١ - ٣ ملم) و يقل بزيادة قطر الجزيئة لمدى(٣ – ١٠) ملم . نقصان الفقدان بالضغط بنسبة ٩٧% مع زيادة المسامية بنسبة ٢٣% . تم الحصول على العديد من المعادلات التجريبية.

1. INTRODUCTION

Many industrial operations include an interacting between solid and fluid. To gain a great proportion of surface area to volume, the liquid may be flowed over a packed bed of the solid medium. The packed beds are served satisfactorily for thermal storage in solar air systems because of their relatively fast response and large surface. Two important factors are involved in the design, optimize and operation of the process equipment involving packed bed. These are (i) the resistance of fluid and power required to force fluid through the packed bed. (ii) The rate at which heat transport through the packed bed. Successful modelling requires a good compromise of heat transfer and pressure drop requirement. Size, shape, porosity of bed, liquid thermal properties, velocity of the fluid and the ratio of particle diameter to the channel diameter must be taken into account to derive a generalized equation relating pressure drop and heat transfer coefficient. Hwang *et al* [1] experimented on the convection heat transfer of Freon-113 in a tube padded with glass beads with diameter 3.5 and 6 mm ,also with chrome steel beads (6.35 mm diameter). The channel was installed vertically with Feron-113 passing versus gravity. Two aluminium walls of test unit employing as the heat origin and the heat sink, and two acrylic walls used as constant temperature side surfaces. It was illustrated that heat transferred in a porous tube three times greater than in an clear tube for Re (Reynolds number depend on interval separated among the plates) ranging from 5000 to 1700. Zhao and Song [2] had been examined analytically and experimentally forced convection in a packed bed heated through a porous surface orthogonal to flow orientation. The orthogonally directed test duct, length 31.5 mm, width 99 mm and depth 28 mm, was surrounded by four orthogonal surfaces. A perforating sheet was joined at the under most of the test unit for maintaining the spherical pad in the required position. A finned copper block placed at the upper surface of the porous pad .This structure consisted of glass beads had mean diameter 1.09 mm that padded as a test duct and the heating element was maintained on the upper of the copper piece. The working fluid was deionized water It was shown that increasing at Peclet number lead to a minimize in the surface temperature. Abbilash and Kumer, [3] were investigated experimentally and numerically the influence of metallic pad integrated into a tube that submitted to constant heat tube. Water has flowed through brass tube (72.8 mm) padded with carbon steel balls (6.35 mm) and (0.4 porosity). Experiments were conducted at (310000 W/m^2) heat flux and velocity numerous from (2mm/sec) to (5mm/sec). It was shown that the heat transfer (Nu) increased as (Re number) increasing and porosity decreasing. Also, a porous pad consist of steel balls used by [4]that had been analysis experimentally enhancement in heat transfer for water passing through horizontal pipe. Steel balls (6.35 mm) were used as a porous pad which packed partially (300 mm long) and (85 mm) horizontal pipe. (Maximum heat transfer/ minimum pressure drop) was reached for the porous pad diameter(55 mm) and porosity (0.44) and higher than (4.6) times as compared with empty pipe .The results showed that the Nusselt number increased with decreasing porosity. The pad having (0.44) porosity (4.76 mm) steel balls have efficient heat transfer than the porosity (0.45) and (6.35mm) steel balls. The experiment on convection heat transfer in the porous media had been carried by [5]through plastic tube described with a diameter of (0.11m) and height of (1.66m) The tube have packed with various porous medium with varying sizes bead and porosity (0.47 &0.53). The finer porosity (0.47) had a more identical flow track deal that permitted a greater interaction between solid and fluid and thus a lot to storage more heat than the rough pad. Lee and Chung [6] were conducted the effect of axial and radial location of one heated bead at unheated porous pads and with whole heated beads in porous beds experimentally. Porous beds height varying from 0.02m to 0.26m and particle Reynolds number was varying 31-389. At one heated sphere, the heat transfer rising at the bead near the side surface or the base of a porous pad. For all heated sphere, the heat transfer decreased with increasing the proportion of height bed to bead diameter. In the present work, an experimental analysis will be carried out to investigate convection heat transport of water flow through vertical porous channel heated a symmetrically with constant heat flux with varying porosity.

2. EXPERIMENTAL SETUP

In this work an experimental setup for forced convection heat transfer and pressure drop measurements in vertical the cylindrical packed channel has been designed and fabricated.

A schematic diagram of the experimental equipment used in this work was shown diagrammatically in Figure 1 and photographically in Figure 2. The test section is vertically oriented with water flowing against gravity. A copper tube is used for the packed bed assembly. It^s dimensions are (48 mm) inside diameter, (54 mm) outer diameter and a length of (1150 mm) , a schematic of the test tube is illustrated in Figure 3 and filled with glass beads (1,3 and 10 mm) diameter respectively. A copper tube, which is considered as a test section heated uniformly by using an electrical heater wound along the tube to get constant heat flux condition. The pressure gradient up the glass bead is measured by ten static pressure taps placed at (100 mm) intervals along the side of the cylinder.

