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Prediction of flow boiling heat transfer data for R134a, R600a and R290 in minichannels

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Abstract In the paper presented is the analysis of the results of calculations using a model to predict flow boiling of refrigerants such as R134a, R600a and R290. The latter two fluids were not used in the development of the model semiempirical correction. For that reason the model was verified with present experimental data. The experimental research was conducted for a full range of quality variation and a relatively wide range of mass velocity. The aim of the present study was also to test the sensitivity of developed model to a selection of the model of two-phase flow multiplier and the nonadiabatic effects. For that purpose two models have been analysed namely the one due to Müller-Steinhagen and Heck, and Friedel. In addition, the work shows the importance of taking surface tension into account in the calculation of the flow structure.

Keywords: Flow boiling; Natural refrigerants; Minichannels

Nomenclature

B	–	blowing parameter
Bo	–	boiling number
C	–	mass concentration of droplets in two-phase core
Con	–	confinement number
c_p	–	specific heat, J/kgK
d	–	diameter, m
f, f_r	–	friction factor
f_1, f_{1z}	–	function

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G	–	mass flux, kg/m^{2}s
M	–	molecular weight, kg/kmol
Nu	–	Nusselt number
P	–	empirical correction
p	–	pressure, Pa
Pr	–	Prandtl number
q	–	heat flux, W/m^2
r	–	specific enthalpy of vaporization, J/kg
R	–	two-phase multiplier
Re	–	Reynolds number
s	–	slip ratio
T	–	temperature, K
x	–	quality

Greek symbols

α	–	heat transfer coefficient, W/m^{2}K
σ	–	surface tension
λ	–	thermal conductivity, W/mK
ρ	–	density, kg/m^3
μ	–	dynamic viscosity, Pa s
$\xi = f_r/4$	–	friction factor

Subscripts

0	–	referencing case
exp	–	experimental
F	–	Friedel correlation
g, v	–	vapor
l	–	liquid
LO	–	total liquid flow rate
MS	–	Müller-Steinhagen and Heck correlation
Pb	–	pool boiling
r	–	reduced
sat	–	saturation
TBP	–	two-phase boiling
th	–	theoretical

1 Introduction

A widely used group of synthetic compounds in refrigeration technology is withdrawn from technical applications under the Montreal Protocol (1987). It is widely acknowledged that these compounds contribute to the reduction of ozone depletion in the upper atmosphere. Natural refrigerants, such as hydrofluorocarbons or hydrocarbons are likely to fully replace them in the very near future. The chlorine-free R134a, introduced in the early 1990s, is a substitute for the commonly used refrigerant R12. R134a is however a synthetic chemical compound which belongs to the group of alkyl halides. This refrigerant contains no chlorine atoms therefore it has no devastating impact on the ozone layer. A drawback in the case of that compound is the presence of fluorine. The global warming potential (GWP) for R134a is equal to 1300, which means that beyond 2030 that refrigerant, together with other synthetic substances, will be withdrawn from perspective applications. In that light the interest is directed towards natural refrigerants as replacements for traditional refrigeration fluids, such as isobutene (R600a) and propane (R290). Isobutane and propane belong to the group of so called saturated hydrocarbons. R600a and R290 do not exhibit a harmful impact on ozone layer and have a negligible global warming potential (GWP for R600a is equal 4 and for R290 is equal 20, respectively). Wongwises *et al.* found that 6/4 mixture of R290 and R600 is the most appropriate refrigerant to replace HFC134a in a domestic refrigerator [1]. Dalkilic *et al.* studied the performance analysis of alternative new refrigerant mixtures as substitute for R134a [2]. Refrigerant blend of R290/R600a (40/60 wt. %) and R 290/R1270 (20/80 wt. %) were found to be the most suitable alternative among refrigerants tested for R12. Agrawal *et al.* found that the binary mixture in the proportion of 64% and 36% of R290 and R600a found to be a retrofit or drop in substitute for R12 for use in the vapour compression refrigeration [3]. Kumar *et al.* studied the behavior of HCFC (hydrochloroflurocarbon)-123/HC-290 refrigerant mixture computationally as well as experimentally and found that refrigerant mixture 7/3 as a promising alternative to R12 system [4]. The known disadvantage of R600a and R290 is their flammability, however they are permitted for use in devices which comply with the specifications outlined in the standard EN/IEC 60335-2-24. In many households refrigerators are using R600a as working fluids without any hazard. That is due to the fact that household refrigeration appliances contain only small amounts of these refrigerants. That amount, assumed to be below 150 g in installation, is insufficient to obtain a dangerous and explosive concentration.

Experimental studies of flow boiling heat transfer of R134a in a tube with internal diameter of 3.4 mm, were carried out by Mancin *et al.* [5]. Study was conducted for constant saturation temperature equal to 30 °C, mass flux ranging from 190 to 755 kg/m^{2}s and three heat fluxes, namely 10, 25, and 50 kW/m^2.

