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Development of an advanced printed circuit heat exchanger analysis code for realistic flow path configurations near header regions

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ABSTRACT

An advanced numerical method for printed circuit heat exchanger (PCHE) analysis and design is developed. It takes into account realistic flow path configurations for cross and parallel flow regions near header locations. Consideration of such header effects is found to be important in evaluating PCHE effectiveness particularly when PCHEs operate with relatively low effectiveness with a considerable heat transfer contribution from the cross and the parallel flow regions. The importance of modeling the header effects manifests even further when pressure drop is considered. Flow distributions subject to flow path configurations leads to a significant pressure drop increase, which is neglected in the widely used sole-counter flow approximation. The developed code can be served to advance the current modeling and design capabilities of PCHEs.

1. Introduction

To achieve effective heat transfer, high pressure gas-cooled systems such as gas-cooled nuclear reactors need a large surface area to compensate for the relatively low heat transferring capacity of gaseous fluids. Consequently, a large heat transfer area in the heat exchangers is critically important to the performance of a high pressure gas-cooled system. Among many heat exchanger candidates, the printed circuit heat exchanger (PCHE) is considered promising because it provides an exceptionally large surface area in a compact component volume [1,2]. The high compactness of the PCHE is rooted in its manufacturing procedure: the PCHE is assembled by stacking multiple etched plates, followed by a diffusion bonding process. As a consequence of the manufacturing process, narrowly spaced semi-circular micro channels (on the order of 10^{-3} m) are formed, which serve as fluid channels. However, while the PCHE provides an exceptionally large surface area for heat transferring fluids, it requires high pumping power and high manufacturing cost.

Previous PCHE design methodologies reported in the literature have generally considered its flow path configurations as either sole counter flow, cross flow, or parallel flow [3,4]. Consequently, the current analytical capabilities of PCHE thermal performance modeling are limited to these simple flow configurations. Yet, depending on header locations, actual PCHE structures consist of various arrangements of cold and hot channels, resulting in regions having counter, cross, and parallel flow in a single component. Fig. 1 illustrates the different hot and cold channel configurations for different header locations.

Although simplified treatments of PCHE flow configurations may provide a reasonable first-hand estimation of PCHE thermal performance, advanced system design requires a more accurate thermal performance prediction with respect to PCHE header locations. It is important to note that header locations affect PCHE arrangements in a system, as well as associated pipe lines. That is, to facilitate PCHE arrangements in a system, certain header locations are preferred or even required. Therefore, developing an advanced PCHE performance analysis method that captures various flow path configurations would achieve optimal system designs as well as accurate thermal hydraulic performance predictions.

In this research, we (1) developed an advanced PCHE analysis code that gives PCHE thermal performance by capturing actual flow configurations with respect to different header locations, and (2) discuss the design implications of flow path dependent PCHE thermal performance.

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Nomenclature

Latin letters

A	area (m^2)
$\dot{m}$	mass flow rate (kg/s)
C_p	specific heat capacity (kJ/kg K)
T	temperature (K)
Q	heat (W)
U	overall heat transfer coefficient (W/m^2 K)
h	convection heat transfer coefficient (W/m^2 K)
R_w	thermal resistance of wall (K/W)
P	pressure
P_{fric}	pressure drop by friction
P_{form}	pressure drop by form loss
D_h	channel's hydraulic diameter (m)
f	frictional pressure drop factors
K	form-loss factor
v	velocity
k	constant multiplier
Nu	Nusselt number
L	length
ε	effectiveness

Greek letters

ρ	fluids density

Subscripts

i	fluid side (1 or 2)
1	Fluid 1 (hot fluid)
2	Fluid 2 (cold fluid)
j	integer
No.	number
in	inlet position
out	outlet position
c	cross-section

2. Development of PCHE analysis code: governing equations

Numerical methods and algorithms were used to get the temperature and pressure profiles of PCHEs that had the eight reference flow configurations shown in Fig. 1. The code developed in this study numerically solves for mass, momentum, and energy balance models for given flow path arrangements, consisting of distinct cross, counter, and parallel flow region. For each distinct flow region, we used previously developed energy balance models.

2.1. Energy equations

2.1.1. Energy balance equations for cross flow channels

Nusselt [5] developed an analytical solution for single-pass cross flow heat exchanger with unmixed fluids. In the study, Nusselt assumed negligible longitudinal conduction. Yoon et al. [4] employed Nusselt's energy balance models for cross flow heat exchanger, shown in Fig. 2.

For each hot and cold channel in Fig. 2, the following energy equations hold.

Fluid 1

$$d\dot{m}_1 C_{p,1} T_1 - d\dot{m}_1 C_{p,1}\left[T_1 + \frac{\partial T_1}{\partial x}\right]dx - dQ = 0 \tag{1}$$

Fluid 2

$$d\dot{m}_2 C_{p,2} T_2 - d\dot{m}_2 C_{p,2}\left[T_2 + \frac{\partial T_2}{\partial y}\right]dy + dQ = 0 \tag{2}$$

The term dQ – the heat transfer rate between the hot and cold fluid across the finite surface area and dA – can be expressed by the overall heat transfer U as follows [4,6,7]:

$$dQ(x,y) = U(x,y)dA(T_1(x,y) - T_2(x,y)) \tag{3}$$

The overall heat transfer U is given as:

$$\frac{1}{UA} = \frac{1}{h_1A_1} + \frac{1}{h_2A_2} + R_w \tag{4}$$

where h is the heat transfer coefficient, A is the surface area, and R_w is the thermal resistance of wall. Eqs. (1) and (2) can be arranged to obtain following temperature fields using the Euler method.

$$T_1(x+\Delta x, y) = T_1(x,y) - \frac{dQ(x,y)}{d\dot{m}_1 C_{p,1}}\Delta x \tag{5}$$

$$T_2(x, y+\Delta y) = T_2(x,y) + \frac{dQ(x,y)}{d\dot{m}_2 C_{p,2}}\Delta y \tag{6}$$

The analytic solutions were obtained by Nusselt [5,8] with the inverse Laplace transform by assuming fluid properties as constant. Also, numbers of cross flow heat exchanger models have been obtained [9–17]. However, this assumption gives unsatisfactory prediction when the fluid properties change dynamically with temperature and pressure. For instance, the supercritical CO_2 brayton cycle near by the critical point cannot be accurately modeled if its properties are assumed to be constant – it undergoes dynamic property changes near the critical point [27]. In this research, we solve the presented governing equations numerically without assuming any fluid properties as constants.

2.1.2. Energy balance equations for counter and parallel flow channels

In a similar way, the counter flow section and parallel flow section[3] can be solved by the Euler method. The analytic model of the cross flow section of the PCHE has been developed using the energy balance between the hot and cold fluid side shown in Fig. 3 [18].

Using the Euler method, following energy equations hold.

Fluid 1

For both counter and parallel flow:

$$T_1(x+\Delta x) = T_1(x) - \frac{dQ(x)}{d\dot{m}_1 C_{p,1}}\Delta x \tag{7}$$

Fluid 2

Counter flow:

$$T_2(x+\Delta x) = T_2(x) - \frac{dQ(x)}{d\dot{m}_2 C_{p,2}}\Delta x \tag{8}$$

Parallel flow:

$$T_2(x+\Delta x) = T_2(x) + \frac{dQ(x)}{d\dot{m}_2 C_{p,2}}\Delta x \tag{9}$$

[3] Parallel flow configuration is relevant in cases (g) and (h) in Fig. 1, only.