Assessment of boiling heat transfer and pressure drop correlations of ammonia/water mixture in a plate heat exchanger

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ABSTRACT

Plate heat exchangers are gaining acceptance in absorption refrigeration systems due to their high transfer rates, compactness, and low refrigerant charge. Nevertheless, boiling heat transfer and pressure drop studies with mixtures in plate heat exchangers (PHEs) are scarce.

In this study, the experimental data of Táboas et al. (2010) on flow boiling of ammonia/water in a plate heat exchanger are compared with the predicted values using the correlations available in the open literature for the boiling heat transfer coefficient and pressure drop. In addition, this study proposes a new correlation based on a separate model by which to obtain the boiling coefficient. The new correlation uses a transition criterion, divided into an apparent nucleate boiling region where pure convective boiling cannot appear, and a region with competition between convective and apparent nucleate boiling. The correlation proposed can predict 98% of data from Táboas et al. (2010) within 20% error.

Keywords: Absorption cycle; Boiling; Heat transfer; Plate exchanger; Pressure drop Ammonia-water

Nomenclature

Bo	boiling number, $Bo = q'' G^{-1} h_{fg}^{-1}$
D_h	hydraulic diameter, $D_h = 2 \cdot b$, *m*
f	friction factor, dimensionless
F	convective enhancement factor, dimensionless
G	mass flux, $kg\ m^{-2}\ s^{-1}$
h	heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
h_{fg}	latent heat, $J\ kg^{-1}$
k	thermal conductivity, $W\ m^{-1} K^{-1}$
L	flow length of the plate, *m*
Nu	Nusselt number, $Nu = h \cdot D_h^{-1} \cdot K^{-1}$, dimensionless
P	pressure, *Pa*
Pr	Prandtl number, $\mathrm{Pr} = \mu \cdot Cp \cdot K^{-1}$, dimensionless
q''	heat flux, $W\ m^{-2}$
Re	Reynolds number, $\mathrm{Re} = G \cdot D_h \cdot \mu^{-1}$
S	nucleate boiling suppression factor, dimensionless
U_{SV}	superficial velocity of vapour, $m\ s^{-1}$, dimensionless
U_{SL}	superficial velocity of liquid, $m\ s^{-1}$, dimensionless
x	vapour quality

Greek symbols

β	corrugation angle
ρ	density, $kg\ m^{-3}$
Λ	corrugation pitch

φ_l two phase friction factor multiplier

υ_m mean specific volume of two phase flow

Subscripts

nb nucleate boiling

cb convective boiling

lo liquid only

l liquid

TP two phase flow

pool pool boiling

eq equivalent

m mean

v vapour phase

1. Introduction

In recent years there has been a growing demand for small capacity ammonia/water air-cooled absorption chillers driven by low temperature (between 80 and 120ºC) heat sources such as waste heat or solar energy. The development of this kind of chillers under these conditions, operating with a single-stage absorption cycle, requires advanced high performance components for the heat and mass transfer processes.

Several researchers (Kan et al., 1998; Lee et al. 2002) have recommended PHEs for the main components of ammonia/water absorption systems in order to enhance heat and mass transfer processes, but these studies have focused on the absorber performance. There are limited studies on desorbers with ammonia/water as the working fluid. Roriz et al. (2004) used a PHE with ammonia/water, in an absorption machine, but their study was focused only on the COP of the absorption machine. Recently Táboas et al. (2010), experimentally obtained pressure drop and heat transfer boiling coefficients for a PHE with the ammonia/water mixture. Considering other geometries, Rivera and Best (1999), Khir et al. (2005a,b) studied the flow boiling mixture in vertical tubes. Although some new developments of ammonia/water absorption machines use a PHE as desorber, the aforementioned literature review clearly reveals that there is limited information currently available in open literature on boiling heat transfer characteristics with ammonia/water.

A survey of the literature shows that there has been increasing experimental work on PHE evaporators, but most of this work concentrates on pure refrigerants (e.g. R22, R134a, R142b, ammonia, propane) or near azeotropic refrigerants (e.g. R410a, R407c) because of the environmental requirement to phase out synthetic chlorofluorocarbons (CFC). Recently, Wang et al. (2007) and Palm and Claesson (2006) presented reviews on thermal and hydraulic performance of PHE evaporators. However, no data have been found in the open literature by the authors of the present study relating to boiling heat transfer and pressure drop in plate heat exchangers with wide boiling range mixtures as the ammonia/water mixture.

Since the beginning of flow boiling research, saturated flow boiling heat transfer has been recognized as the result of two contributions to the heat transfer coefficient, nucleate boiling and convective boiling. In general, the heat transfer coefficient in nucleate boiling is dependent on heat flux, while in convective boiling, heat transfer is mainly dependent on the mass flux and the vapour quality. It is still not clear from the studies presented in the literature whether the saturated flow boiling heat transfer coefficient in plate heat exchangers is governed by nucleate or convective boiling mechanism, since both of them have been reported in the literature depending on the operating conditions. Most authors acknowledge that the main mechanism observed was nucleate boiling (Panchal et al., 1983; Engelhorn and Reinhart, 1990; Osterberger and Slipcevic, 1990; Kumar, 1992; Pelletier and Palm, 1997; Palm and Claesson, 2006; Longo and Gasparella, 2007), others consider only convective effects on the boiling coefficient (Margat et al., 1997; Han et al., 2003), and some consider that both effects are important in the boiling process (Hsieh and Lin, 2003; Táboas et al., 2010).

The literature review showed that fewer experimental studies have been done on pressure drop than on heat transfer in PHEs. The experimental studies agree that the frictional pressure drop of evaporation increases by increasing the vapour quality and the mass velocity, but decreases increasing the system pressure.

For this work, the data of Táboas et al. (2010) obtained with flow boiling of ammonia/water in a PHE is used. The data is analysed with several correlations proposed in the literature to predict the saturated boiling heat transfer coefficient and pressure drop in plate heat exchangers.

For the prediction of evaporation pressure drop, three methods have been suggested in the literature. Probably, the most widely used model for pressure drop prediction in two phase flow in PHEs and other geometries is the Chisholm (1967) model. This model was used in PHEs by Palm and Claesson (2006), Sterner and Sunden (2006), Ouazia (2001), Margat et al. (1997). The second approach to correlate frictional evaporation pressure drop in PHEs was proposed by Longo et al. (2004) and Jassim et al. (2001). These authors proposed a linear equation based on the kinetic energy per unit volume of refrigerant flux. The third approach is that reported by the Yan and Lin (1999) and Hsieh and Lin (2002, 2003). These authors correlated the pressure drop data in terms of the equivalent Reynolds number.

For heat transfer calculations, the correlations considered include: nucleate boiling correlations (as proposed by Palm and Claesson, 2006; Longo and Gasparella, 2007; Pelletier and Palm, 1997; Engelhorn and Reinhart, 1990); correlations found in the open literature for circular tubes (as proposed by Kumar, 1984; Hsieh and Lin, 2003; Djordjevic and Kabelac, 2008); and specific correlations obtained with evaporation experiments in plate heat exchangers (as proposed by Hsieh and Lin, 2002; Han et al. 2003; Yan and Lin, 1999; Margat et al., 1997; Donowski and Kandlikar, 2000).

1.1. Heat transfer correlations proposed in the literature for PHEs

Palm and Claesson (2006) carried out flow boiling experiments with R22 and R134a in different PHEs and concluded that all data could be fitted to Cooper's pool boiling correlation by introducing a constant factor of 1.5. The authors also found that the chevron angle did not have a major influence on heat transfer.

Longo and Gasparella (2007) presented the experimental heat transfer coefficient measured during refrigerant R-134a vaporisation inside a small PHE and the set of saturated boiling heat transfer coefficients were compared with the Cooper (1984) and Gorenflo (1997) pool boiling correlations with a mean absolute percentage deviation of around 8.2% and 12.3%, respectively.

Pelletier and Palm (1997) tested and compared R22 and hydrocarbons (propane, two commercial propane based mixtures and propene) in a small laboratory heat pump with a PHE as evaporator. The experimental results were compared with pool boiling correlations (Cooper, 1984; Gorenflo, 1997; Stephan and Abdelsalam, 1980), correlations for tube-bundle evaporators (Slipcevic, 1988; Gorenflo, 1997), and correlations for flow boiling (Pierre, 1969; Steiner and Taborek, 1992). The authors concluded that the three correlations tested for pool boiling may in most cases be useful to estimate boiling heat transfer coefficients while the other correlations over-predict them.

