AN EMPIRICAL CORRELATION FOR ISOTHERMAL PARALLEL PLATE CHANNEL COMPLETELY FILLED WITH POROUS MEDIA

by

Mohammad O. HAMDAN

Mechanical Engineering Department, United Arab Emirates University, Al Ain, United Arab Emirates

Original scientific paper
DOI: 10.2298/TSCI120419015H

This study reports a simple empirical correlation for friction factor and Nusselt number for laminar, steady-state, hydraulically and thermally fully developed flow in isothermal parallel plate channel completely filled with porous media. The study is carried out using a finite difference numerical analysis. The Darcy-Brinkman-Forchheimer model is used to model the flow inside the porous media. The empirical correlations are developed to relate friction factor and Nusselt number to Darcy and Forchheimer coefficient.

Key words: *heat transfer, porous media, forced convection, empirical formula, friction factor*

Introduction

Heat transfer enhancement strategy has been used for various types of industrial applications such as shell-and-tube type heat exchangers, electronic cooling devices, thermal regenerators, and internal cooling of gas turbine blades. Enhancing the internal cooling of the channels is achieved by different augmentation techniques such as jet impingement [1], porous inserts [2], roughness elements [3], ribs, baffles [4], porous fins [5], or two-phase cooling [6]. Heat exchanger industries are seeking more compact and more cost-effective heat exchanger manufacturing techniques [7], which lead the way to use porous fins to augment heat transfer [8, 9].

Carman [10] and Collins [11] have investigated the fluid flow through porous material using Darcy's law. Beavers and Joseph [12] first investigated the fluid mechanics at the interface between a fluid layer and a porous medium over a flat plate. Closed-form analytical solutions for forced convection in parallel-plate ducts and in circular pipes partially filled with porous materials were obtained by Poulikakos and Kazmierczak [13] for constant wall heat flux. Poulikakos and Renken [14] presented the numerical results computed for a constant wall temperature and completely filled ducts. Vafai and Thiyagaraja [15] obtained an analytical approximate solution for the same problem based on matched asymptotic expansions for the velocity and temperature distributions. Later on, Vafai and Kim [16] presented an exact solution for the same problem. The problem of forced convection in channels partially filled with porous media was numerically investigated by Jang and Chen [17] using Darcy-Brinkman-Forchheimer model.

Implementing Darcy-Brinkman model, analytical solutions were obtained by Chikh *et al*. [18] for the problem of forced convection in an annular duct partially filled with a porous medium. The same problem was investigated numerically by the same group based on the

* Author's e-mail: mohammadH@uaeu.ac.ae

Darcy-Brinkman-Forchheimer model [19]. The transient behavior of a flow inside a parallel-plate channel partially filled with porous media was investigated by Al-Nimr and Alkam [20, 21].

Several experimental and numerical studies have been conducted to provide a deeper understanding of the transport mechanism of the momentum and the heat transfer in porous media. Hwang and Chao [22] showed that the smaller size of pore density can decrease the entrance length and increase the local Nusselt number in their packed bed experiment using sintered material. They introduced two-equation model to overcome heat transfer over prediction of conventional one equation model and showed the existence of non-equilibrium thermal condition between the fluid and the solid matrix. Kim *et al*. [23] experimentally investigated an asymmetrically heated packed bed filled with foam materials. Kim *et al*. [23] have developed correlation equations for friction factor and Nusselt number for different foam materials as function of Darcy number, Reynolds number, and Prandtl number. Recently, Huang *et al*. [24] experimentally and numerically showed that heat augmentation can be achieved by inserting porous medium in the core of the flow. Similar finding was reported earlier by Hamdan *et al*. [25] where numerical analysis of laminar flow between two constant temperature parallel plates shows enhancement in heat transfer by inserting porous substrate in the core of the flow. Rachedi and Chikh [26] numerically studied forced convection cooling in the presence of porous inserts in electronic devices. Results showed that the temperature dropped down by half. The effect of shape and location of porous insert is investigated numerically by Teamah *et al.* [27] to identify the parameters that can offer higher heat transfer with minimum pressure drop. Teamah *et al*. [27] reported the effect of the porous insert thickness and Darcy number on the velocity profiles, the local Nusselt number, the average Nusselt number, and the pressure drop.

