AN IMPROVED METHOD FOR FLOW BOILING HEAT TRANSFER WITH ACCOUNT OF THE REDUCED PRESSURE EFFECT

by

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In the paper are presented the results using the authors own model to predict heat transfer coefficient during flow boiling. The model has been tested against a large selection of experimental data to investigate the sensitivity of the in-house developed model. In the work are presented the results of calculations obtained using the semi-empirical model on selected experimental flow boiling data of the refrigerants: R134a, R1234yf, R600a, R290, NH_3*,* CO_2*, R236fa, R245fa, R152a, and HFE7000. In the present study, particular attention was focused on the influence of reduced pressure on the predictions of the theoretical model. The main purpose of this paper is to show the effect of the reduced pressure on the predictions of heat transfer during flow boiling.*

Key words: *mini-channel, conventional channel, heat transfer coefficient, flow boiling, reduced pressure, synthetic refrigerants, natural refrigerants,* CO_2

Introduction

Nowadays, there is an increasing interest in refrigerants featuring low global warming potential (GWP). The reason for this concern can be attributed to the growing number of regulations and laws prohibiting the use of some synthetic refrigerants. According to these regulations, the new fluids used in *e. g*. air-conditioning and refrigeration applications cannot be implemented with fluorinated greenhouse gases having GWP greater than 150 [1]. Within that document, most of the substances used in the refrigeration system have been regulated due to its ozone depletion potential (ODP). Consequently, one of the working fluids most extensively used in medium evaporation temperatures, such as for example R134a with GWP = 1430 needs to be replaced by more environmentally friendly fluids. Previous studies have considered R152a [2] and the natural refrigerant CO_2 [3] as possible replacements for R134a. On the other hand, CO_2, as compared to contemporary fluids, is a relatively safe one. It is non-toxic, non-flammable, non-explosive, and can be coupled with most metals and plastics. Design of evaporators for use of the CO_2 requires the exact determination of heat transfer coefficient during flow boiling. Available in the literature empirical correlations give different results as compared to the results obtained experimentally [4, 5]. There is hardly any robust and recommended correlation for the purpose of calculation of CO_2 two-phase heat transfer and

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pressure drop, despite some devoted contributions [3]. Moreover, the traditional refrigerants can be also replacement by other natural refrigerants such as R600a and R290 [6]. These working fluids do not exhibit a harmful impact on the ozone layer and have a negligible GWP. Furthermore, the mixture of R290 and R600a can be one of the most appropriate refrigerants to replace R134a [7]. Recently, R1234yf has been believed to be a promising candidate as an alternative of R134a [8]. Its ODP is equal 0, whereas GWP = 4 and its thermophysical properties are similar to those of R134a. In recent years also much attention has been paid to the possible use of fluorinated propene isomers for the substitution of HFC fluids, being in most cases high GWP refrigerants. However, the available hydrofluoroolefins (HFO) cannot cover all the air-conditioning, heat pump and refrigeration systems when used as pure fluids because their thermodynamic properties are not suitable for all operating conditions and therefore some solutions may be found using blends of refrigerants, to satisfy the demand for a wide range of working conditions [9]. In the literature, there are many empirical correlations for modeling of flow boiling heat transfer and pressure drop. One of the recently proposed model for prediction pressure drop is due to Sempertegui-Tapia *et al*. [10] correlation. Ribatski [11] highlighted the problem connecting with differences among data from independent laboratories that can be related to several following aspects: different surface roughness, channel dimensions uncertainties, channel obstructions, inappropriate data reduction procedures, and presence of thermal instabilities [10]. It can be the reason why most of available in the literature empirical correlations give different results as compared to the results obtained experimentally. It should be also noted that developments in many modern applications are associated with the necessity of sizing heat transfer devices and installation. These developments have spurred unprecedented interest in replacing single-phase systems with boiling and condensation counterparts. While computational methods have shown tremendous success in modeling single-phase systems, their effectiveness with phase change systems is limited mostly to simple configurations [12]. This fact has an impact not only on the development of predicting two-phase frictional pressure drop, but also on the prediction of heat transfer coefficient during flow boiling and flow condensation due to the need to dissipate a large heat flux. Flow boiling and condensation phenomena in mini- and micro-scale channels are essential processes involved in a wide range of industrial applications such as heat exchangers, high heat flux cooling and the like in mechanical, chemical, aerospace, energy, automotive and renewable energy, electronics, and also biological and medical engineering [13, 14]. Previously mentioned investigation have shows how reliable prediction methods of two-phase heat transfer coefficient are necessary.

