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Open-Cell Metal Foams for Use in Dehumidifying Heat Exchangers

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Air-side performance of open-cell metal foams for use in dehumidifying heat exchangers

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ABSTRACT

Metal foams with an open-cell structure have physical and mechanical properties suggesting they might have advantages over conventional fin materials for use in air-cooling heat exchangers. For example, metal foams are low weight, have high to very high specific surface area (up to over 10000 m^2/m^3), have high gas permeability, and have relatively high thermal conductivity (for open-cell bodies). Due to these properties, open-cell metal foams have been studied for many heat transfer applications, especially as a material for constructing efficient compact heat exchangers. In this work, dynamic dips tests are undertaken to explore the water-drainage behavior of the metal foams. Experiments are also undertaken in a closed-loop wind tunnel to evaluate the pressure drop and heat transfer performance of metal foam heat exchangers under dry- and wet-surface conditions.

1. INTRODUCTION

In the HVAC industry, compact equipments are rapidly developed, which requires the use of heat exchangers with special designs and surface shapes. Some air cooled condensers for refrigeration are manufactured with enhanced surfaces, both on the external and on the refrigerant side. Metal foams with an open-cell structure have physical and mechanical properties suggesting they might have advantages over conventional fin materials for use in air-cooling heat exchangers. For example, metal foams are low weight; high to very high specific surface area (up to over 10000 m^2/m^3), have high gas permeability, and have relatively high thermal conductivity (for open-cell bodies) (www. metafoam. com). Due to these properties, open-cell metal foams are currently regarded as a highly promising material for constructing efficient compact heat exchangers (Ashby *et al.*, 2000; Bastawros and Evans, 1997; Kaviany, 1995; Ruiz, 2004; Boomsma and Poulikakos, 2001; Zhao *et al*., 2004; Anthohe *et al*, 1996; Boomsma *et al*, 2003).

In many applications, air-cooling heat exchangers operate with the heat-transfer surface below the dew point of the air, in order to dehumidify the conditioned air. Condensate accumulates on the surface and is retained by surface tension unless removed by gravitational or air-flow forces. Retained condensate has profound effects the heat transfer and pressure drop performance, and it plays an important role in the overall performance for the air-conditioning system. It also has implications on air quality, that is, water provides a medium for biological activity on air-handling surfaces, the condensate blown off the heat exchanger surface can directly influence occupant comfort.. With growing concerns about the conditioned air quality, the researchers often strive for heat exchanger designs which hope to be provided efficient condensate drainage in off-cycle operation. For example, Karkhu and Borovkov (1971), Rifert *et al.* (1977), Honda *et al.* (1983), and Rudy and Webb (1985) focused their research on the surface tension force during condensate retention. They proposed that surface tension could be the dominant force in condensate drainage for the integral-fin tube of their studies. Rudy and Webb (1985) investigated the static measurements of the amount of condensate forming on an integral-finned tube, and their model to predict the amount of the surface flooded during condensation on a horizontal, integral-fin tube agreed with experiments to within ±10% over most of the test range. Osada *et al.* (1999, 2001) conducted the heat transfer and condensate visualization studies using single-fin models of flat-tube evaporators. They tested the effects of surface wettability, louver geometry, and heat exchanger inclination. Osada *et al.* (1999, 2001) developed research on corrugated multi-louvered fins under dehumidification and drew a conclusion that fin geometry, wettability, and the characteristics of the airflow, especially at the exit face of the heat exchanger were important factors in condensate drainage. It is also found that coil inclination has a great influence on the thermal performance of an evaporator. McLaughlin and Webb (2000) examined fin geometry effects on drainage and retention characteristics to conduct a single-fin which was brazed to a plate chilled by circulating “ice-water” through a tube brazed to it in a tabletop apparatus. Their scheme allowed optical access to the fin during the formation and subsequent condensate drainage. McLaughlin (1999) compared the retained water measured in their “dip test” to that measured in a wind tunnel. They weighed a dry coil, dipped it in a bucket of water, removed it from the water, the coil began to be weighed after 15 s. The heat exchanger was allowed to drain for 120 s in the vertical position, and then a thin piece of aluminum was touched to the core bottom to remove water clinging to the lower manifold. They found the mass of remaining water to be within 10% of that measured in a wind tunnel. The remaining condensate (per fin) in their dip test was found to be 3% lower than that in their single-fin tests. It should be noted that all wind-tunnel experiments were conducted with the air frontal velocity of 2.4 m/s, and the dip test was studied in quiescent surroundings. Zhong *et al.* (2005) proposed a new method to assess the condensate drainage behavior of the air side surface for the compact heat exchangers, referring to it as a *dynamic dip test*. This method provided highly repeatable data for real-time drainage. Results from experiments for more than 20 flat-tube and round-tube-and-fin heat exchangers were compared to those results obtained in wind tunnel experiments. The compared results showed that geometrical factors have an on condensate drainage. The heat exchangers retaining the most and the least condensate in a steady-state wind-tunnel test, likewise held the most and the least in a dynamic dip test. However, different amounts of water were retained on the air-side surface during dynamic dip tests and wind-tunnel experiments. They also developed a model based on gravity, surface tension and viscous effects to help understand and predict the drainage behavior of heat exchangers. The new model and experimental approach were found to be a useful in screening heat exchangers for condensate retention and for assessing off-cycle drainage behavior. Elsherbini and Jacobi (2006) developed a model for predicting the amount of condensate retained as drops on the air-side of heat exchangers operating under dehumidifying conditions. Although prior research has showed that air-side condensate retention has an important effect on the thermal-hydraulic performance of compact heat exchangers, limited work has been reported on measuring retention and drainage from the air-side surface.