From literature, the availability of general correlation for calculating friction factor and Nusselt number is limited to specific material such aluminum metal foam [23]. In this work, the author intention is to produce general empirical correlations describing the thermal performance via porous media. Such empirical correlations are highly desirable by applied engineers. The present numerical study reports the impact of different dimensionless parameters such as Darcy, Forchheimer, Reynolds, and relative thermal conductivity for steady-state fully developed laminar flow in isothermal parallel-plate channels filled with porous media. Darcy-Brinkman-Forchheimer model is used to describe the flow inside the porous domain.

Mathematical model

A schematic diagram for the problem under consideration is shown in fig. 1. The figure presents a 2-D isothermal parallel-plate channel with porous media sandwished between two parallel plates. A steady-state flow enters the channel with a uniform velocity distribution, U_i, constant temperature, T_i, and constant pressure P_i. The Forchheimer-Brinkman-Darcy model is adopted assuming laminar, single-phase, boundary layer flow with no internal heat generation, and neglecting viscous dissipation and axial conduction. Also, it is assumed that the porous medium is homogeneous, isotropic, consolidated, saturated with fluid, with invariant thermal properties, and chemically stable. The fluid is homogeneous, incompressible, and in-local thermal equilibrium with the solid matrix. On the basis of the dimensionless parameters given in the nomenclature, the equations of continuity, momentum, energy, and integral continuity for flow inside porous domains, reduce to the following dimensionless equations:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{1}$$

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=\frac{1}{\rho}\frac{\partial P}{\partial X}+\nu\frac{\partial^2 U}{\partial Y^2}-\left(\frac{\nu}{\mathrm{Da}}U+AU^2\right) \tag{2}$$

$$U\frac{\partial\theta}{\partial X}+V\frac{\partial\theta}{\partial Y}=k\frac{1}{\mathrm{Pr}_1}\frac{\partial^2\theta}{\partial Y^2} \tag{3}$$

$$\int_0^1 \rho U \mathrm{d}Y = U_\mathrm{o} \tag{4}$$

The equations in this problem are non-linear partial differential equations. These equation are second order in Y-direction and first order in the X-direction and hence two boundary conditions in Y-direction are defined for each variable U, V, and θ. While one boundary condition is defined in X-direction for each variable U, V, θ, and P.

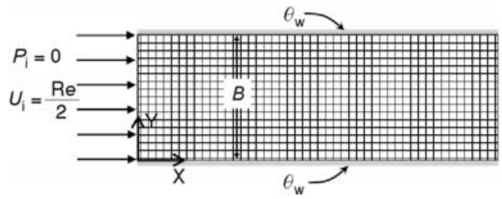

Figure 1. A schematic diagram of the problem under consideration

In eq. 2, the diffusion term in axial direction is neglected following the boundary layer approximation for the developing region which is canceled in the fully developed region due to the fully developed velocity profile. In eq. 3, axial conduction is neglected. Axial conduction, usually, is neglected if the conventional Peclet number is larger than 10 [28]. When the ratio of pore diameter to channel width is of the order of 0.01, it implies that axial conduction may be neglected if the modified Peclet number, Pe*, is of order larger than 0.1 [28].

The momentum equation has the following boundary conditions:

$$\begin{array}{llll}\text{At} & X=0, \text{ and } & 0<Y<B, & U_1=U_i \text{ and } V_1=0\\ \text{For} & X>0, \text{ and } & Y=0, & U_2=U_1=V_2=V_1=0\\ \text{For} & X>0, \text{ and } & Y=B, & U_2=U_1=V_2=V_1=0\end{array}$$

The energy equation has the following boundary conditions:

$$\begin{array}{llll}\text{At} & X=0, \text{ and } & 0<Y<B, & \theta_1=0\\ \text{For} & X>0, \text{ and } & Y=0, & \theta_\mathrm{w}=1\\ \text{For} & X>0, \text{ and } & Y=B, & \theta_\mathrm{w}=1\end{array}$$

The friction factor, and the local Nusselt number are defined as:

$$f=\frac{2\Delta P}{\rho_1 U_i^2}\left(\frac{D_\mathrm{h}}{B}\right) \tag{5}$$

$$\mathrm{Nu}=\frac{2hb}{k_1}=k\frac{2}{1-\theta_\mathrm{MC}}\left.\frac{\partial\theta}{\partial Y}\right|_{Y=0} \tag{6}$$

Numerical solution

The governing differential equations are transformed into the corresponding finite difference equations and are applied to a 2-D uniform grid. The 2-D of the grid under consideration simulate the axial and the transverse variables. The non-linear terms in momentum equation were linearized using the lagging technique by Hoffman and Frankel [29]. Following [29], the linearized implicit finite difference equations are derived using a second-order central difference scheme for the transverse derivatives, and first-order backward scheme for the axial derivatives. The convergence analysis of the present numerical scheme has been performed, and it is found that the derived finite difference equations resemble a consistent representation of the differential equations, and yet, the solution is unconditionally stable (*i. e.* the solution is available and approach true solution for any value of $\Delta X \to 0$ and $\Delta Y \to 0$). The linearized momentum finite difference equations, together with the boundary conditions, are transformed to a set of algebraic equations.

