A Critical Review on Condensation Pressure Drop in Microchannels and Minichannels

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http://dx.doi.org/10.5772/60965

1. Introduction

Condensation in microscales has applications in a wide variety of advanced microthermal devices. For instance, condensation in microscales is widely used in small devices like air-cooled condensers for the air-conditioning and automotive industry, in heat pipes, thermosyphons and other applications for system thermal control. Microchannel condensers are being used to increase heat transfer performance to reduce component size and improve energy efficiency. After 2000s, experimental data became available in open literature in condensation of different refrigerants in small hydraulic diameter microchannels.

This chapter is a continuation of the authors' previous work about a critical review on condensation heat transfer in microchannels and minichannels [1]. The current chapter consists of four sections: Introduction, Literature Review, Recommendations for Future Studies, Summary and Conclusions. The authors used the same style in writing their recent paper about condensation heat transfer in microchannels and minichannels [1].

In the present chapter, the authors use the microchannels and minichannels classification proposed by Kandlikar [2]. According to his classification, the following can be used: for microchannels, d_h = 10-200 μm; for mini-channels, d_h = 200 μm-3 mm. In comparison, conventional channels have hydraulic diameters (d_h) ≥ 3 mm. Therefore, the present chapter covered channels have hydraulic diameters (d_h) in the range 10 μm ≤ d_h 3 mm according to the Kandlikar classification [2].

In macroscale, the gravitational forces are more important than the shear and surface tension forces, and the opposite occurs when the diameter is smaller. Also, Wang and Rose [3] cited another important influence in non-circular microchannel condensation: the viscosity in transverse flow.

Majority of the correlations proposed to predict the frictional pressure gradient during condensation in microscales are based on modifications from the Lockhart and Martinelli [4], Chisholm [5] and Friedel [6] correlations, which were proposed for conventional diameters, and their results show large deviations compared with the experimental data of Dalkilic and Wongwises [7].

2. Literature review

Koyama et al. [8] investigated experimentally the local characteristics of heat transfer and pressure drop for pure refrigerant R134a condensation in two kinds of 865 mm long multi-port extruded tubes having eight channels in hydraulic diameter of 1.11 mm and 19 channels in hydraulic diameter of 0.80 mm. The researchers measured the pressure drop through small pressure measuring ports at an interval of 191 mm. They measured the local heat transfer rate in effective cooling length in every subsection of 75 mm using heat flux sensors. They found that the experimental data of frictional pressure drop agreed with the correlation of Mishima and Hibiki [9], while the correlations of Chisholm and Laird [10], Soliman et al. [11], and Haraguchi et al. [12] overpredicted.

Garimella [13] presented an overview of using the flow visualization in micro- and mini-channel geometries to develop the pressure drop and heat transfer models during condensation of refrigerants. The researcher recorded condensation flow mechanisms for round, rectangular, and square tubes for mass flux (G) of 150 kg/(m^2.s) and 750 kg/(m^2.s) and $0 < x < 1$ with hydraulic diameters (d_h) in the range of 1-5 mm using unique experimental techniques that permitted flow visualization during the condensation process. He documented the influence of miniaturization on the flow regime transitions and channel shape. He categorized the flow mechanisms into four various flow regimes: dispersed flow, intermittent flow, wavy flow, and annular flow. The four various flow regimes were further subdivided into many flow patterns within every regime. He observed that the annular and intermittent flow regimes became larger as the tube hydraulic diameter (d_h) was decreased, and at the expense of the wavy flow regime. These maps and transition lines could be used to predict the flow pattern or regime that would be established for a given tube geometry, mass flux (G), and mass quality (x). He used these pressure drop measurements, together with observed flow mechanisms, to develop experimentally validated models for pressure drop during condensation in every of these flow regimes for different circular and noncircular channels with $0.4 < d_h < 5$ mm. His flow regime-based models yield substantially better pressure drop predictions than the traditionally used correlations that were primarily based on air-water flows for large diameter tubes.

Garimella et al. [14] presented a multiple flow-regime model of refrigerant R134a in horizontal microchannels for pressure drop during condensation. The researchers used five circular channels ranging in hydraulic diameter (d_h) from 0.5 mm to 4.91 mm to measure two-phase pressure drops. For every tube under consideration, they took first pressure drop measurements for five various refrigerant mass fluxes (G) between 150 kg/(m^2.s) and 750 kg/(m^2.s) over the entire range of mass qualities from 100% vapor (x = 1) to 100% liquid (x = 0). In order to assign the applicable flow regime to the data points, they used results from previous work by the author on condensation flow mechanisms in microchannel geometries. They modified and combined pressure drop models for intermittent [15, 16] and annular [17] flow reported earlier by the authors to develop a comprehensive model that addressed the entire progression of the condensation process from the vapor phase to the liquid phase. Their model was based on the R134a flow regime transition criteria and pressure drop data observed by Coleman and Garimella [18-20]. Their model considered the intermittent only, intermittent/discrete wavy annular transition, and annular only flow. The intermittent flow model considered the unit cell. This unit cell consisted of a single vapor bubble and a single liquid slug. For every unit cell, the pressure drop contributions of the liquid slug, vapor bubble, and film/slug transition region were summed together to arrive at the total pressure drop. Therefore, the pressure drop in the intermittent flow regime could be represented as follows:

$$\frac{\Delta P}{L}=\left(\frac{dP}{dz}\right)_{film\ bubble}\left(\frac{L_{bubble}}{L_{unit\ cell}}\right)+\left(\frac{dP}{dz}\right)_{slug}\left(\frac{L_{slug}}{L_{unit\ cell}}\right)+\Delta P_{transition}\left(\frac{N_{unit\ cell}}{L}\right) \tag{1}$$

It can be seen that the frictional gradient in the bubble and film region are independently calculated and then related to the total pressure drop by the relative length of the bubble and slug compared to the total unit cell. Also, the pressure drop associated with the acceleration/deceleration of the liquid phase around the fore and aft regions of the vapor bubble is considered. The total number of unit cells per tube length ($N_{unit\ cell}/L$) establishes the total pressure drop contribution of the transitions. Solution of the above equation requires knowledge of the slug frequency that is empirically correlated as follows:

$$N_{UC}\left(\frac{d_h}{L_{tube}}\right)=\left(\frac{d_h}{L_{UC}}\right)=1.573\ (\mathrm{Re}_{\mathrm{slug}})^{-0.507} \tag{2}$$

Garimella et al. [14] correlated Eq. (2) with pressure drop data from the intermittent and discrete wave flow region [15, 16]. The researchers used data from both flow regimes because as the flow transitions from the intermittent to discrete wavy flow, the vapor bubbles were replaced by stratified well-defined liquid/vapor layers. Therefore, within the discrete wavy flow regime between the pure intermittent and pure annular flow regime, the number of unit cells approached zero. This construct of the intermittent/discrete wavy/annular transition allowed the use of the empirical relation in Eq. (2) in a consistent manner.

