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Modelling of heat transfer during flow condensation of natural refrigerants under conditions of increased saturation pressure

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Abstract

The paper presents a modified in-house model for calculating heat transfer coefficients during flow condensation, which can be applied to a variety of working fluids, but natural refrigerants in particular, at full range thermodynamic parameters with a particular focus on increased saturation pressure. The modified model is based on a strong physical basis, namely the hypothesis of analogy between the heat transfer coefficient and pressure drop in two-phase flow. The model verification is based on a consolidated database that consists of 1286 data points for 7 natural refrigerants and covers the reduced pressure range (the ratio of critical pressure and saturation pressure) from 0.1 to 0.8 for different mass velocities and diameters. The new version of the in-house model, developed earlier by Mikielewicz, was compared with 4 other mathematical models widely recommended for engineering calculations and obtained the best consistency results. The value of the mean absolute percentage error was 28.13% for the modified model, the best result among the scrutinised methods.

Keywords: Condensation, Heat transfer coefficient; Reduced pressure; Saturation pressure; Saturation temperature

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1. Heat transfer during flow condensation in trending application

There is a gap in knowledge about condensation at elevated saturation temperatures and corresponding high reduced pressures. The condensation process in high-temperature heat pumps occurs usually at temperatures higher than 80°C. Most of the existing experimental data has been collected for temperatures below 40°C, which is related to HVAC (heating, ventilation, air conditioning) applications of refrigerants. For temperatures higher than 120°C, most low boiling fluids operate at parameters close to the thermodynamic critical point, where there are significant changes in density and viscosity of the liquid and vapour phases, which has a significant impact on interfacial phenomena. Over the past century, there have been many papers that have dealt with the modelling of the heat transfer coefficient during condensation. Most of the literature models are created for specific experimental parameters and are not intended to be applied to a wide range of refrigerants. In recent years, research on high-temperature heat pumps where saturation temperatures and saturation pressures are much higher than in classical applications have become important [1]. The use of new environmentally neutral refrigerants is also principal. For this reason,

Nomenclature

c_p – specific heat capacity, J/(kg K)
Con – confinement number
d – inner diameter od tube, m
g – gravitational acceleration, m/s^2
G – total mass flux, kg/(m^{2}s)
h_{lv} – latent heat of evaporation, J/kg
Jg – dimensionless vapour velocity, $Jg = \frac{xG}{\sqrt{gd\rho_v(\rho_l-\rho_v)}}$
Nu – Nusselt number
P – pressure, N/m^2
Pr – reduced pressure, $Pr = P_{sat}/P_{crit}$
Pra – Prandtl number
Re – Reynolds number
T – temperature, °C
v – specific volume, m^3/kg
x – vapour quality
Xtt – Martinelli's correlating parameter, $Xtt = \left[\frac{\mu_l}{\mu_v}\right]^{0.1}\left[\frac{1-x}{x}\right]^{0.9}\left[\frac{\rho_v}{\rho_l}\right]^{0.5}$

Greek symbols

α – heat transfer coefficient, W/(m^{2}K)
μ – dynamic viscosity, Pa s
ρ – density, kg/m^3
λ – thermal conductivity, W/(m K)
σ – surface tension, N/m

Subscripts and Superscripts

crit – critical
l – liquid
LO – liquid only
sat – saturation
tf – two phase
v – vapour
VO – vapour only

Abbreviations and Acronyms

MAPE – mean absolute percentage error
RMS – two phase multiplier due to Muller-Steinhagen and Heck

natural refrigerants such as isobutane or propane are gaining popularity in heat pump applications and others. Condensation heat transfer coefficient prediction is also important for the potential application of advanced thermodynamic cycles [2]. Some novel cycles require modelling of such complex phenomena as condensation inside spray ejector [3].

2. Methods of calculating the heat transfer coefficient during condensation

In [4] model for the calculation of heat transfer coefficient during condensation within channels has been published, hereinafter named an in-house model. The method was based on a correlation originally developed for flow boiling. The basis for the analysis involved predicting heat transfer coefficients in flow condensations and assessing them against correlations developed for annular flow structures. In this context, flow boiling and flow condensation were treated as symmetrical phenomena. The comparisons yielded satisfactory results. Additionally, the study examined nonadiabatic effects in the heat transfer coefficient model, incorporating the blowing parameter into the modelling process. The modifications were found to have significant effects on condensation for small qualities, affirming that nonadiabatic effects are less critical for higher qualities. Similarly, in the case of flow boiling, the most substantial changes to the heat transfer coefficient occurred for smaller qualities. Later in [5] a method for calculation of heat transfer coefficient during flow boiling was proposed to take into account the reduced pressure effect. In this paper, a modified Muller-Steinhagen and Heck two-phase multiplier was introduced to consider the effects of increased reduced pressures. Promising results have been obtained, however there was still room left for further modifications. In [6] authors presented the newest version of two phase pressure drop model based on the Müller-Steinhagen and Heck method.

