Numerical Investigation of Forced Convection Heat Transfer for Laminar Flow in Various Parallel Porous Channels

A. Zehforoosh, S. Hossainpour and A. A. Tahery

Abstract—**The laminar field and heat transfer enhancement of forced convection in various counterflow porous channels with a joint aluminum plate was studied, using control volume technique and SIMPLE procedure for the velocity-pressure Coupling. Hydrodynamic and heat transfer results are reported for three configurations: (1) nonporous channels (NPC) (2) both channels filled by porous structure (BPC), and (3) only the channel contains hot fluid, filled by porous structure and other one is nonporous (HPC). The effectiveness, pressure drop and temperature difference are evaluated for wide ranges of Darcy, Reynolds and Prandtl numbers, also effects of thermal conductivity ratio is considered. The results show that as the Darcy, Reynolds or Prandtl number decreases, an increase in effectiveness is obtained. Also increase of thermal conductivity ratio causes to different effects on effectiveness in each case.**

Index Terms— **Parallel channels, Counterflow, Effectiveness, Porous medium, Laminar flow.**

I. Introduction

One of the most important equipments in power and process industries are heat exchangers, which widely used in various engineering applications. Therefore, determining of the optimal design and operating conditions of heat exchangers become a major challenge of engineering.

In response to these demands, different techniques have been used in the past to obtain a well increscent in effectiveness of heat exchangers, including a variety of passive or active heat transfer techniques. One of the promising techniques to enhance the heat transfer rate is the application of the porous structures. By using the porous medium as transport medium, fluid mixing and the heat exchange area between the solid matrix and fluid phase, flowing through the porous medium, increase and this leads to significant enhancement in heat transfer.

There have been numerous investigations with theoretical analyses and numerical simulations of convection heat transfer in fluid-saturated porous media. J. Koh and R. Colony [1] Analysis of cooling effectiveness for porous material in a coolant passage. Kaviany [2] used the Brinkman extended Darcy model to obtain a numerical solution of laminar flow in a porous channel bounded by isothermal parallel plate. Analytical solutions of fully developed forced convection in a porous channel were presented by Vafai and Kim [3] and later by Lauriat and Vafai [4]. Chen and Hadim [5] numerically studied laminar forced flow in a porous channel filled with fibrous medium saturated with a power-law fluid. The result of this investigation showed that in the non-Darcy regime with decreases of power law index, the thermal boundary layer thickness decreases significantly. Consequently, the fully developed Nusselt number increased considerably in the non-Darcy regime. The effect of aluminum foam on the flow and heat transfer characterize, in an asymmetrically heated channel experimentally investigated by Kim et al. [6]. The correlations for the friction factor and the mean Nusselt number were obtained. Angirasa [7] investigated forced convection in a channel filled with metallic fibrous materials. Their results indicated that porous substrate substantially enhance the thermal performance in a channel. Boomsma et al. [8] experimentally investigated pressure drops and Nusselt numbers in the compressed open-cell aluminum foam heat exchangers. They observed significant improvement in the heat transfer efficiency over commercially available heat exchangers. Hadim and North [9] numerically investigated two-dimensional laminar forced convection in a sintered porous channel with inlet and outlet slots. The purpose of this study is to investigate the effect of the porous structure application on the heat transfer enhancement in counterflow heat exchangers.

A numerical investigation is performed on the hydrodynamic and heat transfer characteristics of counterflow heat exchanger. Furthermore, the influences of various thermal characteristics are determined. Also, the heat transfer and pressure drop variation in different cases are evaluated, and the results are compared with those of the nonporous channel case. It is shown that specific choices in certain governing parameters, such as the fluid and solid matrix properties, or permeability of the porous structure, cause significant effects on the effectiveness of the heat exchanger.

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II. Mathematical Formulation

A. Problem description

A schematic of the investigated geometry is shown in Fig. 1. The geometry contains two horizontal channels with length of L and height of H. The hot fluid enters the top channel at uniform temperature T_{h1} with a uniform velocity U_o from left and the cold fluid enters the low channel at uniform temperature T_{c1} with the same velocity from right of the channel. Both channels walls are insulated except between

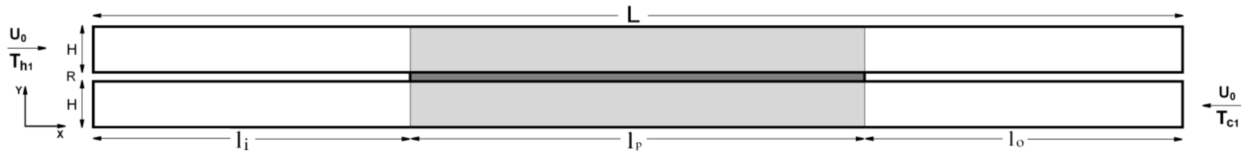

Fig.1. Schematic of channels geometry and coordinate system.

