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Comparative study of flow condensation in conventional and small diameter tubes

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Abstract Flow boiling and flow condensation are often regarded as two opposite or symmetrical phenomena. Their description however with a single correlation has yet to be suggested. In the case of flow boiling in minichannels there is mostly encountered the annular flow structure, where the bubble generation is not present. Similar picture holds for the case of inside tube condensation, where annular flow structure predominates. In such case the heat transfer coefficient is primarily dependent on the convective mechanism. In the paper a method developed earlier by the first author is applied to calculations of heat transfer coefficient for inside tube condensation. The method has been verified using experimental data from literature on several fluids in different microchannels and compared to three well established correlations for calculations of heat transfer coefficient in flow condensation. It clearly stems from the results presented here that the flow condensation can be modeled in terms of appropriately devised pressure drop.

Keywords: Flow condensation; Heat transfer; Minichannels

Nomenclature

c_p	–	specific heat, J/kg K
Con	–	Constraint number, Con=$[\sigma/(\mathrm{g}\ (\rho_L - \rho_G))]^{0.5}/\mathrm{d}$
d	–	channel inner diameter, m
f_1, f_{1z}	–	functions
g	–	gravity acceleration, m/s^2

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G – mass flow rate, kg/m^{2}s
J_G – dimensionless vapour velocity
k_L – liquid thermal conductivity, W/mK
l – channel length, m
p – pressure, Pa
Pr – Prandtl number
R_{MS} – two-phase flow multiplier due to Müller-Steinhagen and Heck
Re – Reynolds number
T – temperature, K
x – quality
X_{tt} – Martinelli parameter
u, w – velocity components, m/s

Greek symbols

α – heat transfer coefficient, W/m^{2}K
μ – dynamic viscosity, Pas
ρ – density, kg/m^3
λ – thermal conductivity, W/mK
δ – liquid film thickness, m
σ – surface tension, N/m
τ – shear stress, Pa

Subscripts

an – annular
G – gas, vapor
L – liquid
LO – liquid only
lt – slug
st – stratified
TPB – two-phase boiling
TPC – two-phase condensation

Superscripts

$+$ – dimensionless

1 Introduction

Flow boiling and flow condensation are often regarded as two opposite or symmetrical phenomena involving the change of phase. There is a temptation to describe both these phenomena with one only correlation, however no such model has yet been suggested. In both cases of phase change the annular flow structure seems to be mostly susceptible to common modeling. Such approach to modeling fail however in cases where other flow structures are present as for example the bubbly flow. In the case of flow boiling in conventional channels one can expect that bubble nucleation renders the

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process of heat transfer not to have its counterpart in the condensation inside tubes. Similarly the collapse of bubbles to form a continuous liquid is the condensation specific phenomenon. Situation seems to be a little less complex in the case of flow boiling in minichannels and microchannels. In such flows the annular flow structure is dominant for most qualities, Thome and Consolini (2008). In such case the heat transfer coefficient is primarily dependent on the convective mechanism. Most of correct modeling of heat transfer in case of condensation inside channels relates the heat transfer coefficient to the friction coefficient, contrary to modeling in case of flow boiling.

The objective of this paper is to present the capability of the flow boiling model, developed earlier, Mikielewicz (2009) to model flow condensation inside tubes. In such case the heat transfer coefficient is a function of the two-phase pressure drop. Therefore some calculations have been accomplished to validate that method with the selection of experimental data due to Bohdal *et al.* (2011). Calculations have been also compared against some well established methods for calculation of heat transfer coefficient due to Cavallini *et al.* (2002), Thome *et al.* (2003) and Bohdal *et al.* (2011).

2 Calculation method

The relation enabling calculation of heat transfer coefficient in flow boiling without bubble generation (α_{TPB}), which is also applicable of calculations of flow condensation (α_{TPC}) yields

$$\frac{\alpha_{TPB}}{\alpha_L} = \frac{\alpha_{TPC}}{\alpha_L} = \sqrt{R_{MS}^n}\,, \tag{1}$$

where α_L is the heat transfer coefficient to liquid only flow. The two-phase flow multiplier R_{MS} due to Müller-Steinhagen and Heck (1986) is recommended for use in case of refrigerants, as confirmed in the state-of-the-art reviews by Ould Didi *et al.* (2002) and Sun and Mishima (2009). Value of the exponent assumes $n = 2$ for laminar flows, whereas for turbulent flows it takes a value of 0.76.