Table 1. Grid independent study by changing mesh size [m]×[n]. The table shows the Nusselt number in the developing region at X = 0.1 and Re = 200 for Da = 0.01 and A = 0.01

	ΔX	0.001	0.0001	0.00005
	[m]	200	2000	4000
ΔY	[n]			
0.02	50	18.8124	18.8449	18.8467
0.01	100	19.048	19.1402	19.1456
0.005	200	19.1199	19.2599	19.2687

The finite difference energy equations are transformed to tridiagonal set of algebraic equations that are solved by Matlab7 [30] matrix inverse techniques. The convergence criteria for all variable are set to the default used in Matlab matrix inverse algorithm which is $1 \cdot 10^{-15}$. A grid refinement procedure has been performed through numerical experimentations and reported in tab. 1. As shown in tab. 1, the optimum choice is found to be ΔX = 0.0001 and ΔY = 0.005.

Results and discussion

The present problem is solved by means of an implicit finite difference scheme over a 2-D mesh which consists of the independent variables, X and Y. The adopted scheme is discussed in more detail by Hamdan *et al*. [25]. In order to examine the validity of this study, several investigation strategies are employed. First, a computer run is made with a high value of the Darcy number, Da $\to \infty$, and a very low value of the microscopic inertial coefficient, A = 0. The result of this run is so close to our knowledge of the clear flow between two-parallel plate ducts. Second, the numerical solution velocity profile for flow between two-parallel-plate channel filled with porous media using Brinkman-Darcy model (Da = 0.01 and A = 0) is compared with Poulikakos and Kazmierczak [13] which shows excellent agreement as shown in fig. 2(a). The value of Nusselt number at the isothermal walls of parallel-plate duct is shown in fig. 2(b) and compared to literature which reports for fully developed a Nusselt value of 7.54 for clear duct [31] and 9.87 for a duct filled with porous media using Brinkman-Darcy model with Da $\to$ 0 and A = 0 [32].

The results obtained in this study have been computed using the following operating and design parameters: Pr = 0.72, $k_R = 1$, $\mu_R = 1$, $\rho_R = 1$, Re = $2U_i$ = 200. The experimental work by Kim *et al*. [23] generally agrees with the correlation suggested by Beavers and Sparrow [33] which represents an attempt to develop a correlations for friction factors (f) and Nusselt number (Nu) for the aluminum foams. This study proposes a general friction factor (f) correlation for fully porous channel with known value of Darcy and Forchheimer number. The following corre-

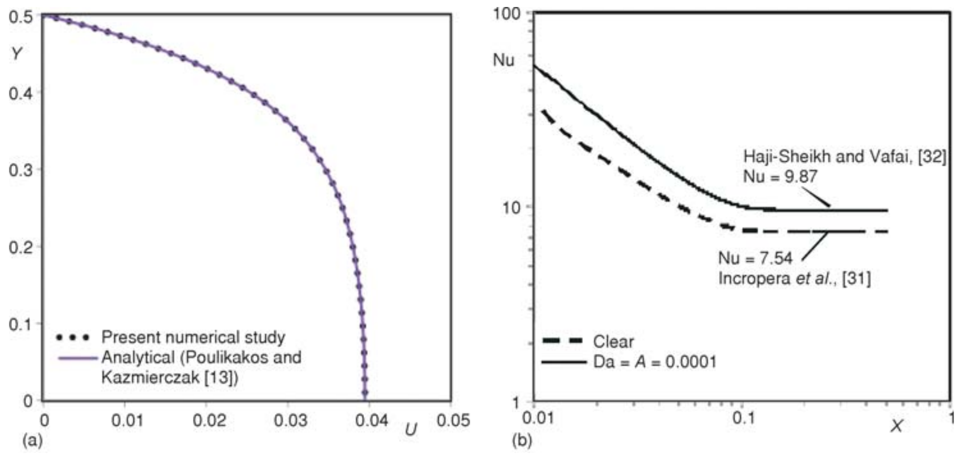

Figure 2. Validation study results; (a) the fully developed velocity profile for Da = 0.01 and A = 0, and (b) local Nusselt number for clear duct and fully filled with porous media with Da $\to$ 0 and A = 0

