Thermal Characteristics Of Ceramic Packed Bed

Raghad H. Hilal*

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Abstract

Convection heat transfer in horizontal channel filled with saturated packed bed has been studied numerically using finite difference technique. The channel wall is heated at constant heat flux and packed with a fluid – saturated spherical ceramic for diameter ratio (0.287). Air , helium and carbon dioxide are used as working fluid at Reynold number ranging (100 – 2500). The results show a significant effect of varying prandtle number on heat transfer rate and friction factor at different Reynold number. The radial temperature profile increase as the prandtle number decrease. The heat transfer rate increase as the prandtle number increase , Carbon dioxide is greater than for air and the last is greater than that for helium at the same flow conditions . Friction factor proportional inversely with prandtl number. New Correlations are obtained in this work:

$$\mathbf{Nu_{av} = 24.548 \quad Re^{0.238} \quad Pr^{0.0787}}$$

$$F = 39.1917 \left(\frac{1-\phi}{\mathrm{Re}_d} \right)^{0.901}$$

Key word: Ceramic Packed Bed , heat transfer .

:الخلاصـــة

يتضمن البحث دراسة نظرية لانتقال الحرارة خلال قناة افقية مملوءة بوسـط مسـامي مشبع ، استخدمت طريق الفروقات المحددة لحل معادلات الطاقة والزخم عدديا . تم تسخين جدار القناة بفيض حراري ثابت ويتكون الوسط المسامي المشبع من كرات سـيراميكية ونسـبة قطـر مقدارها (0.287) . استخدم ثاني اوكسـيد الكاربـون ، الهـواء والهليـوم كمائـع ناقـل للحـرارة لمديـات رقـم رينولدز (Re)(100-2500). تبين النتائج تأثير ملحوظ لرقم برانتل (Pr) على مقدار الحرارة المنتقلة وكذلك معامل الاحتكاك عند ارقام رينولدز المختلفـة . تزداد درجة الحرارة القطرية مع نقصان رقم برانتل اما مقدار كمية الحرارة المنتقلة فتزداد مـع زيادة رقم برانتل فتكون عند استخدام ثاني اوكسيد الكاربون اكبر من الهواء والهواء اكبـر مـن الهليوم عند نفس ظروف التشغيل . يتناسب معامل الاحتكاك عكسيا مع زيادة رقم برانتـل . تـم الحصول على العلاقات التالية من النتائج المستحصلة من التحليل النظري :

$$\mathbf{Nu_{av} = 25.548 \quad Re^{0.238} \quad Pr^{0.0787}}$$

$$\mathbf{F = 39.1917 \left(\frac{1-\phi}{\mathrm{Re}_d} \right)^{0.901}}$$

Nomenclature

B dimensionless pressure drop -
Di inner channel diameter m
dc ceramic particl diameter m
D diameter ratio (dc/ri) -
h convection heat transfer coefficient $w/m^2.k^o$
q_w constant heat flux w/m^2
F friction factor -
K permeability m^2
Ke effective thermal conductivity of porous media $w/m.k^o$
Nu Nusselt number (hDi/ke) -
P Pressure N/m^2
Pr Prandtle number (α_e / γ) -
ri inner radius of channel m
r radial coordinate -
Re Reynold number (ρuDi/μ) -
Re_d particle Reynold number(ρu dc/μ) -
R dimensionless radius -
ΔR Size of the i th mesh interval in r-direction -
T Temperature C^o
Ti Inlet temperature C^o
T\` dimensionless temperature -
u horizontal fluid velocity m/s
uc Constant velocity (slug flow) m/s
u\` dimensionless fluid velocity -
x horizontal coordinate m
X dimensionless horizontal length -
Δx Size of the j th mesh interval in x-direction

Greck

ρ Fluid density kg/m^3
μ Fluid dynamic viscosity kg/m.s
γ Fluid kinernatic viscosity m^2/s
α_e effective thermal diffusivity of porous medium m^2/s
ϕ Porosity

Introduction

Porous media have been employed widely in thermal energy storage system, food processing , geothermal system, chemical reactors engineering , building insulation and infiltration[1,2]. The structure and porosity of the porous media affected the flow patterns and thermal transport phenomena in porous channel. For that various studies investiged convection heat transfer in a channel packed with spherical particle focused on particle diameter and thermal properties. Vafai *etal* [3] performed an experiment on the forced convection of water in a channel filled with glass spheres

(5,8mm in diameter), the average Nusselt number were measured at selected Reynolds number and increased linearly with it. Renken and poulikatos [4] carried out a similar experiment with (3mm) glass spheres and reported the local Nusselt number.