It should be noted however that the selection of a two-phase flow multiplier to be used in the postulated model is arbitrary. In the results presented in the present paper the Muller-Steinhagen and Heck model has been selected for use as it is regarded best for refrigerants such as hydrocarbons, however, a different model could be selected such as for example the

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Lockhart-Martinelli model, where the two-phase flow multiplier is a direct function of the Martinelli parameter, see Mikielewicz (2009). Another conclusion could be drawn from the presented model that in correlations of the type of Eq. (1) the two-phase flow multiplier could also be used for modeling instead of the Martinelli parameter. Author's up to date experience shows that the influence of the two-phase flow multiplier is very important and each fluid could have a different description of a two-phase resistance. In the form applicable to conventional and small diameter channels the Muller-Steinhagen and Heck model yields, Mikielewicz (2009):

$$R_{MS} = \left[1 + 2\left(\frac{1}{f_1} - 1\right) x\,\mathrm{Con}^m\right](1-x)^{\frac{1}{3}} + x^3 \frac{1}{f_{1z}}\,, \tag{2}$$

where $\mathrm{Con} = (\sigma/g/(\rho_L - \rho_G))^{0.5}/d$ and $m = 0$ for conventional channels. Best consistency with experimental data, in case of small diameter and minichannels, is obtained for $m = -1$. In Eq. (2) $f_1 = (\rho_L/\rho_G)(\mu_L/\mu_G)^{0.25}$ for turbulent flow and $f_1 = (\rho_L/\rho_G)(\mu_L/\mu_G)$ for laminar flows. Introduction of the function f_{1z}, expressing the ratio of heat transfer coefficient for liquid only flow (α_{LO}) to the heat transfer coefficient for gas only flow (α_{GO}), is to meet the limiting conditions, i.e. for the quality $x = 0$ the correlation should reduce to a value of heat transfer coefficient for liquid, $\alpha_{TPC} = \alpha_L$ whereas for $x = 1$, approximately that for vapour, i.e. $\alpha_{TPC} \cong \alpha_{G.}$ Hence:

$$f_{1z} = \frac{\alpha_{GO}}{\alpha_{LO}}\,, \tag{3}$$

where $f_{1z} = (\lambda_G/\lambda_L)$ for laminar flows and for turbulent flows $f_{1z} = (\mu_G/\mu_L)(\lambda_L/\lambda_G)^{1.5}(c_{pL}/c_{pG})$. The correlation (1) seems to be general, as the study by Chiou *et al.* (2009) confirms.

3 Condensation inside tubes

Condensation inside tubes has been the topic of interest of not too many investigations. Studies for example by Cavallini *et al.* (2002), Garimella *et al.* (2004) and Thome *et al.* (2003) should here be mentioned. Flow condensation at high heat fluxes enables removal of significant heat fluxes. In case of condensation in small diameter channels the surface phenomena together with the characteristics of the surface itself become more important, as well as interactions between the wall and fluid.

In microchannels we observe domination of forces resulting from action of surface tension and viscosity over the gravitational forces. Hence the

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attempt to extend the range of validity of correlations developed for conventional channels onto the channels with small diameters leads to errors in pressure drop and heat transfer description, making such approach useless. Additionally, the heat transfer coefficient and pressure drop in microchannels strongly depend upon the quality. Hence the detection of flow structures and their influence on pressure drop and heat transfer is indispensable during the condensation of the fluid.

Cavallini *et al.* (2002) proposed separate correlations for the annular, annular-stratified, and stratified, and slug flow regimes. This method is based upon a large data bank, collected for halogenated refrigerants inside tubes with internal diameter $d > 3$ mm at reduced pressure $p_R < 0.75$ and density ratio (ρ_L/ρ_G). Authors of the present article use Cavallini *et al.* (2002) correlations to predict heat transfer coefficient during condensation inside tubes with internal diameter $d > 3$ mm and $d < 3$ mm. The applicable flow regimes were selected based on criteria similar to those proposed by Breber *et al.* (1980), when at the dimensionless vapour velocity $J_G < 2.5$ and Martinelli parameter $X_{tt} < 1.6$ the flow enters the annular-stratified flow transition and stratified flow region. The heat transfer coefficient α_{an-st} is calculated from linear interpolation between heat transfer coefficient at the boundary of the annular flow region $\alpha_{an,JG} = 2.5$ and that for fully stratified flow α_{st}. When $X_{tt} > 1.6$ and $J_G < 2.5$, the flow enters the stratified-slug transition and slug flow pattern region, the heat transfer coefficient is calculated as linear interpolation between the coefficient computed at $X_{tt} = 1.6$ and the one for the liquid flowing with the entire flow rate. Model for annular flow is applied when the dimensionless vapour velocity is lower than 2.5. For the annular flow regime, Cavallini *et al.* (2002) suggested the use of the heat transfer model proposed by Kosky and Staub (1971) with the modified Friedel (1979) correlation for shear stress:

