Experimental analysis of flow boiling pressure drop through copper metallic foam

B. MADANI[a], F. TOPIN[b], L. TADRIST[b]

a. LTPMP, FGMGP, USTHB, BP 32, El Alia, Alger (Algérie).
b. I.U.S.T.I, Polytech'Marseille, Université de Provence, 5 Rue Enrico Fermi Marseille (France).

Résumé:

Ce travail concerne l'ébullition convective lors d'un écoulement ascendant dans un canal rectangulaire rempli de mousse métallique brasée aux parois. La mousse métallique dans le présent travail est en cuivre avec 36 PPI de grade et 97% de porosité. Le fluide de travail est le n-pentane. Les variables d'entrée sont le débit massique qui varie de 10 à 120 kg/m^{2}s et la densité de flux thermique qui varie entre 0 et 25 x 10^4 W/m^2. Les données recueillies sont la pression statique et le titre à la sortie du canal. Les résultats hydrodynamiques sont exprimés en termes du multiplicateur de friction diphasique et comparés à ceux obtenus par l'utilisation des modèles homogène et à phases séparées (corrélation de Friedel) appliqués aux écoulements diphasiques dans un tube lisse.

Abstract

This work deals with the convective boiling during an upward flow inside a rectangular channel filled with metallic foam soldered to lateral walls. The metallic foam sample tested in the present work is made from copper with 36 PPI and 97% of porosity, and the used coolant fluid is n-pentane. The input parameters are the mass flux in the range from 10 to 120 kg/m^{2}s and the heating power with values between 0 and 25 x 10^4 W/m^2. The output data include static pressure and exit mass quality. The hydrodynamic results are compared to those obtained using the multiplier parameter given by both homogeneous and separate flow (Friedel correlation) models for two-phase flow in a plain tube.

Key words: convective boiling, pressure drop, metallic foam, two-phase flow, phase change, porous media.

1. Introduction

Porous media are widely used to enhance heat transfer in thermal exchangers. However, the insertion of these materials increases the pressure drop which in turn inflates the cost of the plumbing and pumps needed for the functioning of the majority of thermal processes. Nowadays, open cell metallic foams which constitute a class of porous media are increasingly used in the heat and mass exchangers. This is due to the opportunities offered in terms of weight and hydrodynamic characteristics, namely: porosity and permeability which are larger than 0.88 and 10^{-7} m^2, respectively. Figure 1 shows photos of a metallic foam sample taken with a Scanning Electrons Microscope (SEM) for three zoom levels (50 X, 150 X and 350 X).

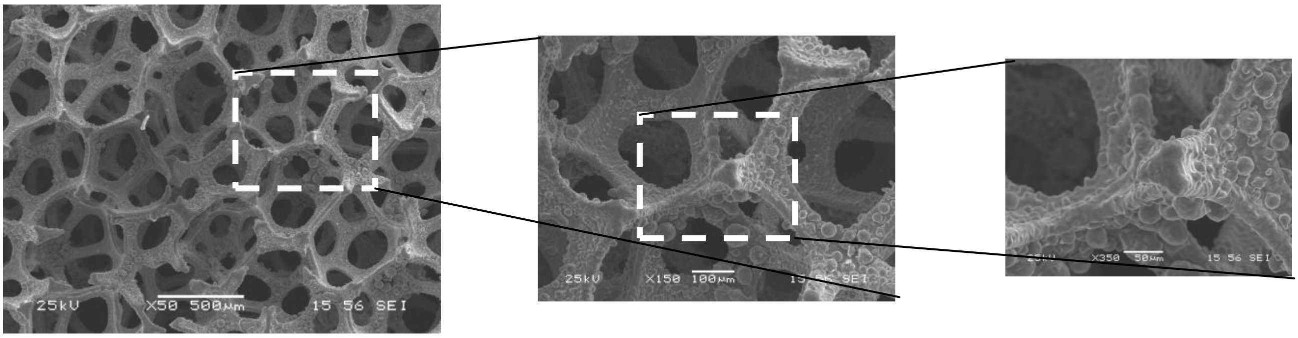

FIG. 1 SEM photos of a metallic foam sample with three zoom levels.