Hwang *etal* [5] used another prandtl number fluid at similer experiment, forced convection of Freon - 113 (Pr : 8.06) in a channel packed with small glass spheres (3,5 and 6mm in diameter) and with chrome steel spheres (6.35mm in diameter). It was found that the Nusselt number increased as the particle diameter was decreased.

Most of heat transfer investigation in packed beds are presented in terms of correlations based on Reynolds number[4], effective thermal conductivity [6] and particle size [7]. The effect of working fluid prandtle number on the pressure drop and heat transfer coefficient have not taken into account in this correlation.

The purpose of the present paper is to investigate numerically forced convection heat transfer in a packed bed of ceramic spherical particles. Air, heluim and carbon dioxide are used as working fluids at constant diameter ratio

2. Physical Problem Formulation

Consider the basic problem of a cylindrical horizontal channel of inner diameter (Di) heated symmetrically with a constant heat flux, packed with a fluid – saturated spherical ceramic of diameter (dc). The physical model being studied is shown in Fig(1). Various kinds of thermal properties fluids at the same temperature (Ti) are used as working fluids like air, heluim and Carbon dioxide.

The ratio of diameter of the channel to the packing diameter should be a minimum of 8:1 to 10:1 for wall effects to be small [8] according the ratio (Di/dc) will be taken 7:1 and the core channel porosity is 0.399 in this study.

The governing equations (momentum and energy) are made on the following assumptions :

1- The fluid and porous medium are in

2- Fluid and ceramic physical properties are constant .

3- Intraparticle conduction is neglected .

The momentum equation based on Darcy flow model which is popular in porous media convective heat transfer investigations because of its simplicity and good performance [4,6] It is essential to know the permeability (K) of the bed in order to relate the fluid flow rate to the pressure gradient with Darcy law :

$$-\frac{dp}{dx} = \frac{\mu}{k} u \qquad \text{------ (1)}$$

For a packed – sphere bed, the permeability of the bed is related to the porosity by [5] :

$$K = \frac{\phi^3 dc^2}{150(1-\phi^2)} \quad \ldots(2)$$

For thermal analysis the following energy equation govering the convection phenomena in the channel [9] :

$$u\frac{\partial T}{\partial x} = \alpha_e . \frac{1}{r} . \frac{\partial}{\partial r}(r\frac{\partial T}{\partial r}) \quad .(3)$$

The boundary conditions for the velocity are

$$r = 0 \qquad u(r) = u_c \qquad \text{------ (4)}$$

r = ri u(r) = u_c

and Eq.(3) is submitted into the following boundary conditions:

$$r = ri \qquad q_w = -ke\,\frac{\partial T}{\partial r}$$

$$r = 0 \qquad \frac{\partial T}{\partial r} = 0 \qquad \text{------ (5)}$$

$$x = 0 \qquad T = Ti$$

Numerical Solution

A solution for the Darcian flow depends on the following non dimensional parameters:

$$u` = \frac{u}{\gamma / r_1}$$

$$R = \frac{r}{ri}$$

$$X = \frac{x}{ri \quad pr} \qquad \text{------ (6)}$$

$$T` = (T-Ti)\,\frac{ke}{q_w\, ri}$$

$$B = -\,\frac{dp}{dx}.\frac{ri^3}{\rho\gamma^2}$$

$$D = \frac{dc}{ri}$$

The dimensional variables represented in Eqs. (1,3) are replaced by new dimensionless variables in Eq.(6) and the following dimension – less governing equations are obtained.

$$u` = C.B \qquad (1\text{-a})$$

where $C = \dfrac{D^2\phi^3}{150(1-\phi)^2}$ ------ (7)

$$u`\frac{\partial T`}{\partial X} = [\frac{\partial^2 T`}{\partial R^2} + \frac{1}{R}.\frac{\partial T`}{\partial R}] \qquad \text{---- (3-a)}$$

There are various numerical techniques to discretize and solve the equations governing the present problem, among which the finite difference approach is the most straight forward method[10] .

In the finite difference approach the flow domain is discretized so that the dependent variables are considered only at discrete points. The typical two- dimensional grid system is shown in Fig(2) , where (i , J) is the grid point in radial and axial coordinates.