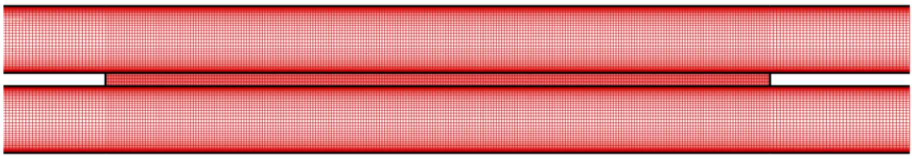

Fig.2. A non-uniform grid distributions, close-up of the region near aluminum plate.

two channels where contain an l_p length and R height aluminum plate. The lengths of l_i and l_o have been chosen to eliminate outlet boundary effects at the exits. In different cases of study up and down of aluminum plate filled with porous structures and results of these cases reported. The porous medium is considered to be homogeneous, isotropic, non-deformable, saturated with fluid and in local thermal equilibrium with the fluid. To neglect the channeling effect, fibrous media are considered which have a relatively constant porosity and permeability even close to the walls [10]. Also the effects of thermal dispersion, natural convection and thermal radiation are neglected. According to [11], the effective viscosity of porous medium can be considered equal to that of the fluid in higher porosities. The general forms of mass, momentum and energy balance governing equations, used in this study are [12]:

$$\nabla . V = 0 \tag{1}$$

$$\frac{\rho}{\varepsilon^2}(V.\nabla)V = -\nabla p + \rho g - \gamma\left[\frac{\mu}{K} + \frac{\rho C}{\sqrt{K}}|V|\right]V + \mu' \nabla^2 V \tag{2}$$

$$(\rho c)_f (V.\nabla T) = k\nabla^2 T \tag{3}$$

In the above equation ε is the value of porosity which equals one for a fluid field and vary between zero and one in a porous field. In (2) **μ′** presents the dynamic viscosity which equals μ, μ/ε in the fluid and porous fields respectively and in the (3) k is equals k_f , k_{eff} respectively. The governing continuity, momentum and energy equations in non-dimensional form are given below:

Continuity:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \tag{4}$$

Momentum:

$$Re\left(U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y}\right) = -\varepsilon\frac{\partial P}{\partial X} + \varepsilon\left(\frac{\partial^2 U}{\partial^2 X} + \frac{\partial^2 U}{\partial^2 Y}\right) - \gamma\left(\frac{\varepsilon^2}{Da^{1/2}}U + \frac{C\varepsilon^2}{Da^{1/2}}(U^2 + V^2)^{1/2}U\right) \tag{5}$$

$$Re\left(U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y}\right) = -\varepsilon\frac{\partial P}{\partial X} + \varepsilon\left(\frac{\partial^2 V}{\partial^2 X} + \frac{\partial^2 V}{\partial^2 Y}\right) - \gamma\left(\frac{\varepsilon^2}{Da^{1/2}}V + \frac{C\varepsilon^2}{Da^{1/2}}(U^2 + V^2)^{1/2}V\right) \tag{6}$$

Energy:

$$Re\left(U\frac{\partial\theta}{\partial X} + V\frac{\partial\theta}{\partial Y}\right) = \frac{1}{Pr}\left(\gamma(R_k - 1) + 1\right)\left(\frac{\partial^2\theta}{\partial^2 X} + \frac{\partial^2\theta}{\partial^2 Y}\right) \tag{7}$$

Where the non-dimensional parameters are given as bellow [13]:

$$X = \frac{x}{H}\ ,\ Y = \frac{y}{H}\ ,\ U = \frac{u}{U_o}\ ,\ V = \frac{v}{U_o}\ ,\ \theta = \frac{T - T_{c1}}{qH/k_f}\ ,$$
$$Re = \frac{U_o H}{\nu_f}\ ,\ P = \frac{pH}{\mu_f U_o}\ ,\ Pr = \frac{\nu}{\alpha}\ ,\ Da = \frac{K}{H^2}$$

R_k stands for the thermal conductivity ratio (k_{eff}/k_f). In (7) γ is either zero or one for fluid and porous reigns respectively.