Ong and Thome accomplished experimental studies of flow boiling of R134a for constant saturation temperature equal to 31 °C [6]. Their studies were carried out for flow boiling in the channel with internal diameter of 1.030 mm. The studied range of parameters was in the case of mass flux from 200 to 1600 kg/m^{2}s and heat flux from 2.3 to 250 kW/m^2. Other studied fluids were R236fa and R245fa.

Tibiriçá i Ribatski conducted the experimental study for flow boiling of R134a and R245fa in a tube with internal diameter of 2.3 mm [7]. Study was conducted for mass flux ranging from 50 to 700 kg/m^{2}s, heat flux from 5 to 55 kW/m^2 and for three saturation temperatures, namely 22, 31, and 41 °C. The results were compared with empirical correlations due to Satioh *et al.* [8], Zhang *et al.* [9], Kandlikar and Balasubramanian [10], Bertsch *et al.* [11], Thome *et al.* [12], Sun and Mishima [13], Lazarek and Black [14], Tran *et al.* [15], and Kew and Cornell [16]. The results showed that the best agreement with experimental data was obtained for the case of Satioh *et al.*, and Sun and Mishima. In the case of these correlations the mean absolute deviation of 21.2% and 20.6% was found.

Comparison between experimental and theoretical studies has also been carried out by Copetti *et al.* [17]. They conducted their research of flow boiling heat transfer of R600a and R134a in tubes with the internal diameter of 2.6 mm. Study was conducted for saturation temperature equal 22 °C. Studied were also two levels of mass flux, namely 240 and 440 kg/m^{2}s. Heat flux was varied from 44 to 95 kW/m^2. The results were compared with empirical correlations due to Kandlikar and Balasubramanian [10], Zhang *et al.* [9], Satioh *et al.* [8], Choi *et al.* [18] and Bertsch *et al.* [11]. The results showed that the best agreement with experimental data were obtained for the case of Kandlikar and Balasubramanian, and Bertsch *et al.*

Choi *et al.* [19] conducted the experimental study for flow boiling of R290, R717 and R1234yf in tubes with internal diameter of 1.5 and 3 mm. The results have been obtained by tests conducted for mass flux ranging from 50 to 600 kg/m^{2}s, heat flux from 5 to 60 kW/m^2 and saturation temperature from 0 to 10 °C.

Comparison between experimental and theoretical studies for R290 has also been carried by Wang *et al.* [20]. Their studies were carried out for flow boiling of R290 in the channels with internal diameter of 6 mm. The studied range of parameters was in the case of mass flux from 62 to 104 kg/m^{2}s, heat flux from 11.7 to 87.1 kW/m^2 and saturation temperature from −35 to −1.9 °C. These experimental data were compared with correlations due to Kandlikar [10], Benett and Chen [21] and Shah [22].

In literature there are many empirical correlations for modeling of boiling heat transfer. Some of them have been mentioned in the introduction. Moreover, it can be seen that correlations mentioned in the introduction, in case of a fluid such as R134a, R600a and R290 prove a good agreement with experimental data. Several recent publications, for example by Ribatski [23], Tibiriçá and Ribatski [24], Sardeshpande and Ranade [25], Kandlikar [26] and Alagesan [27] analyze the experimental data for validation of heat transfer coefficient predictions using the correlations available in literature. These analyzes do not take into account unfortunately the method developed earlier by one of the the authors [28,29]. It is the authors intention to show the performance of this approach and this work is concentrated on the predictions of heat transfer coefficient for three refrigerants, namely R134a, R600a and R290, where the latter two are perspective ones for replacement of R134a.

In the paper the analysis is presented from the point of view of possible replacement of R134a with other natural refrigerants, considered here through R600a and R290. The performance of the model enabling determination of the two-phase flow heat transfer coefficient, developed earlier by one of the authors, has been tested on refrigerants such as R134a, R600a and R290. The latter two fluids were not used in the development of the model semi-empirical correction. The model was verified on some recent data due to Mancin *et al.*, Tibiriçá and Ribatski, Ong and Thome, Copetti *et al.*, Lu *et al.* [30], Choi *et al.* and Wang *et al.*. The experimental research was conducted for a full range of quality variation and a relatively wide range of mass velocity. Another aim of present study was to test the sensitivity of developed earlier model to a selection of the model of two-phase flow multiplier and the nonadiabatic effects. For that purpose two models have been analysed namely the one due to Müller-Steinhagen and Heck, and Friedel. In addition, the work shows the importance of taking into account in the calculation of the flow structure surface tension.

2 The model

The correlation due to Mikielewicz [28,29] has been tested for a significant number of experimental data [29], returning satisfactory results for the case of the flow boiling process and also the flow condensation. Below a brief recapitulation on the model is given. The model [28,29] has been developed for the case of flow boiling on the basis of consideration of dissipation energy in the flow and recently modified to its final form by Mikielewicz *et al.* [28].