lation is proposed based on the knowledge that Darcy-Brickman model is linearly independent while the Forchheimer term is the one forcing the problem to be non-linear for the fully developed steady state flow. Hence, the following empirical friction factor correlation with maximum 5.4% deviation for 100 < Re < 2000 is suggested:

$$f = \frac{96}{\mathrm{Re}} + \frac{8.45}{\mathrm{Da\,Re}} + 5.0A^{0.961} \tag{7}$$

The first term represents the friction factor due to wall shear force ($f1$ = 96/Re), the second term ($f2$ = 8.45/DaRe) represents Darcy friction portion, and third term represents Forchheimer friction part due to the second order velocity, ($f3 = 5.0A^{0.961}$).

To get a correlation of Nusselt number for flow between isothermal parallel plate the Nusselt number is calculated using the developed computation model for different value of Darcy number, Forchheimer number, and Reynolds number. The following empirical Nusselt number correlation with maximum 3.1% deviation for 100 < Re < 2000 is suggested:

$$\mathrm{Nu} = k\left[7.54 + \frac{f2}{f}\left(\frac{0.023}{0.011 + \mathrm{Da}}\right) + \frac{f3}{f}\left(\frac{2.1}{1 + A^{-0.4}}\right)\mathrm{Re}^{0.04}\right] \tag{8}$$

The deviation in the correlation was calculated with respect to the CFD results and were calculated for friction factor as Deviation= $|f_{\mathrm{Correlation}} - f_{\mathrm{CFD}}|/f_{\mathrm{CFD}}$ and for Nusselt number as Deviation = $|\mathrm{Nu}_{\mathrm{Correlation}} - \mathrm{Nu}_{\mathrm{CFD}}|/\mathrm{Nu}_{\mathrm{CFD}}$ .The Nusselt number correlation approaches clear isothermal clear parallel plate channel Nusselt number value of 7.54 for Da $\to \infty$, $A \to 0$, and $k \to 1$. The second term in eq. (8) is due to Darcy effect which does not depend on Reynolds number. Such behavior agrees with earlier work where only Daryc-Brickman model is imposed, such as the work of Poulikakos and Kazmierczak [13]. Poulikakos and Kazmierczak [13] reported that Reynolds has not effect on Nusselt number for laminar fully developed steady-state flow. However as inertia effect increases, the Forchheimer coefficient effect become more pronounce. Hence, for Forchheimer-Brinkman-Darcy model, Reynolds number has moderate effect on Nusselt number as shown in the third term in eq. (8) and hence Reynolds number appeared in the third term with small power value of 0.04. Such behavior also agrees with earlier work done by Kim *et al*. [23] whom proposed an empirical Nusselt number correlation from aluminum foam materials.

The correlations (7) and (8) are assessed against the numerical analysis for different values of Darcy and Forchheimer coefficient and the results are presented in figs. 3 to 6. As shown in fig. 3, as Darcy number increases the friction coefficient decreases and reaches to asymptotic value depending on the Forchheimer coefficient. Also Nusselt number will decrease with the increase of Darcy number and reach asymptotic value depending on the Forchheimer coefficient. From Darcy definition as permeability decrease Darcy number decreases forcing faster flow to pass next to the wall and hence enhances the heat transfer and increase Nusselt number.

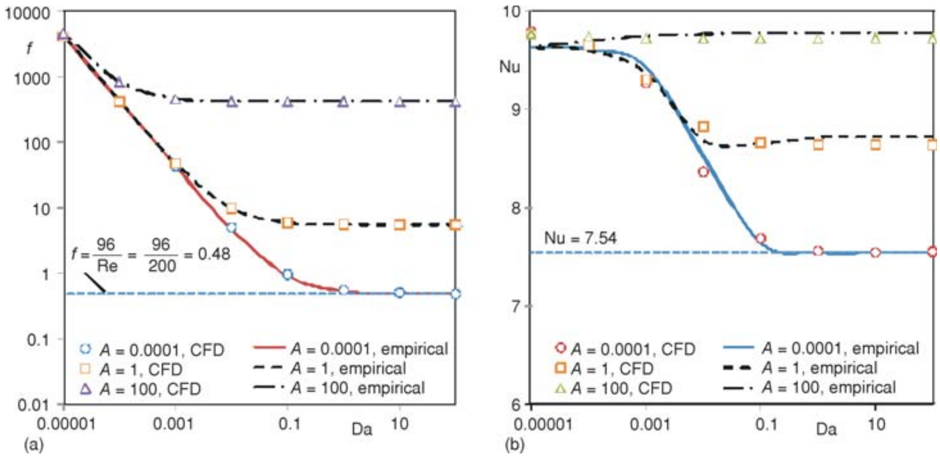

Figure 3. The effect of Darcy number for Re = 200 on (a) friction factor and (b) Nusselt number