$$J_G* = \frac{Gx}{\sqrt{dg\rho_G(\rho_L - \rho_G)}} \,, \tag{4}$$

$$X_{tt} = \left(\frac{\mu_L}{\mu_G}\right)^{0.1} \left(\frac{\rho_G}{\rho_L}\right)^{0.5} \left[\frac{(1-x)}{x}\right]^{0.9} \,, \tag{5}$$

The dimensionless film thickness is based on the liquid-phase Reynolds number:

$$\delta^+ = \begin{cases} \left(\frac{\mathrm{Re}_L}{2}\right) & \text{for} \quad \mathrm{Re}_L \leq 1145 \,, \\ 0.0504\mathrm{Re}_L^{7/8} & \text{for} \quad \mathrm{Re}_L \geq 1145 \,. \end{cases} \tag{6}$$

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The dimensionless temperature is determined based on the dimensionless film thickness and Prandtl number analogously to Traviss and Rohsenow, (1973):

$$T^+ = \begin{cases} \delta^+ \mathrm{Pr}_L & \text{for} \quad \delta^+ \leq 5 \,, \\ 5\left\{\mathrm{Pr}_L + \ln\left[1 + \mathrm{Pr}_L\left(\frac{\delta^+}{5} - 1\right)\right]\right\} & \text{for} \quad 5 < \delta^+ \leq 30 \,, \\ 5\left\{\mathrm{Pr}_L + \ln\left(1 + 5\mathrm{Pr}_L\right) + 0.495\left(\frac{\delta^+}{30}\right)\right\} & \text{for} \quad \delta^+ > 30 \,. \end{cases} \tag{7}$$

Finally, the heat transfer coefficient α_{TPC} is calculated as follows:

$$\alpha_{TPC} = \begin{cases} \alpha_{an} = \frac{\rho_L c_{pL}\left(\frac{\tau}{\rho_L}\right)^{0.5}}{T^+} & \text{for} \quad J_G > 2.5 \,, \\ \alpha_{an-st} = \left(\alpha_{an,J_G=2.5} - \alpha_{st}\right)\left(\frac{J_G}{2.5}\right) + \alpha_{st} & \text{for} \quad 1.6 < J_G \leq 2.5 \,, \\ \alpha_{an-lt} = \alpha_{LO} + \frac{x(\alpha_{1.6} - \alpha_{LO})}{x_{1.6}} \,, \end{cases} \tag{8}$$

where τ is the shear stress. Thome *et al.* (2003) developed a multi-regime heat transfer correlation, in which the regimes identified are either as (a) fully annular forced convective, or as (b) consisting of varying combinations of upper gravity driven, and lower forced convective terms, in case of horizontal flows. Thome *et al.* (2003) proposed heat transfer model for evaporation inside horizontal tubes. They founded that there is a close similarity between the convection mechanisms in annular film condensation and annular film evaporation inside tubes. Finally they proposed the new following model for the annular flow:

$$\alpha_{TPC} = C Re_L^n \mathrm{Pr}_L^m \frac{\lambda_l}{\delta} f_i \,, \tag{9a}$$

where δ is the film thickness determined based on the void fraction model, where f_i is:

$$f_i = 1 + \left(\frac{u_G}{u_L}\right)^{0.5} \left[\frac{(\rho_L \rho_G)\, g \delta^2}{\sigma}\right]^{0.25} \,. \tag{9b}$$

The constants C, n and m were determined to be 0.003, 0.74 and 0.5, respectively, based on best fit to experimental data for tubes with $d_h > 3$ mm. This correlation predicts heat transfer coefficients with an average deviation of 74%. Thome *et al.* (2003) proposed also the general expression for the local condensing heat transfer coefficient, but this method is determined with complicated procedure who which required identification of the flow

