Condensation heat transfer in minichannels: a review of available correlations

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Abstract. Heat exchangers with enhanced performance are demanded in various engineering applications. Very often heat transfer devices are requested to guarantee not only high heat transfer coefficients but also small size and weight, thus limiting the charge of the operative fluid inside the heat exchanger. In order to increase the performance of condensers and to properly design new heat exchangers, it is essential to have predictive tools that are validated with experimental data. Sometimes, even well established semi-empirical correlations can be inaccurate in some microscale flow conditions or with new refrigerants. The present paper starts from the experimental database measured during condensation with different fluids (pure fluids and refrigerant blends) inside small channels (hydraulic diameter around 1 mm) at the Department of Industrial Engineering of the University of Padova. A critical review of available correlations for heat transfer during condensation in minichannels is presented. Predictions of heat transfer coefficients obtained applying selected models are compared with the experimental database that covers various refrigerants: hydrofluorocarbons (HFC, i.e. R32 and R134a), new hydrofluoroolefins (HFOs) with low global warming potential (R1234ze(E)), natural refrigerants (hydrocarbons such as propane) and zeotropic refrigerant blends of HFCs and HFOs (R32/R1234ze(E)). Refrigerant mixtures are studied because for some applications they may be a proper solution. For instance in the air-conditioning industry there are not drop-in pure fluids to replace the high global warming potential (GWP) fluids currently employed (e.g. R410A). Unfortunately, the design of condensers working with zeotropic mixtures poses the additional problem to account for the mass transfer resistance that leads to a penalization of the heat transfer coefficient. Experimental data are necessary for the assessment of predicting correlations that can be used with these new refrigerants blends.

1. Introduction
In convective heat transfer applications, flow passages sizes are shifting towards smaller dimensions in order to enhance the heat transfer coefficient and to reduce the charge of the working fluid inside the heat exchanger, limiting the risks connected to the release of potentially hazardous fluids in the atmosphere (Kandlikar et al. [1]). Minichannels devices are spreading in many applications, such as automotive condensers, air-conditioning systems, heat pipes for electronics cooling or spacecraft thermal control.

Condensation in minichannels differs from condensation in macrochannels due to the different interactions between gravity force, surface tension and gas-liquid interfacial shear stress. In
conventional tubes, the condensation process is dominated by vapor shear stress and gravity, while in minichannels the surface tension force can play an important role over gravitational force (Del Col et al. [2]). Depending on the operating conditions (mass velocity, fluid properties, vapor quality and channel shape), the relative role of shear stress, gravity force and surface tension can affect the transition between flow patterns, and thus the heat transfer mechanism, in a different way.

Basic attempts to develop flow regime maps for minichannels were done in adiabatic flow conditions (air-water, nitrogen-water or air-oil mixtures). Triplett et al. [3] conducted an investigation on flow regimes in adiabatic air-water flow inside circular minichannels of 1.1 and 1.45 mm diameter. As also stated by other investigators, stratified flow was not observed, while bubbly, slug, slug-annular and annular flow patterns were identified. Coleman and Garimella [4,5,6] and Garimella [7] were the first to conduct flow visualizations during condensation of pure fluid R134a inside nine different tubes of round, square and rectangular cross-sections with hydraulic diameter in the range 1-4.91 mm over the mass flux range $150 < G < 750$ kg m$^{-2}$ s$^{-1}$. Four major flow regimes, including annular, intermittent, wavy and dispersed flow, were identified. The size of the intermittent region increases as the hydraulic diameter decreases, with this effect being greater at the low mass fluxes due to the fact that surface tension achieves greater significance in comparison with gravitational force at these dimensions. Furthermore, as the hydraulic diameter decreases, the effects of surface tension increasingly counteract the gravity force, promoting and extending the size of the annular film flow pattern region instead of the stratified wavy flow regime.

