archives
of thermodynamics
Vol. **35**(2014), No. 3, 17–37
DOI: 10.2478/aoter-2014-0019

Comparative study of heat transfer and pressure drop during flow boiling and flow condensation in minichannels

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Abstract In the paper a method developed earlier by authors is applied to calculations of pressure drop and heat transfer coefficient for flow boiling and also flow condensation for some recent data collected from literature for such fluids as R404a, R600a, R290, R32,R134a, R1234yf and other. The modification of interface shear stresses between flow boiling and flow condensation in annular flow structure are considered through incorporation of the so called blowing parameter. The shear stress between vapor phase and liquid phase is generally a function of nonisothermal effects. The mechanism of modification of shear stresses at the vapor-liquid interface has been presented in detail. In case of annular flow it contributes to thickening and thinning of the liquid film, which corresponds to condensation and boiling respectively. There is also a different influence of heat flux on the modification of shear stress in the bubbly flow structure, where it affects bubble nucleation. In that case the effect of applied heat flux is considered. As a result a modified form of the two-phase flow multiplier is obtained, in which the nonadiabatic effect is clearly pronounced.

Keywords: Two-phase pressure drops; Heat transfer coefficient; Boiling; Condensation

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Nomenclature

A	–	cross section area, m^2
B	–	blowing parameter
Bo	–	boiling number, $B_0 = \frac{q}{Gh_{LG}}$
C	–	mass concentration of droplets in two phase core
C_f	–	friction factor
Con	–	confinement number
c_p	–	specific heat, J/kg K
d	–	diameter, m
D	–	deposition term, kg/ms; channel inner diameter, m
E	–	entrainment term, energy dissipation, kg/ms
G	–	mass flux, kg/m^2s
g	–	gravitational acceleration, m/s^2
h	–	enthalpy, J/kg
h_{LG}	–	specific enthalpy of vaporization, J/kg
h_{lv}	–	specific enthalpy of vaporization,J/kg
m_G	–	mass of vapour phase
m_L	–	mass of liquid phase
$\dot{m}$	–	mass flux,kg/s
P	–	perimeter, m
p	–	pressure, Pa
Pr	–	Prandtl number
q	–	density of heat flux, W/m^2
$\dot{q}_w$	–	wall heat flux, W/m^2
Re	–	Reynolds number, $\mathrm{Re} = \frac{Gd}{\mu_L}$
Re_L	–	Reynolds number liquid film only $\mathrm{Re}_L = \frac{G\,d\,(1-x)}{\mu_L}$
$s = u_G/u_L$	–	slip ratio
u	–	velocity, m/s
$u^+ = u/u_h$	–	reduced speed
w	–	velocity, m/s
v_0	–	transverse velocity, m/s
$x = \frac{m_G}{m_G+m_L}$	–	quality,
z	–	longitudinal coordinate, m

Greek symbols

α	–	heat transfer coefficient,W/m^2K
δ	–	film thickness, m
σ	–	surface tension,N/m
λ	–	thermal conductivity, W/mK
ρ	–	liquid density, kg/m^3
μ	–	dynamic viscosity, Pa s
$\xi = \frac{C_f}{4}$	–	friction factor
τ	–	shear stress, N/m^2
φ	–	void fraction
Φ^2	–	two phase multiplier

Subscripts

c	–	core, cross section
cv	–	vapour core
cd	–	dropplets core
e	–	entrainmend
f	–	film
g	–	vapor
G	–	vapour
i	–	internal
lv	–	mixture of liquid and vapor
L	–	liquid
PB	–	pool boiling
sat	–	saturation
TP	–	two-phase flow
TPB	–	two-phase boiling
TPC	–	two-phase condensation
w	–	wall
v	–	vapor
σ	–	for no evaporation of the liquid film
∞	–	undisturbed
0	–	referencing case

1 Introduction

Generally, the nonadiabatic effects modify the friction pressure drop term and subsequently the heat transfer coefficient. That is the reason why it is impossible to use reciprocally existing models for calculations of heat transfer and pressure drop in flow boiling and flow condensation cases. In authors opinion the way to solve that is to incorporate appropriate mechanisms into the friction pressure drop term responsible for modification of shear stresses at the vapor-liquid interface, different for annular flow structure and different for other ones, generally considered here as bubbly flows. Postulated in the paper suggestion of considering the so called 'blowing parameter' in annular flow explains partially the mechanism of liquid film thickening in case of flow condensation and thinning in case of flow boiling. In other flow structures, for example the bubbly flow, there can also be identified other effects, which have yet to attract sufficient attention in literature. One of such effects is the fact that the two-phase pressure drop is modeled in the way that the influence of applied heat flux is not considered.

The objective of this paper is to present the capability of the flow boiling model, developed earlier, Mikielewicz [1] with subsequent modifications, Mikielewicz *et al.* [2], Mikielewicz [3], to model also flow condensation inside

tubes with account of nonadiabatic effects. In such case the heat transfer coefficient is a function of the two-phase pressure drop. Therefore some experimental data have been collected from literature to further validate that method for the case of other fluids. The literature data considered in the paper for relevant comparisons, in case of flow condensation, are due to Bohdal *et al.* [4], Cavallini *et al.* [5], Matkovic *et al.* [6], and in case of flow boiling, due to Lu *et al.* [7], Wang *et al.* [8]. The results of pressure drop calculations have been compared with some correlations from literature for minichannels, namely due to Mishima and Hibiki [10], Zhang and Webb [11] and a modified version of Muller-Steinhagen and Heck [12] model, Mikielewicz *et al.* [2]. Calculations have been also compared against some well established methods for calculation of heat transfer coefficient for condensation due to Cavallini *et al.* [5] and Thome *et al.* [9] and flow boiling.