The fundamental hypothesis in the model under scrutiny here is the fact that heat transfer in flow boiling with bubble generation, regarded as an equivalent flow of liquid with the 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 the energy dissipation due to shearing flow without the bubbles and dissipation resulting from the bubble generation. A final version of the model [28] is

$$\frac{\alpha_{TBP}}{\alpha_{LO}} = \sqrt{R_{MS}^n + \frac{C}{1+P}\left(\frac{\alpha_{Pb}}{\alpha_{LO}}\right)^2} \,. \tag{1}$$

In Eq. (1) α_{LO} is the heat transfer coefficient for the liquid only case. Constant $C = 1$ for flow boiling whereas $C = 0$ for flow condensation. In the case of turbulent flow α_{LO} may be determined using for example the Dittus-Boelter equation, i.e. $\mathrm{Nu} = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}$. In the model (1) introduced is the empirical correction P and a modified two-phase multiplier due to Müller-Steinhagen and Heck R_{MS}. The modified form of the two-phase multiplier R_{MS} is described by the following equation [29]:

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

It should be noted that present in Eq. (1) two-phase multiplier R_{MS} is raised to the power n, where $n = 0.9$ for turbulent flows, and $n = 2$ for laminar flows [29]. In Eq. (2) the exponent $m = 0$ for the case of the flow in conventional size channels and $m = -1$ for minichannels. Functions f_1 and f_{1z} in Eq. (1) denote the ratio of the pressure drop in liquid only flow to gas only flow. For the case of turbulent flow the function assumes the form: $f_1 = (\mu_l/\mu_g)^{0.25}(\rho_g/\rho_l)$. Function f_{1z} denotes the ratio of heat transfer coefficient for liquid only flow to the heat transfer coefficient for gas only flow and is introduced to Eq. (1) to meet the limiting conditions, i.e., for $x = 0$ the correlation (1) should reduce to a value of heat transfer coefficient for liquid, whereas for $x = 1$, to that for the gas respectively. In case of turbulent flow, when the Dittus-Boelter correlation is incorporated $f_{1z} = (\mu_g/\mu_l)(c_{pl}/c_{pg})(\lambda_l/\lambda_g)^{1.5}$. The form of empirical correction P in Eq. (1), must be calculated using following equation:

$$P = 2.53 \times 10^{-3}\,\mathrm{Re}^{1.17}\,\mathrm{Bo}^{0.6}\,(R_{MS} - 1)^{-0.65} \,. \tag{3}$$

For the calculation of the pool-boiling heat transfer coefficient α_{Pb} (1), it is recommended to use a generalized model due to Cooper [31]. This model describes the heat transfer coefficient in the fluid in terms of the reduced pressure, molecular weight and applied wall heat flux. The equation describing the pool-boiling heat transfer coefficient has the form:

$$\alpha_{Pb} = A\, p_r^{0.12} \left(-\log p_r\right)^{-0.55} M^{-0.5}\, q^{\frac{2}{3}} \,. \tag{4}$$

First modification of Eq. (1) which was recently devised was the change of the definition of two-phase multiplier to incorporate the effects associated with the applied wall heat flux, Mikielewicz [32]. In this approach the two-phase multiplier is denoted as R_{TBP}, and the relationship which describes the modifications has the form [32]

$$R_{TPB} = \begin{cases} R_{MS}\left(1-\frac{B}{2}\right) & \text{for annular flow boiling}\; 0 \le x \le 1\,, \\ R_{MS}\sqrt{1+\left(\frac{8\,\alpha_{Pb} d}{\lambda Re \mathrm{Pr} \xi_0 R_{MS}}\right)^2} & \text{for other flow structures}\; 0 \le x < 0.1\,. \end{cases} \tag{5}$$

In Eq. (5) the two-phase multiplier R_{MS} should be calculated using the modified Müller-Steinhagen and Heck correlation, described by the Eq. (2). The blowing parameter B occurring in Eq. (9) is defined as follows [32]:

$$B = \frac{2q\frac{\rho_l}{\rho_v}}{f_r G(s-1)r} \,. \tag{6}$$

In Eq. (6) s is the slip ratio, which can be determined from the relationship by Zivi [29]

$$s = \sqrt[3]{\frac{\rho_l}{\rho_v}} \,. \tag{7}$$

As a result of correction (5), a modified model is obtained, which was adopted for consideration in the present work, which reads:

$$\frac{\alpha_{TBP}}{\alpha_{LO}} = \sqrt{R_{TBP}^n + \frac{C}{1+2.53\times10^{-3}\,\mathrm{Re}^{1.17}\,\mathrm{Bo}^{0.6}\left(R_{MS}-1\right)^{-0.65}}\left(\frac{\alpha_{Pb}}{\alpha_{LO}}\right)^2} \,. \tag{8}$$

