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Analytical solutions for heat transfer efficiency in metallic honeycombs using two-equation method☆

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ABSTRACT

An analytical model based on corrugated wall using two-equation method is proposed to evaluate the heat transfer performance in the sandwiched metallic honeycomb heat exchanger under forced convection conditions. The local thermal non-equilibrium between cooling fluid and solid honeycombs is taken into account compared with corrugated wall model, effective medium model, and transfer matrix model. A general analytical solution carried out using introduced effective specific surface area to overcome the fin efficiency underestimation and corrugated wall overestimation shows extensive applicability in predicting the thermal performance index, which can be restated to represent corrugated wall model, transfer matrix model, and the effective medium model, respectively.

1. Introduction

For decades, superior mechanical, heat transfer, and other physical properties and recent advances in low-cost processing have enabled the wide use of two-dimensional cellular metallic prismatic honeycombs for the heat transfer enhancement applications where high heat transfer performance at minimum weight is required [1]. Continuous channels of sandwiched cellular metallic prismatic structure with a single "easy flow" direction allow internal fluid transport, enabling simultaneous active cooling [2,3].

To optimize the cooling performance of a two-dimensional cellular structure under forced convection conditions, three analytical models are primarily used to describe characteristics of heat transfer. The first one is corrugated wall model [4] (also called the modified fin analogy model [2]). The second one is the effective medium model [5], which is based on one-equation method assuming local thermal equilibrium (LTE) of fluid and solid phases. The third one is transfer matrix model [6]. The corrugated wall method can model the detailed cellular honeycomb structure as a corrugated wall with many fins. The effective medium model uses volume averaging technique and the specific surface area used in the governing equations counts the heat transfer enhancement from the corrugated walls and fins simultaneously. The transfer matrix method can model the heat flux transfer along a slice of honeycomb both in the corrugated walls and fins simultaneously base on the energy conservation. For the three aforementioned analytical models, the fluid temperature field distribution is all assumed to be uniform. Namely, local thermal equilibrium (LTE) of fluid and solid phases is assumed. However, the fluid temperature near the bottom and top heated substrate and the solid honeycomb is much higher, which will lead to significantly different convective heat transfer efficiency between the solid honeycomb wall and the cooling fluid at different transverse position. Therefore, the assumption of having local thermal equilibrium is questionable. Local thermal nonequilibrium (LTNE) model is more accurate than the LTE model, as indicated in Lee and Vafai [7]. The LTNE model has been adopted by many researchers to represent the fluid-solid energy exchange [8–10] in porous media, which suggests that the LTNE model is required.

In the present study, the transverse fluid temperature gradient is taken into account using the two-equation method. A two-equation corrugated wall model is proposed to analyze the heat transfer. A general analytical solution for the thermal performance index is established, so that the available analytical models can be represented by a special value of the introduced effective specific surface area. The analytical predictions are compared with the three conventional analytical solutions.

2. The physical problem

The prototypical heat exchanger filled with a metallic honeycomb sandwiched between two flat rectangular substrates is shown in

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Nomenclature

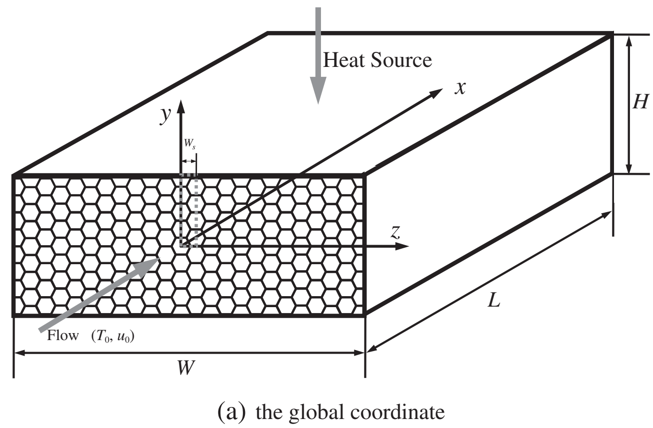
a	Cell size, mm
c_f	Specific heat capacity of fluid, J·kg^{-1}·K^{-1}
h	Local heat transfer coefficient, W·m^{-2}·K^{-1}
$\overline{h}$	Overall heat transfer coefficient, W·m^{-2}·K^{-1}
H,L,W	Thickness, length and width of sandwiched heat exchanger, mm
I	Dimensionless scaling index
k_s, k_f	Thermal conductivity of solid and fluid, W·m^{-1}·K^{-1}
l	Cell wall length, mm
$\dot{m}$	Mass flow rate, kg·s^{-1}
$n, \overline{n}$	Fin efficiency index and its average
Nu	Nusselt number
Δp	pressure drop, Pa
q	Heat flux, W·m^{-2}
t	Cell wall thickness, mm
$T, \overline{T}$	Temperature and its average, K
u	Velocity, m·s^{-1}
(x,y,z)	Global Cartesian coordinates