Correlations developed for condensation inside circular conventional tubes in some cases are even able to predict quite accurately the heat transfer coefficient inside smaller diameters. This agreement occurs mainly at high mass velocities, when the flow is mostly annular and the condensation heat transfer is dominated by the vapour shear stress on the liquid film. However, at lower mass velocities, when the interfacial shear stress is not the dominant force that affects the condensation process, models developed for greater diameter tubes tend to fail (Del Col et al. [2]). At these values of mass velocity, micro-scale characteristics of heat transfer process, and in particular the surface tension, become more important [8, 9].

In the present paper, several models for the heat transfer coefficient prediction during condensation inside minichannels are presented and compared against the experimental database obtained in the Two-phase Heat Transfer Laboratory of the University of Padova. Experimental tests were conducted with pure fluids and with a low-GWP non-azeotropic binary mixture inside a 0.96 mm inner diameter circular minichannel. The test section is a counterflow tube-in-tube heat exchanger in which the refrigerant, entering as a two-phase mixture with the desired value of the vapour quality, is cooled by the water flowing in the external annulus. A detailed description of the experimental apparatus is provided in Matkovic et al. [10]. The tested binary mixture is a blend of R32 and R1234ze(E) at 46/54% by mass composition, presenting a GWP$_{100\text{years}}$ = 309 and a temperature glide of 8.2 K. Details about the database are reported in Table 1. In particular, it is worth underlining that the saturation temperature for the non-azeotropic mixture of R32/R1234ze(E) is intended as the mean value between dew and bubble temperatures.

| Authors         | Fluid            | $T_{\text{sat}}$ [°C] | $T_{\text{sat}} - T_0$ [K] | $G$ [kg m$^{-2}$ s$^{-1}$] |
|-----------------|------------------|------------------------|----------------------------|-----------------------------|
| Matkovic et al. [10] | R134a, R32      | 40                     | 5-18                       | 100-1200                    |
| Del Col et al. [11] | R1234ze(E)     | 40                     | 10-15                      | 100-800                     |
| Cavallini et al. [12] | R245fa        | 40                     | 9-16                       | 100-500                     |
| Del Col et al. [13] | R290 (propane) | 40                     | 11-20                      | 100-1000                    |
| Del Col et al. [14] | R32/R1234ze(E) (46/54% by mass composition) | 40 | 10-16 | 150-800 |

Table 1. Condensation experimental database taken inside the 0.96 mm diameter channel.
2. Prediction of the heat transfer coefficient

In the present paper, several condensation models originally developed for macrochannels \((d > 3 \text{ mm})\) and pure fluids have been considered for the comparison against the experimental data obtained in a 0.96 mm diameter minichannel test section. These models can be classified between shear-driven models, specifically developed for annular flow condition, and multi-regime models, for films under the simultaneous influence of shear and gravity. Furthermore, when condensation models developed for pure fluids are applied to non-azeotropic mixtures, an additional correction is needed to account for the mass transfer and diffusion phenomena that characterize the condensation process.

2.1. Condensation models for pure fluids

The shear-driven model by Akers et al. [15] was developed for annular flow condensation of pure fluids and azeotropic blends. It is based on the definition of the all-liquid flow rate, which provides the same heat transfer coefficient as an annular condensing flow. The all-liquid flow rate is expressed by an “equivalent” Reynolds number and used in a single-phase, turbulent flow equation to predict the condensation coefficient. However, the assumptions on which the equivalent Reynolds number is based result in the underprediction of many researchers’ data.

