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OPEN CELL METAL FOAM HEAT EXCHANGERS FOR AIR-DEHUMIDIFICATION APPLICATIONS

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ABSTRACT

Open cell metal foams have emerged as promising porous materials for various thermal applications such as energy storage and electronics cooling. The physical and mechanical characteristics suggest significant advantages over conventional fin configurations for use in air-cooling heat exchangers. The larger surface area to volume ratio and repetitive cell structure introducing the boundary layer restart and thus enhancing the mixing have been identified as important attributes for such applications. An experimental study has been conducted to evaluate the performance of metal foam heat exchangers under dehumidifying conditions. Closed-loop wind tunnel experiments have been used to determine the pressure drop and heat transfer performance under wet-operating conditions where the parameters such as pore size and upstream face velocity have been varied to understand the potential impact of these parameters.

KEY WORDS: Metal foams, Heat exchangers, Dehumidification, Heat transfer coefficient, Pressure drop

1. INTRODUCTION

Metal foams have been considered for numerous thermal applications including cryogenics, combustion chambers, catalytic beds, compact heat exchangers for airborne equipment, air cooled condensers and compact heat sinks for power electronics. Although the development is in-progress to resolve some manufacturing and implementation issues, these materials have shown remarkable performance for heat exchangers and heat sinks [1]. Characteristics such as open porosity (Fig. 1), low relative density, high thermal conductivity of the cell edges, large accessible surface area per unit volume (Fig. 2) [2-7], and the ability to mix the cooling fluid are important for development of efficient, compact, and light-weight thermal management devices. Several studies have concluded that metal foams have relatively large pressure drop per unit length which is caused by complex geometry of the foams. However, the complex tetrakaidecahedron structure also results in high degree of boundary layer restarting and wake destruction by mixing [8-17]. Overall the increase in pressure drop is mitigated by the considerably higher heat transfer coefficients.

Dai *et al.* [16] compared the heat transfer and pressure drop performance of metal-foam heat exchangers to another state-of-the-art heat exchanger. In the analysis, two heat exchangers were subjected to identical performance requirements, and the resulting volumes, masses, and costs were compared. Metal foam heat exchangers were found to meet the thermal requirements at lower volume and mass, but at a higher cost. Nawaz *et al.* [17] considered open-cell aluminum metal foam as a highly compact replacement for conventional fins in heat exchangers. Heat transfer and pressure drop data for different PPI metal foam heat exchanger were evaluated by wind-tunnel experiments in order to make the comparison with the louver-fin heat exchangers.

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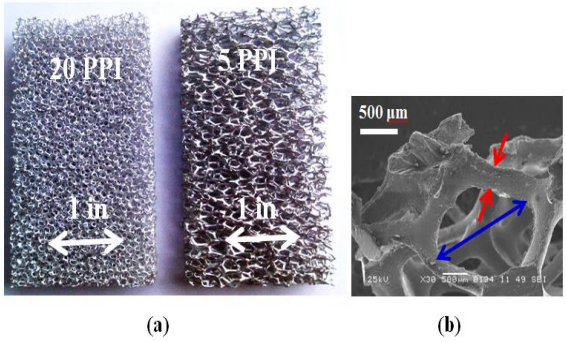

Fig. 1: Definition of length scale for metal foams (a) 20 and 5 PPI (b) Pore diameter (blue), strut diameter (red)

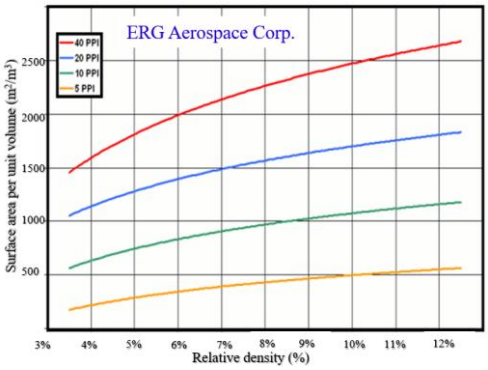

Fig. 2: Surface area per unit volume for different types of metal foams.

Although there are multiple studies on the evaluation of the performance of metal foams heat sinks and heat exchangers under dry operating conditions., there is rare information available about the performance when these devices operate in dehumidifying conditions. For the large-scale deployment of these novel materials, it important to determine their performance since condensate formation, retention and drainage are critical factor which are absent as they operate under dry conditions. In the current study, the thermal-hydraulic performance of metal foam heat exchangers has been evaluated in a closed-loop wind tunnel at varying air flow rates under wet operating conditions.