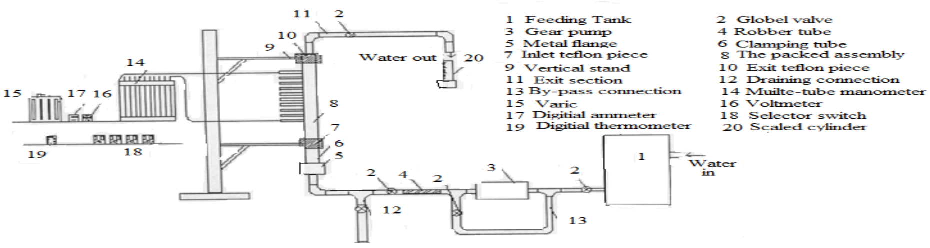

Figure 1 Schematic diagram of experimental apparatus

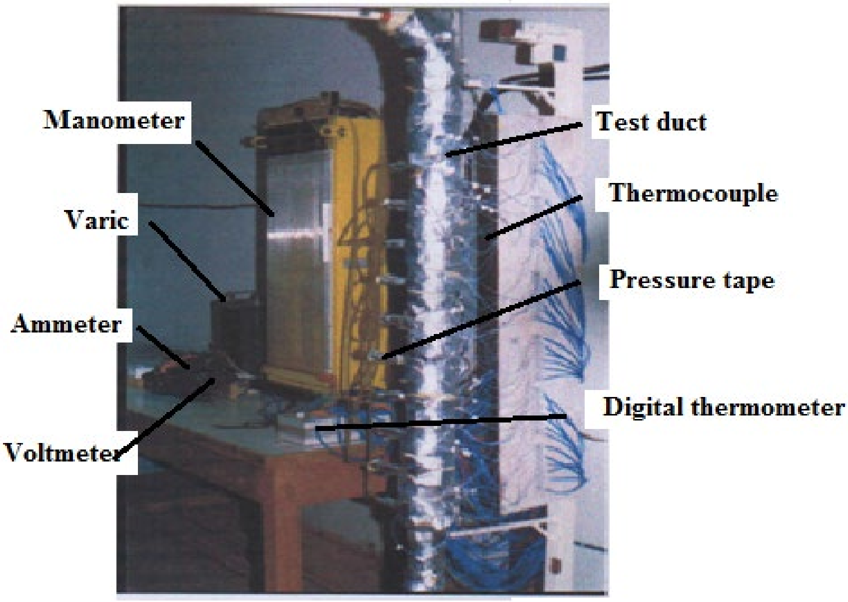

Figure 2 photographic of assembly

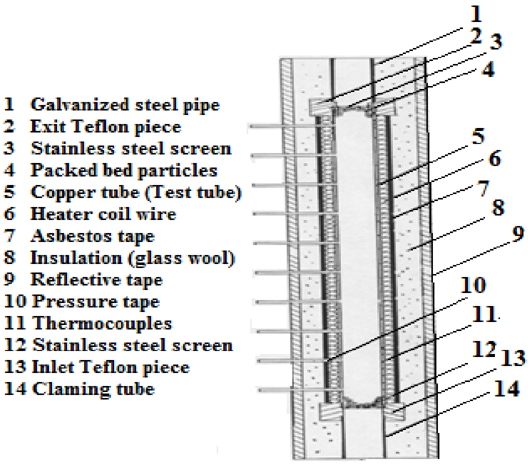

Figure 3 The packed bed assembly

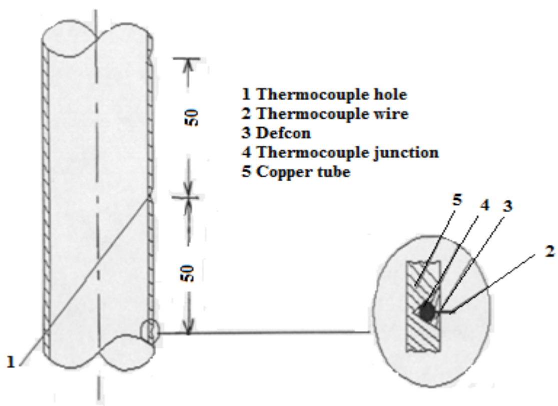

Figure 4 Thermocouple construction

The cylinder surface temperature is measured by (66) copper constantan Type (T) thermocouples, fixed a longitudinal and circumferential the cylinder. The inlet bulk water temperature was measured by three thermocouples placed in the entrance section, also three thermocouples were measured the outlet bulk air temperature located at the exit section. The thermocouples are fixed by drilling holes; 2 mm depth in the surface of copper tube and the measuring junctions are secured permanently in the holes by high temperature application defcon adhesive as in Figure 4. To reduce heat losses from the test section to the surrounding, Teflon connection pieces are represented the test section entrance and exit to minimize the ends losses, also the outside of the test section is thermally insulated. The clamping arms of the vertical stand are fixed on test tube at the Teflon region, to adjust it at a vertical position.

Significant care was possessed in filling the glass balls to assure regularity in the arrangement of the porous media The glass beads were decanted at random way into the test section, levelled and then vibrated to reduce the possibility of any “bridging”. After permitting the glass beads to stabilize, more of the porous media was sited in the tube and padded. These steps were iterated until no additional sphere could be objected into the tube. To fix the glass beads in tube, two stainless steel meshes 54 mm diameter are located at the entrance and exit of the porous tube. Measuring instruments were used in the test rig as shown in Figure 1. The voltage regulator has received voltage between (154-225 Volt) and supplied stable alternating voltage (220 volt). A varic was used to adjust the heater input power as required. Voltmeter and digital ammeter were applied to measure the heater voltage and current respectively. The thermocouples circuit would be connected with digital electronic thermometer to gauge the temperature value. The pressure drop through a packed bed had been depended mainly on the porosity and particle diameter. Two types of pressure gauge were used, a multi- tube manometer was used to measure the pressure drop experimentally with in the porous medium channel consisted of glass bead (3 mm) and (10 mm). The fluid used in the manometer is Carbon Tetrachloride (CCL_4) with (1590 kg/m^3) density and maximum reading of the manometer was approximately (11.854 kPa) and the accuracy is (0.13 %) full scale. Four bourdon gauges are linked to pressure taping to find the pressure drop through the test tube consisted of glass bead (1mm). In the test tube, there are ten pressure taps, only four of them are used at the positions (130,430,730,1030 mm) from the beginning of the test section, the others tap are closed during the experiments which use (1mm) diameter glass bead as a packed bed.