For pressure drop in annular flow, Garimella et al. [14] used in their model the following assumptions: (1) steady flow, (2) equal pressure gradients in the liquid and vapor core, (3) uniform liquid-film thickness, and (4) no liquid entrainment in the vapor core. Therefore, the resulting equation for annular pressure drop could be represented as follows:

$$\frac{\Delta P}{L}=\frac{2\ f_i\rho_g U_g^2}{d_i} \tag{3}$$

Equation (3) could be written in terms of the more convenient tube diameter (d) through the use of Baroczy [21] void fraction model

$$\frac{\Delta P}{L} = \frac{2\ f_i G^2 x^2}{\rho_g \alpha^{2.5} d} \tag{4}$$

The ratio of this interfacial friction factor (f_i) obtained from the experimental data to the corresponding liquid phase Fanning friction factor (f_l) computed using the Churchill [22] equation was then computed and correlated in terms of the Lockhart-Martinelli parameter (X) and the liquid phase Reynolds number (Re_l) as follows:

$$\frac{f_i}{f_l} = aXRe_l + bRe_l + cX \tag{5}$$

$$X = \left[\frac{(dP/dz)_l}{(dP/dz)_g}\right]^{\frac{1}{2}} \tag{6}$$

$$Re_l = \frac{Gd(1-x)}{(1+\sqrt{\alpha})\mu_l} \tag{7}$$

It should be noted that the liquid phase Reynolds number (Re_l) was defined in Eq. (7) in terms of the annular flow area occupied by the liquid phase. Also, the liquid phase Reynolds number (Re_l) would be used to compute the liquid phase pressure drop in the Lockhart-Martinelli parameter (X) in Eq. (6). Similarly, the gas phase Reynolds number required (Re_g) for the calculation of the gas phase pressure drop through the gas core in the Lockhart-Martinelli parameter (X) in Eq. (6) was calculated as follows:

$$Re_g = \frac{Gdx}{\mu_g \sqrt{\alpha}} \tag{8}$$

The various correlation constants (a, b, and c in Eq. (5)) were obtained for 249 points in the laminar (Re_l < 2200) and 24 points in the turbulent (Re_l > 3400) regions, with an appropriate interpolation scheme for 11 points in the transition regime using the observed trends of the friction factor ratio (f_i/f_l). Their model predicted 88% of these data points to within ±20%.

Garimella et al. [14] made many improvements to their preliminary model, even though it was able to successfully predict pressure drop in the annular flow regime for a wide range of circular tubes. Also, the researchers extended the applicability of their model to the mist- and disperse-flow regions through the use of the surface tension parameter (ψ).This non-dimensional parameter (ψ), which accounts for the surface tension effects was introduced by Lee and Lee [23]:

$$\psi = \frac{U_l \mu_l}{\sigma} = \frac{G(1-x)\mu_l}{\rho_l(1-\alpha)} \tag{9}$$

Garimella et al. [14] correlated the ratio of the interfacial friction factor to the liquid phase friction factor (f_i/f_l) in terms of the Lockhart-Martinelli parameter (X), liquid phase Reynolds number (Re_l), and the surface tension parameter (ψ) as follows:

$$\frac{f_i}{f_l} = AX^a Re_l^b \psi^c \tag{10}$$

Garimella et al. [14] computed the Fanning friction factors (f_l and f_g) required for the individual phase pressure drops in the Lockhart-Martinelli parameter (X) using $f=16/Re$ for $Re < 2100$ and the Blasius expression $f = 0.079Re^{-0.25}$ for $Re > 3400$. It should be noted that the Churchill [22] friction factor was used for these single phase pressure drops in the previous paper by Garimella et al. [17]; however, the former expressions were found to yield a better fit to the data in this study. The values for the constant A and the exponents a, b, and c were a function of the liquid film flow regime (Re_l) (laminar or turbulent) and the tube geometry. Regression analysis on data grouped into two regions based on the liquid phase Reynolds number (Re_l) yielded the following values for the respective parameters in Eq. (10):

$A = 1.308 \times 10^{-3}$; $a = 0.427$, $b = 0.930$; $c = -0.121$ for laminar region ($Re_l < 2100$).

$A = 25.64.$; $a = 0.532$, $b = -0.327$; $c = 0.021$ for turbulent region ($Re_l > 3400$).

They used an interpolation technique for liquid film Reynolds numbers in the transition region ($2100 < Re_l < 3400$) to determine the pressure drop.

Garimella et al. [14] recommended interpolation between the two models for data points determined to be in transition between intermittent/discrete and annular flow. The researchers developed empirical transition criteria from intermittent to other flow regimes from the flow visualization studies of Coleman [24]. These criteria were the model transition criteria for transition from intermittent to other flow regimes. The transition quality from intermittent to other flow regimes was predicted by the following transition criteria, where the mass flux (G) has units of kg/(m^2.s).

$$x \leq \frac{a}{G+b} \tag{11}$$

The geometry-dependent constants a and b were functions of hydraulic diameter (d_h, in mm) given as follows:

$$a = 69.57 + 22.60 \exp(0.259 d_h) \tag{12}$$

$$b = -59.99 + 176.8 \exp(0.383 d_h) \tag{13}$$

Their combined model accurately predicted condensation pressure drops in the annular, disperse wave, mist, discrete wave, and intermittent flow regimes. They found that their resulting model predicted 82% of the data within ±20%.

Haui and Koyama [25] investigated experimentally the local characteristics of heat transfer and pressure drop for carbon dioxide (CO_2) condensation in a multi-port extruded aluminum test section, which had 10 circular channels each with 1.31 mm inner diameter. The researchers performed their measurements for the inlet temperature (T) of CO_2 from 21.63 to 31.33°C, pressure (P) ranged from 6.48 to 7.3 MPa, mass flux (G) from 123.2 to 315.2 kg/(m^2.s), vapor quality (x) from 0 to 1, and heat flux (q) from 1.10 to 8.12 kW/m^2. They found that heat transfer coefficient in the two-phase region was higher than that in the single-phase, mass flux had important influence on condensation heat transfer characteristics, and pressure drop was very small along the test section. The influences of vapor quality on the heat transfer coefficients were not evident because of the large scattering of data. Also, they compared their experimental data with previous correlations and observed large discrepancies. Therefore, the existing model failed to predict their experimental data.

Cavallini et al. [26] reviewed published experimental work focusing on condensation flow regimes, pressure drop, and heat transfer in minichannels. New experimental data were available with low pressure (R236ea), medium (R134a) and high pressure (R410A) refrigerants in minichannels of different cross section geometry and with hydraulic diameters (d_h) ranging from 0.4 to 3 mm. The researchers presented a literature review to discuss flow regimes transitions because of the flow regimes effect on pressure drop and heat transfer. They compared the available experimental heat transfer coefficients and frictional pressure gradients with semi empirical and theoretical models developed for conventional channels and with models specifically created for minichannels.

Chowdhury et al. [27] presented an on-going experimental study of condensation pressure drop and heat transfer of refrigerant R134a in a single rectangular microchannel of hydraulic diameter (d_h) = 0.7 mm and high aspect ratio (AR) = 7. Their data would help explore the condensation phenomenon in microchannels that was necessary in the design and development of small-scale heat exchangers and other compact cooling systems. They used the mass fluxes (G) of 130 and 200 kg/(m^2.s) and the inlet vapor qualities (x_i) range was between 20% and 80% in their study. The researchers maintained the microchannel outlet conditions at close to thermodynamic saturated liquid state through a careful experimental procedure. They compared the data recorded trends to that found in recent literature on similar dimension tubes.