Dynamic dip testing is one method, it is simple, inexpensive, relatively fast, easy to assess condensate drainage behavior — a heat exchanger holding more water in a dip test . However, Liu and Jacobi (2008) found the reliability of the method to be effected by many factors which are often ignored. It was observed that dip test measurement is very sensitive to the “dipping rate”, or the speed at which a specimen is withdrawn from the water reservoir. Another method to such measurements is to measure the mass of a heat exchanger operating under dehumidification conditions in a wind tunnel.

Metal foams have been found to exhibit promising heat transfer for use on the air side of heat exchangers, due to their complex geometry and high surface-area-to-volume ratio, etc. These effects enhance the heat transfer performance, but at the same time presumably due to this highly complex structure, condensate retention may be problematic. Very little research in the open literatures has addressed the drainage behavior, especially the drainage under off-cycle conditions for the metal foam. Therefore, in this work, an experimental study of the performance of open-cell aluminum foam heat exchangers are conducted in dipping test apparatus and a closed-loop wind tunnel to evaluate the water-drainage behavior of the metal foams, the pressure drop and heat transfer performance of metal foam heat exchangers under dry- and wet-surface conditions.

2. EXPERIMENT

2.1 Dip test apparatus and procedure

The dynamic dip test apparatus is shown schematically in Figure 1. It includes a moving water reservoir and simple mounting hardware to suspend and weigh the heat exchanger. The moving reservoir had a volume of 0.068m^3, and it was positioned using a hydraulic jack to allow a smooth, consistent lowering. Experiments were conducted with the test specimen in vertical and horizontal orientations. An electronic balance (A&D GF-8000) was used to record the variation of weight with time. A data acquisition system based on computer with a minimum recording interval of 0.1 s was used for measuring the mass, its uncertainty is adopted as the mass measurement uncertainty for these computer-timed data.

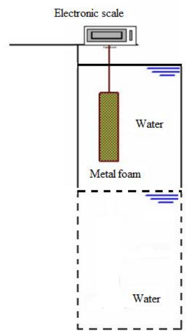

Figure 1: Schematic of dynamic dip test

A dry test metal foam was suspended over the water reservoir and the alignment was confirmed. After the balance was zeroed, the water reservoir was raised to immerse the specimen. The water was agitated to remove air trapped on the air-side heat transfer surfaces, before the reservoir was lowered. Beginning at the instant when the water level reached the bottom of the heat exchanger, mass readings were recorded at 5-s intervals for 90 s and then at 30-s intervals for additional 240s. Experiments of longer duration were also conducted to helpfully characterize the nature of water drainage.