The outcomes of modelling focused on heat transfer and pressure drop during the condensation of R134a and R404A refrigerants in minichannels have been presented in [7]. Internal diameters ranged from 0.31 mm to 3.30 mm. The study specifically examines the local heat transfer coefficient and pressure drop in individual minichannels, with a comparative analysis against methods proposed by other researchers. Drawing from the measured data, the authors introduced their own correlation for calculating the local heat transfer coefficient.

The study [8] introduces a novel and straightforward model for determining the local heat transfer coefficient during flow condensation within plain pipes. The proposed model accounts for two distinct regimes distinguished by the value of mass flux. For both regions, a new correlation is presented, resembling the single-phase heat transfer coefficient model. Corresponding Reynolds and Prandtl numbers are based on the sum of superficial liquid and vapour versions. These models emphasize the significant role of the superficial vapour Reynolds number in governing the heat transfer coefficient. The model demonstrates predictive capability for estimating the heat transfer coefficient across a range of channels including microchannels and conventional channels, encompassing various refrigerants.

Shah [9] presented a straightforward dimensionless correlation designed for predicting heat-transfer coefficients in the context of film condensation inside pipes. The dataset encompasses various fluids such as water, R-11, R-12, R-22, R-113, methanol, ethanol, benzene, toluene, and trichloroethylene undergoing condensation in pipes of horizontal, vertical, and inclined orientations with diameters ranging from 7 mm to 40 mm. Additionally, a dataset for condensation within an annulus has been analysed. The parameter range covered includes reduced pressures spanning from 0.002 to 0.44, saturation temperatures ranging from 21°C to 31°C and vapour velocities from 3 m/s to 300 m/s, and vapour qualities from 0% to 100%. A new version [10] of correlation extended for a wider range of parameters was published in 2009. The presented method has been shown to be in

Table 1. Formulas of analysed methods.

