Two-Phase Ammonia-Water Absorption in Mini-Channel Annulus

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ABSTRACT

In order to reduce investment costs and refrigerant charge for heat pump equipment, the design of the required heat exchangers should be optimized. Mini-channels heat exchangers are proposed since they can dissipate a higher heat flux and they can be more compact. An accurate prediction of the two-phase heat transfer coefficient in mini-channels is necessary for the heat exchangers design. Several correlations have been proposed in the literature but they cannot cover the wide ranges of working fluids, operating conditions and different mini-channels dimensions. This paper presents and compares the most suitable correlations available in the literature for mini-channel heat transfer and pressure drop during two-phase flow. Also a new theoretical model is proposed for absorption of ammonia into ammonia-water in mini-channel annuli. The suitability of previously published correlations for prediction of heat transfer coefficients and pressure drop in mini-channels during absorption of ammonia-water mixture is then evaluated. For this purpose, experimental data have been collected in a mini-channel with the hydraulic diameter of 0.4 mm and 0.8 m length, so that the predictions of the correlations can be compared with the values from the experiments. The effect of vapour quality and mass flux on the two-phase heat transfer coefficient is investigated. From the comparison with the experimental data the prediction accuracy of different methods is evaluated.

1. INTRODUCTION

Mini-channels heat exchangers are proposed to be used for heat pumps since they can dissipate a higher heat flux and they can be more compact. In the past, many experiments have been done to determine the two phase flow behavior in mini-channels. Bandhauer et al. (2006) did experiments with R-134a in micro-channels with diameters between 0.5 and 1.6 mm under different operating conditions. The measured data showed large discrepancies from the models developed for larger channels mainly because the flow regime and interfacial shear force play a more significant role. Experimental data for the condensation of R-410A and R-134a in mini-channels has been presented by Cavallini et al. (2006). They suggested a heat transfer model from an analogy between heat and momentum transfer where the effect of droplet entrainment rate from liquid film has been introduced. The model applies to the annular and annular-mist flow regimes and agrees well with the previously published experimental data.

Empirical correlations for condensation in mini-channels have been proposed based mainly on data for R-134a. It has been shown that when these correlations are applied for ammonia, the results are not in agreement with each other (Su et al., 2009).

Phase change flow visualization studies of R-134a have been performed by Coleman and Garimella (1999) and they showed that in the intermittent flow regime the vapor phase travels as long solitary bubbles surrounded by an annular liquid film and separated by liquid slugs. As the tube size decreases, surface tension forces at the bubble interface begin to dominate the gravitational forces and the bubble tends to take a cylindrical shape.

Two-phase flow patterns in annuli have been investigated by a few researchers. Kelessidis and Dukler (1989) conducted an experimental study of air-water two-phase flow regimes for vertical upward flow in an annulus with a hydraulic diameter of 25.4 mm. They identified the following flow regimes: bubbly, slug, churn, annular and annular with lumps. Ekberg et al. (1999) also performed experiments on narrow annuli with water and air. Bubbly, slug/plug, churn and annular were the major flow patterns observed when inner and outer diameter of 6.6 and 8.6 mm and 33.2 and 35.2 mm, respectively were used. A two-phase flow model that predicts the saturated flow boiling heat transfer coefficient based on the annular flow pattern has been presented by Qu and Mudawar (2003).

In the present paper, the influence of mini-channel heat exchanger operating conditions on the heat transfer coefficient and pressure losses during absorption of ammonia-water mixture in vertically oriented annulus is investigated. For this purpose, a numerical model for the prediction of absorption of ammonia-water in mini-channel annulus is developed based on the corresponding flow pattern. The suitability of existing condensation heat transfer correlations for the conditions encountered in ammonia-water absorption in mini-annuli would be also investigated.