The opposite trend is expected for Forchheimer coefficient friction factors as shown in fig. 4 nevertheless with higher order effect since as shown in eq. (2) the Forchheimer coefficient is related to the second order of velocity. The Forchheimer coefficient has been investigated by different scientist and many relations have been developed based on porous cavity shapes. As shown in fig. 4(a), as Forchheimer coefficient increase the friction factor increase.

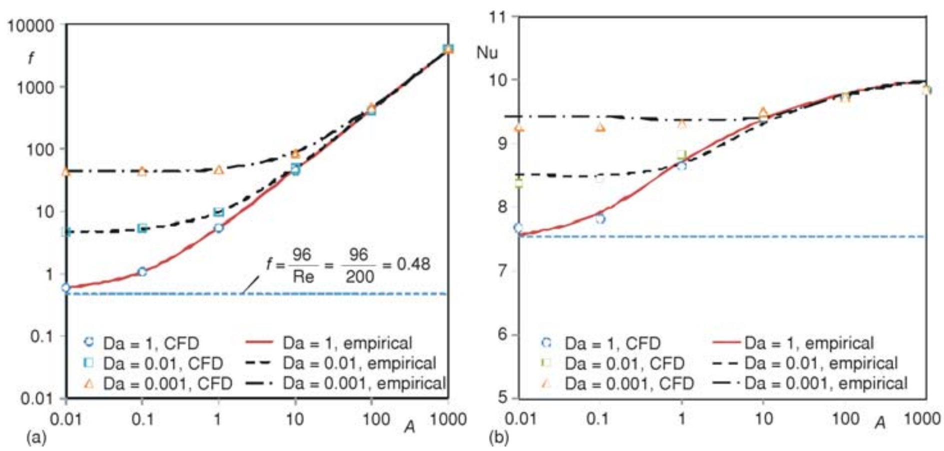

Figure 4. The effect of Forchheimer coefficient for Re = 200 on (a) friction factor and (b) Nusselt number

From fig. 5(a), it is clear that as Reynolds number increases the friction factor decreases since the effect of shear to inertia forces decreases with the increase in Reynolds number. However for high Forchheimer coefficient, the friction factor asymptotic value as low Reynolds number which mean that inertia forces is more pronounce effect and that Forchheimer coefficient is more related to pressure drag. As shown in fig. 5(b) (CFD results), the Reynolds number has no effect on Nusselt number when Forchheimer coefficient is very small. While for high Forchheimer coefficient, the effect of Reynolds number on Nusselt number is more pronounced.

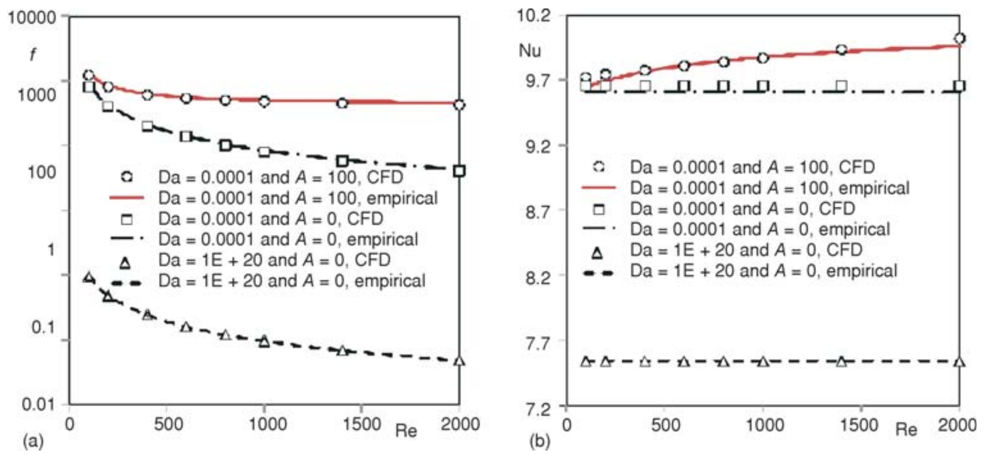

Figure 5. The effect of Reynolds number for different values of Darcy number and Forchheimer coefficient on (a) friction factor, and (b) Nusselt number