Works on the two-phase flow pressure drop through channels filled with metallic foam began at the last decade of the last century. The analysis of the literature gives that metallic foam made from nickel-chrome, copper and aluminum are the most commonly used. Noting that the experimental approach is the main mean used in investigations. The deficient in direct simulation on two-phase flow pressure drop in metallic foam is flagrant. This is due in one hand to the fact that the experimental database is not yet sufficient and in the other hand, to the lack in morphological studies dealing with this material. The mixture of water and air is largely used as working fluid in the works on adiabatic two-phase flow [1-7], except for [3] where glycerol and propanol are added in moderate proportions to the mixture of air-water. For this regime of flow, two configurations are studied: firstly, co-current upward flow [3], down-flow [3, 4] and horizontal flow [2, 6, 7]; secondly, counter- current upward flow [1,5]. Certain investigations are accompanied by visualizations [5, 6] and numerical simulations [5, 6]. To explore the experimental results, a number of parameters are used: liquid hold up, two-phase friction multiplier and residence time distribution. Unlike adiabatic regime, studies on pressure drop in metallic foam carried in diabatic regime are lesser [8, 9, 10]. In these studies, only copper metallic foams were used, whereas varied working fluid are utilized: R135 in horizontal flow [8], n-pentane in upward flow [9] and water ionized horizontal flow [10]. Although the obtained results in the works cited above give useful information, even correlations [10], the available data remains still insufficient to establish universal model for two-phase pressure in metallic foams and hence numerical simulation of the physical phenomenon.

2. Experimental Facilities

The experimental set-up consists of three main parts: a test section, a fluid loop and a data acquisition system. The set-up used in the present study is similar to that in [9]. The difference between the present experimental set up and the previous one is that, in this case the channel as shown in Fig. 2(b) is 10 (Width) x 50 (Length) x 100 (Height) mm^3 with metallic foam sheets soldered to the walls. Thus only 10 cartridges heaters with P = 250 W/unit are used to heat the lateral walls.

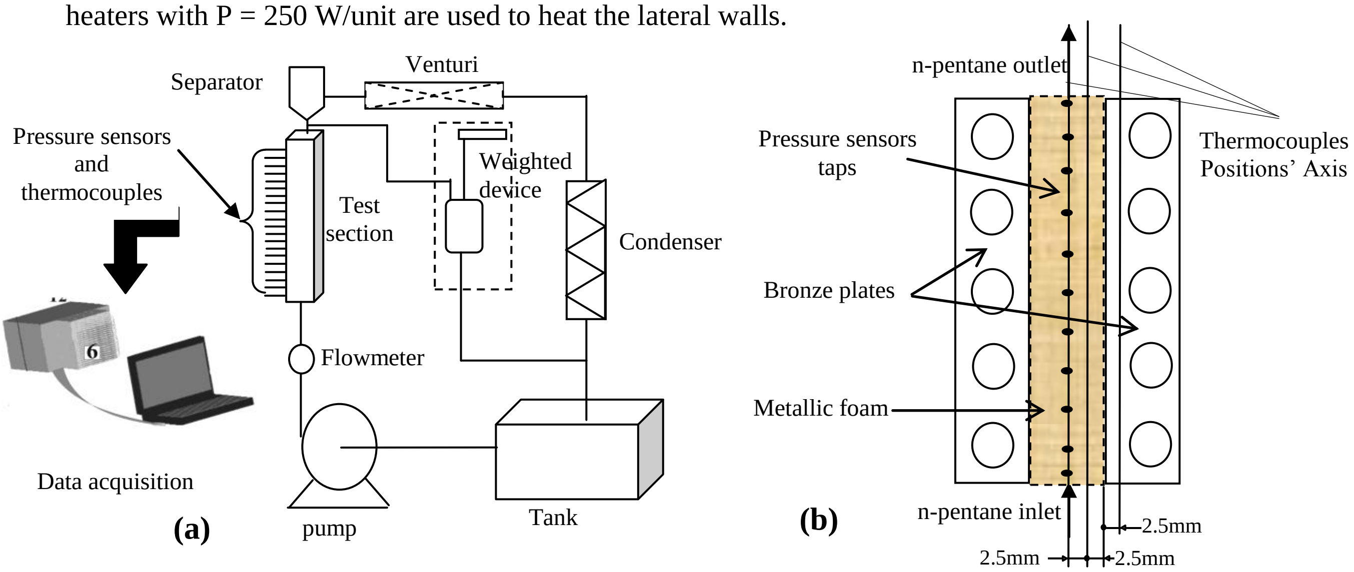

FIG. 2 Experimental set up for boiling experiment (a) the experimental loop (b) the test section