Model	Formula	Purpose
Mikielewicz [4,5]	$\frac{\alpha_{TPB}}{\alpha_{LO}} = \sqrt{R_{MS}^{n}}$ $R_{MS} = \left[1 + 2\left(\frac{1}{f_1} - 1\right) x\mathrm{Con}^{m}\right](1-x)^{\frac{1}{3}} + x^{3}\frac{1}{f_{1z}}$ Turbulent flow: $f_1 = \frac{\left(\frac{\rho_l}{\rho_v}\right)}{\left(\frac{\mu_l}{\mu_v}\right)^{0.25}}$, $f_{1z} = \frac{\left(\frac{\mu_v}{\mu_l}\right)}{\left(\frac{\lambda_l}{\lambda_v}\right)^{1.5}}\left(\frac{c_{pl}}{c_{pv}}\right)$ Laminar flow: $f_1 = \frac{\frac{\rho_l}{\rho_v}}{\frac{\mu_l}{\mu_v}}$, $f_{1z} = \left(\frac{\lambda_v}{\lambda_l}\right)$	General correlation for condensation inside conventional and mini/ microtubes
Bohdal [7]	$\mathrm{Nu} = 25.084\mathrm{Re}_l^{0.258}\mathrm{Pra}_l^{-0.495}Pr^{-0.288}\left(\frac{x}{1-x}\right)^{0.266}$	Condensation in mini/ microchannels
Dorao and Fernandiono [8]	$\mathrm{Nu}J = 0.023\mathrm{Re}_{tf}^{0.8}\mathrm{Pra}_{tf}^{0.3}$ $\mathrm{Nu}JJ = 41.5d^{0.6}\mathrm{Re}_{tf}^{0.4}\mathrm{Pra}_{tf}^{0.3}$ $\mathrm{Nu} = \sqrt[9]{(\mathrm{Nu}J^9 + \mathrm{Nu}JJ^9)}$	General correlation for condensation in tubes
Shah 1979 [9]	$\alpha = 0.023\mathrm{Pra}_l^{0.4}\mathrm{Re}_l^{0.8}\frac{(1-x)^{0.8} + [3.8x^{0.76}(1-x)^{0.04}]}{Pr^{0.38}}\frac{\lambda_l}{d}$ $\mathrm{Re}_l > 3000$	Correlation for condensation in tubes for annular flow
Shah 2009 [10]	$\alpha = hi$ for $Jg \geq JgC$ $\alpha = hi + h_{\mathrm{Nu}}$ for $Jg < JgC$ $hi = 0.023\mathrm{Re}_l^{0.8}\mathrm{Pra}_l^{0.4}\left(\frac{\mu_l}{14\mu_v}\right)^{n2}(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{Pr^{0.38}}\frac{\lambda_l}{d}$ $n2 = 0.0058 + 0.557Pr$ $h_{\mathrm{Nu}} = 1.32\mathrm{Re}_l^{-\frac{1}{3}}\left[\frac{\rho_l(\rho_l-\rho_v)gk_l^3}{\mu_l^2}\right]^{1/3}$ $Z = \left(\frac{1}{x-1}\right)^{0.8}Pr^{0.4}$ $JgC = 0.98(Z+0.263)^{-0.62}$ $Jg = \frac{xG}{\sqrt{gD\rho v(\rho_l-\rho_v)}}$	General correlation for condensation in plain tubes
Shah 2019 [11]	$\alpha = hi$ for $Jg \geq JgC$, $Frl > 0.012$ and $Wef \geq 100$ $\alpha = hi + h_{\mathrm{Nu}}$ for $Jg < JgC$, $Frl > 0.012$ for other regimes $h = hII + h_{\mathrm{Nu}}$ $hi = 0.023\mathrm{Re}_l^{0.8}\mathrm{Pra}_l^{0.4}\left(\frac{\mu_l}{14\mu v}\right)^{n2}(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{Pr^{0.38}}\frac{\lambda_l}{d}$ $n2 = 0.0058 + 0.557Pr$ $hII = 0.023\mathrm{Re}_l^{0.8}\mathrm{Pra}_l^{0.4}(1 + 1.128x^{0.8170}\frac{\rho_l}{\rho_v}^{0.3685}\frac{\mu_l}{\mu_v}^{0.2363}\frac{1-\mu_v}{\mu_l}^{2.144}\mathrm{Pra}_l^{-0.1}\frac{\lambda_l}{d}$ $h_{\mathrm{Nu}} = 1.32\mathrm{Re}_l^{-\frac{1}{3}}\left[\frac{\rho_l(\rho_l-\rho_v)gk_l^3}{\mu_l^2}\right]^{1/3}$ $Z = \left(\frac{1}{x-1}\right)^{0.8}Pr^{0.4}$ $JgCC = 0.95(1.254 + 2.27Z^{1.249})^{-1}$ $JgC = 0.98(Z+0.263)^{-0.62}$ $Jg = \frac{xG}{\sqrt{gD\rho v(\rho_l-\rho_v)}}$	General correlation for condensation in conventional and mini/micro tubes
Cavallini 2006 [12]	$J_G = \frac{xG}{\sqrt{gD\rho v(\rho_l-\rho_v)}}$ $J_G^T = \sqrt[-3]{\left(\frac{7.5}{4.3Xtt^{1.111}}\right)^{-3} + CT^{-3}}$ For hydrocarbons: CT = 1.6 ; for other refrigerants: CT = 2.6 Modified Dittus-Boelter correlation: $\alpha_{LO} = 0.023\mathrm{Re}_l^{0.8}\mathrm{Pra}_l^{0.4}\frac{\lambda_l}{d}$ ΔT-independent flow regime ($J_G > J_G^T$) $\alpha_A = \alpha_{LO}\left[1 + 1.128x^{0.817}\left(\frac{\rho_l}{\rho_v}\right)^{0.3685}\left(\frac{\mu_l}{\mu_v}\right)^{0.2363}\left(1-\frac{\mu_v}{\mu_l}\right)^{2.144}\mathrm{Pra}_l^{-0.1}\right]$ ΔT - dependent flow regime ($J_G < J_G^T$) $\alpha_{strat} = 0.725\left\{1 + 0.741\left[\frac{1-x}{x}\right]^{0.3321}\right\}^{-1}\left[\frac{\lambda_l^3\rho_l(\rho_l-\rho_v)gh_{lv}}{\mu_l D\Delta T}\right]^{0.25} + (1-x^{0.087})\alpha_{LO}$ $\alpha_D = \left[\alpha_A\left(\frac{J_G^T}{J_G}\right)^{0.8} - \alpha_{strat}\right]\left(\frac{J_G}{J_G^T}\right) + \alpha_{strat}$	General correlation for condensation in channels with D > 3 mm

good agreement with data ranging from highly turbulent flows to laminar flow conditions. The ability to predict heat transfer in mini- and microchannels was introduced in [11]. Improved versions of correlation are much more complex in comparison to the original form.

Cavallini et al. [12] introduced a novel method for determining the condensation heat transfer coefficient of refrigerants within horizontal plain tubes with inner diameters exceeding 3 mm (D > 3 mm). The proposed method was intentionally designed for simplicity, facilitating its immediate application in heat exchanger modelling and design for both traditional and emerging fluids in the HVAC industry. Notably, the method prioritizes accuracy and has been tested across an experimental database from various researchers, ensuring reliability with reduced experimental uncertainties. To maintain simplicity, the method employs two equations, each catering to temperature difference independent and dependent fluid flows. Parameters influencing condensation heat transfer are incorporated into the equations. The accuracy of the method was validated through a comprehensive comparison with data from HCFCs (hydrochlorofluorocarbons), HFCs (hydrofluorocarbons), HCs (hydrocarbons), carbon dioxide, ammonia, and water. Application of the temperature-dependent part of correlation is often complicated because wall temperature is usually not known. Nearly no data source provides precise information on the experimental value of the wall temperature. For design purposes, wall temperature has to be calculated iteratively. Equations for analysed methods are presented in Table 1.