3. PACKED BED PROPERTIES

The properties of the porous medium will be found to use in the experimental calculation. The average glass sphere diameter was determined by randomly choosing fifteen glass spheres for each packed (1mm, 3mm, 10mm). Measured the diameter of this bead is made by fixing the bead through the clamping arms of the micrometres and rotated the bead to checked the spherical shape of it, then measure the bead’s diameter. The measurements illustrated that the average glasses sphere diameter are (1.01mm, 3.01mm and 10mm) for the three kinds of packed bed used in this work. The bead diameters used in the present work are (1mm, 3 mm and 10 mm) and made of different glass composition. First weighting the sample and then finding its volume by a water displacement method obtain the average density of the sample containing each particle diameter. Distilled water at 20 ºC is added to the cylinder until the water surface touches a certain level. The volume of the water added is measured. Glass spheres are added to the cylinder and there volumes are equalled water volume displaced. The average density is obtained from dividing the mass of the sample to the volume of the water displaced. This measuring is iterated many times until no variation detected. The average densities of the glass spheres (1mm, 3 mm and 10 mm) are found to be (2500,2700 and 2800 kg/m^3) respectively. A significant property for porous media is the medium thermal conductivity (K_m). In prior research [7] a standard mix ruling depend on the volume portion has generally been used:

$$K_m = \varepsilon K_f + (1-\varepsilon) K_s \tag{1}$$

Where (K_f) is water thermal conductivity and (K_s) glass spheres thermal conductivity. Values obtain from Eq. (1) were accepted only if $K_f \approx K_s$, the larger difference between fluid and the solid thermal conductivities, the more the measured values of (K_m) vary from the predictions made by Eq. (1).

The thermal properties of glass spheres are taken from [8],[9]&[10], by comparing the average density of glass spheres found with the density of various kind of glass given in [8],[9]&[10] to determine the kind of glass that the spheres made from it. The average porosity of the porous media (packed bed) which calculate for each glass beads diameter from the defining equation:

$$\varepsilon = \frac{V_t - V_p}{V_t} \tag{2}$$

Where Vp is the ratio of the measured weight of the spherical glass added to the bed to the density of packed bed and (Vt) is the copper tube (test section) volume measured by knowing the height and the inner diameter of the tube. For the glass beads of (1, 3 and 10) mm diameter the average porosity is obtained as (0.356, 0.382 and 0.4621) respectively. These values are obtained by repeating the measurements three times for each particle diameter to have an accurate value of average porosity. The data obtained is correlated by the following equation:

$$\varepsilon = 0.3454 + 11.6985\,(Dp) \tag{3}$$

4. EXPERIMENTAL DATA ANALYSIS

The average heat flux supplied to packed bed is gained by measuring the average increase in bulk temperature of the water across the porous duct. The average increase at bulk temperature obtain by measured the bulk temperature at the inlet and outlet of the packed bed as the average for three-thermocouple reading [11].

$$Q = m^{\cdot}\,(Cp_o\, t_o - Cp_i\, t_i) \tag{4}$$

The total input power provided to test duct can be computed:

$$Q_t = V * I \tag{5}$$

The heat balance between heat flux supplied and heat flux calculated from Eqs. (4&5) illustrated that the difference between them does not exceed (5%), this was due to good insulation and high thermal conductivity for copper.

The wall heat flux can be calculated by:

$$q_w = \frac{Q}{A_s} \tag{6}$$

$$A_s = 2\ \pi\, r_i\, L \tag{7}$$

The local heat transfer coefficient may be obtained by:

$$h_x = \frac{q_w}{(\Delta t)_x} \tag{8}$$

Where $(\Delta t)_x$ is the temperature difference between local wall temperature $(t_s)_x$ and local bulk water temperature $(t_b)_x$ at length (x) from the test tube entrance.

The local bulk water temperature $(t_b)_x$ at each measuring level was obtained by assumed a linear water temperature variation along the flow channel.

$$(t_b)_x = t_i + \frac{x}{L}\,(t_o - t_i) \tag{9}$$

Then the local heat transfer coefficient is gained by: -

$$h_x = \frac{q_w}{(t_s)_x - (t_b)_x} \tag{10}$$

The local Nusselt number Nu_x is calculated by: -

$$Nu_x = \frac{h_x * di}{K_m} \tag{11}$$

The stagnant thermal conductivity of water –porous bed for sphere (K_m) was calculated from Eq. (11).