2. EXPERIMENTAL APPARATUS

A closed-loop wind tunnel apparatus was used to investigate the thermal-hydraulic performance of metal foam heat exchangers. As shown in the schematic of wind tunnel (Fig. 3), air downstream of test section passed through a set of electric strip heaters, past a steam injector, through an axial blower and another set of strip heaters, a flow nozzle, a mixing chamber, a flow conditioning section, a flow contraction, and the test section, completing the loop. Evenly spaced T-Type thermocouples were used both upstream (16-TC grid) and downstream (32-TC grid) of the test section to measure the air temperature both upstream and downstream. The chilled-mirror hygrometers were used to measure the upstream and downstream dew points. The pressure drops across the test sample and the flow nozzle were measured using pressure transducers. The cooling-fluid (ethylene glycol-water solution 40%-60% respectively) was used to run the system in cooling mode where the fluid was cooled using a chiller to achieve a constant inlet temperature. The cooling-fluid temperatures at the inlet and exit of the heat exchangers were measured using RTDs. All experiments were conducted under dehumidifying conditions and had an energy balance ($(Q_a-Q_c)/Q_{avg}$, ANSI/ASHRAE-33 [18]) within 5%. The exit and inlet temperature of air and cooling fluid were monitored to ensure that the heat exchanger is completely wet during the tests. The heat transfer rate on air-side and on the water-side was determined using the mass flow rate, specific heats for fluids and the relative temperature gradients at the inlet and the exit of the heat exchanger.

Metal foam heat exchangers had the exactly same geometry (face area, flow depth), however, the metal foam pore size changed (10, 20, 40 PPI). Fig. 4 presents a representative metal foam heat exchanger where the metal foam fins have been stacked between flat channels. The fluid flows through the channel while the air passes in a cross-flow formation through the metal foam fins. The empty spaces between the header and the fins surface is blocked to ensure that there is no bypass flow of air.

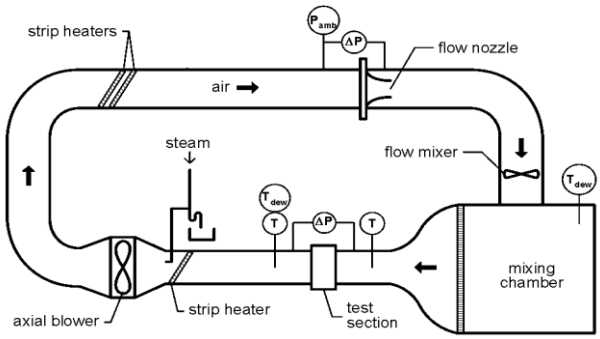

Fig. 3: A closed loop wind tunnel apparatus

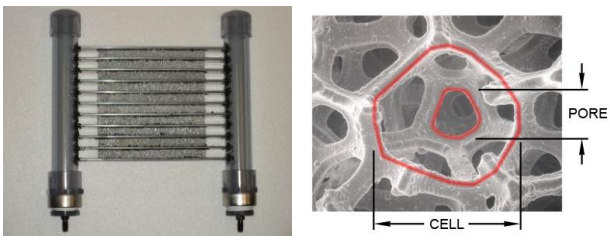

Fig. 4: Metal foam heat exchanger and geometry of a pore

3. RESULTS

The air-side pressure drop and heat transfer rate was determined from the experimental data at various face velocities. All these experiments were conducted under completely wet operating conditions i.e., the exit condition on the tube side was always less than the dew point temperature for the exiting air. Figures 5, 6 and 7 present the air-side pressure gradient (pressure drop per unit length), air-side latent and total heat transfer rate and the heat transfer coefficients respectively. It can be observed that pressure gradient increases exponentially with the face velocity as it happens in any channel flow situation. The pressure drop for a 40 PPI metal foam is larger compared to 20 PPI and 10 PPI metal foam samples which can be explained in terms of relative resistance to the flow based on the average pore diameter for different samples. A similar trend is observed for the air-side capacity where a 40 PPI metal foam samples shows a significantly higher performance compared to the 20 PPI and 10 PPI metal foam. A like-wise trend is observed when the performance of a 20 PPI metal foam samples is compared to a 10 PPI metal foam sample. Here the larger PPI value indicates relatively smaller pore size which leads to not only a relatively larger surface area per unit volume, but also, ensures better mixing of fluid; thus resulting in higher heat transfer coefficient (Fig. 7). The overall impact is higher heat transfer rate. The air-side capacity increases at relatively faster rate for lower face velocities; however, the increment is suppressed for higher face velocities. Although similar pattern can be observed for operation under completely dry conditions but for the dehumidifying case, the suppression is more obvious mainly due to the noticeably smaller increase in latent heat transfer as the face velocity increases. This can be attributed to the insufficient condensate drainage. The air-side heat transfer coefficient is calculated using an adiabatic boundary conditions. The adiabatic fin condition is used to determine the total air-side effectiveness. Following equations have been used to determine the air-side heat transfer coefficient.