Garimella [28] reviewed a large number of the existing studies on mini- to microchannel condensation covering the flow pattern, pressure drop, void fraction, and heat transfer prediction methods. The researcher presented the available relevant information on pressure drops in condensing flows through relatively small channels and primarily adiabatic flows through microchannels in tabular form. Also, he compared different techniques for predicting the frictional pressure gradient during condensation of refrigerant R-134a flowing through a 1 mm diameter tube, at a mean quality of 0.5, at a mass flux of 300 kg/(m^2.s), and a pressure of 1500 kPa. He showed graphically a comparison of the pressure drops predicted by these different techniques. He found that the predicted pressure drops varied considerably, from 4.8 to 32.3 kPa. He attributed this large variation to the considerably various two-phase multipliers developed by the different investigators. He recommended choosing a model that was based on the geometry, fluid and operating conditions similar to those of interest for a given application.

Agarwal and Garimella [29] presented a multiple flow-regime model for pressure drop during condensation of refrigerant R134a in horizontal microchannels. The researchers considered in their study condensation pressure drops measured in two circular and six noncircular channels with hydraulic diameter (d_h) = 0.42 -0.8 mm. For every tube under consideration, they took pressure drop measurements for five various refrigerant mass fluxes (G) between 150 kg/(m^2.s) and 750 kg/(m^2.s) over the entire range of qualities from 100% vapor (x = 1) to 100% liquid (x = 0). In order to assign the applicable flow regime to the data points, they used results from previous work by the authors on condensation flow mechanisms in microchannel geometries. For example, Garimella et al. [14] reported a comprehensive model for circular tubes that addressed the progression of the condensation process from the vapor phase to the liquid phase by modifying and combining the pressure drop models for intermittent [15, 16] and annular [17] flows reported earlier by them. Like the previous work on circular channels, they presented new condensation pressure drop data on six noncircular channels over the same flow conditions. Similar to the multiple flow-regime mode developed earlier by Garimella et al. [14] for circular microchannels, they developed a multiple flow-regime model for these new cross sections. Their combined model accurately predicted condensation pressure drops in the intermittent, disperse-wave, mist, discrete-wave, and annular flow regimes for both circular and noncircular microchannels of similar hydraulic diameters. In addition, they addressed overlap and transition regions between the respective regimes to yield relatively smooth transitions between the predicted pressure drops. They found that their resulting model predicted 80% of the data within ±25%. Moreover, they demonstrated the influence of tube shape on pressure drop.

Cavallini et al. [30] presented a model for calculation of the frictional pressure gradient during condensation or adiabatic liquid-gas flow inside minichannels with different surface roughness. The researchers used new experimental frictional pressure gradient data associated to single-phase flow and adiabatic two-phase flow of R134a inside a single horizontal mini tube with rough wall in their modelling to account for the effects of surface roughness. It was a Friedel [6] based model and it took into account fluid properties, tube diameter, mass flux, vapor quality, reduced pressure, entrainment ratio and surface roughness. With respect to the flow pattern prediction capability, they built for shear dominated flow regimes inside pipes, thus, annular, annular-mist and mist flow were here predicted. However, they extended the suggested procedure to the intermittent flow in minichannels and applied it also with success to horizontal macro tubes. Cavallini et al. [30] suggested the following equations to calculate the frictional pressure gradient during adiabatic flow or during condensation, when the dimensionless gas velocity (Jg) 2.5

$$\left(\frac{dP}{dz}\right)_f = \phi_{lo}^2 \left(\frac{dP}{dz}\right)_{f,lo} = \phi_{lo}^2 \frac{2 f_{lo} G^2}{\rho_l d_h} \tag{14}$$

$$\mathrm{f_{lo}} = 0.046\ \mathrm{Re_{lo}^{-0.2}} = 0.046 \left(\frac{\mathrm{Gd_h}}{\mu_\mathrm{l}}\right)^{-0.2} \qquad \text{for any } \mathrm{Re_{lo}} \tag{15}$$

The friction factor from Eq. (15) refers to surfaces with negligible surface roughness.

$$\phi_{lo}^{2} = \mathrm{Z} + 3.595 \cdot \mathrm{F} \cdot \mathrm{H} \cdot (1 - \mathrm{E})^{\mathrm{W}} \tag{16}$$

$$\mathrm{W} = 1.398\mathrm{P_r} = 1.398\frac{\mathrm{P}}{\mathrm{P_{cr}}} \tag{17}$$

$$\mathrm{Z} = (1 - \mathrm{x})^{2} + \mathrm{x}^{2}\left(\frac{\rho_l}{\rho_g}\right)\left(\frac{\mu_g}{\mu_l}\right)^{0.2} \tag{18}$$

$$\mathrm{F} = \mathrm{x}^{0.9525}(1 - \mathrm{x})^{0.414} \tag{19}$$

$$H = \left(\frac{\rho_l}{\rho_g}\right)^{1.132}\left(\frac{\mu_g}{\mu_l}\right)^{0.44}\left(1 - \frac{\mu_g}{\mu_l}\right)^{3.542} \tag{20}$$

The entrainment ratio (E) in Eq. (16) must be calculated as suggested by Paleev and Filippovich [31]:

$$\begin{aligned} &\mathrm{E} = 0.015 + 0.44 \cdot \log\left[\left(\frac{\rho_{gc}}{\rho_l}\right)\left(\frac{\mu_l j_g}{\sigma}\right)^{2} 10^{4}\right] \\ &\text{if E} \ \geq \ 0.95, \quad \mathrm{E} = 0.95 \text{if E} \ \geq \ 0.95, \quad \mathrm{E} = 0.95 \\ &\text{if E} \ \leq \ 0, \quad \mathrm{E} = 0 \end{aligned} \tag{21}$$

where the homogeneous gas core density (ρ_{gc}) was given by

$$\begin{aligned} &\rho_{\mathrm{gc}} = \left(\frac{\mathrm{x} + (1 - \mathrm{x})\mathrm{E}}{\frac{\mathrm{x}}{\rho_{\mathrm{g}}} + \frac{(1 - \mathrm{x})\mathrm{E}}{\rho_{\mathrm{l}}}}\right) \\ &\rho_{\mathrm{gc}} \approx \rho_{\mathrm{g}}\left(1 + \frac{(1 - \mathrm{x})\mathrm{E}}{x}\right) \quad \text{for } \rho_{\mathrm{l}} >> \ \rho_{\mathrm{g}} \end{aligned} \tag{22}$$

