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

Dariusz Mikielewicz[1,*], Rafał Andrzejczyk[1], Blanka Jakubowska[1], Jarosław Mikielewicz[2]

* Corresponding author: Tel.: ++44 (0)1895 267132; Fax: ++44 (0)1895 256392; Email: Dariusz.Mikielewicz@pg.gda.pl, rafal.andrzejczyk@pg.gda.pl, jakubowska.blanka@gmail.com, jarekm@imp.gda.pl
1: Gdansk University of Technology, Faculty of Mechanical Engineering, Department of Energy and Industrial Apparatus, ul. Narutowicza 11/12, 80-233 Gdansk, Poland

2: The Szewalski Institute of Fluid-Flow Machinery PAS, ul. Fiszera 14, 80-231 Gdansk, Poland

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 R245fa, R600a, R134a, R1234yf and other. The modification of interface shear stresses between flow boiling and flow condensation in annular flow structure is considered through incorporation of the so called blowing parameter. The shear stress between vapor phase and liquid phase is generally a function of non-isothermal 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 the 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 non-adiabatic effect is clearly pronounced.

Keywords: two-phase pressure drops, heat transfer coefficient, boiling, condensation

1. Introduction

Generally, the non-adiabatic effects modify the friction pressure drop term and subsequently the heat transfer coefficient. That is the reason why it is impossible to use reciprocally the 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 issue 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 annular flow structures. 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 by Mikielewicz [1] with subsequent modifications, Mikielewicz et al [2], Mikielewicz [3], to model also flow condensation inside tubes with account of non-adiabatic 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 are due to

Bohdal et al. [4], Cavallini et al. [5], Matkovic et. al. [6], for flow condensation and due to Lu et al. [7] and Wang et al. [8] for flow boiling. 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]. Finally, authors compared their pressure drop calculations in minichanells with some correlations from literature, namely due to Mishima and Hibiki [10], Zhang and Webb [11] and a modified version of Muller-Steinhagen and Heck [12] model, [2], and Tran et al. [13].

2. Two-phase pressure drop model based on dissipation

Flow resistance due to friction is greater than that 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, $(dp/dz)_0$, present:

$$\Phi^2 = \left(\frac{dp}{dz}\right)_{TP} \left(\frac{dp}{dz}\right)_0^{-1} \tag{1}$$

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].

2.1 Dissipation based model for pressure drop calculations in flow boiling and flow condensation

The fundamental hypothesis in the model under scrutiny here is the fact that 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}, [1]:

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

Dissipation energy 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 the 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/collapse 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 applied wall heat flux. That term will be modified in the remainder of the text to include the heat flux dependence. 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 be considered as a pressure drop in the equivalent flow of a fluid flowing 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 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 flow regime. A critical difference of the method (1) in comparison to other authors models is incorporation of the two-phase flow multiplier into modeling. There are specific effects related to the shear stress modifications, named here the non-adiabatic effects, which will be described below. One of the effects is pertinent to annular flows, whereas the other one to the bubbly flow.

2.2 Non-adiabatic effects in annular flow

The shear stress between vapor phase and liquid phase is generally a function of non-adiabatic 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. We will attempt now to modify the shear stress between liquid and vapor phase in annular flow 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, Mikielewicz (1978). The formula for modification of shear stresses in the boundary layer reads:

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

In (4) $\tau^{+}=\tau/\tau_w$, $\tau_0^{+}=\tau_w/\tau_{w0}$, where τ_{w0} is the wall shear stress in case where the non-adiabatic effects are not considered, and $B=2\vartheta_0/(c_f\ u_\infty)$ is the so called "blowing parameter". Additionally, ϑ_0 denotes the transverse velocity, which in case of condensation or boiling is equal to $q_w/(h_{lv}\ \rho_l)$. In case of small values of B the relation (4) reduces to the form, and such a form will be used later in relevant modifications:

$$\tau_0^{+} = \left(1 \pm \frac{B}{2}\right) \tag{5}$$

The blowing parameter is hence defined as:

$$B = \frac{2\vartheta_0}{c_f u_\infty} = \frac{2q}{c_{f0}(u_G - u_L)h_{lv}\rho_G} = \frac{2q\dfrac{\rho_L}{\rho_G}}{c_{f0}G(s-1)h_{lv}} \tag{6}$$

In (6) s denotes the slip velocity and G - mass velocity. In the present paper a new approach to determination of the blowing parameter in function of vapor quality is presented.