2.2 Experimental apparatus and procedures for the closed-loop wind tunnel

The closed-loop wind tunnel is shown schematically in Figure 2. It included a closed-loop wind tunnel, a test section of heat exchangers exposed to horizontal air-flow, and a coolant loop which circulated a single-phase coolant. The closed-loop wind tunnel allowed temperature, humidity, and air flow rate control. Air temperature was controlled by varying the power supplied to four electrical resistance heaters which were capable of adding 4.0 kW to the air flow. Evenly spaced T-Type thermocouples were used both upstream (11-TC grid) and downstream (30-TC grid) of the test section to measure the air temperature (see Figure 3). Each thermocouple was individually referenced to a thermocouple located in an ice bath, and calibrated to a NIST-traceable mercury-in-glass thermometer using a thermostatic bath.

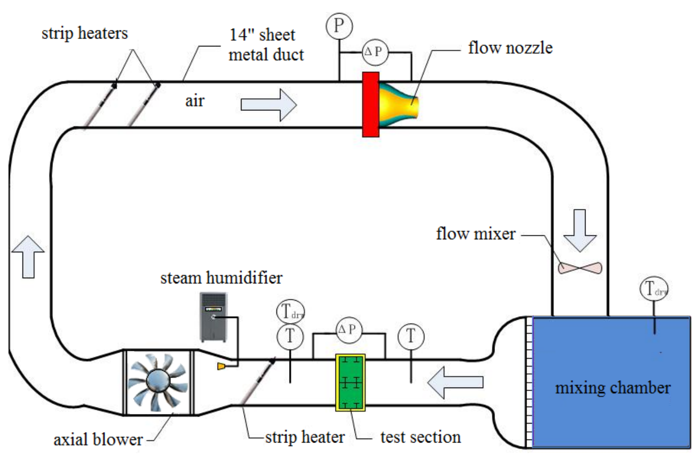

Figure 2: The flowchart of a closed-loop wind tunnel

Calibration data were fit with fifth order polynomials for each thermocouple to give an uncertainty of ± 0.2 ℃. The dew point of the air was maintained by injecting steam at a controlled rate. They were measured by chilled mirror hygrometers with a measurement uncertainty of ± 0.2 ℃. The measurement of the inlet dew point was used to provide a control signal for a closed-loop dew point control. The control signal was transmitted to a PID controller which could adjust the output of the humidifier to attain the desired inlet air dew point. The heaters and the steam injection were located downstream of the test section before the axial fan (Model: Dayton 42380A, USA). The axial fan had a mixture with the conditioned air and provided volumetric flow rates up to 6.55 m^3/min. After air was drawn from a thermal mixing chamber , it passed through a set of screens, honeycomb flow straightness, and a 9:1 contraction to get a steady laminar flow before passing through the test section.

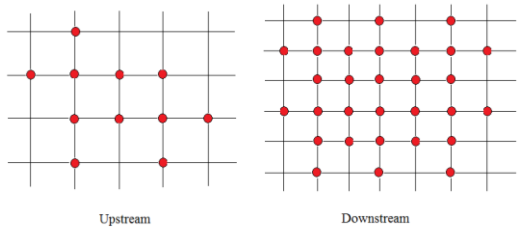

Figure 3: Measuring temperature distribution

A unique test section, shown in Figure 4, was designed for testing dry and wet heat exchangers. The design allows for both real-time and steady-state measurements of the mass of retained condensate. The test section was constructed using clear acrylic to allow for optical access and insulated with 1.27cm thick foam. An interchangeable frame was implemented to allow for testing of different heat exchangers. Pressure taps from upstream and downstream were located on all four sides of the rectangular test section to measure the pressure drop across the heat exchanger. They were located about three inches upstream and downstream of the heat exchanger and on the centerline of each side of the test section. Face velocities were measured at the test section using a constant temperature thermal anemometer. The face velocity was measured by the anemometer (Model: 8355, USA) . A single-phase ethylene glycol (DOWTHERM 4000) and water mixture was circulated on the tube side of the heat exchanger. A chiller controlled the coolant temperature, and the mixture was circulated through a copper tubing loop by two pumps. The copper tubing loop was insulated with 9.5mm thick foam. The heat exchanger was connected to the copper tubing with flexible, reinforced, PVC tubing also insulated by 9.5 mm thick foam.
Coolant-side temperatures were measured using PT100 Platinum resistance thermometer located approximately two meters upstream and downstream of the heat exchanger. Coolant flow rate was measured using a coriolis mass flow meters with a manufacturer reported measurement uncertainty of $\pm 0.1\%$. All data were logged by the Ni transducer.
Pressure and pressure drop were measured by An electric manometer (Model: 1430, USA) with an uncertainty of ± 0.124 Pa was used to measure the air-side pressure drop across the heat exchanger.. OMEFA RH-usb probes were used to measure the inlet and outlet air relative humidity. In this work, metal foam heat exchanger is selected as follows, shown in Figure 4 and Table 1.