This model, presented above, for the frictional pressure gradient could be extended to lower vapor qualities and mass fluxes when the dimensionless gas velocity (Jg) 2.5, with the constraint to take the higher value between $(dP/dz)_f$ from Eqs. (14)-(22) and the all-liquid frictional pressure gradient $(dP/dz)_{frlo}$ for the considered channel geometry Eqs. (23)-(25)

http://dx.doi.org/10.5772/60965

$$\left(\frac{dP}{dz}\right)_{f,lo} = \frac{2 f_{lo} G^2}{\rho_l d_h} \tag{23}$$

$$\begin{aligned} &\text{for } \mathrm{Re}_{\mathrm{lo}} > 2000, \quad \mathrm{f}_{\mathrm{lo}} = 0.046[\ \mathrm{Gd}_{\mathrm{h}}/\mu_{\mathrm{l}}]^{-0.2} \\ &\text{for } \mathrm{Re}_{\mathrm{lo}} < 2000, \quad \mathrm{f}_{\mathrm{lo}} = \mathrm{C}/[\ \mathrm{Gd}_{\mathrm{h}}/\mu_{\mathrm{l}}] \end{aligned} \tag{24}$$

$$C = 16 \text{ for the circular section and } C = 14.3 \text{ for the square section} \tag{25}$$

Cavallini et al. [32] had set up a new test apparatus for heat transfer and fluid flow studies in single minichannels during the condensation and adiabatic flow of R134a and R32 in a single circular section minitube with a much higher surface roughness. The researchers presented new experimental frictional pressure gradient data, relative to single-phase flow and adiabatic two-phase flow of R134a and R32 inside a single horizontal minitube, The test tube was a commercial copper tube with an inner diameter of 0.96 mm and a length of 228.5 mm. The uncertainty associated to the diameter was equal to ±0.02 mm. The arithmetical mean deviation of the assessed profile (*Ra*) of the inner surface was $Ra = 1.3$ μm, the maximum height of profile *Rz* was 10 μm. Because of high values of *Ra* and *Rz*, surface roughness was not negligible. They compared successfully the new all-liquid and all-vapor data against predictions of single-phase flow models. Also, they compared the two-phase flow data against a model previously developed by Cavallini et al. [30] for adiabatic flow or flow during condensation of halogenated refrigerants inside smooth minichannels. They discussed surface roughness effects on the liquid-vapor flow. In this respect, they modified the friction factor in the proposed model previously developed by Cavallini et al. [30] to take into consideration also effects due to wall roughness. The all-liquid friction factor (f_{lo}) of the model of Cavallini et al. [30] was corrected in the following way:

$$\mathrm{f}_{\mathrm{lo}} = 0.046\ \mathrm{Re}_{\mathrm{lo}}^{-0.2} + 0.7 \cdot \mathrm{RR} = 0.046 \left(\frac{\mathrm{Gd}_{\mathrm{h}}}{\mu_{\mathrm{l}}}\right)^{-0.2} + 0.7 \cdot \mathrm{RR} \quad \text{for RR} < 0.0027 \tag{26}$$

The above friction factor (f_{lo}) was in good agreement with the Churchill curve [22] in the range 3000 Re_{lo} 6000.

Park and Hrnjak [33] investigated the carbon dioxide (CO_2) flow condensation heat transfer coefficients and pressure drop in multi-port microchannels made of aluminum having a hydraulic diameter (d_h) of 0.89 mm at low temperatures in horizontal flow conditions. The researchers performed their measurements at mass fluxes (G) from 200 to 800 kg/(m^2.s), saturation temperatures (T_s) of -15 and -25°C, and wall subcooling temperatures from 2 to 4°C. They predicted the flow patterns for experimental conditions using the Akbar et al. [34] and the Breber et al. [35] flow pattern maps before investigating pressure drop and heat transfer. These flow pattern maps demonstrated that the flow patterns were intermittent flow patterns could occur at low and medium vapor qualities for the mass fluxes of 200 kg/(m^2.s), and annular flow patterns in most of flow conditions. Many correlations such as the Thome et al. [36] model could predict the heat transfer coefficients within acceptable error range. Many correlations such as the Mishima and Hibiki model [37] could predict their measured pressure drop relatively well.

Agarwal and Garimella [38] measured condensation heat transfer coefficients and pressure drops for refrigerant R134a flowing through rectangular microchannels with hydraulic diameters (d_h) ranging from 100 μm to 200 μm in small quality increments. These microchannels were fabricated on a copper substrate by electroforming copper onto a mask patterned by X-ray lithography and sealed by diffusion bonding. The researchers measured heat transfer coefficients for 0 < mass quality (x) < 1 for mass fluxes (G) ranging from 200 kg/(m^2.s) to 800 kg/(m^2.s) at several different saturation temperatures. They conducted conjugate heat transfer analyses in conjunction with local pressure drop profiles to obtain accurate driving temperature differences and heat transfer coefficients. They illustrated the influences of mass flux, mass quality, and saturation temperature on condensation heat transfer coefficients and pressure drops through their experiments.

Song et al. [39] reported preliminary results from a new research program for making accurate pressure drop and heat transfer measurements during condensation in microchannels. The researchers used a dummy test section with identical channel and header geometry to that to be used in the main test program. While measuring the vapor flow rate and total heat transfer rate based on coolant measurements, they took the opportunity to make accurate pressure drop measurements. They obtained data for steam and FC72. In addition, they presented approximate comparisons with available pressure drop calculation methods.

Keinath and Garimella [40] investigated R404a condensation in channels diameter of 0.5-3 mm. The researchers obtained quantitative information on flow mechanisms using image analysis techniques on high speed video. They conducted experiments on condensing R404a at vapor qualities (x) of 0.05 to 0.95 at mass fluxes (G) ranging from 200 to 800 kg/(m^2.s). They conducted the experiments at the high pressures representative of actual operation of air-conditioning and refrigeration equipment. Also, they measured pressure drops as a function of operating conditions and geometry during the flow visualization experiments. Quantitative image analysis enabled detailed computation of void fraction and vapor bubble parameters like diameter, length, frequency, and velocity with a high degree of repeatability and with low uncertainties. They documented the effect of operating conditions on the flow patterns, void fractions, and vapor bubble parameters for saturation temperatures (T_s) = 30-60°C. The resulting void fraction models provided closure for heat transfer and pressure drop models that had thus far not been possible in the reported works in the literature.

Fronk and Garimella [41] measured pressure drops and heat transfer coefficients during carbon dioxide (CO_2) condensation in small quality increments in microchannels of 100 < d_h < 200 μm. The researchers measured heat transfer coefficients for 0 < mass quality (x) < 1 for mass flux (G) = 600 kg/(m^2.s) and multiple saturation temperatures. They presented preliminary results for a 300 × 100 μm (15 channels) test section. They used these data to evaluate the applicability of correlations developed for larger hydraulic diameters and different fluids for predicting pressure drops and heat transfer coefficients during carbon dioxide (CO_2) condensation.

Kuo and Pan [42] investigated experimentally condensation of steam in rectangular microchannels with uniform and converging cross-sections and a mean hydraulic diameter (d_h) of 135 μm. The researchers determined the flow patterns, condensation heat transfer coefficient, and two-phase flow pressure drop. They found that heat transfer coefficient was higher for the microchannels with the uniform cross-section design than those with the converging cross-section under condensation in the mist/annular flow regimes, although the latter worked best for draining two-phase fluids composed of liquid water and uncondensed steam, which was consistent with the result of their previous study [43]. Using their experimental results, they developed dimensionless correlations of a two-phase frictional multiplier for the microchannels with both types of cross-section designs and condensation heat transfer for the mist and annular flow regions. Their experimental data agreed well with the obtained correlations, with the maximum mean absolute errors of 6.0% for the condensation heat transfer and 6.4% for the two-phase frictional multiplier.