2.3 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. Fig. 1 shows the considered schematic of the annular flow model. The analysis will be conducted with the reference to condensation.

Conservation of mass requires that the mass flow rate of liquid in the film, liquid in the form of droplets in the core and vapor in the core is constant:

$$\dot{m} = \dot{m}_f + \dot{m}_{cd} + \dot{m}_{cv} \tag{7}$$

In the model presented below the following notation is used. The liquid film cross-section area is expressed by the expression $A_f=\pi D\delta_f$, while the core cross-section area as $A_c=\pi(D-\delta_f)^2/4$. 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 $u_i=2u_f$.

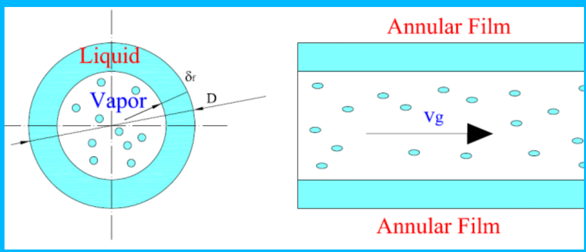

Fig. 1 Annular flow structure model.

Mass balance in liquid film and core

Liquid film:

$$\frac{dm_f}{dz} = -\Gamma_{lv} + D - E \tag{8}$$

Two-phase flow vapor core:

$$\frac{dm_{cd}}{dz} = -D + E \tag{9}$$

Vapor in vapor core:

$$\frac{dm_{cv}}{dz} = -\Gamma_{lv} \tag{10}$$

In (8) and (9) the terms D and E denote deposition and entrainment in the annular flow. The remaining term in equation, namely $\Gamma_{lv}=q_w P/h_{lv}$, 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.

$$C = \frac{\dot{m}_{cf}}{\dot{m}_{cv} v_g + \dot{m}_{ef} v_f} \qquad (11)$$

The combined mass flow rate of the core results from combination of (9) and (10):

$$\frac{dm_c}{dz} = -\Gamma_{lv} - D + E \qquad (12)$$

The amount of entrained droplets in (11) can be determined from the mass balance:

$$\dot{m}_{ef} = \dot{m} - \dot{m}_f - \dot{m}_{cv} \qquad (13)$$

Momentum balance in liquid film and 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 Figure 2.

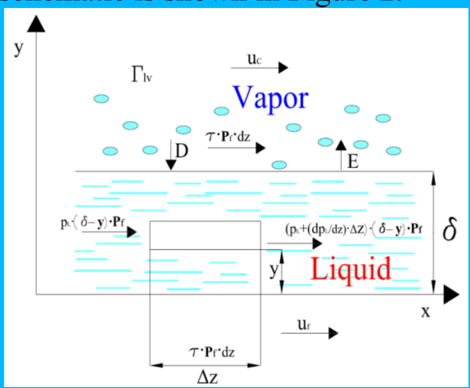

Fig. 2. Flow diagram for the momentum analysis in the liquid film

Momentum equation for liquid film

Momentum equation for the liquid film reads:

$$-\frac{dp_L}{dz}\Delta z \cdot (\delta - y)\cdot P_f - \tau P_f \Delta z + \tau_i P_f \Delta z = \left(\Gamma_{lv} u_i + D u_c - E u_i\right)\cdot \Delta z \qquad (14)$$

The shear stresses in the liquid film can be expressed by:

$$\tau = -\frac{dp_L}{dz}(\delta - y) + \tau_i + \frac{1}{P_f}\left(\Gamma_{lv} u_i + D u_c - E u_i\right) \qquad (15)$$

The relation between vapor-liquid equilibrium results from the Laplace equation:

$$p_v - p_l = \frac{\sigma}{r} \qquad (15)$$

After differentiation, (15) takes the form:

$$\frac{dp_v}{dz} - \frac{dp_l}{dz} = \frac{\sigma}{r^2}\frac{dr}{dz} \qquad (16)$$