In the article is presented a semi-empirical model for calculations of flow boiling and flow condensation in mini and conventional size channels. It was authors' intention to show the performance of their own approach in predicting the flow boiling heat transfer coefficient on the example of the data collected from the literature using the in-house developed model [15-18]. Based on the evidence of comparisons with experimental data a correction incorporating the effect of reduced pressure has been postulated to the authors own model to provide a better consistency of the predictions with the experimental data.

The modeling

The versatile semi-empirical model for calculations of flow boiling and flow condensation due to Mikielewicz [15] and the final version for the calculations of saturated flow boiling due to Mikielewicz and Mikielewicz [16] and Mikielewicz *et al*. [17-19] has been tested for a significant number of experimental data and has returned satisfactory results for the case of the flow boiling process for numerous fluids. The fundamental hypothesis of the model is the fact that heat transfer during flow boiling with bubble generation can be modeled as a sum of two contributions constituting the total energy dissipation in the flow, namely the energy dissipation due to the shearing flow without the bubbles and dissipation resulting from the bubble generation. The final version of the model reads:

$$\frac{\alpha_{TPB}}{\alpha_{LO}} = \sqrt{\left(\phi_{LO}^2\right)^n + \frac{C}{1+P}\left(\frac{\alpha_{Pb}}{\alpha_{LO}}\right)^2} \tag{1}$$

In eq. (1) $C = 1$ for flow boiling and $C = 0$ for flow condensation, eq. (1) also includes the empirical correction, P, defined by eq. (2). Occurring in the eq. (1) the two-phase multiplier is raised to the power n ($n = 0.76$ for turbulent, $n = 2$ for laminar flow):

$$P = 2.53\cdot 10^{-3}\,\mathrm{Re}^{1.17}\,\mathrm{Bo}^{0.6}\left[\left(\phi_{\mathrm{LO}}^2\right)_{\mathrm{MS}} - 1\right]^{-0.65} \tag{2}$$

The model has been recently extended to cases of subcooled flow boiling [17] and post dry out modeling [20]. In presented calculations tested was the sensitivity of the developed heat transfer model to the selection of the two-phase flow multiplier. For that purpose four well established two-phase flow multiplier models were introduced and tested in eq. (1), namely modified correlation due to Muller-Steinhagen and Heck [21], along with relationships which have been outlined in tab 1.

Table 1. Two-phase frictional pressure drop correlation

Authors	Equations
Muller-Steinhagen and Heck [21]	$$\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{MS}} = \phi_{LO}^2\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}}$$ $$\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{MS}} = \left\{\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}} + 2\left[\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{GO}} - \left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}}\right]x\right\}(1-x)^{1/3} + \left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{GO}} x^3 \tag{3}$$
Modified Muller-Steinhagen and Heck [22]	$$\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{MS_1} = \phi_{LO}^2\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{LO}$$ $$\phi_{LO}^2 = \left[1 + 2\left(\frac{1}{f_1} - 1\right)x\,Con^m\right](1-x)^{1/3} + \frac{1}{f_{1z}}x \tag{4}$$ $$Con = \frac{1}{d_h}\sqrt{\frac{\sigma}{\mathrm{g}\left(\rho_l - \rho_g\right)}}$$ Turbulent flow $$f_1 = (\rho_l/\rho_g)(\mu_l/\mu_g)^{0.25}, \quad f_{1z} = (\mu_g/\mu_l)(\lambda_l/\lambda_g)^{1.5}(c_{p,\mathrm{l}}/c_{p,\mathrm{g}})$$ Laminar flow $$f_1 = (\rho_l/\rho_g)(\mu_l/\mu_g), \quad f_{1z} = (\lambda_g/\lambda_l)$$ $m = 0$ for flow in conventional channels, $m = -1$ for flow in mini-channels