All the porosities are found at the mean film water temperature:

$$(t_m)_x = \frac{(t_s)_x + (t_b)_x}{2} \tag{12}$$

The average values of Nusselt number (Nu) can be calculated as:

$$Nu = \frac{1}{L}\int_{x=0}^{x=L} Nu_x\ dx \tag{13}$$

The water velocity at inlet of the test tube (U_i) can be calculated by:

$$U_i = \frac{4\, \dot{m}}{\pi\, di^2\, \rho} \tag{14}$$

And the porous medium effective thermal diffusivity (α_e) was

$$\alpha_e = \frac{K_m}{\rho\ Cp} \tag{15}$$

Reynolds number was realized as the proportion of inertia forces to viscous force can be found as:

1. Water velocity at entrance tube and inner diameter of test tube

$$Re = \frac{U_i\ di}{\upsilon} \tag{16}$$

2. Water velocity in inlet porous tube and glass bead diameter.

$$Re_d = \frac{U_i\ Dp}{\upsilon} \tag{17}$$

3. Water velocity through porous media and glass sphere diameter.

$$Re_d{}^* = \frac{Re_d}{(1-\varepsilon)} \tag{18}$$

5. EXPERIMENTAL RESULTS

The experiments were conducted to cover a range of Reynolds number from (473) to (1760), heat flux varied from (2.1 kW/ m²) to (7.4 kW/ m²) and porosity between (0.356 – 0.4621).

5.1.Temperature distribution

The direct reading of temperature along the axial direction is plotted in Figure 5 to Figure 7 , in which the temperature profiles for different types of packing, Reynolds number and heat flux. It is shown that, increasing local wall temperature as the axial position increased, decreasing in Reynolds number, or increasing in the heat flux. The outlet bulk water temperature increased with increases in the amount of heat flux while decreased with increases in the bead Reynolds number. The thermal conductivity of three kind of spherical glass bead used in the experiments are (1.4, 0.78, 0.81 W/ m. K) for (Dp = 1, 3, 10) mm respectively.