According to Fig. 2 the radius of vapor is r=(D-2δ)/2, which after differentiation yields dr/dz=-0.5(dδ/dz). Shear stress in the liquid is:

$$\tau = \mu \frac{du_f}{dy} \qquad (17)$$

Using equation (15) and (17), we obtain the velocity profile in the liquid film:

$$u_f = \frac{1}{\mu_l}\left(\delta_y - \frac{1}{2}y^2\right)\left(-\frac{dp_l}{dz}\right) + \frac{y}{\mu_l}\tau_i - \frac{y}{\mu_l u P_f}\left(\Gamma_{lv} u_i + D u_c - E u_i\right) \qquad (18)$$

Mass flow rate of the liquid film is defined as:

$$\dot{m}_f = \rho_l P_f \int_0^{\delta} u_f \, dy \qquad (19)$$

Substituting (22) into (21) and integrating:

$$\dot{m}_f = \frac{P_f \rho_f \delta^3}{3\mu_l}\left(-\frac{dp_l}{dz}\right) + \frac{3 P_f \rho_f \delta^3}{6\mu_l \delta}\tau_i - \frac{3\rho_f P_f \delta^3}{6\mu\ \delta P_f}\left(\Gamma_{lv} u_i + D u_f - E u_i\right) \qquad (20)$$

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

$$-\left(\frac{dp_l}{dz}\right) = \frac{3\mu_f \dot{m}_f}{P_f \rho_f \delta^3} - \frac{3\tau_i}{2\delta} + \frac{3\left(\Gamma_{lv} u_i + D u_f - E u_i\right)}{2\delta P_f} \qquad (21)$$

The momentum balance for the core flow

Control volume for the flow core is shown in Fig. 3 where momentum equation for the mixture in the core is given by equation:

$$\rho_{TP} u_c^2 A_c + \frac{d}{dz}\left(\rho_{TP} u_c^2 A_c\right)\Delta z - \rho_{TP} u_c^2 A_c + \left[-\Gamma_{lv} u_i - D u_c + E u_i\right]\Delta z = p_v A_c - \left[p_v A_c + \frac{d\left(p_v A_c\right)}{dz}\Delta z\right] - \tau_i P \Delta z \qquad (22)$$

From equation (22) it follows that interfacial shear stress are:

$$\tau_i = \frac{1}{P}\left[A_c\left(-\frac{dp_v}{dz}\right) - p_v\frac{dA_c}{dz}\right] - \frac{1}{P}\frac{d}{dz}\cdot \left(\rho_{TP}u_c^2 A_c\right) - \frac{1}{P}\left(-\Gamma_{lv}u_i - Du_c + Eu_i\right) \quad (23)$$

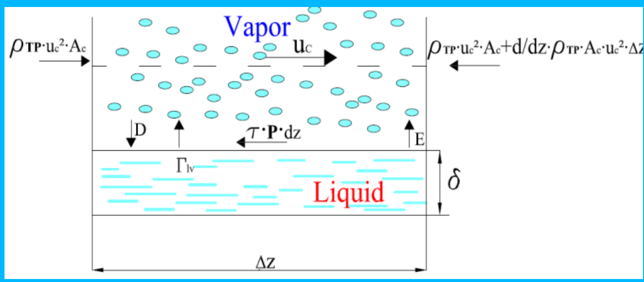

Fig. 3 Control volume for the flow core

In (23) it is assumed that the perimeter of the vapor core P≈π(D-2δ). Respectively the differentiated cross-section has the form:

$$\frac{dA_c}{dz} = -\frac{2\pi}{2}(D-\delta)\frac{d\delta}{dz} = -\pi(D-2\delta)\frac{d\delta}{dz} \quad (24)$$

The interfacial shear stress is defined as:

$$\tau_i = \frac{1}{2}\rho_{TP}(u_c - u_i)^2 \quad (25)$$

The Reynolds number for the core is:

$$\mathrm{Re}_c = \frac{\rho_{TP}(u_c - u_i)d_c}{\mu_g} \quad (26)$$

The interface friction factor can be taken from:

$$f_i = 0.005\left(1 + 300\frac{\delta}{D}\right) \quad (27)$$

In order to use equation (27) is necessary liquid film thickness. It has been determined according to Thome et al. [9] as:

$$\delta = \sqrt{\frac{f_w}{f_i}\frac{\rho_l}{\rho_g}\frac{J_l}{J_g}}D \quad (28)$$