→

Zhang and Webb [23]	$$\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{ZW}} = \phi_{LO}^2 \left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}}$$ $$\phi_{LO}^2 = (1-x)^2 + \frac{2.87x^2}{p_r} + 1.68x^{0.8}(1-x)^{0.25} p_r^{-1.64}$$	(5)
Cioncolini *et al.* [24]	$$-\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_C = 2 f_{tp} \frac{G_c^2}{\rho_c d} + G^2 \frac{d}{\mathrm{d}z}\left[\frac{x^2}{\rho_g \varepsilon} + \frac{e^2(1-x)^2}{\rho_l \gamma (1-\varepsilon)} + \frac{(1-e)^2(1-x)^2}{\rho_l (1-\gamma)(1-\varepsilon)}\right]$$ The full calculation algorithm was presented in [33]	(6)
Sempertegui-Tapia and Ribatski [10]	$$\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{STR}} = F(1-x)^{1/\lambda} + \left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{GO}} x^{\lambda}$$ $$F = \left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}} + \omega\left[\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{GO}} - \left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}}\right]x$$ $$\omega = a e^{b\,\mathrm{Re_{GO}}/1000},\ a = 3.01,\ b = -0.00464,\ \lambda = 2.31$$	(7)
Tran [25]	$$\phi_{\mathrm{LO}}^2 = 1 + \left[4.3\frac{\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{GO}}}{\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}}}\right]\left[N_{\mathrm{conf}} x^{0.875}(1-x)^{0.875} + x^{1.75}\right]$$ $$N_{\mathrm{conf}} = \sqrt{\frac{\sigma}{\mathrm{g}(\rho_l - \rho_g) d_h^2}}$$	(8)
Friedel [26]	$$\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_F = \phi_{\mathrm{LO}}^2 \left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}}$$ $$\phi_{LO}^2 = (1-x)^2 + \frac{x^2\left(\frac{\rho_l}{\rho_g}\right)\left(\frac{f_g}{f_l}\right) + 3.24x^{0.78}(1-x)^{0.224}\left(\frac{\rho_l}{\rho_g}\right)^{0.91}\left(\frac{\mu_g}{\mu_l}\right)^{0.19}\left(1-\frac{\mu_g}{\mu_l}\right)^{0.7}}{\mathrm{Fr}_h^{0.045}\mathrm{We}_l^{0.035}}$$	(9)
New predictive method	$$\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{MS_2}} = \phi_{\mathrm{LO}}^2 \left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{\mathrm{LO}}$$ $$\phi_{\mathrm{LO}}^2 = \phi_{\mathrm{MS_1}}^2\left[1 - \left(\frac{p_{\mathrm{sat}}}{p_{\mathrm{crit}}}\right)^{a_1}\right] + 1,\quad a_1 = 1$$ $$\phi_{\mathrm{MS_1}}^2 = \left[1 + 2\left(\frac{1}{f_1} - 1\right) x Con^m\right](1-x)^{1/3} + \frac{1}{f_{1z}} x^3$$	(10)

It was expected that the accuracy of model predictions could be improved by some modifications to the empirical correction, P, in this case by incorporation of the reduced pressure effect. Taking into account the reduced pressure in the empirical correction, P, in case of modeling the condensation process in the flow will not affect the obtained results of the calculations. This is due to the fact that the empirical correction, P, and thus the reduced pressure included in the calculations, are considered only in the part associated with the generation of bubbles, which is non-existent when modeling the condensation process. The modified in the present work empirical correction, P, yields:

$$P = \left(\frac{p_{\mathrm{sat}}}{p_{\mathrm{crit}}}\right)^{a_2} 2.53 \cdot 10^{-3}\, \mathrm{Re}^{1.17}\, \mathrm{Bo}^{0.6}\left[\left(\phi_{\mathrm{LO}}^2\right)_{\mathrm{MS}} - 1\right]^{-0.65} \tag{11}$$

The exponent a_2, present in the modified eq. (11) was adjusted to the available data bank for flow boiling.