The following central finite difference formules are used :

$$\frac{\partial T`}{\partial R} = \frac{T`_{i+1,J} - T`_{i-1,J}}{2\Delta R}$$

$$\frac{\partial^2 T`}{\partial R^2} = \frac{T`_{i-1,J} - 2T`_{i,J} + T`_{i+1,J}}{\Delta R^2} \qquad \text{---- (8)}$$

$$\frac{\partial T`}{\partial X} = \frac{T`_{i,J+1} - T`_{i,J-1}}{2\Delta X}$$

Eqs. (1-a , 3 , a) are formulated in central difference form, the linearzation equation together with the boundary condition and transformed into an equivalent tridiagonal set of algebraic equations.

Knowing the velocity distribustion from numerical solution of momentum energy then applied at the analysis of the energy equation . The temperature profiles and the local Nusselt number at any cross section can be calculated. The average values of Nusselt number (Nu_x) can be found:

$$Nu_{av} = \frac{1}{L}\int_{x=0}^{x=L} Nu_x\, dx \qquad \text{------ (9)}$$

Also pressure drop across the ceramic packed bed are obtained from numerical solution and presented in terms of friction factor (F) [11].

$$F = \frac{\Delta P}{L} . \frac{dc}{\rho u^2} . \frac{\varepsilon^2}{(1-\varepsilon)} \quad \text{------ (10)}$$

4. **Results and Discussion**

Results of the 2-D model ceramic packed bed analysis taken into account the effect of prandtl number on heat transfer rate and friction factor are presented for Reynold number ranging (100 – 2500) and diameter ratio (0.287). The gases considered in this work are CO_2 , air and helium. These gases have different heat transfer properties (specfic heat, thermal conductivity and prandtl number) as shown in table (1) [12].

This difference in properties has a significant effect on the radial temperature distribution as shown in Fig (3). The dimensionless temperature profile across the channel radius at volumetric flow rate (0.5 lit/sec)at different position of X are illustrated . It can be seen that the temperature in the packed bed progressively decreases away from the heated channel wall. The temperature profile increase as the prandtle number decrease for that helium is greater than air and carbon dioxide.

The local Nusselt number along the flow direction for different volumetric flow rate are presented in Figs (4-6) . It begin with a high value at the inlet of the channel and decreases greatly with an increase in X. As the volumetric flow rate in -terms of Reynold number increase the local Nusselt number profile will be increased.

Fig. (7) shows the effect of prandtl number on the heat transfer rate at the same inlet and boundary condition . It is seen that the local Nusselt number profile increase as the prandtle number increase, CO_2 is greater than that for air and the last is greater than that for helium at the same flow conditions.

Fig. (8) shows the correlation of average Nusselt number with Reynolds number. Increasing (Re) yields increasing (Nu_{av}) , but incremente at CO_2 is greater than that for air and helium. This variables are correlated by :

$$Nu_{av} = 24.548\ Re^{0.238}\ Pr^{0.0787} \quad \text{---(11)}$$

The shap of curves in Fig(8) is similer to those presented in ref(5) and shown in Fig(9). It is seen that average Nusselt number increase as Reynolds number increase at the different particle diameter.

From the numerical solution (ΔP/L) is obtained the value of friction factor (F) is evaluation by using Eq.(10). Fig (9) shows the value of (F) versus [Red / (1-ϕ)] where Red is defined as particle Reynolds number based on ceramic particle diameter. It is noted that (F) proportional inversely with prandtl number. The following equation is obtained to prdict the friction factor across the ceramic packed bed .

$$F = 39.1917 \left(\frac{1-\phi}{Re_d} \right)^{0.901} \quad \text{---- (12)}$$

Table (1) Physical Properties of the gases at 27C^{O}[12]

Properties	CO_2	Air	Helium
Cp (J / kg.C^o)	852	1005	5197
K (w / m.C^o)	0.0166	0.026	0.153
Pr No.	0.768	0.712	0.69