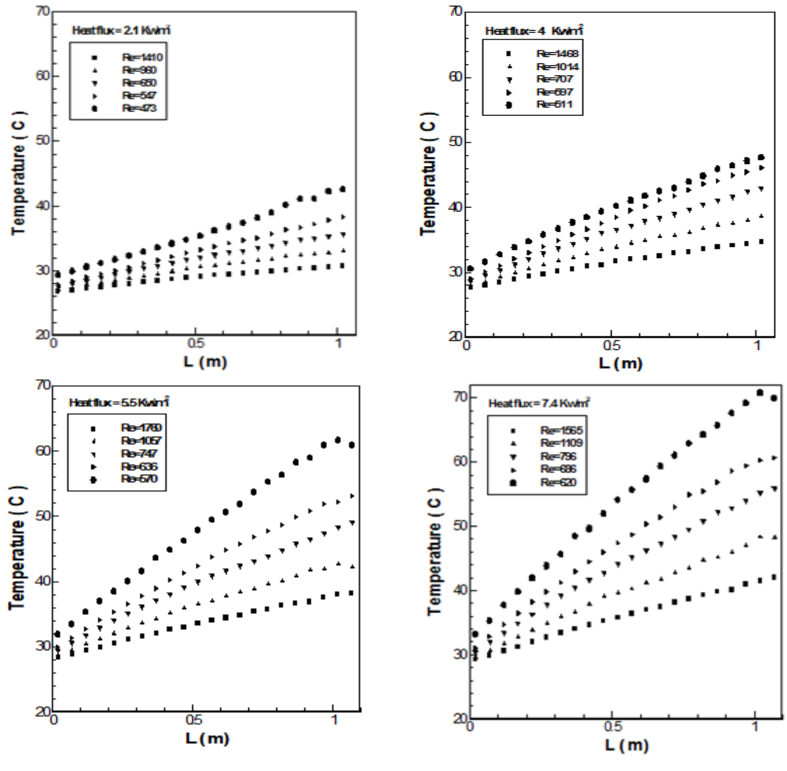

Figure 5 Channel wall temperature distributions with channel length for Dp=1 mm

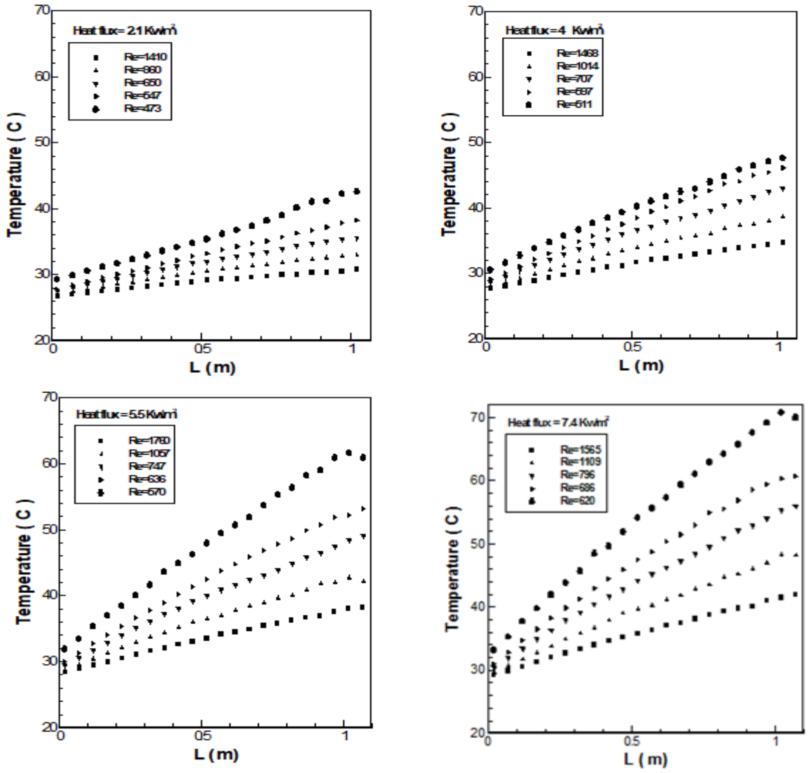

Figure 6 Channel wall temperature distributions with channel length for Dp=3 mm

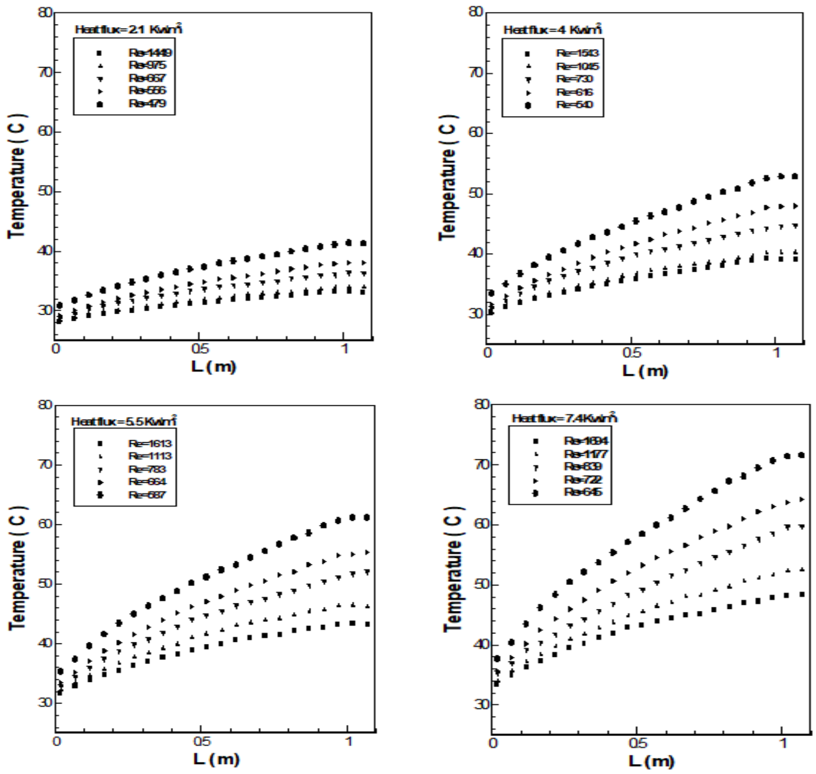

Figure 7 Channel wall temperature distributions with channel length for Dp=10 mm

For that the effect of types of packing and bead diameter isn't clear appeared in Figure 8 at (Re = 473 – 479). It is seen that the highest temperature is at (Dp = 10, 3, 1 mm) respectively. Actually, if the thermal conductivity is equal for all of them the highest temperature will be occurred at (Dp = 10, 1, 3 mm) respectively.

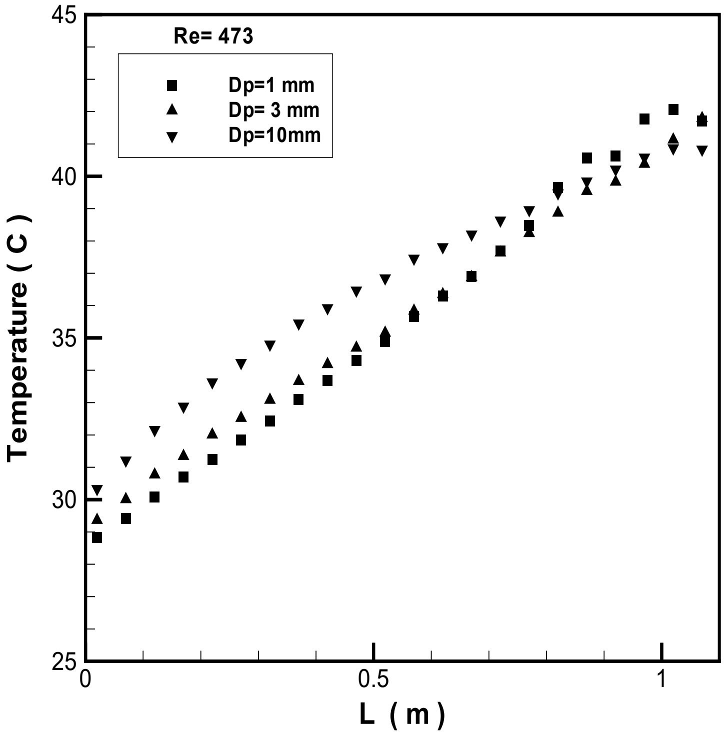

Figure 8 The influence of particle diameter and axial position on the temperature distributions at Re=473

5.2. Local Nusselt number

The difference of local Nusselt number (Nu_x) with channel length is illustrated in Figures 9 to 11. It is observed that the local Nusselt number decrease as increasing in the axial position due to the difference between the bulk water and wall temperature and increasing thickness of the thermal boundary layer. At a given heat flux and particle diameter, the local Nusselt number increase with an increase in the Reynolds number where the thermal entrance length is shorter for the small Reynolds number than for the layer one. For above results, local Nusselt number increased at 34% with increasing Reynolds number at 65% while increased at 11% with increasing heat flux at 71% .