2.1. Modification of in-house model

Mikielewicz and Mikielewicz [4] developed earlier a semi-empirical method for modelling heat transfer during boiling and condensation in minichannels. It is based on the hypothesis the total energy dissipation in the two-phase flow with boiling is a sum of two contributions, namely the dissipation of energy from the shearing flow and dissipation due to bubble generation. In case of condensation in the flow, the dissipation due to bubble generation is not present. The main equation includes the term modelling the resistance in two-phase flow and the term modelling the flow resistance associated with the nucleation process during boiling. The modified main equation presents only the term corresponding to the flow resistance because this paper focus on the modelling of heat transfer during condensation:

$$\frac{\alpha_{TPB}}{\alpha_{LO}} = \sqrt{R_{MS}^{n}\mathrm{Con}^{0.34}}, \tag{1}$$

where n = 0.76 for $\mathrm{Re_l}$ > 2300 and n = 2 for $\mathrm{Re_l}$ < 2300.

The flow resistance term is modelled using the modified Muller-Steinhagen and Heck correlation [5]: presented in Eq. (2),

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

where for turbulent flow

$$f_1 = \frac{\left(\frac{\rho_l}{\rho_v}\right)}{\left(\frac{\mu_l}{\mu_v}\right)^{0.25}}, \quad f_{1z} = \frac{\left(\frac{\mu_v}{\mu_l}\right)}{\left(\frac{\lambda_l}{\lambda_v}\right)^{1.5}}\left(\frac{c_{pl}}{c_{pv}}\right) \tag{3}$$

and for laminar flow

$$f_1 = \frac{\frac{\rho_l}{\rho_v}}{\frac{\mu_l}{\mu_v}}, \quad f_{1z} = \left(\frac{\lambda_v}{\lambda_l}\right). \tag{4}$$

Parameter α_{LO} is the heat transfer coefficient for the single-phase liquid flow and α_{TPB} represents the heat transfer coefficient during condensation in Eq. (1). Authors recommend standard Dittus-Boelter correlation for prediction of forced internal convection for single liquid phase heat transfer coefficient. The modification introduced in the present work is based on the introduction of the confinement number (Con), which is variation of Bond number, into the two-phase flow multiplier RMS. The flow resistance part was multiplied by the confinement number raised to the power of 0.34. The Con number,

$$\mathrm{Con} = \frac{\sqrt[2]{\frac{\sigma}{g(\rho_l-\rho_v)}}}{D}, \tag{5}$$

describes changes in the nature of the flow at high values of reduced pressure. It includes gas and liquid phase densities, surface tension, diameter and gravitational acceleration. This modification allows to take into account the change in physical properties that affect the heat transfer process at high values of reduced pressure. Surface tension alongside density has an immense effect on two phase flow structures which has been shown in experimental research [13]. For reduced pressure 0.2 where surface tension and difference in phase specific densities are high annular flow was observed at quality as low as 7%. For a reduced pressure 0.8 in the vicinity of the critical point annular flow occurred at around 50% quality. Surface tension and difference in phase densities decrease significantly for high values of reduced pressure.

2.2. Consolidated database

The modelling was based on a consolidated database that consists of 1286 measurement points published in 7 research papers. It includes data for 7 different natural refrigerants, different diameters, mass velocities and reduced pressures. The database includes values of reduced pressures ranging from 0.1 to as high as 0.809. Internal tube diameters vary from 0.76 mm to 9.43 mm. Mass velocity ranges from 75 kg/m^{2}s to 1000 kg/m^{2}s. Variation of quality is from 0% to 100%. Consolidated database is presented in Table 2.

2.3. Results

The consolidated database was compared with 7 methods for predicting the heat transfer coefficient during condensation due to Bohdal et al. [7], Dorao and Fernandiono [8], Shah [9–11] and Cavallini et al [12]. The calculation code was written in Engineering Equation Solver EES.

The mean absolute percentage error is given by the equation

$$\mathrm{MAPE} = \frac{1}{N}\sum\frac{A-F}{A} \times 100\%, \tag{6}$$

where: N – number of points, A – real value, F – forecast value.

Results of calculations are presented in Table 3. The best result was the modified Mikielewicz correlation for which the

Table 2. Consolidated database.

	Authors	Diameter [mm]	Fluid	Mass velocity [kg/(m²·s)]	Reduced pressure	Number of measurement points
1	Macdonald et al. [14,15]	0.76–1.45	R290	150–450	0.254–0.809	260
2	Zhuang et al. [16]	4	Ethane	101–255	0.22–0.522	230
3	Zhuang et al. [17]	4	Methane	99–254	0.43–0.76	286
4	Milkie et al. [18]	7.75	n-Pentane	150–600	0.04–0.17	163
5	Moreira et al. [19]	9.43	Propylene, R290, R600a	50–250	0.12–0.32	110
6	Del Col et al. [20]	1	Propylene	80–1000	0.35	109
7	Longo [21]	4	Propylene, R290	75–400	0.15–0.88	125

mean absolute percentage error (MAPE) was 28.13%. The second best model was the Cavallini [12] correlation. The third best correlation was the original Mikielewicz correlation which achieved MAPE = 38.97%. The fourth best method was the model due to Shah [11] which resulted in MAPE = 40.04%. Doaro and Fernandiono achieved a slightly worse result of 40.78%. The other correlations returned worse results due to their empirical nature – they were created for completely different conditions and specified refrigerants. The results of the calculations are shown in Table 3 and the histogram of discrepancies is in Fig. 1. Figure 2 presents histogram with modelling results. Figure 3 shows the results of modelling with the modified Mikielewicz correlation for all considered data sources separately.