This can be explained that Darcy number is related to friction drag while Forchheimer coefficient is related to pressure drag. Hence when Darcy number is the dominating factor compared to Forchheimer coefficient, then the flow behave as internal fully developed flow where Nusslet number is expected to be independent of Reynolds number for laminar flow. However when Forchheimer coefficient is the dominated factor compared to Nusslet number, it is expected that the flow behave more likely as external flow where inertia forces becomes important and hence Nusselt number is highly dependent on Reynolds number.

Finally in fig. 6, as expected the relative thermal conductivity have direct effect on Nusselt number and as relative thermal conductivity increases the Nusselt number increases linearly as shown in fig. 6 and as indicated in eq. (8).

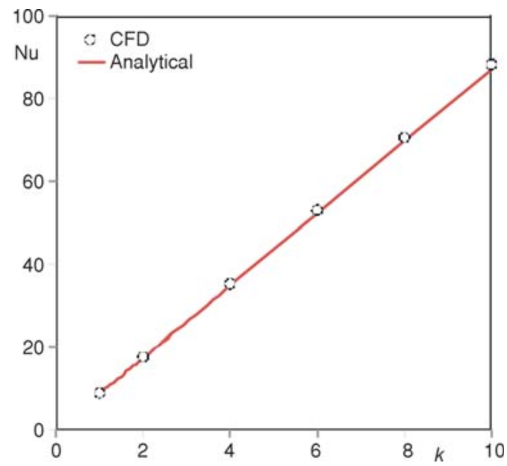

Figure 6. The effect of thermal conductivity ratio on Nusselt number for Re = 200, Da = 0.01, and *A* = 1

Conclusions

A numerical investigation on the flow and convective heat transfer characteristics for isothermal parallel plate channel filled with porous media has been performed. The following conclusion can be stated.

- Friction factor and Nusselt number for flow between isothermal walls of parallel-plate duct can be described using simple empirical correlations that depend on Darcy number and Forchheimer coefficient and is suggested by this study.
- Darcy friction term is directly related to friction drag inside the porous matrix. So as Darcy decrease then permeability decrease which is highly dominated by increase in surface area and hence by friction drag.
- Forchheimer friction term is directly related to pressure drag inside the porous matrix. So as Forchheimer coefficient is highly dominated by frontal and blockage area and hence by pressure drag.
- For porous media that consists of straight holes, one expects that friction drag is the dominated force and hence Darcy number plays the main role in describing friction factor.
- For porous media that consists of spherical hole and many blockage areas, one expects that pressure drag force is the dominated force and hence Forchheimer coefficient plays a role in describing friction factor.

Nomenclature

A – microscopic inertial or form drag coefficient, $((= \varepsilon Fb/\rho_R(K)^{1/2}))$, [–]
B – dimensionless channel height, (= b/b = 1), [–]
b – dimensional channel height, [m]
C_1 – specific heat constant of the fluid, [$\mathrm{kJkg^{-1}K^{-1}}$]
Da – Darcy number, $(= K/b^2)$, [–]
D_h – hydraulic diameter, $(= 2b)$,[m]
d – dimensional pore diameter, [m]
F – Forchheimer coefficient $((= 1.8)/(180\varepsilon^5))$, [–]
f – friction factor, $(= 2\tau/\rho u^2)$, [–]
$f1$ – friction factor component due to wall, [–]
$f2$ – friction factor due to Darcy effect, [–]
$f3$ – friction factor due to Forchheimer effect, [–]
h – local heat transfer coefficient, [$\mathrm{Wm^{-2}K^{-1}}$]
K – permeability of the porous substrate, [$\mathrm{m^{-2}}$]
k_1 – thermal conductivity of the fluid, [$\mathrm{Wm^{-1}K^{-1}}$]
k_2 – thermal conductivity of porous domain $[= k_f\varepsilon + k_s(1-\varepsilon)]$, [$\mathrm{Wm^{-1}K^{-1}}$]
k – thermal conductivity ratio, $(= k_2b/k_1)$, [–]
m – number of grid in X-direction, [–]
Nu – local Nusselt number $(= h_2b/k_1)$, [–]
n – number of grid in Y-direction, [–]
P – dimensionless pressure, $(= pb^2/\rho 1 v_1^2)$, [–]
Pe* – modified Peclet number $(= U_i d^2/D_h\alpha)$, [–]
Pr – Prandtl number of the fluid, $(= C_1\mu_1/k_1)$, [–]
p – pressure, [Pa]
T – temperature, [K]
T_{CM} – mixing cup temperature $[= \int_0^y C_2\rho_2u_2T_2\mathrm{d}y/(C_2\rho_1U_iy)]$, [K]
U – dimensionless volume averaged axial velocity $(= uB/\nu_1)$, [–]
u – axial velocity, [$\mathrm{ms^{-1}}$]
υ – transverse velocity, [$\mathrm{ms^{-1}}$]
V – dimensionless transverse velocity $(= ub/\nu_1)$, [–]
X – dimensionless axial co-ordinate $(= x/b)$, [–]
x – dimensional axial co-ordinate, [m]
Y – dimensionless transverse co-ordinate $(= y/b)$, [–]
y – dimensional transverse co-ordinate, [m]