In the subsequent analyzes, the two-phase flow multiplier was also considered by application of the Friedel correlation [33]. That is a method, which could be more appropriate for modeling of the flow resistance of isobutane and propane. According to this method the two-phase multiplier R_F can be determined in the following manner:

$$R_F = E + \frac{3.24\, FH}{\mathrm{Fr}^{0.045}\,\mathrm{We}^{0.035}} \tag{9}$$

The terms E, F, H are defined as follows:

$$E = (1-x)^2 + x^2 \left(\frac{\rho_l f_v}{\rho_v f_l}\right) , \tag{10}$$

$$F = x^{0.78} (1-x)^{0.2224} , \tag{11}$$

$$H = \left(\frac{\rho_l}{\rho_v}\right)^{0.91} \left(\frac{\mu_v}{\mu_l}\right)^{0.19} \left(1 - \frac{\mu_v}{\mu_l}\right)^{0.7} . \tag{12}$$

After taking into account the expressions for another definition of the two-phase flow multiplier, namely Eqs. (9) and (10), the model, which was used to calculate the flow boiling heat transfer of carbon dioxide can be written in the form

$$\frac{\alpha_{TBP}}{\alpha_{LO}} = \sqrt{R_F^n + \frac{C}{1 + 2.53 \times 10^{-3} \,\mathrm{Re}^{1.17}\, \mathrm{Bo}^{0.6} (R_{MS} - 1)^{-0.65}} \left(\frac{\alpha_{Pb}}{\alpha_{LO}}\right)^2} . \tag{13}$$

In the following part, the basic model and its subsequent modifications, which have been selected for discussion will be analysed with respect to predictions of heat transfer. These models are denoted as: model I – Eq. (1), model II – Eq. (8), and model III – (Eq. (13).

3 The results

Analysis presented in the following parts is conducted on the basis of available experimental data from literature due to Mancin *et al.* [5], Ong and Thome [6], Tibiriçá and Ribatski [7], Lu *et al.* [30], Copetti *et al.* [17] for R14a, Copetti *et al.* [17] for R600a and Choi *et al.* [19], Wang *et al.* [20] for R290. Table 1 shows the characteristic range of parameters analyzed in the experiments.

Analysis of the parameters from Tab. 1 indicates the fact that in most cases the experimental research was conducted for a full range of quality variation and a relatively wide range of mass velocity.

One of the concepts of distinguishing between the minichannels and conventional size channels is due to Kandlikar's [35] systematization of channel sizes with respect to the diameter. It is as follows:

- Conventional channels: hydraulic diameters greater than 3 mm.
- Minichannels: hydraulic diameters to range of 600 μm – 3 mm.

Table 1: The range of variation of experimental data for flow boiling of R134s, R600a and R290.

The author data	Refrigerant	T_{sat} [°C]	x [–]	d [mm]	G [kg/m^{2}s]	q [kW/m^2]
Mancin *et al.* [5]	R134a	30	0–1	3.4	190–775	10 25 50
Ong and Thome [6]	R134a	31	0–0.9	1.030	200 400 1200	21.5–111.3
Tibiriçá and Ribatski [7]	R134a	22 31 41	0–1	2.3	100–600	5–35
Copetti *et al.* [17]	R134a	22	0–0.75	2.6	240 440	44
Lu *et al.* [30]	R134a	10	0–0.9	3.9	200 300 400	5.67 11.35 18.91
Copetti *et al.* [17]	R600a	22	0–0.7	2.6	240 440	95 63 44
Choi *et al.* [19]	R290	0 10 11	0–1	1.5 3	150 200 250	5 10 15
Wang *et al.* [20]	R290	-35 -14.1 -1.9	0–1	6	63.9–102.8	11.7 33.6 53.2

- Microchannels: hydraulic diameters to range of 50 m – 600 μm.

In accordance with that systematization the data due to experimental data of Mancin *et al.* [5], Lu *et al.* [30], and Wang *et al.* [20] correspond to flow boiling in conventional channels.

Different concept of distinguishing between conventional channels and minichannels was suggested by Kew and Cornwell [16]. The transition threshold from conventional size channels to minichannels is based on the physical mechanism and not only based on the hydraulic diameter. Their criterion is based on the so called confinement number Con, defined as

$$\mathrm{Con} = \frac{1}{d}\sqrt{\frac{\sigma}{d(\rho_l - \rho_v)}} \,. \tag{14}$$

When the confinement number Con is greater than 0.5 then the two-phase flow has the properties of the flow in minichannel, in which the surface tension plays a dominant role. The values of Con number for the fluids considered in the present study are presented in Tab. 2. According to the division that Con greater than 0.5 corresponds to minichannels it can be concluded that experimental data due to Ong and Thome [6] and Choi *et al.* [19] for internal diameter equal to 1.5 mm belong to the concept of minichannel size, whereas other data belong to the conventional size channels.

Table 2: The confinement number Con for experimental data.

