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Perforated plate convective heat transfer analysis

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ABSTRACT

The need for high performance heat exchangers has led to the development of many types of surfaces that enhance the rate of heat transfer. One of the heat exchanger types are the matrix heat exchangers. Matrix heat exchangers consist of a series of perforated plates, separated by a series of spacers. The present study experimentally investigates the overall heat transfer characteristics of flow through a perforated plate with 2 mm in diameter, the hole length to diameter ratio of 1 and 25.6% porosity. Numerical simulations were performed to determine the overall heat transfer in the function of geometric parameters. Reynolds numbers based on the perforated plate pitch were set in the range from 50 to 500. The results of the average Nusselt number prediction were compared with the related experimental correlations. The experimental data agreed on qualitatively with the results obtained using a CFD. Using these data, a Nusselt criterial equation was obtained.

1. Introduction

One of the most important properties of heat exchangers, apart from having high effectiveness, is the need to be very compact, *i.e.* they must accommodate a large surface to volume ratio. This helps in controlling the heat exchanger exposure to the surroundings by reducing the exposed surface area. A small mass also means a lower heat inertia. This requirement is particularly important for small refrigerators operating at liquid helium temperature. The need to attain high effectiveness and a high level of compactness together in one unit has led to the invention of matrix heat exchangers (MHE) by McMation et al. [1]. A matrix heat exchanger consists of a package of perforated plates with a multitude of flow passages aligned in the direction of flow, allowing high heat transfer in a properly designed unit. This exchanger can have up to 6000 m^2/m^3 surfaces to volume ratio [2,3]. The MHE scheme is presented in Fig. 1.

The convective heat transfer characteristics of any heat exchanger surface can be determined by using steady state, periodic test, and transient test techniques [3]. For a steady-state method, the temperatures of hot and cold fluids entering and leaving the heat exchanger, as well as the flow rates are measured. When the steady state is achieved, it is possible to determine the heat flux, thus the overall heat transfer coefficient. In the transient technique method, after the steady state is reached, the temperature of the fluid entering the heat exchanger is suddenly changed. The heat transfer coefficient can be determined from temperature-time history data. The periodic test techniques represent a variation of the transient method in which the temperature of the fluid entering the heat exchanger is continuously varied.

In 1966, an extensive experimental study of convective heat transfer and flow friction based on transient technique was published for eight different perforated surfaces [4]. G. Venkatarathnam and Ragab M. Moheisen gave a good literature review of MHE, their constructions, and Nusselt criteria [3,5,6]. The MHE were used mainly as helium liquefieres and low power cryorefrigerators based on Claude and reverse Brayton cycles [6], but also perforated plates could be used as fins for improving cooling effect of the electronic equipment [7]. The goal of this paper was to investigate thermal and fluid flow processes on the air side of an air/water perforated plate heat exchanger at ambiental conditions. The experimental research was conducted over a single perforated plate with a porosity of 25.6%, while a numerical experiment was done in the range of porosity between 10 and 50%.

2. Literature review

The heat transfer improvement may be achieved by increasing the heat transfer coefficient, heat transfer surface areas, or both. In papers [8–11] the authors conclude that for certain values of perforation dimension, a perforated plate enhances heat transfer in comparison to the

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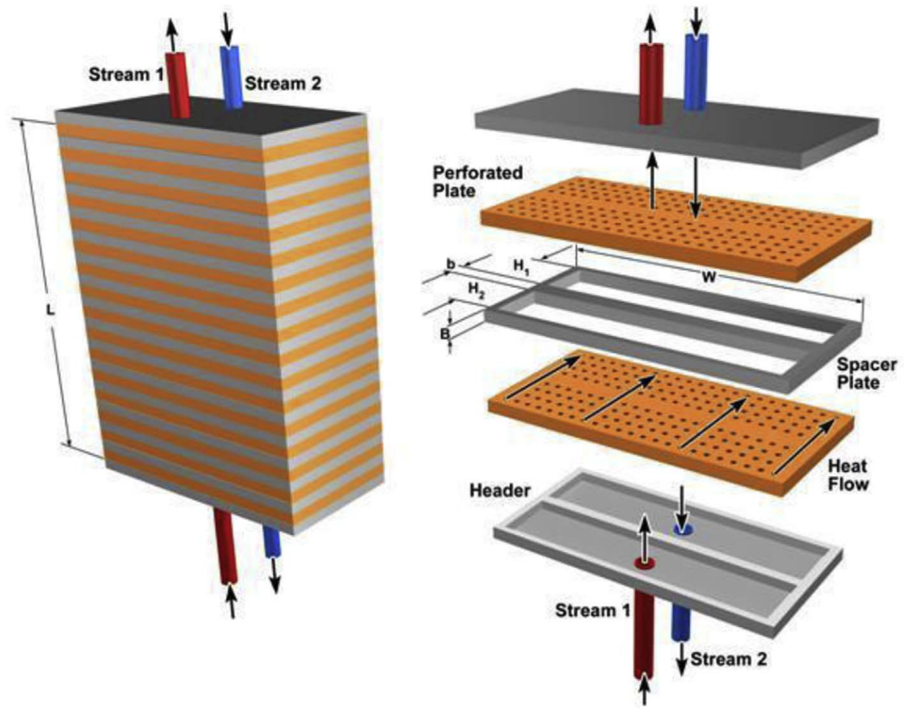