Engelhorn and Reinhart (1990) tested a plate evaporator with R22 as working fluid. The experiments were performed with and without a distributor for the liquid refrigerant at the entrance. The results achieved for the case with a liquid distributor and saturated conditions of the refrigerant at the evaporator outlet were compared with the predicted values of two correlations for pool boiling in tube bundles, the correlations of Gorenflo (1997) and Slipcevic (1988). It was concluded that the deviation between experimental and predicted values was within 25% and 10%, respectively.

Kumar (1984) suggested a Chen-type correlation for predicting evaporation heat transfer coefficient for refrigerants. The working fluids in the related experiments were R22 and ammonia. In the proposed correlation, the boiling coefficient is the result of the addition of the contribution from nucleate boiling and the contribution from two-phase convective boiling term as shown in equation (1). The nucleate boiling coefficient is calculated as the product of a pool boiling coefficient and a suppression factor, which is a function of flow conditions. The two-phase convective boiling term is the product of the single-phase convective coefficient and an enhancement factor which is always greater than unity. It has been found that this enhancement factor varies greatly with the corrugation angle and is a function of the Lockhart-Martinelli parameter. However no information is given in the work reported by Kumar (1984) regarding these terms.

$$h = h_{nb} \cdot S_{nb} + h_{cb} \cdot F_{cb} \tag{1}$$

Margat et al. (1997) investigated the evaporation of refrigerant R134a in a single channel of a compact PHE. They concluded that heat transfer in this process was independent of the heat flux and strongly dependent on the vapour quality, which indicated that the effect of nucleate boiling was not significant. The heat transfer coefficient was correlated according to equation (2).

$$h = h_{lo} \cdot F_{TP} \tag{2}$$

where h_{lo} is the liquid heat transfer coefficient assuming all the fluid flows as liquid, and is calculated through the specific correlation for corrugated channels. The enhancement factor F_{tp} was correlated using the Chisholm two-phase multiplier Φ_l^2 calculated as a function of the Lockhart-Martinelli parameter:

$$F_{TP} = \left(\Phi_l^2\right)^{0.5} = \left(1 + \frac{C}{X} + \frac{1}{X^2}\right)^{0.5} \tag{3}$$

where Margat et al. (1997) proposed $C=3$.

Hsieh and Lin (2003) performed experiments of R410a evaporation in a vertical PHE with a single channel arrangement for the refrigerant side to avoid maldistribution phenomena. Based on experimental data, the authors proposed the Gungor and Winterton (1986) correlation to calculate the corresponding heat transfer coefficient in terms of convective and nucleate boiling contributions as shown in equation (4):

$$h = h_{pool} \cdot S + h_l \cdot E \tag{4}$$

where h_l and h_{pool} are respectively given by the Dittus and Boelter (1985) correlation and Cooper (1984) pool boiling correlation. The term S represents the nucleate boiling suppression factor and the term E the convective boiling enhancement factor. These terms are dependent on the boiling number Bo, the Lockhart-Martinelli parameter X, and liquid Reynolds number Re_l. The expressions suggested for E and S were:

$$E = 1 + 24000 \cdot Bo^{1.16} + 1.37 \cdot \left(\frac{1}{X}\right)^{0.86} \tag{5}$$

$$S = \left(1 + 1.15 \times 10^{-6} \cdot E^2 \cdot \mathrm{Re}_l^{1.17}\right)^{-1} \tag{6}$$

Djordjevic and Kabelac (2008) obtained experimental results on heat transfer for flow boiling of ammonia and of R134a in a PHE. The experimental data was compared with the predicted values using Steiner and Taborek (1992) correlation for vertical tubes. The authors concluded that this correlation has to be scaled down by a factor of 0.51 to match the experimental data points.

The literature survey showed that most of researchers' work done on evaporation in PHEs has suggested specific correlations that were claimed to be only valid for their specific situations. For instance, Hsieh and Lin (2002) in their experiments performed with R410a as refrigerant, suggested the following correlation:

$$h = h_{lo} \cdot \left(88 \cdot Bo^{0.5}\right) \tag{7}$$

where h_{lo} is the single-phase heat transfer coefficient assuming all fluid flow as liquid. The single-phase heat transfer coefficient is determined from an empirical correlation obtained by the authors. The boiling number is obtained from equation (8):

$$Bo = \frac{q^{"}}{G \cdot h_{fg}} \tag{8}$$

In the same set-up, Yan and Lin (1999) carried out an evaporation test using R134a as refrigerant. The experimental data showed that both the evaporation heat transfer coefficient and the pressure drop increase with vapour quality. Based on their experimental data the authors correlated the heat transfer coefficient according to equation (9):

$$\left(\frac{h \cdot D_h}{k_l}\right) \cdot \Pr_l^{-1/3} \cdot \mathrm{Re}^{0.5} \cdot Bo_{eq}^{-0.3} = 1.926 \cdot \mathrm{Re}_{eq} \tag{9}$$

where Re_{eq} and Bo_{eq} are, respectively, the equivalent Reynolds and Boiling numbers in which an equivalent mass flux is used in their definitions first proposed by Akers et al. (1958):

$$\mathrm{Re}_{eq} = \frac{G_{eq} \cdot D_h}{\mu_l} \quad \text{and} \quad Bo_{eq} = \frac{q''}{G_{eq} \cdot h_{fg}} \tag{10}$$

where

$$G_{eq} = G \cdot \left[\left(1 - x_m\right) + x_m \cdot \left(\frac{\rho_l}{\rho_v} \right)^{1/2} \right] \tag{11}$$

Donowski and Kandlikar (2000), using the experimental data reported by Yan and Lin (1999) also developed improved correlations for single-phase and two-phase heat transfer coefficients for evaporation of refrigerant R-134a. The authors proposed a model for correlating flow boiling heat transfer coefficient in a PHE based upon the previous work of Kandlikar (1991):

$$h = \left[3.312 \cdot Co^{-0.3} \cdot E_{CB} + 667.3 \cdot Bo^{2.8} \cdot F_{fl} \cdot E_{NB} \right] \cdot \left(1 - x\right)^{0.003} \cdot h_l \tag{12}$$

The convective number is defined as:

$$Co = \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \cdot \left(\frac{1 - x}{x} \right)^{0.8} \tag{13}$$

The factors E_{CB} and E_{NB} are the multiplying enhancement factors to the convective boiling and nucleate boiling contributions, respectively.

Fitting the correlation to the experimental data, the values obtained were:

$$E_{CB} = 0.512 \text{ and } E_{NB} = 0.338 \tag{14}$$

The fluid-dependent parameter, F_{fl}, was taken to be 1.0 for the stainless steel plate.

Han et al. (2003) performed experiments on the evaporative heat transfer with refrigerants R410a and R22 in PHEs with different 45°, 35° and 20° chevron angles. Based on their experimental data, the authors established the following empirical correlation for Nu of the tested PHEs, including the geometric parameters.

$$Nu = Ge_1 \cdot \mathrm{Re}_{eq}^{Ge_2} \cdot Bo_{eq}^{0.3} \cdot \mathrm{Pr}^{0.4} \tag{15}$$

$$Ge_1 = 2.81 \cdot \left(\frac{\Lambda}{D_h}\right)^{-0.041} \cdot \left(\frac{\pi}{2} - \beta\right)^{-2.83} \tag{16}$$

$$Ge_2 = 0.746 \cdot \left(\frac{\Lambda}{D_h}\right)^{-0.082} \cdot \left(\frac{\pi}{2} - \beta\right)^{-0.61} \tag{17}$$

where Ge_1 and Ge_1 are non-dimensional geometric parameters that involve the corrugation pitch (Λ), the hydraulic diameter and the chevron angle. Re_{eq} and Bo_{eq} are the equivalent Reynolds and Boiling numbers, respectively, defined above. The value of the exponent of the Boiling number was set to 0.3 as proposed, previously, by Yan and Lin (1999) for PHEs.

1.2. Pressure drop correlations proposed in the literature for PHEs

As was stated earlier, the model that is most often used to correlate the data for PHEs and other geometries is that suggested by Chisholm (1967) in equation (18):

$$\varphi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{18}$$

The Lockhart-Martinelli parameter X being defined as in equation (19) and the two-phase friction multiplier for liquid φ_l as in equation(20):

$$X^2 = \frac{\Delta P_l}{\Delta P_v} \tag{19}$$

$$\varphi_l^2 = \frac{\Delta P_{TP}}{\Delta P_l} \tag{20}$$

where ΔP_l and ΔP_g are the liquid and vapour frictional pressure drops, respectively, calculated considering that each phase flows alone in the PHE. In practice, however, the values of C presented in the literature are quite different between authors. Thonon (1995) reported C=8, although no information on the experiments or the channel geometry was included. However a value of C=3 was suggested by the same group based on the experimental work of Margat et al. (1997).