Table 3. Results of calculations for eight modelling methods.

Model	Mean relative absolute error	Number of points within ±30%	Percentage of points within ±30%
Mikielewicz modified	28.13%	892	69.36%
Mikielewicz [4]	38.97%	662	51.48%
Bohdal [7]	163.46%	223	17.34%
Dorao and Fernandiono [8]	40.78%	724	56.30%
Shah 1979 [9]	93.17%	273	21.23%
Shah 2009 [10]	45.55%	727	56.53%
Shah 2019 [11]	40.04%	727	56.53%
Cavallini 2006 [12]	38.38%	772	60.03%

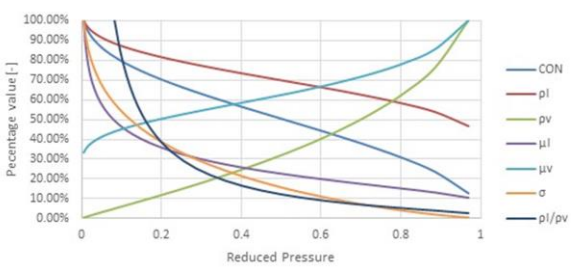

Fig. 1. Properties change in function of reduced pressure for R290.

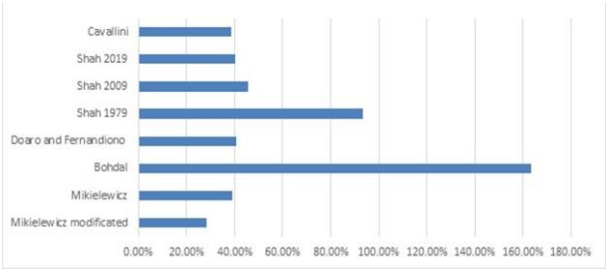

Fig. 2. Comparison of MAPE for calculated methods.

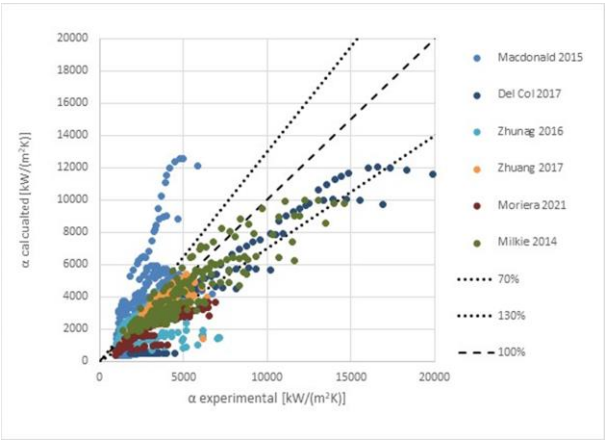

Fig. 3. Results of modeling with modified Mikielewicz correlation.

Results of modelling for data above reduced pressure 0.5 is presented in Table 4. Table 5 presents results for data below reduced pressure 0.5. The new modification slightly lowered values of error for low pressure experiments to 26.29% from 30.89%. Both values are close to the typical measurement error for two phase flow heat transfer measurement which is usually between 20% and 30%. The main improvement lies in data points with high pressure. In this region, new correlation has an error 31.63% whereas the old version has 54.33% and the best other correlations have values of MAPE around 50%. The main goal of the research was to correct modelling results for high values of saturation pressure, but new modification managed to increase accuracy in the whole spectrum of reduced pressure.

Table 4. Results of calculations for eight modelling methods for data points with reduced pressure higher than 50%.

Model	Mean relative absolute error	Number of points within ±30%	Percentage of points within ±30%
Mikielewicz modified	31.63%	319	72.01%
Mikielewicz [4]	54.33%	167	37.70%
Bohdal [7]	176.58%	47	10.61%
Dorao and Fernandiono [8]	57.58%	242	54.63%
Shah 1979 [9]	167.00%	19	4.29%
Shah 2009 [10]	49.67%	272	61.40%
Shah 2019 [11]	52.32%	272	61.40%
Cavallini 2006 [12]	51.61%	278	62.75%

Table 5. Results of calculations for eight modelling methods for data points with reduced pressure lower than 50%

Model	Mean relative absolute error	Number of points within ±30%	Percentage of points within ±30%
Mikielewicz modified	26.29%	573	67.97%
Mikielewicz [4]	30.89%	495	58.72%
Bohdal [7]	156.57%	176	20.88%
Dorao and Fernandiono [8]	31.95%	482	57.18%
Shah 1979 [9]	54.38%	254	30.13%
Shah 2009 [10]	43.39%	455	53.97%
Shah 2019 [11]	33.58%	455	53.97%
Cavallini 2006 [12]	31.43%	494	58.60%

Although changes below a reduced pressure of 0.5 are not spectacular, corrections for high saturation pressure are significant. The new version also managed to fit the most points inside the 30% error area for both regions.

