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A common method for calculation of flow boiling and flow condensation heat transfer coefficients in minichannels with account of non-adiabatic effects

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Abstract Flow boiling and flow condensation are often regarded as two opposite or symmetrical phenomena, however their description with a single correlation has yet to be suggested. In the case of flow boiling in minichannels there is mostly encountered the annular flow structure, where the bubble generation is not present. Similar picture holds for the case of inside tube condensation, where annular flow structure predominates. In such case the heat transfer coefficient is primarily dependent on the convective mechanism. In the paper a method developed earlier by D. Mikielewicz et al. [1] is applied to calculations of heat transfer coefficient for flow condensation. The modification of interface shear stresses between flow boiling and flow condensation are considered through incorporation of the so called blowing parameter, which differentiates between these two modes of heat transfer. Satisfactory consistency with well established correlations for condensation has been found as well as with selected experimental data.

Keywords: Flow boiling, Condensation inside tubes, Minichannels

1. Introduction

Flow boiling and flow condensation are often regarded as two opposite or symmetrical phenomena involving the change of phase. There is a temptation to describe both these phenomena with one only correlation, however no such model has yet been suggested. In both cases of phase change there is found an annular structure, which seems to be mostly appropriate to common modeling. However, in the case of flow boiling in conventional channels one can expect that bubble nucleation renders the process of heat transfer not to have its counterpart in the condensation inside tubes. Similarly in as the case of inside tube condensation, where the collapse of bubbles to form a continuous liquid structure is the condensation specific phenomenon. Situation seems to be a little less complex in the case of flow boiling in minichannels and microchannels. In such flows the annular flow structure is dominant for most qualities, Thome and Consolini [2]. Then the heat transfer coefficient is primarily dependent on the convective mechanism. Most of correct modeling of heat transfer in case of condensation inside channels relates the heat transfer coefficient to the friction coefficient. Such modeling is rather not commonly used in case of flow boiling. In that case, all existing approaches are either the empirical fits to the experimental data, or form an attempt to combine two major influences to heat transfer, namely the convective flow boiling without bubble generation and nucleate boiling. Generally that is done in a linear or non-linear manner. Alternatively, there is a group of modern approaches based on models which start from modeling a specific flow structure and in such way postulate more accurate flow boiling models, usually pertinent to slug and annular flows. The most popular approach, however, to model flow boiling is to present the resulting heat transfer coefficient in terms of a combination of nucleate boiling heat transfer coefficient and convective boiling heat transfer coefficient:

$$\alpha_{TP} = \left[(\alpha_{cb} F)^n + (\alpha_{PB} S)^n \right]^{1/n} \tag{1}$$

where α_{PB} – pool boiling heat transfer coefficient, α_{cb} – liquid convective heat transfer coefficient, which can be evaluated using for example the Dittus-Boelter type of correlation. Exponent n is an experimentally fitted coefficient without recourse to any theoretical foundations. Function S is the so called suppression factor which accounts for the fact that together with increase of vapour flow rate the effect related to forced convection increases, which on the other hand impairs the contribution from nucleate boiling, as the thermal layer is reduced. The parameter F accounts for the increase of convective heat transfer with increase of vapour quality. That parameter always assumes values greater than unity, as flow velocities in two-phase flow are always greater than in the case of single phase flow. The approach represented by equation (1) is usually dedicated to Rohsenow [3], who suggested a linear superposition with n=1, which has been later modified by Chen [4], who incorporated the suppression and enhancement functions, S and F respectively. The correlation due to Chen is used up to date with a significant appreciation in case of flows in conventional size tubes. It was also Kutateladze [5], who recommended a superposition approach, but combined in a geometrical rather than linear manner with the value of exponent n=2. A similar summative non-linear approach has been recommended later by Steiner and Taborek [6] with n=3.

The objective of the present paper is to present the capability of the flow boiling model, presented earlier by authors in [1] to model condensation inside tubes. The model presented here is enhanced in comparison to its earlier version to incorporate a "blowing parameter" which redefines the shear stresses on the vapour-liquid interface to yield a more correct behaviour.

2. Dissipation based model of flow boiling

A fundamental hypothesis in the original model under scrutiny here is the fact that heat transfer in flow boiling with bubble generation, treated here as an equivalent flow of liquid with properties of a two-phase flow, can be modeled as a sum of two contributions leading to the total energy dissipation in the flow, namely energy dissipation due to shearing flow without the bubbles, E_{TP}, and dissipation resulting from bubble generation, E_{PB}, J. Mikielewicz [7]:

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

Energy dissipation under steady state conditions in the two-phase flow can be approximated as energy dissipation in the laminar boundary layer, which dominates in heat and momentum transfer in the considered process. Analogically the energy dissipation due to bubble generation in the two-phase can be expressed. Substituting the definition of respective energies into (2) a geometrical relation between respective friction factors is obtained:

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

It is difficult to imagine the flow resistance during the generation of bubbles, however in Russian literature there is a number of contributions, where such studies into flow resistance caused merely by the generation of bubbles on the wall were reported, see for example Ananiev [8], which confirm that the modeling approach presented in the paper is possible. Making use of the analogy between the momentum and heat we can generalize the above result to extend it over to heat transfer coefficients to yield heat transfer coefficient in flow boiling with bubble generation in terms of simpler modes of heat transfer, namely heat transfer coefficient in flow without bubble generation and heat transfer coefficient in nucleate boiling:

$$\alpha_{TPB}^2 = \alpha_{TP}^2 + \alpha_{PB}^2 \tag{4}$$

Equation (4) returns a value of n=2 and hence it confirms on physical grounds why in equation (1) there should be used that value of exponent n.