Sterner and Sunden (2006) used ammonia in four commercial plate heat exchangers. The Chisholm constant was estimated for the different heat exchangers as a function of the Reynolds number, and may vary from 1 to 100. The results clearly deviate from other works and the authors conclude that the discrepancy may be due to flow maldistribution between the channels of the commercial heat exchangers, or the existence of oil (which increases the pressure drop). Finally, Claesson (2004) suggests a single value of C=4.67, based on measurements of adiabatic flow of R-134a, even though the value of C for the individual points varied between 2 and 15.

The second model presented in the literature to calculate frictional pressure drop in PHE is based on the kinetic energy, KE, per unit volume is defined in equation (21):

$$\frac{KE}{Volume} = \frac{G^2}{2 \cdot \rho} \tag{21}$$

Longo and Gasparella (2007) presented experimental data of pressure drop measured during HFC 134a, 410a and 236fa vaporization inside a small PHE. The following fitting equation was derived for all experimental data using the homogenous model to compute the kinetic energy per unit volume:

$$\Delta P_f = 1.49 \frac{KE}{Volume} \tag{22}$$

In the third model reported in the literature, the evaporation frictional pressure drop is calculated with a two-phase friction factor, which is correlated in term of an equivalent Reynolds number as in equation (24), being G_{eq} obtained by (25):

$$\Delta P_f = f_{TP} \cdot \frac{2 \cdot G^2 \cdot \upsilon_m \cdot L}{D_h} \tag{23}$$

$$\mathrm{Re}_{eq} = \frac{G_{eq} \cdot D_h}{\mu_l} \tag{24}$$

$$G_{eq} = G \cdot \left[(1-x) + x \cdot \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \right] \tag{25}$$

This method was used by Yan and Lin (1999) and Hsieh and Lin (2002, 2003), who performed evaporation experiments with R134a and R410a in a vertical PHE channel.

Hsieh and Lin (2003) presented a simple expression to correlate the two-phase friction factor during evaporation of refrigerant R-410 in a vertical plate heat exchanger.

$$f_{tp} = 2.382 \times 10^4 \cdot \mathrm{Re}_{eq}^{-1.12} \quad \text{for } 2000<\mathrm{Re}_{eq}<6000 \text{ and } 0.0002<Bo<0.0020 \tag{26}$$

and in Hsieh and Lin (2002) the correlation was:

$$f_{TP} = 6.1 \times 10^4 \cdot \mathrm{Re}_{eq}^{-1.25} \tag{27}$$

Yan and Lin (1999), based on their experiments on the evaporation pressure drop of refrigerant R-134a, proposed two correlations depending on equivalent Reynolds number

$$\text{for } Re_{eq}<6000 \qquad f_{TP} \cdot \mathrm{Re}^{0.5} = 6.947 \times 10^5 \cdot \mathrm{Re}_{eq}^{-1.109} \tag{28}$$

for $Re_{eq} \geq 6000$ $$f_{TP} \cdot \mathrm{Re}^{0.5} = 31.21 \cdot \mathrm{Re}_{eq}^{-0.04557} \quad (29)$$

Han et al. (2003) also used this approach but included the geometric parameters of the PHE as the authors of the present study also did in the *Nu* correlation. The correlation presented is as follows:

$$f_{tp} = Ge_3 \cdot \mathrm{Re}_{eq}^{Ge_4} \quad (30)$$

$$Ge_3 = 64710 \cdot \left(\frac{\Lambda}{D_h} \right)^{-5.27} \cdot \left(\frac{\pi}{2} - \beta \right)^{-3.03} \quad (31)$$

$$Ge_4 = -1.314 \cdot \left(\frac{\Lambda}{D_h} \right)^{-0.62} \cdot \left(\frac{\pi}{2} - \beta \right)^{-0.47} \quad (32)$$

2. Experimental results versus empirical correlations prediction

The saturated flow boiling heat transfer coefficient and the two-phase pressure drop were measured in an experimental set-up described in Táboas et al. (2010) and experimental data was presented to show the effects of heat flux between 20 and 50 kW.m^{-2}, mass flux between 70 and 140 kg m^{-2} s^{-1}, mean vapour quality from 0.0 to 0.22 and pressure between 7 and 15 bar, for a series of data of an ammonia mass fraction ranging from 0.35 to 0.62. The original experimental data reported by Táboas et al. (2010) is compared in the present work with several correlations published in the open literature. The last section contains a proposal for a new correlation that takes into account the different trends observed experimentally.

2.1 Pressure drop

The experimental pressure drop in flow boiling of the ammonia/water mixture in the PHE channel was correlated following the three correlation models introduced in the literature review: Chisholm correlation, correlations based on equivalent Reynolds number and finally a linear equation based on the kinetic energy per unit volume of refrigerant flow.

Due to its flexibility, the Chisholm correlation was used for pressure gradient prediction, in a variety of channels of irregular geometry. The drawback of this correlation is that single-phase pressure drop should be obtained previously. In cases where this information is available, only a constant is required. Therefore, the relationship between friction factor and Reynolds number is required in order to calculate the single-phase pressure drop. While there is a fairly large amount of literature dealing with the determination of single-phase pressure drop in PHEs, there is little consensus between the different sets of experimental results and correlation proposed. As a consequence, the fitting of Chisholm constant gives quite different results depending on the correlation of friction factor used. This could explain the differences of Chisholm’s constant found in the literature.

In Táboas et al. (2010) the relationship between friction factor and Reynolds number were established during single-phase experiments with water. As a result, the correlation that best fitted throughout the experimental data was:

$$f = 4.779 \cdot \mathrm{Re}^{-0.118} \tag{33}$$

With this correlation, and using the homogenous model to predict the void fraction, the pressure drop was successfully correlated with the Chisholm constant C of equation

(34), equal to 3. This value is in the same range as the results obtained from recent works. Margat et al. (1997) reported values of C=8 and C=3.3 for PHEs with a chevron angle of 30° and 67.5°, respectively. In experiments with air/water flowing in a PHE channel, Tribbe and Muller-Steinhagen (2001) found C values between 3.5 and 15, depending on the plate geometry and hydraulic diameter. The calculated two-phase frictional pressure drop from equation (34) was compared with experimental data from Figure 1.

$$\varphi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{34}$$

In order to analyse the influence of a single-phase friction factor correlation, the Chisholm constant C of equation (34) was fitted using the single-phase pressure drop correlations of Muley and Manglik (1999), Wanniarachchi et al. (1995) and Kumar (1984). The Chisholm constant yields results of 16.8, 18, and 11.5, respectively. The differences are quite significant, and therefore, to calculate the two-phase pressure drop according to the Chisholm model, it is necessary to have a good prediction of the single-phase friction factor.

The use of kinetic energy per unit volume to characterize two-phase flow pressure drop is very easy to apply because it is only required to know the two-phase flow density. Figure 2 shows the two-phase frictional pressure drop of ammonia/water mixture against the kinetic energy of the solution flow per unit volume calculated by the homogenous model. The relationship between the frictional pressure drop and the kinetic energy per unit volume is quite linear as was already reported by Longo et al. (2004) and Jassim et al. (2001). The following best fitting equation has been derived from the experimental data:

$$\Delta P_f = 0.33 \cdot \frac{KE}{Volume} \tag{35}$$

The constant value of 0.33 is very close to that of Jassim et al. (2001) who reported a constant value of 0.5335 for evaporation of refrigerant R134a in a PHE, whereas the value reported by Longo et al. (2004) for refrigerant R134a, R410A and R236fa evaporating inside a small PHE was 1.49.

Finally, Han et al. (2003), Hsieh and Lin (2002, 2003), Jokar et al. (2006) and Yan and Lin (1999) proposed the correlation of the two-phase friction factor in terms of the equivalent Reynolds number. The comparison of these correlations presented above with the experimental results shows that the two-phase friction factor did not appear to be strongly correlated with the equivalent Reynolds number, as can be observed in Figure 3.

Consequently, the pressure drop prediction with these correlations for the data of Táboas et al. (2010) is poor compared with the other models. To illustrate this, Figure 4 compares the experimental data against the predicted result with the best fit obtained with the correlation proposed by Hsieh and Lin (2002):

$$f_{TP} = 23799.8423 \cdot Re_{eq}^{-1.25} \tag{36}$$

The prediction results are summarized in Table 1.