Graphical comparison of correlations for all analysed data sources are shown in Figs. 4–10. Figure 4 presents graph for Macdonald et al. [14,15] data. Figure 5 presents graph for Zhuang et al. [16] data. Figure 6 presents graph for Zhuang et al. [17] data. Figure 7 presents graph for Del Col et al. [18] data. Figure 8 presents graph for Moreira et al. [19] data. Figure 9 presents graph for Milkie [20] data. Figure 10 presents graph for Longo et al. [21] data. Distributions present data for R290, ethane, methane, propylene and R600a. The modified Mikielewicz method performed the best of all analysed methods for all data sources.

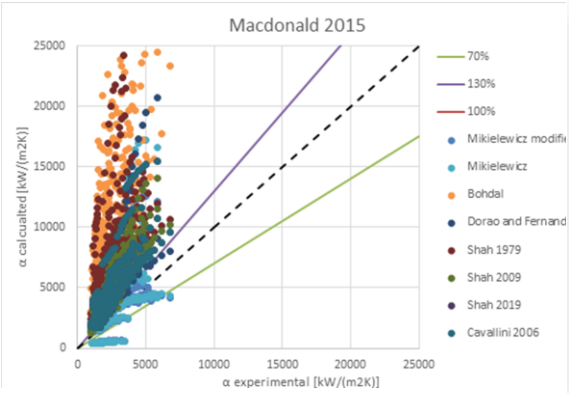

Fig. 4. Comparison of correlations for R290 [14,15].

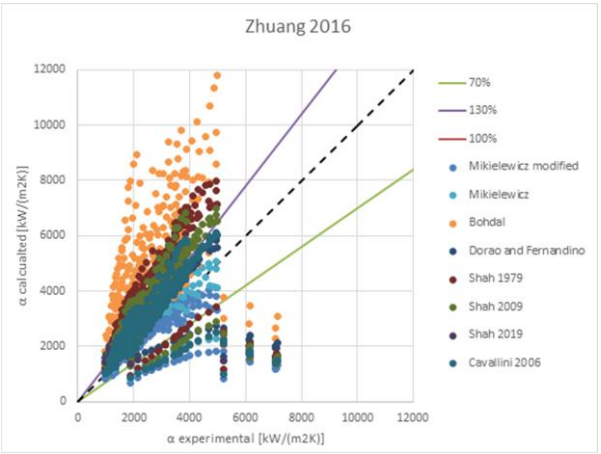

Fig. 5. Comparison of correlations for ethane [16].

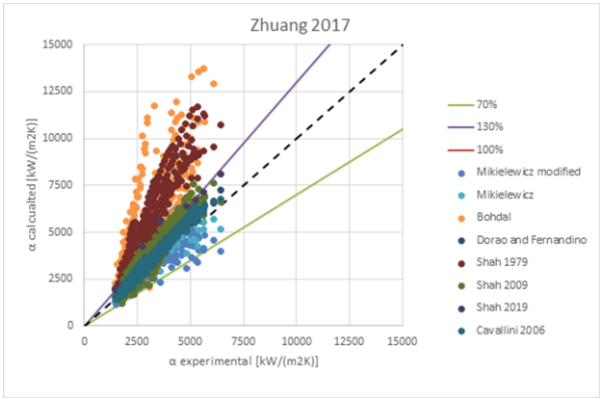

Fig. 6. Comparison of correlations for methane [17].

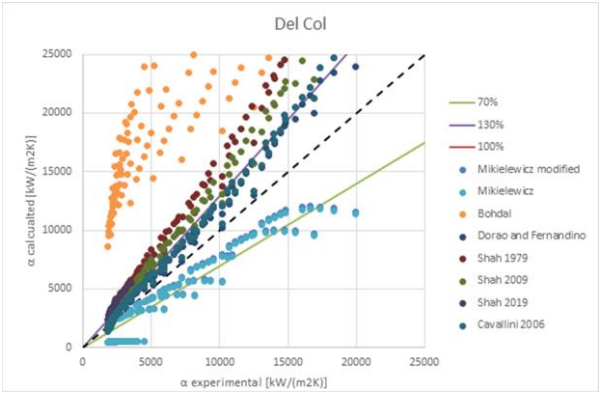

Fig. 7. Comparison of correlations for propylene [20].

