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Analytical model of non-adiabatic effects for pressure drop and heat transfer of flow boiling and flow condensation in conventional tubes and 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 flow condensation with account of non-adiabatic effects for some recent data collected from literature. The first effect, the modification of interface shear stresses in annular flow pattern is considered through incorporation of the so called "blowing parameter". The mechanism of modification of shear stresses at the vapor-liquid interface for such case has been presented in detail in the paper. 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 another influence of heat flux, where it is influencing the bubble nucleation in the case of the bubbly flow pattern. As a result a modified general form of the two-phase flow multiplier, applicable both to flow boiling and flow condensation, is obtained, in which the non-adiabatic effects are clearly pronounced. The model of two-phase multiplier is additionally used in predictions of heat transfer coefficient.

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

INTRODUCTION

It is very tempting to postulate that the non-adiabatic effects modify the frictional pressure drop term in two-phase flow and subsequently the heat transfer coefficient. That is the reason why it is impossible to use reciprocally the existing adiabatic models for calculations of heat transfer and pressure drop in the cases of flow boiling and flow condensation. Heat transfer coefficients in flow boiling are different however from their counterpart in the condensation inside tubes. Considerations presented in the paper relate both to the case of flow boiling and flow condensation in conventional channels as well as small diameter ones. Authors devoted all the possible attention that the modeling presented is applicable to the whole range of quality variation in cases of condensation and boiling. The form of the two-phase flow multiplier, which is a major factor in the modeling presented here, should be capable of capturing both the mentioned above cases, however in case of some fluids more studies will be required to devise a more accurate, structure dependent, version of the two-phase flow multiplier. 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 [1]. In such case the heat transfer coefficient is primarily dependent on the convective mechanism.

In authors approach to solve the issue of non-adiabatic effects in flow boiling and flow condensation is to incorporate appropriate mechanisms into the liquid-vapour interface shear stress modelling. That can be done through consideration of the two-phase flow multiplier term, present in the model of flow boiling, developed firstly by Mikielewicz [2] with subsequent modifications by Mikielewicz et al. [3] and Mikielewicz and Mikielewicz [4]. In the latter paper presented have been responsible factors for modification of shear stresses at the vapor-liquid interface, different for annular flow structure and for other structures, generally considered in case of boiling flows in minichannels as bubbly flows. Postulated in these papers 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 pattern. In other flow structures, for example the bubbly flow, there can also be identified the effects related to the applied heat flux. One of such effects is the fact that the available in literature two-phase pressure models in literature is modeled in the way that the influence of applied heat flux is not considered.

The objective of this paper is to present the model of annular two-phase flow, allowing to introduce the blowing parameter in the analytical manner, confirming in such a way the capability of the type of modeling presented in authors earlier approaches to model the flow boiling and flow condensation inside tubes with account of non-adiabatic effects. Therefore some experimental data have been collected from literature to further validate that method for the case of other than considered earlier fluids. Additionally, authors compared the developed model for two-phase pressure drop calculations in minichannels with some relevant correlations from literature, namely due to Mishima and Hibiki [5], Zhang and Webb [6], Tran et al. [8] and a modified version of Muller-Steinhagen and Heck [7] model, [4]. Calculations have also been compared for the heat transfer data in flow boiling and flow condensation. The literature data considered in the paper for relevant comparisons of flow condensation are due to Bohdal et al. [9], Cavallini et al. [10], Matkovic et al. [11], and due to Lu et al. [12], Wang et al. [13] and Copetti et al. [14] 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. [15] and Thome et al. [16] as well as Shah [17] and Kandlikar et al. [18].

TWO-PHASE FRICTION PRESSURE DROP MODEL BASED ON THE CONCEPT OF ENERGY DISSIPATION

Flow resistance in two-phase flow friction is greater than that in the 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. Some of them will be presented in the later part of the text.

Dissipation energy 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 energy in two-phase flow can be modeled as a sum of two contributions, namely the energy dissipation due to the shearing flow without bubbles, E_{TP}, and the energy dissipation resulting solely from bubble generation, E_{PB}, [2-4]:

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

Dissipation energy is expressed as power lost in the control volume. It refers to two-phase flow friction losses. Analogically can be expressed the energy dissipation due to bubble generation in the two-phase flow. A geometrical relation between the friction factors in two-phase flow is analitically 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 applied wall heat flux, as the phenomenon it describes is the bubble generation (in case of flow boiling). That term will be modified in the remainder of the text to include the heat flux dependence in bubbly flow. The first term on the right hand side of (3) can be determined from the definition of the two-phase flow multiplier for the adiabatic two-phase flow, described in general by equation (1), i.e. $\Delta p_{TP} = \Phi^2 \, \Delta p_{LO}$ 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 average velocity w_{TP}. The pressure drop of liquid or vapour flowing alone can be determined from a corresponding single phase flow relations. In case of turbulent flow we can use the Blasius formula for determination of the appropriate 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 discernible difference of the method (3) in comparison to other authors models is the fact that the relation between the friction factors is of geometrical character, the approach which has no counterpart in other approaches to two-phase flow 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.

Non-adiabatic effects in annular flow

In the following sections presented is the idea of shear stress modification at the liquid-vapour interface in the annular flow due to the applied heat flux.

Modification of interface shear stress using the blowing parameter

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 (the term borrowed from the theory of blowing or suction into the boundary layer), which contributes to the liquid film thickening in case of flow condensation and thinning in case of flow boiling. That idea has been derived from the boundary layer modification in case of injecting or suction of fluid into the boundary layer, Mikielewicz [19], where a relation describing the blowing into the boundary layer has been obtained. Mikielewicz derived a general formula for modification of shear stresses in the boundary layer which reads:

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

In (4) $\tau^+=\tau/\tau_w$, is the ratio of shear stress in the flow with bubbles injection into the boundary layer referred to the wall shear stress at the presence of injection, whereas $\tau_0^+=\tau_w/\tau_{w0}$, where τ_{w0} is the wall shear stress in case where the bubble injections effects are not considered, and $B=2\vartheta_0/(c_f\, u_\infty)$ is the so called "blowing parameter". In (4) ϑ_0 denotes the transverse velocity, c_f is the friction factor and u_∞ is the velocity of undisturbed flow in the boundary layer. The idea of modeling the boundary layer disturbance by means of the bubble injection was adopted to the case of flow condensation or flow boiling where migration of vapour due to applied heat flux is present. In such case the transverse velocity can be expressed as $q_w/(h_{lv}\rho_l)$. The formulae (4) was developed for the case of blowing, so for the case of suction a negative sign instead of positive one should be used in equation (4).