As in the previous facilities the channel is provided with forty thermocouples (type-K, diameter 0.5 mm) implanted in the walls and in the foam along the axis a, b, c as shown in the Fig (2-b), uniformly spaced 10 mm apart. The sensitivity of these thermocouples is 41 μV/°C. The Pressure is measured using Sensym pressure sensors which are attached to needles (d = 0.5 mm). These are located in the foam and uniformly spaced 10 mm apart along the central axis (Fig. 2-b). The sensors' range is from 0 to 1.03 bar and their sensitivity is about 1.7 μV/Pa. The inlet test section flow rate is monitored using two flowmeters turbine-type for optimal accuracy over the full range of experiments. The first flowmeter works between 0.2 *l*/min and 2 *l*/min with a sensitivity of 2.5 V for 1 *l*/min, while the second covers the range from 1 *l*/min to 40 *l*/min with a sensitivity of 1.2 V for 1 *l*/min. The measurement of liquid mass flow rate both upstream the test section and coming to the weighted tank allows the determination of the exit mass vapor quality.

3. Experimental results

Figure 3 shows a typical pressure signal given by the sensor located at z = 0.06 m for E = 1600.87 W, G = 5.71 kg/m^{2}s and T_{in} = 33.13 °C. The mean value, standard deviation, as well as uncertainty related to this measure of pressure are also mentioned in this figure. The pressure loss along the channel for a heating power Q = (16 ± 1) x 10^4 W/m^2 and for several mass velocities is given in Fig. 4. Note that the exit mass quality is given for each mass velocity and the indicated power Q concerns only that one injected in the two-phase zone.

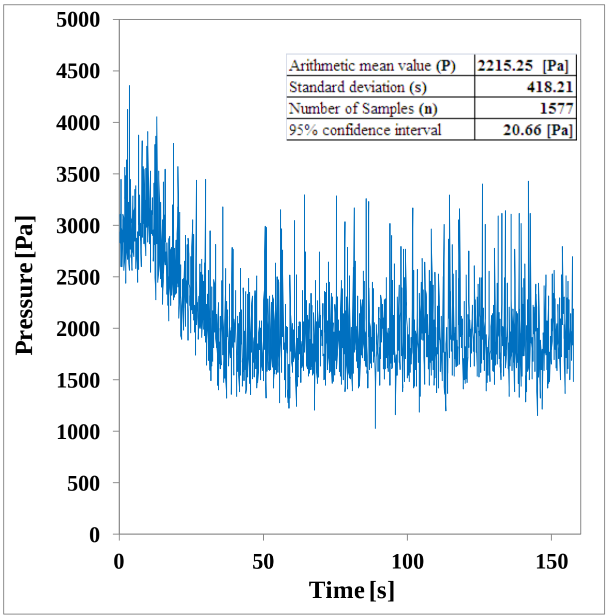

FIG. 3 Typical pressure signal

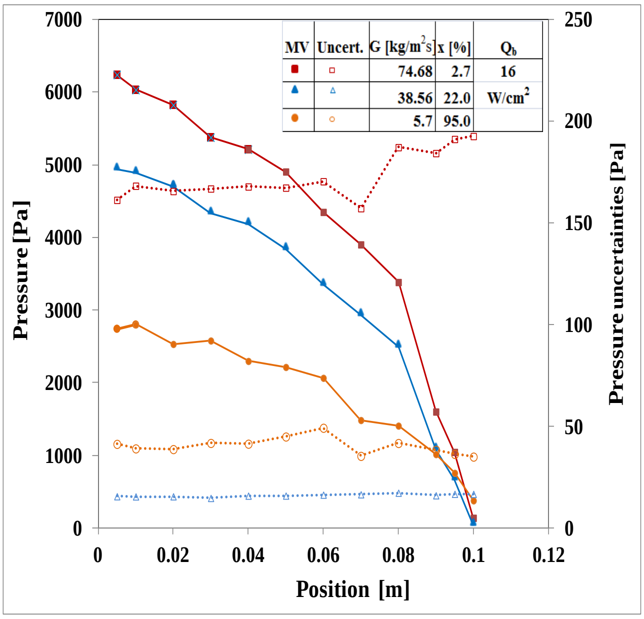

FIG. 4. Pressure loss along the channel for several exit qualities and mass velocities (MV : Mean Value).