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pattern. In such approach the intermittent flow is very complex, and therefore it is assumed that it is can be predicted approximately by annular flow equations (9a). Recently there appeared also an experimental fit to authors own data, considered in the present paper, due to Bohdal *et al.* (2011). The expression yields:

$$\mathrm{Nu} = 25.084\mathrm{Re}_L^{0.258}\mathrm{Pr}_L^{-0495}p_r^{-0.288}\left(\frac{x}{1-x}\right)^{0.266} . \tag{10}$$

4 Results of comparisons

Presented below is a comparison of best established literature correlations for calculations of flow condensation with the model presented in the first part of the paper, namely relation (1). From available literature the correlations due to Cavallini *et al.* (2002) and Thome *et al.* (2003) have been selected. That set of equations was supplemented by the method due to Bohdal *et al.* (2011). Bearing in mind that the proposed model (1) was thoroughly tested for the conditions of flow boiling, see for example Mikielewicz (2009), showing satisfactory performance, the prediction of condensation inside tubes commenced. The considered methods have been verified by experimental data collected from literature, Bohdal *et al.* (2011). Results of calculations in the form of distributions of heat transfer coefficient with respect to quality are presented in Figs. 1–12. The data has been collected for several tube diameters ranging from 0.45 to 3.3 mm and two different fluids, namely R134a and R404A. The method Eq. (1) has yet to be tested on the case of R404A.

It can be seen that Eq. (1) describes reasonably well the heat transfer coefficients during the flow condensation, see Figs. 1 to 13. The results of comparisons, which are presented in these figures are very promising. The biggest advantage offered by Eq. (1) is the fact that it has a general character and does not require any specific fluid-related constants. Relation (1) does not require prior knowledge of flow maps which are indispensable in case of more accurate methods for calculation of heat transfer coefficients. Calculation show that the method described by (1) can predict heat transfer coefficient in conventional channels and minichannels. However, it must be admitted that in case of smaller channels the discrepancy increases. That can be partially attributed to the fact that in the case of flow condensation, as well as flow boiling, the two-phase flow multiplier is developed for adiabatic flow conditions. In such case the non-isothermal effects on the

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interface of annular flow are neglected. That, in some case can lead to significant modifications of heat transfer coefficient, especially for smaller channel diameters. In flow boiling the liquid film layer is thinning, whereas in case of flow condensation it is thickening. Some way to solve this problem has been presented in *D*. Mikielewicz and J. Mikielewicz (2010) where the so called blowing parameter was introduced to model that effect. The presented in that paper data will be reconsidered in that light in further studies.

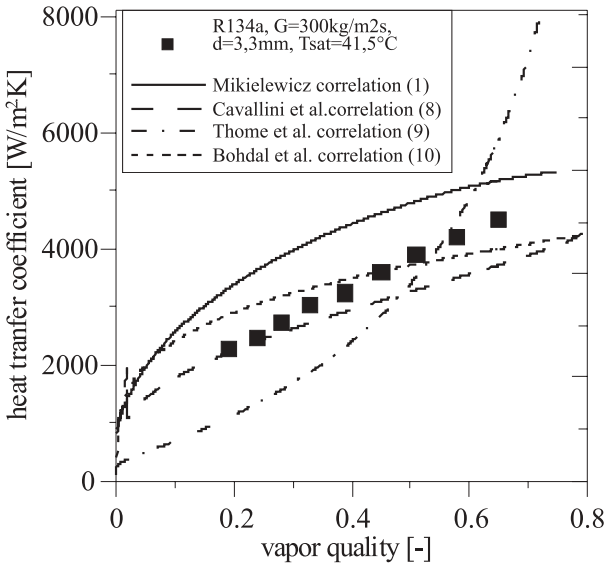

Figure 1. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 3.3$ mm, $G = 300$ kg/m^{2}s, $T_{sat} = 41.5$ °C.

The poor performance of (1) in case of small mass velocity may be attributed to the fact that the bubbly flow is encountered. In such case the annular flow approach is not valid, however the authors would like to stress that still the formulae dependent on the two-phase flow multiplier is applicable. In such case authors plan to incorporate into the two-phase flow multiplier the term responsible for the presence of heat flux. Works on that topic are also underway.

