EXPERIMENTAL INVESTIGATION OF CONVECTIVE BOILING HEAT TRANSFER FOR R-134A FLOW IN METAL FOAM FILLED VERTICAL TUBE

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Abstract

The present work reports an experimental investigation for the convective boiling heat transfer of R-134a in vertical tube filled with metal foam. High porosity (0.95-0.98) with PPI (40-80) metal foams (open-cell) are being considered to improve heat transfer process. Both of hydrodynamic and heat thermal performance are investigated. The results indicate that the metal foams significantly increases both heat transfer coefficient and Nusselt number but at the expense of increasing the pressure drop with mass flux rang 3-40 kg/m\textsuperscript{2}.s. New correlations are proposed to predict the pressure drop and Nusselt number and show good agreements with previous experimental and numerical works.

Keywords: Metal Foam; Forced Convection; Boiling; Experimental Study; Pressure Drop; Vertical Tube

I. Introduction

Interest in using metal foam in heat transfer and fluid flow applications motivate the need for fully characterizing them more imperious. Examples for application include thermally enhanced oil recovery, heaters, compact heat exchangers, silencers, flame arrester, mine heat exchanger, and electrochemical applications and geothermal energy exploitation, \textsuperscript{[xv]} Two-phase flow in tube filled by metal foam has not been investigated thoroughly. \textsuperscript{[XIV]} investigated the enhancement of boiling of R134a, and R1234ze(E) in a 5-PPI copper foam. The experimental measurements studied the effect of changing heat fluxes, mass flux and vapour quality on the heat transfer coefficient, dryout phenomenon, and pressure drop.
at constant saturation temperature of 30 °C; Furthermore, the flow boiling heat transfer was observed using a high speed video camera allowing for a detailed analysis of the experimental results. [I] investigated flow boiling heat transfer inside a horizontal high porosity copper foam with 5 PPI of R1234yf and R1234ze(E). Also, they studied the effect of changing heat fluxes, mass flux and vapour quality on the heat transfer coefficient. The two-phase heat transfer and pressure drop performance of the two new HFO refrigerants was compared against that of the more traditional R134a.

[XVII] investigated experimentally the influence of copper foam (porosity 95 % with 5, 10 PPI) on enhancements of heat transfer for oil mixture and refrigerant convective flow boiling through copper metallic-foam inserted in tubes. The conditions of experimental work with saturation pressure of 995 kPa, heat flux range 3.1 - 9.3 kW/m², mass flux from 10 to 30 kg/ s. m², oil concentration between 0% - 5% and inlet vapor quality ranges 0.175 - 0.775. The results show that the coefficient of boiling heat transfer increased by a 1.85 times, on the other hand, the effect of metallic foam is reduced at oil presence case. A new empirical equation was developed.

[VII] studied the phase-change flow inside a copper mini tube filled by metallic-foam. The test section was visualized by high-speed imaging in glass tubes. Flow pressure drop, the heat transfer coefficient, and pattern maps, are displayed for vapor quality of 0.1–0.7, heat flux between 20–40 kW/m², and mass flow between 400–700 kg/m² s. The test section is a horizontal copper tube with an inner diameter of 4 mm and outer diameter of 6 mm and length is 500 mm”, metallic-foams of Copper were supplied (90% porosity and PPI 20-30). For this range, the coefficient of heat transfer was compared with empty tube, the metallic-foam increased by 320%. Also, the pressure drop is adversely affected by the copper foams inside the channel. [VII] compared experimental data with the developed correlations for metallic-foam-filled tubes. The correlations for both pressure drop and heat transfer coefficient were established by using new method based on [VI]. The new correlations showed a very low error with the literature and experimental data.