2 Dissipation based two-phase pressure drop model

Flow resistance due to friction is greater than in case of single phase flow with the same flow rate. The two-phase flow multiplier is defined as a ratio of pressure drop in two-phase flow, $(dp/dz)_{TP}$, to the total pressure drop in the flow with either liquid of vapor, (dp/dz), present

$$\Phi^2 = \left(\frac{dp}{dz}\right)_{TP} \left(\frac{dp}{dz}\right)^{-1} . \tag{1}$$

Unfortunately, the correlations developed for conventional size tubes cannot be used in calculations of pressure drop in minichannels. In case of small diameter channels there are other correlations advised for use. Their major modification is the inclusion of the surface tension effect into existing conventional size tube correlations. Amongst the most acknowledged ones are those due to Mishima and Hibiki [10], Tran *et al.* [13] and Zhang and Webb [11].

The pressure drop model for two-phase flow condensation or flow boiling is developed on the basis of dissipation energy analysis, which is a fundamental hypothesis in the model under scrutiny here.The dissipation in two-phase flow can be modeled as a sum of two contributions, namely the energy dissipation due to shearing flow without the bubbles, E_{TP}, and dissipation resulting from the bubble generation, E_{PB}, Mikielewicz [1]

$$E_{TPB} = E_{TP} + E_{PB} . \tag{2}$$

Dissipation energy, E_{TP}, is expressed as power lost in the control volume. The term power refers to compensation of two-phase flow friction losses and is expressed through a product of shear stress and flow velocity. Analogically can be expressed the energy dissipation due to bubble generation in the two-phase flow. A geometrical relation between the friction factor in two-phase flow is obtained which forms a geometrical sum of two contributions, namely the friction factor due to the shearing flow without bubbles and the friction factor due to generation of bubbles, in the form

$$\xi_{TPB}^2 = \xi_{TP}^2 + \xi_{PB}^2 \, . \tag{3}$$

In the considered case ξ_{PB} is prone to be dependent on the applied wall heat flux. As can be seen from (3) the friction factors in two phase flow are summed up in a geometrical manner. The first term on the right hand side of (3) can be determined from the definition of the two-phase flow multiplier (1). Pressure drop in the two-phase flow without bubble generation can also be considered as a pressure drop in the equivalent flow of a fluid with velocity w_{TP}. The pressure drop of the liquid flowing alone can be determined from a corresponding single phase flow relation. In case of turbulent flow we will use the Blasius equation for determination of the friction factor, whereas in case of laminar flow the friction factor can be evaluated from the corresponding expression valid in the laminar regime. A critical difference of the method in comparison to other authors models is incorporation of the two-phase flow multiplier into modeling (1). There are specific effects related to the shear stress modifications, named here the nonadiabatic effects, which will be described below. One of the effects is pertinent to annular flows, whereas the other one to the bubbly flow.

2.1 Nonadiabatic effects in annular flow

The shear stress between vapor phase and liquid phase is generally a function of nonadiabatic effects. That is a major reason why up to date approaches, considering the issue of flow boiling and flow condensation as symmetric phenomena, are failing in that respect. The way forward is to incorporate a mechanism into the convective term responsible for modification of shear stresses at the vapor-liquid interface. The relationship describing the shear stress between liquid and vapor phase in annular flow can be modified by incorporation of the so called 'blowing parameter', B, which contributes to the liquid film thickening in case of flow condensation and thinning in case of flow boiling. Such idea stems from the earlier work

on the topic of the 'boundary layer intensification' by introduction of air transversely into the boundary layer as presented in Fig. 1, Mikielewicz [14]. Considered was a turbulent flow of incompressible fluid without pressure gradient over the interface between liquid and vapour, with the presence of transverse mass flux. On the basis of analysis of the continuity of mass and momentum equations derived has been the expression for the modification of shear stress in the boundary layer, which reads

$$\tau^+ = 1 + \frac{B}{\tau_0^+} u^+ \,. \tag{4}$$

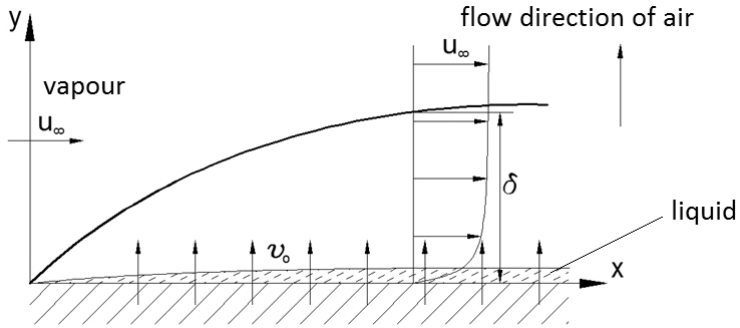

Figure 1: Injection of air into the boundary layer.

In (4) v_0 denotes the transverse velocity, $u^+ = u/u_h$, $\tau^+ = \tau/\tau_w$, $\tau_0^+ = \tau_w/\tau_{w0}$, where τ_{w0} is the wall shear stress in case where the air is not injected into the boundary layer, and $B = 2v_0/(C_f u_\infty)$ is the so called 'blowing parameter'. Using that idea it has been decided that the mechanism of liquid film thinning or thickening close to the wall can be modeled similarly. A possible confirmation of that comes from the works by Kutateladze and Leontiev [15] and Wallis [16], who studied the effect of shear stress modifications in flow boiling in vertical channels, and who developed the expressions linking the shear stress at the wall and the blowing parameter. The relation due to Kutateladze and Leontiev reads

$$\tau_0^+ = \left(1 - \frac{B}{4}\right)^2 \,. \tag{5}$$

On the other hand, in case of small values of B the relation given by Eq. (5)

reduces to that recommended by Wallis

$$\tau_0^+ = \left(1 - \frac{B}{2}\right) . \tag{6}$$

The expression (4) actually reduces to the expression (6) for the values of Reynolds number tending to infinity, encouraging to define the transverse velocity to be equal to $v_0 = \dot{q}_w/(h_{lv}\rho_l)$ in case of condensation or boiling. In case of small values of the blowing parameter B the relation (4) reduces to the form:

$$\tau_0^+ = \left(1 \pm \frac{B}{2}\right) . \tag{7}$$

The blowing parameter is hence defined as

$$B = \frac{2v_0}{C_f u_\infty} = \frac{2\dot{q}}{C_{f0}\left(u_G - u_L\right) h_{LG}\rho_G} = \frac{2q\frac{\rho_L}{\rho_G}}{C_{f0}G\left(s-1\right) h_{LG}} . \tag{8}$$

In the present paper another new approach to determine the blowing parameter B in function of vapor quality is presented.