The use of the kinetic energy per unit volume has the main advantage of its easy calculation, and has the disadvantage of a lack of consensus with the constants obtained by different authors. The use of the equivalent Reynolds number is not recommended, at least for the ammonia/water data from Táboas et al. (2010) because it does not provide a good prediction of pressure drop. Considering the three approaches to calculating the pressure drop in PHEs, we recommend the Chisholm (1967) correlation to calculate the pressure drop because it gives a better prediction of the pressure drop and the constant is in the order of magnitude of other experiments that were obtained with different PHEs and different fluids. This correlation will be further used to calculate the convective term in the heat transfer correlation proposed.

2.2 Heat transfer coefficient versus nucleate boiling correlations

The boiling process in PHEs has been mostly obtained for evaporators with refrigerant fluids. The operating conditions of evaporators are characterized by a low mass flux, thus many authors reported a dominant influence of nucleate boiling.

According to Collier and Thome (1994) the correlation of Cooper (1984) is the most suitable for nucleate boiling of refrigerants in these geometries. Pelletier and Palm (1997), Palm and Claesson (2006) and Longo et al. (2004) reported good agreement between experimental data and the original Cooper correlation, mainly if no superheating is produced in the evaporator.

However, the Cooper (1984) correlation was developed for pure fluids and the ammonia/water is a mixture with a wide boiling range. Therefore, it seems more reasonable to compare the experimental data with a nucleate boiling correlation that takes into account the mixture effects. Recently, Táboas et al. (2007) proposed a new correlation for calculating the pool boiling heat transfer of ammonia/water mixture. Based on the experimental data of Inoue et al. (2002a,b) and Arima et al. (2006), the correlation combines the well known approaches of Schlünder (1982) and Thome and Shakir (1987). The combination is particularly effective in the case of ammonia/water because the variation of $|x\text{-}y|$ and ΔT_{bp} with composition is very asymmetric. As a consequence, if only one of the two approaches is used to calculate the nucleate boiling of the mixture, great deviations may arise in compositions near the pure components.

In Figure 5 the experimental data are compared with the calculated values using the Cooper (1984) and the Táboas et al. (2007) correlations. To calculate the heat transfer coefficient with the Cooper correlation, the mixture was considered as pure fluid with the properties of the mixture. As can be seen, there are large deviations between the experimental results and the values calculated from correlations. Cooper's correlation under-predicts most of the results. This means that convective boiling plays an important role in most of the experimental results, at least under the conditions in which the experiments were performed. The Táboas et al. correlation (2007) gives an even worse prediction, because the calculated pool boiling coefficient is lower than that calculated by the Cooper correlation. The renewal of the diffusion boundary layer at the liquid-vapour interface could improve the heat transfer process, resulting in heat transfer coefficients higher than those predicted by nucleate boiling correlations for mixtures.

2.3 Heat transfer coefficient versus in-tube boiling correlations

As was mentioned above, different authors proposed in-tube boiling heat transfer correlations to calculate the boiling coefficient in plate heat exchangers. The experimental results were compared with the correlations presented above, namely Hsieh and Lin (2003), who proposed the Gungor and Winterton (1986) equation with the single-phase heat transfer coefficient given by the Dittus-Boelter (1985) equation and Djordjevic and Kabelac (2008), which suggested the correlation of Steiner and Taborek (1992), applying a scaling down factor of K=0.51, but considering a single-phase correlation specific for PHEs. In the present work the correlation of Kumar (1992) was used.

Figure 6 compares the experimental results with these two correlations. The Hsieh and Lin (2003) equation with the single-phase heat transfer coefficient given by the Dittus-Boelter (1985) equation under-predicts the boiling heat transfer coefficient. This behaviour is to be expected, considering the conclusions of other authors, who observed a higher boiling coefficient than in tubes (Kumar, 1992; Yan and Lin, 1999; Palm and Claesson, 2006). In the case of the Djordjevic and Kabelac (2008), correlation and the scaling down factor of K=0.51, there is also a tendency to over-predict the experimental results. The best fitting is achieved when the correlation is scaled down by a factor of 0.41, lower than the value of 0.51 obtained by Djordjevic and Kabelac (2008) with ammonia in a plate with similar chevron angle. Kumar (1992) postulated that the bubble generation promotes more turbulence than the corrugation angle, and thus the difference on the heat transfer coefficient between single phase-flow and two-phase flow in PHEs is smaller than for tubes. As a consequence, the enhancement factor of the convective boiling term in PHEs should be smaller than in tubes. This has also been observed by Ohara et al. (1990) and Margat et al. (1997).

If a flow boiling correlation for vertical tube is considered, as was proposed by Hsieh and Lin (2002), with a single-phase correlation for tubes, the prediction results are below the experimental data, but again a scaling down factor is required to correlate the data in a single-phase correlation for PHEs. The two correlations' prediction performances are shown in Table 2.

2.4 PHE heat transfer specific correlations

Most of the authors that have analysed flow boiling in PHEs have proposed specific correlations based on their data. Depending on the main mechanism observed, the specific correlations proposed in the literature are nucleate boiling oriented or convective boiling oriented. The specific correlations for nucleate boiling dominant experimental data are less common in practice because most authors used the Cooper (1984) correlation directly. The specific nucleate boiling correlation for PHEs considered in this study is that of Danilova (1981). Regarding the convective boiling dominant experimental data, the specific correlations proposed are Donowski and Kandlikar (2000), Feldman et al. (2000), Yan and Lin (1999), Hsieh and Lin (2002) and Han et al. (2003). In addition, the Hsieh and Lin (2002) and Han et al. (2003) correlations take into account the effects of heat flux in the boiling coefficient.

The comparison of experimental data with specific empirical correlations developed from the boiling experiments in PHEs presented above is summarized in Table 3. As can be seen a good agreement with the experimental data is only found for the correlations of Donowski and Kandlikar (2000), with nearly 90% of data within 30% of error and an average error of 3.2%, and Han et al. (2003), with 86% of data within 30% error and an average error of 3.2%.

However, it can be seen that this correlation does not match at all the trends of the boiling curves for the experimental data. To illustrate these differences, Figure 7-10 show the trends of the predictions for selected data at different mass and heat fluxes. The Donowski and Kandlikar (2000) correlation predicts well the mass flux effect in the convective zone (Figure 7, obtained with a constant heat flux of 30 kW m^{-2}, and mass fluxes of G=50 to 140 kg m^{-2} s^{-1}), but under-predicts the heat flux effect in the nucleate boiling zone (Figure 8 obtained with a constant mass flux of G=100 kg m^{-2} s^{-1}, and heat fluxes q”=20 to 50 kW m^{-2}). On the other hand, the correlation of Han et al. (2003) fits better the experimental data for different heat fluxes (Figure 10), but shows a poor behaviour for different mass fluxes (Figure 9). In addition, as can be seen in Figure 10, the experimental data are less sensitive to heat flux than the correlation of Han et al. (2003), highlighting the need to modify the boiling exponent of this correlation.

Han et al. (2003) suggested that the strong turbulence of flow inside the PHE reduces the effect of the nucleate boiling even in the low quality and low mass flux region. As a consequence, the exponents of boiling number in the correlations of Nusselt number for the PHE should be lower than the typical value of 0.7 for plain tubes. Han et al. (2003) reported a value of 0.3 and Hsieh and Lin (2002) a value of 0.5. Nevertheless, it should be noted that these experimental works were conducted with pure components or near azeotropic mixtures while ammonia/water is a wide boiling range mixture. The preferential evaporation of the more volatile ammonia component is an additional factor in degrading the mixture nucleate heat transfer coefficient. As a result, there is an additional resistance due to mass transfer (diffusion) that could justify a reduction of the boiling number exponent.

2.5 Proposed saturated flow boiling correlation for ammonia/water in PHEs

In the experimental results presented by Táboas et al. (2010) different trends on the boiling coefficient were observed. For mass fluxes of G=100 kg $m^{-2}s^{-1}$ and higher, there was a transition from nucleate boiling to convective boiling such as can be observed in flow boiling in tubes. For mass fluxes of G=50 and 70 kg m^{-2} s^{-1} the boiling coefficient is insensitive to the vapour quality. This trend is typical of a nucleate boiling dominant process. However, the data show a mass flux effect on the boiling coefficient.

Based on these observations, a criterion has been established in order to decide whether only apparent nucleate boiling is present or whether a competition between convective and apparent nucleate boiling occurs. This criterion is based on the superficial velocities of vapour and liquid in the heat exchanger.