The modification of interfacial shear stress by the action of the transverse mass flow yields:

$$\tau_i = f_i\left[\frac{1}{2}\rho_{TP}(u_c - u_i)^2\right] - \frac{\Gamma_{lv}}{2P}(u_c - u_i) \quad (29)$$

The sought unknowns in this issue are: liquid film mass flow $\dot{m}_f$, liquid film thickness δ, interfacial shear stress τ_i. In the mini-channel the dominating flow structure is annular flow. Let us now focus at the effect of phase change impact on modification of shear stress τ_i. Shear stress resulting from the model yields:

$$\tau_i = \frac{1}{P}\left[A_c\left(-\frac{dp_v}{dz}\right)\right] - \frac{1}{P}\left[-\Gamma_{lv}u_i - Du_c + Eu_i\right] \quad (30)$$

Pressure liquid and vapor p_l and p_v are linked through the Laplace equation $p_l - p_v = \sigma/r$. We ignore the effect of the surface tension of the liquid in a first approximation. In this case, equation (30) will adopt the form:

$$-\frac{A_c}{P}\frac{dp_v}{dz} = \tau_i + \frac{1}{P}\left[-\Gamma_{lv}u_i - Du_c + Eu_i\right] \quad (31)$$

Comparing (31) and (21), which are the expressions for pressure drop in liquid and vapor returns a relationship on the shear stress:

$$\tau_i = \frac{-\frac{1}{A_c}\left(-\Gamma_{lv} - Du_c + Eu_i\right) + \frac{3\mu_f \dot{m}_f}{P_f\rho_f\delta^3}}{\frac{P}{A_c} + \frac{3}{2\delta}} + \frac{\frac{3}{2\delta\cdot P_f}\left(\Gamma_{lv}u_i - Du_f + Eu_i\right)}{\frac{P}{A_c} + \frac{3}{2\delta}} \quad (32)$$

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

$$\tau_{io} = \frac{-\frac{1}{A_c}\left(-Du_c + Eu_i\right) + \frac{3\mu_f \dot{m}_f}{P_f\rho_f\delta^3}}{\frac{P_f}{A_c} + \frac{3}{2\delta}} + \frac{\frac{3}{2\delta P_f}\left(-Du_f + Eu_i\right)}{\frac{P_f}{A_c} + \frac{3}{2\delta}} \quad (33)$$

In case we can neglect the entrainment and deposition i.e. by assigning E = 0 and D = 0, we obtain a very simplified form of the diabatic two-phase flow effect in the form:

$$\frac{\tau_i}{\tau_{io}} = 1 + \frac{2 q_w \delta \left(\frac{4\delta}{D} + \frac{3}{2} \right)}{3 \mu_f h_{lv}} = (1 + B) \qquad (34)$$

Fig. 4 presents sample calculations of the blowing parameter for condensation of HFE7100 at parameters: G = 483kg/m^{2}s, T_{sat} = 74 °C, and for R134a: G=300 kg/m^{2}s, T_{sat} =10°C in a 1mm tube. When the parameter is calculated by equation (13) then B = 0.137 for HFE7100 and B=0.014 for R134a. The result from application of (34) is B=0.127 and B=0.016, respectively. This shows satisfactory consistency of calculations.

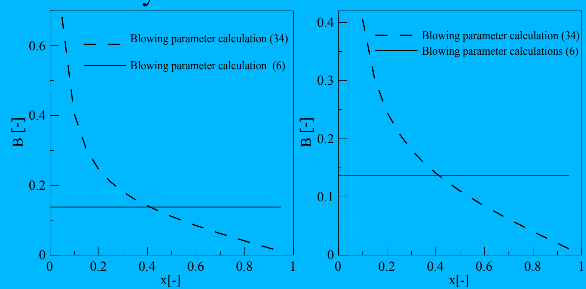

Fig. 4 Blowing/suction parameter as a function vapor quality for HFE7100 and R134a.