2.2 Model of blowing parameter

Analysis of the liquid and vapor phase is based on examination of mass and momentum balance equations with respect to the non-adiabatic effect influence. Figure 2 shows the considered schematic of the annular flow model. The analysis will be conducted with the reference to condensation. In the model presented below the following notation is used. The liquid film cross-section area on the wall is expressed by $A_f = \pi d\delta$, while the core cross-section area as $A_c = \pi(d - 2\delta)^2/4$. The wetted perimeter is given by the relation $P_f = \pi d$, where d is the channel inner diameter. The mean liquid film velocity is given as $u_f = \dot{m}/(\rho_f A_f)$. Authors assumed that the interfacial velocity can be determined from the relationship $u_i = 2u_f$.

2.2.1 Mass balance in liquid film and core

Liquid film:

$$\frac{d\dot{m}_f}{dz} = -\Gamma_{lv} + D - E . \tag{9}$$

Two-phase core:

$$\frac{d\dot{m}_{cd}}{dz} = -D + E . \tag{10}$$

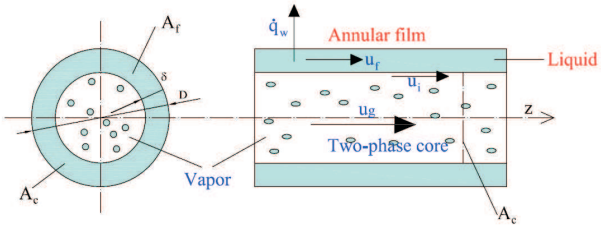

Figure 2: Annular flow structure model.

Vapor in the two-phase vapor core:

$$\frac{d\dot{m}_{cv}}{dz} = -\Gamma_{lv} \, . \tag{11}$$

In (9) and (10) the terms D and E denote deposition and entrainment in the annular flow. The remaining term in equation, $\Gamma_{lv} = \dot{q}_w P/h_{lv}$, is responsible for the condensation of vapor. Concentration of droplets in the core is defined as a ratio of mass flow rate droplets in the core to the sum of mass flow rate vapor and entrained liquid droplets from the flow:

$$C = \frac{\dot{m}_{cf}}{\dot{m}_{cv} v_g + \dot{m}_{ef} v_f} \, . \tag{12}$$

The combined mass flow rate of the core results from combination of (10) and (11):

$$\frac{d\dot{m}_c}{dz} = -\Gamma_{lv} - D + E \, . \tag{13}$$

The amount of entrained droplets in (12) can be determined from the mass balance:

$$\dot{m}_{ef} = \dot{m} - \dot{m}_f - \dot{m}_{cv} \, . \tag{14}$$

2.2.2 Momentum balance in liquid film and two-phase core

The change of momentum is mainly due to the mass exchange between the core of flow and liquid film (evaporation, droplet deposition or entrainment). Acceleration is neglected. The flow schematic is shown in Fig. 3.

2.2.3 Momentum equation for liquid film

Momentum equation for the liquid film reads:

$$-\frac{dp_L}{dz} dz \, (\delta - y) \, P_f - \tau P_f dz + \tau_i P_f dz = \left(\Gamma_{lv} u_i + D u_c - E u_i\right) dz \, . \tag{15}$$

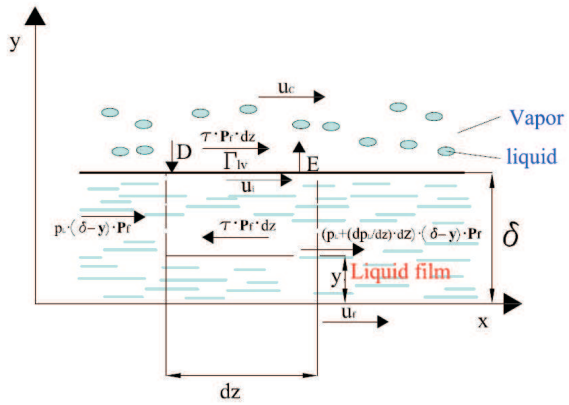

Figure 3: Flow diagram for the momentum analysis in the liquid film.

Pressure gradient in the liquid film is therefore (assuming that $\rho_f = \rho_l$ and $\mu_f = \mu_l$)

$$-\left(\frac{dp_l}{dz}\right) = \frac{3\mu_f \dot{m}_f}{P_f \rho_f \delta^3} - \frac{3\tau_i}{2\delta} + \frac{3\left(\Gamma_{lv} u_i + D u_f - E u_i\right)}{2\delta P_f} \,. \tag{16}$$

2.2.4 The momentum balance for the core flow

Control volume for the two-phase core is shown in Fig. 4.Momentum equation for the mixture in the core is given by equation:

$$\begin{aligned} &\rho_{TP} u_c^2 A_c + \tfrac{d}{dz}\left(\rho_{TP} u_c^2 A_c\right) dz - \rho_{TP} u_c^2 A_c + \left[-\Gamma_{lv} u_i - D u_c + E u_i\right] dz \\ &= p_v A_c - \left[p_v A_c + \tfrac{d(p_v A_c)}{dz}\Delta z\right] - \tau_i P dz \,. \end{aligned} \tag{17}$$