If the vapour velocity is lower than that obtained by equation (37), only nucleate boiling is present, and if it is higher, the heat transfer coefficient is the highest of the calculated nucleate and convective boiling term:

$$u_{SV} = -111.88 \cdot u_{SL} + 11.848 \tag{37}$$

being u_{SL} and u_{SV} obtained by equations (38), and (39), respectively

$$u_{SL} = \frac{G \cdot (1-x)}{\rho_l} \tag{38}$$

$$u_{SV} = \frac{G \cdot x}{\rho_g} \tag{39}$$

In the apparent nucleate boiling region the heat transfer coefficient increases with the imposed mass flux. This can be attributed to the fact that the mass flux could increase the mass diffusion coefficient at the boundaries of the bubble interface, improving the nucleate boiling coefficient. To take into account the mass flux effect in this region, a correlation with the single-phase liquid fraction heat transfer coefficient, $h_{L,}$ and the boiling number, *Bo*, is proposed, modified from that of Hsieh and Lin (2002).

$$h_{TP} = 5 \cdot Bo^{0.15} \cdot h_{lo} \tag{40}$$

When convective boiling occurs, (no heat flux effect on the boiling coefficient) the convective boiling heat transfer coefficient is very often expressed by equation (41).

$$h_{CB} = F \cdot h_{lo} \tag{41}$$

where *F* is the two-phase convective enhancement factor.

To calculate the enhancement factor, Bennett and Chen (1980) proposed the use of a two-phase multiplier of Chisholm. Margat et al. (1997) successfully used this enhancement factor in plate heat exchangers. The enhancement factor proposed here responds to equation (42).

$$F = \frac{h_{TP}}{h_l} = \left(\Phi_l^2\right)^{0.2} = \left(1 + \frac{C}{X_{tt}} + \frac{1}{X_{tt}^2}\right)^{0.2} \tag{42}$$

For the Chisholm's constant, the same value of C=3 is selected as was obtained with the pressure drop experiments. As mentioned above, the value of this constant is the same as that obtained by Margat et al. (1997) but the exponent of the two-phase multiplier of the Chisholm correlation is 0.2 instead of the 0.5 proposed by Margat et al. (1997).

The final correlation proposed is summarized as follows:

$$If\ \ u_{SV} < -111.88 \cdot u_{SL}\ \ +11.848\ \left[\frac{m}{s}\right] \quad \Rightarrow \quad h_{TP} = 5 \cdot Bo^{0.15} \cdot h_{lo} \tag{43}$$

$$If\ \ u_{SV} > -111.88 \cdot u_{SL}\ \ +11.848\ \left[\frac{m}{s}\right]\ \ h_{TP}\ \ the\ greater\ of \begin{cases} h_{nb} = 5 \cdot Bo^{0.15} \cdot h_{lo} \\ h_{cb} = \left(\phi_{Chisholm}^2\right)^{0.2} \cdot h_{lo} \end{cases} \tag{44}$$

Figure 11 and 12 compare the experimental data with the model proposed. These figures indicate that the model can satisfactorily correlate our data for different heat and mass fluxes. Figure 11 shows the correlation prediction for data obtained at different heat fluxes. The proposed correlation is able to yield a very reasonable estimation of the transition from apparent nucleate to convective boiling. Figure 12 shows the correlation prediction in comparison with data at different mass fluxes. In this case, the correlation prediction, when only apparent nucleate boiling is present, is quite accurate. The correlation can predict the database with a RMS error of 8% and also can predict the transition between nucleate and convective boiling. Table 4 summarizes the errors obtained with the proposed correlation.

3. Conclusions

The experimental data of ammonia/water mixture boiling in a plate heat exchanger presented in Táboas et al. (2010) is used to compare heat transfer and pressure drop measurements with predicted values of correlations available in the open literature. The major conclusions can be summarized as follows:

1. The frictional pressure drop in two-phase flow may be calculated according to the Chisholm model. The numerical value of the Chisholm constant found with the experimental data is 3 but it should be noted that this value is very dependent on the single-phase friction factor correlation used.

2. The frictional pressure drop shows an almost linear dependence on the kinetic energy per unit volume of the flow. The linear fit shows a slope value of 0.33 when the kinetic energy per unit volume is calculated by the homogenous model.

3. The two-phase friction factor correlation with the equivalent Reynolds number yielded a relatively large spread.

4. The pool boiling correlations analysed show large deviations with respect to the experimental data.

5. The analysis of convective boiling correlations in tubes showed that the two-phase enhancement factor should be lower than for tubes.

6. Six specific PHE flow boiling correlations were compared with the experimental data. The Donowski and Kandlikar (2000) correlation gives the best fitting with experimental data although this correlation does not take into account the heat flux effect on the boiling coefficient for the data considered.

7. A new PHE flow boiling correlation is proposed in this paper. There was an attempt to correlate the data, based on previous results from other authors. As a consequence, the convective boiling enhancement factor was correlated by the Margat et al. (1997) proposal. The apparent nucleate boiling term is calculated based on the Hsieh and Lin (2002) correlation. The proposed correlation is able to predict the transition between nucleate and convective boiling and 98 % of the heat transfer coefficient calculated within 20% of error for the data considered.

Acknowledgements

This work is part of a R&D project funded by the Spanish Ministry of Science and Technology: DPI2002-04536-C02-01.

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Table 1 - Prediction performance obtained for pressure drop flow boiling correlations applied to the PHE.

	AVG ERROR	MEAN ERROR	30% RANGE	20% RANGE	10% RANGE
Chisholm (1967)	-5.42	12.05	95.18	81.67	49.52
KE/Volume	10.70	15.11	88.10	71.70	41.80
Re equivalent	8.81	30.53	63.67	45.66	25.72

Table 2 - Prediction performance obtained for in-tube flow boiling correlations applied to PHE.

	AVG ERROR	MEAN ERROR	30% RANGE	20% RANGE	10% RANGE
Hsied and Lin (2003) (Gungor and Winterton, 1987)	-35.31	36.49	78.86	50.68	7.05
Djordjevic and Kabelac (2008) (Steiner and Taborek, 1992; K=0.51)	25.19	30.12	61.50	42.80	17.60

Table 3 - Prediction performance obtained for specific PHE correlations.

	AVG ERROR	MEAN ERROR	30% RANGE	20% RANGE	10% RANGE
Danilova (1981) (mult. factor=1.15)	-21.17	27.56	52.03	30.89	15.72
Feldman et al. (2000)	90.01	90.02	9.49	4.34	1.08
Hsieh and Lin (2002)	35.1	36.3	37.0	21.2	5.4
Donowski and Kandlikar (2000)	3.2	14.3	89.9	76.4	49.2
Yan and Lin (1999)	84.5	84.5	0.0	0.0	0.0
Han et al. (2003)	2.3	18.4	86.1	62.8	29.6

Table 4 - Prediction performance of the proposed model.

	%
AVG. ERROR	2.11
RMS ERROR	8.29
30 % RANGE	99.73
20 % RANGE	98.10
10 % RANGE	76.90

Figure_1

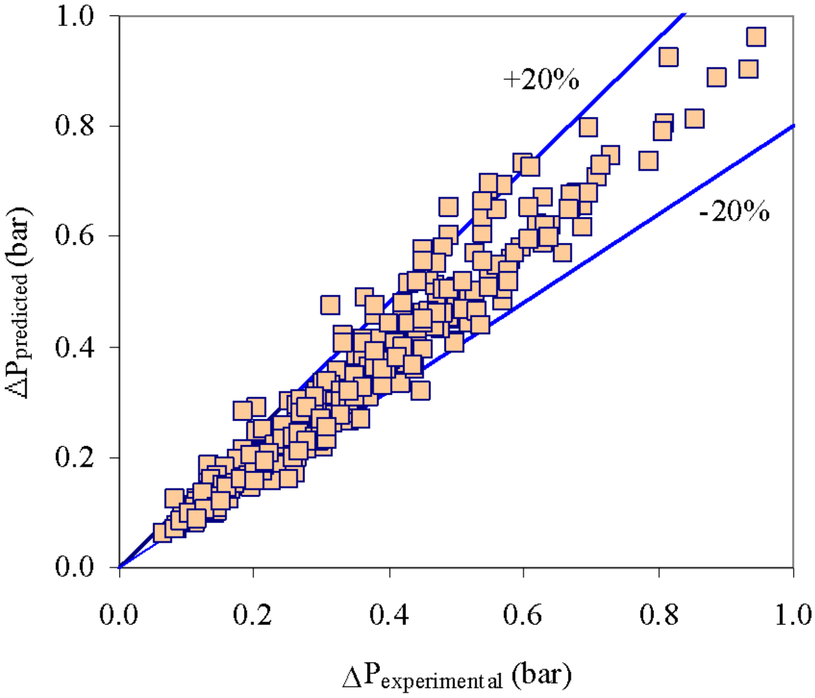

Figure 1 - Two-phase pressure drop comparison between predicted and experimental values using the Chisholm (1967) approach.