Note that the curvature of the plots is due to the flow acceleration caused by the vapor generation. As the figure shows, the acceleration phenomenon is present even for the weak exit mass qualities. Furthermore, the figure gives uncertainties linked to each experimental point. As one could deduce from the figure, the pressure fluctuations and the gradient of this one are more influenced by the mass velocity than the exit quality.

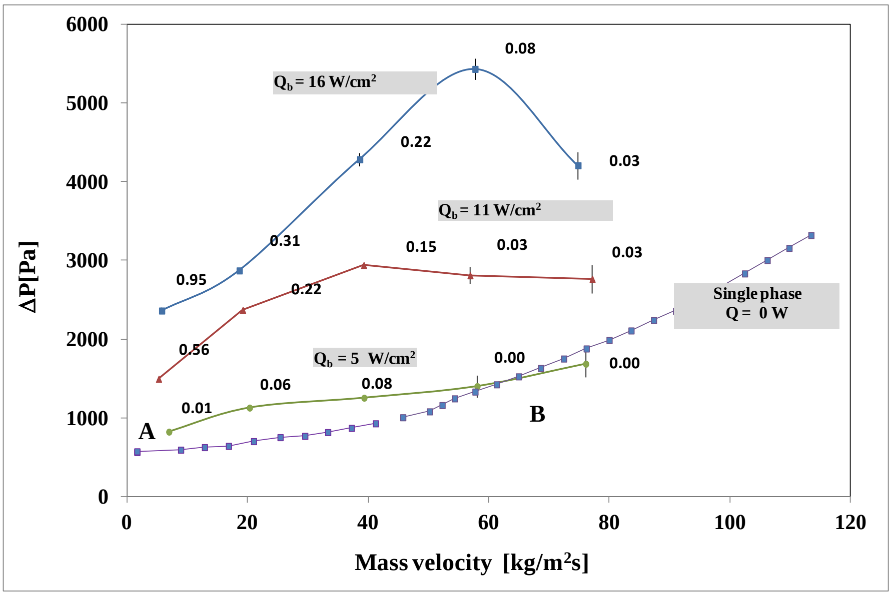

FIG. 5. Pressure loss *Vs* mass velocities for several heating powers.

Pressure loss function of mass velocity is given in Fig. 5 for several diphasic heating powers. The choice of heating power values allows the analysis of the pressure loss evolution for three levels of heat: high, intermediate and low. As one could note, the shape of the pressure loss for heating power null is parabolic while the curves for the two-phase mode have a different shape. They are of a higher order and look like the S-shape. The pressure loss curve for heat power $Q = 5 \times 10^4$ W/m^2 increases monotonously with an inflexion point at mass velocity $G = 40$ kg/m^{2}s, the start and the end parts of this curve approach the single phase one. The two extremities of the 5×10^4 W/m^2 curve mark the disappearance of liquid in point A and vapor in point B. For this last point the heat power is insufficient to keep boiling in the channel. However, we note that the curve for the heat power $Q = 11 \times 10^4$ W/m^2 is away from the single phase curve, meaning that the applied power in this case is sufficient to maintained the boiling regime over the tested mass velocity domain. Note that for this heating power, the pressure loss curve increases up to mass velocity $G = 40$ kg/m^{2}s and remains constant to join the single phase curve at mass velocity not tested here. The pressure loss curve for $Q = 16 \times 10^4$ W/m^2 presents a particular shape. The curve increases until a max value at mass velocity $G = 60$ kg/m^{2}s then decreases towards the single phase curve. As mentioned previously the applied power in this case is not sufficient to maintain boiling for $G > 60$ kg/m^{2}s.