2. EXPERIMENTAL SET-UP

2.1 Test facilities

The experimental test section and facility used to perform the experiments is presented in Figure 1. The test section consists of twelve heat exchangers using stainless steel tubes with inner diameters of 2.0, 1.1 and 0.5 mm and lengths of 0.2, 0.4, 0.6 and 0.8 m. The heat exchangers consist of an inner and outer vertical channel with absorption and desorption streams in counter flow. Absorption takes place in the outer channel and dissipates heat to the tube side of the heat exchanger. Ideally almost saturated ammonia-water vapor enters the top of the channel and leaves the bottom as almost saturated liquid. In the inner channel the desorption process takes place by receiving the heat from the absorption process. Rock wool insulation was wrapped around the test section in order to reduce heat losses to the ambient, and all lines were insulated with heat resistant foam.

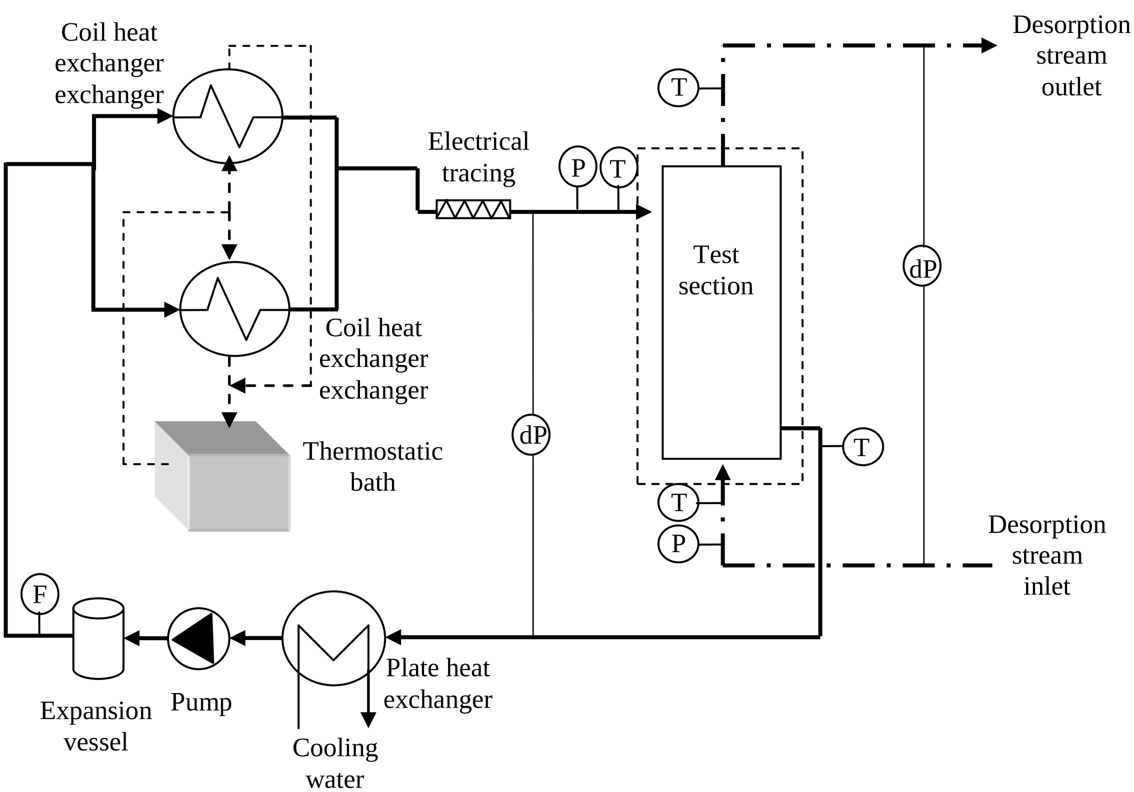

Figure 1: Schematic diagram of the test facility

The inlet temperature and pressure conditions of the absorption stream are controlled with a utility section. The loop for the absorption stream consists of a plate heat exchanger, a pump, an expansion vessel and two coil heat exchangers. The stream is cooled in the plate heat exchanger from which it is leaving in a subcooled liquid state. The piston pump provides circulation and pressurizes the stream. The two coil heat exchangers are used for heating the absorption stream to almost saturated vapor. The inlet temperature of the stream is controlled with a thermostatic bath and with electrical tracing heating. The measurements of temperature and pressure at this point are used to determine the thermodynamic state of the working fluid. A similar loop is provided for the desorption stream.

