14th IEA Heat Pump Conference
15-18 May 2023, Chicago, Illinois

Pool boiling on metal-foam enhanced tube bundle: heat transfer characteristics and flow visualization

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Abstract

A flooded evaporator configuration is common in large central air conditioning or process cooling systems. It is basically a shell and tube heat exchanger, in which a secondary fluid (brine or water) circulates inside the tube bundle and is cooled by the vaporization of the refrigerant on the outside surface of the tubes. The enhanced pool boiling process enables the compact design of flooded evaporators, which substantially reduces the refrigerant charge. High-porosity metal foam, with a large surface-area-to-volume ratio, could provide an extended heat transfer area and a high-density of nucleation sites. This study experimentally investigated the pool boiling heat transfer and flow characteristics on metal-foam enhanced tube bundles. The enhanced bundle consists of four aluminum tubes with aluminum foam brazed around the outer surface, which are horizontally mounted in a staggered arrangement. The results showed that the metal-foam enhanced tube bundles improved the heat transfer coefficient by 100-160% with a lower wall temperature difference of 1-10°C, compared to the baseline. In addition, the tube pitch played a significant role in determining the pool boiling behavior of the tube bundles.

Keywords: Tube bundle;metal foam;pool boiling;heat transfrer; flow visualization

1. Introduction

Tube bundles have been widely employed in flooded evaporators for air-conditioning and refrigeration, desalination, and absorption chiller industries. In order to reduce the size of heat exchangers and maintain a lower temperature difference between the tube wall and working fluid, improving the pool boiling heat transfer is pivotal. In this regard, several enhanced tube geometries have been developed for industrial applications, such as GEWA, TURBO, and HIGH FLUX tubes. Ayub and Bergles [1] demonstrated 1.5- and 1.75-times heat transfer enhancements for the R-113 pool boiling using GEWA-K and GEWA-T tubes, respectively. Thome [2] demonstrated a 4-10 fold improvement in GEWA-TX tubes compared to the plain tubes.

Recently, utilizing open-cell metal foam to enhance the pool boiling heat transfer has gained significant attention due to its larger area-to-volume ratio and liquid replenishment ability through capillary action. These features of metal foam structures could aid in improving the heat transfer rate and delaying the critical flux. Nosrati et al. [3] studied the flow boiling of R-134a in metal foam filled tubes (metal foam on the tube side). They reported a 220% improvement in the heat transfer coefficient. Similarly, Hu et al. [4] tested the wettability effect of metal foam filled tubes during the flow boiling using R-134a. In their study, they compared the flow boiling behavior of hydrophilic, hydrophobic, and uncoated metal foams. The results showed that the hydrophobic coating offered a 6-30% enhancement in heat transfer compared to the uncoated one, whereas the hydrophilic coating reduced the heat transfer by 2-18%. Manetti et al. [5]

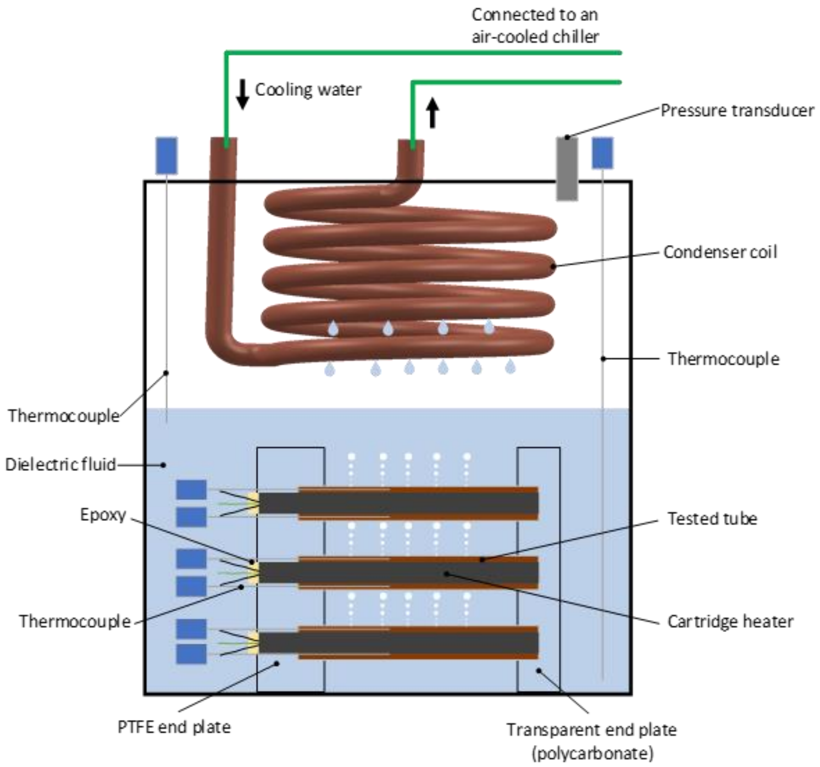
performed the pool boiling experiments on a copper metal foam surface (flat surface) using HFE-7100. They suggested that the metal foam surfaces can overcome the temperature overshoot problem associated with electronic cooling due to their interconnected porous structure that increases the wetted surface area and nucleation site density. Shi et al. [6] analyzed the behavior of a flat copper foam surface under pool boiling conditions for different wettability's. They reported that the superhydrophilic coated foams offered a higher heat transfer rate when the heat flux was above 20 W cm^{-2}, whereas the superhydrophobic coatings outperformed the superhydrophilic ones when the heat flux was below 20 W cm^{-2}. The recent developments in phase change heat transfer in metal foam structures are summarized in [7, 8].