Uncertainties on experimental points in Fig.(5) are given in Figs.(6, 7). Uncertainty on pressure drop (Fig. 6) is a mean value estimated on the basis of uncertainties related to each pressure gages implanted along the channel. Plots show that the boiling regime amplifies uncertainties by a factor higher than 10, compared to those for a single phase regime, this factor is proportional to mass velocity. Uncertainties on mass velocities (Fig. 7) are obtained from a statistical analysis of flowmeter output signal. For the majority of measurement points up to 60 kg/m^{2}s, in boiling regime uncertainties are less amplified, about 5 times compared to the single phase regime.

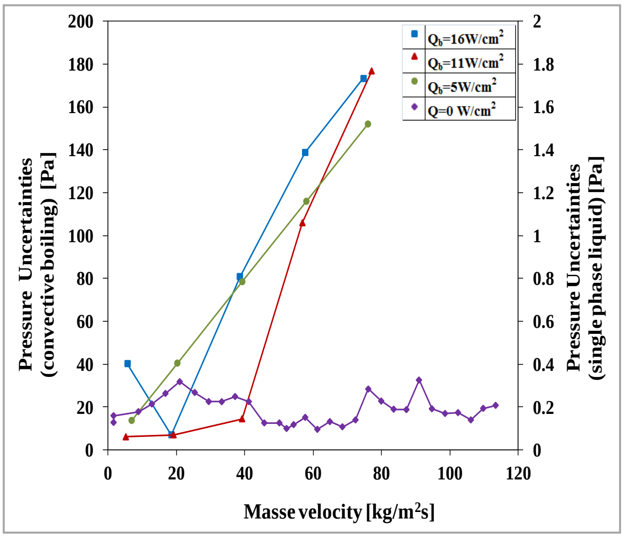

FIG. 6. Uncertainty on pressure drop (figure 5) for both single phase and two-phase flow.

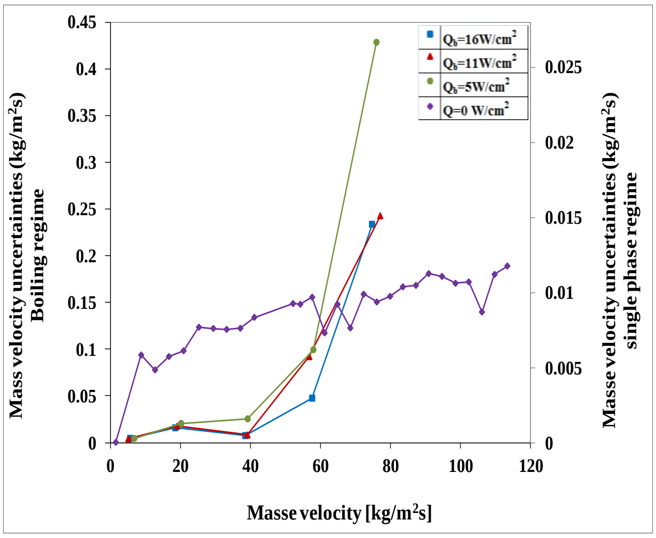

FIG. 7. Uncertainty on mass velocity (figure 5) for both single phase and two-phase flow.

4. DISCUSSION AND COMPARISON WITH THE LITERATURE

Since results relating to boiling in porous media are not yet agreeing, the experimental results obtained in the present work are compared to those given for convective boiling in plain tubes.

First, recall that the total pressure loss in two-phase flow is constituted from three components: friction, acceleration and gravity. Colier and Thome [11] developed a simple 1D formulation to describe these three components. In their development, the friction term $(\partial P/\partial z)_f$ is not explicitly defined as the two other terms. Among others, the homogenous and separate models are frequently used to express this component in terms of single phase pressure loss and the parameter Φ^2_{LO} named: the two-phase friction multiplier. This last parameter is given by the Eq.(1). Wallis [12] develops the Eq. (2) for the homogeneous flow. Whalley [13], on the basis of a comparative study and for $(\mu_l/\mu_v) < 1000$ and G < 2000 kg/m^{2}s, recommended the use of Friedel [14] correlation (Eq. 3) for the separate flow. Note that the Whalley's conditions are largely verified in the present work.