A differential pressure transducer (Siemens Sitrans P DSIII) is used to measure the pressure drop between the inlet and outlet of the channel. The temperature of the fluids is measured by PT100 AA class type temperature sensors. The mass flow and density are measured by a Coriolis flow meter (mini Cori-Flow M14 manufactured by Bronkhorst) with uncertainty of ± 0.046 kg/h.

The inlet pressure and temperature for the absorption and desorption streams are kept constant. The operating conditions of the set-up have been selected to approach the conditions expected during industrial heat pump operation. For the absorption side, the pressure is maintained at about 13 bar and the inlet temperature is around 170°C. Absorption tests were carried out at mass fluxes ranging between 200 kg/m^{2}s and 400 kg/m^{2}s.

2.2 Data analysis

The energy balance for the test section is:

$$\dot{Q}_{abs} = \dot{m}(i_{in} - i_{out}) \tag{1}$$

The enthalpies and qualities of the absorption stream at the inlet and outlet are obtained using FluidProp (Colonna and Van der Stelt, 2004). The log-mean temperature difference is:

$$LMTD = \frac{(T_{in,abs} - T_{out,des}) - (T_{out,abs} - T_{in,des})}{\ln \dfrac{T_{in,abs} - T_{out,des}}{T_{out,abs} - T_{in,des}}} \tag{2}$$

The overall heat transfer coefficient is:

$$U_{abs} = \frac{\dot{Q}}{A \cdot LMTD} \tag{3}$$

The Wilson plot method has been applied to determine the heat transfer coefficient on the absorption side of the heat exchanger.

3. THEORETICAL MODEL

The experimental conditions for the absorption side have been drawn in the Ekberg et al. (1999) flow pattern map. For a mass flux per unit area of 100 kg/m^{2}s, the map indicates that the flow starts as an annular flow which subsequently develops into an annular slug flow, stratified slug flow and finally a slug/plug flow. The dominant flow regime is the slug/plug flow, as indicated in Figure 2. Taking this into account, the model developed is based on a slug/plug flow pattern. This flow is assumed to be comprised of a liquid slug and a vapor bubble surrounded by a thin annular liquid film on both sides of the annulus. This is illustrated in Figure 3. The frictional pressure drop for this flow pattern includes contributions from the liquid slug and the vapor bubble:

$$\frac{\Delta P}{L} = \left(\frac{dP}{dz}\right)_{bubble} \frac{L_{bubble}}{L_{bubble} + L_{slug}} + \left(\frac{dP}{dz}\right)_{slug} \frac{L_{slug}}{L_{bubble} + L_{slug}} \tag{4}$$