To illustrate the effect of packing type on the (Nu_x), Figure 12 is plotted. It can be seen that (Dp = 3 mm) gives the highest amount of Nusselt number. That is because of the large surface of contact area and lowest values of porosity which is (0.382) compared to (0.4621) for the (Dp = 10 mm). For porosity between (0.356 – 0.382), the local Nusselt number decrease with decrease the porosity and particle diameter, although the contact area is increased. The small particle diameter causes narrow passage through the pore, low velocity and the channelling effect happened at a small distance closer to the wall, all these parameter decrease the heat transfer at particle diameter between (1, 3) mm.

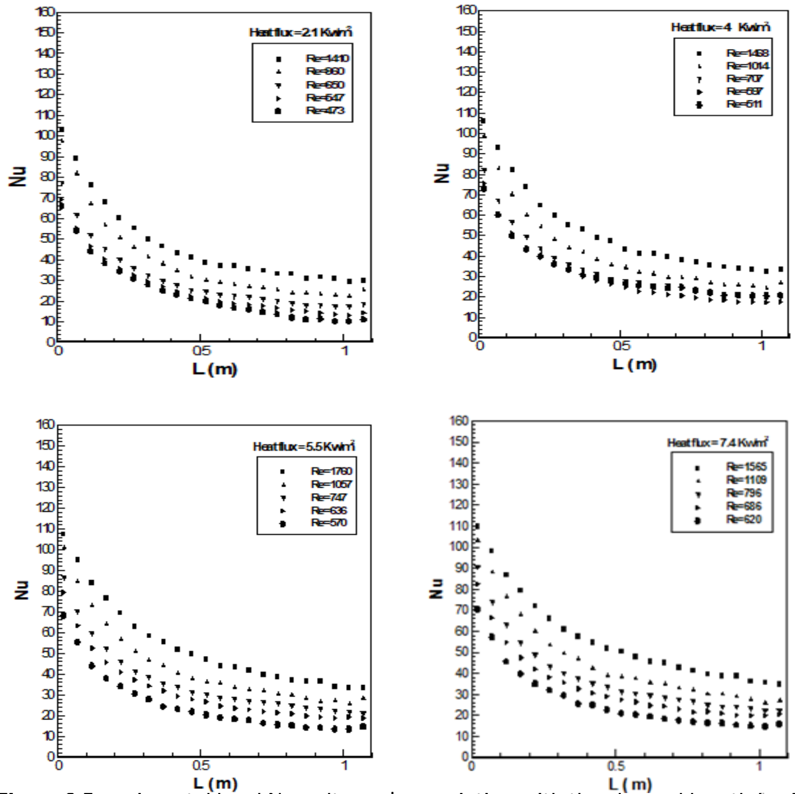

Figure 9 Experimental local Nusselt number variation with the channel length for Dp=1 mm

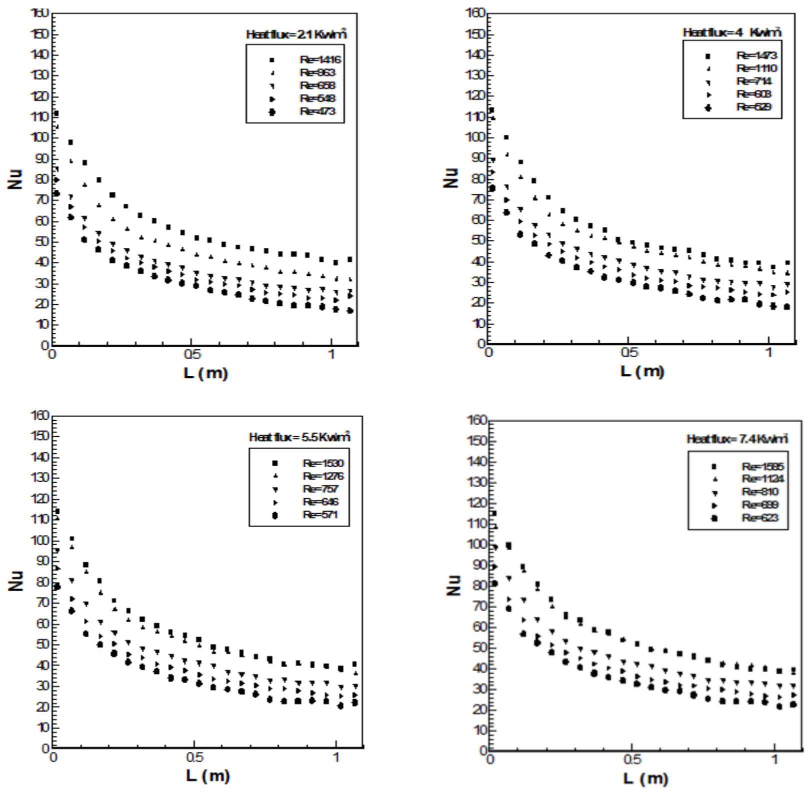

Figure 10 Experimental local Nusselt number variation with the channel length for Dp=3 mm