[VIII] Investigated experimentally the boiling of two phase flow in metallic-foam filled a plate heat exchanger. The working fluid was zeotropic mixture of R245fa/R134a (molar ratio is 0.6/0.4) and a pure R245fa. The metal foam samples were (20, 30, 60 PPI) with constant porosity. The results indicated that both zeotropic mixture and R245fa refrigerant were presented. Compared to a conventional evaporator, the (20 PPI) foam caused an increased 230% in the coefficient of heat transfer, while for (30 and 60 PPI) the increases were 200% and 130%, respectively. Comparing with the empty channel evaporator, the mixture evaporator filled by foams enhanced the coefficient of convection heat transfer by 230%, 190%, and
128% for metal foam PPI (20, 30, and 60), which is almost identical to increase criteria's for the R245fa. The coefficient of the heat transfer degradation for the mixtures was up to 11% compared with the empty tube and up to 14% compared with 20-PPI foam. Because of low ratio of volume-to-surface-area and unique structure which produced good fluid mixing, which is required for wide range of heat transfer applications. [X] performed the performance of thermal-hydraulic for metal foam filled open cell heat exchanger, four types of metal foams have been measured during closed-loop experiments which was done in a wind tunnel at different mass fluxes. The results showed that for (40 PPI) convection heat transfer coefficient is greater than that for (5 PPI). However, foams with lower PPI produced relatively smaller pressure drop.

Finally, [III], [IV] investigated both natural and forced convection boiling heat transfer for Refrigerant R134a flow in a metal foam filled vertical tube is numerically by using the Multiphase Mixture Model MMM and modified MMM for forced convection at constant heat flux for both transient and steady state behaviour under different parameters. The effect of changing imposed heat flux, porosity, PPI (pore per inch), and tube diameter was analysed. The two phase flow in metal foam is not investigated thoroughly in literature. The aim of the present experimental study is to verify the enhancement of flow boiling heat transfer coefficient of R-134a as working fluid inside vertical tube filled with copper foam and to study the pressure drop across the foam tube.

II. Experimental Study

Experimental Setup

A closed loop was prepared for the experiment, as shown in Fig. 1. The main part of the loop is a refrigerant cycle where a compressor is used to deliver refrigerant to the air cooled condenser then main storage before entering the test section. A R-134a rotameter is installed after the capillary tube. The volume of flow entering the test section is controlled by adjusting the by-pass flow branch. The R-134a rotameter had a measure range 0–60 LPH. After the test section, there is a separator to make remaining liquid flow down then pass through 2nd R-134a rotameter (to measure the evaporation rate through test section). An auxiliary heater is electrically isolated from the rest of the setup and its power is controlled by adjusting the applied voltage to make sure that all refrigerant coming back to compressor is gas phase.

The test section is a vertical copper tube with a diameter of 5/8 in and 7 mm thickness, as shown in Fig. 2. The copper tube length is 140 mm, which is long enough to make sure that the entrance effects are negligible and that the flow is fully developed. three thermocouples are installed on the outer surface of the copper tube at inlet, midpoint, and outlet, with a total of 9 thermocouples. A tape heater is tightly
installed on the copper tube, and the whole system is thermally insulated by woolglass insulation around the tube. The heat flux is controlled by the voltage applied to the wire heater. Before and after the test section there were a pressure gauge to check the pressure loss through it.

Copper metal foams were prepared with 0.95 porosity and 40, 60 and 80 PPI and 40 PPI with 0.96 and 0.98 porosities then cut into cylinders. As shown in plate 2. R-134a refrigerant was used as the working fluid.

Plate (1): The Experimental Apparatus.
Data reduction

Heat Transfer Calculations

Heat Flux

Heat losses from the heater across the insulation layers were calculated to be 3.65%. These losses are subtracted from the electric power to obtain the net heat transfer rate.

In the experiments, the liquid and vapor were separated when they flow out of the test section. The volume of the separated liquid was measured by the graduated flask with time. The flow rate of this amount of liquid is calculated by dividing its volume by the measured time. Such flow rate of the liquid was subtracted from the inlet flow rate of the liquid to give the amount of vapor generated in the system. This mass fraction of vapor is expressed by the vapor quality ($x$) and determined on the basis of the liquid flow rates at both the inlet and outlet of the test section as follows;

\[
x = [1 - (\frac{Vs l}{Vin})] \times 100
\]

(1)

Where;

$Vs l$ = The flow rate of the separated liquid $m^3/s$.

$Vin$ = The inlet flow rate of the liquid to the test section $m^3/s$. 

Fig. (1): Schematic of the Experimental Apparatus.