The author data	Mancin *et al.* [5]	Ong and Thome [6]	Tibiriçá and Ribatski [7]	Copetti *et al.* [17]	Lu *et al.* [30]	Copetti et al. [17]	Choi *et al.* [19]		Wang *et al.* [20]
Refrigerant	R134a	R134a	R134a	R134a	R134a	R600a	R290		R290
d [mm]	3.4	1.030	2.3	2.6	3.9	2.6	1.5	3	6
Con	0.2369	0.782	0.3709-0.3256	0.3281	0.234	0.4711	0.9412-0.8898	0.4706-0.4473	0.2717-0.2505

Figures 1 to 6 show the results of calculations of heat transfer coefficient for considered refrigerants such as R134a, R600a, R290 using three mentioned earlier models based on Eqs. (1), (8), and (13). In the applied model formulation the surface tension effects were included into the analysis by considering the effect of the confinement number in equations through Con. Hence the version of the model applicable to minichannels was used.

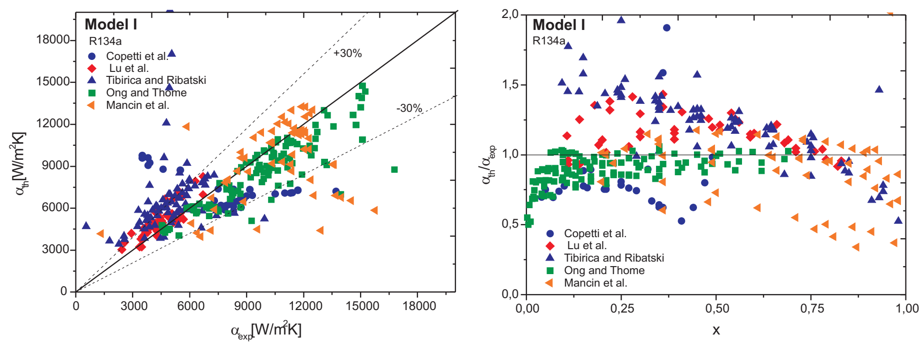

Figure 1: Comparison of test results, α_{exp}, [5,6,7,17,30] with predictions obtained using Eq. (1), α_{th}.

Figures 7 to 12 show the results of calculations of heat transfer coefficient for flow boiling of R134a, R600a and R290 using the version of models I to

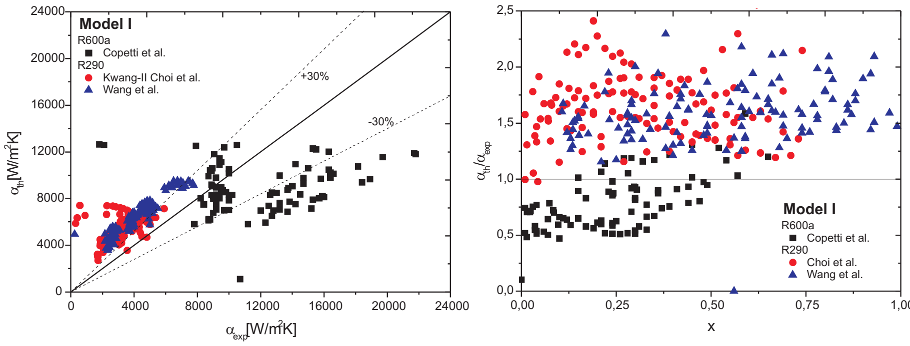

Figure 2: Comparison of test results, α_{exp}, [17,19,20] with the predictions obtained using Eq. (1), α_{th}.

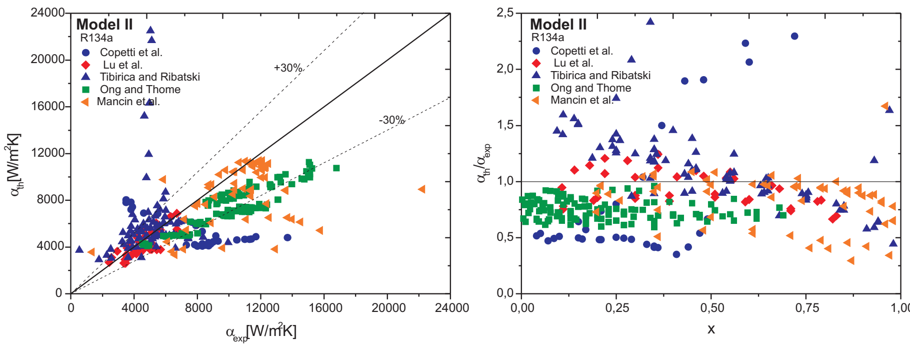

Figure 3: Comparison of test results, α_{exp}, [5–7,17,30] with the predictions obtained using Eq. (8), α_{th}.