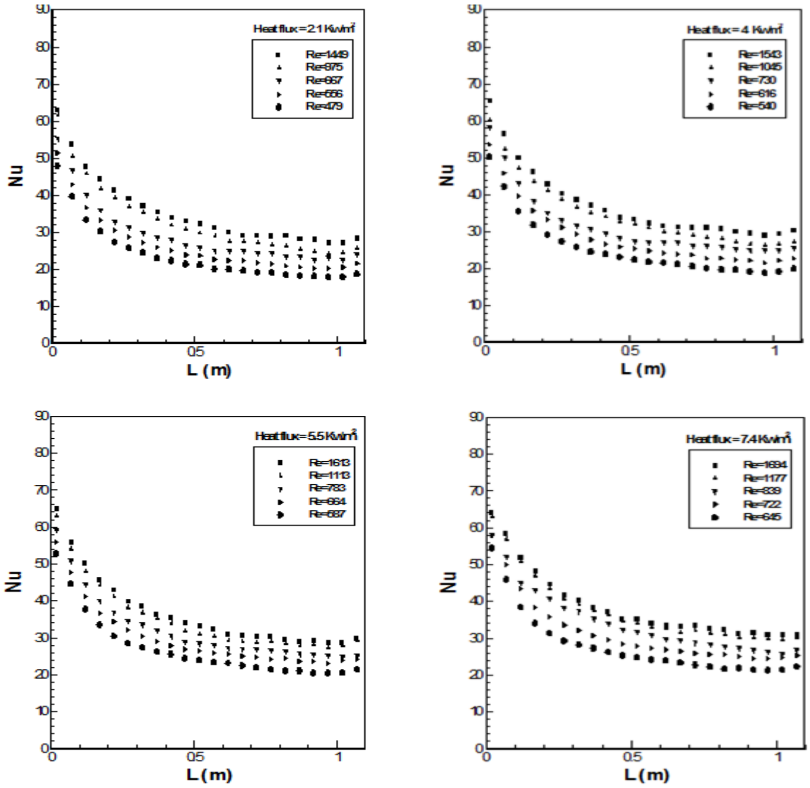

Figure 11 Experimental local Nusselt number variation with the channel length for Dp=10 mm

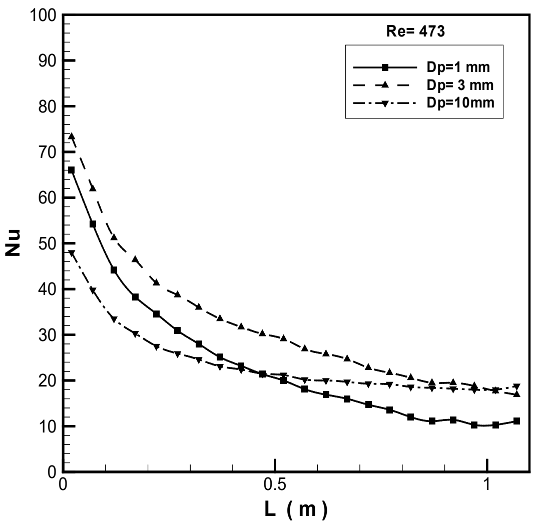

Figure 12 The influence of particle diameter and axial position at experimental Nusselt number at Re=473

5.3. Pressure drop

Figure 13 shows the pressure drop functional to Reynolds number to (1, 3 and 10 mm) particle size and (0.356, 0.382 and 0.4621) porosity. It is seen that the pressure drop minimize at 97% as porosity increase at 23% due to the resistance in a packed bed was the sum of the viscous and turbulent forces

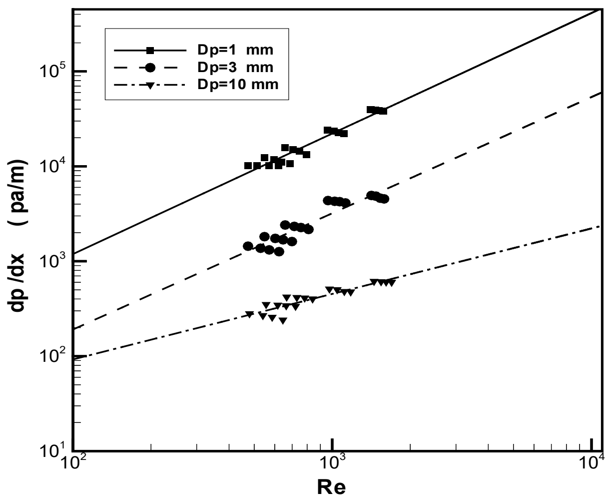

Figure 13 Experimental pressure drop versus Reynolds number for various particle diameters

5.4. Correlation of heat transfer relations

Figures 14 & 15 show the correlation of average Nusselt number (Nu) and average Nusselt number depending on particle diameter with Reynolds number (Re)) by the equations of the form:

$$Nu = n1 \ Re^{m1} \tag{19}$$

$$Nu = n2 \, {Re_d}^{m2} \tag{20}$$

Where n1,m1,n2 and m2 are constant illustrated at Table 1. Figure shown that as Reynolds number increased Nusselt number increased for all particle size used in experiments