B. Boundary conditions

As notified in the following equations the fluid enters the channel with a parabolic velocity and uniform temperature profile.

$$X = 0,\ 1.2 \le Y \le 2.2\ ,\ U = 1\ , V = 0\ , \theta = 0 \tag{8}$$
$$X = L,\ 0 \le Y \le 1\ ,\ U = -1\ , V = 0\ , \theta = 0 \tag{9}$$

The flow is supposed to be fully developed at the outlets. Also the diffusion fluxes for all flow variables are set to be zero.

$$X = 0,\ 0 \le Y \le 1\ ,\ \frac{\partial U}{\partial X} = \frac{\partial V}{\partial X} = \frac{\partial\theta}{\partial X} = 0 \tag{10}$$
$$X = L,\ 1.2 \le Y \le 2.2\ ,\ \frac{\partial U}{\partial X} = \frac{\partial V}{\partial X} = \frac{\partial\theta}{\partial X} = 0 \tag{11}$$

Upper and lower walls of channels are impermeable and no slip condition is applied. Also they are insulated except in contact with aluminum plate.

$$0 \le X \le L,\ Y = 0 \text{ and } Y = 2.2,\ U = 0,\ V = 0,\ \frac{\partial\theta}{\partial Y} = 0 \tag{12}$$

$$0 \le X \le L,\ Y = 1 \text{ and } Y = 1.2\quad U = 0,\ V = 0,$$
$$\begin{cases} k_{Al}\dfrac{\partial\theta_{Al}}{\partial Y} = k_{eff}\dfrac{\partial\theta}{\partial Y} & \textit{on aluminum plate} \\ \dfrac{\partial\theta}{\partial Y} = 0 & \textit{no aluminum plate} \end{cases} \tag{13}$$

Effectiveness of heat exchanger is defined as the ratio of actual heat transfer in heat exchanger to the maximum possible heat transfer between hot and cold fluids.

$$E = (T_{h1} - T_{h2}) / (T_{h1} - T_{c1}) \tag{14}$$

And local dimensionless pressure drop defined as difference between the inlet and the average pressure at a cross section.

$$\Delta P = \frac{\Delta p}{\mu_f} \tag{15}$$

III. Numerical Procedure and Validation

A control volume technique employed to discrete the differential equation. For the convective and diffusive terms, a first order upwind method was used while the SIMPLE procedure was introduced for the velocity-pressure Coupling [14]. A typical Grids distribution which used for the current study presented in Fig. 2 .The discretization grid is finer on the porous layers and near the walls where the velocity and temperature gradients are larger. Several different grid distributions have been examined on effectiveness of heat exchanger to ensure that the calculated results are grid independent. three different sets of grid systems, 480×115, 530×115, and 650×135 were investigated in this work for a case $Da = 10^{-6}$, C = 0.1 , $k_{eff}/k_f = 1$, Re = 200, $L_i = L_o = 3.5$, $L_P = 5$ and Pr = 0.69. A suitable grid is chosen by checking for the error between two successive levels of grid refinement to be less than 1% for the effectiveness values. It is found that the best distribution among these grids is the arrangement of 530×115 so all of the computations for this study were based on this grid system.

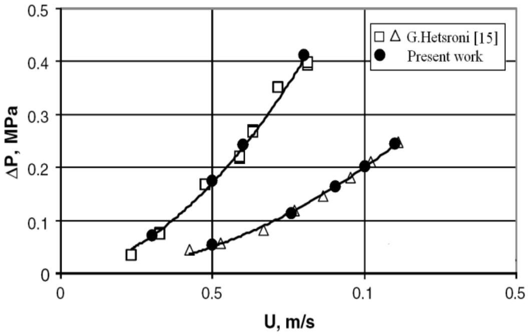

Fig.3. Comparison of the calculated pressure drop, with the experimental results of G. Hetsroni [15].

In Figs. 3 and 4 the numerical result of this work is compared with the relevant available experimental data of G. Hetsroni [15], which is in a rectangular channel with a porous block made of 316LSS and mounted heat source. (i.e., W=1/5, $L_T = 15$, $200 \le Re \le 2800$, (1) $Da = 8.6\times10^{-7}$, C = 1.24 and $R_k = 18.8$ (2) $Da = 3.84\times10^{-5}$, C = 3.07 and $R_k = 15.65$). As it is shown both of the calculated pressure drop and the mean Nusselt number are in good agreement with the experimental results. The maximum difference is less than 5 percent and therefore, the numerical code is reliable and can be used for the analysis of varying permeability effect.