It results from examination of presented above results that heat transfer coefficient in flow condensation is a function of the two-phase multiplier. Careful examinations presented above results show two distinct trends. In case of data for higher diameter tubes (conventional size tubes) the results

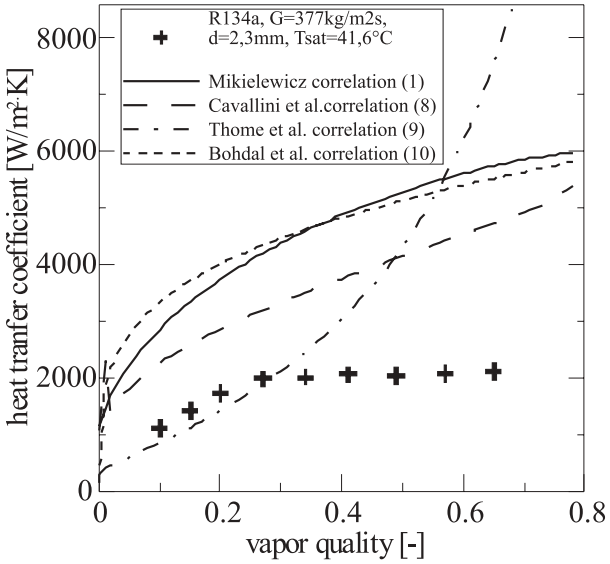

Figure 2. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 2.3$ mm, $G = 377$ kg/m^{2}s, $T_{sat} = 41.6$ °C.

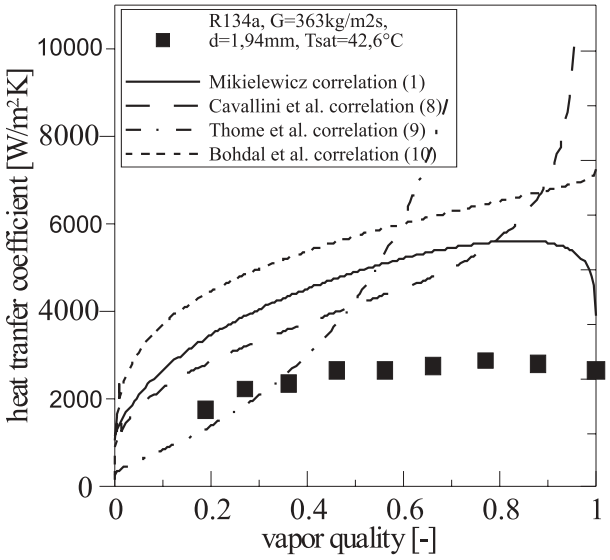

Figure 3. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 1.94$ mm, $G = 363$ kg/m^{2}s, $T_{sat} = 41.6$ °C.

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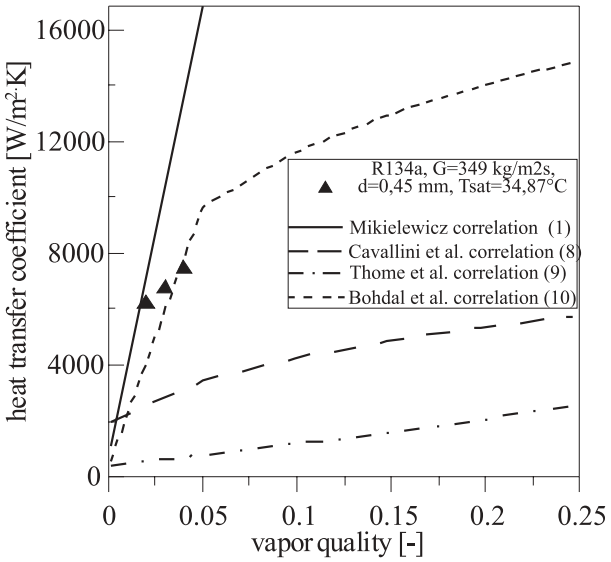

Figure 4. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 0.45$ mm, $G = 349$ kg/m^{2}s, $T_{sat} = 35.0$ °C.