Starting from the work of Akers et al. [15], Moser et al. [16] developed a condensation model for smooth tubes. A new equivalent Reynolds number approach was proposed, based on the heat-momentum analogy, which states that a relationship exists between heat transfer and wall shear stress. During single-phase flow, an increase in wall-shear stress results in a heat transfer increase. Similarly, in the two-phase flow, it is expected that, as the wall shear stress increases, the heat transfer will also increase. Therefore, if a condensing two-phase flow is replaced by an all-liquid flow with the same wall shear stress, the analogy suggests that the single-phase heat transfer coefficient would be equal to the condensation coefficient, providing that the driving temperature differences, which define the single-phase and condensing heat transfer coefficients, are the same. The two-phase multiplier for the evaluation of the pressure gradient can be predicted using several methods, such as the correlation by Zhang and Webb [17], developed for small diameter channels. The Moser et al. [16] model was validated against 1197 data points from 18 sources, regarding refrigerants R12, R22, R113, R134a, R410A and R125, inner diameters from 3.14 to 20 mm. Figure 1 shows the comparison between experimental heat transfer coefficients taken inside the 0.96 mm diameter minichannel and calculated values with the Moser et al. [16] model. On average, the Moser et al. [16] correlation shows a good agreement with the present database, with a mean absolute deviation \(\epsilon_{AB}\) equal to 13.9% and a standard deviation \(\sigma_{N}\) of 19.7%. The disagreement between experimental and calculated values is greater at low mass fluxes: indeed, at mass velocity in the range \(G = 100-200 \text{ kg m}^{-2} \text{s}^{-1}\) the mean absolute deviation is equal to 15.6% and the standard deviation is of 20.5%. This should be due to the fact that the basic assumption of annular flow condensation may not be satisfied in this case. When considering the pure fluid R245fa, it is overpredicted at almost all the mass fluxes; this may be related to the different thermodynamic properties (low vapour density, high liquid density and kinematic viscosity) and to the lower operating pressures of R245fa with respect to the other tested fluids. It is interesting also to notice that the model by Moser et al. [16] is able to predict with good accuracy the experimental data of R290, except some points at low mass velocity, although it has not been developed for hydrocarbons \((\epsilon_{AB} = 10.9\%, \sigma_{N} = 7.83\%)\).

Cavallini et al. [18] developed a semi-empirical model to compute the heat transfer coefficient during condensation inside horizontal smooth tubes with inner diameter greater than 3 mm. This simple and easy-to-use model includes only one transition between two different regions: \(\Delta T\)-dependent and \(\Delta T\)-independent (where \(\Delta T\) is the saturation-minus-wall temperature difference). The model includes in these two categories all possible flow regimes (annular flow, stratified-wavy flow, stratified smooth flow and slug flow). This transition criterion was obtained from an analysis of condensation heat transfer data inside macrochannels. The model was validated against a total number of 4471 data points relative to HCFCs, HFCs, HCs, carbon dioxide, ammonia and water from several independent laboratories. As can be seen from Fig. 2, the model by Cavallini et al. [18] is able to predict the
experimental data inside the 0.96 mm circular minichannel with a mean absolute deviation $e_{AB}$ equal to 9.5% and a standard deviation $\sigma_N$ equal to 12.3%. Only data points with mass velocity in the range $G = 100-200$ kg m$^{-2}$ s$^{-1}$ are clearly underpredicted by more than 20% using this model; in this range of mass velocity the mean absolute deviation between calculated and experimental data is equal to 12.3% and the standard deviation is of 14.3%. This higher deviation may be related to the different flow patterns occurring in minichannel at low mass velocity as compared to macrochannels. Indeed, in the Cavallini et al. [18] model the effect of the surface tension is not taken into account in the development of the flow pattern map and this can lead to inaccurate predictions of the heat transfer coefficient at low mass fluxes.

Shah [19] presented two comprehensive correlations for the prediction of the heat transfer coefficient, starting from the Shah [20] model developed for condensation inside macrochannels. Three different regimes were identified: Regime I corresponding to intermittent, annular and mist flow, Regime II corresponding to wavy flow and Regime III corresponding to stratified flow. The ultimate correlations proposed in Shah [19] can be applied also to minichannels, with specific adjustments to the original correlation. Indeed, at high mass flow rates, inertia force dominates the surface tension force while at low flow rates the surface tension force dominates. For this reason, an additional condition on the Weber number for all mass flowings as a vapour $We_\beta$ (which expresses the ratio between surface tension force and inertia force) was added to the original correlation. Furthermore, for $\Delta T$-independent flow regime, one of the two versions of the correlation uses the equation of Cavallini et al. [18] to improve its accuracy. The Shah [19] correlations were validated by comparison with a database that includes 33 fluids, diameters from 0.10 to 49.0 mm, reduced pressures in the range 0.0008-0.946, mass velocity from 1.1 to 1400 kg m$^{-2}$ s$^{-1}$, various tube shapes and orientations.