In summary, the pool boiling behavior of metal foam structures is widely studied for the flat surface configuration, and the flow boiling characteristics are analyzed for the metal foam filled tubes (i.e., metal foams are placed inside the tubes). In our previous study, the pool boiling heat transfer over an externally embedded metal-foam tube was experimentally investigated [9]. Yang et al. reported that the effective heat transfer coefficient of the metal-foam enhanced tube was 2.1~4.8 times higher than a bare aluminum tube. However, the studies that pertain to the pool boiling behavior of externally embedded metal- foam *tube bundles* (i.e., shell side) are limited in the open literature. Therefore, the current study aims to develop a metal foam tube bundle to improve the shell side heat transfer rate. The performance of the metal foam tube bundle is compared to the bare tube bundle. In addition, a visualization study is carried out to understand the bubble dynamics on both bare and metal-foam enhanced tube bundles.

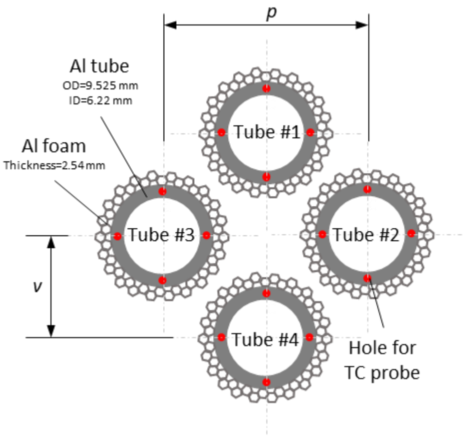
2. Experiments

The schematic of the pool boiling apparatus for tube bundle experiments is shown in Fig. 1 (a). It consists of a pool boiling chamber, a heat transfer test section, a condenser coil, and a cooling system. The pool boiling chamber is made of clear polycarbonate, and its internal dimensions are 222.3 mm × 212.7 mm × 222.3 mm. The chamber is charged with 4 Liters of dielectric liquid at atmospheric pressure, and the heat transfer test section is submerged in the liquid pool. The system pressure is monitored with an Omega absolute pressure transducer with an accuracy of ±0.2% Upper range limit (URL) ranging from 0 to 103.4 kPa. The bulk temperature of the liquid pool is measured with T-type thermocouples at two different locations. A coil condenser located at the upper part of the chamber is designed for condensing the dielectric vapor generated through the pool boiling process. The evaporated vapor rises and condense on the outer surface of the coil. The condensed liquid is dropped and returned to the pool to create a natural circulation. The cooling water for the coil is supplied by a recirculating chiller with a cooling capacity up to 7.5 kW (Thermo Scientific, ThermoFlex 7500).

(a)



(b)



	Case 1	Case 2
Tube pitch p	19.05 mm	25.4 mm
Vertical distance between tubes v	16.5 mm	22.0 mm

Fig. 1. Schematic diagrams of the (a) experimental apparatus and (b) detailed geometry of the tested tube bundles.

The heat transfer test section of the lab-scale tube bundle experiments has four 3/8” OD aluminum tubes in a staggered arrangement. As demonstrated in Fig. 2(a) and (b), the baseline case consists of bare tubes, whereas the enhanced tube bundle comprises metal-foam tubes. Fig. 1(b) provides the detailed geometry of the tested tube bundles. The tubes have an inner diameter D_i=6.22 mm and an outer diameter D_o=9.525 mm. Two tube pitches of the bundle are investigated in this study. The externally enhanced tubes (metal foam tubes) have a 2.54 mm thick layer of open cell aluminum foam brazed around the outer surface. The aluminum foam used in this study is 40 PPI Duocel foam made of 6101 alloy with heat treated to T6 condition from ERG Aerospace Corporation. The porosity of the metal foam is 81%, estimated through X-ray computed tomography (XCT) 3D scanning. The tested tubes are 76.5 mm long, and they are horizontally mounted to two end plates. One is made of PTFE, while the other is made of transparent polycarbonate for visualizing the boiling process. Two brass pieces are fastened to the end plates as additional weight to ensure the heat transfer test section is submerged. More details of the tube assembly can be found in the work of Yang et al. [9].