Fig. (2): Schematic Diagram of the Test Section.
The Reynolds number can be defined according to the length scale $\sqrt{K}$ and the velocity at the inlet of the test section as:

$$Re_K = \frac{\rho v_i \sqrt{K}}{\mu_i} \tag{2}$$

Where:
- $v_i$ = velocity in m/s
- $k$ = Absolute permeability in m$^2$
- $\rho$ = Density in kg/m$^3$
- $\mu_i$ = Dynamic viscosity in kg m/s

For the flow boiling, the dimensionless group, boiling number can be defined as a function to the heat flux, mass flux and latent heat of evaporation:

$$Bo = \frac{q_w}{\rho v_i h_{fg}} \tag{3}$$

Where:
- $q_w$ = heat flux in W/m$^2$
- $v_i$ = velocity in m/s
- $\rho$ = Density in kg/m$^3$
- $h_{fg}$ = Latent heat of liquid/vapor phase change in J/kg

The effect of fluid motion on the heat transfer in enclosure of fluid saturated porous material is evaluated by computing conduction referenced Nusselt number. Hence, the local Nusselt number can be written as [XII]:

$$Nu_Z = \frac{h_Z D_h}{k_{eff}} = \frac{q_w D}{k_{eff}(T_{wZ} - T_{sat})} \tag{4}$$

The average Nusselt number can be calculated by integrating Eq. (4) over the entire length of the heated wall using a numerical integrating method (Simpson’s rule) as:

$$\overline{Nu} = \frac{1}{L} \int_{Z=0}^{L} Nu_Z \, dZ \tag{5}$$

**Pressure drop**

The total pressure difference between the inlet and outlet of the refrigerant side is directly measured by an accurate pressure difference across top and bottom of the tube. Two type of pressure drop should be removed from the total pressure drop; [XIII], [XVI]
1- Pressure drop due to the acceleration of the flow (the two-phase flow)

\[ \Delta P_{\text{acceleration}} = \frac{G^2}{\rho_0} \left( \frac{1}{\rho_0} - \frac{1}{\rho_1} \right) \Delta x \]  

2- Static pressure drop (vertical configuration)

\[ \Delta P_{\text{static}} = gL\rho_{\text{mean}} \]  

The remaining head difference is interpreted as the friction pressure drop:

\[ \Delta P_{\text{friction}} = \Delta P_{\text{total}} - \Delta P_{\text{static}} - \Delta P_{\text{acceleration}} \]  

The experimental conditions are summarized in Table 1. The measurement system was made up a chain of components, each of which was subjected to individual accuracy. The uncertainties of experimental quantities were estimated using the method presented by [IX] (see table 2).

Table 1. Experimental conditions.

| No. | Parameter         | Range                  |
|-----|------------------|------------------------|
| 1   | Working fluid    | R-134a                 |
| 2   | Pressure         | 6 bar                  |
| 3   | Copper foam      | 40-80 PPI, 0.95-0.98 Porosity |
| 4   | Mass flux        | 3-40 kg/m².s           |
| 5   | Heat flux        | 10 – 80 Kw/m²          |

Table 2. Experimental uncertainties

| Independent parameter (e) | Uncertainty (w) |
|---------------------------|-----------------|
| Temperature               | ± 0.25 °C       |
| Voltage                   | ± 0.1 V         |
| Current                   | ± 0.01 A        |
| R-134a flow rate          | ± 0.4 LPh       |
| Diameter                  | ±0.0007         |
| Length                    | ±0.0007         |
| Pressure drop             | ± 1.2 pa        |
III. Results and Discussions

Experimental measurements of refrigerant(R-134a), wall temperatures and pressure drop are presented in this part of the present work to characterize the features of boiling heat transfer in the tubes filled by copper foam flows tested.

Hydrodynamic Performance

Fig. (3) shows the variation of pressure drop per unit length for flow boiling with respect to mass flux for five foam test sections and smooth tube. As expected, the results showed that pressure drop dramatically increases as the mass flux rises due to the friction losses. At the same porosity (95%), the 80 PPI foam produces higher-pressure drop compared to the 60 and 40 PPI. This is due to higher surface area density for this foam compared with 60 and 40 PPI foam. The change in porosity has small effect on the drop of pressure for the 40PPI foam and the curves for 98%, 96% and 95% porosity are close.