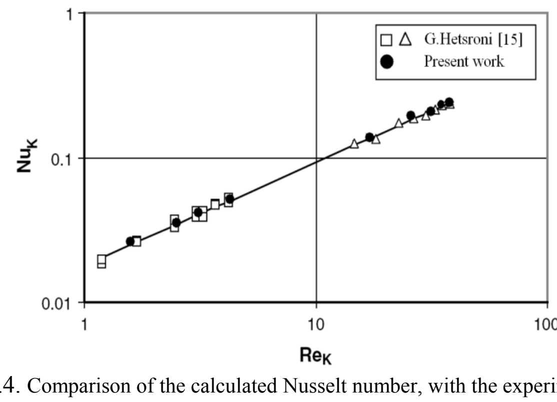

Fig.4. Comparison of the calculated Nusselt number, with the experimental results of G. Hetsroni [15].

IV. Results and Discussion

The fixed values selected for all cases are $L_i = L_o = 3.5$, $L_P = 5$, R = 0.2 and C = 0.1. The effects of governing physical parameters, such as Darcy number (Da), Reynolds number (Re), thermal conductivity ratio (R_k) and Prandtl number (Pr) on the hydrodynamic and energy field were investigated numerically. Results are presented as variation of effectiveness, pressure drop and temperature difference in the flow domain.

Streamlines in the **BPC** case for Re=200, $Da=10^{-6}$ is shown in Fig. 5. As shown in this figure, streamlines diverge when flow approaches porous media; also streamlines converge and form the fully developed flow after leaving porous media. This phenomenon leads to velocity profile become more flat inside the porous media.

Fig.5 streamlines in the BCP case for Re=200, Da=10-6.

The combined effects of Darcy and Reynolds numbers on the effectiveness of heat exchanger in **BPC** and **HPC** cases are depicted in Figs .6 and 7, respectively. It is shown that by increasing Re number from 50 to 800 the effectiveness of both cases decrease. This is due to the fact that by increasing of Re, more fluid passes from channels which causes heat transfer in less time.

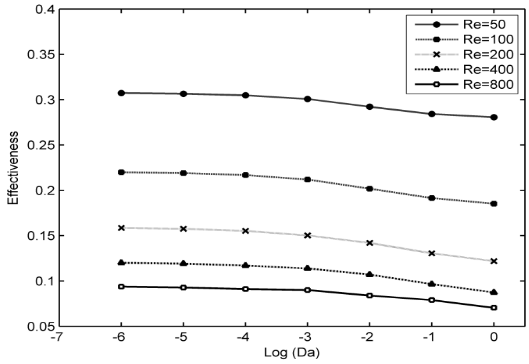

Fig.6. Variation of effectiveness in the BPC case with Darcy and Reynolds numbers for Rk = 1 and Pr = 0.69.

In the other hand As the Darcy number decreases, there is a small increase in effectiveness and this effect in **BPC** case is more visible than the **HPC** case. Finally according to the results it is observed that in special Darcy and Re numbers, the effectiveness of **BPC** is slightly more than **HPC** case.

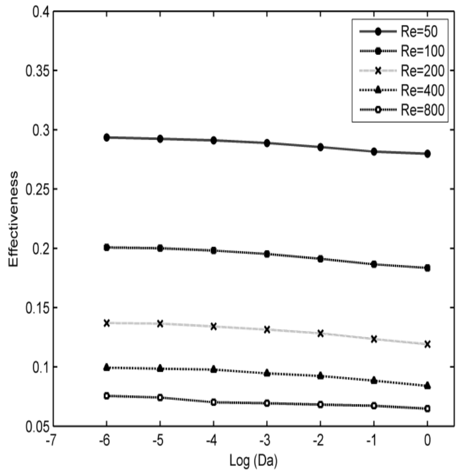

Fig.7. Variation of effectiveness in the HPC case with Darcy and Reynolds numbers for Rk = 1 and Pr = 0.69.

Eff/Eff_C Factor shows the ratio of effectiveness of porous heat exchanger to the effectiveness of nonporous heat exchanger. Variations of this factor in **BPC** and **HPC** cases are indicated in Figs .8 and 9, respectively.

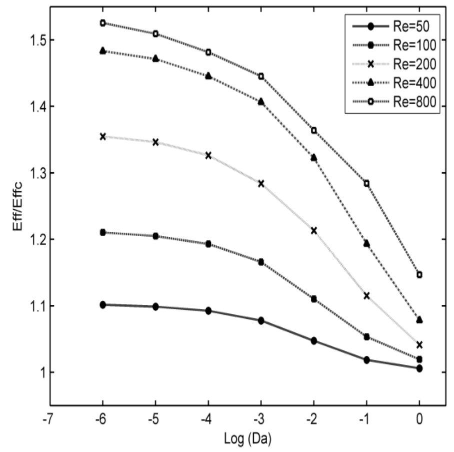

Fig.8. Variation of effectiveness factor (Eff/EffC) in the BPC case with Darcy and Reynolds numbers for Rk = 1 and Pr = 0.69.