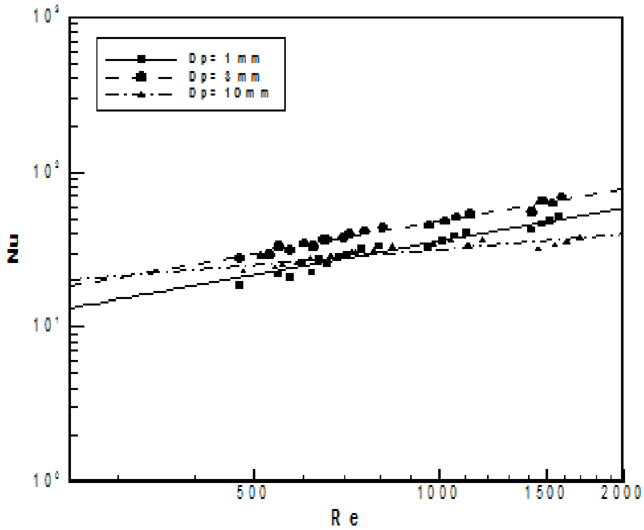

Figure 14 Experimental average Nusselt number versus Reynolds number for various particle diameters

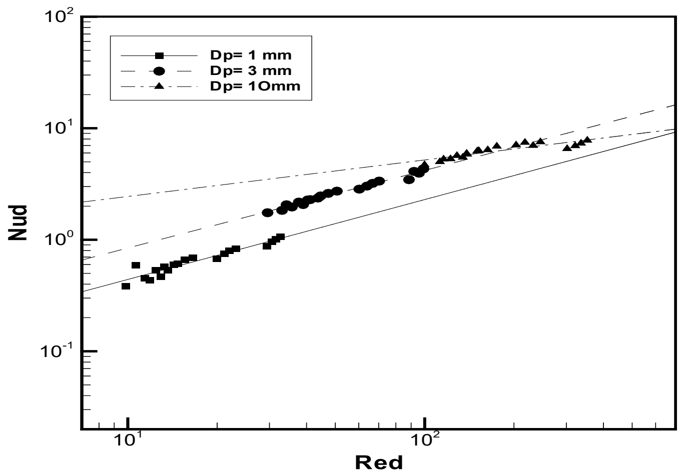

Figure 15 Nusselt number based on particle diameter with the Reynolds number based on particle The variation of the experimental average diameter

Table 1 Empirical constant for Eqs. 19 & 20

	Dp = 1 mm	Dp = 3 mm	Dp = 10 mm
n1	0.2544	0.3947	3.3034
m1	0.7160	0.6958	0.3274
n2	0.0847	0.1698	1.1501
m2	0.7160	0.6957	0.3273
Re Range	472 - 1569	473 - 1585	479 – 1694

5.5. Comparison with previous work

In the present work, Nusselt number was increased as particle diameter decreased between (10 –3 mm), the same results obtained by [12] for particle diameter (7.2 and 16.9 mm in diameter) is present in Figure 16.

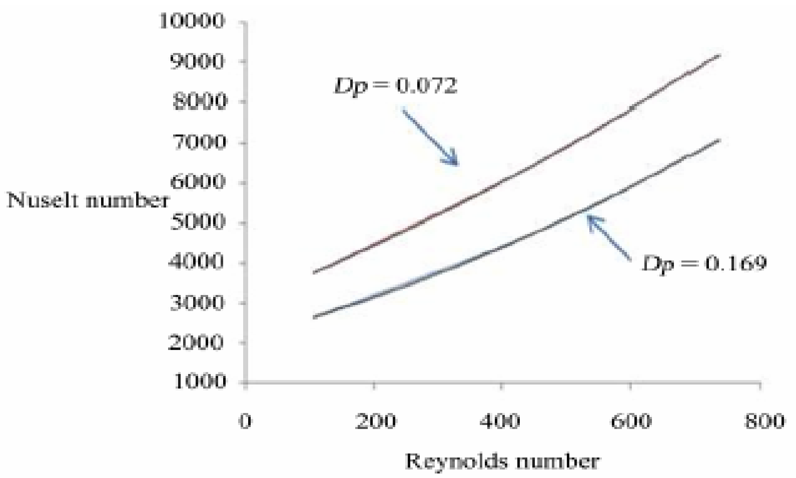

Figure 16 Average Nusselt number in packed channel [12]

6. CONCLUSION

In this work, the heat transfer characteristics of forced convection flow in a vertical porous channel subjected to constant heat flux boundary condition have performed experimentally. The local wall temperature decreases as Reynolds number increase and the heat flux decrease. The local Nusselt number increasing as the Reynolds number and heat flux increase. The average Nusselt number increase as Reynolds number increase for all particle size used in experiments. The pressure drop through porous tube minimize at 97% as porosity increase at 23%.

NOMENCLATURE

A_s	Surface area m^2	Q	Power absorbed by water W
Cp	Water specific heat at constant pressure kJ/ kg. K	Q_T	Total input power W
Dp	Particle diameter m	ri	Inner channel radius m
di	Inner channel diameter m	Re	Reynolds number
h_x	local heat transfer coefficient W/m^2. °C	Re_d	Particle Reynolds number
I	Electrical current Amp.	t_{sx}	Local surface temperature °C
K_f	Water thermal conductivity W/m. °C	t_{bx}	Local bulk water temperature °C
K_s	Glass sphere thermal conductivity W/m. °C	Δt_x	Local mean film temperature °C
K_m	Packed bed thermal conductivity W/m. °C	U_i	Water velocity at channel entrance section m/s
L	Channel length m	V_T	Test duct volume m^3
m	Water mass flow rate m	V_p	Packing volume m^3
Nu_x	Local Nusselt number	V	Voltage Volt
Nu	Average Nusselt number	ρ	Air density kg/m^3
Pr	Prandtl number	ε	Porosity
q_w	Wall heat flux W/m^2	υ	Kinematics viscosity $^2/s$

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