Figure 4: Metal foam heat exchanger of flat tube configuration

Table 1: Heat exchanger design specifications

Name	**Specification**
Sample	10 PPI
Base metal	Al 6061 alloy
Porosity	0.942
Pore diameter	1.8mm
ligament diameter	0.2 mm
Tube side configuration	Flat tube
Number of fins	10
Fin depth	15 mm
Fin thickness	15 mm
Bonding method	Artic silver, thermal compound
Face area	200 mm×174 mm
Tube width	25.4 mm
Tube wall thickness	0.5 mm

Table 2: Instrument specification and precision

Instrument	Model	Range	Accuracy
Thermocouple	T-type	-50 - 200 ℃	±0.1 ℃
Resistance thermometer	Pt100, three-wire	-50 - 100 ℃	±0. 3 ℃
Microtector	1430	0 - 0.689 MPa	±0.02% F.S.
Mass flow meter	Micro Motion, 3950S1239U	0 -2180 kg / h	±0.1 %
Axis blower	Dayton 42380A	0-30A, 0-30V	±0.2 %
Temperature controller	WaTlow Series 982		
16-Channel Thermocouple Input Module	NI 9213	-40 ℃～70 ℃	0.02
Analog Input Module	NI 9205	-40 ℃～70 ℃	
Anemometer	8355		

At the beginning of each experiment, the chiller temperature was set. When the liquid supply temperature reached steady state, the temperature controller and axial blower for the air-side were turned on, and the air-side temperature and velocity were set. The air and coolant inlet and outlet temperatures were then monitored until steady-state conditions were obtained (usually about 60 minutes). Steady-state data (temperature, air velocity, air-side pressure, mass flow rate of coolant) were then recorded via the DAQ system. The recorded data were then reduced using the equations provided later to calculate the overall heat transfer coefficient.

Wet experiments were conducted to evaluate the effect of the condensate on the overall thermal performance of the same heat exchanger. For these tests, the inlet coolant temperature which was lower than the local dew point temperature was held constant,, a humidifier and a large boiling water reservoir were used to increase relative humidity above 85%. A special thermal-conditioning chamber was also used to help maintain these moisture levels at the wind tunnel inlet. The heat exchanger was exposed to these wet conditions for approximately 1.5 hour for each air face velocity before data were recorded. The specific operating conditions for these dry and wet experiments are shown below in Table 3.

Table 3: Specific experimental conditions

Parameters	Dry condition	Wet condition
Coolant temperature	3.7℃ , 7.0℃	3.7℃
Air inlet velocity	0-7m/s	0-7m/s
Inlet relative humidity	20%-30%	50%-70%
Air inlet temperature	20℃-35℃	20℃-35℃

3. DATA REDUCTION

To determine the wet and dry air-side heat transfer coefficients for the heat exchanger, the following data reduction procedure was used. On the basis of an energy balance for both the air stream and the coolant stream, the following heat transfer rate expression can be written:

$$Q_{air} = \dot{m}_{air} c_{pair} \left(T_{airdown} - T_{airup} \right) + \dot{m}_{condensate} h_{fg} \quad (1)$$

$$Q_{cool} = \dot{m}_{cool} c_{pcool} \left(T_{coolin} - T_{coolout} \right) \quad (2)$$

$$Q_{ave} = \left(Q_{air} + Q_{cool} \right) / 2 \quad (3)$$

In this work, Reynolds Number *Re* and Nusselt number *Nu* are calculated following

$$\mathrm{Re} = \rho V D_h / \mu \quad (4)$$

Where V is the velocity based on the actual cross section area of the duct or pipe and D_h is from:

$$D_h = 4A_{\min}L / A_T \tag{5}$$

Where the minimum free flow area, $A_{\min}$, is related to the frontal area directly by the porosity, ε. For metal foam,

$$A_{\min} = \varepsilon A_{fr} \tag{6}$$

A_T is the total surface area which is comprised of the exposed tube area , A_{base}, and the surface area of the metal foam, A_{foam}

$$A_T = A_{base} + A_{foam} \tag{7}$$

Where A_{foam} is determined from manufacturer's data for foam surface area per unit volume. The Nusselt number is calculated by the following equation

$$Nu = \frac{hD_h}{k} \tag{8}$$

Where h is obtained by the following procedure. For the data reduction purposes, the metal foam was considerred as a porous fin with an adiabatic tip condition. A relation developed by Calmidi and Majahan (2000) was used. The LMTD (log-mean temperature difference) was used to reduce the data for dry condition test, while LMED (log-mean enthalpy difference) was used for wet conditions as both sensible and latent heat transfer were involved. For a metal foam heat exchanger of the dry condition

$$q = UA \cdot LTMD \tag{9}$$

where

$$LMTD = F\,\frac{(T_{air,in} - T_{coolant,out}) - (T_{air,out} - T_{coolant,in})}{\ln\left(\dfrac{T_{air,in} - T_{coolant,out}}{T_{air,out} - T_{coolant,in}}\right)} \tag{10}$$

The flow configuration factor, F, from Incropera and Dewitt (1996).

The overall thermal conductance of the heat exchanger, UA, is formulated by neglecting the conduction resistance of the tube wall

$$\frac{1}{UA} = \left(\frac{1}{\eta_o Ah}\right)_{air} + \left(\frac{1}{Ah}\right)_{coolant} + R_{bond} \tag{11}$$

The thermal contact resistance due to bonding the foam to the tubes, R_{bond}, was determined from ancillary experiments. The coolant-side convection coefficient is determined for the in-tube single- phase flow during the experiments.

The air-side fins are accounted for using the surface efficiency

$$\eta_o = 1 - \frac{A_{foam}\left(1-\eta_f\right)}{A_{foam} + \varepsilon A_{base}} \tag{12}$$

The fin efficiency η_f is then calculated assuming a straight fin with an adiabatic tip, following Dai *et al.*(2012)

$$\eta_f = \frac{\tanh(m_{foam}L_f)}{m_{foam}L_f} \tag{13}$$

where the fin parameter accounts for the ligament and pore diameters, D_f and D_p, respectively

$$m_{foam} = \sqrt{3\pi D_f h / (D_p^2 k_{eff})} \tag{14}$$

For metal foam PPI=10, D_f is 0.4 mm, D_p is 3.13 mm.

The effective thermal conductivity of the foam is taken as the solid-only effective thermal conductivity (Dai *et al.*, 2010)

$$k_{eff} = (1-\varepsilon)k_{solid} / 2 \tag{15}$$

For a metal foam heat exchanger of the wet condition, LMED is used to calculate the heat transfer coefficient (Xia *et al.*, 2009)

$$q = HA\Delta h_{LM} \tag{16}$$

$$\frac{1}{HA} = \frac{1}{\left(\eta_0 hA/C_p\right)_a} + \frac{1}{\left(hA/b\right)_w} \tag{17}$$

$$\Delta h_{LM} = F\frac{\left(h_{air,in} - h_{as,out}\right) - \left(h_{air,out} - h_{as,in}\right)}{\ln\left(\dfrac{h_{air,in} - h_{as,out}}{h_{air,out} - h_{as,in}}\right)} \tag{18}$$

Where Δh_{LM} is the log-mean enthalpy difference, *HA* is the overall heat transfer coefficient based upon the enthalpy difference, C_{pa} the air side sensible heat transfer coefficient.

4. RESULTS AND DISCUSSION

4.1Results and discussion for dip testing

In this work, metal foams with 5 10, 20 and 40 PPI (shown in Fig. 5) were used to conduct the dip testing. The water retention in grams per unit volume for samples with four different porosities is presented in Figures 6-7. Experiments on all the samples were conducted under same conditions and equal time was given to analyze the steady state behavior for the water retained in the sample. It can be observed from the curves that porosity has a high impact on the water retention, as a 40 PPI sample with smaller sized pores retains much more water than does the 10 PPI sample, and the position has an effect on water retention for different porosities.