Heat transfer without bubble generation, α_{TPB}, can be modeled in terms of the two-phase flow multiplier. From the definition of the two-phase flow multiplier the pressure drop in two-phase flow can be related to the pressure drop of a flow where only liquid at a flow rate G is present:

$$\Delta p_{TP} = R\,\Delta p_L \tag{5}$$

In (5), R denotes the two-phase flow multiplier. The 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 flowing with velocity w_{TP}:

$$\Delta p_{TP} = \frac{l}{d}\xi_{TP}\rho_L \frac{w_{TP}^2}{2} \tag{6}$$

The pressure drop of the liquid flowing alone can be determined from a corresponding single phase flow relation:

$$\Delta p_L = \frac{l}{d}\xi_L \rho_L \frac{w_L^2}{2} \tag{7}$$

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 laminar valid expression. In effect obtained is a relation enabling calculation of heat transfer coefficient in flow boiling without bubble generation in the form:

$$\frac{\alpha_{TPB}}{\alpha_L} = \sqrt{R_{MS}^n} \tag{8}$$

That is also the form which will be used later in calculations of condensation inside tubes. In case of flow boiling in (8) n=2 for laminar flows, whereas for turbulent flows that value is taking up a value of 0.9. The two-phase flow multiplier R_{MS} due to Müller-Steinhagen and Heck [9] is recommended for use in case of refrigerants, Ould Didi et al. [10] and Sun and Mishima [11]. In case of consideration of bubble generation the following expression is valid for calculation of heat transfer, D. Mikielewicz et al [1]:

$$\frac{\alpha_{TPB}}{\alpha_L} = \sqrt{R_{MS}^n + \frac{1}{1+P}\left(\frac{\alpha_{PB}}{\alpha_L}\right)^2} \tag{9}$$

In (9) the correction term, $P = 2.53 \times 10^{-3}\, \mathrm{Re}^{1.17}\, Bo^{0.6}\left(R_{MS} - 1\right)^{-0.65}$, has been established by a method of multiple regression fitting. The pool boiling heat transfer coefficient α_{PB}, is to be calculated from the relation due to Cooper [12]. The applied heat flux is incorporated through the boiling number Bo, defined as, $Bo=q/(Gh_{LG})$. For the same difference between the wall and saturation temperature there is a different temperature gradient in the fluid in case of pool boiling and flow boiling. In the case of flow boiling the boundary layer is thinner and hence the gradient of temperature is more pronounced, which suppresses generation of bubbles in flow boiling. That is the reason why heat flux is included in modeling. That term is more important for conventional size tubes, but cannot be totally neglected in small diameter tubes in the bubbly flow regime, where it is important. Postulated form of correction has an appearance preventing it from assuming values greater than one, which was a fundamental weakness of the model in earlier modifications.

It should be noted however that the choice of a two-phase flow multiplier to be used in the postulated model is arbitrary. In the frame of works into modeling heat transfer to refrigerants the Muller-Steinhagen and Heck model has been selected for use as it is regarded best for refrigerants such as hydrocarbons in predicting appropriate consistency with experimental data, however, a different model could be selected such in case of dealing with other fluids as for example the Lockhart-Martinelli model, where the two-phase flow multiplier is a direct function of the Martinelli parameter, see Sun and Mishima [11]. The latter model is often found in correlations of flow boiling without bubble generation similar to (1). Another conclusion could be drawn from the presented model that in correlations of the type of equation (1) the two-phase flow multiplier could also be used for modeling instead of the Martinelli parameter. Author's up to date experience shows that the influence of the two-phase flow multiplier is very important, D. Mikielewicz [13]. In the presented model the R_{MS} acts in the correction P as a sort of convective number, known from other correlations. In the form applicable to conventional and small diameter channels the Muller-Steinhagen and Heck model yields:

$$R_{MS} = \left[1 + 2\left(\frac{1}{f_1} - 1\right) x\, Con^{m}\right] \cdot (1-x)^{1/3} + x^3 \frac{1}{f_{1z}} \tag{10}$$

where $Con=(\sigma/g/(\rho_L-\rho_G))^{0.5}/d$ and m=0 for conventional channels. Best consistency with experimental data, in case of small diameter and minichannels, is obtained for m=-1. In (10) $f_1=(\rho_L/\rho_G)\,(\mu_L/\mu_G)^{0.25}$ 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 vapour, i.e. $\alpha_{TPB}\cong\alpha_{G.}$ Hence:

$$f_{1z} = \frac{\alpha_G}{\alpha_L} \tag{11}$$

where $f_{1z}=(\lambda_G/\lambda_L)$ for laminar flows and for turbulent flows $f_{1z}=(\mu_G/\mu_L)(\lambda_L/\lambda_G)^{1.5}(c_{pL}/c_{pG})$. The correlation (9) seems to be quite general, as confirmed for example by the study by Chiou et al. (2009).

3. Condensation inside tubes

Condensation inside tubes has been a topic of interest of not too many investigations. Mentioned here should be studies by Cavallini et al. [14], El Hajal et al. [15], Thome et al. [16] and Garimella [17]. Flow condensation at high heat fluxes enables removal of significant heat fluxes. In case of condensation in small diameter channels the surface phenomena together with the characteristics of the surface itself become more important, as well as interactions between the wall and fluid.

In microchannels we observe domination of forces resulting from action of surface tension and viscosity over the gravitational forces. Hence the attempt to extend the range of validity of correlations developed for conventional channels onto the channels with small diameters often leads to errors in pressure drop and heat transfer description, making such approaches useless. Additionally, the heat transfer coefficient and pressure drop in microchannels strongly depend upon the quality, especially in case when annular flow structure is encountered. Hence the detection of flow structures and their influence on pressure drop and heat transfer is indispensable during the condensation of the fluid.