From Eq. (17) it follows that interfacial shear stress are:

$$\tau_i = \frac{1}{P}\left[A_c\left(-\frac{dp_v}{dz}\right) - p_v \frac{dA_c}{dz}\right] - \frac{1}{P}\frac{d}{dz}\left(\rho_{TP} u_c^2 A_c\right) - \frac{1}{P}\left(-\Gamma_{lv} u_i - D u_c + E u_i\right) \tag{18}$$

The relationship expresses the interfacial shear stress for the two-phase flow (here condensation), and included are the non-adiabatic effects: liquid film condensation, droplets deposition and entrainment. When there is no evaporation of the liquid film, but entrainment and deposition are, the interfacial shear stress distribution takes the form

$$\tau_{io} = \frac{-\frac{1}{A_c}\left(-D u_c + E u_i\right) + \frac{3\mu_f \dot{m}_f}{P_f \rho_f \delta^3} + \frac{3}{2\delta P_f}\left(-D u_f + E u_i\right)}{\frac{P_f}{A_c} + \frac{3}{2\delta}} \,. \tag{19}$$

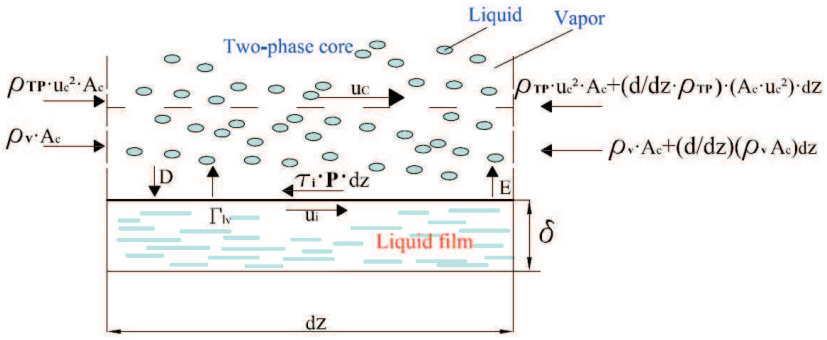

Figure 4: Control volume for the two-phase core.

In case one could neglect the entrainment and deposition, i.e., by assigning $E = 0$ and $D = 0$, were obtained a very simplified form of the diabatic two-phase flow effect in the form

$$\frac{\tau_i}{\tau_{io}} = 1 + \frac{2q_w\delta\left(\frac{4\delta}{D} + \frac{3}{2}\right)}{3\mu_f h_{lv}} = (1 + B)\,. \tag{20}$$

Figures 5 and 6 present the results of sample calculations of the blowing parameter for boiling of R290 at parameters: $G = 74$ kg/m^{2}s, $T_{sat} = -1.9$ °C in a 2.6 mm tube, and for R600a: $G = 440$ kg/(m^{2}s), $T_{sat} = 22$ °C in a 2.6 mm tube. When the parameter is calculated by Eq. (8) then $B = 0.133$ for R290 and $B = 0.023$ for R600a. The result from application of Eq. (19) is $B = 0.095$, and 0.025, respectively. This shows satisfactory consistency of calculations.

2.3 Nonadiabatic effects in other than annular flow

In case of the nonadiabatic effects in other than annular structures authors presented their idea in Mikielewicz [3]. The two-phase flow multiplier, which incorporates the non-adiabatic effect, resulting from (3), reads:

$$\Phi^2_{TPB} = \frac{\xi_{TPB}}{\xi_0} = \sqrt{\Phi^2 + \frac{\xi^2_{PB}}{\xi^2_0}} = \Phi^2\sqrt{1 + \frac{\left(\frac{8\alpha_{PB}d}{\lambda \mathrm{ReP_r}}\right)^2}{\xi^2_0\Phi^2}}\,. \tag{21}$$

The two-phase flow multiplier presented by the above equation reduces to adiabatic formulation in case when the applied wall heat flux is tending to zero.

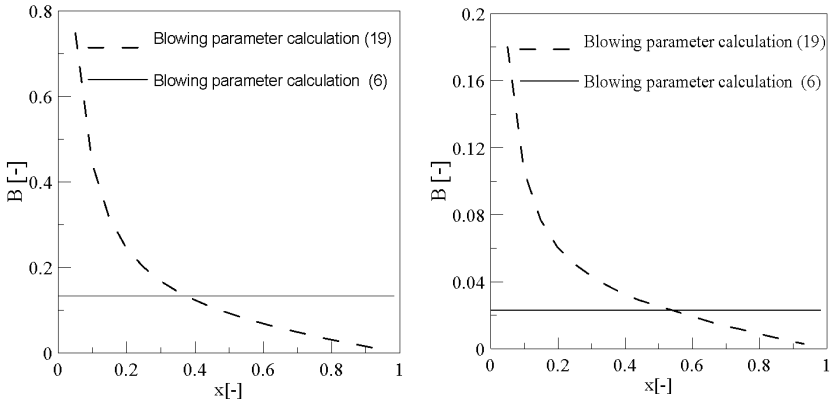

Figure 5: Blowing parameter as a function vapor quality at the left for R290 and at the right for R 600a.

Generalizing the obtained above results it can be said that the two-phase flow multiplier inclusive of non-adiabatic effects can be calculated, depending upon the particular flow case and the flow structure in the following way:

$$\Phi^2_{TPC} = \Phi^2_{TPB} = \frac{\xi_{TPB}}{\xi_0} = \begin{cases} \Phi^2\left(1 \pm \frac{B}{2}\right) & \text{for annular structure, condensation and boiling} \\ \Phi^2\sqrt{1 + \left(\frac{8\alpha_{PB}D}{\lambda \mathrm{RePr}\xi_0\Phi^2}\right)^2} & \text{for other flow structures} \end{cases} \tag{22}$$

In (21) there is no specification on which two-phase flow multiplier model should be applied. That issue is dependent upon the type of considered fluid. The effect of incorporation of the blowing parameter into pressure drop predictions is shown in Figs. 6–8.