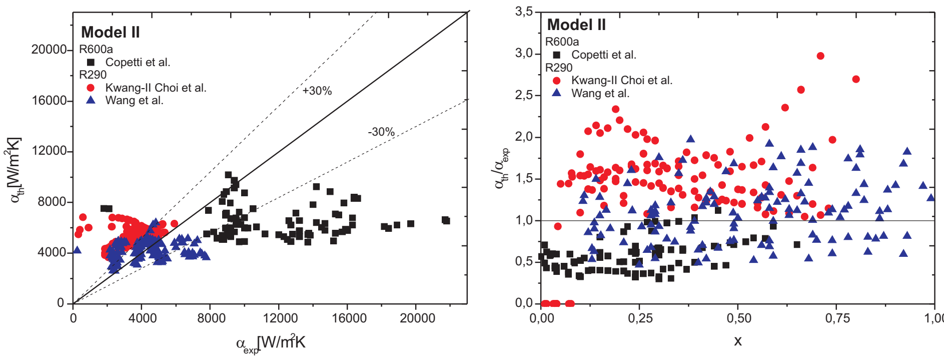

Figure 4: Comparison of test results, α_{exp}, [17,19,20] with the predictions obtained using Eq. (8), α_{th}.

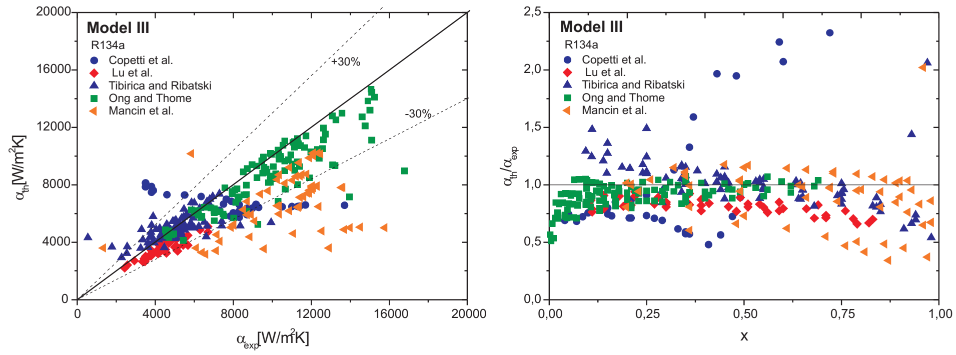

Figure 5: Comparison of test results, α_{exp}, [5–7,17,30] with the predictions obtained using Eq. (13), α_{th}.

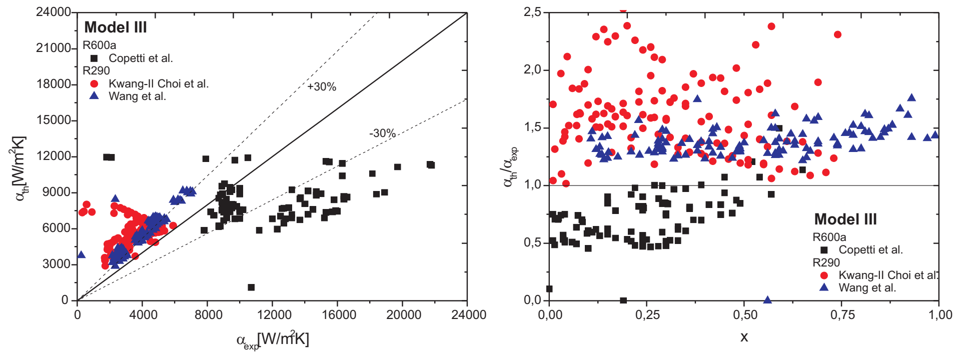

Figure 6: Comparison of test results, α_{exp}, [17,19,20] with the predictions obtained using Eq. (13), α_{th}.

III, where the surface tension was not included into the analysis. Hence the version of the model relevant to conventional channels was applied. This is equivalent to setting the exponent at the Con in Eq. (2) equal to zero. That is also the reason why data due to Ong and Thome and Choi *et al.* for tubes with internal diameter of 1.5 mm was not considered here, as the data can be regarded as being that for minichannels.

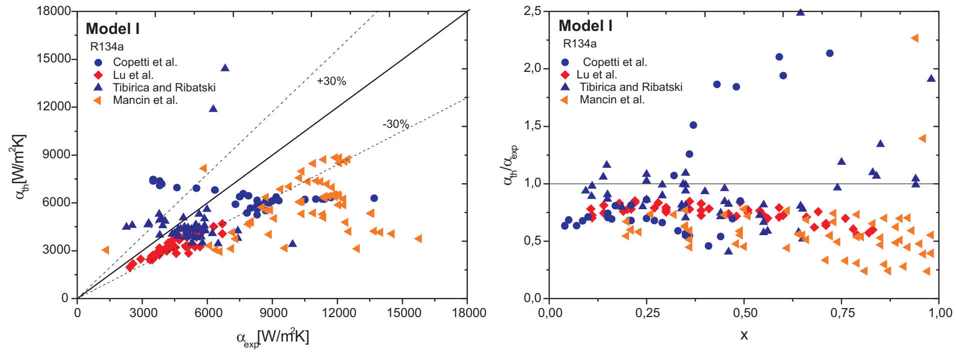

Figure 7: Comparison of test results, α_{exp}, [5,7,17,30] with the predictions obtained using Eq. (1).