A pioneering work to modeling of flow condensation was presented by Akers et al. [18], valid for the most commonly found flow structure, namely the annular flow:

$$\alpha_{TPK} = 0.026\frac{\lambda_l}{d}\Pr_l^{1/3}\left[\frac{G(1-x)}{\mu_l}\right]^{0,8}\left[\frac{x}{(1-x)}\left(\frac{\rho_l}{\rho_g}\right)^{0,5}+1\right] \tag{12}$$

Empirical correlation due to Shah [19] is one of the most general and widely used for calculations of heat transfer coefficients in flow condensation. It has been developed on the basis of experimental data accomplished for water, R11, R12, R22, R113, methanol, ethanol, toluene and trichloroethylene in flowing in vertical, horizontal and inclined tubes. In the development of that model it was concluded that in the case of lack of nucleate boiling, which is the case for condensation, the heat transfer coefficient should be close to the one for the annular flow structure:

$$\frac{\alpha_{TPK}}{\alpha_{LO}} = (1-x)^{0,8} + \frac{3,8x^{0,76}(1-x)^{0,04}}{(p/p_{kr})^{0,38}} \tag{13}$$

Traviss and Rohsenow [20] used the analogy between exchange of heat and universal velocity distribution to obtain correlation for heat transfer coefficient in the annular flow. On the basis of assumed velocity profile the authors obtained a relation describing the heat transfer coefficient during condensation as a function of turbulent liquid film thickness. Assuming that the stresses at the interface and wall stresses were comparable the following relation for heat transfer coefficient has been obtained:

$$\frac{\alpha_{TPK} D}{\lambda_l} = \frac{0{,}15\,\mathrm{Re}_l^{0,9}\,\mathrm{Pr}_l}{F_T}\left(\frac{1}{X_{tt}} + \frac{1}{X_{tt}^{0,476}}\right) \tag{14}$$

$$F_T = 5\,\mathrm{Pr}_l + 5\ln\left(1 + 5\,\mathrm{Pr}_l\right) + 2{,}5\ln\left(0.0031\,\mathrm{Re}_l^{0,812}\right) \quad for \quad \mathrm{Re}_l > 1125 \tag{15}$$

$$F_T = 5\,\mathrm{Pr}_l + 5\ln\left(1 + 5\,\mathrm{Pr}_l\left(0{,}0964\,\mathrm{Re}_l^{0,585} - 1\right)\right) \quad for \quad 50 < \mathrm{Re}_l < 1125 \tag{16}$$

$$F_T = 0{,}707\,\mathrm{Pr}_l\,\mathrm{Re}_l^{0,5} \quad for \quad \mathrm{Re}_l < 50 \tag{17}$$

The Reynolds number in the above equations is calculated from formula $\mathrm{Re}_l = G(1-x)d/\mu_l$.

Dobson and Chato [21] noticed that the metod of analysis of the boundary layer, used by some researchers and in that light by Traviss and Rohsenow [20] in particular, is similar to the approach utilizing the two-phase flow multiplier, used by other authors. They found that the foundation of thermal resistance in the annular flow are the laminar and buffer sublayers. They regarded necessary incorporation of multi-zone model of thermal resistance in liquid film, considering also the presence of waves at the phase interface or variation of liquid film thickness. With such assumptions the following correlation for annular flow has been postulated:

$$Nu_{TPK} = 0{,}023\,\mathrm{Re}_l^{0,8}\,\mathrm{Pr}_l^{0,3}\left(1 + \frac{2}{X_{tt}^{0,89}}\right) \tag{18}$$

The authors recommended a separate heat transfer model describing the heat transfer for the case of wavy flow structure and suggested to use the Nusselt number as for the annular flow in case when G>500 kg/m^{2}s, whereas in case when G<500 kg/m^{2}s together with the value of Froude number is greater than 20 to use the Nusselt number as for the annular flow, and in case where Froude number is smaller than 20, then to use the Nusselt number as for the wavy flow structure.

The accuracy of the methods presented above is not fully satisfactory. One of the possible reasons for underestimation of data is that the models based on the annular flow structure, are derived using the stresses determined for the conventional size channels.

4. Non-adiabatic effects in flow boiling and flow condensation

The shear stress between vapour phase and liquid phase is generally a function of non-adiabatic effects. That is a major reason why that up to date approaches, considering the issue of flow boiling and flow condensation as symmetric, are failing in that respect. The way forward is to incorporate a mechanism into the convective boiling term responsible for modification of shear stresses at the vapour-liquid interface. As our objective is to devise a model applicable both to flow boiling and flow condensation we will attempt to improve the model (9) 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, J. Mikielewicz [22]. The devised formula for modification of shear stresses in the boundary layer reads:

$$\tau^{+} = 1 + \frac{B}{\tau_0^{+}} u^{+} \tag{19}$$

In (19) $\tau^{+}=\tau/\tau_w$, $\tau_0^{+}=\tau_w/\tau_{w0}$, where τ_{w0} is the wall shear stress in case where the non-adiabatic effects are not considered, and $B=2\vartheta_0/(c_f\, u_\infty)$. 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 when $Re\rightarrow\infty$ the relation (19) tends to that suggested by Kutateladze and Leontiev [23], which reads:

$$\tau_0^{+} = \left(1 - \frac{B}{4}\right)^2 \tag{20}$$

On the other hand, in case of small values of B the relation (20) reduces to that recommended by Wallis [24]:

$$\tau_0^{+} = \left(1 - \frac{B}{2}\right) \tag{21}$$

The analyses due to Wallis [24] and Kutateladze and Leontiev [23] were carried out for the case of flow boiling.