Figure_2

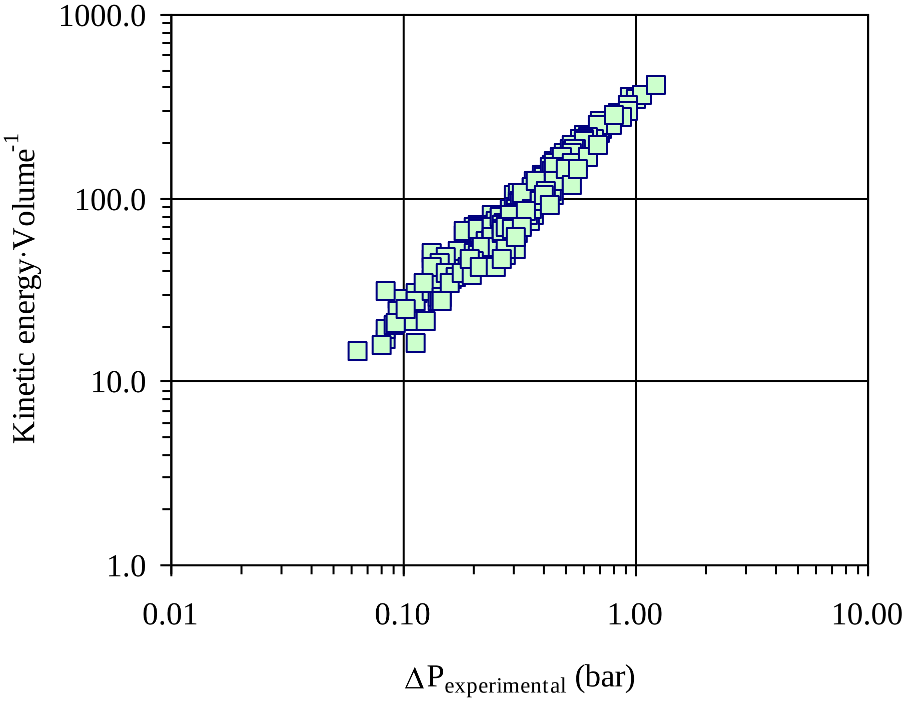

Figure 2 - Kinetic energy per unit volume vs. experimental pressure drop.

Figure_3

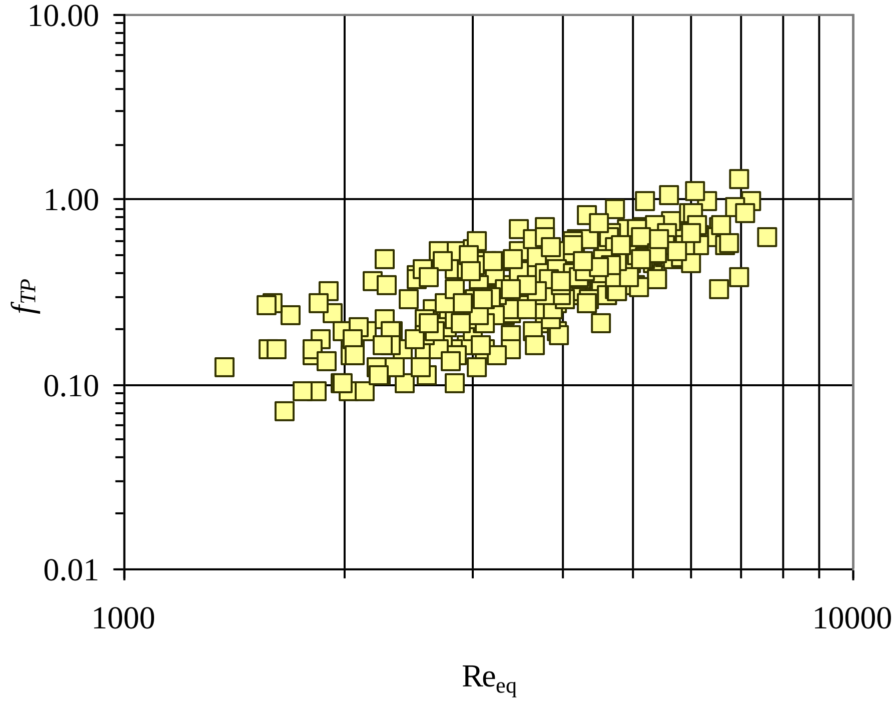

Figure 3 - Experimental friction factor vs. Equivalent Reynolds number.

Figure_4

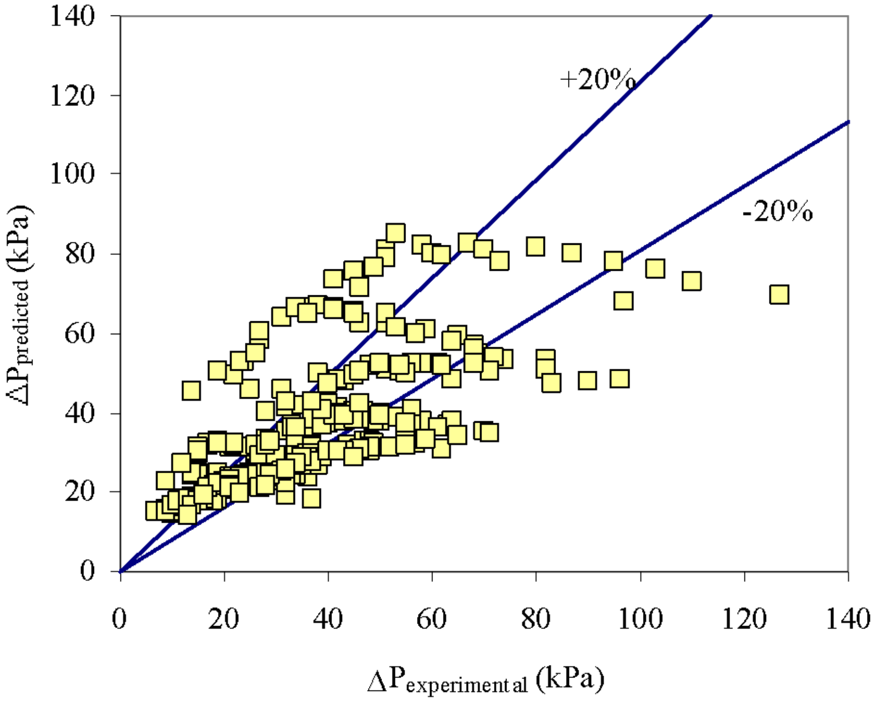

Figure 4 - ΔP predicted by equation (36) vs. ΔP experimental.

Figure_5

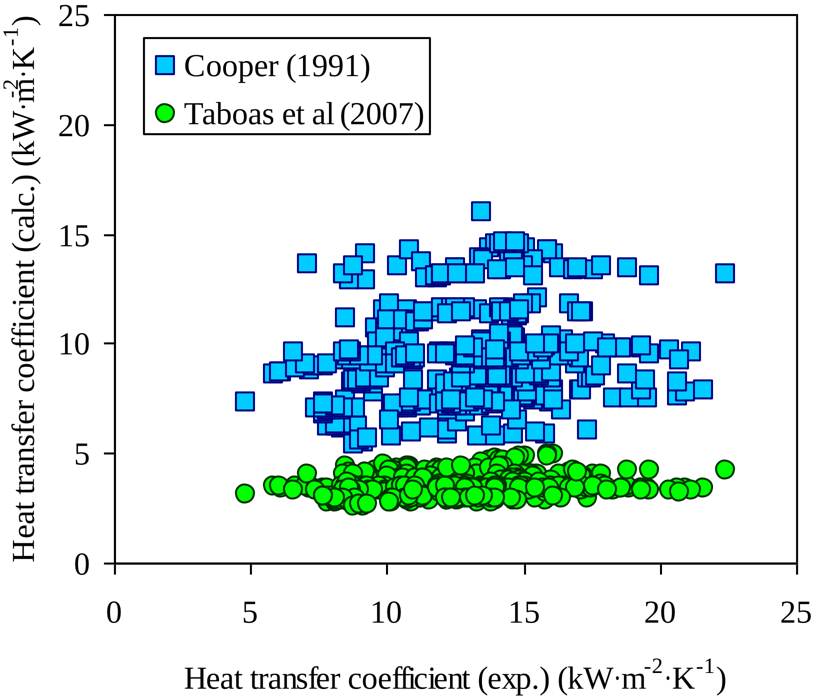

Figure 5 - Heat transfer coefficient calculated by Cooper (1984) and Táboas et al. (2007) vs. experimental results from Táboas et al. (2010).