Fig. (4) shows the comparison of pressure drop for copper tubes. It can be seen that the pressure drop is higher with foam than in smooth tube by a factor 1.8 for 40PPI and 0.98 porosity, a factor 2-2.15 for 40PPI and 0.95 porosity and a factor 2.9-3.3 for 80PPI and 0.95 porosity. The reason is that in the foam tubes three types of flow resistance affect on the flow; tube surface drag resistance, Darcy viscous resistance and form resistance induced by metal foam structures while in case of smooth tube only the tube surface drag resistance causes the pressure drop.

The pressure drop as a function of Reynolds number based on the permeability is presented in Fig. (5). As shown in the figure the pressure increases as Reynolds number increased from 2.5 to 25 (Forchheimer regime) with same behavior of Fig. (3).

![Graph showing pressure drop versus mass flux for five metal foams and smooth tube.](image)

Fig.(3): The Measured Pressure Drop Versus Mass Flux for Five Metal Foams and Smooth Tube
Fig.(4): Comparison of Pressure Drop for Metal Foam Tubes with Smooth Tube

Fig.(5): The Pressure Drop Versus Reynolds Number for Five Metal Foams

**Nusselt Number and Heat Transfer Coefficient**

Figs. (6) and (7) present the effect of the mass flux on the local heat transfer coefficients along the heated wall for different metal foam types for minimum and maximum mass fluxes 10 and 30 kg/s.m² respectively. It can be seen that the flow regimes for subcooled flow in a metal foam tube can be classified in a manner similar to those for conventional tubes or ducts. First, just past the entrance, a single-phase liquid forced-convection regime exists, which is more obvious at the low heat flux and lowest pore density of metal foam. This is followed by a subcooled boiling regime and then a saturated flow-boiling regime. This phenomenon is obviously
shown in low mass flux (10 kg/s.m$^2$) (Fig. (6)). For 0.95 porosity and 60 and 80 PPI metal foam the heat transfer value increases to maximum value near thermocouple No.2 where the difference between $T_w$ and $T_{sat}$ is very low. This will lead to higher heat transfer coefficient. The reason is that for the low mass flux, the heat transfer coefficient becomes smaller as more vapor is generated and more foam structure and tube wall surfaces are occupied by vapor as the vapor quality rises and this reduces heat transfer capability. As mass flux increased to 30 kg/m$^2$.s (Fig. (7)), the heat transfer coefficient decreases from peak values at the starting of two phase regime (sub boiling regime) and difference between the local wall temperature $T_w$ and the saturation temperature $T_{sat}$ reaches lowest value. The reason is that the high velocity of the vapor at outlet of the tube prevents collection of vapor between the structure of the metal foam and more vapor is generated. The same behavior is occurred for the locale Nusselt number along the heated wall tube as shown in Figs. (8) and (9).

Fig.(6): Local Heat Transfer Coefficient along the Heated Wall for Constant Mass Flux and Different Metal Foams Cell Sizes

G=10 kg/m$^2$.s

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Fig.(7): Local Heat Transfer Coefficient along the Heated Wall for Constant Mass Flux and Different Metal Foams Cell Sizes

Fig.(8): Local Nusselt Number along the Heated Wall for Mass Flux 10 kg/s.m² and Different Metal Foams Cell Sizes.
Figs. (10) and (11) present the effect of the mass flux on the average heat transfer coefficient and Nusselt number respectively on the heated wall for different cell sizes of metal foam and different heat fluxes. It can be seen from these figures that the heat transfer enhances moderately as mass flux increases and more enhancements where the metal foam has lower permeability (high pore density). This tends to homogenize the flow field and enhance heat transfer. This result highlights the interesting heat transfer augmentation capabilities of metal foams during the phase change process.
Fig. (11): Average Nusselt Number with the Mass Flux for Different Metal foam Cell Sizes.