There are also other studies which appeared only recently, where a way of describing the shear stress in flow condensation is presented, Bai et al. [25]. These authors recommended to consider the stresses as the interaction between the liquid and vapour phases including both frictional shear stress due to different velocities in the liquid and vapour phases and a momentum-transfer shear stress due to vapour condensation in the following form:

$$\tau_0 = \tau_f + \tau_{mt} \tag{22}$$

In (22):

$$\tau_f = \pm \frac{1}{2} f \rho_v (u_v - u_l)^2 \tag{23}$$

The "+" sign is applied when the average velocity of vapour phase is greater than that of liquid at the liquid/vapour interface, otherwise "-" is applied. The momentum-transfer shear stress due to vapour condensation reads:

$$\tau_{mt} = \frac{q_w}{h_{lv}} (u_v - u_l) \tag{24}$$

Relation (24) can be expressed in the following form:

$$\tau_0^+ = \frac{\tau_0}{\tau_f} = \left(1 + \frac{\dfrac{q_w}{h_{lv}} (u_v - u_l)}{\pm \dfrac{1}{2} f \rho_v (u_v - u_l)^2} \right) = \left(1 + \frac{B'}{2} \right) \tag{25}$$

The general form of (25) reminds closely that of (21), confirming that the shear stresses at the liquid/vapour interface can be modified also in case of flow condensations.

Such approach is to be also incorporated in the present work. In the present paper the blowing parameter is defined in the following way:

$$B = \frac{2\vartheta_0}{c_f u_\infty} = \frac{2q}{c_{f0}(u_v - u_l)h_{lv}\rho_l} \tag{26}$$

The considerations by J. Mikielewicz [22] were pertaining to both cases, i.e. flow boiling and flow condensation. Having acquired the way to modify the stresses in flow boiling and flow condensation it is relatively straightforward to implement (21) in the model descrided above by equation (9). A general form applicable both to flow boiling and flow condensation, utilising the definition of blowing parameter (26), therefore reads:

$$\frac{\alpha_{TPB}}{\alpha_L} = \sqrt{\left[R_{MS}\left(1 \pm \frac{B}{2}\right)\right]^n + \frac{1}{1+P}\left(\frac{\alpha_{PB}}{\alpha_L}\right)^2} \tag{27}$$

Obviously, in case of calculations for flow condensation the second term on the right hand side, responsible for bubble generation, has to be omitted, ie. $1/(1+P)(\alpha_{PB}/\alpha_L)^2=0$. The simplicity of the blowing parameter is not disregarding its practical importance. The shear stresses are modified due to non-adiabatic effects on convection and that simple formulation enables to consider that fact.

4. Results of calculations

Presented below is comparison of selected correlations for calculations of flow condensation with the model presented in the first part of the paper, namely relation (8). The proposed model was thoroughly tested for the conditions of flow boiling, see for example D. Mikielewicz [13], showing satisfactory performance. A good agreement has also been obtained in case of fluids, for which relation (9) was not tested yet, Chiou et al. [26]. That encouraged to pursuit further the extension of the model to flow condensation conditions. Obviously in that case the full form of the flow boiling correlation (9) cannot be used, as in that case the bubble generation is not present. Firstly, the comparisons have been carried out for two fluids, namely R123 and R134a for two channel diameters, i.e. 1.15mm and 2.3mm using the relation (8). Calculations have been carried out for the assumed condensation temperature $t_k=50^oC$, heat flux density $q=20\,000\ W/m^2$ and mass velocity $G=600\ kg/m^2s$, which corresponds to turbulent flow conditions. The results of calculations of heat transfer coefficient against quality have been presented in Figures 1 to 4.

It can be seen that equation (8) describes well the heat transfer coefficients during the flow condensation. In the majority of calculations it is consistent with the correlation due to Traviss et al. [20], which, on the other hand, is regarded as one of the most accurate models for calculations of heat transfer coefficients in flow condensation. The biggest advantage offered by equation (8) is the fact that it has a general character and does not require any specific fluid-related constants. Relation (8) does not require prior knowledge of flow maps which are indispensable in case of more accurate methods for calculation of heat transfer coefficients. The accuracy of predictions can be improved through a selection of a more appropriate expression for the two-phase pressure drop. In the selection of appropriate model the recent study by Sun and Mishima [11] may be helpful.

Next, attention is focused on incorporation of the non-adiabatic effects, i.e. expression (27). The calculations have been completed for the same conditions and fluids as in figures 1 to 4. The results of calculations showing the influence of non-adiabatic effects are presented in figures 5 to 8. Examination of these figures show that there is a clear effect of the modifications of shear stresses in condensation on the value of heat transfer coefficient.

Mainly the effect is marked for smaller values of quality, where the correction is reaching even 200%. For values of higher qualities the discrepancy is much lower, not exceeding 10%. The modifications of the heat transfer coefficient are occurring qualitatively in the correct direction. The modified model (8) has been further tested against other experimental data. Finally, some comparisons were made for the data of Matkovic et al. [27]. The results of comparisons are very promising. First of all, the model of heat transfer coefficient with non-adiabatic correction performs qualitatively correctly, confirming the fact that the heat transfer is a function of two-phase pressure drop. The biggest discrepancy is found in case of higher qualities. That probably could be improved if better models for pressure drop are developed. The exact behaviour of the blowing parameter should also attract further and more comprehensive attention as the phenomenon of condensation may require more detailed description of velocity changes at interface.