In case when Re→∞ the equation (4) tends to that suggested by Kutateladze and Leontiev [20]:

$$\tau = \left(1 - \frac{B}{4}\right)^2 \tag{5}$$

On the other hand, in case of small values of B the relation given by equation (15) reduces to that recommended by Wallis [21]:

$$\tau = \left(1 - \frac{B}{2}\right) \tag{6}$$

The analyses due to Wallis as well as Kutateladze and Leontiev were carried out for the case of flow boiling. That confirms the appropiateness of assumptions.

The blowing parameter, applicable now to the case of flow boiling or flow condensation, is hence defined as:

$$B = \frac{2\vartheta_0}{c_f u_\infty} = \frac{2q}{c_{f0}(u_G - u_L)h_{lv}\rho_G} = \frac{2q\dfrac{\rho_L}{\rho_G}}{c_{f0}G(s-1)h_{lv}} \tag{7}$$

In (7) s denotes the slip velocity and G - mass velocity. For the sake of incorporation of the blowing/suction parameter the following form of expression will be used, in which the plus sign stands for the case of condensation and minus for boiling respectively:

$$\tau = (1 \pm B) \tag{8}$$

In the next section a new approach to determination of the blowing parameter incorporating deposition and entrainment of droplets in function of vapor quality is presented.

Annular flow model with incorporation of blowing parameter

Analysis of the liquid and vapor phase in two-phase flow is based on examination of mass and momentum balance equations with respect to the applied heat flux. Fig. 1 shows the considered schematic of the annular flow model. Presented below analysis will be conducted with the reference to the condensation in the flow.

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:

$$\dot{m} = \dot{m}_f + \dot{m}_{cd} + \dot{m}_{cv} \tag{9}$$

In the model presented below the following notation is used. The liquid film cross-section area is expressed by the product of film perimeter and the film thickness $A_f=\pi d\delta_f$, while the core cross-section area as the area occupied by vapour in the channel, i.e. $A_c=\pi(d-2\delta_f)^2/4$. The wetted perimeter by the liquid film is given by the relation $P_f=\pi d$, where d is the channel inner diameter. The mean liquid film velocity results from the definition of mass flow rate in the film, i.e. $u_f=\dot{m}_f/(\rho_f A_f)$. Authors assumed that the interfacial velocity can be determined similarly as in the laminar flow from a known relationship $u_i=2u_f$.

Mass balance in liquid film and core

-Liquid film**:**

$$\frac{dm_f}{dz} = \Gamma_{lv} + D - E \tag{10}$$

-Two-phase flow vapor core**:**

$$\frac{dm_{cd}}{dz} = -D + E \tag{11}$$

-Vapor in vapor core:

$$\frac{dm_{cv}}{dz} = -\Gamma_{lv} \tag{12}$$

In (10) and (11) the terms D and E denote deposition and entrainment in the annular flow. The remaining term in equation, namely the mass transfer term $\Gamma_{lv}=q_w P/h_{lv}$, is responsible for the vapour condensation on the interface, ($P=\pi(d-2\delta_f)$). 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:

$$C = \frac{\dot{m}_{cf}}{\dot{m}_{cv}\vartheta_g + \dot{m}_{ef}\vartheta_f} \tag{13}$$

The combined mass flow rate of the core results from combination of (11) and (12):

$$\frac{dm_c}{dz} = -\Gamma_{lv} - D + E \tag{14}$$

The amount of entrained droplets $\dot{m}_{ef}$ in (13) can be determined from the mass balance:

$$\dot{m}_{ef} = \dot{m} - \dot{m}_f - \dot{m}_{cv} \tag{15}$$

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/condensation, droplet deposition or entrainment). Acceleration of the flow is neglected. The flow schematic is shown in Figure 2.

-Liquid film

Momentum equation for the liquid film reads:

$$-\frac{dp_L}{dz}\Delta z \cdot (\delta - y) \cdot P_f - \tau P_f \Delta z + \tau_i P_f \Delta z = (\Gamma_{lv} u_i + D u_c - E u_i) \cdot \Delta z \tag{16}$$

Re-arranging (16) leads to obtaining the shear stresses in the liquid film expressed by:

$$\tau = -\frac{dp_L}{dz}(\delta - y) + \tau_i - \frac{1}{P_f}(\Gamma_{lv}u_i + Du_c - Eu_i) \tag{17}$$

With the view to obtain the film velocity distribution we express the shear stress in the liquid in the form:

$$\tau = \mu_l \frac{du_f}{dy} \tag{18}$$

Substituting equation (18) into (17) and performing integration with respect to radial coordinate, we obtain the velocity profile in the liquid film:

$$u_f = \frac{1}{\mu_l}\left(\delta y - \frac{1}{2}y^2\right)\left(-\frac{dp_l}{dz}\right) + \frac{y}{\mu_l}\tau_i - \frac{y}{\mu_l P_f}(\Gamma_{lv}u_i + Du_c - Eu_i) \tag{19}$$

The mass flow rate of the liquid film is defined as:

$$\dot{m}_f = \rho_l P_f \int_0^\delta u_f dy \tag{20}$$

Substituting (19) into (20) and integrating allows to obtain the mass flow rate of liquid film:

$$\dot{m}_f = \frac{P_f \rho_f \delta^3}{3\mu_l}\left(-\frac{dp_l}{dz}\right) + \frac{3P_f \rho_f \delta^3}{6\mu_l \delta}\tau_i - \frac{3\rho_f P_f \delta^3}{6\mu\,\delta P_f}(\Gamma_{lv}u_i + Du_f - Eu_i) \tag{21}$$

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

$$-\left(\frac{dp_l}{dz}\right) = \frac{3\mu_f \dot{m}_f}{P_f \rho_f \delta^3} - \frac{3\tau_i}{2\delta} + \frac{3(\Gamma_{lv}u_i + Du_f - Eu_i)}{2\delta P_f} \tag{22}$$

- 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:

$$\left[\rho_{TP}u_c^2A_c+\frac{d}{dz}\left(\rho_{TP}u_c^2A_c\right)\Delta z-\rho_{TP}u_c^2A_c\right]+\left[-\Gamma_{lv}u_i-Du_c+Eu_i\right]\Delta z$$
$$=p_vA_c-\left[p_vA_c+\frac{d\left(p_vA_c\right)}{dz}\Delta z\right]-\tau_iP\Delta z \tag{23}$$

From equation (23) it follows that interfacial shear stress are:

$$\tau_i=\frac{1}{P}\left[A_c\left(-\frac{dp_v}{dz}\right)-p_v\frac{dA_c}{dz}\right]-\frac{1}{P}\frac{d}{dz}\left(\rho_{TP}u_c^2A_c\right)-\frac{1}{P}\left(-\Gamma_{lv}u_i-Du_c+Eu_i\right) \tag{24}$$

In (24) it is assumed that the perimeter of the vapor core $P\approx\pi(d-2\delta)$. In subsequent calculations it will be assumed that the longitudinal change of the core flow is small, and hence can be neglected ($dA_c/dz\approx0$).