Fig. 1. Matrix heat exchanger schema [5].

solid one. Perforated plate convective heat transfer takes place on three surfaces:

- The front area surface,
- The tubular surface of perforations and
- The back surface of the plate.

The flow through the tubular section can be considered as a developing flow with a very high heat transfer coefficient. The heat transfer coefficient in the developing flow is well studied, and it can be calculated in the function of the Pecklet number [12]:

$$\alpha = \zeta 0.0465 Pe^{0.75} \frac{\lambda}{d} \tag{1}$$

where ζ is the function of pipe length L and the length of the pipe needed for the developed flow L\`. The length L′ is equal to

$$L' = 0.015 Pe \cdot d \tag{2}$$

Value Pe represents the Pecklet number, the product of the Reynolds and Prandtl number, and d is the pipe diameter. The value of ζ is presented in Table 1.

Linghui et al. [13] studied the flow through hexagonally arranged perforations on the plate. The purpose of the research was to determine how the length to diameter ratio (δ/d) of the plate's holes affected the heat transfer coefficient. They studied the ratios varying from 0.333 to 1.1666, holding the diameter constant while varying the thickness. Their experiments used the naphthalene sublimation technique to determine the heat transfer of the plate. The research led to the conclusion that there was little change in the heat transfer coefficients between the (δ/d) ratios of 0.5 and 1.1. The final equation for the Nusselt number inside the tube was

$$Nu = 2.058\,\mathrm{Re}^{0.487} \tag{3}$$

Table 1
The value of ζ in the function of developing length.

L/L'	0	0.01	0.05	0.1	0.2	0.4	0.6	0.8	1.0	∞
ζ	∞	1.26	1.16	1.12	1.08	1.05	1.03	1.01	1.00	1.00

The heat transfer from the front face of the plate was studied by Sparrow and Ortiz [14]. In their experiments the Reynolds number and the hole's pitch to diameter ratio was varied per hole. The suggested Nusselt criteria was in the function of the Reynolds and Prandtl number, but the characteristic length in the Nusselt criteria was the ratio of the module surface area of the pitch for $2000 < \mathrm{Re} < 20000$

$$Nu = 0.881\,\mathrm{Re}^{0.476}\,\mathrm{Pr}^{1/3}, \tag{4}$$

and

$$Nu = \frac{\alpha A}{\lambda p}. \tag{5}$$

The result was established for the limited case, where the relative spacing was $2 < p/d < 2.5$.

Dorignac et al. [15] conducted a series of experiments of air flow leading to the result for the Reynolds number of 1000–1200

$$Nu = 1.202\left(\frac{p}{A^{0.5}}\right)^{1.879}\left(\frac{p}{d}\right)^{0.163}\mathrm{Re}^{0.409} \tag{6}$$

where p is the pitch length and A is the area.

Heat transfer rate in the back face of the last plate was high due to flow separation and resulting turbulence [16]. Brunger et al. [17] studied the effectiveness for each of the three zones of heat transfer on a perforated plate: the front of the plate, the inside of the tube and the back of the plate. In their study, they considered large pitch to diameter ratios (> 6.67). For each of the heat transfer regions, an equation for effectiveness was given. The authors also stated that under typical operating conditions, about 62% of the ultimate temperature rise of the air was predicted to occur on the front surface, 28% in the hole, and 10% on the back of the plate.

An average heat transfer for the perforated plate may be defined as

$$\alpha = \frac{\sum_{i=1}^{n} \alpha_i A_i}{\sum_{i=1}^{n} A_i}. \tag{7}$$

The Nusselt number correlation as a function of the Reynolds number, Prandtl number and geometry factors apply to higher Reynolds numbers and lower plate porosities. The majority of authors derive empirical correlations for the Nusselt number and the friction factor