Figure_6

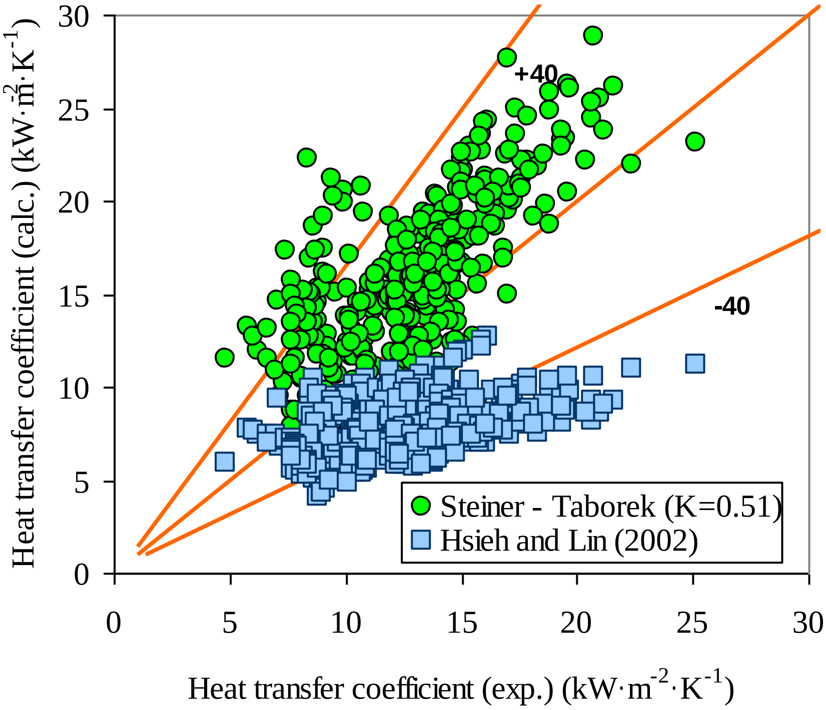

Figure 6 - Prediction performance of the Steiner and Taborek (1992) correlation multiplied by the constant K=0.51 proposed by Djordjevic and Kabelac (2008), and the correlation of Gungor and Winterton (1986) proposed by Hsieh and Lin (2003).

Figure_7

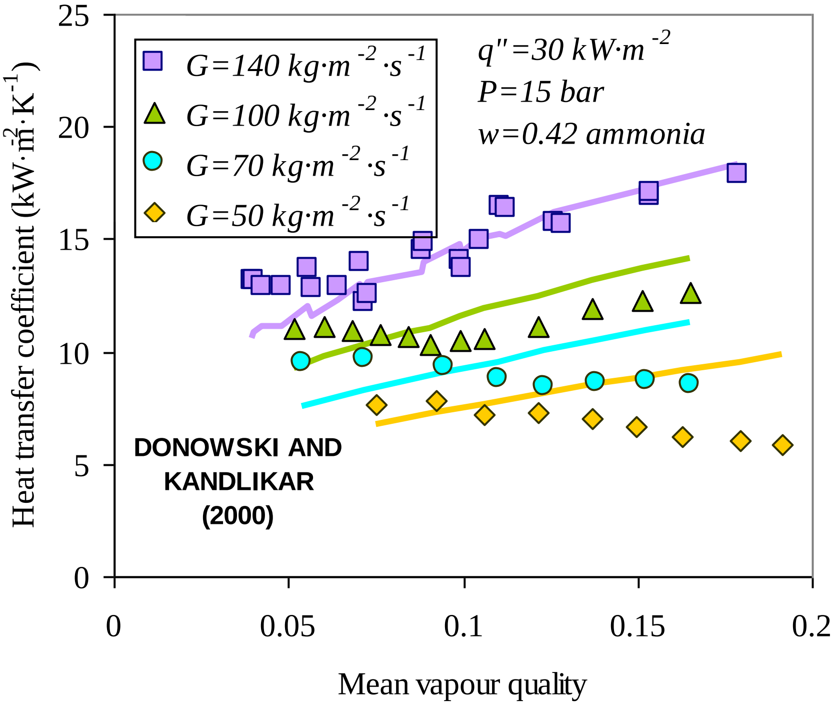

Figure 7 - Heat transfer coefficient comparison between calculated and experimental values using the Donowski and Kandlikar (2000) correlation for different mass fluxes and a constant heat flux of 30 kW m^{-2}.

Figure_8

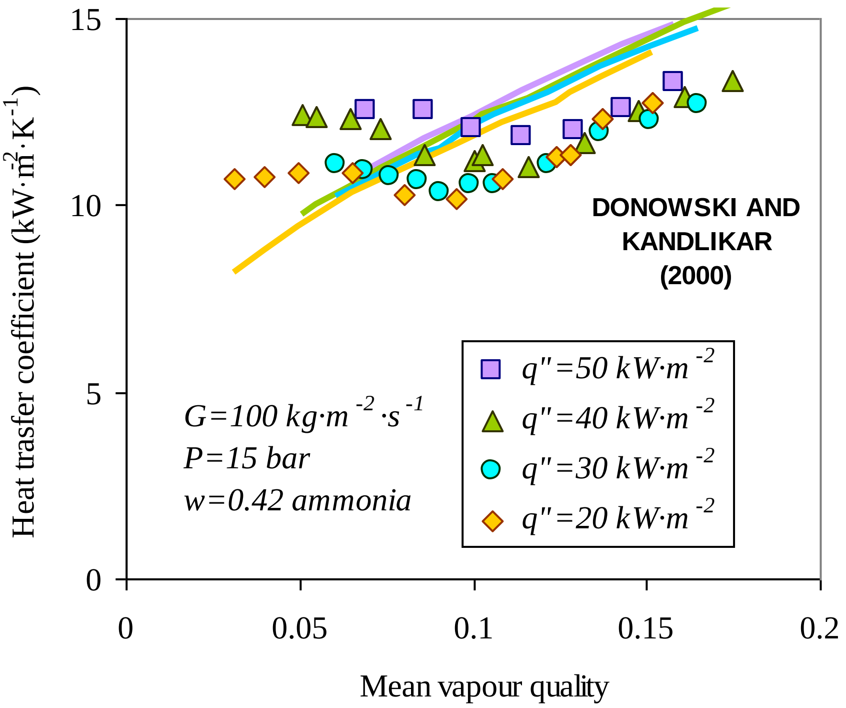

Figure 8 - Heat transfer coefficient comparison between calculated and experimental values using the Donowski and Kandlikar (2000) correlation for different heat fluxes and a constant mass flux of 100 kg $m^{-2}s^{-1}$.

Figure_9

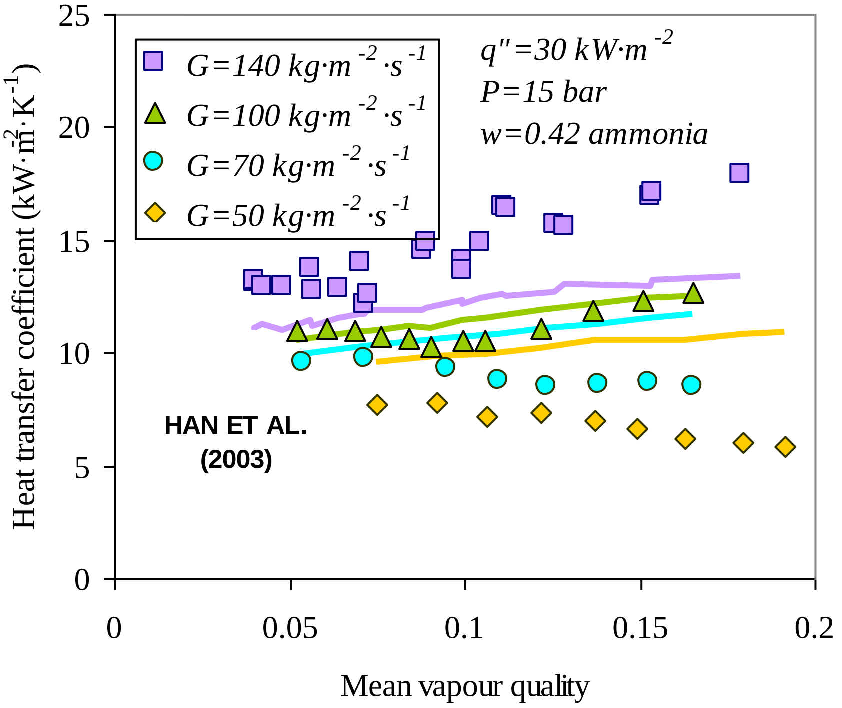

Figure 9 - Heat transfer coefficient comparison between calculated and experimental values using the Han et al. (2003) correlation for different mass fluxes and a constant heat flux of 30 kW m^{-2}.