The interfacial shear stress is usually defined as:

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

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

$$\tau_i=f_i\left[\frac{1}{2}\rho_{TP}\left(u_c-u_i\right)^2\right]-\frac{\Gamma_{lv}}{2P}\left(u_c-u_i\right) \tag{26}$$

The sought unknowns in our issue are: liquid film mass flow $\dot{m}_f$, liquid film thickness δ and the interfacial shear stress τ_i. In order to find the value of the latter term the value of Reynolds number for the core is required:

$$\mathrm{Re}_c=\frac{\rho_{TP}\left(u_c-u_i\right)d_c}{\mu_g} \tag{27}$$

The interface friction coefficient can be taken from:

$$f_i=0.005\left(1+300\frac{\delta}{d}\right) \tag{28}$$

In order to use equation (28) the knowledge of liquid film thickness is necessary. For the sake of simplicity in the present analysis it has been determined according to Thome et al. [9] as:

$$\delta = \sqrt{\frac{f_w}{f_i}\frac{\rho_l}{\rho_g}\frac{J_l}{J_g}}d \tag{29}$$

In the minichannel the dominating flow structure is the annular flow. Let us now focus on the effect of phase change impact on modification of the interface shear stress τ_i. Shear stress resulting from the momentum equation (24) yields:

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

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

$$p_v - p_l = \frac{\sigma}{r} \tag{31}$$

After differentiation, (31) takes the form:

$$\frac{dp_v}{dz} - \frac{dp_l}{dz} = \frac{\sigma}{r^2}\frac{dr}{dz} \tag{32}$$

As a first approximation, however, we ignore the effect of the surface tension of the liquid, so the pressure gradient in liquid will be the same as in vapour.

With the view to determine the pressure drop in equation (30) we can re-arrange it to the form:

$$-\left(\frac{dp_v}{dz}\right) = \frac{P}{A_c}\tau_i + \frac{1}{A_c}\left[-\Gamma_{lv}u_i - Du_c + Eu_i\right] \tag{33}$$

Comparing (33) and (22), which are the expressions for pressure drop in vapour and liquid respectively returns a relationship for the interfacial shear stress:

$$\tau_i = \frac{\dfrac{3\mu_f \dot{m}_f}{P_f \rho_f \delta^3} + \left(\dfrac{3}{2\delta P_f} + \dfrac{1}{A_c}\right)\left(\Gamma_{lv} u_i + Du_f - Eu_i\right)}{\dfrac{P}{A_c} + \dfrac{3}{2\delta}} \tag{34}$$

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/condensation, droplet deposition and entrainment. When there is no evaporation of the liquid film, but the other two are, the interfacial shear stress distribution is:

$$\tau_{io} = \frac{\dfrac{3\mu_f \dot{m}_f}{P_f \rho_f \delta^3} + \left(\dfrac{3}{2\delta P_f} + \dfrac{1}{A_c}\right)\left(Du_f - Eu_i\right)}{\dfrac{P_f}{A_c} + \dfrac{3}{2\delta}} \tag{35}$$

In order to investigate the effect of the mass transfer between liquid film and the core we form a ratio of equation (34) to equation (35) to obtain:

$$\frac{\tau_i}{\tau_{io}} = 1 + \frac{\left(\dfrac{3}{2\delta P_f} + \dfrac{1}{A_c}\right)\left(\Gamma_{lv} u_i\right)}{\dfrac{3\mu_f \dot{m}_f}{P_f \rho_f \delta^3} + \left(\dfrac{3}{2\delta P_f} + \dfrac{1}{A_c}\right)\left(Du_f - Eu_i\right)} = (1 + B') \tag{36}$$

In equation (36) we obtained a general expression for the blowing parameter in the flow condensation inclusive of mass transfer between film and core, as well as deposition and entrainment. For the sake of present illustration the case of equation (36) has been developed where we neglect the entrainment and deposition i.e. by assigning E = 0 and D = 0. In such case we obtain a very simplified form of the diabatic two-phase flow effect in the form:

$$\frac{\tau_i}{\tau_{io}} = 1 + \frac{2 q_w \delta\left(\dfrac{4\delta}{d} + \dfrac{3}{2}\right)}{3\mu_f h_{lv}} = (1 + B) \tag{37}$$

Non-adiabatic effects in other than annular two-phase flows

In case of the non-adiabatic effects in other than annular flow structures author presented his idea in [4]. Therefore only the final form of the modified two-phase flow multiplier is presented here. The two-phase flow multiplier, applicable to bubbly flow, reads:

$$\Phi^2_{TPB} = \frac{\xi_{TPB}}{\xi_0} = \sqrt{\Phi^2 + \frac{\xi^2_{PB}}{\xi^2_0}} = \Phi^2\sqrt{1 + \frac{\left(\dfrac{8\alpha_{PB} d}{\lambda \operatorname{Re}\Pr}\right)^2}{\xi^2_0 \Phi^2}} \tag{38}$$

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:

$$\Phi^2{}_{TPC} = \Phi^2{}_{TPB} = \frac{\xi_{TPB}}{\xi_0} = \begin{cases} \Phi^2\left(1 \pm \dfrac{B}{2}\right) & \textit{for annular structure, condensation} \\ & \textit{and boiling} \\ \Phi^2\sqrt{1 + \left(\dfrac{8\alpha_{PB} d}{\lambda \operatorname{Re}\Pr \xi_0 \Phi^2}\right)^2} & \textit{for other flow structures} \end{cases} \tag{39}$$

In (39) 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 specifications.

General relation of heat transfer during the phase change

The heat transfer model applicable both to the case of flow boiling and flow condensation can be written as:

$$\frac{\alpha_{TP}}{\alpha_{LO}} = \sqrt{\left(\Phi^2\right)^n + \frac{C}{1+P}\left(\frac{\alpha_{TP}}{\alpha_{LO}}\right)^2} \tag{40}$$

In case of condensation the constant C=0, whereas in case of flow boiling C=1. In Eq. (40) the correction, P has been devised based in the experimental data, and reads: $P = 2.53 \cdot 10^{-3} \cdot \mathrm{Re}_l^{1.17} \cdot Bo^{0.6} \cdot \left(\Phi^2 - 1\right)^{-0.65}$. The Boiling number Bo is: Bo=$q_w/(G\, h_{lv})$.

In the form applicable to conventional and small-diameter channels, the modified Muller-Steinhagen and Heck adiabatic multiplier model is advised, Mikielewicz et al. [3]:

$$\Phi^2 = \left[1 = 2\left(\frac{1}{f_l} - 1\right)Con^m\right]\left(1-x\right)^{\frac{1}{3}} + x^3 \frac{1}{f_{lz}} \tag{41}$$

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 (39) $f_1=(\rho_L/\rho_G)\,(\mu_L/\mu_G)^{0.25}$ 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 advised to be calculated from a relation due to Cooper [4].

