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SINGLE-PHASE PRESSURE DROP AND HEAT TRANSFER IN MICRO-PIN FIN HEAT EXCHANGERS

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1. ABSTRACT

Single-phase flow of HFE-7100 in a micro-pin fin heat sink was investigated and the results described in this paper. Both adiabatic and diabatic experiments were carried out at a system pressure of 1 bar, inlet fluid temperature of 19 ^oC and Reynolds number ranging from 86 to 850. Different existing correlations of friction factor and Nusselt number were evaluated and assessed. A good agreement between the present results and some correlations was found.

2. INTRODUCTION

Micro-pin fin heat sinks have recently received a significant attention by industrial and scientific communities. As described in the published literature, different fin geometries were designed and tested, including circular, square, diamond, triangle, teardrop, piranha, hexagon and elliptic, see [1]. Moreover, the effects of pin dimensions and arrangement, i.e. in-line or staggered arrays, were also studied. These parameters could influence the hydraulic and thermal performance leading to high heat transfer enhancement due to the promotion of fluid mixing. However, more investigations using dielectric and eco-friendly fluids are still needed in order to establish design correlations predicting pressure drop and heat transfer in order to promote the use these geometries in, for example, electronics cooling. Acceptable design correlations should also be proposed for different working fluids, geometries and operating conditions. The present study aims to evaluate some existing correlations using single-phase flow of HFE-7100 in staggered diamond micro-pin fins.

3. METHDOLOGY

The experimental facility used in this study is depicted in Fig. 1(a). The construction of the test section is presented in Fig. 1(b). Staggered diamond micro-pin fins were manufactured using a high precision micro-milling machine with a total base area of 20 x 25 mm. Fig. 1(c) shows the details of the pins, having pin height, H_{pin}, of 1 mm, a square cross section, W_{pin}, of 0.6 mm width, 0.95 mm pin space. The number of pins on the base area was 207. The heat transfer area, including the pin efficency, η_{pin}, compared to the base area (without fins) was 1.94.

$$A_{ht} = A_b - N_{pin}W_{pin}^2 + N_{pin}\eta_{pin}H_{pin}4W_{pin} + 2H_{pin}L_b \quad (1)$$

where A_b, A_{ht}, N_{pin} and L_b are base area, total heat transfer area, number of pins and base length, respectively. In our case, A_{ht}= 0.00097 m^2. The pin efficiency was found for a straight fin of uniform cross-section area with an adiabatic tip, see [2].

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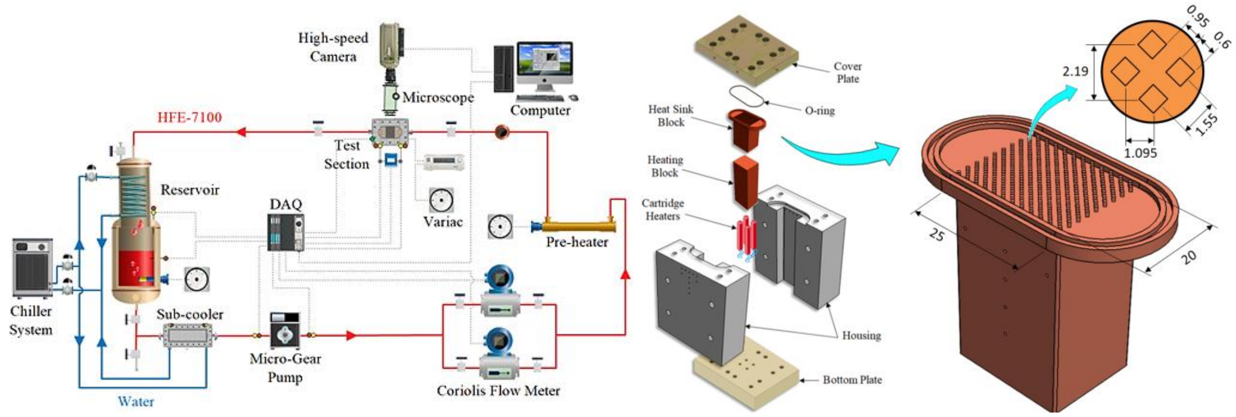

(a) (b) (c)

Fig. 1 Schematic diagram of (a) Experimental rig, [3] (b) Test section (c) Heat sink (dimensions in mm).

Both adiabatic and diabatic experiments were carried out by changing mass flow rate in steps. In the diabatic experiments, the base heat flux was kept constant at 117 kW/m^2. All the data were recorded for two minutes after reaching the steady state condition. Steady state was considered when the change in the measured signals was 5%. For comparison purposes, the Fanning friction factor could be calculated using Eq. (2), originally based on the work presented in [4]. This allows comparisons with recent publications in micro-pin fin geometries [5-7] who adopted this equation. An alternative approach was proposed in [8] for an in-line micro-pin fin heat sink.

$$fr = \frac{\Delta P_{pin} \rho_l}{2 G_{max}^2 N} \tag{2}$$

where ΔP_{pin}, ρ_l, G_{max} and N are pin pressure drop, liquid density, maximum mass flux and number of pin rows in the direction of the flow, respectively. The maximum mass flux was obtained by dividing the inlet mass flow rate by the minimum area between pins, see Eq. (3). Where W_b and S_T are the base width (20 mm) and the transverse pitch (2.19 mm), respectively.

