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Technical Note

Analytical solutions of force convective heat transfer in plate heat exchangers partially filled with metal foams

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ABSTRACT

This paper investigated the flow and thermal characterization of parallel-plate heat exchangers partially filled with metal foams analytically. Firstly, the velocity and temperature distributions in partially filled symmetrical channel have been plotted and the effect of key parameters (e.g. porosity, pore density, height of metal foams, flow ratio in metal-foam filled region) on its flow and heat transfer performance have been analyzed. Then, the results have been applied for combined heat exchangers with symmetrical and asymmetrical channels to have optimized heat transfer performance and relatively low and identical pressure drop for each channel.

1. Introduction

Open-cell metal foam has outstanding surface volume density (790–2740 m^2/m^3), high thermal conductivity and high porosity ($\varepsilon > 0.85$), which improves the ratio of the heat transfer surface area to the volume, therefore, it can be used to improve the performance of heat exchanger. Open cell metal foam, which is a kind of porous media, has been widely studied using experimental methods, numerical simulations and analytical solutions. Zhao [1] investigated the thermal conductance in open cell cellular metal foams and indicated that metal foams provide better heat transfer compared with other mechanism at various conditions. Dukhan and Chen [2] studied the forced convection heat transfer in completely filled metal foam plate using deduction analysis methods. Abadi et al. [3] carried out experiments for a plate heat exchanger with metal-foam filled channel to investigate the heat transfer and pressure drop.

For theoretical studies of the convective heat transfer in metal foams, the Brinkman-extended Darcy model is commonly used for momentum equation. Dukhan et al. [4] measured the fluid temperature inside metal foam and compared that with the numerically studies with Brinkman–Darcy model for small velocity (e.g. less than 0.3 m/s). The predictions from Brinkman–Darcy model is well agreed with experiment measurement. When $Re_K < 10$, Brinkman-extended Darcy model is accurate for flow characteristics predictions [5]. Therefore, the Brinkman-extended Darcy model has been adopted as momentum equation in this paper. For energy equation of heat transfer modelling in porous medium, Lee and Vafai [6] indicated that temperature difference between local fluid and solid cannot be ignored in metal foam filled heat exchanger, otherwise, the results are not reliable. Therefore, the two-equation non-equilibrium heat transfer model are introduced in this paper to analyze the heat transfer in metal-foam region to obtain more accurate results.

When metal foam is used to enhance heat transfer in a channel, it can be fully filled [7] or partially filled [8,9]. The fully filled channel has the best heat transfer performance, but the pressure drop is significant. Meanwhile, multi-channel heat exchanger with the fully filled metal foam is required to be produced as a whole, which is difficult to make, modify and customize. The partially filled heat exchangercan be made in parts and have lower pressure loss. Xu et al. [10] studied the heat transfer in a counter-flow double-pipe heat exchanger filled with metal foams. They found that decreasing porosity or increasing pore density could improve the heat transfer performance. For plate channel, which is commonly used for compact heat exchangers, Mahmoudi and Maerefat [11] studied the plate channel with a layer of metal-foam in the middle without attachment to heated wall. The results show the heat transfer can be enhanced because the fluid velocity near the heated wall is higher with blockage of metal foams in the middle of channel, but the metal-foam needs to be fixed and maintained in the position. To solve this problem, the plate channel with metal-foam layers attached to both side of plates has been analyzed in this paper,

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because this kind of plate can be easily made and arranged to get customized heat exchangers. When several metal-foam enhanced plates have been put together to make heat exchanger, the outermost plates do not require heat transported through it, so they have insulations without metal foams, and the geometry of this kind of heat exchanger is presented in Fig. 1(a). As the channels in the middle are the same as each other, they can be represented by one channel while the heat transfer coefficients and pressure drop of inner channels equal to those of the outermost channels. Then, the heat exchanger in Fig. 1(a) can be simplified as the heat exchanger in Fig. 1(b), which is combined of a symmetrical filled channel (inner channel) and two asymmetrical filled channels (outer channels). The asymmetrical filled channels has been studied in previous work [12]. In this paper, symmetrical filled channel is investigated with the theoretical model based on the Brinkman-extend Darcy momentum equation and Non-equilibrium energy equations. The analytical solutions are deduced to predict the flow and the heat transfer characterizations of symmetrical filled channel (as shown in Fig. 1(c)) and eventually the proposed heat exchanger.

2. Physical problem and solutions

2.1. The physical problem

To simplify the problem, general assumptions for theoretical analysis have been applied, such as that the flow through the channel is fully developed laminar flow, all the thermo-physical properties of solid and fluid are temperature independent, the natural convection and radiation are neglected, and the porous medium is assumed to be homogeneous and isentropic. For the simplified heat exchanger shown in Fig. 1(b), the *Re* numbers for both sides are set to make the contraflow fluids having the same temperature gradient along z direction and the wall with heat transfer through it can be regarded as uniformly heated boundary. Thus, the heat exchanger can be considered as one dimensional problem containing two kinds of channels (symmetrical filled inner channel and asymmetrical filled outer channel (studied by author's previous work [12])) with uniformly heated boundary. Then, the symmetrical filled channel, which has metal foam attached to both walls of the channel as shown in Fig. 1(c), is the channel going to be studied in this paper to eventually study the heat exchanger shown in Fig. 1(a). Half of the channel has been taken and analyzed, because the channel is symmetric.