Fig. (12) shows comparison between smooth tube and three metal foam tubes (80 PPI-0.95 porosity, 40 PPI-0.95 porosity and 40 PPI-0.98 porosity) of the overall heat transfer coefficients for different mass flux. It is clear that the use of metal foams significantly enhances the flow boiling heat transfer. The heat transfer coefficients of tube filled with 80PPI and 0.95 porosity metal foam is approximately more 2.7 to 3.1 times those of smooth tube while 0.95 porosity and 40 PPI metal foam is approximately from 1.3 to 2.2 times of smooth tube. Finally, 0.98 porosity and 40 PPI metal foam approximately from 1.3 to 7.2 times of smooth tube. As mentioned previously the mixing properties of the foam structure associated to the high shear stress. Liquid and vapour are highly mixed and the fluid streams are deviated by the foam’s fibers through a tortuous path; in this way, the convective contribution is extremely enhanced.
Mass Fraction Vapour

In the experiments, the liquid and vapour were separated when they flow out of the test tube. The separated liquid was measured by R-134a rotameter for each run. The measured value of the liquid was subtracted from the inlet flow rate of the liquid, and the result would be the vapour generated in the system. As shown in Fig. (13), the vapour quality had an inverse proportionality with mass flux for different types of metal foams. Generally, a smaller mass flux and presence of metal foam suggests a longer residence time during which liquid remains in the test tube. The fluid absorbs more heat, leading to the generation of more vapour. As the cell sizes of the metal foam is increased (higher PPI and lower Porosity), more heat absorbs and more vapour generates because the surface of heat exchange is increased with presence of metal foam.
Empirical Correlations

Friction Factor Correlation

In the present study, the homogeneous frictional pressure gradient was calculated based on all liquid Reynolds number with respect to the permeability $Re_K$. The suggested model for the pressure gradient calculation was calculate via the friction factor $f_K$ as a function of the experimental permeability values $[\mu]$, as;

$$f_K = \frac{\Delta P}{\frac{\mu}{\rho_0 \nu^2}}$$

(10)

The friction factor variations of water with Reynolds number for different test tubes filled with different metal foams are shown in Fig. (14). The friction factor test data have been correlated as follows:

$$f_K = \frac{59.939}{Re_K^{0.664}}$$

(11)

Nusselt Number Correlation

Notably, there are no well-established evaporation heat transfer correlations for copper foam in the literature. The experimental Nusselt number defined by eq. (5,6) was correlated using the Reynolds number (accounts for the inertia and viscous forces), the liquid boiling number (account the heat and mass flux effect) and the Prandtl number. The porosity and pore size effect are included via the permeability and definition of $Re_K$. To correlate the experimental data of the current study, the function form given by eq. (12a) was proposed. The constant and the exponents in eq. (12b) are determined using the multi-parameters nonlinear least square fitting and the final form of the correlation is given by eq. (12c).
The above equation is valid for R-134a as a working fluid, mass flux range (8 – 40 kg/sm²), and copper foam with specification porosity range (0.95- 0.98) PPI=40-80. Fig. (15) illustrates how the above dimensionless groups managed to correlate the current experimental data with insignificant scattering.

\[ \text{Nu}_{\text{Exp.}} = C \text{Re}_k^{a} \text{Bo}^b \text{Pr}^c \]  
\[ \text{Nu}_{\text{Exp.}} = 5.75 \text{Re}_k^{0.631} \text{Bo}^{0.21} \text{Pr}^{0.74} \] 

\( f_k = 59.939 / \text{Re}_k^{1.494} \)

Fig.(14): Friction Factor Empirical Correlation

Fig.(15): The Experimental Nusselt Number Correlation of (Gholamreza et al.2017).
Verification of the Results

In this section, a verification of the experimental results was achieved by comparing the present and previous experimental work has been made for the pressured drop and heat transfer coefficient. Also, comparisons between the numerical algorithm from pervious works and experimental results of the present work have been made for the values of the local Nusselt number.