(a) (b)

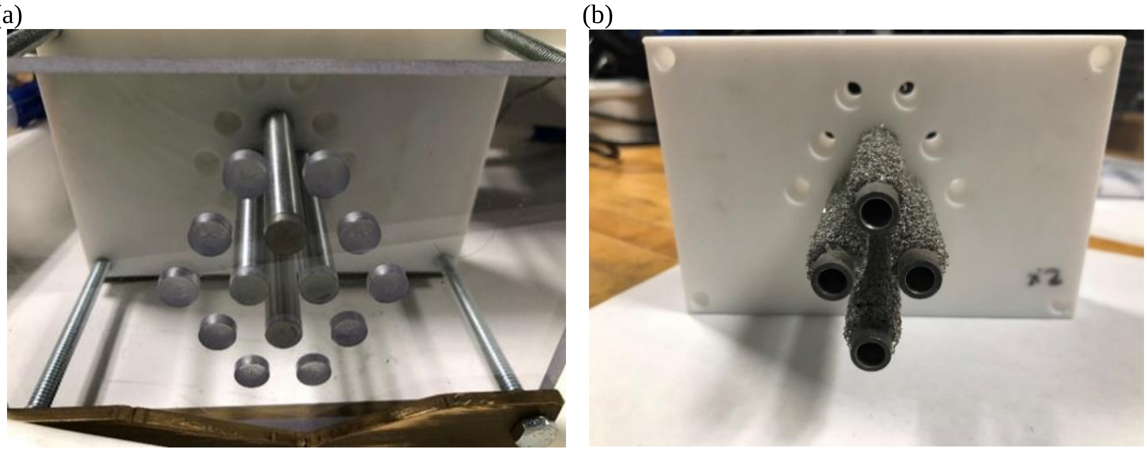


Fig. 2. (a) bare aluminum tube bundle and (b) metal foam enhanced tube bundle.

For each tube, a Watlow cartridge heater with 579W heating capacity is inserted to provide the heat. The outer diameter and the heating length of the heater are, 6.22 mm and 63.5 mm, respectively. To be noted that epoxy is applied to the end of the cartridge heaters to reduce the heat loss in the axial direction. The power to the cartridge heaters is controlled using pulse width modulation, and different outputs can be approximated over a time span of 1 second. The actual heating power is measured using a watt transducer ranging from 0 to 3 kW (Ohio Semitronics, #PC5-020X5Y25). The wall temperatures for each tube are measured at four locations with T-type thermocouple probes. Four holes with the diameter of 1.02 mm are drilled alongside the tube wall at 90 degrees apart. The first 25.4 mm of the thermocouple probes stays inside the tube wall, and the remaining portion that is directly exposed to the dielectric fluid is insulated with a PVC sleeve. All the thermocouple probes were calibrated against a NIST-certified precision thermometer with an accuracy of ±0.05°C, and all the data are collected and recorded with a Campbell Scientific CR1000 measurement and control datalogger accompanied with an AM25T thermocouple multiplexer.

For the pool boiling experiments of the tube bundles, 3M Novec engineered fluid HFE-7000 is used as the working fluid. HFE-7000 is a type of clear, colorless, thermally stable dielectric fluid with 34°C boiling point under atmospheric pressure. Prior to the experiments, the pool boiling chamber is charged with 4 Liters of the HFE-7000, which provides around 25.4 mm liquid level above the heat transfer test section. At the beginning of the experiments, the cartridge heaters are set to 10% of its maximin heating capacity to heat up the liquid pool to reach saturation temperature. The liquid pool is heated with the same power setting for an additional hour to remove the non-condensable gases dissolved in the fluid. The power of the cartridge heaters is then reduced to 5% to start the heat transfer measurement. Once the steady state is reached, the data is collected and recorded for 10 mins. The experiments are repeated by increasing the heater power with 5% increments.

3. Data Reduction

The effective pool boiling heat transfer coefficient (*HTC*) on the outside of the tube is calculated as the ratio of the heat flux to wall superheat.

$$HTC = \frac{q''}{\Delta T_{\mathrm{sup}}} \tag{1}$$

The heat flux q'' is estimated based on the outer surface area of the bare tube:

$$q'' = \frac{\dot{Q}_{elec} - \dot{Q}_{loss}}{\pi D_o L_h} \tag{2}$$

where D_o and L_h are the outer diameter and the effective heating length of the tested tube. The effective heat transfer rate from the tube wall to the dielectric fluid is determined by the electrical power delivered to the heater, $\dot{Q}_{elec}$ subtracting the conduction heat loss in the axial direction, $\dot{Q}_{loss}$. The conduction heat loss in the axial direction is estimated using a special tube, in which the thermocouple holes alongside the tube have two different depths.

$$\dot{Q}_{loss} = kA_{cs} \frac{T_{wall,R} - T_{wall,L}}{\Delta x} \tag{3}$$

where k and A_{cs} are the thermal conductivity and the cross-sectional area of the tube material, respectively. Δx is the depth difference between the two thermocouple holes in the axial direction. The subscript of the wall temperature shows the location of the thermocouple probe, and R and L represent the right and left, respectively.

The wall superheat, ΔT_{sup} , is the temperature between the tube wall temperature and the saturation temperature of the dielectric fluid.

$$\Delta T_{\mathrm{sup}} = T_{wall} - T_{sat} \tag{4}$$

The saturation temperature T_{sat} is determined by the average values of the two T-type thermocouple probes located at the top and bottom of the pool boiling chamber. During the experiments, the pool temperature is maintained within 0.3°C. The wall temperature T_{wall} for each tube is the mean temperature of the four thermocouples inserted into the holes alongside the tube wall and corrected using the 1-D heat conduction equation.

$$T_{wall} = T_{m,avg} - \left(\frac{\dot{Q}_{elec} - \dot{Q}_{loss}}{2\pi k L_h} \right) \ln\left(\frac{r_o}{r_m} \right) \tag{5}$$

where r_o is the outer tube radius, and r_m is the mean value of inner and outer tube radius, which is the location of the tip of thermocouple probes.