Reference

1. Sheridan J . , Williams A . and Close D. J. " An experimental study of natural convection with coupled heat and mass transfer in porous media ", Int. J. Heat mass transfer , Vol. 35 , PP. 2131-2141, 1992.
2. Zhao T. S. and Song y. J. " Forced convection in a porous medium heated by a permeable wall perpendicular to flow direction analyses and measurements " , Int. J. Heat mass transfer , Vol. 44 , PP. 1031- 1037, 2001.
3. Vafai k., R.L Alkire and Tien C. L. " An experimental investigation of heat transfer in variable porosity Media" , Transactions of the ASME, Vol. 107 , PP. 642-647, 1985.
4. Renken K. J. and D. " Experiment and analysis of forced convective heat transport in a paked bed of spheres , Int. J. Heat mass transfer , Vol. 31 , PP. 1399-1408 , 1988.
5. Hwang T.H., Cai Y. and.Cheng P " An experimental study of forced convection in a packed channel with asymmetric heating", Int. J. Heat mass transfer,Vol.35, pp.3029-3038,1992 .
6. Boomsma K. and Poulilcakes D." on the effective thermal conductivity of a three – dimensionally structured fluid saturated metal foam" , Int. J. Heat mass transfer , Vol. 44 , PP. 827-836 , 2001.
7. Nasr, K. S. and Ramadhyani Viskanta R. " An experimental investigation on forced convection heat transfer from a cylinder embedded in a packed bed ", J. Heat transfer , Vol. 116 , PP. 73 – 80 , 1994.
8. Ani F. AL -, " Temperature distribution in paked beds using Numerical technique ", A thesis submitted to college of Engineering , Baghdad , 1998.
9. Poulikoks D. Renken K. " forced convection in a channel filled with porous medium , including the effects of flow inertia, variable porosity and brink man friction", transactions of the ASME , Vol. 109, PP.880-888 , 1987.
10. . Zhene W, " Convection of water Near $4C^o$ in An isotropic porous Medium " , M. Sc. Thesis , Universite` de Montere ` al , 2000 .
11. Bird R. B. , Stewart W. and light foot " Transport phenomena " , John wiley and sons Inc 1st Edition 1960.
12. Yunus A.C , " Heat Transfer, A practical Approach " , WCB / MCGRAW HILL , 1998.

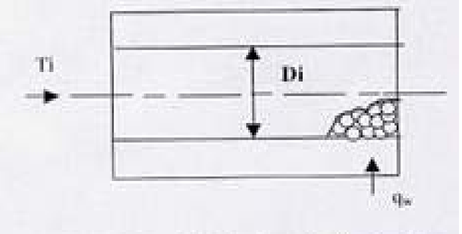


Fig (1) Schematic of horizontal ceramic packed bed

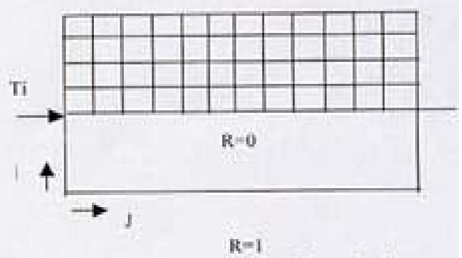


Fig (2) Schematic of flow domine

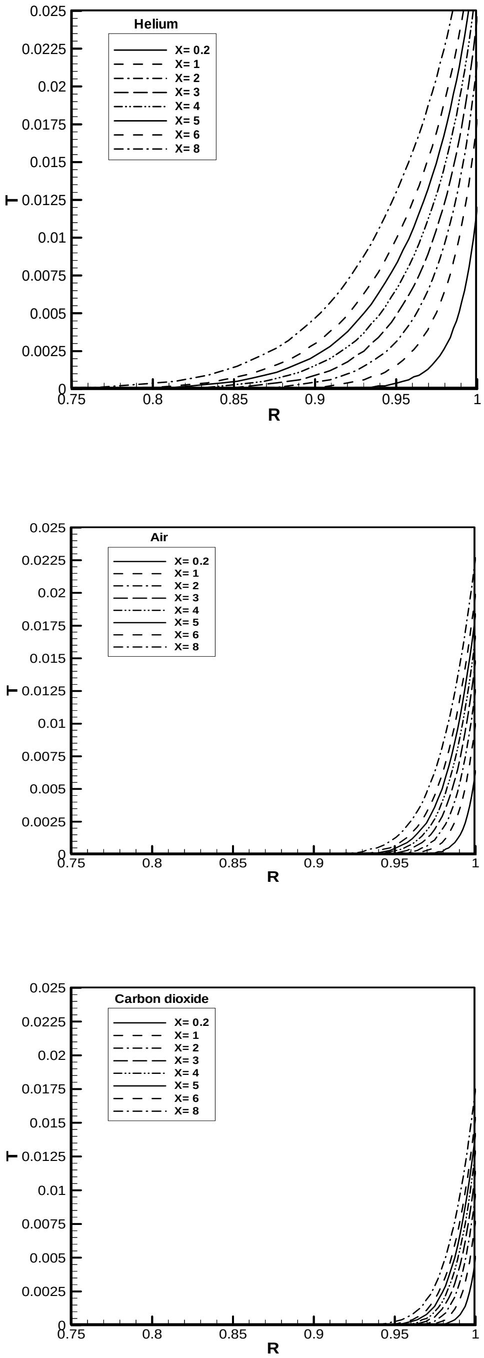


Fig (3) Dimensionless temperature distribution across the channel radius(V=0.5 lit/Sec)