It is shown that contrary to effectiveness, by increasing of Re number the effectiveness factor of both cases increase. Also with decrease of Darcy number a considerable increase is shown in amount of the effectiveness and with the same Re the **BPC** case increases more in effectiveness factor than the **HPC** case.

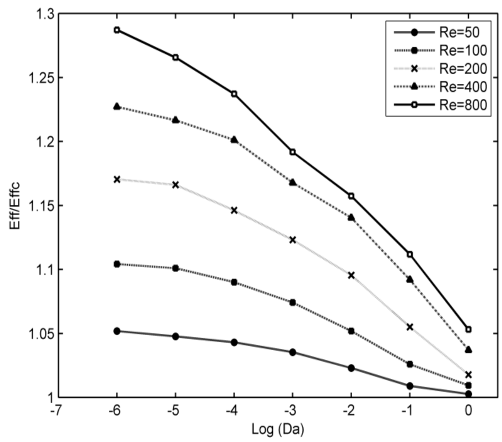

Fig.9. Variation of effectiveness factor (Eff/EffC) in the HPC case with Darcy and Reynolds numbers for Rk = 1 and Pr = 0.69.

Figs. 10 and 11 display the effects of Darcy and Reynolds numbers on the pressure drop in the **BPC** and **HPC** cases. With Darcy number decrease, the pressure drop in both cases rises suddenly.

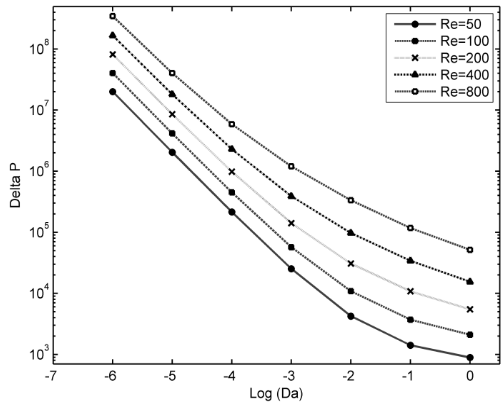

Fig.10. Total dimensionless pressure drop in the BPC case versus Darcy numbers at various Reynolds number.

Also the pressure drop in **BPC** case at the same Darcy numbers is approximately two times of the **HPC** case and this is due to negligible pressure drop in the nonporous channel compared with the porous channel. As expect in both cases pressure drop increases as Reynolds number increases.

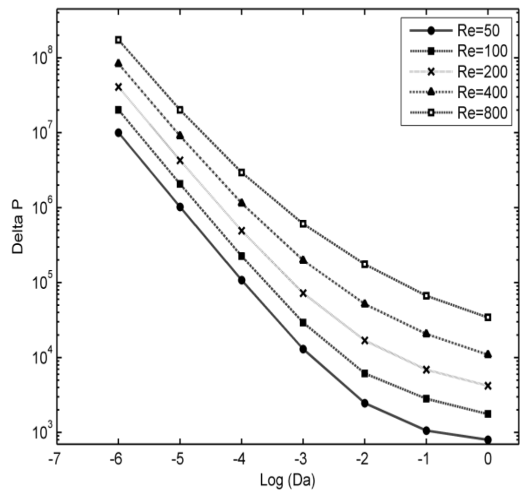

Fig.11. Total dimensionless pressure drop in the HPC case versus Darcy numbers at various Reynolds number.

The effects of thermal conductivity ratio on the effectiveness of **BPC** and **HPC** cases are shown in Figs. 12 and 13, respectively. These figures show that the effectiveness of heat exchangers will increase with the decrease of Darcy number. But, the amount of this increase is not considerable. On the other hand with increase of $R_{k,}$ significant increase effectiveness is obtained.

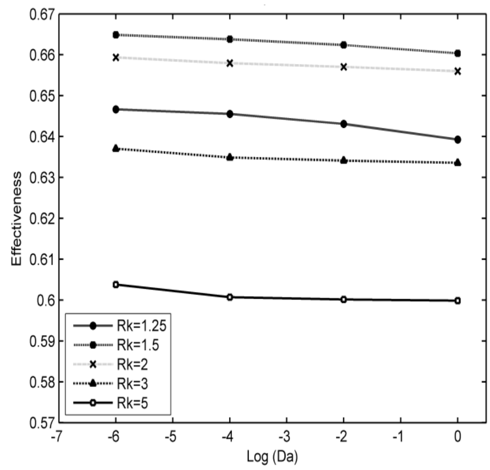

Fig.12. Variation of effectiveness in the BPC case with thermal conductivity ratio and Darcy number for Re = 200 and Pr = 0.69.