clearly show that the method described by Eq. (1) outperforms other models for greater tube sizes. On the other hand in case of simulations for smaller tube diameters the picture of calculations looks different. The model described by Eq. (1) performs poorly for smaller tube diameters. It ought to be stressed that the qualitative trends are very well revealed by Eq. (1) and the qualitative consistency is very satisfactory. That cannot be said about other considered methods. A significant observation is that at high qualities the heat transfer coefficient tends to the value corresponding for vapour, whereas none of the empirical correlations notice that fact. It is difficult to judge why the models due to Cavallini *et al.* (2002) and Thome *et al.* (2003) exhibit the accuracy presented in the paper. In case of the model described by Eq. (1) the situation is different, as the model predicts depend on the two-phase multiplier. In the present work the halogenated refrigerants were examined in which case the two-phase multiplier due to Muller-Steinhagen and Heck (1986) is recommended. That model is also developed on the adjustment to experimental data. In some calculations, in order to increase the accuracy of predictions, the more appropriate two-phase flow multiplier could be used, specifically developed for a particular fluid.

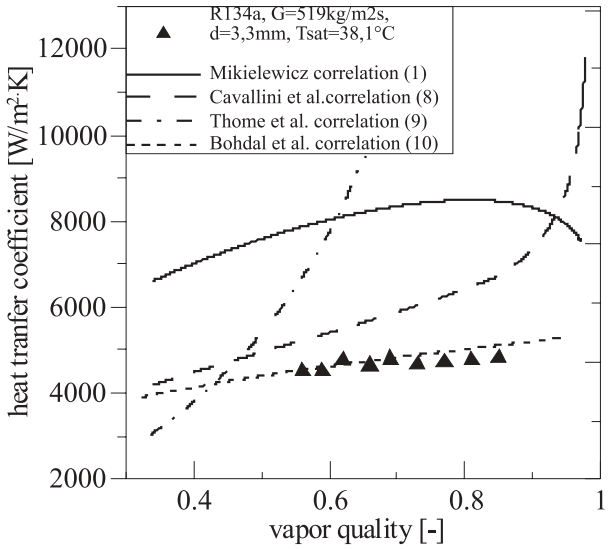

Figure 5. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 3.3$ mm, $G = 519$ kg/m^{2}s, $T_{sat} = 38.1$ °C.

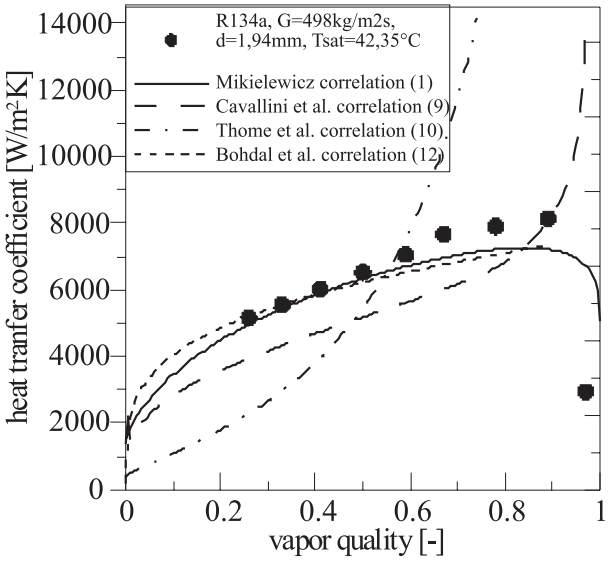

Figure 6. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 1.94$ mm, $G = 498$ kg/m^{2}s, $T_{sat} = 42.4$ °C.

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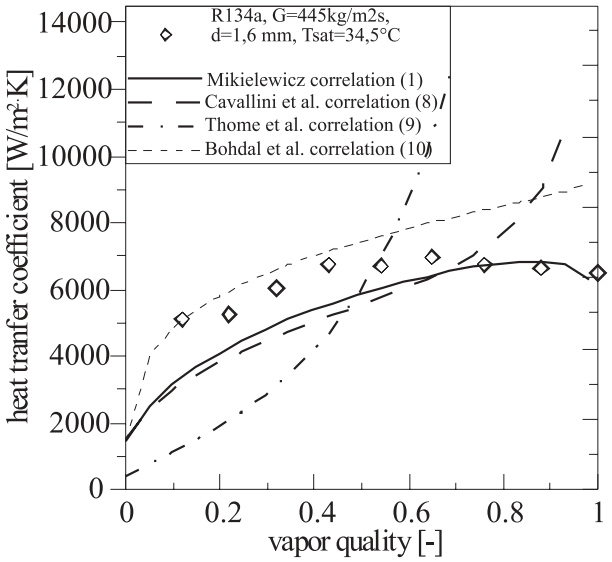

Figure 7. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 1.6$ mm, $G = 445$ kg/m^{2}s, $T_{sat} = 34.5$ °C.