Results and discussions

The collected experimental data came from various studies from literature [28-48] and in case of the HFE7000 for own experimental researches, which were conducted for a full range of quality variation and the relatively wide range of mass velocity and saturation temperature. A full range of variability of parameters for experimental data can be found in [49]. In the work are presented the results of calculations obtained by using the in-house developed semi-empirical model on selected experimental flow boiling data of the refrigerants such as: R134a [27-40], R1234yf [37, 39], R600a [41], R290 [42], NH_3 [36], CO_2 [38, 40, 42-47], R236fa [34], R245fa [34, 48], and HFE7000. The range of the confinement number *Con* and reduced pressure for all considered refrigerants in the analysis are presented in tab. 2. For the collected experimental database, the confinement number *Con* was determined based on *Con* $Con = 1/d_h[\sigma/g(\rho_l - \rho_g)]^{1/2}$. While, the value of reduced pressure is the ratio of saturation pressure and critical pressure. Using the Kew and Cornwell [50] criterion, the available data bank was divided into conventional size channels and mini-channels and when the confinement number is greater than 0.5 then the flow corresponds to the flow in the mini-channel. Belyaev *et al.* [51] tried to confirm the hypothesis that in the case of high value of reduced pressure, the two-phase flow structures in small diameter channels are similar to those occurring in conventional size diameter. Based on their study they observed that when reduced pressure is greater than 0.4 then is no differences between heat transfer during flow boiling in mini-channels and conventional channels. Mauro [52] reports that at the same value of reduced pressure, the thermodynamic properties of refrigerants are very similar, where in in the case of transported properties, these properties are more divergent. The example for selected thermodynamic and transported properties has been shown in figs 1 and 2.

Table 2.The range of confinement number *Con* and reduced pressure

Fluid	*Con*	p_{sat}/p_{crit}
CO_2	0.071-**1.611**	0.359-0.777
NH_3	**0.75-1.167**	0.0455-0.1125
R600a	**0.537-0.855**	0.0884-0.1023
R290	0.021-**0.889**	0.0328-0.273
R152a	**0.639-0.656**	0.14-0.162
R134a	0.234-**1.65**	0.094-0.257
R1234yf	0.218-**0.736**	0.111-0.244
R236fa	**0.826**	0.104
R245fa	0.441-**1.592**	0.050-0.071
HFE7000	0,361-0,396	0.034-0.080

Figure 3 displays the evolution of the two-phase pressure drop gradient with the vapour quality according to the predictive methods from the literature listed in tab. 1. As can be seen the discrepancies in the presented values of two-phase pressure drop are very large. In addition, the slight differences in pressure drop can be observed for the original model due to Muller-Steinhagen and Heck [21] eq. (3), Zhang and Webb [23] eq. (5), and Sempertegui-Tapia and Ribatski [10] eq. (7). Also, the newly proposed model eq. (10) and Zhang and Webb eq. (5) models give very similar results of calculation, but on the other hand, the two-phase pressure drop obtained by the new prediction method eq. (10) is about 20% greater than obtained by Zhang and Webb correlation eq. (5).

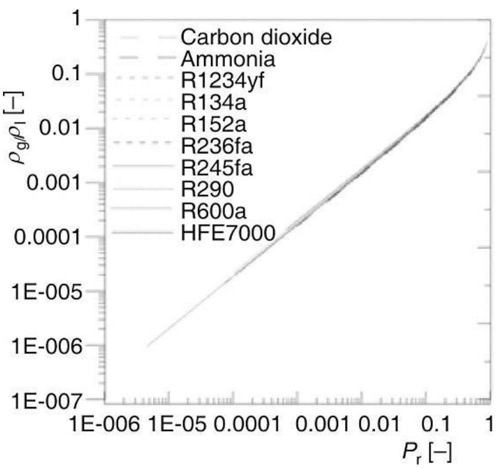

Figure 1. Ratio of gas density to liquid density as a function of reduced pressure

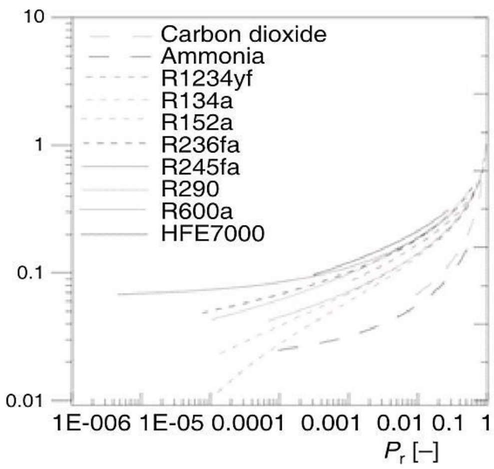

Figure 2. Ratio of gas conductivity to liquid conductivity as a function of reduced pressure