Greek symbols

α_e	Effective specific surface area [m^{-1}]
ε	Porosity
μ	Dynamic viscosity, kg·m^{-1}·s^{-1}
ν	Kinematic viscosity, m^2·s^{-1}
ρ_f	Density of fluid, kg·m^{-3}
ρ	Relative density of the honeycomb structure

Subscripts

e	Effective
f	Fluid
s	Solid
w	Substrate wall
0	Inlet



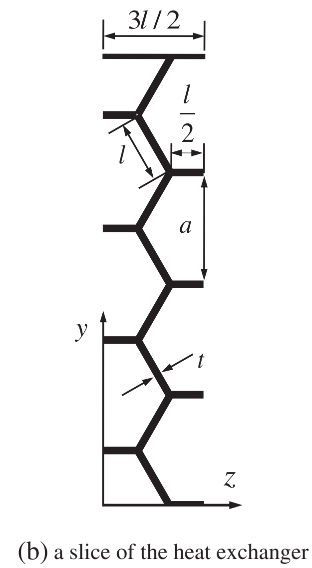


Fig. 1. A prototypical design of a heat exchanger cooled by forced convection through metallic honeycomb: (a) the global coordinate, (b) a slice of the heat exchanger.

Fig. 1(a). The cooling fluid is forced to flow across the sandwiched honeycomb of thickness H at the inlet ($x=0$) and exits at the outlet ($x=L$). The module is thermally insulated on the left and right sides ($y=\pm W/2$). The prototypical heat exchanger is divided into periodic slices of equal width, and only half of the slice is studied due to the symmetry, as illustrated in Fig. 1(b). The regular hexagonal cells of cell wall length l and thickness t (Fig. 1(b)) is considered. For simplicity, the length of the channel is assumed sufficiently long such that the problem can be considered as two dimensional, and the other assumptions upon which the theoretical model is based are summarized as follows:

(1) The thermal–physical properties of fluid and solid are assumed to be constant.
(2) The substrates are assumed to be thin and have large thermal conductivity.
(3) The width of the channel, W, is assumed to be much larger than the cell size so that the thermal and hydraulic fields are independent of the coordinate z.
(4) Flow and heat transfer are steady and fully developed.

The heat transfer analysis based on two-equation method is accomplished in two steps. Firstly, the analysis of heat transfer is performed for the corrugated walls by excluding the effect of fins. Then, the heat loss contributed from the fins is added according to the energy balance. Micro topology and volume averaging techniques are used to establish the governing equations of the solid and fluid, respectively.

3. Micro topology technique

3.1. Two-equation method

With reference to Fig. 1(b), the governing energy equation for the solid phase can be established by an energy balance analysis, as

$$k_s t_s \frac{\partial^2 T_s(x,y)}{\partial y^2} - \frac{2h}{\sin(\pi/3)}\left(T_s(x,y) - T_f(x,y)\right) = 0 \tag{1}$$

where k_s is the thermal conductivity of the solid of which the cell wall is made, $t_s = 2t/\sqrt{3}$ is the transverse cell wall thickness, $T_s(x,y)$ and $T_f(x,y)$ are the solid and fluid temperatures at the location with coordinate (x,y) respectively, h is the local heat transfer coefficient and can be denoted by following expression for regular hexagonal structures:

$$h = \frac{Nu \cdot k_f}{\sqrt{3}l\sqrt{1-\rho}} \tag{2}$$

where Nu is Nusselt number, k_f is the thermal conductivity of cooling fluid, ρ is the relative density of the honeycomb structure [5]. According to the study of Lu [4] and Shutian Liu [6], the resulting expression of the overall heat transfer coefficient $\overline{h}$ is identical for the two classic types of heat transfer boundary conditions. The only difference is that the Nu is 3.35 for the isothermal surface and is 4.021 for the constant heat flux surface. This conclusion is still stood in this study.