5. Conclusions

In the paper presented is a comparison of predictions of condensation inside channels with the correlation developed for flow boiling on the basis of predictions of heat transfer coefficients in flow condensations. The comparisons were made with correlations developed for the annular flow structure, where flow boiling and flow condensation can be regarded as symmetrical phenomena. The comparison is satisfactory.

Additionally studied were the non-adiabatic effects in the description of the model of heat transfer coefficient (9). The blowing parameter has been incorporated into modeling. The modification leads in case of condensation to biggest effects for small qualities, confirming the fact that non-adiabatic effects for higher qualities are of smaller importance. Also in case of flow boiling the biggest modifications to heat transfer coefficient occur for smaller qualities (smaller than 0.4).

In case of modeling of flow condensation the postulated model (9) and (27) could further be enhanced through incorporation of and additional term related to the work of bubble collapse to capture better the transition from the annular flow structure to bubbly flow. More detailed experimental data regarding that flow regime should be collected.

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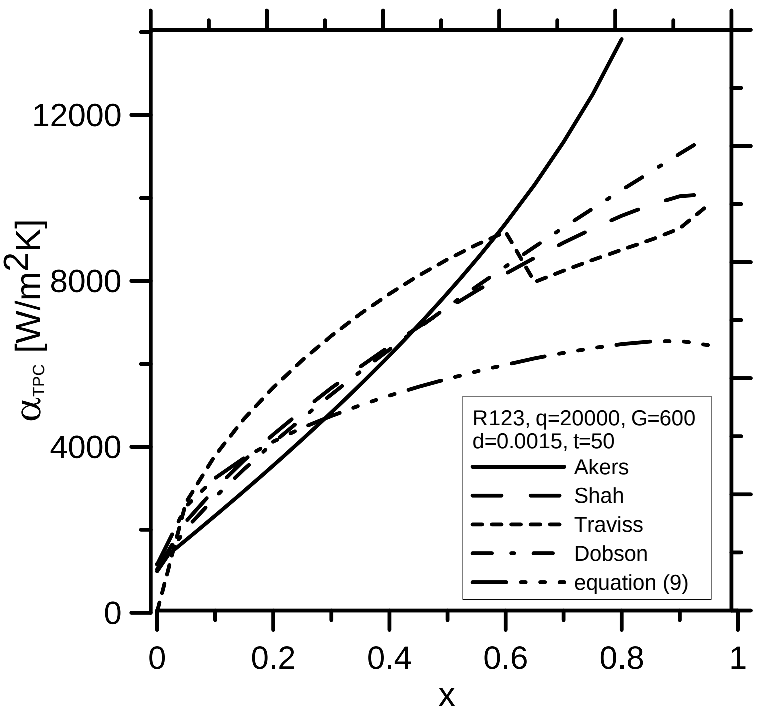

Fig. 1. Comparison of predictions of heat transfer coefficient for R123 using (8) and other correlations, d=1,5 mm

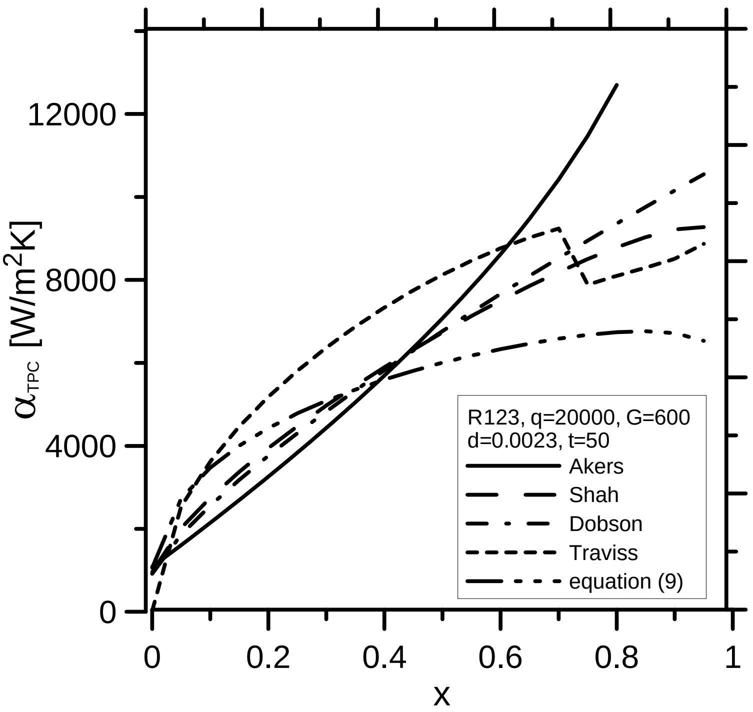

Fig. 2. Comparison of predictions of heat transfer coefficient for R123 using (8) and other correlations, d=2,3 mm