RESULTS OF CALCULATIONS

Calculations have been accomplished to show the performance of the model (39) and (40) in comparison to some experimental data from literature for flow boiling and flow condensation. Before these comparisons the comparisons have been made between the value of the blowing parameter defined by equation (7) and the model (35). Fig. 4 presents sample calculations of the blowing parameter for condensation for HFE7100 at parameters: G = 483kg/m^{2}s, T_{sat} = 74 °C, whereas Fig. 5 for R134a: G=300 kg/m^{2}s, T_{sat} =10°C in a 1mm tube. When the parameter is calculated by equation (7) then B=0.137 for HFE7100 and B=0.014 for R134a respectively. The result from calculations using equation (37) is B=0.127 and B=0.016, respectively. That has been obtained as a result of integration of the distribution of the parameter B from the beginning of annular flow to the end of it, i.e. from x=0.1 to x=1.0. Satisfactory consistency of calculations is observed.

The examples of the effect of incorporation of the modification (7) on pressure drop predictions in flow condensation is shown in Fig. 6-7 for the experimental data due to Bohdal et al. [9]. In the presented case the effect of considering the modification may reach even 20% effect exhibiting a good consistency with the results. Additional few examples of comparisons are presented in Fig. 8-11 for Δp_{TPB} predictions in flow boiling of R134a and R1234yf for the data due to Lu et al. [12]. Also a good consistency with experimental data is seen. In the case of comparisons with the experimental data due to Lu et al. apart from the model (39) also the good agreement with experimental data is obtained with Mishima and Hibiki at al. [10] correlation and relatively good correctness shows Tran et al model. In Fig. 12-15 presented are comparisons from the point of view of heat transfer coefficient for the flow boiling data due to Copetti et al. [14], Wang et al. [13] and Lu et al. [13]. A satisfactory consistency is clearly seen. In these calculations the non-adiabatic effect was modeled through incorporation of (7) to the model (38). In Fig. 16-21 presented are some comparisons for the case of flow condensation for the experimental data due to Bohdal et al. [9], Matkowic et al. [11] for three different fluids, namely R404A, R32 and R134a respectively. Again a satisfactory agreement is seen.

CONCLUSIONS

In the paper presented is a model to incorporate the non-adiabatic effects in predictions of the pressure drop and heat transfer in flow boiling and flow condensation. The model is based on the modification of the interface shear stress through incorporation of the so called blowing parameter. A general expression for the blowing parameter in the flow condensation inclusive of mass transfer between film and core, as well as deposition and entrainment has been derived from the annular flow model. The results of calculations between the description of blowing parameter given by equations (7) and (37) returned consistent results. In effect the blowing parameter can be included into the definition of the two-phase flow multiplier. In the present work such model has been incorporated into authors own model of pressure drop and subsequently heat transfer. 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 proposed model 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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NOMENCLATURE

A_f – cross section area wetted by liquid film [m^2]

A_c – cross section area of the core [m^2]

B – blowing parameter [-]

Bo – boiling number [-]

C – parameter in Lockhart-Martinelli correlation [-]

c_f – friction factor, [-]

Con – Constraint number [-]

D – droplet deposition [-]

d – diameter [m]

E – droplet entrainment [-]

E_{TP} – energy dissipation without the bubbles [W/m^3]

E_{PB} – energy dissipation from the bubble generation [W/m^3]

f_i – the interface friction coefficient [-]

G – mass flux, [kg/m^{2}s]

g – acceleration due to gravity [m/s^2]

h_{lv} – specific enthalpy of vaporization [kJ/kg]

l– channel length [m]

$\dot{m}$ – total mass flow rate [kg/s]

$\dot{m}_f$ – liquid film mass flow rate [kg/s]

$\dot{m}_{cd}$ – droplets in the core of flow mass flow rate [kg/s]

$\dot{m}_{cv}$ – vapor mass flow rate [kg/s]

M – molecular weight, [kg/mol]

P – wetted perimeter [m]

p_v – vapor pressure [Pa]

p_l – liquid pressure [Pa]

Δp_{LO} – pressure drop for liquid phase only [Pa]

Δp_{TP} – pressure drop in two-phase flow [Pa]

Δp_0 – total pressure drop in single phase flow [Pa]

p_{kr} – critical pressure [Pa]

Re – Reynolds number, [-]

s – slip ratio [-]

u_g – liquid phase velocity [m/s]

u_v – vapor phase velocity [m/s]

u^* – friction velocity, m/s,

w_{TP} – two-phase velocity, m/s

w_{LO} – velocity liquid film only, m/s

x – quality [-]

Greek symbols

δ - liquid film thickness [m]

σ – surface tension [N/m]

ρ –density [kg/m^3]

ξ – friction factor [-]

Φ^2 – two-phase flow multiplier

Γ_{lv} - evaporation/condensation of vapor

τ^* – dimensionless shear stress [-]

ϑ_0 – transverse velocity [m/s]

τ_{w0} –wall shear stress in case where non-adiabatic effects are not considered

q_w – wall heat flux [kJ/kg]

α –heat transfer coefficient [W/m^{2}K]

λ – thermal conductivity [W/mK]

Superscripts

+ – non-dimensional

cb – convective boiling

f – forced flow

G - gas

h – hydraulic

kr – critical

L – liquid

LO – liquid only

PB – pool boiling

sat – saturation

TP – two-phase flow

TPB – two-phase boiling

TPK – two-phase condensation

v – saturated vapour

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Figure captions

Fig. 1 Annular flow structure model.

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

Fig. 3 Control volume for the flow core

Fig. 4. Blowing/suction parameter as a function vapor quality for HFE7100, G=483kg/m^2s, T_{sat} = 74°C, d=1mm

Fig. 5. Blowing/suction parameter as a function vapor quality for R134a, G=300 kg/m^2s, T_{sat} =10°C, d=1mm

Fig. 6. Condensation pressure drop distribution in function of quality, Bohdal et al. [9].

Fig. 7. Condensation pressure drop distribution in function of quality, Bohdal et al. [9].

Fig. 8. Flow boiling pressure drop in function of quality for R134a, Lu et al. [12].

Fig. 9. Flow boiling pressure drop in function of quality for R134a, Lu et al. [12].

Fig. 10. Flow boiling pressure drop in function of quality, boiling R1234yf, Lu et al. [12].

Fig. 11. Flow boiling pressure drop in function of quality, boiling R1234yf, Lu et al.

Fig. 12. Flow boiling heat transfer coefficient for R134a, Copetti et al. [14].