Greek symbols

α – thermal diffusivity $(= k\rho/C)$, [$\mathrm{m^2s^{-1}}$]
Δ – increment in numerical mesh network space
ε – porosity, [–]
θ – dimensionless temperature $[= (T - T_i)/(Tw - T_i)]$, [–]
θ_{MC} – dimensionless mixing cup temperature $[= (T_{MC} - T_i)/(T_w - T_i)]$, [–]
μ – dynamic viscosity, [Pa·s]
ν – kinematics viscosity, [$\mathrm{m^2s^{-1}}$]
ρ – density, [$\mathrm{kgm^{-3}}$]
τ – shear, [$\mathrm{Nm^{-2}}$]

Subscripts

1 – fluid domain property
2 – porous domain property
MC – mixing cup
i – inlet
w – wall

References

[1] Hamdan, M. O., *et al*., Measurement and Modeling of Confined Jet Discharged Tangentially on a Concave Semicylindrical Hot Surface, *ASME Journal of Heat Transfer, 133* (2011), 12, DOI:10.1115/1.4004529
[2] Al-Nimr, M.A., *et al*., On Forced Convection in Channels Partially Filled with Porous Substrates, *Heat and Mass Transfer, 38* (2002), 4-5, pp. 337-342
[3] Eifel, M., *et al*., Experimental and Numerical Analysis of Gas Turbine Blades with Different Internal Cooling Geometries, *ASME J. Turbomach, 133* (2011), 1
[4] Dutta, P., Hossain, A., Internal Cooling Augmentation in Rectangular Channel Using Two Inclined Baffles, *International Journal of Heat Fluid Flow, 26* (2005), 2, pp. 223-232
[5] Mahjoob, S., Vafai, K., A Synthesis of Fluid and Thermal Transport Models for Metal Foam Heat Exchangers, *International Journal of Heat Mass Transfer, 51* (2008), 15-16, pp. 3701-3711
[6] Hamdan, M. O., Elnajjar, E., Thermodynamic Analytical Model of a Loop Heat Pipe, *Journal of Heat and Mass Transfer, 46* (2009), pp. 167-173, DOI: 10.1007/s00231-009-0555-0
[7] Narasimhan, S., Majdalani, J., Characterization of Compact Heat Sink Models in Natural Convection, *IEEE Transaction on Components and Packaging Technologies, 25* (2002), 1, pp. 78-86
[8] Hamdan, M. O., Al-Nimr, M. A., The Use of Porous Fins for Heat Transfer Augmentation in Parallel Plate Channels, *Transport in Porous Media, 84* (2009), 2. pp. 409-420
[9] Ko, K.-H., Anand, N. K., Use of Porous Baffles to Enhance Heat Transfer in a Rectangular Channel, *International Journal of Heat and Mass Transfer, 46* (2003), 22, pp. 4191-4199
[10] Carman, P. C., *Flow of Gases through Porous Material*, Academic Press, New York, USA, 1965
[11] Collins, R. E., *Flow of Fluids through Porous Material*, Reinhold, New York, USA, 1961
[12] Beavers, G. S., Joseph, D. D., *Boundary Conditions at Naturally Permeable Wall*, *Journal of Fluid Mechanics, 13* (1967), 1, pp. 197-207
[13] Poulikakos, D., Kazmierczak, M., Forced Convection in a Duct Partially Filled with a Porous Material, *ASME Heat Transfer, 109* (1987), 3, pp. 653-663
[14] Poulikakos, D., Renken, K., Forced Convection in a Channel Filled with a Porous Medium, Including the Effect of Flow Inertia, Variable Porosity, and Brinkman Friction, *ASME Heat Transfer, 109* (1987), 4, pp. 880-889
[15] Vafai, K., Thiyagaraja, R., Analysis of Flow and Heat Transfer at the Interface Region of a Porous Medium, *International Journal of Heat Mass Transfer, 30* (1987), 7, pp. 1391-1405