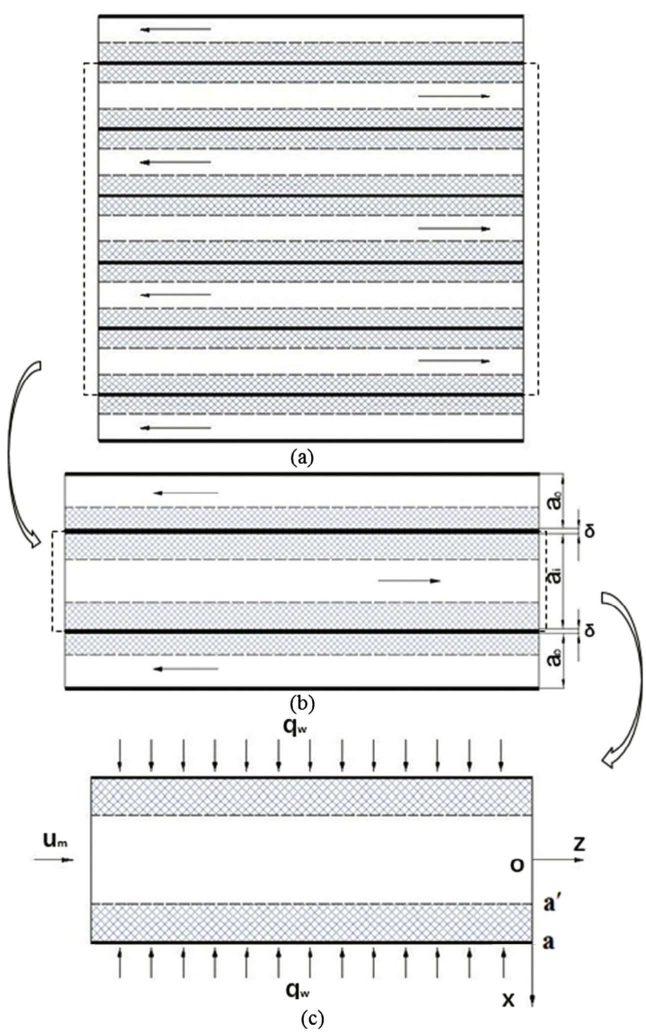

Fig. 1. Schematic drawing of (a) multi-channel heat exchanger (b) simplified heat exchanger (c) the symmetrical channel.

2.2. Governing equations and boundary conditions

Comparing with the asymmetrical channel [12], the symmetrical channel is governed by the same momentum and heat transfer equations, but different boundary conditions make the problem unique and lead to different solutions. To simplify the form of equations and solutions, the following dimensionless variables are introduced to normalise the governing equations and boundary conditions.

$$X = \frac{x}{a}, \quad U = \frac{u}{u_m}, \quad \theta_s = \frac{(T_s - T_w)k_{se}}{q_w a}, \quad \theta_f = \frac{(T_f - T_w)k_{se}}{q_w a},$$
$$H = \frac{a'}{a}, \quad Da = \frac{K}{a^2}, \quad P = \frac{K}{\mu_f u_m}\cdot\frac{dp}{dz}, \quad A = \frac{h'_{sf}a}{k_s}, \quad B = \frac{k_f}{k_{se}},$$
$$C = \frac{k_{fe}}{k_{se}}, \quad D = \frac{h_{sf}a_{sf}a^2}{k_{se}}, \quad S = \sqrt{\frac{\varepsilon}{Da}}, \quad R = \sqrt{\frac{D(C+1)}{C}} \tag{1}$$

2.2.1. Governing equations for foam free part (0 < X < H)

$$\frac{\partial^2 U}{\partial X^2} - \frac{P}{Da} = 0 \tag{2}$$

$$B\frac{\partial^2 \theta_f}{\partial X^2} = U \tag{3}$$

2.2.2. Governing equations for metal-foam filled part (H < X < 1)

The Brinkman-extended Darcy mode and two-equation non-equilibrium heat transfer model were employed for the momentum equation and energy equations.

$$\frac{\partial^2 U}{\partial X^2} - S^2 U = S^2 P \tag{4}$$

$$\frac{\partial^2 \theta_s}{\partial X^2} - D(\theta_s - \theta_f) = 0 \tag{5}$$

$$C\frac{\partial^2 \theta_f}{\partial X^2} + D(\theta_s - \theta_f) = U \tag{6}$$

2.2.3. Boundary conditions

At the centre of the channel (X = 0),

$$\frac{\partial U}{\partial X} = 0, \quad \frac{\partial \theta_f}{\partial X} = 0 \tag{7}$$