Goss et al. [44] investigated experimentally the local heat transfer coefficient and pressure drop during the convective condensation of R-134a inside eight round (d = 0.8 mm) horizontal and parallel microchannels over a mass flux range of $57 < G < 125$ kg/(m^2.s) and pressure range of $6.8 < P < 11.2$ bar. The researchers compared their experimental results with correlations and semi-empirical models described in the literature. For the pressure drop, they tested five correlations and that proposed by Zhang and Webb [45] gave the best results, with a deviation of 30%.

Keinath and Garimella [46] used the Garimella et al. [14] model on pressure drop data for R404A in circular tubes with diameter ranging from 0.5 mm to 3.0 mm. The researchers found that this model tended to overpredict the data. They observed the poorest agreement for the 3-mm tube data. They surmised that at d = 3 mm, the various flow regime arising from the significantly higher reduced pressures encountered with R404A compared to R134a accounted for the difference. Also, Andresen [47] found that the Garimella et al. [14] model overpredicted data for R410A at high reduced pressures (P_r = 0.8 to 0.9) for tubes with diameter from 0.76 mm to 3.05 mm. Therefore, while the underlying mechanisms in the model appeared to be sound, further work was required to extend the applicability of the model to very high reduced pressures and somewhat larger diameters than for which the model was developed. Moreover, it could be seen that the annular flow pressure drop using the void fraction model of Baroczy [21] (Eq. (4)) was inversely proportional to the void fraction to the 2.5 power ($\alpha^{2.5}$). This shows the importance of developing void fraction models specifically for condensation of refrigerants in small-diameter channels.

Bohdal et al. [48] investigated experimentally the two-phase pressure drop of the environmentally friendly refrigerant R134a (an R12 substitute) during its condensation in pipe minichannels with internal diameter (d_i) = 0.1-3.3 mm in compact condensers. Pipe minichannels could be used in the construction of compact refrigeration condensers. These were channels with a circular section that in the condensation process of refrigerants intensify the heat transfer process while guaranteeing a high degree of structural compactness. The researchers established local and average pressure drops in the whole range of mass quality (x) = 0-1. They illustrated the effect of the mass quality and mass flux on pressure drop. They compared their results of experimental investigations to the results of calculation according to correlations proposed by other authors. They obtained many experimental results in this range, with was essential to make their owns correlation. They found that the pressure drop in two-phase flow during condensation of refrigerant R134a was dependent on: the agent type, process parameters and the structure of two-phase flow. In the literature, there were no generalized maps of structures for two-phase flow of refrigerants that significantly limited the choice of an appropriate correlation in the design of compact condensers. Therefore, they obtained systematic experimental studies of condensing refrigerant R134a in smooth pipe minichannels (stainless steel) with an inside diameter (d_i) = 0.31-3.3 mm. Their database included measurements of average and local values of pressure drop in pipe minichannels within the range of parameters: saturation temperature (T_s) = 30-40°C, mass quality (x) = 0-1, and refrigerant mass flux (G) = 0-1200 kg/(m^2.s). They used their database to develop their own correlation for determination of pressure drop under these conditions.

Bohdal et al. [49] presented the results of experimental investigations of heat transfer and pressure drop during R134a and R404A condensation in pipe minichannels with internal diameters (d_i) = 0.31-3.30 mm. Their results concerned investigations of the pressure drop and the local heat transfer coefficient in single mini-channels. The researchers found that the values of the pressure drop and the local heat transfer coefficient in the above mentioned minichannels were higher for R134a than R404A. Also, they compared their results with calculations according to the correlations proposed by other authors. They showed that a pressure drop during the condensation of the R134a and R404A refrigerants was described in a satisfactory manner with Friedel [6] and Garimella [13] correlations.

Bohdal et al. [50] investigated experimentally the pressure drop during R134a, R404a and R407C condensation in pipe minichannels with internal diameter (d_i) of 0.31-3.30 mm. Their results concerned investigations of the local and mean pressure drop in single minichannels. The researchers compared the experimental investigations results with the calculations according to the correlations proposed by other authors. They found that a pressure drop during the refrigerants condensation was described in a satisfactory manner with Friedel [6] and Garimella [13] correlations. They proposed their own correlation for calculation of local pressure drop during condensation in single minichannels using their experimental investigations in the range of two-phase flow structures: annular and annular-stratified for the following parameters: refrigerant mass flux (G) = 0-1300 kg/(m^2.s), mass quality (x) = 0-1, and saturation temperature (T_{sat}) = 20-50°C. Their pressure drop correlation was

$$\left(\frac{dP}{dz}\right)_f = \left(\frac{dP}{dz}\right)_{f,lo}\left[0.003\left(\frac{\mathrm{P_{sat}}}{\mathrm{P_{cr}}}\right)^{-4.722}\mathrm{E}^{-0.992} + 143.74\left(\frac{\mathrm{F}^{0.671}\mathrm{H}^{-0.019}}{\mathrm{We}^{0.308}}\right)\right] \tag{27}$$

http://dx.doi.org/10.5772/60965

$$E = (1 - x)^2 + x^2 \left(\frac{\rho_l}{\rho_g} \right) \left(\frac{f_{go}}{f_{lo}} \right) \tag{28}$$

$$F = x^{0.98} (1 - x)^{0.24} \tag{29}$$

$$H = \left(\frac{\rho_l}{\rho_g} \right)^{0.91} \left(\frac{\mu_g}{\mu_l} \right)^{0.19} \left(1 - \frac{\mu_g}{\mu_l} \right)^{0.7} \tag{30}$$

$$We = \frac{G^2 d}{\sigma \rho_g} \tag{31}$$

The friction coefficients f_{lo} and f_{go} for Eq. (28) were determined for a single-phase flow for the liquid and gaseous phases respectively, from the Baroczy dependence [21] of the following form:

$$f_x = 8 \left[\left(\frac{8}{\mathrm{Re}_x} \right)^{12} + \left\{ 2.457 \ln \left[\left(\frac{\mathrm{Re}_x}{7} \right)^{0.9} \right]^{16} + \left(\frac{37530}{\mathrm{Re}_x} \right)^{16} \right\}^{-1.5} \right]^{1/12} \tag{32}$$

where the lower index $x = go$ was applied in the case of the calculation of f_{go} and $x = lo$ for f_{lo}. The same denotations applied to Reynolds' numbers: Re_{lo} and Re_{go}. They found that the results of the experimental tests and calculations from Eq. (27) fell within the compatibility range of ±25%.