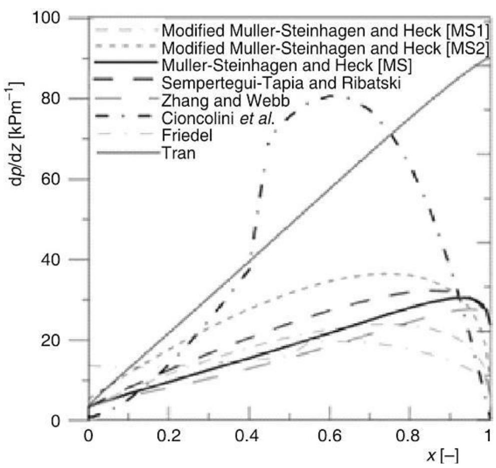

Figure 3. The comparison of correlations describing the two-phase pressure drop for CO_2 at G = 400 kgm^{-2}s^{-1}, T_{sat} = 0°C, and d_h = 1 mm

In tab. 3 have been presented the values of mean absolute error as a results of calculations of heat transfer coefficient using the model presented pressure drops related to boiling in flow using the relationships specified in tab. 1 with exclusion the model described by eq. (3) and without taking into account the effect of reduced pressure in empirical correction, P, eq. (11).

The analysis showed that the most satisfactory results have been obtained if taken into account Friedel and based on Muller-Steinhagen and Heck correlations. In this case, only the new predictive method eq. (10) takes into account the influence of the reduced pressure effect. Because, it was expected that the model can be improved by taking into account the effect of reduced pressure in empirical correction, P, for further analysis in eq. (1), has been adopted the new predictive method eq. (10) as a two-phase flow multiplier and modified empirical correction, P, eq. (11). Therefore the general form of the model eq. (1) with an account of reduced pressure in the convection term and bubble generation term reads:

Table 3. The values of mean absolute error

Model	*MAE* [%]
Modified Muller-Steinhagen and Heck (MS_1) [22]	37.53
Zhang and Webb [23]	70.61
Cioncolini *et al*. [24]	63.70
Sempértegui-Tapia and Ribatski [10]	46.35
Tran [25]	49.67
Friedel [26]	33.78
Modified Muller-Steinhagen and Heck (MS_2) [21]	36.73

$$\frac{\alpha_{\mathrm{TBP}}}{\alpha_{\mathrm{LO}}}=\sqrt{\left\{\left(\phi_{\mathrm{LO}}^{2}\right)_{\mathrm{MS_1}}\left[1-\left(\frac{p_{\mathrm{sat}}}{p_{\mathrm{crit}}}\right)\right]^{a_1}+1\right\}^{n}+\frac{C}{1+\left(\frac{p_{\mathrm{sat}}}{p_{\mathrm{crit}}}\right)^{a_2}2.53\cdot10^{-3}\,\mathrm{Re}^{1.17}\,\mathrm{Bo}^{0.6}\left[\left(\phi_{\mathrm{LO}}^{2}\right)_{\mathrm{MS_1}}-1\right]^{-0.65}}\left(\frac{\alpha_{\mathrm{Pb}}}{\alpha_{\mathrm{LO}}}\right)^{2}} \quad (12)$$

where $(\phi_{\mathrm{LO}}^{2})_{\mathrm{MS_1}}$ is two-phase flow multiplier due to modified Muller-Steinhagen and Heck eq. (4). It should be also noted that there is another term in eq. (1) and the final version of that eq. (12), which should account for the fact that it may be prone to the reduced pressure. That is the pool boiling heat transfer coefficient. In the considered model the generalized Cooper [53] model is used. That model is featuring reduced pressure as one of the independent parameters. For that reason, no amendments are required to that issue. The final form of the generalized pool boiling model yields:

$$\alpha_{\mathrm{Pb}}=Aq^{2/3}M^{-0.5}p_r^{0.12}\left(-\lg p_r\right)^{-0.55} \quad (13)$$

The term A in the eq. (13) is a constant dependent on the type of refrigerant and in the case of freons this value is 55.

In order to improve the predictive compliance of the model eq. (12) with the collected database, in the first analysis approach was performed for individual groups of fluids, *i. e.* synthetic, natural and CO_2. The CO_2 is a separate group due to its specificity. However, this approach did not give satisfactory results. The conducted research has shown that the best convergence of the experimental data with the model could be obtained considering all the data without a division into a group of refrigerants. The values of the exponents a_1 and a_2 in eq. (12) were adjusted using the regression analysis and the following results were obtained a_1 = 1 and a_2 = –0.985. The results of calculations, which were obtained with taking into account the reduced pressure are presented in figs. 4-7. The information about the mean absolute error is given in tab. 4.