The overall heat transfer coefficient for the tube bundle is estimated based on the average value of four single-tube heat transfer coefficients calculated using Eq. (1).

The experimental uncertainties of the effective heat transfer coefficient, the heat flux, and the wall superheat are estimated following the error propagation analysis documented in the work of Moffat [10]. The root-sum-square combination of the effects of the individual measurements gives the overall uncertainty of the target parameter. All the uncertainties are based on a confidence level of 95%.

$$\delta U\left(X_1, X_2, \cdots, X_N\right) = \sqrt{\sum_{i=1}^{N}\left(\frac{\partial U}{\partial X_i}\delta X_i\right)^2} \tag{6}$$

4. Results and Discussion

4.1. Validation of experimental setup

The pool boiling experiments are first performed on a single bare tube to validate the experimental facility. The experimental data are compared against the Cooper [11] and Cornwell and Houston [12] correlations, as shown in Fig. 3. The Cooper correlation and the Cornwell-Houston correlation are given in Eqns. (7) and (8), respectively.

$$HTC_{Cooper} = 90M^{-0.5}\left(q''\right)^{0.67}\left(-\log\left(P_r\right)\right)^{-0.55}\left(P_r\right)^{\left(0.12-0.2\log\left(R_p\right)\right)} \tag{7}$$

where M and P_r are the molecular weight and reduced pressure. R_p is the average surface roughness of the plain tube, and 1 μm was used in the current study.

$$HTC_{Cornwell-Houston} = 9.7P_c^{0.5}\,\mathrm{Re}_b^{0.67}\,\mathrm{Pr}^{0.4}\left(1.8P_r^{0.17} + 4P_r^{1.2} + 10P_r^{10}\right)\left(\frac{k_l}{D_o}\right) \tag{8}$$

Noted that P_c is critical pressure in bar and k_l is the thermal conductivity of dielectric liquid. Re_b is boiling Reynolds number, which is directly related to the heat flux under the saturated pool boiling, as follows.

$$\mathrm{Re}_b = \frac{GD_o}{\mu_l} = \frac{qD_o}{\mu_l h_{fg}} \tag{9}$$

where μ_l is the dynamic viscosity, while h_{fg} is the latent heat of vaporization.

Fig. 3 shows the variation of heat transfer coefficient of HFE-7000 against heat load ranging from 5 to 100 kW/m^2. Similar to the trend predicted by the correlations [11][12], the heat transfer coefficient increases with heat flux. In addition, the Cooper [11] correlation predicts the heat transfer coefficients within a 7% deviation when the heat flux is below 50 kW m^{-2}. At higher heat fluxes (q > 50 kW m^{-2}), the deviation between experimental and predicted data grows. In contrast, the Cornwell-Houston [12] correlation showed better agreement at higher heat fluxes with a maximum deviation of 7% and exhibited a larger deviation at smaller heat fluxes with a maximum deviation of 13%. Overall, the experimental data matches well with the standard correlations, which validates the experimental setup.

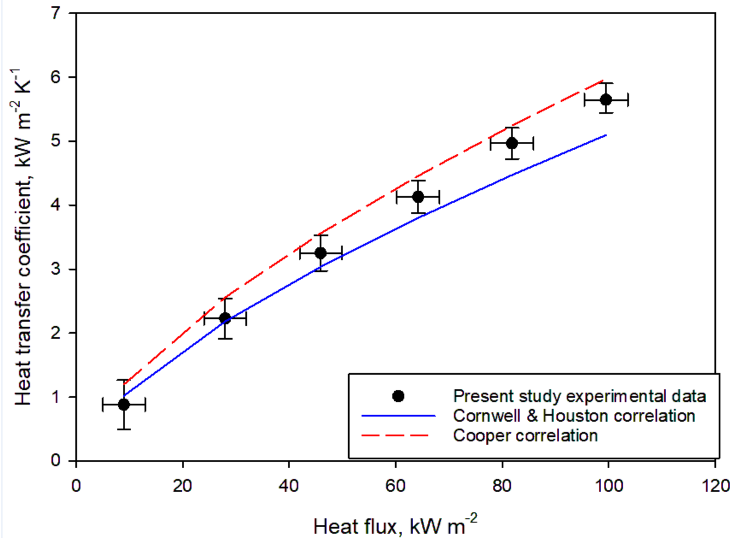


Fig. 3. Validation of experimental facility through comparing single tube pool boiling data against Cooper [11] and Cornwell-Houston [12] correlations.