Alshqirate et al. [51] obtained the experimental results of the pressure drop and convection heat transfer coefficient during condensation and evaporation of CO_2 at various operating conditions for flow inside micropipes of 0.6, 1.0, and 1.6 mm internal diameter. The Reynolds number (Re_d) range was between 2000 and 15000. The researchers used the dimensional analysis technique to develop correlations for pressure drops and Nusselt numbers. They assumed that the pressure drop of any gas during condensation (ΔP_{cond}) was dependent on the following dimensional variables: the micropipe internal diameter (d_i); the micropipe condenser length (L); micropipe condenser inlet pressure ($P_{in,cond}$); the mean velocity (U_m); and the mean value of the density (ρ_m) and the dynamic viscosity (μ_m). They obtained a general formula for the non-dimensional pressure drop during condensation. Solving for carbon dioxide (CO_2) data generated by their experimental work using the Multiple Linear Regression Method would give the following correlation:

$$\frac{\Delta P_{cond}}{P_{in,cond}} = 1.56 * \left[\left(\frac{1}{Eu * \mathrm{Re}_d} \right)^{0.27} \left(\frac{L}{d_i} \right)^{0.14} \right] \tag{33}$$

$$Eu = \frac{P_{in,cond}}{U_m^2 \rho_m} \tag{34}$$

$$\mathrm{Re}_d = \frac{\rho_m d_i U_m}{\mu_m} \tag{35}$$

It should be noted that the mean values of the properties were defined by liquid and gas properties. For example

$$\rho_m = \frac{\rho_l + \rho_g}{2} \tag{36}$$

$$\mu_m = \frac{\mu_l + \mu_g}{2} \tag{37}$$

Alshqirate et al. [51] carried out a comparison between experimental and correlated results. The results showed that for the condensation process, the bias errors were 0.4% and 5.25% for Nusselt number and pressure drops respectively. Consequently, Average Standard Deviation (ASD) values reached 4.62% and 17.94% for both respectively. On the other hand, the Nusselt number error for the evaporation process was 3.8% with an ASD of 4.14%. Their correlations could be used in calculating heat transfer coefficients and pressure drops for phase change flows in mini and micro tubes. Also, their correlations could help to enhance design calculations of evaporators, condensers and heat exchangers.

Kim and Mudawar [52] examined the heat transfer characteristics and pressure drop of annular condensation in rectangular micro-channels with three-sided cooling walls. The researchers proposed a theoretical control-volume-based model using the assumptions of smooth interface between the vapor core and annular liquid film, and uniform film thickness around the channel's circumference. They applied mass and momentum conservation to control volumes encompassing the vapor core and the liquid film separately. They compared their model predictions with experimental heat transfer and pressure drop data for annular condensation of FC-72 along 1×1 mm^2 parallel channels. The data spanned FC-72 saturation temperatures (T_s) of 57.8-62.3°C, mass fluxes (G) of 248-367 kg/(m^2.s), mass qualities (x) of 0.23-1.0, and water mass flow rates of 3-6 g/s. Also, they compared the data to predictions of previous separated flow mini/micro-channel and macro-channel correlations. Their new model accurately captured the heat transfer coefficient and pressure drop data in both magnitude and trend, evidenced by mean absolute error values of 9.3% and 3.6%, respectively.

In their first part of a two-part study, Kim et al. [53] performed experiments to investigate FC-72 condensation along parallel, square micro-channels with a length (L) of 29.9 cm and a hydraulic diameter (d_h) of 1 mm that were formed in the top surface of a solid copper plate. The operating conditions included FC-72 saturation temperatures (T_s) of 57.2-62.3°C, FC-72 mass fluxes (G) of 68-367 kg/(m^2.s), and water mass flow rates of 3-6 g/s. The researchers identified five distinct flow regimes: smooth-annular, wavy-annular, transition, slug, and bubbly using high-speed video imaging and photomicrographic techniques, with the smooth-annular and wavy-annular regimes being most prevalent. They presented a detailed pressure model that included all pressure drop components across the micro-channel. They examined various sub-models for the frictional and accelerational pressure gradients using the homogenous equilibrium model (with various two-phase friction factor relations) as well as previous macro-channel and mini/micro-channel separated flow correlations. The homogenous flow model provided far more accurate predictions of pressure drop than the separated flow models. They achieved better predictions with those for adiabatic and mini/micro-channels than those for flow boiling and macro-channels among the separated flow models.

Rose and Wang [54] investigated the annular laminar flow pressure drop, or more precisely pressure gradient, during condensation in microchannels. The annular laminar flow was the only flow regime permitting wholly theoretical solution without having recourse to experimental data. The researchers obtained solutions and made comparisons with empirical formulae for void fraction (needed to calculate the momentum pressure gradient) when obtaining the friction pressure gradient from experimentally measured or "total" pressure gradient. They restricted to date calculations and comparisons to one fluid (R134a), one channel section and one flow condition. They found that earlier approximate models for estimating void fraction agreed quite well with the theoretical annular flow solutions. However, there was significant difference between momentum pressure gradients obtained from approximate models used in the earlier investigations and that given by the theoretical annular flow solution that was (numerically) higher than all of them. The annular flow solution indicated that the momentum pressure gradient was not small in comparison with the friction pressure gradient. The friction pressure gradient in the annular flow case was appreciably smaller than given by the earlier correlations.

Fronk and Garimella [55] investigated experimentally pressure drop and heat transfer during Ammonia condensation in a single circular tube of d = 1.435 mm. The researchers chose Ammonia (NH_3) as a working fluid because of its use in thermal systems was attractive due to its favorable transport properties, high latent heat, zero global warming potential (GWP), and zero ozone depletion (ODP) as well as there were few data on ammonia condensation at the microscale while there was a growing body of research on conventional refrigerants condensation (i.e., R134a, R404A, etc.) in microchannels. Ammonia has significantly different fluid properties than synthetic HFC and HCFC refrigerants. For instance, ammonia has an enthalpy of vaporization 7.2 times and a surface tension 3.2 times greater than those of R134a at T_s = 60°C. As a result, models validated with data for synthetic refrigerants might not predict ammonia condensation with sufficient accuracy. They determined two-phase frictional pressure gradient and condensation heat transfer coefficient at multiple saturation temperatures (corresponding to P_r = 0.10-0.23), mass fluxes (G) of 75 and 150 kg/(m^2.s), and in small quality increments (Δx ~ 15-25%) from 0 to 1. They discussed trends in pressure drops and heat transfer coefficients and used the results to assess the applicability of models developed for both macro and microscale geometries for predicting the ammonia condensation. The dependence of heat transfer and pressure drop on mass quality and mass flux was as expected. However, they found that existing models were not able to predict accurately the results. The coupled influences of ammonia properties and microscale geometry were outside the applicable range of most condensation pressure drop and heat transfer models. Additional reliable data pressure drop and heat transfer for smaller tube diameters and with working fluids like ammonia were necessary. These results would enable the development of models, which allow better prediction of condensation over a wider range of working fluids, hydraulic diameters, and operating conditions.

Charun [56] investigated experimentally the heat transfer and pressure drop during the R404A condensation in 1.4-3.30 mm stainless steel pipe minichannels. The researcher provided a review of the present state of knowledge concerning the R404A condensation in conventional channels and in small-diameter channels. He found that there were few prior publications concerning this issue. The test setup is described as well as the results of the experimental tests. He discussed the dependence of the heat transfer coefficient and the pressure drop of the R404A on the minichannel diameter (d), the mass flux (G), and mass quality (x). The pressure drop during the R404A condensation was satisfactorily described by the Friedel [6] and Garimella [13] correlations.