Fig. 13. Flow boiling heat transfer coefficient for R1234yf, Lu et al. [12].

Fig. 14. Flow boiling heat transfer coefficient for R600a, Copetti et al. [14]

Fig. 15. Flow boiling heat transfer coefficient for R290, Wang et al. [13].

Fig. 16. Flow condensation heat transfer coefficient for R134a, Bohdal [9], d=3.3 mm.

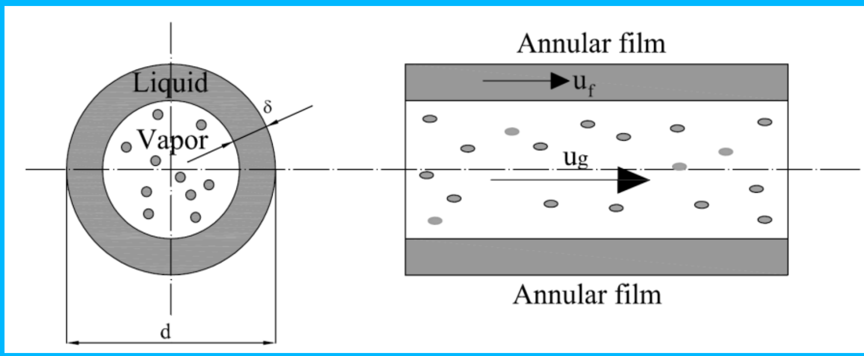
Fig. 17. Flow condensation heat transfer coefficient for R134a, Bohdal [9], d=1.94mm.

Fig. 18. Flow condensation heat transfer coefficient for R404A, Bohdal [9], d=3.3m.

Fig. 19. Flow condensation heat transfer coefficient for R404A, Bohdal [9], d=1.94mm.

Fig. 20. Flow condensation heat transfer coefficient for R32, Matkovic et al. [11], d=0.96mm.

Fig. 21. Flow condensation heat transfer coefficient for R134a, Matkovic et al. [11], d=8mm.



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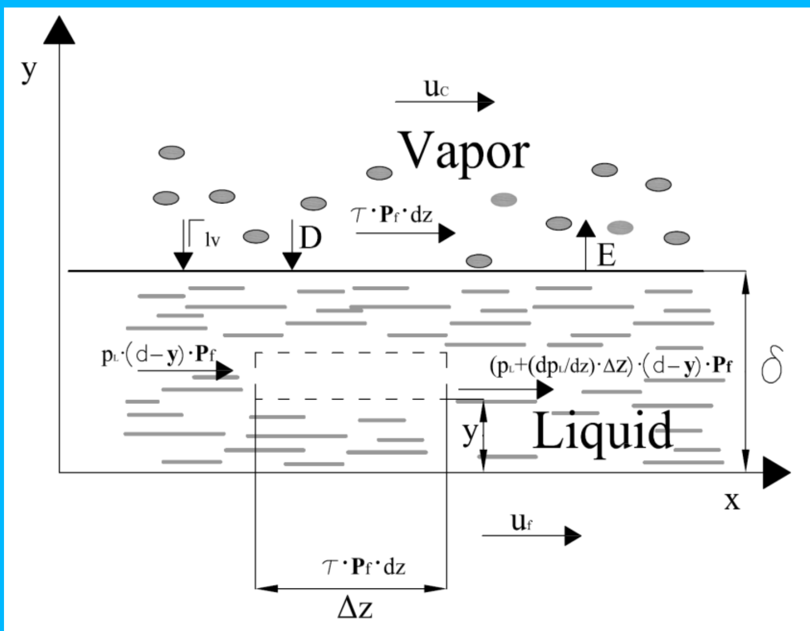

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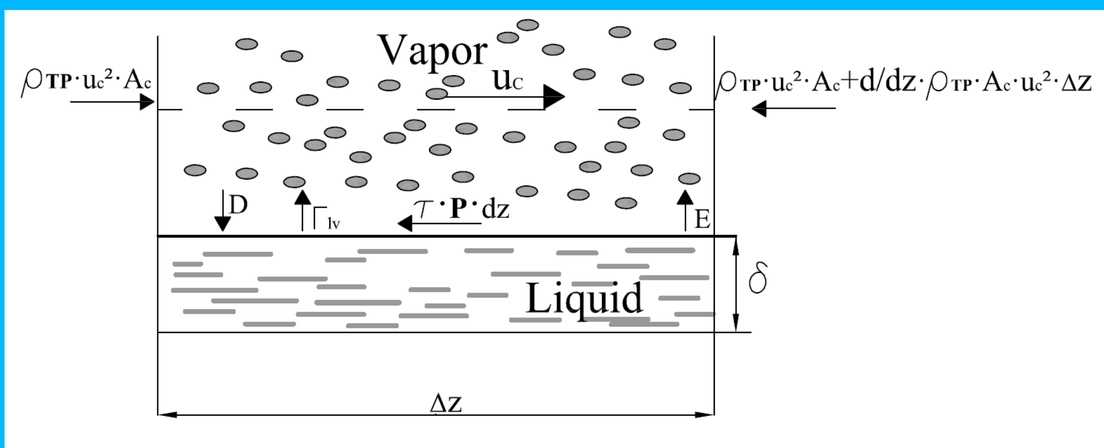

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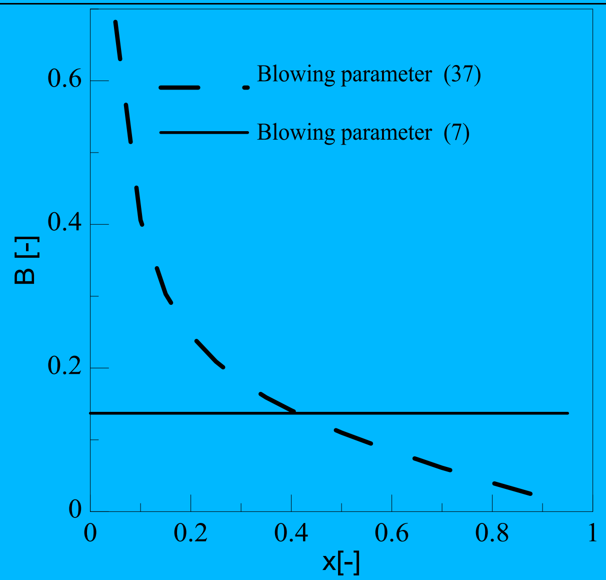