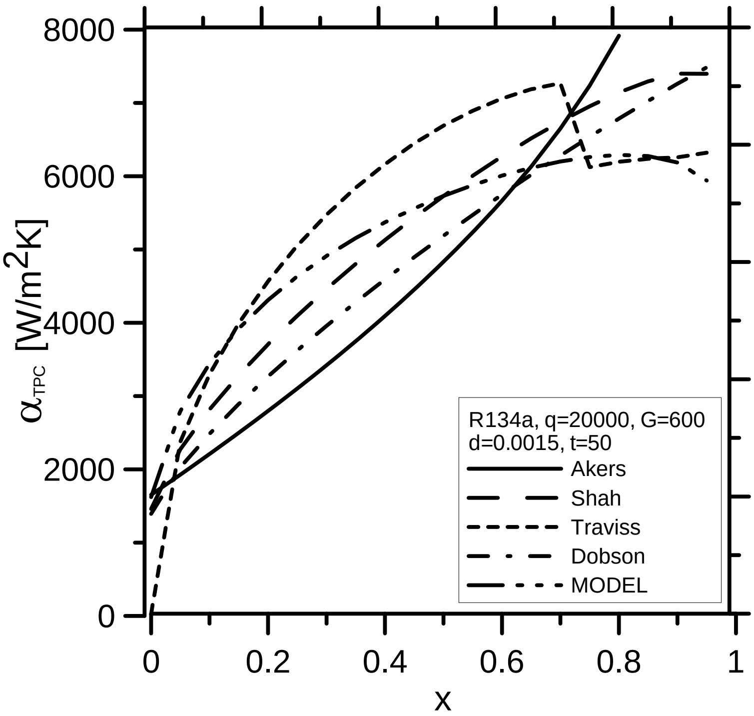

Fig. 3. Comparison of predictions of heat transfer coefficient for R134a using (8) and other correlations, d=1,5 mm

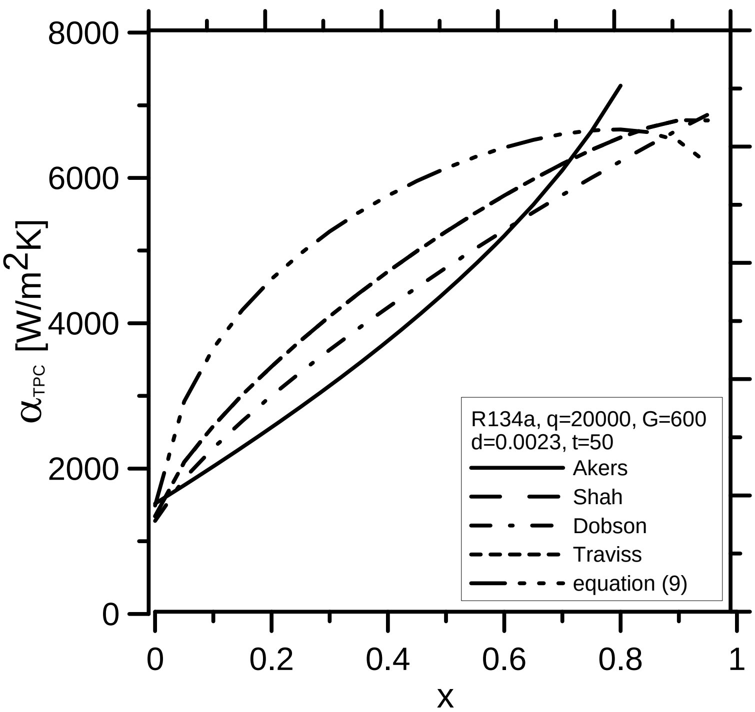

Fig. 4. Comparison of predictions of heat transfer coefficient for R134a using (8) and other correlations, d=2,3 mm

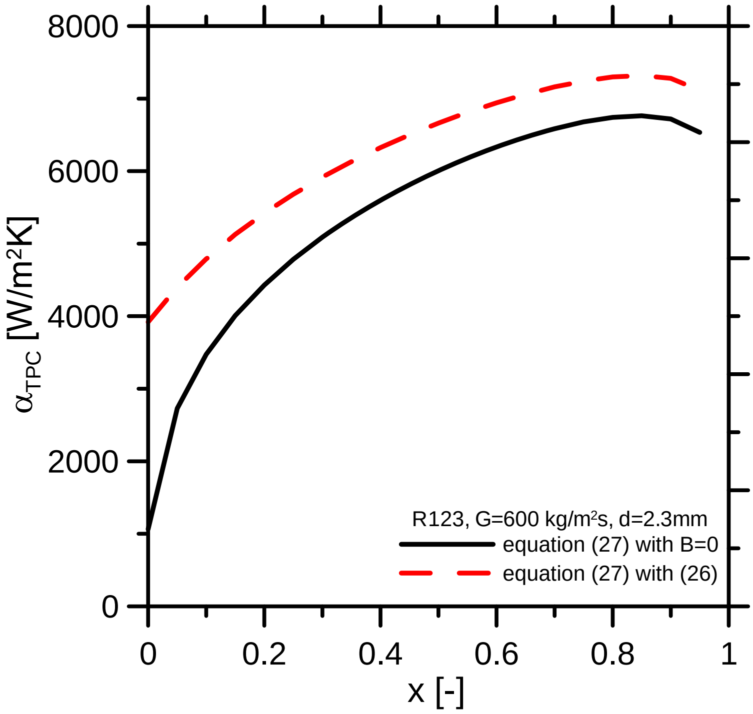

Fig. 5. Comparison of predictions of heat transfer coefficient for R123 using relation (27) with and without blowing parameter, d=2,3 mm