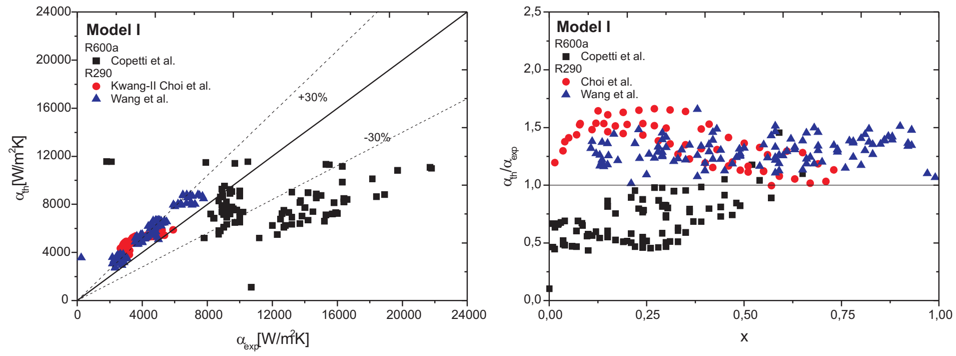

Figure 8: Comparison of test results, α_{exp}, [17,19,20] with the predictions obtained using Eq. (1), α_{th}.

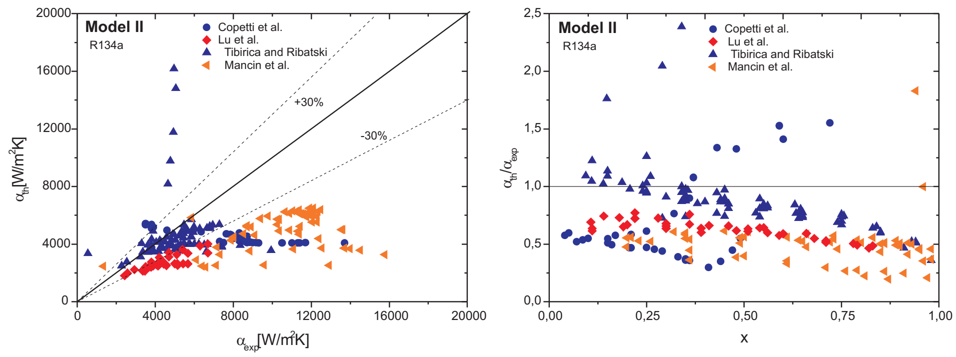

Figure 9: Comparison of test results, α_{exp}, [5,7,17,30] with the predictions obtained using Eq. (8), α_{th}.

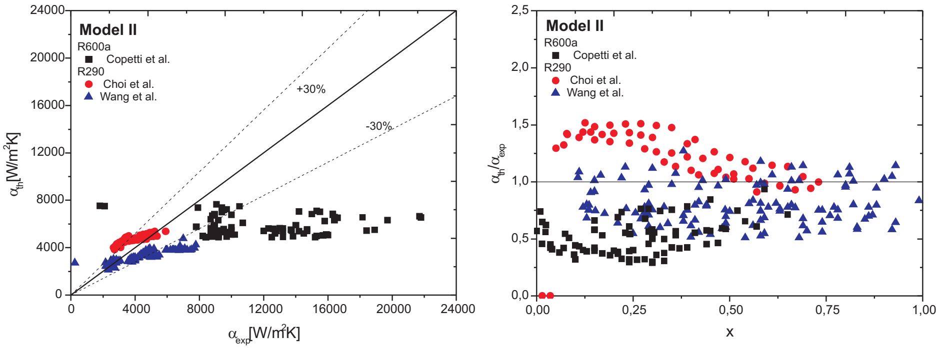

Figure 10: Comparison of test results, α_{exp}, [17,19,20] with the predictions obtained using Eq. (8), α_{th}.

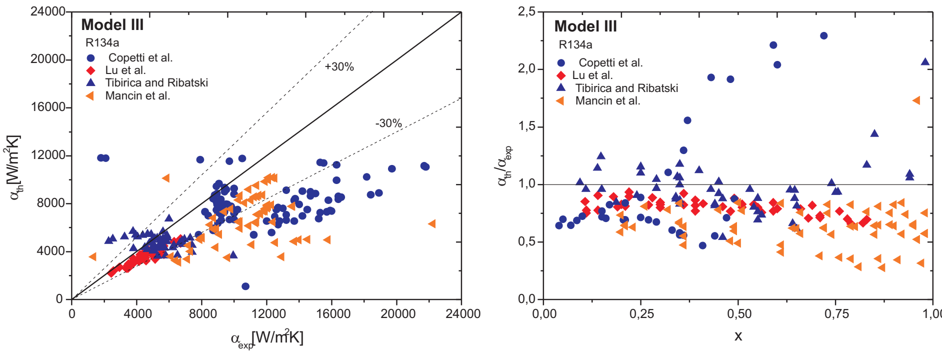

Figure 11: Comparison of test results [5,7,17,30] with the predictions obtained using Eq. (13), α_{th}.