Previous correlations and models for the pressure drop prediction in adiabatic and condensing mini/micro-channel flows had been validated for only a few working fluids and relatively narrow ranges of relevant parameters. Therefore, Kim and Mudawar [57] developed a universal approach for the prediction of pressure drop in adiabatic and condensing mini/micro-channel flows that was capable of tackling many fluids with drastically various thermophysical properties and very broad ranges of all geometrical and flow parameters of practical interest. The researchers amassed a new consolidated database of 7115 frictional pressure gradient data points for both adiabatic and condensing mini/micro-channel flows from 36 sources to achieve this goal. The database consisted of 17 working fluids (air/CO_2/N_2-water mixtures, N_2–ethanol mixture, R12, R22, R134a, R236ea, R245fa, R404A, R410A, R407C, propane, methane, ammonia, CO_2, and water), hydraulic diameters (d_h) from 0.0695 to 6.22 mm, mass fluxes (G) from 4.0 to 8528 kg/(m^2.s), flow qualities (x) from 0 to 1, liquid-only Reynolds numbers (Re_{lo}) from 3.9 to 89,798, and reduced pressures (P_r) from 0.0052 to 0.91. They showed that, while a few prior models and correlations provided fair predictions of the consolidated database, their predictive accuracy was highly compromised for certain subsets of the database. They proposed a universal approach to predict two-phase frictional pressure drop by incorporating appropriate dimensionless relations in a separated flow model to account for both small channel size and various combinations of liquid and vapor states. Their new pressure drop correlation for mini/micro-channels in both single- and multichannel configurations was

$$\left(\frac{dP}{dz}\right)_f = \left(\frac{dP}{dz}\right)_l \phi_l^2 \tag{38}$$

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{39}$$

$$X^2 = \frac{(dP/dz)_l}{(dP/dz)_g} \tag{40}$$

$$\left(\frac{dP}{dz}\right)_l = \frac{2 f_l G^2 (1-x)^2}{\rho_l d_h} \tag{41}$$

$$\left(\frac{dP}{dz}\right)_g = \frac{2 f_g G^2 x^2}{\rho_g d_h} \tag{42}$$

$$f_k = \frac{16}{\mathrm{Re}_k} \quad \mathrm{Re}_k < 2000 \tag{43}$$

$$f_k = \frac{0.079}{\mathrm{Re}_k^{0.25}} \quad 2000 \le \mathrm{Re}_k < 20000 \tag{44}$$

$$f_k = \frac{0.046}{\mathrm{Re}_k^{0.2}} \quad \mathrm{Re}_k \ge 20000 \tag{45}$$

For laminar flow forced convection in rectangular ducts, the Shah and London relation [58] can be used. This relation can be written as a function of the aspect ratio (AR) as follows:

$$\mathrm{f_k Re_k} = 24(1 - 1.3553AR + 1.9467AR^2 - 1.7012AR^3 + 0.9564AR^4 - 0.2537AR^5) \tag{46}$$

where subscript k denotes l or g for liquid and vapor phases, respectively.

In Eq. (46), $AR = 0$ is the case of parallel plates and $f_k Re_k = 24$ in that case while $AR = 1$ is the case of square shape and $f_k Re_k = 14.23$ in that case.

$$\mathrm{Re}_l = \frac{G(1-x)d_h}{\mu_l} \tag{47}$$

$$\mathrm{Re}_g = \frac{Gxd_h}{\mu_g} \tag{48}$$

The different correlations for C for different laminar and turbulent liquid and vapor flow states were

For turbulent liquid-turbulent vapor flow ($Re_l > 2000$ and $Re_g > 2000$)

$$C = 0.39\mathrm{Re}_{\mathrm{lo}}^{0.03}\ \mathrm{Su}_{go}^{0.10}\left(\frac{\rho_l}{\rho_g}\right)^{0.35} \tag{49}$$

For turbulent liquid-turbulent vapor flow ($Re_l > 2000$ and $Re_g > 2000$)

$$C = 8.7 \text{ x } 10^{-4}\mathrm{Re}_{\mathrm{lo}}^{0.17}\ \mathrm{Su}_{go}^{0.50}\left(\frac{\rho_l}{\rho_g}\right)^{0.14} \tag{50}$$

For laminar liquid-turbulent vapor flow ($Re_l < 2000$ and $Re_g > 2000$)

$$C = 0.0015\mathrm{Re}_{\mathrm{lo}}^{0.59}\ \mathrm{Su}_{go}^{0.19}\left(\frac{\rho_l}{\rho_g}\right)^{0.36} \tag{51}$$

For laminar liquid-laminar vapor flow ($Re_l < 2000$ and $Re_g < 2000$)

$$C = 3.5 \text{ x } 10^{-5}\mathrm{Re}_{\mathrm{lo}}^{0.44}\ \mathrm{Su}_{go}^{0.50}\left(\frac{\rho_l}{\rho_g}\right)^{0.48} \tag{52}$$

In the equations of the parameter (C), the Reynolds number for all flow as liquid (Re_{lo}), and the Suratman number for all flow as vapor (Su_{go}) were defined as

$$\mathrm{Re}_{lo} = \frac{G d_h}{\mu_l} \tag{53}$$

$$Su_{go} = \frac{\rho_g \sigma d_h}{\mu_g^2} \tag{54}$$

Kim and Mudawar [57] showed their new two-phase frictional pressure drop correlation predicted the entire 7115 experimental mini/micro-channel database quite accurately, with Mean Absolute Error (MAE) values of 26.3%, 22.4%, 26.8%, and 21.1% for the laminar-laminar (vv), laminar-turbulent (vt), turbulent-laminar (tv), turbulent-turbulent (tt) flow regimes, respectively. These low values of MAE could be attributed to the large database (7115 data points) upon which it was based. Also, their approach was capable of tackling single and multiple channels as well as situations involving significant flow deceleration due to condensation.

Zhang et al. [59] investigated experimentally condensation pressure drop and heat transfer of R22, R410A and R407C in two single round stainless steel tubes with d = 1.088 mm and 1.289 mm. The researchers measured two phase pressure drop and condensation heat transfer coefficients at the saturation temperatures of 30°C and 40°C. The vapor quality (x) varied from 0.1 to 0.9 and the mass flux (G) varied from 300 to 600 kg/(m^2.s). They investigated the influences of vapor quality and mass flux and their results indicated that condensation heat transfer coefficients increased with vapor quality and mass flux, increasing faster in the high vapor quality region. Two phase pressure drop and condensation heat transfer coefficients of R22 and R407C were equivalent but both higher than those of R410A. As a substitute for R22, R410A had more advantages than R407C in view of the characteristics of pressure drop and condensation heat transfer.

Mikielewicz et al. [60] presented a general method for calculation of two-phase flow pressure drop in flow boiling and flow condensation because flow boiling and flow condensation were often regarded as two opposite or symmetrical phenomena, however their description with a single correlation had yet to be suggested. This task was a little easier in the case of flow boiling/ flow condensation in minichannels in comparison to the case of flow boiling/flow condensation in conventional size tubes (diameters greater than 3 mm). This was because they were dealing with two major structures of two-phase flow, namely bubbly flow and annular flow in conventional size tubes while they were dealing with the annular flow structure only in minichannels where the bubble generation/collapse was not present. The difficulty in devising a general method for pressure drop calculations, applicable to both flow condensation and flow boiling, lay in the fact that the non-adiabatic effects were excluded into the present in literature models. In case of bubbly flow the applied heat flux effect was not encountered, similarly the heat flux effect in annular flow was excluded.