In the presented case the effect of considering the blowing parameter may reach even 20% effect. The authors own correlation is shows best compatibility with the experimental data. In the case of pressure drops the good agreement with experimental data shows also Mishima and Hibiki *et al.* [10] correlation and relatively good correctness shows Tran *et al.* relationship [13] .

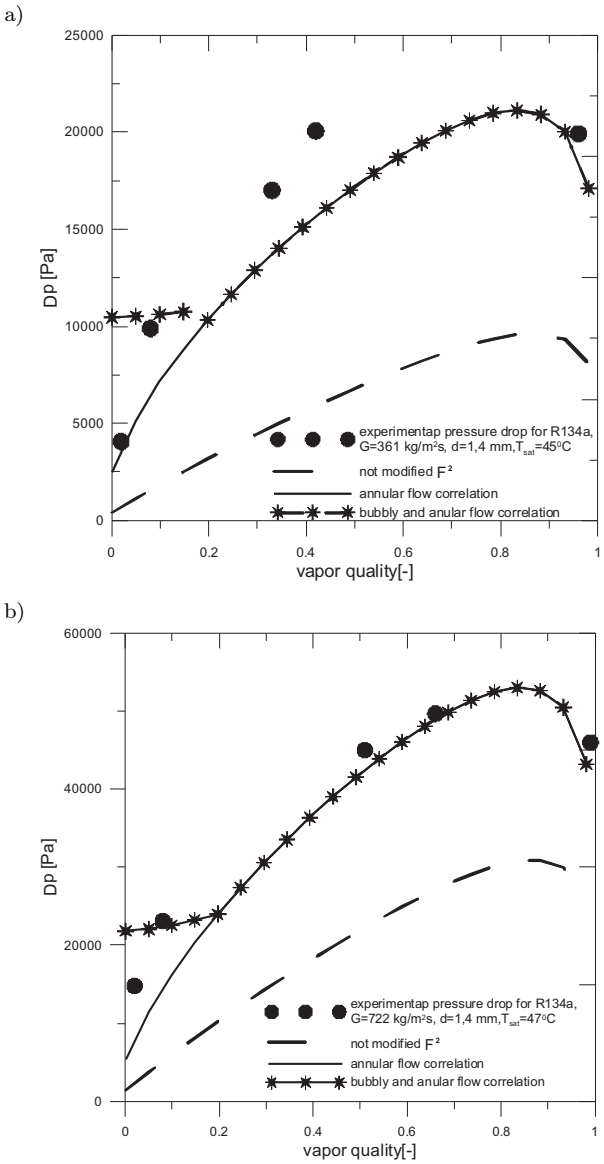

Figure 6: Condensation pressure drop in function of quality, Bohdal *et al.* [4], R134a: a) $G = 361$ kg/m^{2}s, $T_{sat} = 45$ °C, $d = 1.4$ mm; b). $G = 722$ kg/m^{2}s, $T_{sat} = 47$ °C, $d = 1.4$ mm.

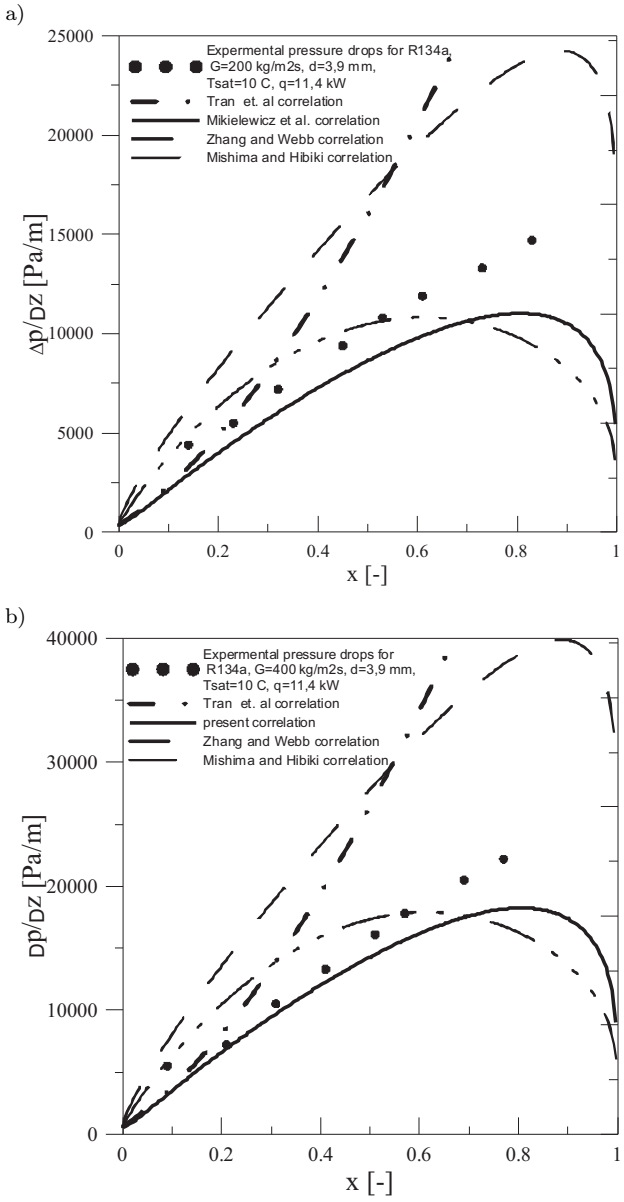

Figure 7: Flow boiling pressure drop in function of quality for R134a, Lu *et al.* [7], T_{sat} = 10 °C, $q = 11.4$ kW/m^2, $d = 3.9$ mm: a) $G = 200$ kg/m^{2}s, b) $G = 400$ kg/m^{2}s.