As it can be seen from Fig. 3, most of the data of the present experimental database are predicted within the ±20% error band, with a mean absolute deviation of 10.4% and a standard deviation of 13.3%. As for the previous correlations, a higher disagreement is found at $G = 100-200$ kg m$^{-2}$ s$^{-1}$ for all fluids ($e_{AB} = 15.2\%$ and $\sigma_N = 17.7\%$), except for R134a. The main discrepancies between experimental and predicted values at low mass fluxes are found at low vapour qualities, mainly due to the transition criterion from Regime I (intermittent, annular and mist flow) to Regime II (wavy flow pattern) and to the increased effect of the surface tension.

![Fig. 1](image1.png)  
Fig. 1 Comparison between experimental and predicted values by the Moser et al. [16] model in the 0.96 mm diameter test section.

![Fig. 2](image2.png)  
Fig. 2 Comparison between experimental and predicted values by the Cavallini et al. [18] model in the 0.96 mm diameter test section.
The condensation heat transfer model developed by Thome et al. [21] is based on a flow pattern map for condensation in horizontal smooth tubes, which considers fully stratified, stratified-wavy, intermittent, annular and mist flow. This model incorporates a method for the calculation of vapour void fraction spanning from low pressures up to pressures near the critical point. The proposed condensation model includes the effect of liquid-vapor interfacial roughness on the heat transfer by means of an enhancement factor. The model was validated against a very broad experimental database (fifteen fluids) including propane (R290), R134a, R32, R404A and R410A. It has been tested for mass velocities from 24 to 1022 kg m$^{-2}$ s$^{-1}$, vapour qualities from 0.03 to 0.97, reduced pressures from 0.02 to 0.80 and tube diameters from 3.1 to 21.4 mm. Fig. 4 shows the comparison between experimental and calculated heat transfer coefficients with the Thome et al. [21] model. Most of the experimental data are predicted in the range ±20%, in particular the mean absolute deviation between calculated and experimental values is equal to 12.4% while the standard deviation is 14.8%. It should be stressed that, at low mass velocity ($G = 100-300$ kg m$^{-2}$ s$^{-1}$), the predicting accuracy of the model tends to fail ($e_{AB} = 15.07\%$, $\sigma_N = 19.02\%$). In the whole range of mass velocities, the agreement between calculated and experimental data for all fluids results to be lower at very high and very low vapour qualities, in particular with hydrocarbon R290. A similar trend, but with reference to a 8.0 mm diameter macrochannel, was observed in Thome et al. [21] with hydrocarbons.

2.2. Condensation models for non-azeotropic mixtures

All the aforementioned condensation models for pure fluids and azeotropic blends are extendable to the prediction of the heat transfer coefficients of zeotropic mixtures by use of the Bell and Ghaly [22] method. This method essentially consists of assuming two thermal resistances in series for the evaluation of the heat transfer coefficient of the zeotropic mixture, as indicated in equation (1):

$$
\alpha_{\text{mix}} = \left( \frac{1}{\alpha_f} + \frac{\partial q_{SG}}{\partial q_f} / \frac{\partial q_r}{\alpha_f} \right)^{-1} \tag{1}
$$
where the first resistance $1/\alpha_f$ is across the condensate film itself and the corresponding heat transfer coefficient is evaluated with a condensation model for pure fluid but using the thermophysical properties of the mixture. The second resistance, instead, is for the convective heat transfer in the vapor phase from the bulk temperature to the temperature at the vapour-liquid interface. In this term $\alpha_g$ is the convective heat transfer coefficient of the vapour flowing alone in the tube, calculated with the Dittus-Boelter equation, while $\frac{\partial q_{sg}}{\partial t} / \partial q_t$ expresses the ratio between the sensible vapour heat flow rate and the total heat flow rate. This ratio can be calculated as a function of the vapour quality and the temperature glide of the mixture, as indicated in equation (2):

$$\frac{\partial q_{sg}}{\partial t} \approx x_c p_g \left( \frac{\Delta T_{\text{glide}}}{\Delta h_{\text{mix}}} \right)$$ (2)

The additional thermal resistance is required to account for the different volatility of the pure components of the mixture. Indeed, the temperature glide of the blend is responsible for the creation of a mass transfer resistance owing to the fact that the less volatile component must diffuse through the vapour phase before it can condense at the liquid-vapor interface.