$$A_{min} = W_b H_{pin} \left[1 - \frac{W_{pin}}{S_T}\right] \tag{3}$$

The pin section pressure drop was found by subtracting the sudden contraction and expansion pressure drop components from the measured pressure drop. The local heat transfer coefficient was calculated from Eq. (4), then the average heat transfer coefficient and the average Nusselt number were found from Eq. (5-6).

$$h_{(z)} = \frac{q"_b A_b}{A_{ht}(T_{w(z)} - T_{l(z)})} \tag{4}$$

$$\bar{h} = \frac{1}{L_b} \int_0^{L_b} h_{(z)} dz \tag{5}$$

$$Nu = \frac{\bar{h} W_{pin}}{k_l} \tag{6}$$

where $q"_b$, $T_{w(z)}$, $T_{l(z)}$ and k_l are base heat flux, local wall temperature, local fluid temperature and fluid thermal conductivity, respectively. The mean absolute error was found from Eq. (7). Where n, fr_{pred} and fr_{exp} are the number of data points, the predicted friction factor and the experimental friction factor, respectively.

$$MAE = \frac{1}{n} \sum \left| \frac{fr_{pred} - fr_{exp}}{fr_{exp}} \right| 100\% \tag{7}$$

The present experiments were conducted at a system pressure of 1 bar, inlet fluid temperature of 19 ^{o}C and Reynolds number of 86-850. The maximum uncertainty in the friction factor and average Nusselt number were found to be 5% and 12%, respectively.

4. RESULTS

The Fanning friction factor versus Reynolds number of the adiabatic results is plotted in Fig. 2(a) for Re number less than 800, i.e. laminar flow. The Fanning friction factor decreased with increasing Reynolds number within the experimental range. The results were compared with the correlations by [5-7]. It can be seen that the correlation by Xu and Wu [5] predicted the results well with a MAE of 7%. In contrast, the MAE of the correlations [6] and [7] was found to be 35% and 39%, respectively. This good agreement by [5] could be due to the fact that their correlation was proposed for staggered diamond pins, which is similar to our geometry. Konishi et al. [6] proposed their correlation for square pins, while Brunschwiler et al. [7] used circular pins. Fig. 2(b) shows the experimental average Nu number versus Re number of the diabatic process. Showing, as expected, an increase of the Nu number with Re number. Three correlations proposed by [5, 7, 9] were also assessed. It was found that the mean absolute error between the results and the correlations by [5], [9] and [7] was 13%, 26% and 61%, respectively. Again, this good agreement with the correlation proposed by [5] could be due to the same diamond pin geometry, while square or circular pins were tested by [7,9].

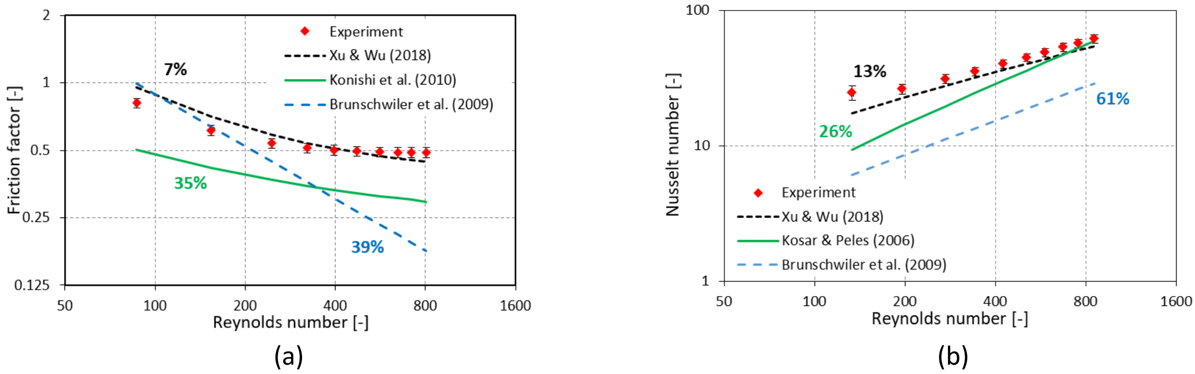

Fig. 2 Single-phase results: (a) Adiabatic process (b) Diabatic process.

The geometry of micro-pin fins could have a clear influence on the fluid mixing process. This could affect the hydrodynamic boundary layer (during adiabatic process) and the thermal boundary layer (during diabatic process). Accordingly, both velocity and temperature profiles could be affected leading to wall shear stress and different friction factor and local and average heat transfer coefficient and hence Nu number.

5. CONCLUSIONS

Single-phase flow experiments of HFE-7100 in micro-pin fin heat sinks were carried out. Adiabatic and diabatic processes were tested at 1 bar inlet pressure and 19 ^{o}C inlet fluid temperature during laminar flow. The experimental results were used to assess existing correlations. It was found that the correlations by [4] provided the best prediction. The geometry of micro-pin fins affects the single-phase results and hence resulting correlations. The results were compared with micro-channels and micro-gap (channel of same height and base area but without fins). The experiments parameters will be extended to include different heat fluxes and eventually flow boiling.

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