Non-adiabatic effects in other than annular flows

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

$$\Phi^2_{TPB} = \frac{\xi_{TPB}}{\xi_0} = \sqrt{\Phi^2 + \frac{\xi^2_{PB}}{\xi^2_0}} = \Phi^2 \sqrt{1 + \frac{\left(\frac{8 \alpha_{PB} d}{\lambda \operatorname{Re} \Pr} \right)^2}{\xi^2_0 \Phi^2}} \qquad (35)$$

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.

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:

$$\Phi^2{}_{TPC} = \Phi^2{}_{TPB} = \frac{\xi_{TPB}}{\xi_0} = \begin{cases} \Phi^2 \left(1 \pm \frac{B}{2} \right) & \textit{for annular structure, condensation and boiling} \\ \Phi^2 \sqrt{1 + \left(\frac{8 \alpha_{PB} D}{\lambda \operatorname{Re} \Pr \xi_0 \Phi^2} \right)^2} & \textit{for other flow structures} \end{cases} \qquad (36)$$

In (36) there is no specification of which two-phase flow multiplier model should be applied. That issue is dependent upon the type of considered fluid and other recommendations.

The effect of incorporation of the blowing parameter on pressure drop predictions is shown in Fig. 6-7. In the presented case the effect of considering the blowing parameter may reach even 20% effect.

Heat transfer in phase change

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

$$\frac{\alpha_{TP}}{\alpha_l} = \sqrt{\left(\Phi^2\right)^n + \frac{C}{1+P} \left(\frac{\alpha_{TP}}{\alpha_l} \right)^2} \qquad (37)$$

In case of condensation the constant C=0, whereas in case of flow boiling C=1. In Eq. (37) B=q_w/(G h_{lv}) and the correction, P, is: $P = 2.53 \cdot 10^{-3} \cdot \operatorname{Re}_l^{1.17} \cdot Bo^{0.6} \cdot \left(\Phi^2 - 1\right)^{-0.65}$.

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

$$\Phi^2 = \left[1 = 2 \left(\frac{1}{f_l} - 1 \right) Con^m \right] (1-x)^{\frac{1}{3}} + x^3 \frac{1}{f_{lz}} \qquad (38)$$

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 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_1 must be used. In (38) $f_1=(\rho_L/\rho_G)\ (\mu_L/\mu_G)^{0.25}$ 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 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 $f_{1z}=\alpha_{GO}/\alpha_{LO}$, 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 pool boiling heat transfer coefficient α_{PB} is calculated from a relation due to Cooper.

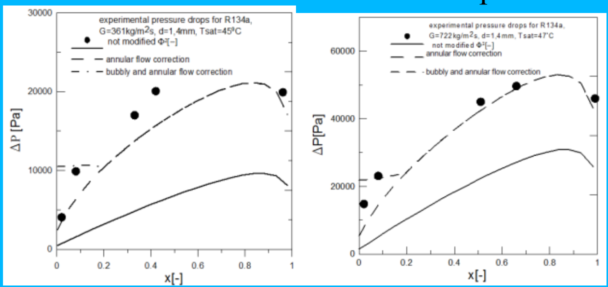

Fig 6 and 7. Condensation pressure drop distribution in function of quality, Bohdal et al. [4].

The correctness of the calculations was compared due to experimental data and the own correlation (37). A few examples of comparisons are presented in Fig. 8-11 for Δp_{TPB} in flow boiling of R134a and R1234yf.

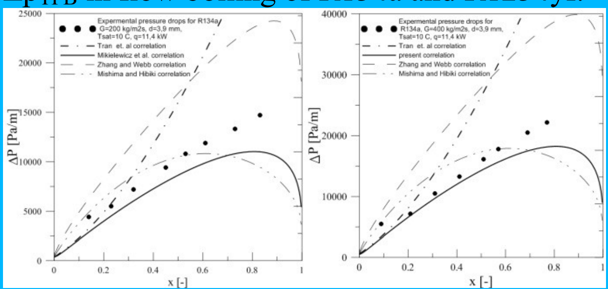

Fig. 8 and 9. Pressure drop in function of quality for R134a, Lu et al. [7].