A. Comparison with Previous Experimental Work

Fig. (16) presents a comparison of the pressure drop with mass flux for the present work (0.95 porosity and 40 PPI), the work of [II] (40PPI and 91 % porosity) with working fluid water and the work of [XI] (40 PPI and porosity 95 %) with the working fluid N-pentane flowing through a vertical channel filled with copper foam. The three curves have the same trend and behavior, as mass flux is increased, the pressure drop increases. The main difference in the pressure drop of three works is due to the work conditions which are different, like the nature of N-pentane other than water and the different values of viscosity and density.

![Comparison of experimental Pressure Drop with Mass Flux](image)

Fig.(16): Comparison of experimental Pressure Drop with Mass Flux between the Present Work, the Work of (Ali Lefta .2015) and the Work of (Madani et al.2010).

Fig. (17) shows a comparison of the experimental results of the mean heat transfer coefficient of the present work with the work of [VIII], who investigated experimentally the flow boiling of R245fa/R-134a through a vertical channel filled with 60 PPI copper foam. At first, the test rig was run for several times to get the value of mean heat transfer coefficient at high mass flux range to compare it with [VIII]. It can be seen that the behavior of increasing the heat transfer coefficient with
heat flux is the same. Also the difference in values is due to difference in working pressure, different geometry, also different in porosity, effective thermal conductivity and the work conditions such as the nature of working fluid (different values of viscosity and density).

Fig. (17): Comparison of the Mean Heat Transfer Coefficient with Mass Flux between the Present Work, (Gholamreza et al.2017).

Comparison of Numerical and Experimental Results of the Present Work

Figure (18) presents a comparison between the numerical model of [IV] and the experimental local Nusselt number of present for 0.95 and 0.98 porosity and 40PPI metal foam. It is clear from the figure that values are close together for small heat flux at the first part of the tube and intersect with each other before the exit section of the foam tube. It is clear from the figure that for higher heat fluxes and with the onset of boiling, the values of the experimental and numerical local Nusselt number are deviate from each other towards the tube exit section with relatively small difference in the location of the boiling starting point. This can be attributed to the fact that for the two-phase flow the local Nusselt number is no longer defined from the boiling onset point to the tube exit section and it was calculated in the experimental part from the tube inlet section to the point on the tube wall that exist downstream of the boiling onset point, while in the numerical part the local Nusselt number was computed from the tube inlet section to the point just before the boiling onset section, and as a result, a difference was observed in the location of the boiling starting point. Another reason is that the contact thermal resistance between tube wall and metal foams has significant influence on the overall heat transfer performance.
Fig. (18): Comparison Between the Experimental and Numerical Local Nusselt Number Along the tube Filled with 40PPI with 0.95 and 0.98 porosity Copper Foam and for Different Heat Flux

The comparison of vapour quality with mass flux, between the numerical modelling, measured in experimental part (0.98 porosity and 40 PPI) and the work of [II] for water as working fluid (40PPI and 0.91 porosity) is presented in Fig. (19). It can be seen that the three curves have the same behaviour and vapour quality decreases as the mass flux increase. The comparison also shows that experimental results are higher than numerical modelling with low relative average error. The reason may be due to the surface roughness which was high and the type structure of the copper foam which is different slightly in practical. All that makes the vapour generated experimentally higher. The large difference between experimental test of the present work and [II] is due to difference in working fluid, heat flux and test condition.
V. Conclusion

In the present work, an experimental investigation was done for refrigerant (R-134a) forced convection boiling heat transfer in a vertical tube filled with metal foam. The following are the conclusions drawn from the present investigation;

I. As mass flux increases the pressure drop is increased, and will be higher for low permeability (high PPI), and is lower affected by porosity.

II. Inserting metal foams inside tube will leads to considerable enhancement in heat transfer coefficient for 80 PPI and 0.95 porosity 270 to 310 % and for 0.95 porosity and 40 PPI is 130 to 220 % compared with smooth tube.

III. Depending on the imposed heat flux, both values of the local Nusselt number and convection heat transfer coefficient increased slightly and reached its maximum value before the exit with the onset of boiling, depending on the values of the imposed heat flux.

IV. Both Nusselt number and convection heat transfer coefficient increase moderately as mass flux increases and increase more when the metal foam has lower permeability (high pore density).

V. A high mass fraction is expected when the inlet mass flux decreases and more vapor generated as the cell sizes of metal foam is increased.
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