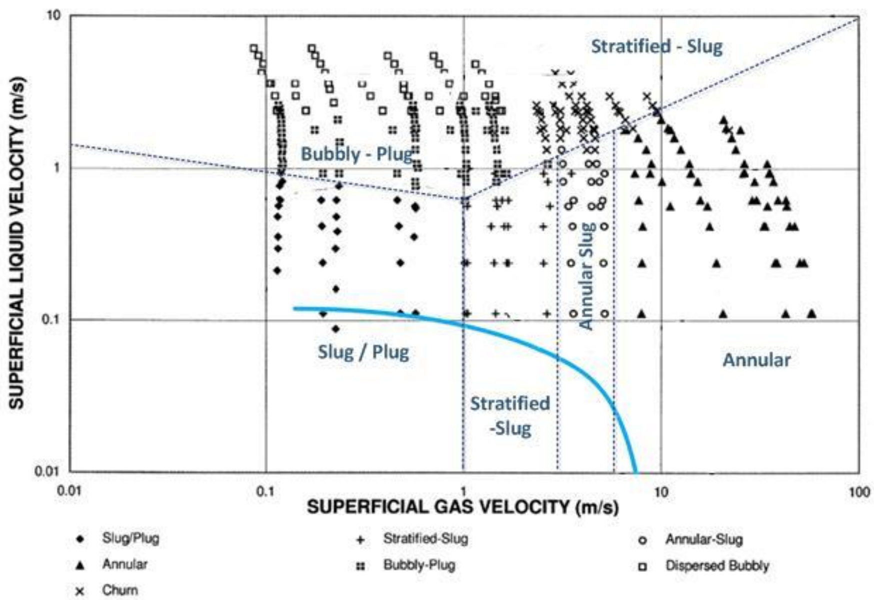

Figure 2: Experimental conditions of the absorption process in the flow pattern map (Ekberg et al., 1999)

The following assumptions have been made in the Navier-Stokes equations in cylindrical form: steady state, no flow in radial or rotational direction, no change of flow velocity in axial direction, constant pressure in radial and rotational direction and negligible gravity and surface tension effects.

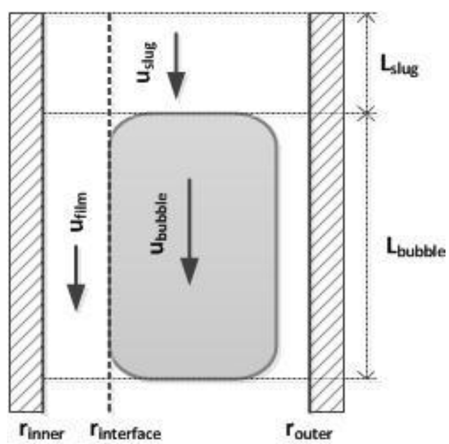

Figure 3: Assumed flow pattern

Following the method proposed by Garimella et al. (2002), the film thickness can be calculated by determining the ratio between the slug and bubble length, followed by stating that the pressure drop in the film should be equal to the pressure drop in the bubble.

The inner film thickness, outer film thickness and pressure loss are calculated from a system of simultaneous equations including a shear balance at the bubble-film interface. The velocity and shear stress distribution in the annulus is presented in Figure 4. The section where a vapor bubble covers the annulus is divided in an inner liquid film flow, an outer liquid film flow and a vapor bubble flow.

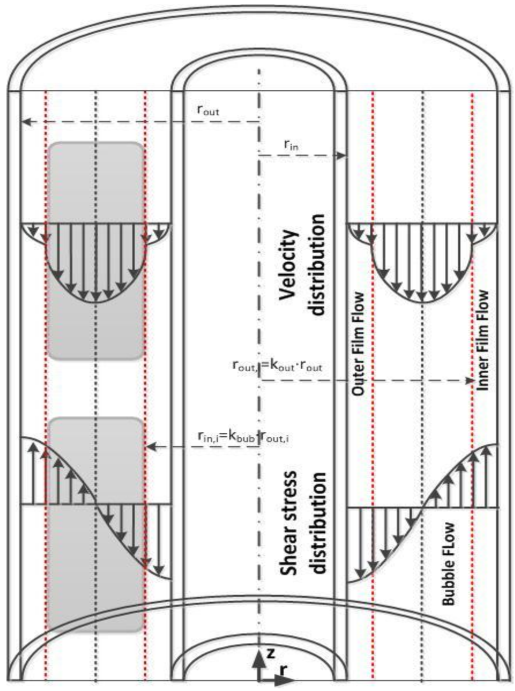

Figure 4: Velocity and shear distribution in the annulus

The velocity in the liquid slug is calculated with the mass flux per unit area and quality as suggested by Suo and Griffith (1964):

$$j_L = \frac{(1-q)\cdot G}{\rho_L} \tag{5}$$

The slug velocity is calculated with the mass flux per unit area and quality:

$$u_{slug} = \frac{(1-q)\cdot G}{\rho_L} + \frac{q \cdot G}{\rho_V} \tag{6}$$

It can be derived that the velocity profile for an annular single phase laminar viscous flow is (van Leeuwen, 2011):

$$u_z = -\frac{dP}{dz}\frac{R}{4\cdot\mu}\left[\left(\frac{r}{R}\right)^2 - C_1 \ln\left(\frac{r}{R}\right) + C_2\right] \tag{7}$$