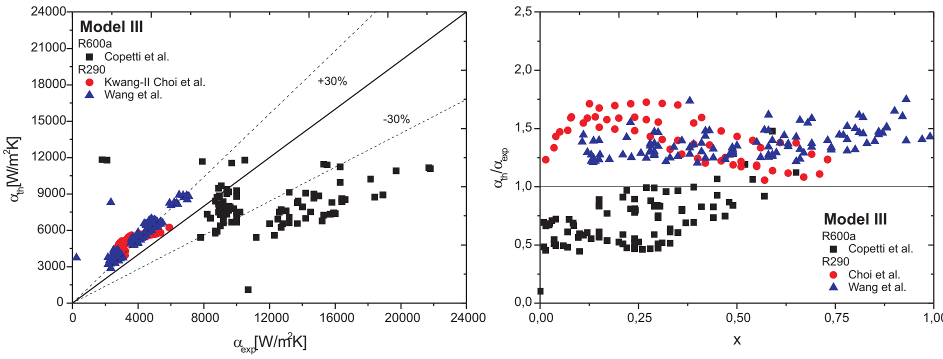

Figure 12: Comparison of test results, α_{exp}, [17, 19, 20] with the predictions obtained using Eq. (13), α_{th}.

Table 3: The mean absolute deviation for the considered experimental data.

The mean absolute deviation [%]			Model I	Model II	Model III
Mancin *et al.* [5]	R134a	minichannel	21.4	22.8	40.2
		conventional channel	46.7	52.9	40.4
Ong and Thome [6]	R134a	minichannel	11.9	11.1	25.2
Tibiriçá and Ribatski [7]	R134a	minichannel	33.4	29.8	21.6
		conventional channel	23.8	26.4	21.6
Copetti *et al.* [17]	R134a	minichannel	28.6	52.1	34.2
		conventional channel	37.7	47.0	35.4
Lu *et al.* [30]	R134a	minichannel	17.31	25.8	19.1
		conventional channel	14.3	37.2	19.6
Copetti *et al.* [17]	R600a	minichannel	28.2	47.7	30.3
		conventional channel	32.7	57.5	31.5
Choi *et al.* [19]	R290	minichannel	56.3	47.5	51.9
		conventional channel	35.3	25.4	41.4
Wang *et al.* [20]	R290	minichannel	57.0	31.6	38.9
		conventional channel	30.0	22.6	31.6

Based on the presented results of calculations, which were obtained using the expressions (1), (8), and (13), it can be seen that the agreement between the model predictions and the experimental data for 134a is very good, especially for the case of calculations obtained using the version of the model applicable to minichannels. In case of R600a and R290 the agreement is worse and is dependent on the kind of used model. The selection of the appropriate flow resitstance model is important and requires further study. At the moment it is not certain which two-phase flow multiplier model is the most appropriate to simulate the relevant phenomena in the flows of hydrocarbons such as R290 and R600a. In case of calculations for R134a the best agreement with experimental data is obtained for Ong and Thome data if model I and model II are applied. In these cases the mean absolute deviation for model I is equal to 11.9% and for model II is equal to 11.1%, respectively, see Tab. 3. In case of other R134a data the agreement is slightly less satisfactory. In case of the analysis of data for R600a the best predictions are obtained using the model I with the experimental data due to Copetti *et al.* In this case the mean absolute deviation is 28.2%. The worst agreement with the experimental data by Choi *et al.* and Wang *et al.* is obtained using the model I, in which the surface tension effects have been included. In this case the mean absolute deviation for the model I is about 57%, while the best agreement with the all experimental data for R290 obtained using the model II, in which the surface tension effects have been omitted. In this case the mean absolute deviation is 25.4% for the experimental data by Choi *et al.* and 22.6% for experimental data by Wang *et al.*

4 Conclusions

The paper presents the analysis of the results of calculations using a model developed earlier by one of the authors to study flow boiling of different refrigerants. In the paper such refrigerants as R134a, R600a and R290 were examined. The results of the calculations show satisfactory compliance with experimental data. There is an outstanding issue of the appropriate flow resistance model to be recommended in calculations of flow boiling of hydrocarbons as the Müller-Steinhagen and Heck model, recommended for other refrigerants, is not performing better than the model due to Friedel. Calculations for R290 are very prone to the selection of the appropriate version of the correlation, namely with or without surface tension effects. It can be concluded that the all models described by (1), (8) and (13) can be used for the purposes of the engineering calculation of heat transfer coefficient for flow boiling of refrigerants such as R134a, R600a and R290.

In general it can be concluded that the form of the semiempirical correction present in the considered model requires another scrutiny. Experimental data bear statistical errors between different facilities and exhibit a reduced pressure dependence. Works will be continued in that direction.

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