4.2. Performance assessment of bare and metal foam tube bundles

The performances of a bare single tube and a metal foam tube bundle are characterized in terms of heat transfer coefficient and wall temperature. Note that the results presented in this section pertain to a tube pitch of 19 mm. The experiments are performed for a heat flux range of 5-100 kW m^{-2}, and the corresponding average heat transfer coefficient values are presented in Fig. 4. The HTC of the bare tube bundle is almost 3-5% higher than that of the single bare tube. It indicates that the bundle effect caused by the vapor flow induced convection is minimum for this tube bundle configuration as it only has three rows. However, it can be seen in Fig. 4 that the metal foam tube bundle exhibited a superior performance over the bare tube bundle. The trend of the metal foam tube bundle is consistent with that of the single metal foam tube published in the earlier work [9]. The heat transfer coefficient of the metal foam tube bundle is about 100-160% higher than that of the bare tube bundle for the tested heat flux range. The improvement in the metal foam tube bundle is primarily attributed to the increased surface area. Besides, the metal foam structures possess many active nucleation sites compared to the plain tube, as shown in Fig. 5. The large number of active nucleation sites in metal foam tubes could ameliorate the nucleate boiling heat transfer compared to the bare tube, which in turn increases the heat transfer coefficient.

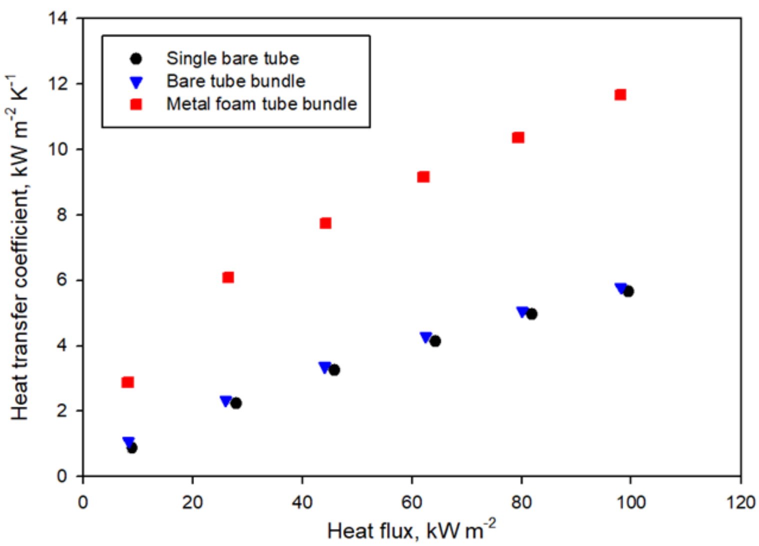
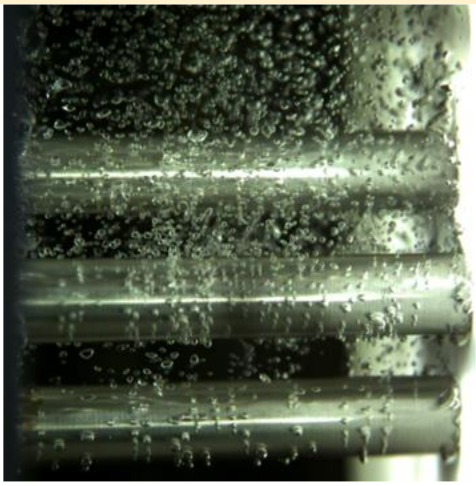


Fig. 4. Performance comparison of bare and metal foam tube bundles.

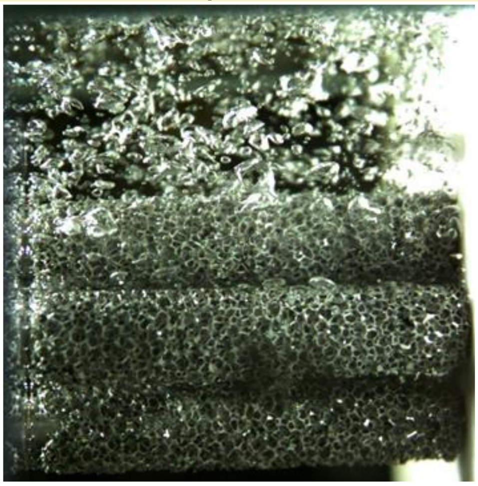
The photographic comparison of the pool boiling behavior in bare and metal foam tube bundles can be seen in Fig. 5. At lower fluxes (q < 10 kW m^{-2}), the wall temperature is close to the saturation temperature of the working fluid (i.e., HFE-7000), as shown in Fig. 6. At this stage, the number of nucleation sites is relatively low, as shown in Fig. 5(a). When the heat flux is increased, the nucleation sites get activated, leading to more bubble formation, as shown in Fig. 5(b). This leads to a bubble coalescence and forms local vapor blankets on the tubes, which hinder the heat transfer from the tubes to the fluid. As a result, the wall temperature keeps increasing with heat load, as shown in Fig. 6. It can be noted that the wall temperature of the bottom tube (i.e., tube 4) is higher than that of other tubes for both bare and metal foam configurations. This is because of the bubble flow pattern. The bubble or vapor column's departure from the bottom tube is mainly impeded by the top (i.e., tube 1) and side (i.e., tubes 2 and 3) tubes. However, the vapor column leaving the bottom tube mainly passes through the gap between tube 2 and tube 3, and finally hits the bottom of tube 1, as shown in Fig. 5 This phenomenon enhances the convection induced heat transfer in tubes 1, 2, and 3. As a result, tubes 1-3 experienced a lower wall temperature compared to that of the bottom tube, as shown in Fig. 6. When comparing Fig. 6(a) and (b), it can be understood that the wall temperature of the metal foam tubes is almost 4-10°C lower than that of the bare tubes. The lower wall temperature of metal foam tubes is attributed to the combination of the following effects: The metal foam on the outside of the tube increases the surface area and thereby the active nucleation site density, which leads to increased latent heat transfer (i.e., pool boiling). Additionally, the open cell metal foam structure could lead to a smaller bubble departure diameter even at the higher heat fluxes, which can potentially delay the local vapor blanket formation. Furthermore, the porous structure of the metal foam can act as a capillary wick to locally feed the liquid to the tube wall due to the capillary action. The combined effect of the above causes could result in a lower wall temperature for the metal foam tubes, as shown in Fig. 6(b).