The relative bubble length is calculated by:

$$L_{bubble} = \frac{j_V}{j_L + j_V} \tag{8}$$

The relative slug length is calculated by:

$$\frac{L_{slug}}{L_{slug} + L_{bubble}} = C_K \frac{j_L}{j_L + j_V} \tag{9}$$

The C_K has been taken equal to 1.

The shear force in axial direction is calculated by Newton's law of viscosity:

$$\tau_z = \mu \frac{du_z}{dr} = -\frac{dP}{dz}\frac{R}{2}\left[\frac{r}{R} - \frac{1}{2}C_1 \ln\left(\frac{r}{R}\right)\right] \tag{10}$$

The volumetric flow rate is constant through any plane. Thus, the volumetric flow rate of the slug is in balance with the combined volumetric flow rate of the films and bubble.

$$\bar{u}_{slug} \cdot A_{slug} = \bar{u}_{z,in\_film} \cdot A_{in\_film} + \bar{u}_{z,out\_film} \cdot A_{out\_film} + \bar{u}_{z,bubble} \cdot A_{bubble} \tag{11}$$

Mass balance for the bubble and film phases:

$$\dot{m}_L = \bar{u}_{z,in\_film} \cdot A_{in\_film} \cdot \rho_L + \bar{u}_{z,out\_film} \cdot A_{out\_film} \cdot \rho_L \tag{12}$$

$$\dot{m}_V = \bar{u}_{z,bubble} \cdot A_{bubble} \cdot \rho_V \tag{13}$$

The transfer of mass and energy from the vapor phase to the liquid phase is modeled assuming equilibrium at the liquid vapor interface.

3.1 Mass transfer

During the absorption process, the interface mass transfer resistance plays a significant role, so that mass transfer must be included in the model.
The mass transfer coefficient per unit area for liquid and vapor phases is obtained based on the Chilton and Colburn analogy:

$$F = \frac{c \cdot h_c}{\rho \cdot c_p} \Pr^{\frac{2}{3}} \cdot Sc^{-\frac{2}{3}} \tag{14}$$

The diffusivity coefficient for the liquid phase is calculated using the correlation of Wilke-Chang (1955). For the gas phase, the diffusion coefficient is calculated with the empirical relation by Fuller et al. (1966). The thermodynamic properties of ammonia-water have been calculated using REFPROP (Lemmon et al., 2010) and correlations from M. Conde Engineering (2006).

3.2 Heat transfer

The annular vapor bubble is classified as an annular flow where either the inner surface, outer surface or both surfaces of the annulus transfer heat. In this case, the inner surface of the annular bubble transfers heat and the outer surface is considered insulated.

The annular liquid film flow is generally laminar and the heat transfer occurs only by conduction. In this case, the Nusselt number equals unity. The Nusselt number between the film and annuli wall equals the Nusselt number of the film at the liquid vapor interface.

4. RESULTS

4.1 Heat transfer

The local heat transfer coefficients have been investigated. For this purpose, different mass fluxes and vapor fractions have been applied.

Previously developed empirical two-phase flow correlations are used to determine the heat transfer coefficient using explicit functions of measurable parameters. These correlations have been developed and validated only for specific flow configurations and operating conditions.

In the heat transfer model of Cavallini et al. (2006) for condensation inside mini-channels, the heat transfer coefficient is correlated to the frictional pressure gradient through the interfacial shear stress. Koyama et al. (2003) also indicated that the shear stress plays an important role in determining the heat transfer coefficient. They proposed a correlation that takes into account the effect of both forced and free convection. In this correlation, the forced convection term is calculated based on the frictional pressure drop.