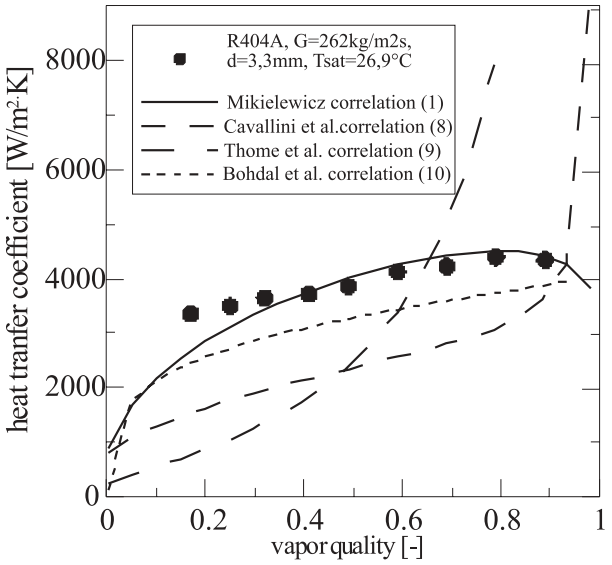

Figure 8. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 3.3$ mm, $G = 262$ kg/m^{2}s, $T_{sat} = 26.9$ °C.

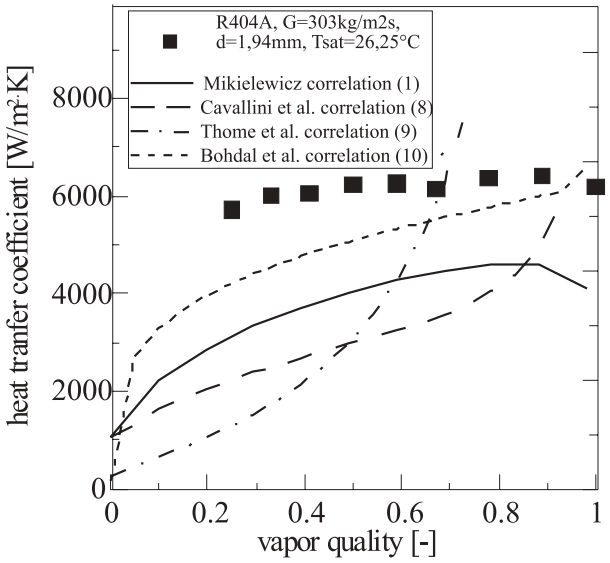

Figure 9. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 1.94$ mm, $G = 303$ kg/m^{2}s, $T_{sat} = 26.3$ °C.

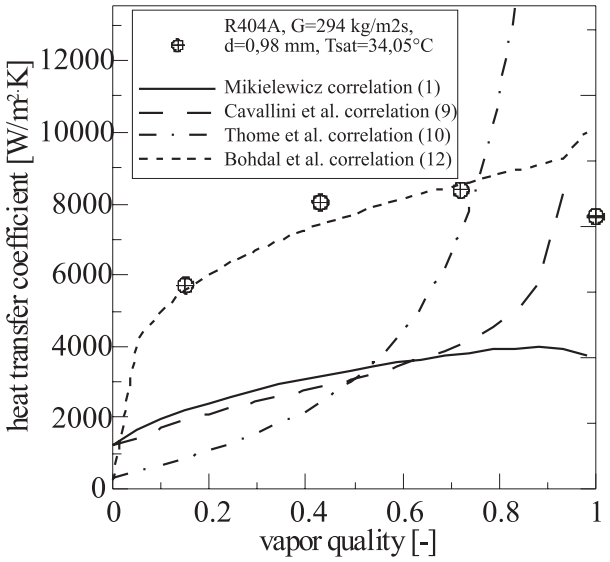

Figure 10. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 0.98$ mm, $G = 294$ kg/m^{2}s, $T_{sat} = 34.0$ °C.