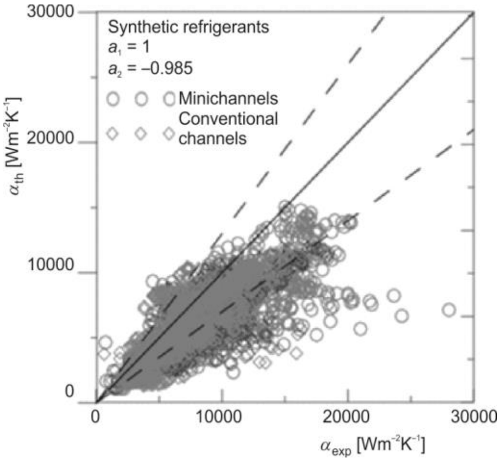

Figure 4. Comparison of the test results α_{exp} with predictions α_{th} using eq. (12) for synthetic refrigerants at a_1 = 1 and a_2 = –0.985

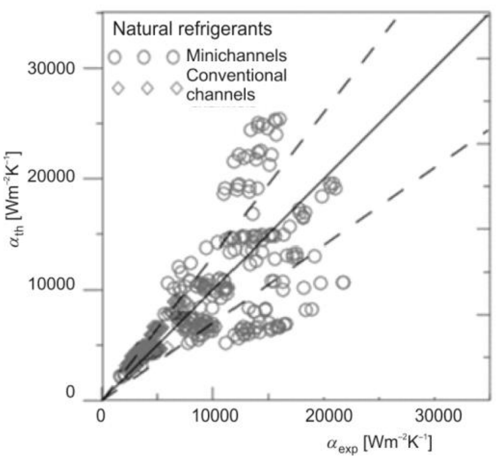

Figure 5. Comparison of the test results α_{exp} with predictions α_{th} using eq. (12) for natural refrigerants at a_1 = 1 and a_2 = –0.985

The graphs show that the correlation equations adopted for analysis reflect the experimental data in a satisfactory manner.

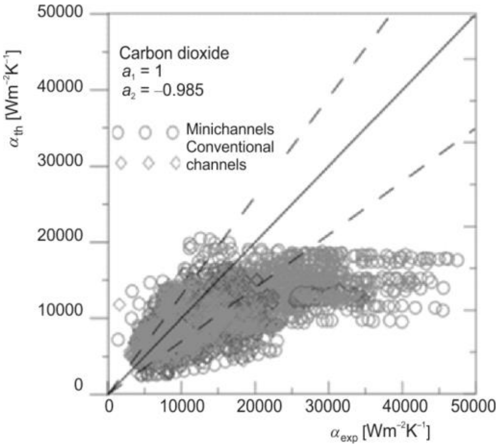

Figure 6. Comparison of the test results α_{exp} with predictions α_{th} using eq. (12) for CO_2 at $a_1 = 1$ and $a_2 = –0.985$

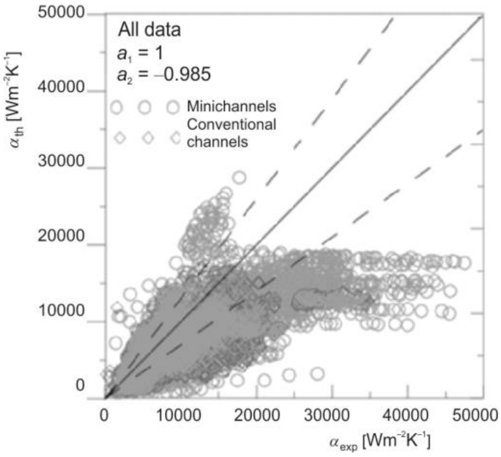

Figure 7. Comparison of the test results α_{exp} with predictions α_{th} using eq. (12) for CO_2 at $a_1 = 1$ and $a_2 = –0.985$

Table 4. The values of mean absolute error for the model described by eq. (12)

Group of refrigerants	*MAE* [%]
Synthetic	25.20
Natural	22.40
CO_2	31.14
All data	28.20

Furthermore, it should be added that in addition to reducing the average relative error, the amount of experimental data falling within the error limits of ±30% in relation to the original form of the model adopted for analysis also increased. For the case of considered group of fluids, synthetic refrigerants data it amounts to 70%, natural refrigerants data it amounts to 76% and CO_2 data it amounts to 52%. The histogram of deviations for all experimental data is presented in fig. 8.