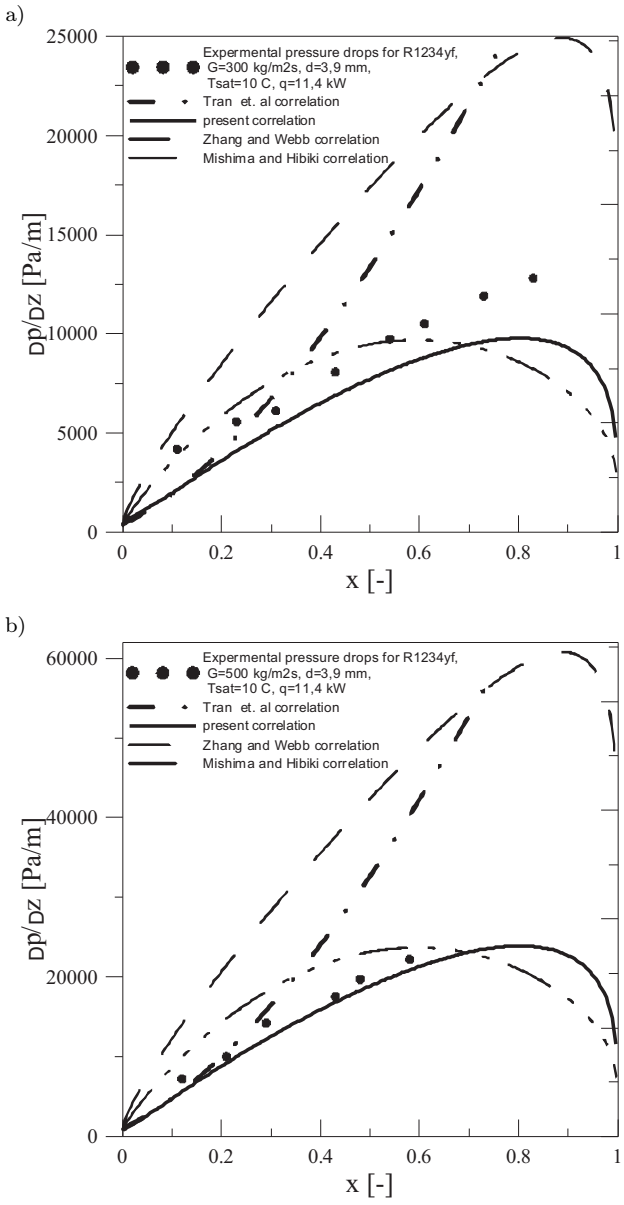

Figure 8: Flow boiling pressure drop in function of quality for R1234yf, Lu *et al.* [7], $T_{sat} = 10$ °C, $q = 11.4$ kW/m^2, $d = 3.9$ mm: a) $G = 300$ kg/m^{2}s, b) $G = 500$ kg/m^{2}s.

3 Heat transfer in phase change

The heat transfer correlation applicable both to the case of flow boiling and flow condensation:

$$\frac{\alpha_{TP}}{\alpha_l} = \sqrt{(\Phi^2)^n + \frac{C_1}{1+P_1}\left(\frac{\alpha_{TP}}{\alpha_l}\right)^2} \,. \tag{23}$$

In case of condensation the constant $C_1 = 0$, whereas in case of flow boiling $C_1 = 1$. In Eq. (22) $B = q_w/(Gh_{lv})$ and the correction factor is

$$P_1 = 2.53 \times 10^{-3} \times \mathrm{Re}_l^{1.17} \times \mathrm{Bo}^{0.6} \times \left(\Phi^2 - 1\right)^{-0.65} \,. \tag{24}$$

In the form applicable to conventional and small-diameter channels, the modified Muller-Steinhagen and Heck model is advised, Mikielewicz *et al.* [2]

$$\Phi^2 = \left[1 = 2\left(\frac{1}{f_l} - 1\right)\mathrm{Con}^m\right](1-x)^{\frac{1}{3}} + x^3\frac{1}{f_{lz}} \,. \tag{25}$$

The exponent at the confinement number m assumes a value $m = 0$ for conventional channels and $m = -1$ in case of small diameter and minichannels. Within the correction factor P the modified version of the Muller-Steinhagen and Heck model should be used, however instead of the f_{1z} a value of the function f_l must be used. In (24) $f_l{=}(\rho_L/\rho_G)\ (\mu_L/\mu_G)^{0.25}$ for turbulent flow and $f_l{=}(\rho_L/\rho_G)(\mu_L/\mu_G)$ for laminar flows. Introduction of the function f_{lz}, expressing the ratio of heat transfer coefficient for liquid only flow to the heat transfer coefficient for gas only flow, is to meet the limiting conditions, i.e., for $x = 0$ the correlation should reduce to a value of heat transfer coefficient for liquid, $\alpha_{TPB} = \alpha_L$ whereas for $x = 1$, approximately that for vapor, i.e. $\alpha_{TPB} \cong \alpha_G$. Hence $f_{lz} = \alpha_{GO}/\alpha_{LO}$, where $f_{lz}{=}(\lambda_G/\lambda_L)$ for laminar flows and for turbulent flows $f_{lz}{=}(\mu_G/\mu_L)(\lambda_L/\lambda_G)^{1.5}(\mathrm{c}_{pL}/\mathrm{c}_{pG})$. The pool boiling heat transfer coefficient α_{PB} is calculated from a relation due to Cooper [18].

The correctness of the calculations was compared due to experimental data and the own correlation (22). A few examples of comparisons are presented in Figs. 9,10 for flow boiling of R134a and R1234yf. Presented next is a comparison of selected correlations for calculations of flow condensation with the model presented earlier, Figs. 11,12.