(a) q=8 kWm^{-2}

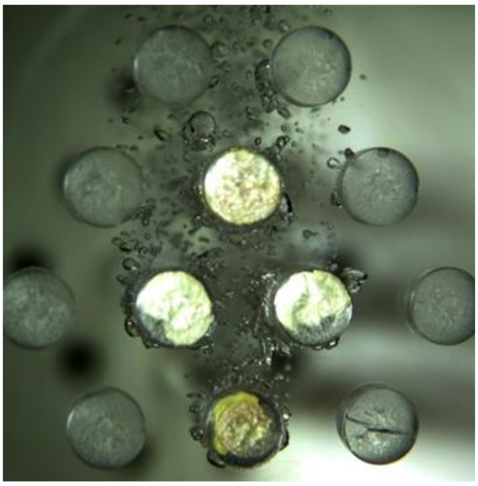
Bare tubes	***Metal foam tubes***



Front view



Front view



Side View

Side View

(b) q=98 kWm^{-2}

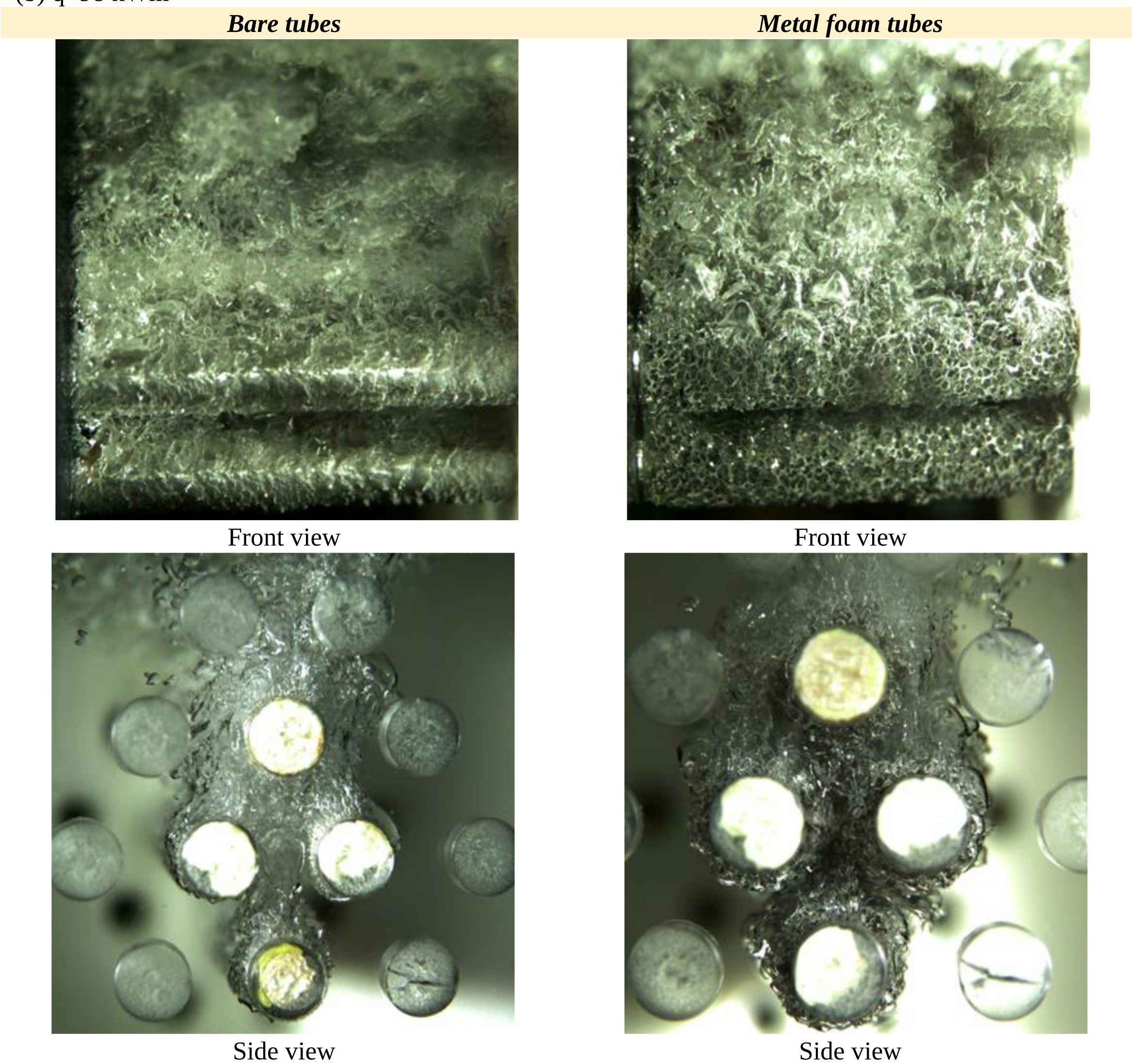
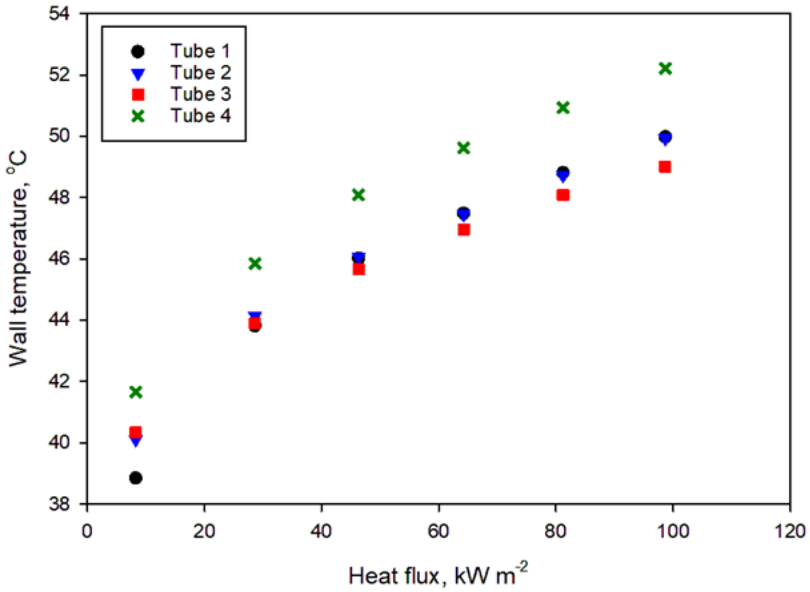