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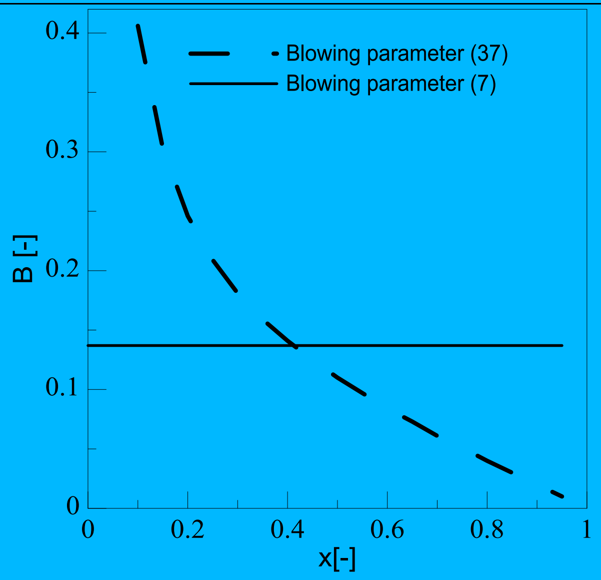

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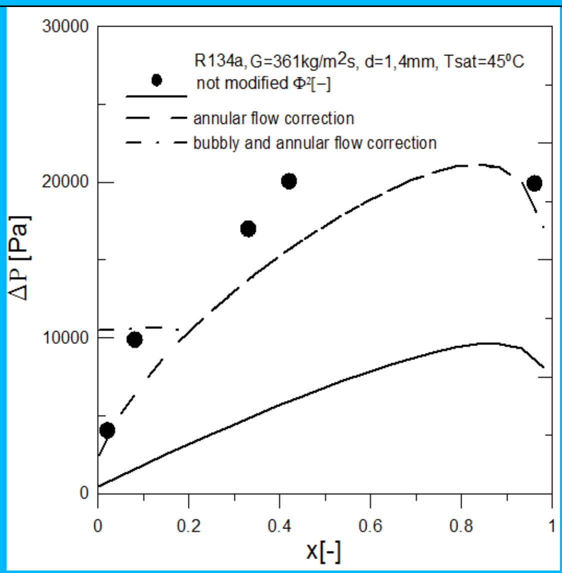

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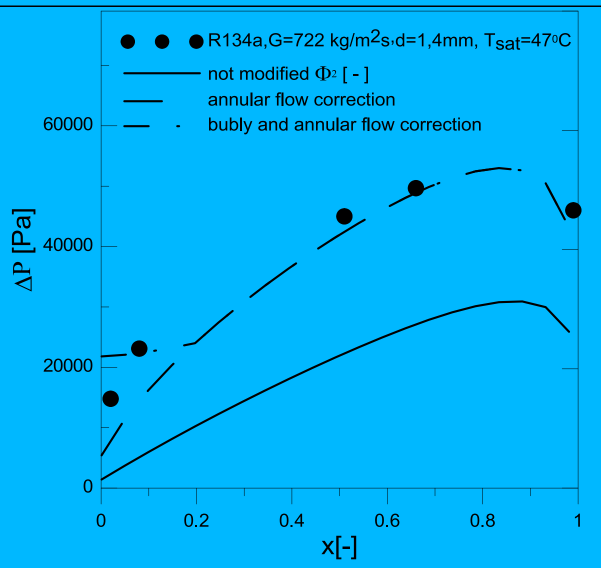

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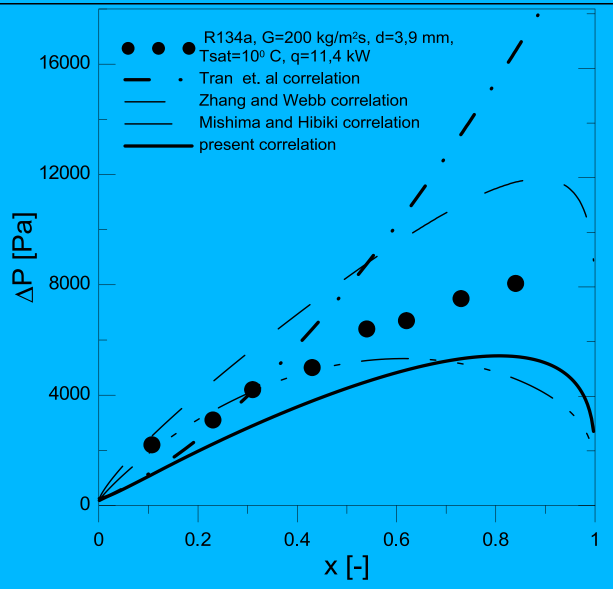


Fig. 8. Flow boiling pressure drop in function of quality for R134a, Lu et al. [12].

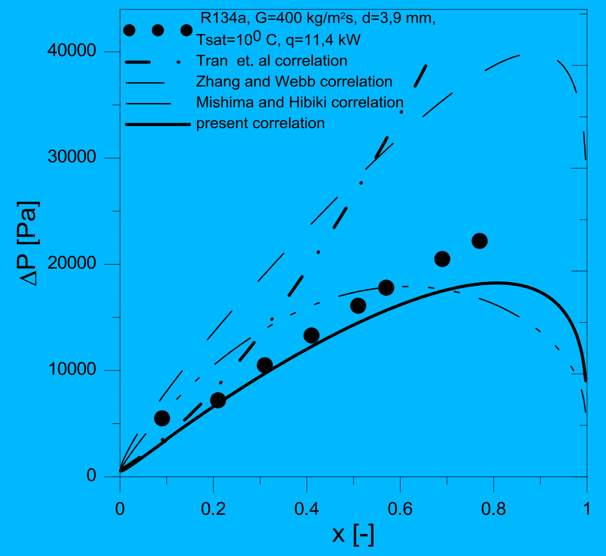


Fig. 9. Flow boiling pressure drop in function of quality for R134a, Lu et al. [12].

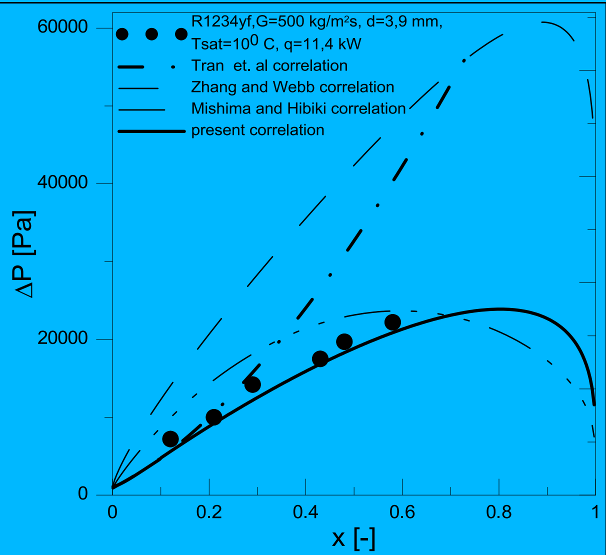
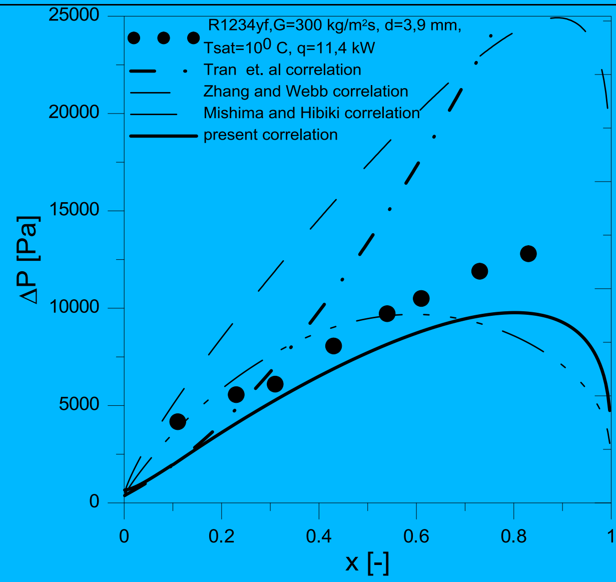