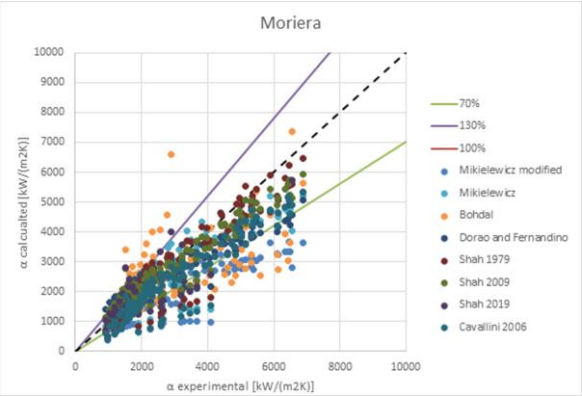

Fig. 8. Comparison of correlations for propylene, R290, R600a [19].

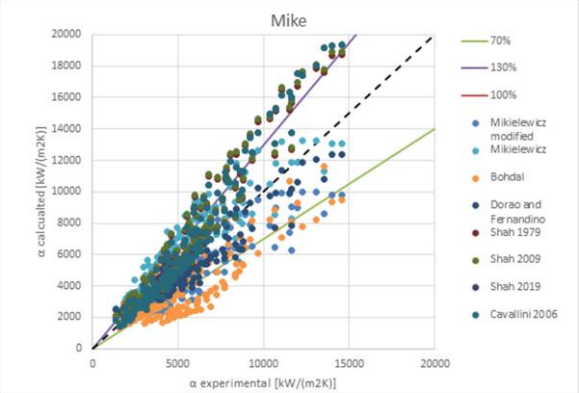

Fig. 9. Comparison of correlations for R290 [18].

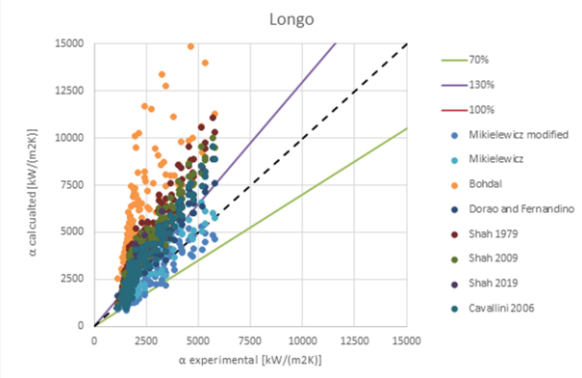

Fig. 10. Comparison of correlations for R290 and propylene [21].

3. Conclusions

A modified in-house method for the prediction of heat transfer coefficients during condensation of natural refrigerants with special emphasis on high values of reduced pressure has been proposed. A developed model has been formulated with the aim of achieving high accuracy while maintaining a straightforward structure. The modification is based on the fact that the two-phase flow multiplier used in the in-house model is modified with the adjusted value of the confinement number. Significant improvements have been obtained especially in the high range of reduced pressure values.

The modelling was based on an accumulated database of experimental measurement data, which consists of 1286 measurement points for 7 natural refrigerants. The database includes values of reduced pressures ranging from 0.1 to as high as 0.809. Channel internal diameters range from 0.76 mm to 9.43 mm. Mass velocity range from 75 kg/(m^{2}s) to 1000 kg/(m^{2}s). A full range of distribution of quality from 0% to 100% is considered. The new method was compared with 7 models from the literature and obtained the smallest value of mean absolute percentage error (MAPE) 28.13% while 69.36% had an error value lower than 30%. The second best method achieved 38.38% MAPE and 60.03% points below 30%error value. For data above 50% of reduced pressure model achieved 31.64% MAPE while second best method in this region achieved 49.67%. The other methods surpassed 50% MAPE. The outcomes reveal a highly satisfactory agreement with experimental data.

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References

[1] Pysz, M., Głuch, S., & Mikielewicz, D. (2023). Experimental study of flow boiling pressure drop and heat transfer of R1233zd(E) at moderate and high saturation temperatures. *International Journal of Heat and Mass Transfer*, 204, 123855. doi: 10.1016/j.ijheatmasstransfer. 2023.123855

[2] Szewczuk-Krypa, N., Drosińska-Komor, M., Głuch, J., & Breńkacz, L. (2018). Comparison analysis of selected nuclear power plants supplied with helium from high-temperature gas-cooled reactor. *Polish Maritime Research*, 25(S1), 204–210. doi: 10.2478/pomr-2018-0043

[3] Głuch, S.J., Ziółkowski, P., Witanowski, Ł., & Badur, J. (2021). Design and computational fluid dynamics analysis of the last stage of innovative gas-steam turbine. *Archives of Thermodynamics*, 42(3), 255–258. doi: 10.24425/ ather.2021.138119

[4] Mikielewicz, D., & Mikielewicz, J. (2011). A common method for calculation of flow boiling and flow condensation heat transfer coefficients in minichannels with account of nonadiabatic effects. *Heat Transfer Engineering*, 32(13-14), 1173–1181. doi: 10.1080/01457632.2011. 562728