Fig. 5. Visualization of pool boiling behavior in bare and metal foam tube bundles.

(a)



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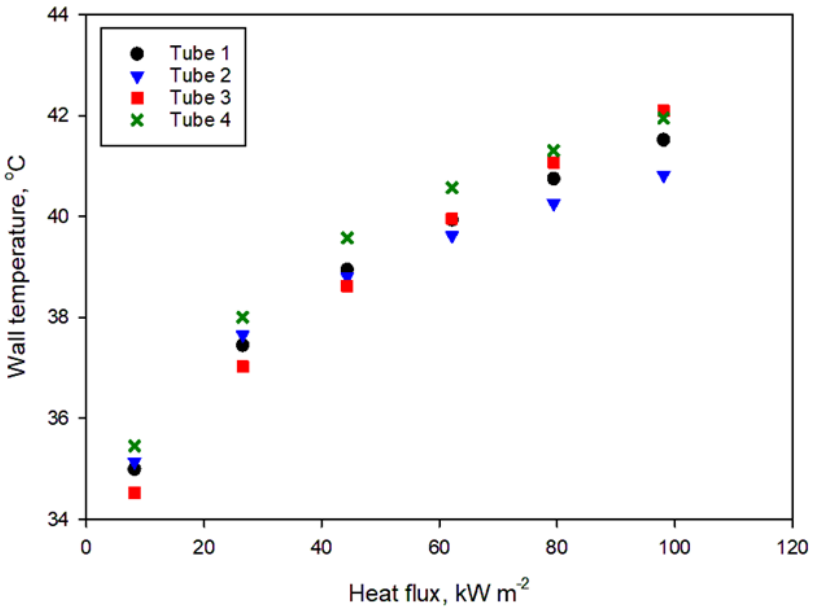
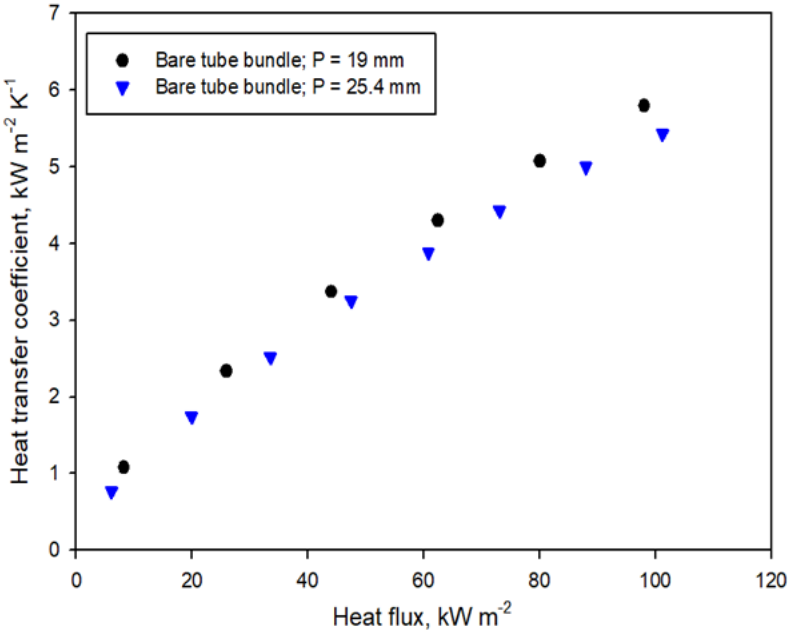