Fig. 10. Flow boiling pressure drop in function of quality, boiling R1234yf, Lu et al. [12].



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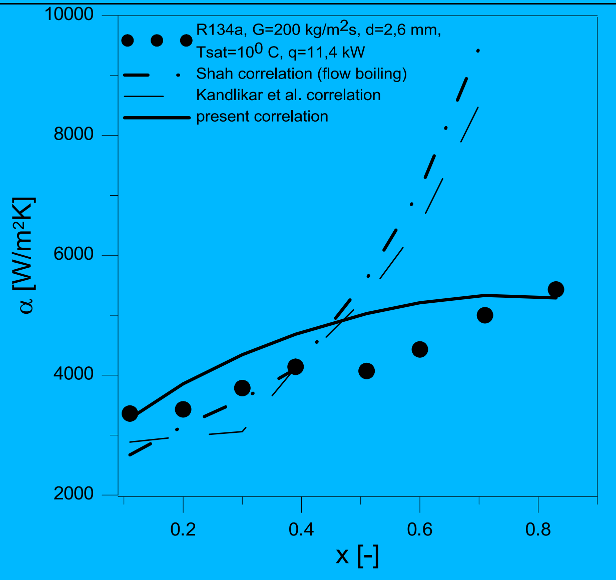


Fig. 12. Flow boiling heat transfer coefficient for R134a, Copetti et al. [14].

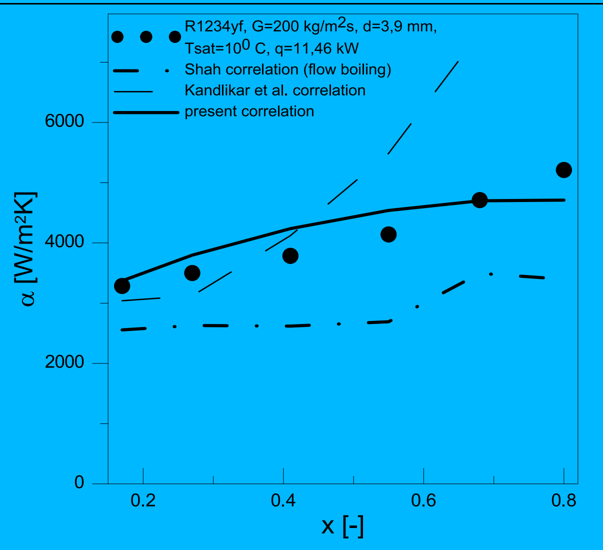


Fig. 13. Flow boiling heat transfer coefficient for R1234yf, Lu et al. [12].

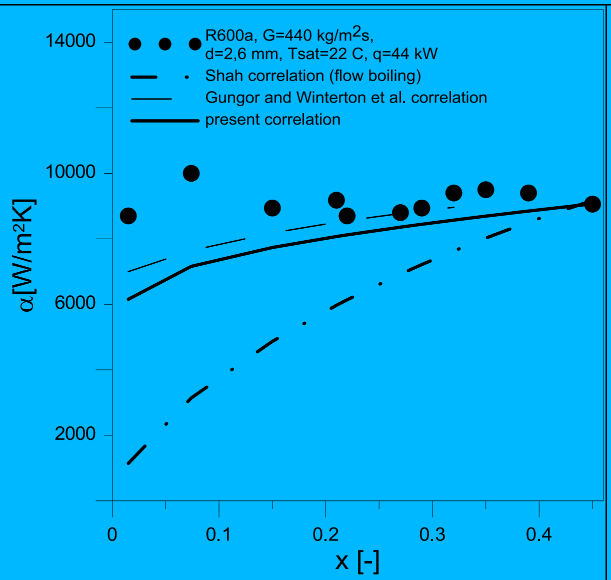


Fig. 14. Flow boiling heat transfer coefficient for R600a, Copetti et al. [14]

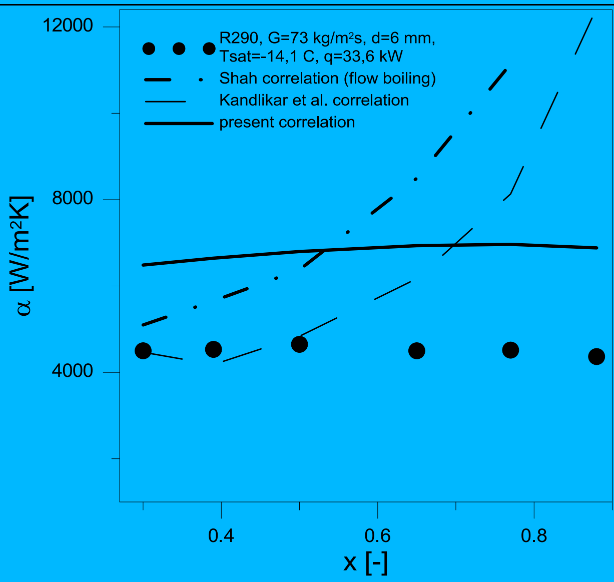


Fig. 15. Flow boiling heat transfer coefficient for R290, Wang et al. [13].