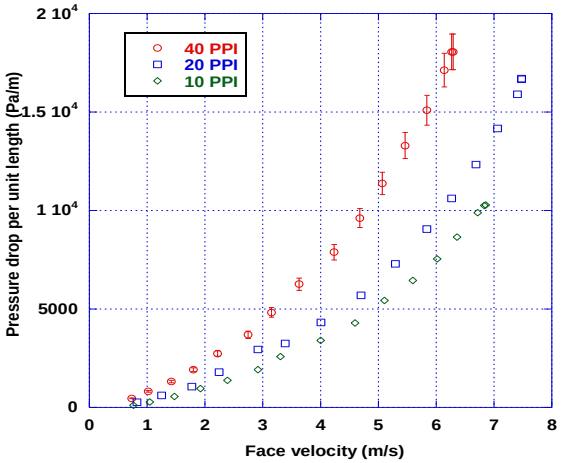

Fig. 5: Variation of air-side pressure drop with face velocity

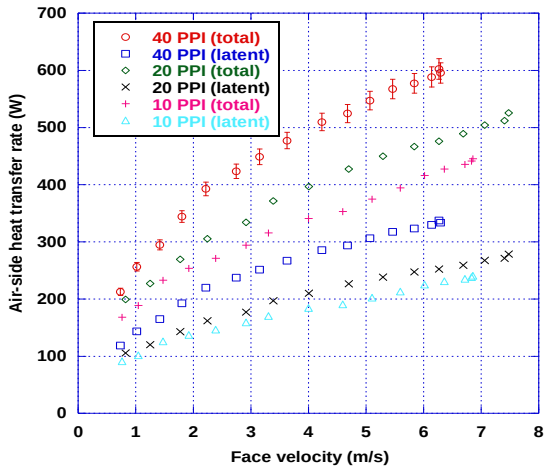

Fig. 6: Variation of air-side heat transfer with face velocity

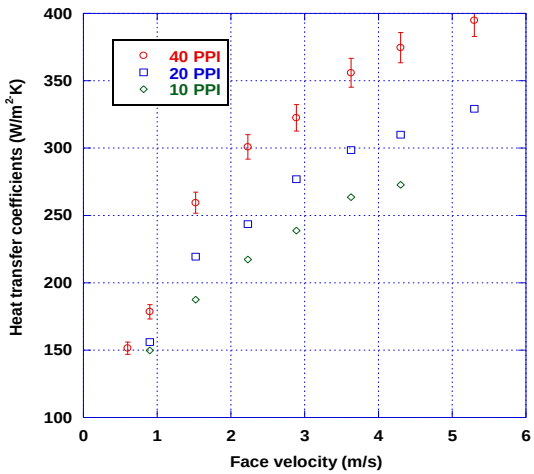

Fig. 7: Variation of heat transfer coefficient with face velocity

$$q = HAi_{LM} \quad (1)$$

$$\frac{1}{HA} = \frac{1}{(\eta_o hA/c_p)_a} + \frac{1}{(hA/b)_c} + R_b \quad (2)$$

$$\eta_f = \frac{\tanh(m_{fo}L_f)}{m_{fo}L_f} \qquad \eta_o = 1 - \frac{A_{foam}(1-\eta_f)}{A_{foam} + \varepsilon A_{base}} \quad (3,4)$$

$$\Delta i_{LM} = F(\Delta i_1 - \Delta i_2)/(\Delta i_1 / \Delta i_2) \quad (5)$$

$$\Delta i_1 = i_{a,i} - i_{a,sat(Tc,o)} \qquad \Delta i_2 = i_{a,o} - i_{a,sat(Tc,i)} \quad (6a, 6b)$$

$$m_{fo} = \sqrt{3\pi D_f h/(D_p^2 k_{ef})} \qquad k_{ef} = (1-\varepsilon)k_s/2 \quad (7, 8)$$

In equations (6a) and (6b) $i_{a,sat(Tc,o)}$ and $i_{a,sat(Tc,i)}$ present the enthalpy of the saturated air calculated at the exiting and entering coolant temperature respectively. b is the slope of saturated line.

It can be observed from above figures that the metal foam pore diameter (often defined as PPI- Pores per inch) can significantly alter the performance. It's important to note that the geometric factors such as surface area per unit volume and pore diameters are equally critical for performance under dry conditions, however, the impact becomes significant due to condensation.

4. CONCLUSIONS

The thermal-hydraulic performance of metal foam heat exchangers has been evaluated under dehumidifying operating conditions while varying the face velocity. The associated pressure drop and heat transfer rate (heat transfer coefficient) show a strong dependence on the pore diameter of the metal foam used in the sample. In all cases, the latent heat is suppressed which causes a decrease in the increment for the total heat transfer rate as the face velocity is increased.

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NOMENCLATURE

A	Heat transfer area (m^e)	h	Heat transfer coefficient (W/m^2-K)
c_{ap}	Specific heat (J/kg-K)	H,i	Enthalpy (kJ/kg)
D_f	Ligament diameter (m)	k_{ef}	Effective conductivity of foam (W/m-K)
D_P	Pore diameter (m)	Lf	Ligament diameter (m)
F	Correction factor for LMTD	q	Heat transfer rate (W)
		R_b	Bonding resistance (K/W)
η_0	Overall surface efficiency		
η_f	Fin efficiency		

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