The flow regimes transition is taken into account in the correlation developed by Wang et al. (2002). They showed that at lower mass fluxes or qualities, a stratified flow regime will prevail and thus the heat transfer is dominated by conduction across the condensate film. The factors that influence the heat transfer in this regime are the condensate film thickness and the fraction of the tube circumference over which the film exists. As mass flux or quality increases, an annular film forms and forced convection prevails.

Surface tension is an important parameter for a two-phase flow in small diameter tubes. The effect of this parameter is taken into account only in the correlations of Cavallini et al. (2006) and Bandhauer et al. (2006).

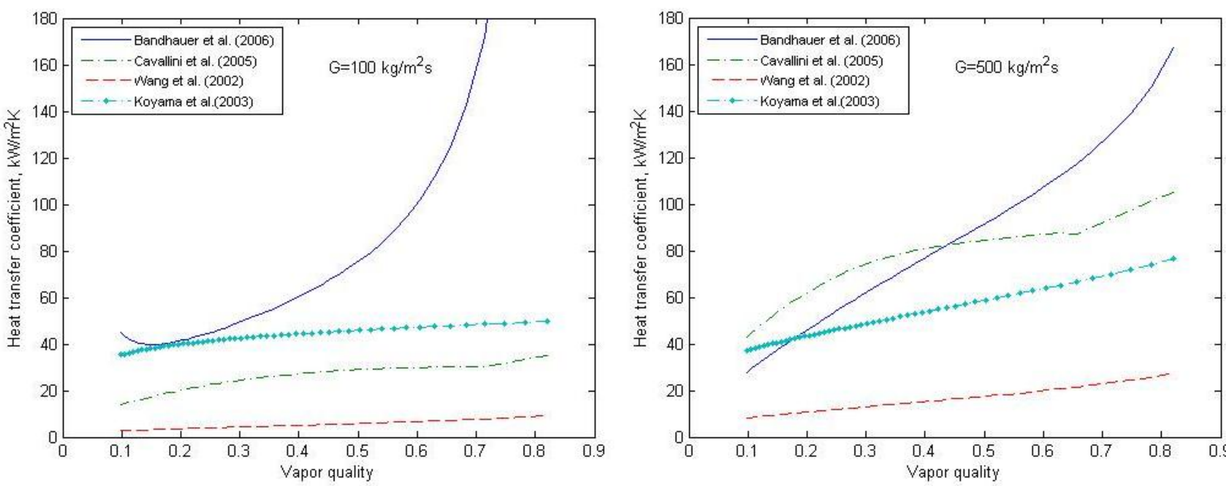

Figure 6: Comparison of condensation correlations when applied for ammonia-water absorption conditions

Figure 6 shows the heat transfer coefficient predicted with the available condensation correlations for small tubes when applied to absorption of ammonia-water mixture with 35% ammonia concentration in a channel with hydraulic diameter of 0.4 mm and 0.8 m length. The heat transfer coefficients decrease with decreasing vapor quality and the slope of the data decreases as the mass flux decreases. During the absorption process the temperature changes significantly so that the mixture properties change more significantly than for condensation processes. It can be seen that at small mass fluxes and at vapor quality higher than 0.6, the method proposed by Bandhauer et al. (2006) predicts significantly larger heat transfer coefficient values than the other methods. The correlation proposed by Wang et al. (2002) presents the lowest values. At high mass fluxes and at vapor quality up to 0.4, the highest heat transfer coefficients are obtained using the correlation of Cavallini et al. (2006).

The predictions of the correlations for heat transfer coefficient during absorption of ammonia-water are compared with experimental data for the tube with hydraulic diameter of 0.4 mm in Figure 7. The Wilson plot technique has been applied to calculate the heat transfer coefficient from overall measurements. This method is based on data measured at the inlet and outlet of the mini-annulus and is subject to uncertainties which are difficult to quantify. The correlations of Bandhauer et al. (2006), Cavallini et al. (2006) and Koyama et al. (2003) overpredict significantly the values obtained from the experiments. The correlation by Wang et al. (2002) underpredicts the experimental data, but presents the lowest deviation. When the results of the theoretical model are compared with the experimental data, the proposed method gives the best prediction of the experimental data. The average deviation is 50%. If a simple correlation would be proposed, then the experimental results indicate that Wang et al. (2002) correlation would give conservative but acceptable results.