[5] Jakubowska, B., & Mikielewicz, D. (2019). An improved method for flow boiling heat transfer with account of the reduced pressure effect. *Thermal Science*, 23(4), S1261–S1272. doi: 10.2298/TSCI19S4261J

[6] Mikielewicz, D., & Mikielewicz, J. (2022). An improved Müller-Steinhagen and Heck model for two phase pressure drop modelling at high reduced pressures. *Journal of Power Technologies*, 102(3), 81–87. https://papers.itc.pw.edu.pl/index.php/JPT/article/view/1783

[7] Bohdal, T., Charun, H., & Sikora, M. (2011). Comparative investigations of the condensation of R134a and R404A refrigerants in pipe minichannels. *International Journal of Heat and Mass Transfer*, 54(9-10), 1963–1974. doi: 10.1016/j.ijheatmasstransfer.2011.01.005

[8] Dorao, C.A., & Fernandino, M. (2018). Simple and general correlation for heat transfer during flow condensation inside plain pipes. *International Journal of Heat and Mass Transfer*, 122, 290–305. doi: 10.1016/j.ijheatmasstransfer.2018.01.097

[9] Shah, M.M. (1979). A general correlation for heat transfer during film condensation inside pipes. *International Journal of Heat and Mass Transfer*, 22(4), 547–556. doi: 10.1016/0017-9310(79)90058-9

[10] Shah, M.M. (2009). An improved and extended general correlation for heat transfer during condensation in plain tubes. *HVAC and R Research*, 15(5), 889–913. doi: 10.1080/10789669.2009. 10390871

[11] Shah, M.M. (2019). Improved correlation for heat transfer during condensation in conventional and mini/micro channels. *International Journal of Refrigeration*, 98, 222–237. doi: 10.1016/ j.ijrefrig.2018.07.037

[12] Cavallini, A., Col, D.D, Doretti, L., Matkovic, M., Rossetto, L., Zilio, C., & Censi, G. (2006). Condensation in horizontal smooth tubes: A new heat transfer model for heat exchanger design. *Heat Transfer Engineering*, 27(8), 31–38. doi: 10.1080/ 01457630600793970

[13] Głuch, S., Pysz, M., & Mikielewicz, D. (2023). Flow maps and flow patterns of R1233zd(E) in a circular minichannel at low, medium and high values of saturation pressure. *36th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, ECOS 2023*, 402–413. 25-30 June, Las Palmas De Gran Canaria, Spain. doi: 10.52202/069564-0037

[14] Macdonald, M., & Garimella, S. (2016). Hydrocarbon condensation in horizontal smooth tubes: Part I – Measurements. *International Journal of Heat and Mass Transfer*, 93, 75–85. doi: 10.1016/j.ijheatmasstransfer.2015.09. 018

[15] Macdonald, M., & Garimella, S. (2016). Hydrocarbon condensation in horizontal smooth tubes: Part II - Heat transfer coefficient and pressure drop modeling. *International Journal of Heat and Mass Transfer*, 93, 1248–1261. doi: 10.1016/j.ijheatmasstransfer.2015.09.019

[16] Zhuang, X.R., Gong, M.Q., Zou, X., Chen, G.F., & Wu, J.F. (2016). Experimental investigation on flow condensation heat transfer and pressure drop of R170 in a horizontal tube. *International Journal of Refrigeration*, 66, 105–120. doi: 10.1016/j.ijrefrig.2016.02.010

[17] Zhuang, X.R., Chen, G.F., Zou, X., Song, Q.L., & Gong, M.Q. (2017). Experimental investigation on flow condensation of methane in a horizontal smooth tube. *International Journal of Refrigeration*, 78, 193–214. doi: 10.1016/j.ijrefrig.2017.03.021

[18] Col, D.D., Azzolin, M., Bortolin, S., & Berto, A. (2017). Experimental results and design procedures for minichannel condensers and evaporators using propylene. *International Journal of Refrigeration*, 83, 23–38. doi: 10.1016/ j.ijrefrig.2017.07.012

[19] Moreira, T.A., Ayub, Z H., & Ribatski, G. (2021). Convective condensation of R600a, R290, R1270 and their zeotropic binary mixtures in horizontal tubes. *International Journal of Refrigeration*, 130, 27–43. doi: 10.1016/j.ijrefrig.2021.06.031

[20] Milkie, J.A. (2014). *Condensation of hydrocarbons and zeotropic hydrocarbon/refrigerant mixtures in horizontal tubes*. PhD thesis, Georgia Institute of Technology. http://hdl.handle.net/1853/51825

[21] Longo, G.A., Mancin, S., Righetti, G., & Zilio, C. (2017). Saturated vapour condensation of HFC404A inside a 4 mm ID horizontal smooth tube: Comparison with the long-term low GWP substitutes HC290 (Propane) and HC1270 (Propylene). *International Journal of Heat and Mass Transfer*, 108, 2088–2099. doi: 10.1016/j.ijheatmasstransfer.2016. 12.087