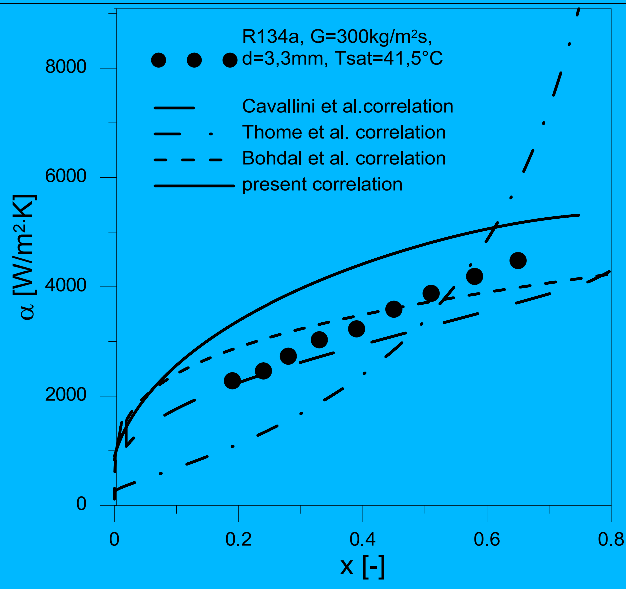
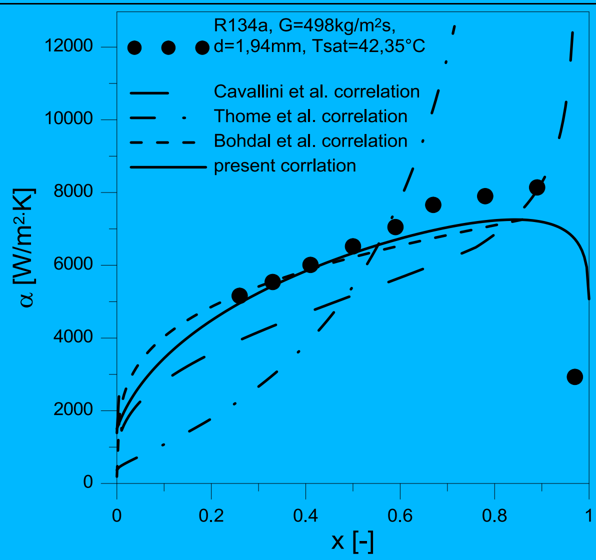


Fig. 16. Flow condensation heat transfer coefficient for R134a, Bohdal [9], d=3.3 mm.



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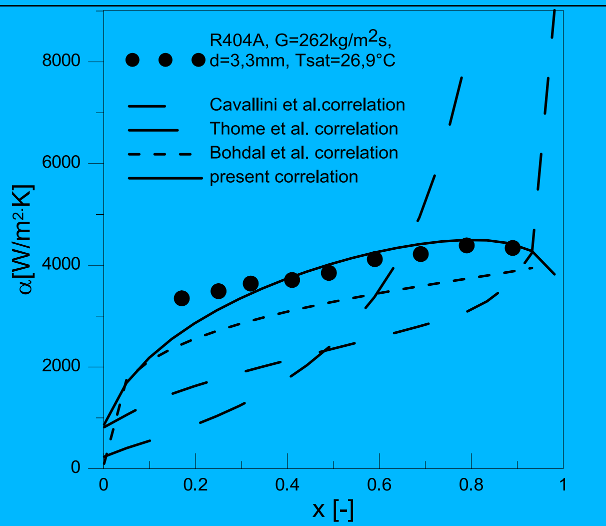

Fig. 18. Flow condensation heat transfer coefficient for R404A, Bohdal [9], d=3.3m.

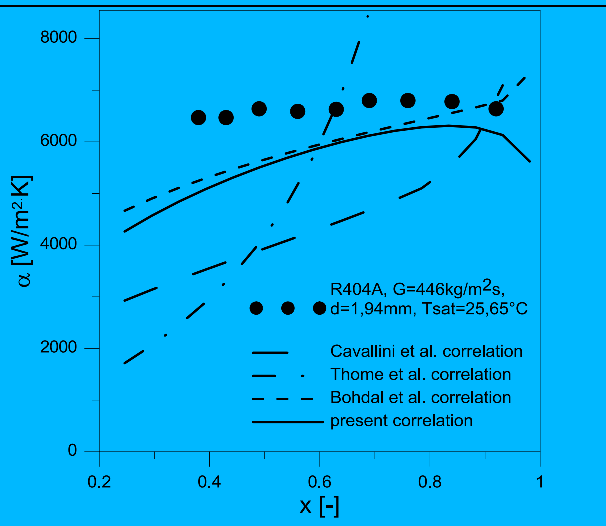

Fig. 19. Flow condensation heat transfer coefficient for R404A, Bohdal [9], d=1.94mm.

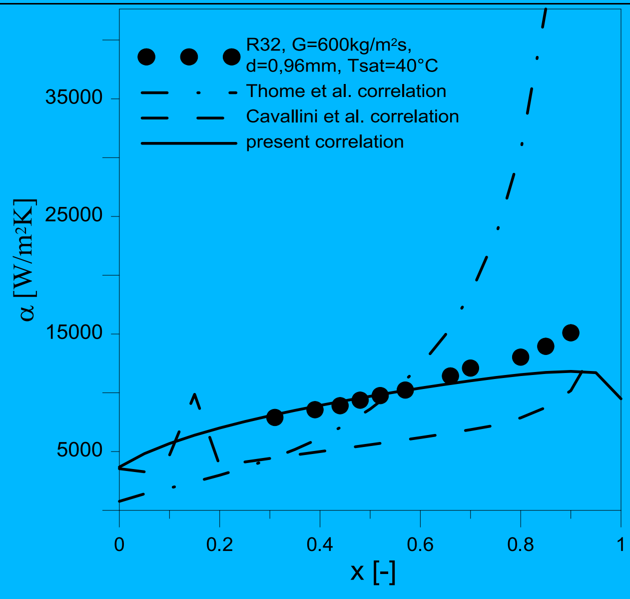

Fig. 20. Flow condensation heat transfer coefficient for R32, Matkovic et al. [11], d=0.96mm.

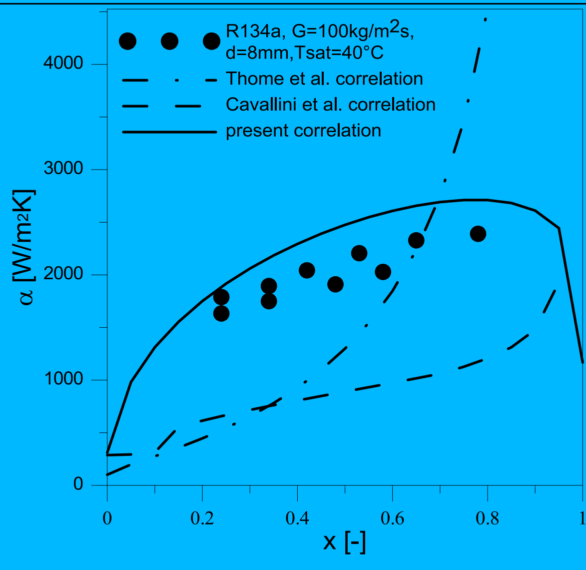

Fig. 21. Flow condensation heat transfer coefficient for R134a, Matkovic et al. [11], d=8mm.