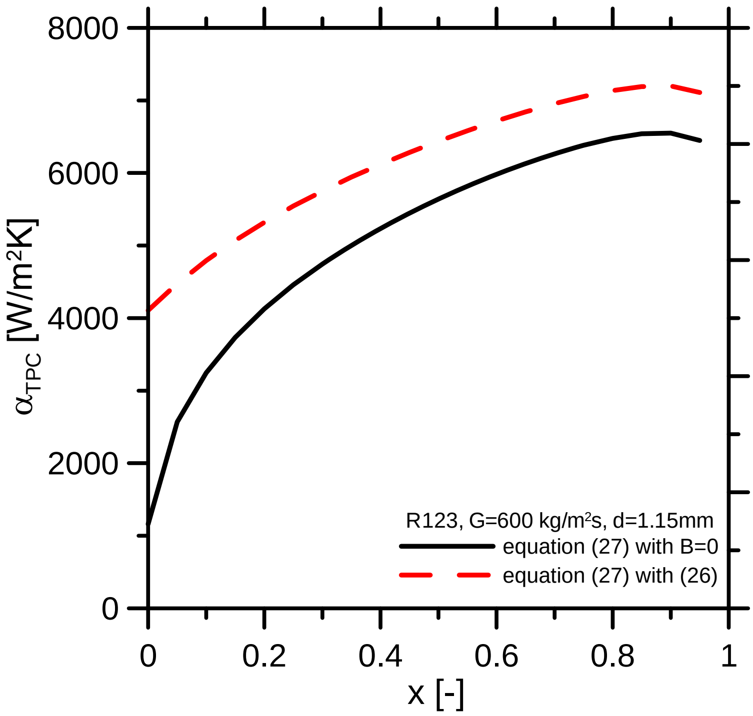

Fig. 6. Comparison of predictions of heat transfer coefficient for R123 using relation (27) with and without blowing parameter, d=1,15 mm

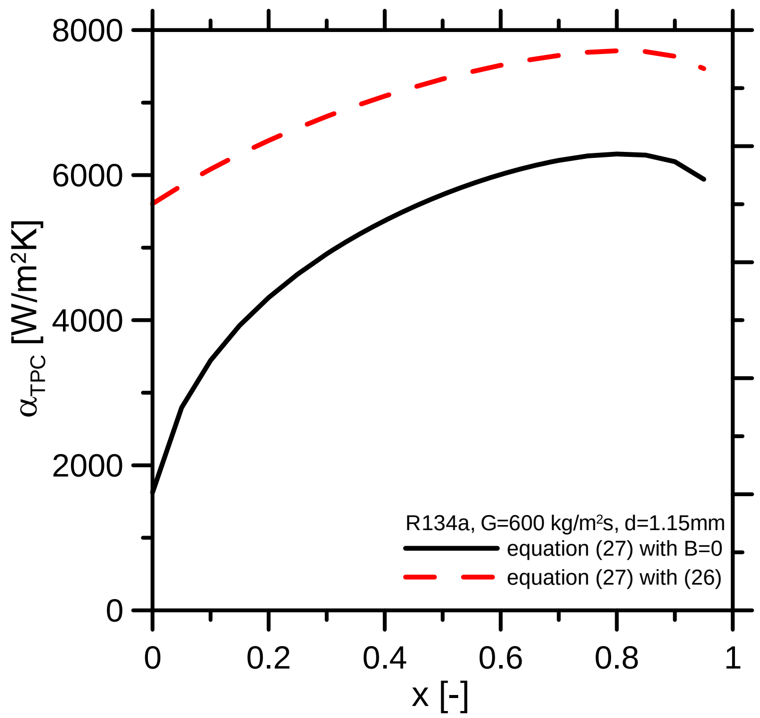

Fig. 7. Comparison of predictions of heat transfer coefficient for R134a using relation (27) with and without blowing parameter, d=2,3 mm

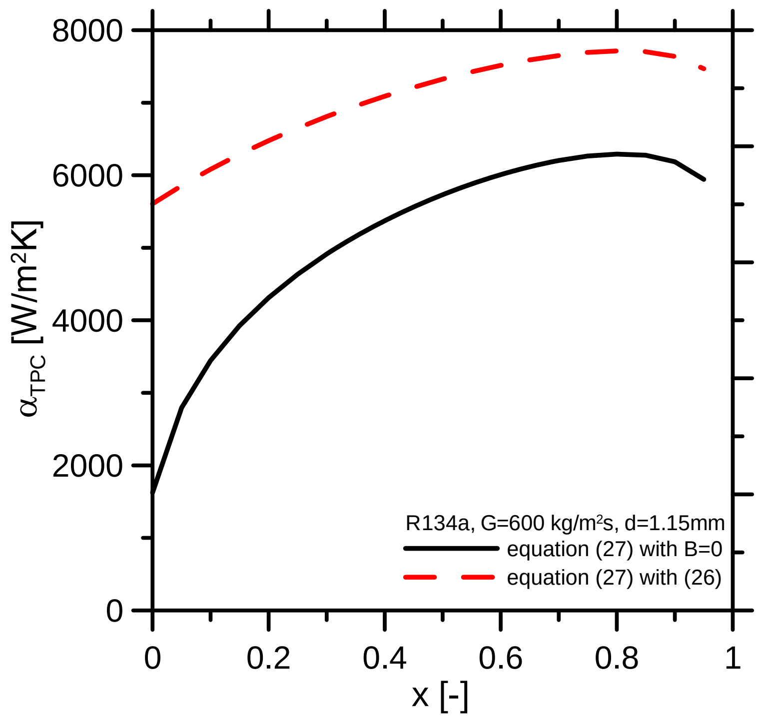

Fig. 8. Comparison of predictions of heat transfer coefficient for R134a using relation (27) with and without blowing parameter, d=1,15 mm