$$\left(\frac{\partial P}{\partial z}\right)_f = \Phi_{\mathrm{LO}}^2 \left(\frac{\partial P}{\partial z}\right)_{LO} \qquad (1) \qquad\qquad \Phi_{\mathrm{LO}}^2 = \left(1 + x\frac{\rho_l - \rho_v}{\rho_v}\right)\left(1 + x\frac{\mu_l - \mu_v}{\mu_v}\right)^{-1/4} \qquad (2)$$

$$\Phi_{\mathrm{LO}}^2 = \mathrm{E} + \frac{3.24\,\mathrm{F\,H}}{\mathrm{Fr}_h^{0.045} We_l^{0.035}} \qquad (3)$$

Table 1 gives the dimensionless numbers E, F, H, Fr_h and We_l. The liquid and vapor friction factors are determined using the results given in [15], where it is shown that the single phase flow in metallic foam is mainly inertial and follows the Forcheimer law, $\Delta P/L = \alpha u + \beta u^2$, with $\Delta P/L$ as pressure gradient, α viscosity factor and β inertial factor

Table 1. Dimensionless number used in Friedel correlation

$$E = (1-x)^2 + x^2\frac{\rho_l f_v}{\rho_v f_l} \qquad H = \left(\frac{\rho_l}{\rho_v}\right)^{0.91}\left(\frac{\mu_v}{\mu_l}\right)^{0.19}\left(1 - \frac{\mu_v}{\mu_l}\right)^{0.7} \qquad F = x^{0.78}(1-x)^{0.224}$$

$$We_l = \frac{G^2 d_i}{\sigma\,\rho_h} \qquad\qquad Fr_h = \frac{G^2}{g\, d_i \rho_h^2}$$

The mass quality profile along the channel is taken linear fixed on the exit mass quality Eq. (4).

$$x = \frac{z - z_{dz}}{L_{dz}} x_{out} \qquad (4)$$

The experimental two phase friction multiplier compared to homogeneous and separate flow model (Friedel correlation) is given in Fig. 8. Note that the experimental Φ_{LO}^2 is calculated on the basis of the measured total pressure drop. Uncertainties on all experimental points are also given in this figure. One could see that the experiment results are qualitatively compatible with those given by the models. The parameter Φ_{LO}^2 increases for the three methods as mass quality increases. However, for quality near unity, we can see that the Friedel correlation presents a net decrease of Φ_{LO}^2, physical phenomenon that is not reproduced neither by the experiment nor by the homogeneous model. Nevertheless these two last methods present a significant decrease of Φ_{LO}^2 gradient when x is near unity.

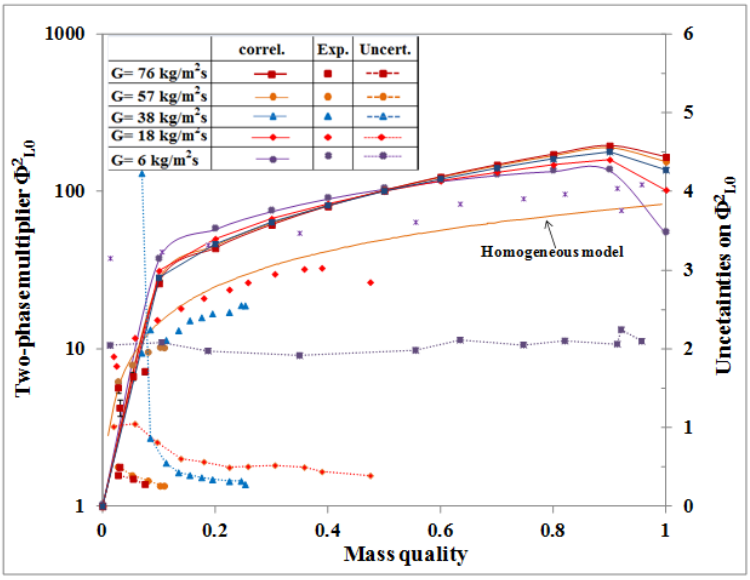

Fig. 8. Experiment two-phase flow multiplier Φ_{LO}^2 compared to the two-phase flow models (a) homogenous model (b) separate flow model (Friedel correlation).

The application of Friedel correlation shows that for mass quality lower than 60 % the parameter Φ_{LO}^2 decreases when mass velocity increases, whereas for x upper than 60 %, the two-phase multiplier and velocity evolve in the same way. This tendency is clearly reproduced by the experimental measurement in the part where x < 60%. The maximum discreapancy between the experimental results and those given by the homogeneous model stands for ± 25 %. The discrepancy concerning Friedel correlation is between 50 % and 90 %.