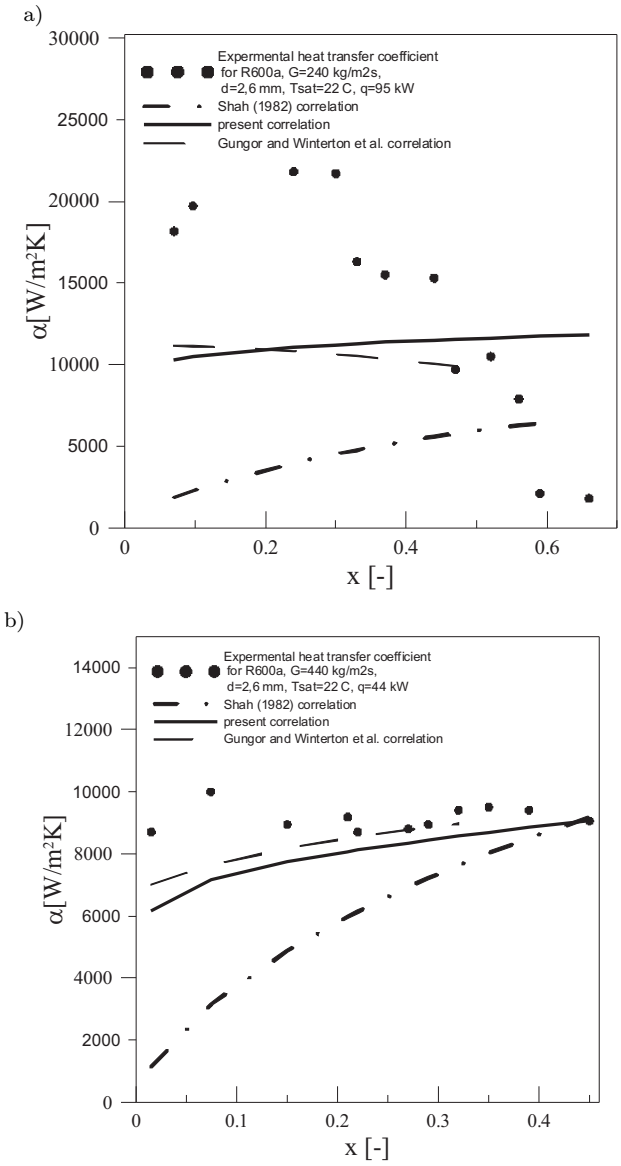

Figure 9: Flow boiling heat transfer coefficient in function of quality for R600a Copetti et al. [17]; $T_{sat} = 22\ ^{\circ}\mathrm{C}$, $d = 2.6$ mm; a) $G = 240$ kg/m^{2}s, $q = 95$ kW/m^2; b) $G = 440$ kg/m^{2}s, $q = 44$ kW/m^2

4 Conclusions

In the paper presented is a model of annular flow to incorporate the non-adiabatic effects in predictions of pressure drop and heat transfer for the

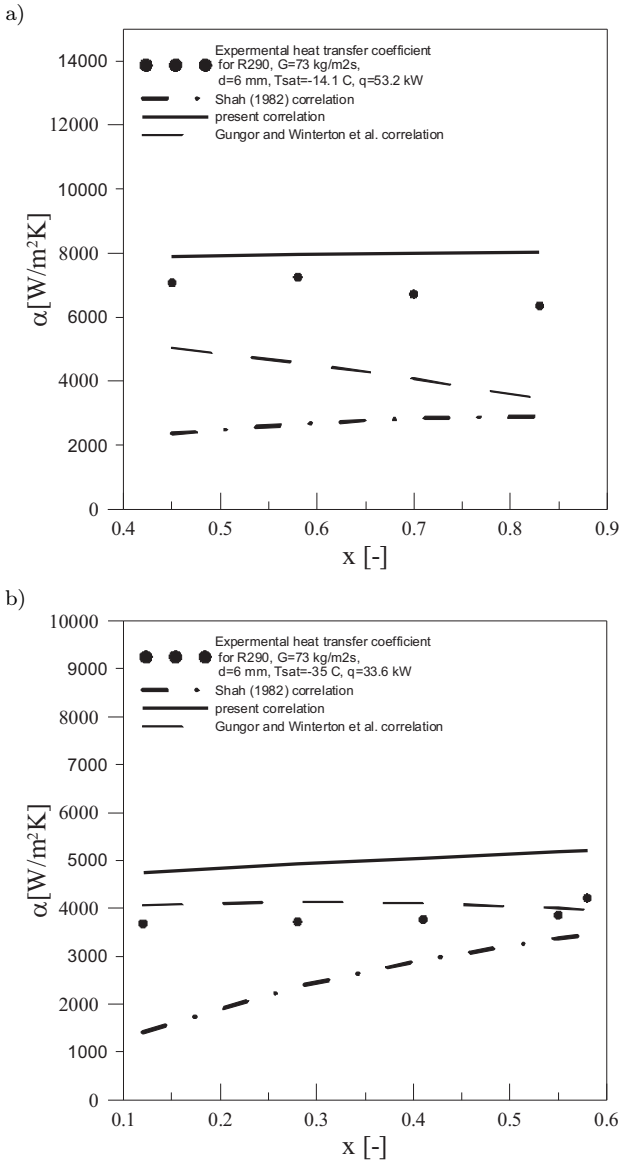

Figure 10: Flow boiling heat transfer coefficient in function of quality for R290 Wang *et al.* [8]; $G = 73$ kg/m^{2}s, $d = 6$ mm; a) $T_{sat} = 14.1$ °C, $q = 53.2$ kW/m^2; b) $T_{sat} = 35$ °C, $q = 44$ kW/m^2