Figure_10

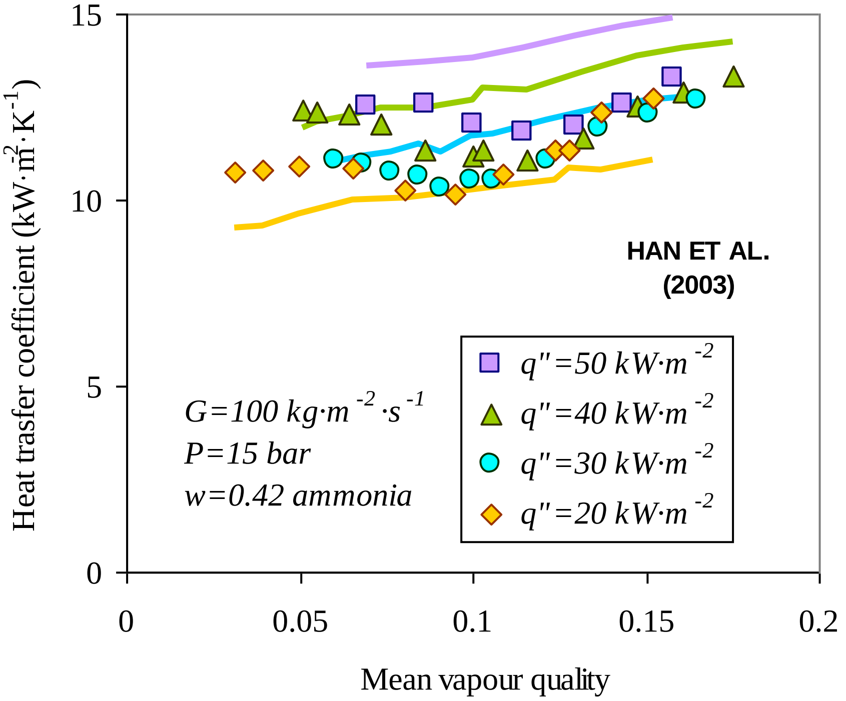

Figure 10 - Heat transfer coefficient comparison between calculated and experimental values using Han et al. (2003) correlation for different heat fluxes and a constant mass flux of 100 kg $m^{-2}s^{-1}$.

Figure_11

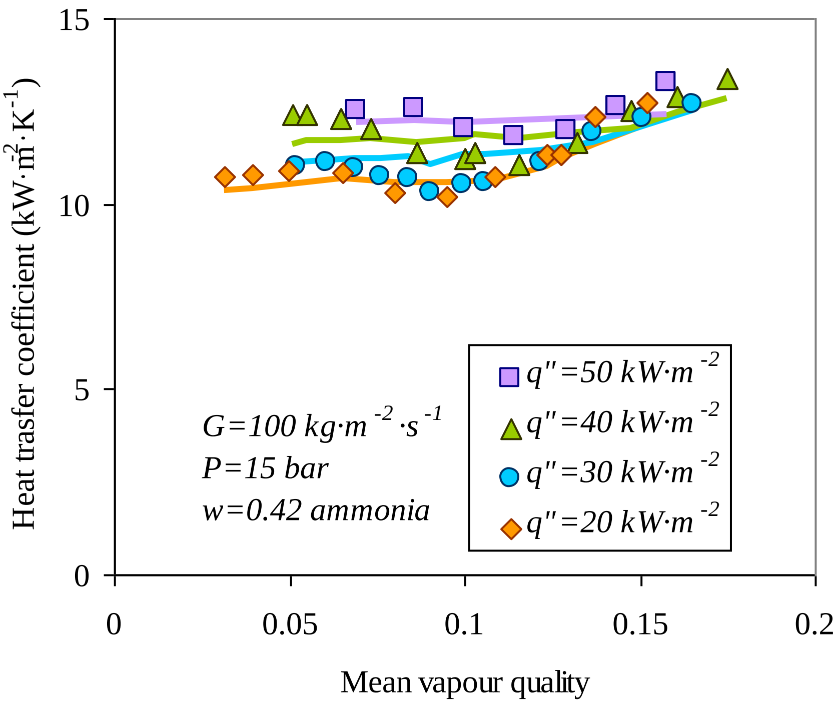

Figure 11 - Proposed model heat transfer coefficient prediction at a constant mass flux of 100 kg.m^{-2}.s^{-1} and different heat fluxes.

Figure_12

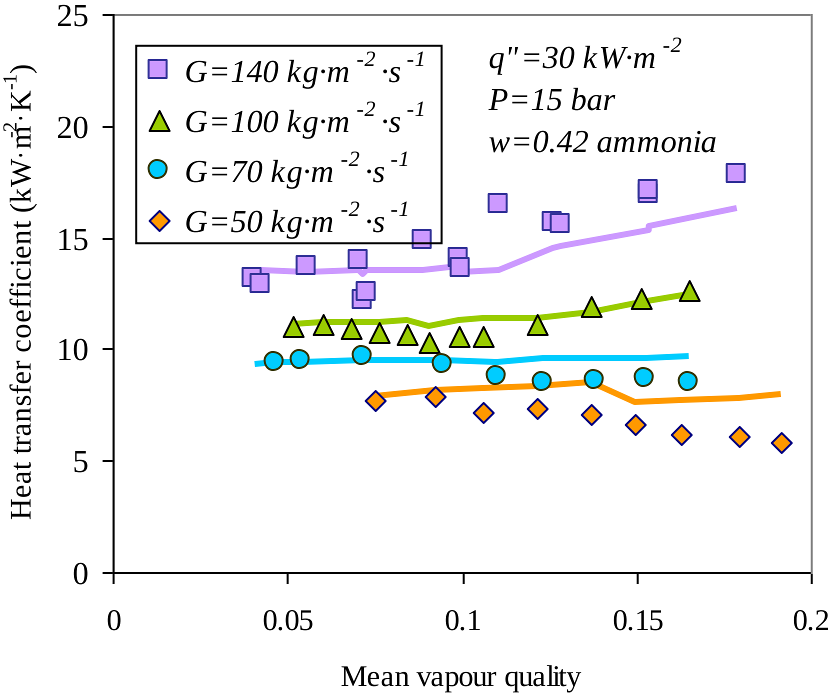

Figure 12 - Proposed model heat transfer coefficient prediction at a constant mass flux of 30 kg m^{-2} s^{-1} and different heat fluxes.

Figure captions

Figure 1 - Two-phase pressure drop comparison between predicted and experimental values using the Chisholm (1967) approach.

Figure 2 - Kinetic energy per unit volume vs. experimental pressure drop.

Figure 3 - Experimental friction factor vs. equivalent Reynolds number.

Figure 4 - ΔP predicted by equation (36) vs. ΔP experimental.

Figure 5 - Heat transfer coefficient calculated by Cooper (1984) and Táboas et al. (2007) vs. experimental results from Táboas et al. (2010).

Figure 6 - Prediction performance of the Steiner and Taborek (1992) correlation multiplied by the constant K=0.51 proposed by Djordjevic and Kabelac (2008), and the correlation of Gungor and Winterton (1986) proposed by Hsieh and Lin (2003).

Figure 7 - Heat transfer coefficient comparison between calculated and experimental values using the Donowski and Kandlikar (2000) correlation for different mass fluxes and a constant heat flux of 30 kW m^{-2}.

Figure 8 - Heat transfer coefficient comparison between calculated and experimental values using the Donowski and Kandlikar (2000) correlation for different heat fluxes and a constant mass flux of 100 kg.m^{-2} s^{-1}.

Figure 9 - Heat transfer coefficient comparison between calculated and experimental values using the Han et al. (2003) correlation for different mass fluxes and a constant heat flux of 30 kW m^{-2}.

Figure 10 - Heat transfer coefficient comparison between calculated and experimental values using Han et al. (2003) correlation for different heat fluxes and a constant mass flux of 100 kg.m^{-2} s^{-1}.

Figure 11 - Proposed model heat transfer coefficient prediction at a constant mass flux of 100 kg.m^{-2} s^{-1} and different heat fluxes.

Figure 12 - Proposed model heat transfer coefficient prediction at a constant mass flux of 30 kg.m^{-2} s^{-1} and different heat fluxes.