5. Conclusion

This experimental study deals with convective boiling pressure loss in channel filled with metallic foams brazed on the lateral walls. The measured pressure profiles make obvious the effects of the flow acceleration. The homogeneous model results are more convenient than those given by the correlation of Friedel. The discrepancy of the first model is less than 25 %, whereas the second model presents more than 50 %.
The results obtained in this work show that the hydrodynamic conditions (flow patterns, fluid properties,..) created by the metallic foam insert in normal conditions of pressure and flow velocity are similar to those, which may be obtained in the case of plain tube, but in severe operating conditions.

Nomenclature

d diameter [mm]
E heating power [W
f friction coefficient
Fr Froude number
LO liquid only
P pressure [Pa]
u superficial velocity [m/s]

We Weber number
x quality
z ordinate [m]

Greek Symbols
μ viscosity [Pa.s]
ρ density [kg/m^3]
σ superficial tension [N/m]

Subscripts
dz two-phase
i hydraulic
h homogeneous
l liquid
v vapor

References

[1] Stemmet, CP., Jongmans, J. N., Schaaf, J., Kuster, B. F. M., Schouten, J. C., Hydrodynamics of gas-liquid counter-current flow in solid foam packings, Chem. Eng. Sci., 60, 6422-6429, Elsevier, 2005.
[2] Topin, F., Bonnet, J. P, Madani, B., Tadrist, L., Experimental analysis of multiphase flow in metallic foam: flow laws, heat transfer and convective boiling. Adv. Eng. Mat., 8, 890- 899, 2006.
[3] Stemmet, C. P., Bartelds F., Schaaf J., Kuster B. F. M., Schouten J. C., Influence of liquid viscosity and surface tension on the gas-liquid mass transfer coefficient for solid foam packings in co-current two-phase flow, Chem. Eng. Res. Design, 86, 1094-1106, 2008.
[4] Edouard D., Maxime L., Pham C., Mbodji M., Pham-Huu C., Experimental measurements and multiphase flow models in solid SiC foam beds, AIChE Journal, 54, 11, 2823- 2832, 2008.
[5] Calvo, S., D. Beugre, M., Crine, A., Léonard, P., Marchot and D., Toye, Phase distribution measurements in metallic foam packing using X-ray radiography and micro-tomography, Chem. Eng. Proc. : Process intensification, 48, 1030-1039, 2009.
[6] Gerbaux, O., Vercueil, T., Memponteil, A., Bador, B., Experimental characterization of single and two-phase flow through nickel foams, Chem. Eng. Sci., 64, 4186-4195, 2009.
[7] Bonnet, J.P., Topin, F., Vincent, J., Tadrist, L., Cocurrent gaz-liquid flow in metal foam: an experimental investigation of pressure gradient, Journal Porous Media, 13, 497-510, 2010.
[8] Lu, T. J., Investigation of heat transfer in metal foam tubes, PHD thesis, Brunel University, 2008.
[9] Madani, B., Topin, F., Tadrist, L., Convective boiling in metallic foam: Experimental analysis of the pressure loss, Fluid Dynamics Materials Processing, 6, 351-368, 2010.
[10] Ji X., Xu J., Experimental study on the two-phase pressure drop in copper foams, Heat Mass Transfer, 48, 1, 153-164, 2012.
[11] Collier J. G, Thome, J. R., Convective boiling and condensation, Clarendon Press Oxford, 1994.
[12] Wallis G. B., One dimensional two-phase flow, Mac Graw Hill, 1969.
[13] Whalley, P. B.,(1980) see Hewitt, G. F., Multiphase flow and pressure drop, heat exchanger design handbook, Hemisphere, Washington, DC, Vol. 2, 2.3.2-1, 1983.
[14] Friedel, L., Improved friction pressure drop correlations for horizontal and vertical two-phase pipe flow, European Two-phase Flow Group Meeting, Ispra, Italy, Paper E2, 1979.
[15] Madani, B., Topin, F., Rigollet, C, Tadrist, L., Flow laws in metallic foams: Experimental determination of inerial and viscous contributions, Journal Porous Media, 10, 51-70, 2007.