In **BPC** case, there is an optimum R_k , for which the effectiveness is maximum. Any increase or decrease in R_k will result in the decrease of the effectiveness. For a fixed Darecy number, it is shown that the effectiveness distribution is sharply increased by changing R_k from 1 to 1.5, and drops less steeply by changing R_k from 1.5 to 5.

But in **HPC** case, it was observed that the more the thermal conductivity ratio increases, the more the effectiveness increases. (i.e., the effectiveness in R_k=1 shown in Figs.6 and 7.)

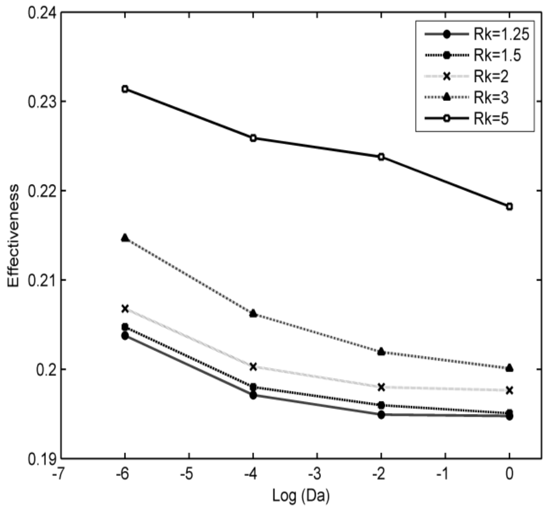

Fig.13. Variation of effectiveness in the HPC case with thermal conductivity ratio and Darcy number for Re = 200 and Pr = 0.69.

T/T_{C1} is defined to indicate the ratio of local temperature of hot and cold fluids in the centerline of porous channels to the cold fluid inlet temperature. Fig. 14 show this ratio versus X, for different R_k in Re=200, Da=10^{-6} and Pr=0.69 for **BPC** case. With increasing thermal conductivity ratio, the rate of decrease and increase of the hot and cold fluids temperature is increased and their temperatures faster get closer. But this phenomenon dose not leads to more decline of hot fluid temperature and hence effectiveness does not increase.

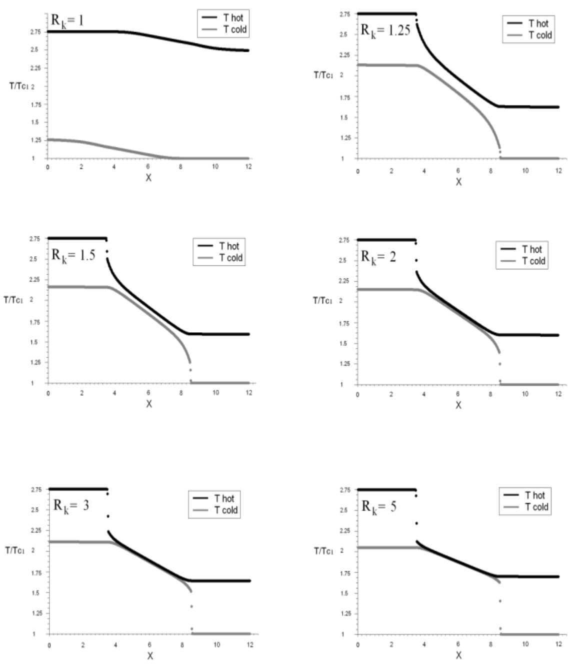

Fig.14. local dimensionless temperature of hot and cold fluids in the BPC case in the centerline of porous channels for Re=200, Da=10-6, Pr=0.69 and various thermal conductivity ratio.

Effectiveness of **BPC** and **HPC** cases in different Prandtl numbers are displayed in Figs. 15 and 16 respectively.

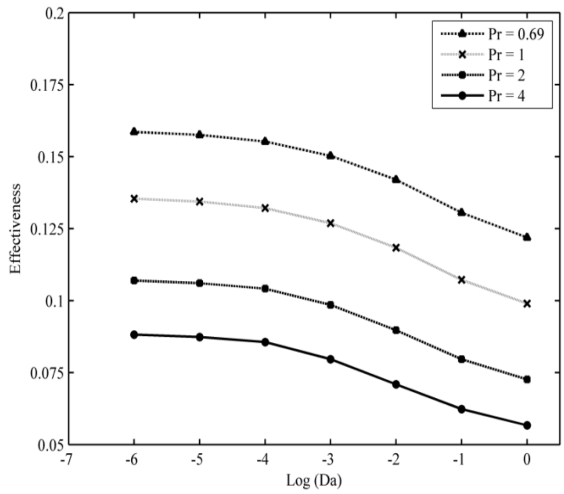

Fig.15. Variation of effectiveness in the BPC case with Darcy and Prandtl numbers for Re = 200 and Rk = 1.