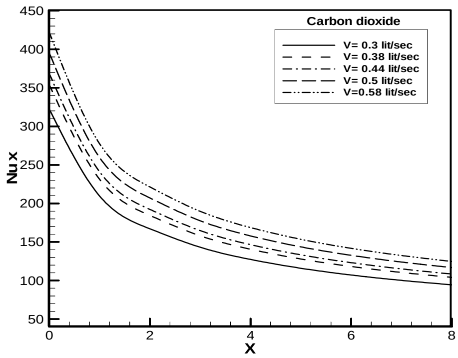


Fig (4) Local Nusselt number variation with the horizontal coordinate for Carbon dioxide

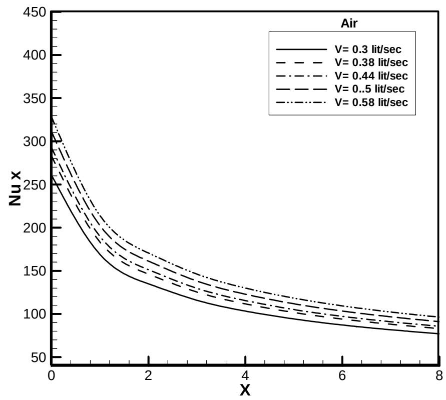


Fig (5) Local Nusselt number variation with the horizontal coordinate for Air

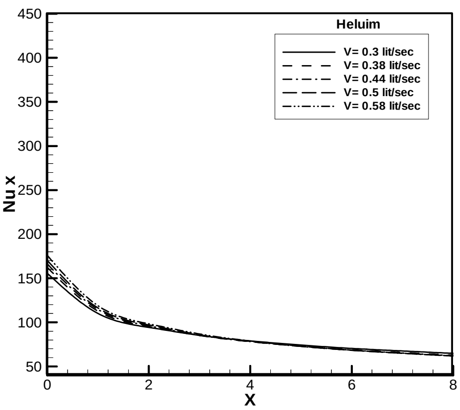
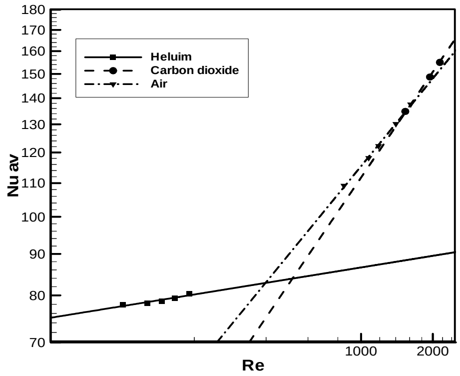


Fig (6) Local Nusselt number variation with the horizontal coordinate for Heluim



Fig(8) The relation between average Nusselt number with the Reynolds number

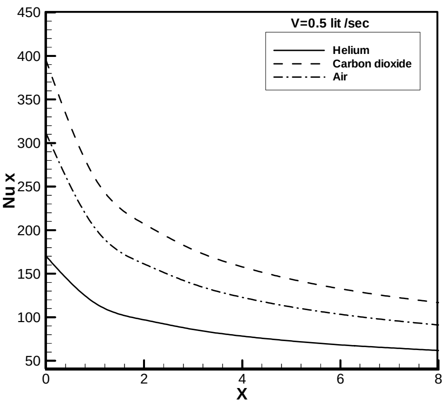
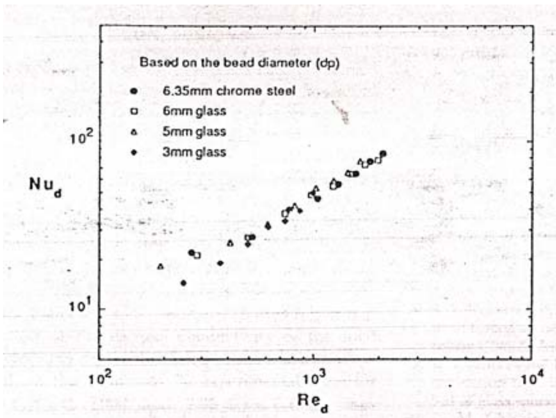


Fig (7) The influence of Prandtl number on the Nusselt number for



Fig(9) Average Nusselt Number (Nu_d) in apacked channel[5]