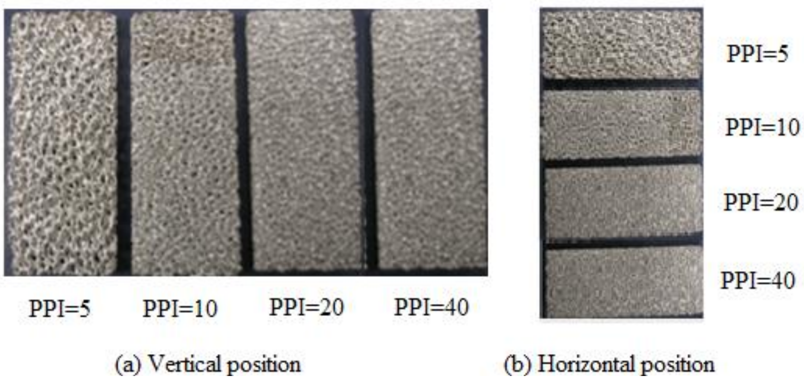

Figure 5: Metal foam samples with different porosities (5, 10, 20 and 40PPI)

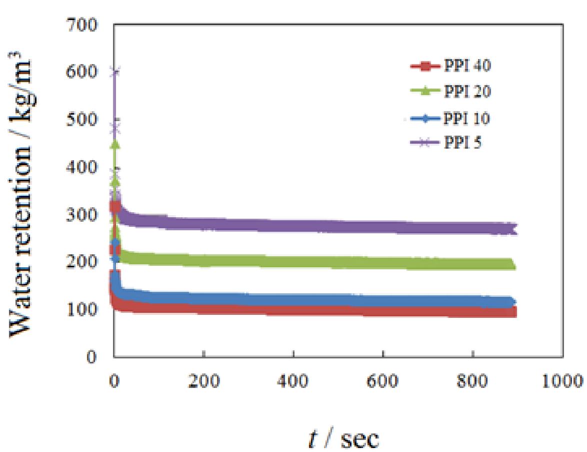

Figure 6: Water retention for metal samples with different porosities in vertical position

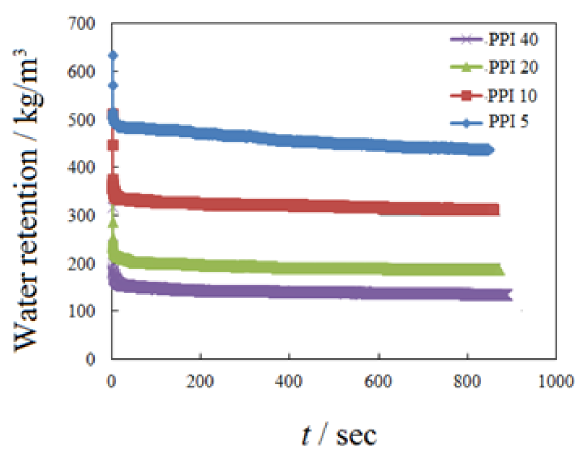

Figure 7: Water retention for metal samples with different porosities in horizontal position

4.2 The results and discussion for the wind tunnel experiment

In order to determine the effect of condensation accumulation on the air-said heat transfer of the metal foam heat exchanger, both dry and wet condition experiments were conducted on the wind tunnel experimental apparatus. The dry experiments were mainly done to provide a baseline for comparison with the wet experiments. The energy balance for the experiments varied between 0 and 10 % for the dry test conditions and reached as high as 15 % for the wet test conditions. The uncertainty in the heat transfer rate to the coolant on tube side was small compared to that from the air side. The results for the pressure drop per unit length are plotted against the face velocity under dry surface conditions and wet surface conditions in Figure 8.