It is found out that the effectiveness of each case in different Darcy number decreases as Pr increases. But the rate of this reduction decelerates by more increase in Prandtl number.

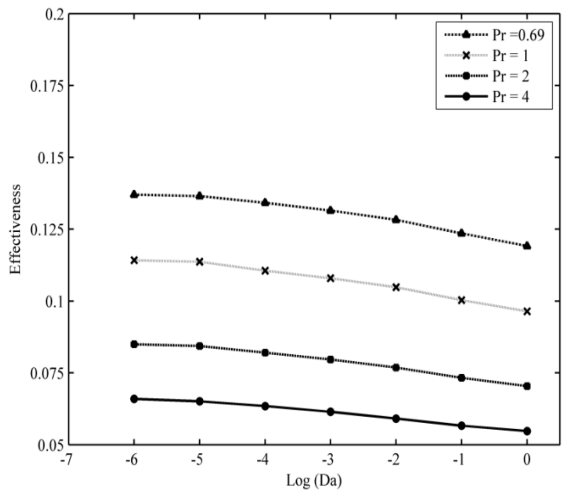

Fig.16. Variation of effectiveness in the HPC case with Darcy and Prandtl numbers for Re = 200 and Rk = 1.

The variation of Eff/Eff_C Factor of **BPC** and **HPC** cases in different Pr are shown in Figs.17and 18 respectively. In **BPC** case, the more the Pr number increases, the more effectiveness increases, but in HPC case there is an optimum for the Pr in which the effectiveness is maximum.

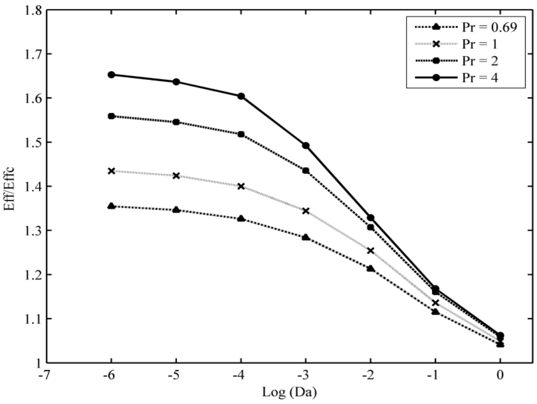

Fig.17. Variation of effectiveness Factor (Eff/EffC) in the BPC case with Darcy and Prandtl numbers for Re = 200 and Rk = 1.

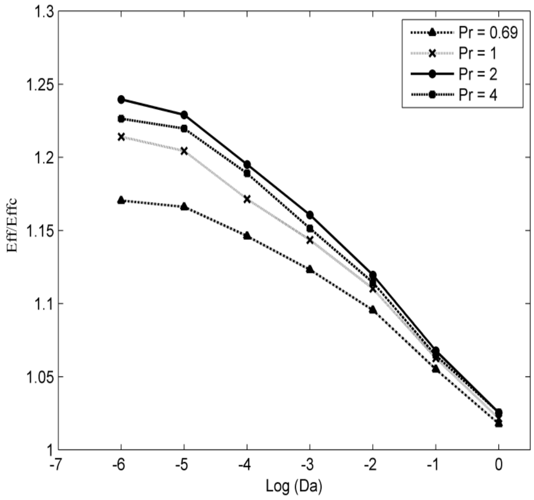

Fig.18. Variation of effectiveness Factor (Eff/EffC) in the HPC case with Darcy and Prandtl numbers for Re = 200 and Rk = 1.

V. Conclusion

In order to investigate the effects of using porous medium on flow structure and the heat transfer in parallel channel with a joint aluminum plate, a study of laminar forced convection was performed. Hydrodynamic and heat transfer results are reported for three cases of nonporous channels (**NPC**) , both channels filled by porous structure (**BPC**), the channel contains hot fluid, filled by porous structure and the other one is nonporous (**HPC**).

The results show that the more the Darcy, Reynolds or Prandtl number decreases, the more effectiveness of heat exchanger increases. But increase of thermal conductivity ratio causes to different effects on effectiveness of **BPC** and **HPC** cases. But in all cases significant increase in effectiveness is obtained.

Eff/Eff_C Factor is introduced to compare effectiveness of **BPC** and **HPC** cases with effectiveness of nonporous heat exchanger. It is indicated that, contrary to effectiveness, by increasing of Reynolds and Prandtl number the effectiveness factor of both cases increase.