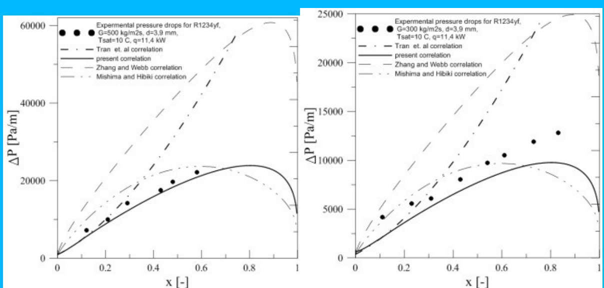

Fig. 10 and 11. Pressure drop in function of quality, boiling R1234yf, Lu et al. [7].

As we can see the authors own correlation shows best compatibility with the experimental data. The good agreement with experimental data is obtained with Mishima

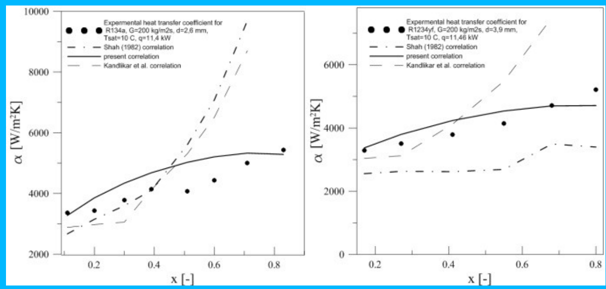

Fig. 12 and 13. Heat transfer coeff. for R134a, Copetti et al. [14], and R1234yf, Lu et al. [7].

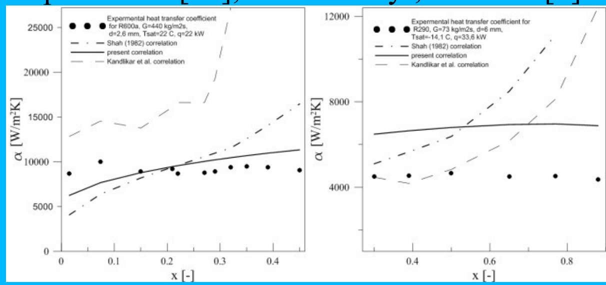

Fig. 14 and 15. Heat transfer coeff. for R600a, Copetti et al. [14], and R290, Wang et al. [8].

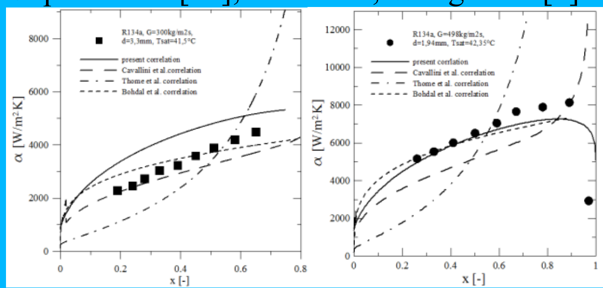

Fig. 16 and 17. Heat transfer coeff. for R134a, Bohdal [4], d=3.3 mm, and d=1.94mm.

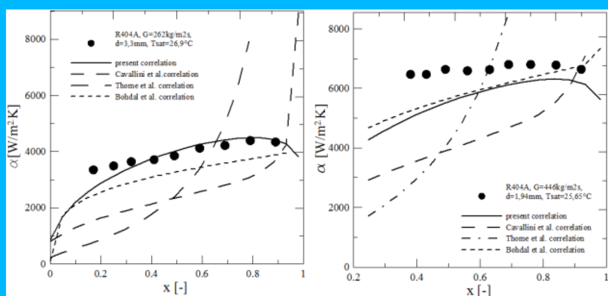

Fig. 18 and 19. Heat transfer coeff. for R404A, Bohdal [4], d=3.3m and d=1.94mm.