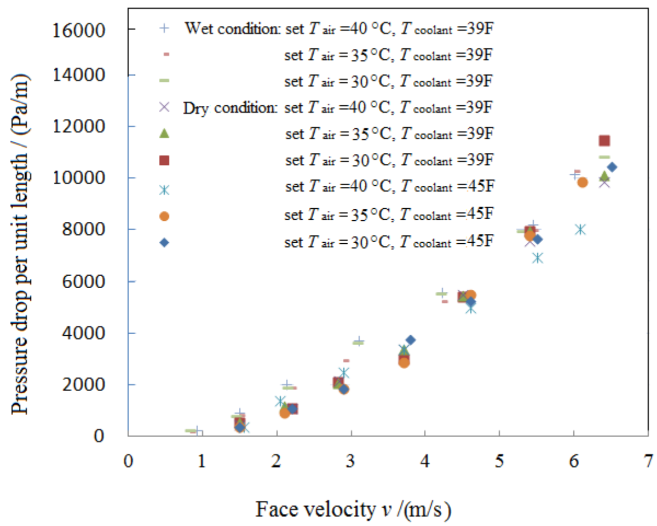

Figure 8: Pressured drop per unit length for PPI 10 metal foam under dry conditions and wet conditions

From Figure 8, it is known that for PPI 10 metal foam, the pressure drops per unit length in wet conditions are slightly bigger than those in the dry conditions at the same set temperatures. For the dry condition, the differences for the pressure drops per unit length are very small in different coolant temperatures. For the same set coolant temperature, differences for the pressure drops per unit length in different set air temperatures are negligible.

Figure 9 and Figure 10 presents the air-side heat transfer coefficient *h* as a function of face velocity, *V* under both dry and wet conditions for the metal foam heat exchanger. From Figures 9-10, it can be seen that *h* increases with increasing face velocity. Compared to dry conditions, the heat transfer coefficient *h* under wet conditions is slightly higher.

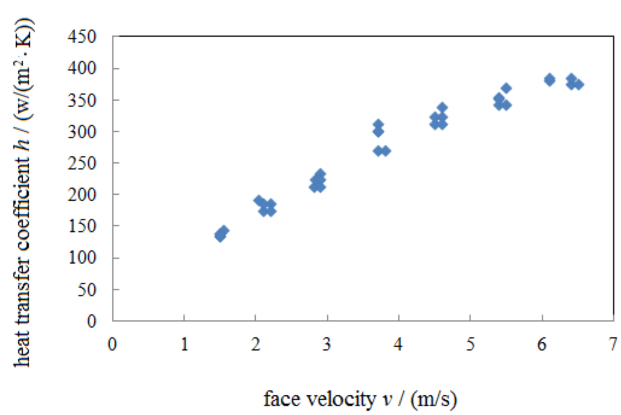

Figure 9: The air-side heat transfer coefficient, *h*, as a function of face velocity ,*V* ,under dry conditions

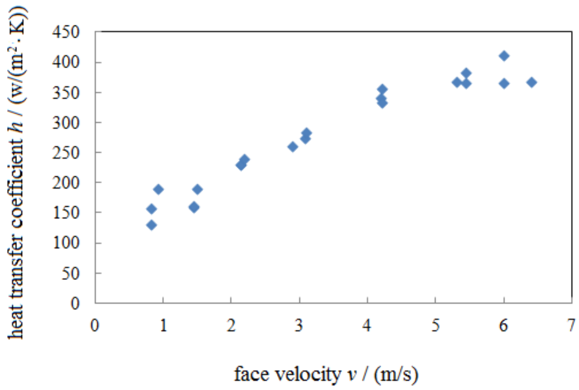

Figure 10: The air-side heat transfer coefficient, *h*, as a function of face velocity, *V*, under wet conditions

5. CONCLUSIONS

Open-cell metal foams have been studied for many heat transfer applications, especially as a material for constructing efficient compact heat exchangers. In this work, dynamic dips tests were undertaken to explore the water-drainage behavior of the metal foams. The drainage from metal foams is as good as or better than drainage from louver-fin heat exchangers. The experimental results showed that the porosity has a high impact on the water retention, as a 40 PPI sample with smaller sized pores retains much more water than does the 10 PPI sample, and the orientation has an effect on water retention for different porosities. Experiments were also undertaken in a closed-loop wind tunnel to evaluate the pressure drop and heat transfer performance of metal foam heat exchangers under dry- and wet-surface conditions. The heat transfer coefficient, *h*, increases with increasing face velocity, *V*, under dry and wet conditions. Compared to dry conditions, the heat transfer coefficient *h* under wet condition is slightly higher.

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