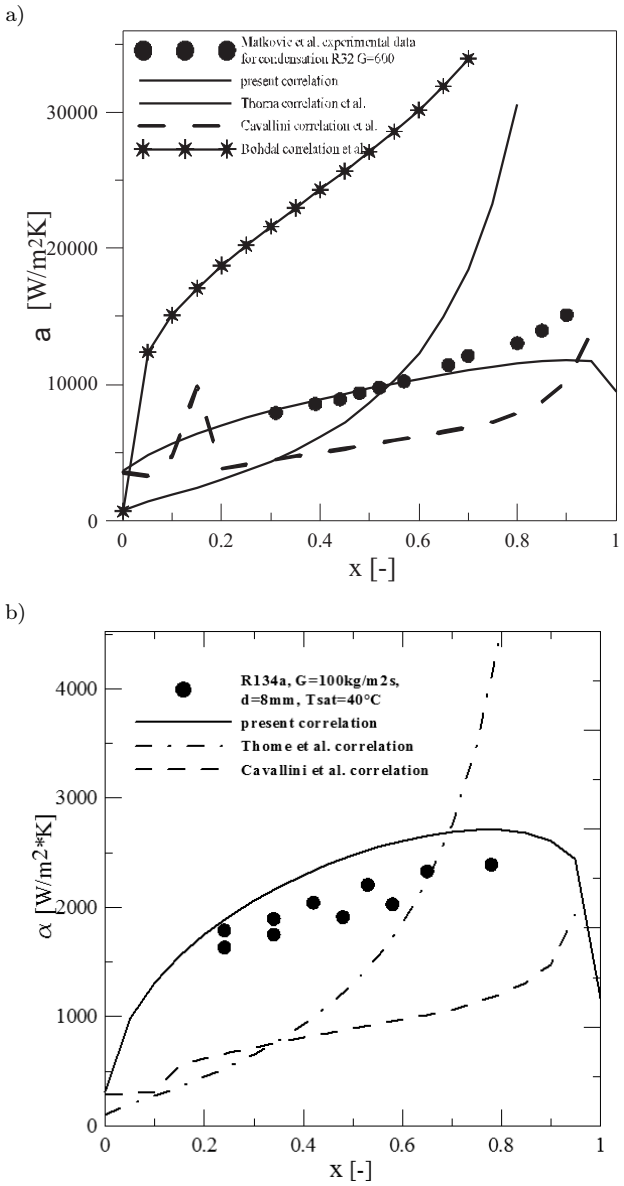

Figure 11: Flow boiling heat transfer coefficient in function of quality for R32 Matkovic *et al.* [6]; a) $G = 600$ kg/m^2, $T_{sat} = 14.1$ °C, $d = 0.96$ mm; b) $G = 100$ kg/m^2, $T_{sat} = 40$ °C, $d = 8$ mm.

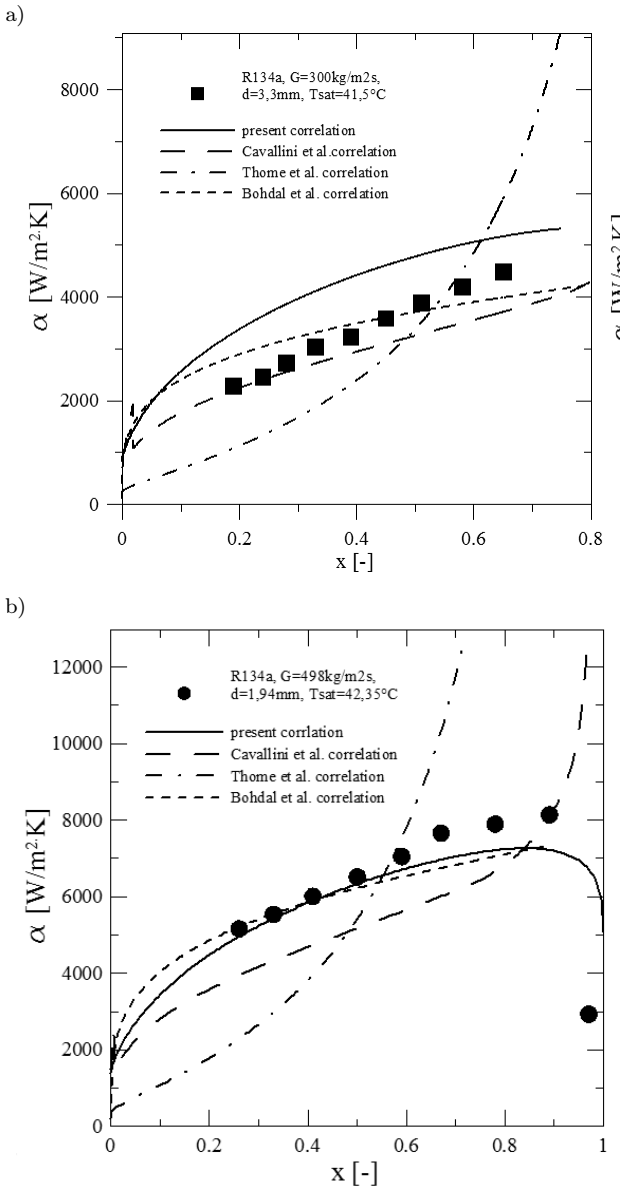

Figure 12: Flow boiling heat transfer coefficient in function of quality for R134a Bohdal *et al.* [4]; a) $G = 300$ kg/m^2, $T_{sat} = 41.5$ °C, $d = 3.3$ mm; b) $G = 498$ kg/m^2 $T_{sat} = 42.35$ °C, $d = 1.94$ mm.

case of flow boiling and flow condensation. The model is general, and is applicable to flow boiling and flow condensation. As a result of the model the expression for modification of interface shear stress has been postulated. In effect the modification is presented in relation to quality. The model can be included into any two-phase flow multiplier definition. In the present work such model has been incorporated into authors own model, which is a modification of the Muller-Steinhagen and Heck model. The comparison of predictions of boiling and condensation pressure drop and heat transfer coefficient inside minichannels have been presented together with the recommended correlations from literature. Calculations show that the model outperforms other ones, is universal and can be used to predict heat transfer due to flow boiling and flow condensation in different halogeneousand natural refrigerants.

Acknowledgments The work presented in the paper has been partially funded from the statute activity of the Faculty of Mechanical Engineering of Gdańsk University of Technology in 2014.

Received 18 June 2014

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