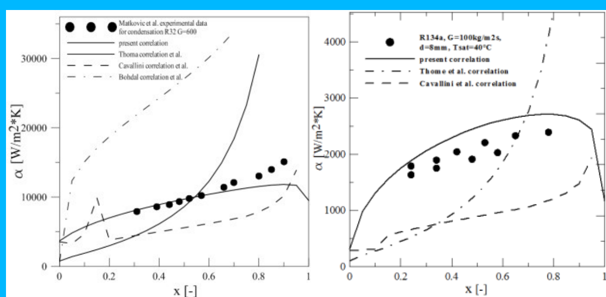

Fig. 20 and 21. Heat transfer coeff. for R32, Matkovic et al. [6], d=0.96mm,R134a d=8mm.

and Hibiki at al. [10] correlation and relatively good correctness shows Tran et al model. In Fig. 12-21 presented are comparisons from the point of view of heat transfer coefficient.

CONCLUSIONS

In the paper presented is a model to incorporate the non-adiabatic effects in predictions of pressure drop and heat transfer. The model is general as it enables to be included into any two-phase flow multiplier definition. In the present work such model has been incorporated into authors own 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 halogeneous refrigerants and other fluids.

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References

[1] Mikielewicz J: Semi-empirical method of determining the heat transfer coefficient for subcooled saturated boiling in a channel, Int. J. Heat Transf. 1973, 1129-1134.
[2] Mikielewicz, D., Mikielewicz, J., Tesmar, J., Improved Semi-Empirical Method for Determination of Heat Transfer Coefficient in Flow Boiling in Conventional and Small Diameter Tubes, Int. J. of Heat and Mass Transfer, 50, 3949–3956, 2007
[3] Mikielewicz D., Mikielewicz J., A common method for calculation of flow boiling and flow condensation heat transfer coefficients in minichannels with account of non-adiabatic effects, Heat Tr. Engng, 32, 1173-1181, 2011.
[4] Bohdal, T., Charun, H., Sikora, M. (2011). Comparative investigations of the condensation of R134a and R404A refrigerants in pipe minichannels, Int. J. of Heat and Mass Transfer, 54, 1963–1974.
[5] Cavallini, A., Censi G., Del Col D., Doretti L., Longo G.A., Rossetto L., Condensation of Halogenated Refrigerants inside Smooth Tubes, HVAC and Res., 8, 429-451, 2002.
[6] Matkovic M., Cavallini A., Del Col D., Rossetto L., Experimental study on condensation heat transfer inside a single circular minichannel, Int. J. of Heat and Mass Transfer, 52, 2311-2323, 2009
[7] Lu M-C, Tong J-R, Wang C-C, Investigation of the two-phase convective boiling of HFO-1234yf in a 3.9 mm diameter tube, Int. J. of Heat and Mass Transfer, 65, (2013) 545–551.
[8] Wang S., Gong M.Q., Chen G.F., Sun Z.H., Wu J.F., Two-phase heat transfer and pressure drop of propane during saturated flow boiling inside a horizontal tube, Int. Journal of Refrigeration (2013),
[9] Thome, J.R., El Hajal, J., Cavallini, A., Condensation in Horizontal Tubes, Part 2: New Heat Transfer Model Based on Flow Regimes, Int. J. of Heat and Mass Transfer, 46, 3365–3387, 2003.
[10]Sun, L., Mishima, K., Evaluation Analysis of Prediction Methods for Two-Phase Flow Pressure Drop in Mini-Chann., Int. J. of Multiphase Flows, 35, 47–54, 2009.
[11]Zhang M., Webb R.L., Correlation of Two-Phase Friction for Refrigerants in Small-Diameter Tubes, Experimental Thermal and Fluid Science, vol. 25, nos. 3–4, pp. 131–139, 2001.
[12]Müller-Steinhagen, R., and Heck, K., A Simple Friction Pressure Drop Correlation for Two-Phase Flow in Pipes, Chemical Enging Progress, 20, 297–308,1986.
[13]Tran, T. N., Chyu, M.-C., Wambsganss, M. W., and France, D. M., Two-phase pressure drop of refrigerants during flow boiling in small channels: an experimental investigation and correlation development, Int. J. Multiphase Flow, **26(11)**, 1739–1754, 2000.
[14]Copetti J.B., Macagnan M.H., Zinani F., Kunsler N.L.F., Flow boiling heat transfer and pressure drop of R134a in a mini tube: an experimental investigation, Exp. Thermal and Fluid Sc, 35, (2011) 636–644.