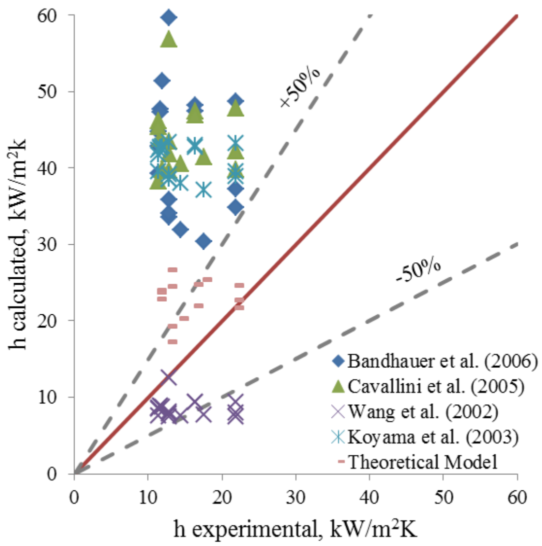

Figure 7: Comparison of data from experiments and predictions of the correlations

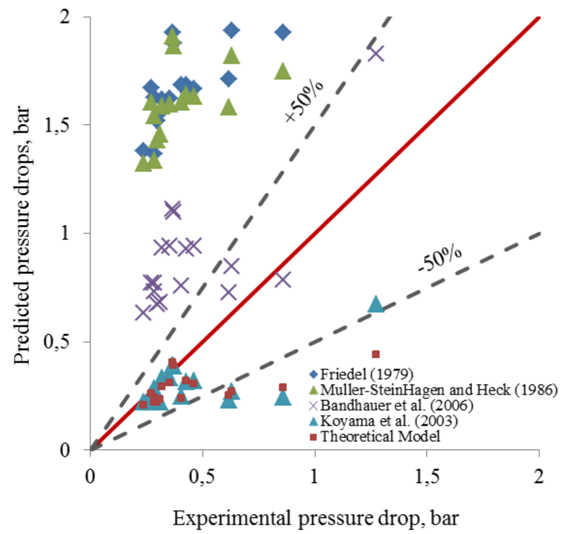

Figure 8: Comparison of pressure drops data from experiments and predictions of the correlations

4.2 Pressure Drop

The two-phase pressure drop has also been measured during the experiments and different predictive approaches were examined for suitability to mini-channel pressure-drop prediction. The theoretical numerical model and several empirical correlations have been considered. Figure 9 shows the frictional pressure drop predicted with different correlations as a function of flow quality and mass flux. The pressure drop increases with the increase in flow quality and mass flux. The method proposed by Koyama et al. (2003) predicts significantly lower pressure drop values than the other methods and it has a different trend.