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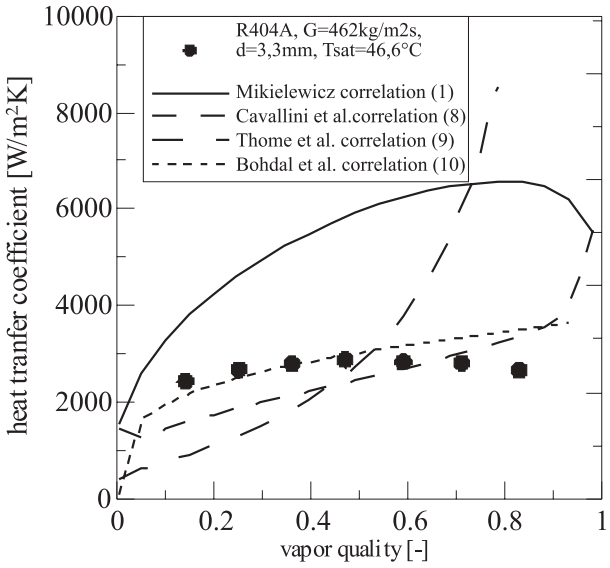

Figure 11. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 3.3$ mm, $G = 462$ kg/m^{2}s, $T_{sat} = 46.6$ °C.

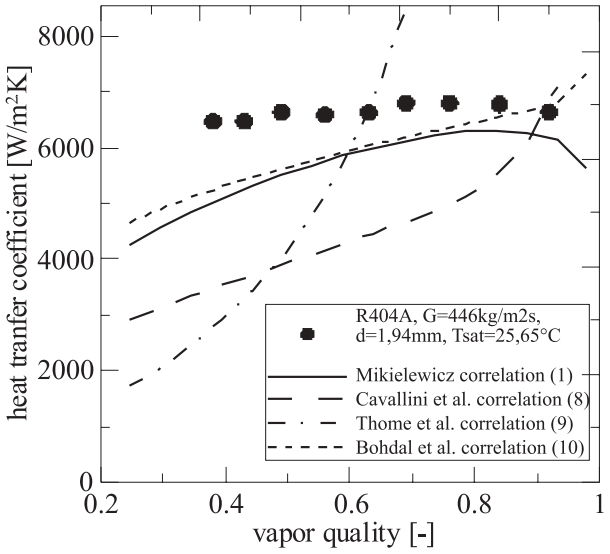

Figure 12. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 1.94$ mm, $G = 446$ kg/m^{2}s, $T_{sat} = 25.6$ °C.

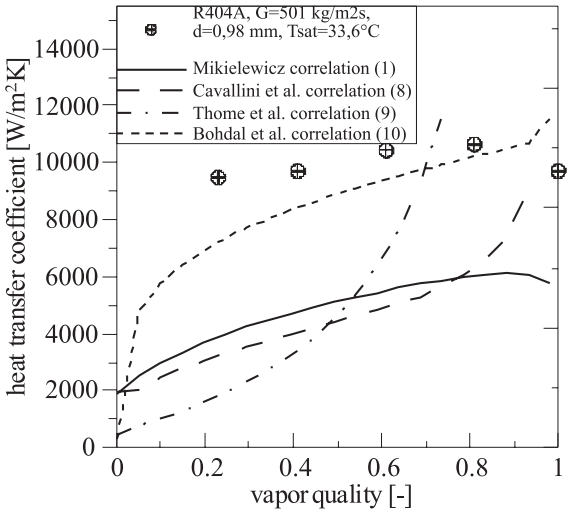

Figure 13. Comparison of heat transfer coefficient for R134a in relation to experimental data due to Bohdal *et al.* (2011), $d = 0.98$ mm, $G = 501$ kg/m^{2}s, $T_{sat} = 33.6$ °C.

5 Conclusions

The paper presents is a comparison of predictions of condensation inside conventional tubes for tube diameters, namely $d = 3.3$ mm and minichannels ($d < 3$ mm) with the Mikielewicz (2009) correlation originally developed for flow boiling. In the paper that method has been applied to predictions of heat transfer coefficient in flow condensation and has been verified by experimental data due to Bohdal *et al.* [1] and also compared to Cavallini's correlation (5), Thome's correlation (6), and Bohdal *et al.* (8) correlation. The comparison is satisfactory. The calculation shows that Eq. (1) outperforms other models, but is universal and can be used to predict heat transfer due to condensation for different halogeneous refrigerants and other fluids. Ways to improve the performance of correlation (1) have been presented. There are based on selection of more appropriate two-phase flow multiplier or to introduce non-isothermal or heat flow dependent terms into the two-phase flow multiplier.

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