Fig. 6. Wall temperature of different tubes: (a) bare tube bundle and (b) metal-foam enhanced tube bundle.

4.3. Effect of tube pitch on tube bundle performance

To understand the influence of the tube pitch on the heat transfer performance of the tube bundle, two tube pitches, 19 mm and 25.4 mm, are experimentally analyzed for both bare and metal foam tube bundles. The tube pitch effects on heat transfer coefficient are presented in Fig. 7. For both bare and metal foam tube bundles, the heat transfer coefficient value reduces marginally when the tube pitch is increased from 19 mm to 25.4 mm. A maximum of 10% reduction in HTC is observed with an increase in pitch for the bare tube bundle, whereas the same for the metal foam tube bundle is about 14%. In other words, the two-phase flow is more intensive for the tube bundle with lower pitch, and this trend is in agreement with that reported in the work of Fujita and Hidaka [13] and Swain et al. [14]. Since the increased tube pitch creates a larger gap between the tubes, the vapor column leaving the bottom tubes may not shear along the side tubes. Similarly, the vapor column leaving the side tubes may not shear along the top tubes. As a result, the HTC enhancement caused by the convection effects deteriorates and lead to a lower HTC at a larger tube pitch.

(a)



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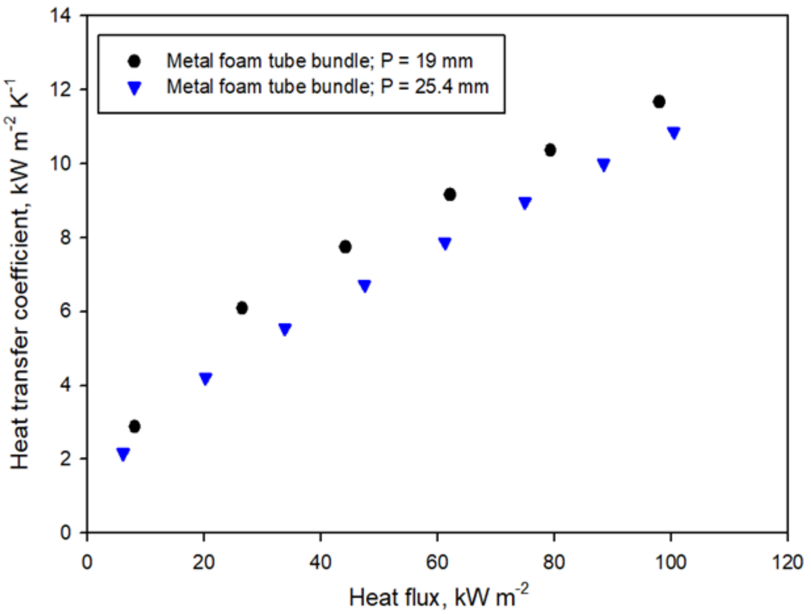

Fig. 7. Effect of tube pitch on heat transfer coefficient (a) bare tube bundle and (b) metal foam tube bundle.

5. Conclusions

In this study, open-cell metal foam tubes are proposed to enhance the shell-side performance of the flooded evaporators. The performance of the bare tube bundle is compared against the proposed metal foam tube bundle under pool boiling conditions. The experiments are carried out for a heat flux range of 5-100 kW m^{-2} and two different tube pitches (19 mm and 25.4 mm). Based on the experimental results, the key outcomes of the study can be outlined as follows:

- The heat transfer coefficient of the bare tube bundle is just 3-5% higher than that of the single tube.
- As compared to the bare tube bundle, the metal foam tube bundle reported a 100-160% enhancement in heat transfer coefficient.
- The wall temperature of the metal foam tubes is about 4-10°C lower than that of the plain tubes due to the combination of increased surface area and nucleation sites, reduced bubble departure diameter, and enhanced liquid suction to the tubes' wall through capillary action.
- The tube pitch played a significant role in determining the pool boiling behavior of the tube bundles. When the tube pitch is increased from 19 mm to 25.4 mm, the heat transfer coefficient values decrease by a maximum of 10% and 14% for the bare and metal foam tube bundles, respectively.

Acknowledgements

The authors are grateful to the colleagues at Oak Ridge National Laboratory who provided useful comments and suggestions to improve the quality of the paper. In addition, the technical support provided by Anthony Gehl, Jeff Taylor, Brian Goins, and Michael Day is greatly appreciated. The authors also acknowledge the support provided by US Department of Energy Building Technologies Office (BTO) and the technology manager, Mr. Antonio Bouza.

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