[16] Vafai, K., Kim, S. J., On the Limitations of the Brinkman-Forchheimer-Extended Darcy Equation, *International Journal of Heat Mass Transfer, 1*6 (1995), 1, pp. 11-15
[17] Jang, J. Y., Chen, J. L., Forced Convection in a Parallel Plate Channel Partially Filled with a High Porosity Medium, *International Communications in Heat Mass Transfer, 19* (1992), 2, pp. 263-273
[18] Chikh, S., *et al.,* Analytical Solution of Non-Darcian Forced Convection in an Annular Duct Partially Filled with a Porous Medium, *International Journal of Heat Mass Transfer, 38* (1995), 9, pp. 1543-1551
[19] Chikh, S., *et al*., Non-Darcian Forced Convection Analysis in an Annular Partially Filled with a Porous Material, *Numerical Heat Transfer, 28* (1995), 6, pp. 707-722
[20] Al-Nimr, M. A., Alkam, M., Unsteady Non-Darcian Forced Convection Analysis in an Annulus Partially Filled with a Porous Material, *ASME Heat Transfer, 119* (1997), 4, pp. 799-785
[21] Al-Nimr, M. A., Alkam, M. K., Unsteady Non-Darcian Fluid Flow in Parallel Plates Channels Partially Filled with Porous Materials, *Heat Mass Transfer, 33* (1998), 4, pp. 315-318
[22] Hwang, G. J, Chao, C. H., Heat Transfer Measurement and Analysis for Sintered Porous Channels, *ASME Heat Transfer, 116* (1994), 2, pp. 456-465
[23] Kim, S. Y., *et al*, Forced Convection from Aluminum Foam Materials in an Asymmetrically Heated Channel, *International Journal Heat Mass Transfer, 44* (2001), 7, pp. 1451-1454
[24] Huang, Z. F., *et al.*, Enhancing Heat Transfer in the Core Flow by Using Porous Medium Insert in a Tube, *International Journal of Heat and Mass Transfer, 53* (2010), 5-6, pp. 1164-1174
[25] Hamdan, M., *et al*., Enhancing Forced Convection by Inserting Porous Substrate in the Core of a Parallel-Plate Channel, *International Journal of Numerical Methods for Heat and Fluid Flow, 10* (2000), 5, pp. 502-518
[26] Rachedi, R., Chikh, S., Enhancement of Electronic Cooling by Insertion of Foam Materials, *Heat Mass Transfer, 37* (2001), 4-5, pp. 371-378
[27] Teamah, M. A., *et al*., Numerical Simulation of Laminar Forced Convection in Horizontal Pipe Partially or Completely Filled with Porous Material, *International Journal of Thermal Sciences, 50* (2011), 8, pp. 1512-1522

[28] Alkam, M., *et al*., Enhancing Heat Transfer in Parallel-Plate Channels by Using Porous Inserts, *International Journal of Heat Mass Transfer, 44* (2001), 5, pp. 931-938
[29] Hoffman, J. D., Frankel, S., *Numerical Method for Engineering and Scientists*, 2nd ed., CRC Press, Boca Raion, Fla., USA, 1992
[30] ***, Matlab® 7, Users Guide, Matlab, 2007
[31] Incropera, F. P., *et al*., *Introduction to Heat Transfer*, 5th ed., John Wiley and Sons, New York, USA, 2007
[32] Haji-Sheikh, A., Vafai, K., Analysis of Flow and Heat Transfer in Porous Media Imbedded Inside Various-Shaped Ducts, *47* (2004), 8-9, pp. 1889-1905
[33] Beavers, G. S., Sparrow, E. M., Non-Darcy Flow through Fibrous Porous Media, *Journal of Applied Mechanics, 36* (1969), 4, pp. 711-715

Paper submitted: April 19, 2012
Paper revised: January 14, 2013
Paper accepted: April 3, 2013