In the present paper, heat transfer coefficients of a non-azeotropic mixture of R32 and R1234ze(E) (46/54% by mass composition) measured during condensation inside a 0.96 mm diameter minichannel have been considered. Figure 5 reports the predicted heat transfer coefficients of the mixture calculated with the Cavallini et al. [18] model for pure fluids without the Bell and Ghaly [22] correction term, plotted against the corresponding experimental values. As can be seen, the model overestimates the experimental data with a mean absolute deviation of 11.8% and a standard deviation of 6.1%. The application of Bell and Ghaly [22] correction, as shown in Fig. 6, allows a more accurate prediction of the heat transfer coefficients with a mean absolute deviation of 6.7% and a standard deviation of 4.9%.

**Fig. 5** Comparison between experimental and calculated heat transfer coefficients using the Cavallini et al. [18] model for pure fluids. $G$ is the mass velocity in kg m$^{-2}$ s$^{-1}$.

**Fig. 6** Comparison between experimental and calculated heat transfer coefficients using the Cavallini et al. [18] model coupled with the Bell-Ghaly correction [22]. $G$ is the mass velocity in kg m$^{-2}$ s$^{-1}$.
3. Conclusions

In the present paper, condensation models for the prediction of the heat transfer coefficient inside minichannels have been compared against an experimental database taken inside a 0.96 mm diameter test section. The heat transfer models by Moser et al. [16], Cavallini et al. [18], Shah [19] and Thome et al. [21] have been considered for this comparison. Except for the model by Shah [19], the others have been developed for conventional channels, but they have proven to be applicable also to minichannels at high mass velocities. At low mass velocities, these models tend to fail because they are based on datasets obtained with conventional tubes. Above all, the surface tension effect may differ inside minichannels. At mass flux $G = 100 \text{ kg m}^{-2}\text{s}^{-1}$ the considered models display the highest disagreement with the experimental data. At these conditions they may need proper correction to predict the heat transfer mechanism. Considering the present database, the model of Cavallini et al. [18] provided the best prediction having a mean absolute error $e_{AB}$ equal to 9.5% and a standard deviation $\sigma_N$ equal to 12.3%. Furthermore, condensation models developed for pure fluids are suitable for the prediction of the heat transfer coefficient of non-azeotropic mixtures inside minichannels coupled with the Bell and Ghaly [22] correction to account for the additional mass transfer resistance.

### Nomenclature

- $c_p$ [J kg$^{-1}$ K$^{-1}$]: Specific heat
- $d$ [m]: Hydraulic diameter
- $e_{AB}$ [-]: Mean absolute deviation
- $G$ [kg m$^{-2}$ s$^{-1}$]: Mass velocity
- HTC [W m$^{-2}$ K$^{-1}$]: Heat transfer coefficient
- $q$ [W]: Heat flow rate
- $t$ [°C]: Temperature
- $T$ [K]: Temperature
- $x$ [-]: Thermodynamic vapour quality
- $\alpha$ [W m$^{-2}$ K$^{-1}$]: Heat transfer coefficient
- $\Delta h$ [J kg$^{-1}$]: Specific enthalpy difference
- $\Delta T$ [K]: Temperature difference
- $\sigma_N$ [-]: Standard deviation

### Subscripts

- $f$: Film
- $s$: Sensible
- $t$: Total
- $g$: Vapor
- $sat$: Saturation
- $w$: Wall
- $mix$: Mixture

### Greek symbols

- $\alpha$: Heat transfer coefficient
- $\Delta T$: Temperature difference
- $\sigma_N$: Standard deviation

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