The results of this study clearly demonstrate that using porous structure in the parallel channels with counterflow is an effective method to augment the effectiveness.

Nomenclature

c specific heat ($J.kg^{-1}.K^{-1}$)
C inertia coefficient
Da Darcy number (K/R^2)
E Effectiveness of heat exchanger ($(T_{h1}-T_{h2})/(T_{h1}-T_{c1})$)
Eff Effectiveness of heat exchanger
Eff_C Effectiveness of nonporous heat exchanger
H channels height (m)
k thermal conductivity ($W.m^{-1}.K^{-1}$)
K permeability of the porous medium (m^2)
l_i length of channel upstream from the aluminum plate (m)
l_o length of channel downstream from the aluminum plate (m)
l length of the channel (m)
P dimensionless pressure, $pH/\mu U_{av}$
Pr Prandtl number, ν/α
q heat flux, transferred between two fluid ($W.m^{-2}$)
R thickness of aluminum plate (m)
Re Reynold number, U_0H/ν
R_k thermal conductivity ratio, k_{eff}/k_f
T temperature (K)
u x-component velocity ($m.s^{-1}$)
v y-component velocity ($m.s^{-1}$)
V velocity vector ($m.s^{-1}$)
W width of heat source (m)
x horizontal coordinate (m)
y vertical coordinate (m)

Greek symbols

α thermal diffusivity ($m^2.s$)
γ binary parameter defined in Eq. (2-5-6-7)
ε porosity of the porous medium
Θ dimensionless temperature, $(T-T_i)/(qH/k_f)$
μ dynamic viscosity ($kg.m^{-1}.s^{-1}$)
ν kinematic viscosity ($m^2.s^{-1}$)

Subscripts

c cold
eff Effective property in porous region
h hot
i inlet
o outlet
p porous
1 inlet condition
2 outlet condition

References

[1] J.C.Y. Koh, R. Colony, Analysis of cooling effectiveness for porous material in a coolant passage, ASME J. Heat Transfer 96 (1974) 324–330.
[2] M. Kaviany, Laminar flow through a porous channel bounded by isothermal parallel plates, Int. J. Heat Mass Transfer 28 (4) (1985) 851–858.
[3] Vafai, K., and Kim, C. L., 1989, "Forced Convection in a Channel Filled With a Porous Medium: An Exact Solution," ASME JOURNAL OF HEAT TRANSFER, Vol. I 11, pp. 1103-1106.
[4] Lauriat, G., and Vafai, K., 1991, "Forced Convection and Heat Transfer Through a Porous Medium Exposed to a Flat Plate or a Channel," in: *Convective Heat and Mass Transfer in Porous Media,* S. Kakac, B. Kilkis,

[5] Chen, G., Hadim, H.A.: Forced convection of a power-law fluid in a porous channel—numerical solutions. Heat Mass Transf. 34, 221–228 (1998)

[6] Kim, S.Y., Kang, B.H., Kim, J.H.: Forced convection from aluminum foam materials in an asymmetrically heated channel. Int. J. Heat Mass Tran. **44**, 1451–1454 (2001)

[7] D. Angirasa, Forced convective heat transfer in metallic fibrous materials, ASME J. Heat Transfer 124 (2002) 739–745.

[8] Boomsma, K., Poulikakos, D., Zwick, F.: Metal foams as compact high performance heat exchangers. Mech. Mater. **35**, 1161–1176 (2003)

[9] Hadim, H., North, M.: Forced convection in a sintered porous channel with inlet and outlet slots. Int. J. Therm.Sci. 44, 33–42 (2005)

[10] Hunt, M. L., and Tien, C. L, Effects of Thermal Dispersion on Forced Convection in Fibrous Media, *Int. J. Heat Mass Transfer* 31 (1988a) pp. 301-310.

[11] Lundgren, T.S.: Slow flow through stationary random beds and suspensions of spheres. J. Fluid Mech. 51(1972) 273–299.

[12] A. Hadim, Forced convection in a porous channel with localized heat sources, ASME J. Heat Transfer 8 (1994) 465–472.

[13] Yong Zeng , Kambiz Vafai, An investigation of convective cooling of an array of channel-mounted obstacles, Numerical Heat Transfer, Part A, 55 (2009) 967–982.

[14] S.W. Patankar, Numerical Heat Transfer and Fluid Flow, McGraw-Hill, New York, 1980

[15] G. Hetsroni , M. Gurevich, R. Rozenblit, Sintered porous medium heat sink for cooling of high-power mini-devices, Int. J. Heat and Fluid Flow 27 (2006) 259–266.

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