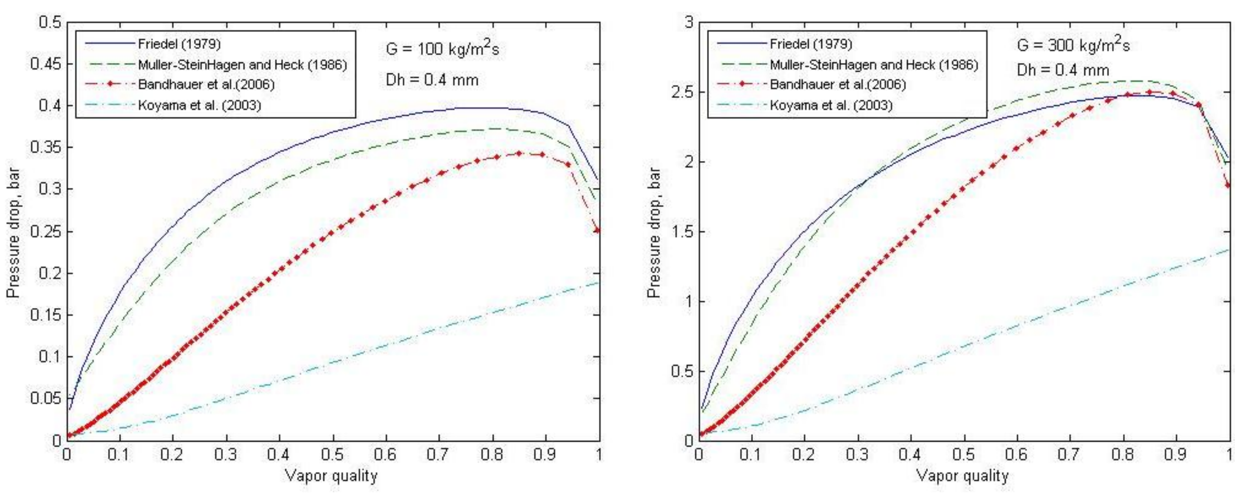

Figure 9: Comparison of pressure drop correlations when applied for ammonia-water absorption conditions

The correlations of Friedel (1979), Müller-Steinhagen and Heck (1986) and Bandhauer et al. (2006) overpredict significantly the present ammonia-water experimental data (Figure 8). This figure gives the total pressure drop .The best prediction is obtained with the method proposed by Koyama et al. (2003). The theoretical numerical model appears to give a good prediction and it is comparable with the prediction of Koyama et al. (2003).

5. CONCLUSIONS

Experiments have been carried out in order to determine two-phase heat transfer coefficients and pressure drop during absorption of ammonia-water in the annulus of a mini-channel tube-in-tube heat exchanger. The inlet and outlet temperatures and pressures were measured for a mini-channel with 0.4 mm hydraulic diameter and a tube length of 0.8 m. A numerical model has been developed based on the flow pattern that is present during the absorption of ammonia-water. The experimental data have been compared against the theoretical model and several microchannel condensation models available in the literature.

The results of the theoretical model overpredicted the experimental data for the heat transfer coefficient. The correlation by Wang et al. (2002) and the theoretical model presents the lowest deviation. However Wang's correlation underpredicts the experimental data, while the model overpredicts the experiments data. The correlations of Bandhauer et al. (2006), Cavallini et al. (2006) and Koyama et al. (2003) cannot reproduce the experimental trend of ammonia-water data, giving a large overprediction of the experimental values.

For the pressure drop, the correlations of Friedel (1979), Müller-Steinhagen and Heck (1986) and Bandhauer et al. (2006) overpredict significantly the present ammonia-water experimental data. The best prediction of the pressure drop is obtained with the method proposed by Koyama et al. (2003) and the theoretical model.

NOMENCLATURE

A	area (m^2)
c	concentration (mol m^{-3})
c_P	specific heat capacity (J $kg^{-1}K^{-1}$)
D_h	hydraulic diameter (m)
F	mass transfer coefficient per unit area (mol $m^{-2}s^{-1}$)
G	mass flux per unit area (kg m^{-2} s^{-1})
h	heat transfer coefficient (W m^{-2} K^{-1})
L	length (m)
i	enthalpy (J kg^{-1})
j	superficial velocity (m s^{-1})
$\dot{m}$	mass flux (kg s^{-1})
P	pressure (N m^{-2})
Pr	Prandtl number
R	outer radius, m
r	inner radius, m
$\dot{Q}$	heat flux (W)
Sc	Schmidt number
q	vapor fraction (-)
T	temperature (K)
t	time, s
u	velocity (m s^{-1})
$\bar{u}$	average velocity (m s^{-1})
U	overall heat transfer coefficient (W m^{-2} K^{-1})
V	volume (m^3)

Greek

ρ	density (kg m^{-3})
μ	dynamic viscosity (Pa s)
τ	shear force

Subscript

abs	absorption
c	convection
des	desorption
in	inlet
L	liquid
out	outlet
V	vapor
z	axial coordinate

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