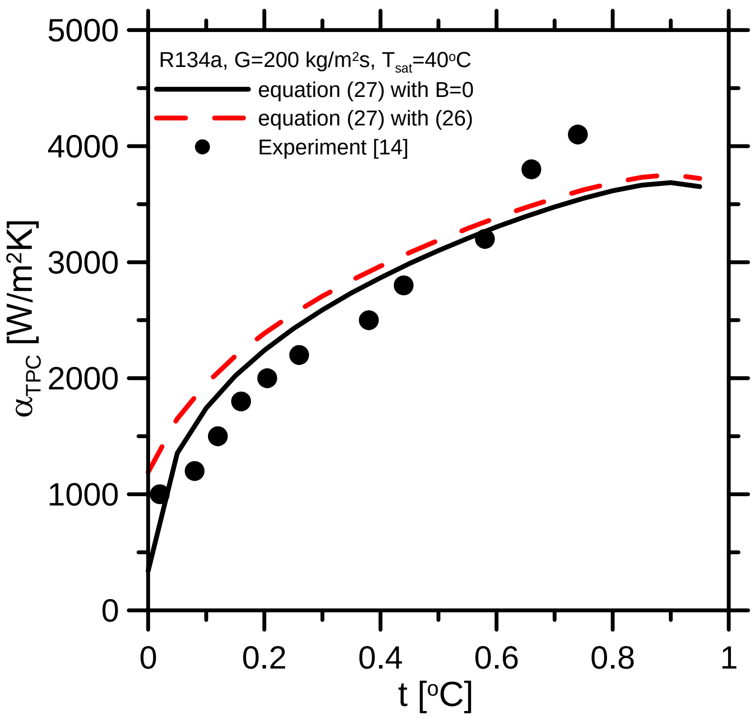

Fig. 9. Comparison of predictions of heat transfer coefficient for R134a in comparison to experimental data due to Matkovic et al. [27] using relation (27) with and without blowing parameter, d=0,96 mm, G=200 kg/m^{2}s.

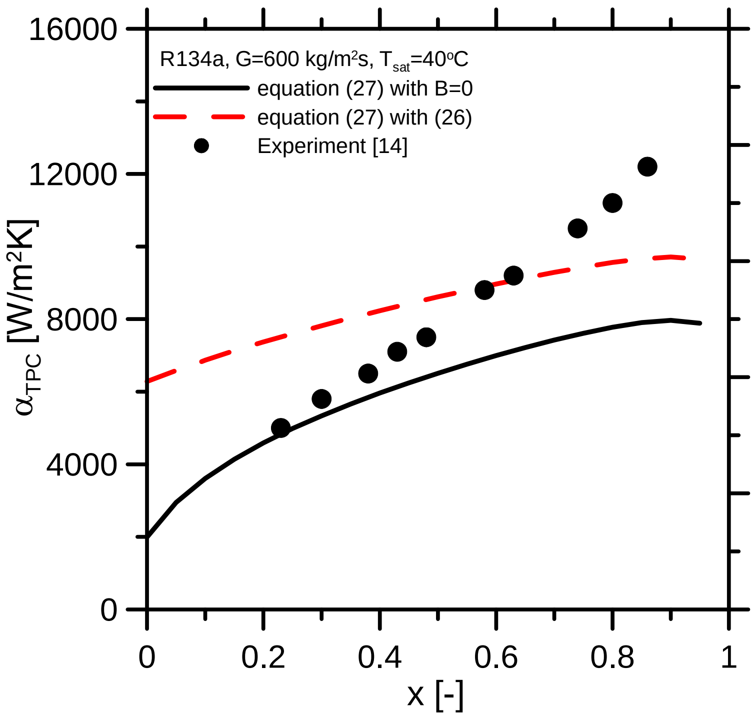

Fig. 10. Comparison of predictions of heat transfer coefficient for R134a in comparison to experimental data due to Matkovic et al. [27] using relation (27) with and without blowing parameter, d=0,96 mm, G=600 kg/m^{2}s.

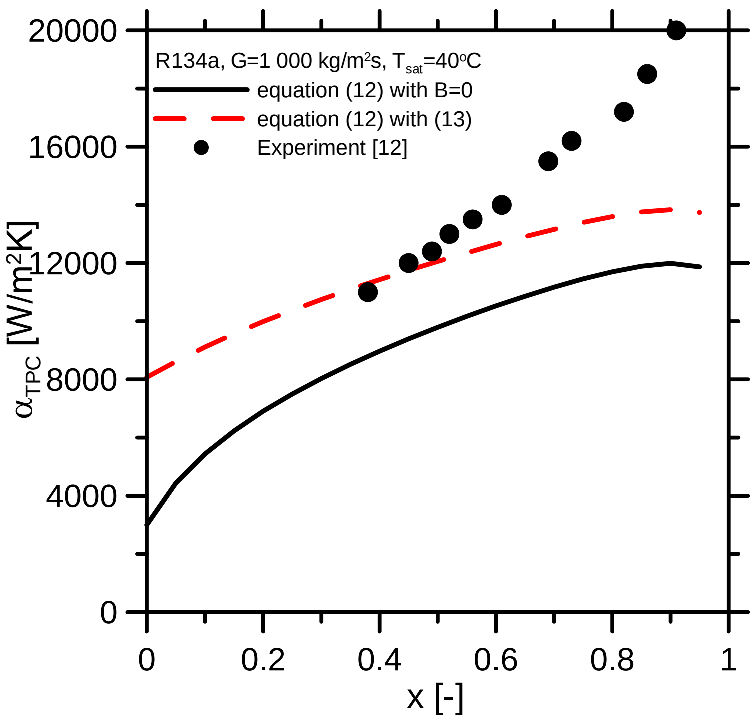

Fig. 11. Comparison of predictions of heat transfer coefficient for R134a in comparison to experimental data due to Matkovic et al. [27] using relation (27) with and without blowing parameter, d=0,96 mm, G=1 000 kg/m^{2}s.