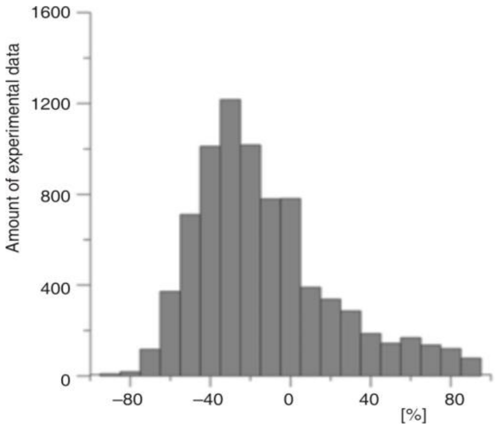

Figure 8. Histogram of deviations of the results α_{th}

Conclusions

The paper presents the analysis of the results of flow boiling calculations using the authors' own model to predict the heat transfer coefficient in a wide range of reduced pressures. The special correction has been postulated to the in-house model of flow boiling and condensation in which modified was the two-phase flow multiplier. Eight two-phase flow multiplier models were tested for this purpose, *i. e*. due to Muller-Steinhagen and Heck, modified Muller-Steinhagen and Heck, Zhang and Webb, Cioncolini *et al*., Sempertegui-Tapia and Ribatski, Tran, Friedel and finally Muller-

Steinhagen and Heck in-house modification with the effect of reduced pressure. The model has been tested against a large selection of experimental data collected from various researchers to investigate the sensitivity of the in-house developed model. The collected experimental data were conducted for the full range of quality variation and a wide range of mass velocity and saturation temperature. The results show that change of the model which describes the two-phase flow multiplier is significant in the case of CO_2, where the best compliance with experimental data obtained using the Tran correlation. The results also show that taking into account appropriate two-phase multiplier model and reduced pressure effects can significantly contribute to the convergence with experimental data compared to the original model. In the authors' opinion, the proposed method to calculate the heat transfer coefficient is a reliable tool in engineering calculations. An example of an application can be found in [54-56].

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Nomenclature

Bo – boiling number, (= q/Gh_{lg}), [–]
C – mass concentration of droplets in two-phase core, [–]
Con – confinement number, (=$1/d_h[\sigma/g(\rho_l - \rho_g)]^{1/2}$, [–]
c_p – specific heat, [$\mathrm{Jkg^{-1}K^{-1}}$]
d_h – hydraulic diameter, [m]
G – mass flux, [$\mathrm{kgm^{-2}s^{-1}}$]
h – heat transfer coefficient, [$\mathrm{Wm^{-2}K^{-1}}$]
h_{lg} – specific enthalpy of vaporization, [$\mathrm{Jkg^{-1}}$]
M – molecular weight, [$\mathrm{kgkmol^{-1}}$]
MAE – mean absolute error, [%]
Nu – Nusselt number,(= $\alpha d_h/\lambda$), [–]
P – empirical correction, [–]
p – pressure, [Pa]
q – heat flux, [$\mathrm{Wm^{-2}}$]
Re – Reynolds number, (= Gd_h/μ), [–]
T – temperature, [K]
x – quality, [–]

Greek symbols

α – heat transfer coefficient, [$\mathrm{Wm^{-2}K^{-1}}$]
σ – surface tension, [$\mathrm{Nm^{-1}}$]
λ – thermal conductivity, [$\mathrm{Wm^{-1}K^{-1}}$]
ρ – density, [$\mathrm{kgm^{-3}}$]
μ – dynamic viscosity, [Pa s]
ϕ^2_{LO} – two-phase flow multiplier, [–]

Subscripts

crit – critical
exp – experimental
g – vapor
GO – total vapour flow rate
l – liquid
LO – total liquid flow rate
Pb – pool boiling
r – reduced
sat – saturation
th – theoretical

Acronyms

ODP – Ozone Depletion Potential [–]
GWP – Global Warming Potential [–]
TBP – two-phase boiling
TP – two-phase

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