The key feature of their method was the approach to model the modification of interface shear stresses in flow boiling and flow condensation due to mass flux and heat flux on interface. In case of annular flow structure incorporation of the so called "blowing parameter" that differentiated these two modes of heat transfer, was considered. The researchers devoted that effect to a correct mass flux modeling on interface. The differences in shear stress between vapor phase and liquid phase was generally a function of non-adiabatic effect. Correct modeling of that heat flux enabled to predict a thinner liquid film thickness in boiling and thicker in condensations at otherwise exactly the same flow conditions. That was a major reason why that up to date approaches, considering the issue of flow boiling and flow condensation as symmetric, were failing in successful predictions. In case of bubbly flow structure the applied heat flux effect was considered. Therefore, a modified form of the two-phase flow multiplier was obtained, in which the non-adiabatic effect was clearly pronounced. They made comparisons with some well established experimental data from literature for many fluids. These data would be carefully scrutinized to extract the applied heat flux effect. Preliminary calculations showed a satisfactory consistency of their model with experimental data. Also, they made comparisons with well established empirical correlations for calculations of heat transfer coefficient. Their calculations showed that their method presented above was universal and could be used to predict heat transfer in flow boiling and flow condensation for various halogeneous refrigerants and other fluids. They mentioned that their model could be suggested for a wider use amongst engineers, but further validation with experimental data would add value to its robustness.

Son and Oh [61] investigated experimentally the condensation pressure drop characteristics for pure refrigerants R22, R134a, and a binary refrigerant mixture R410A without lubricating oil in a single circular microtube. Their test section consisted of 1220 mm length with horizontal copper tube of 3.38 mm outer diameter and 1.77 mm inner diameter. The researchers conducted their experiments at refrigerant mass flux (G) of 450-1050 kg/(m^2.s), and saturation temperature (T_s) of 40°C. For the same mass flux, they found that the condensation pressure drop of R134a was higher than that of R22 and R410A. They compared their experimental data against 14 two-phase pressure drop correlations. They presented a new pressure drop model that was based on a superposition model for refrigerants condensing in the single circular tube. They related the experimental pressure drop during condensation inside the single circular tube to mass flux, inner diameter and thermophysical properties such as surface tension, density and viscosity. Therefore, they developed the Chisholm factor (C) as a function of the two-phase Weber number (We_{tp}), and two-phase Reynolds number (Re_{tp}). Their correlation was

$$\mathrm{C} = \left(\varphi^2 - 1 - \frac{1}{X^2}\right) X = 2485\ \mathrm{We}_{\mathrm{tp}}^{0.407}\ \mathrm{Re}_{tp}^{0.34} \tag{55}$$

$$We_{tp} = \frac{G^2 d}{\sigma \rho_{tp}} \tag{56}$$

$$\mathrm{Re}_{tp} = \frac{Gd}{\mu_{tp}} \tag{57}$$

$$\rho_{tp} = \left[\frac{x}{\rho_g} + \frac{(1 - x)}{\rho_l} \right]^{-1} \tag{58}$$

$$\mu_{tp} = \left[\frac{x}{\mu_g} + \frac{(1 - x)}{\mu_l} \right]^{-1} \tag{59}$$

Based on their experimental database and using a regression method with 108 data points,, their correlation provided a mean deviation of 2.31% and an average deviation of -8.7%.

Zhang et al. [62] presented the heat transfer characteristics of CO_2 condensation in a mini-channel condenser. The condenser consisted of seven tubes in parallel whose inner diameter was 0.9 mm that were thermally connected to two aluminium base-plates by using thermal glue. They obtained the CO_2 condensation heat transfer coefficients, ranging from 1700 to 4500 W/(m^2.K) at saturation temperatures ranging from -5°C to 15°C, with average vapor qualities from 0.2 to 0.8, and, mass fluxes of 180, 360 and 540 kg/(m^2.s), respectively. Also, they found that the measured pressure drop over the condenser increased with the vapor quality and the mass flux, but decreased with the saturation temperature.

Garimella and Fronk [63] conducted a systematic series of experiments on condensation flow regimes, heat transfer, and pressure drop using innovative visualization and measurement techniques for condensation of synthetic and natural refrigerants and their azeotropic and zeotropic mixtures through micro-channels with a wide range of diameters ($0.1 < d_h < 5$ mm), shapes, and operating conditions. These experiments resulted in flow-regime-based pressure drop and heat transfer models with very good predictive capabilities for such micro-channel geometries.

Wang and Rose [64] investigated pressure drop and heat transfer during laminar annular flow condensation in micro-channels. The annular laminar condensate flow permitted wholly theoretical solution without recourse to empirical input. Channel geometry, flow parameters and tube wall temperatures, local pressure gradient, and local heat transfer-coefficient could be calculated as well as local quality and void fraction for laminar annular flow condensation in micro-channels and for specified fluid. The researchers outlined the theory in this article, and discussed recent developments. They summarized and compared results for pressure drop and heat transfer for laminar annular flow condensation in micro-channels with experimental data. They found that correlations of experimental data for both pressure drop and heat transfer could only be expected to have validity for fluids and conditions close to those used when obtaining the data on which the correlations were based. They found that the results for pressure gradient given by the annular laminar flow model were generally lower than those given by the correlations.

Liu et al. [65] presented experimental data for pressure drop and heat transfer during R152a condensation in square and circular microchannels with hydraulic diameters (d_h) of 0.952 mm and 1.152 mm, respectively. Saturation temperatures (T_s) were 40°C and 50°C with vapor mass qualities (x) varying from 0.1 to 0.9 and mass fluxes (G) from 200 to 800 kg/(m^2.s). The researchers investigated effects of vapor mass quality, mass flux, and channel geometry on pressure drop and heat transfer. They found that pressure gradients and heat transfer coefficients during condensation decreased with increasing saturation temperature while increased with increasing vapor mass quality and mass flux both in square and circular microchannels. Channel geometry had little influence on two-phase pressure gradients. They used three pressure drop correlations based on experimental data of R134a in microchannels to predict experimental data. These three pressure drop correlations were correlations of Koyama et al. [8], Agarwal and Garimella [29], and Cavallini et al. [30]. They found that Koyama et al. [8] underestimated the data for both square and circular microchannels while Agarwal and Garimella [29] overestimated the data for the square microchannel. Predictions of Cavallini et al. [30] showed large root-mean-square errors for data in both square and circular microchannels.

Wang et al. [66] calculated the frictional pressure gradient for the laminar annular flow condensation in microchannels. The laminar annular flow was the only flow regime permitting theoretical solution without having recourse to experimental data. The researchers made comparisons with correlations using experimental data for R134a. The correlations were different somewhat among themselves with the highest to lowest predicted friction pressure gradient ratio typically around 1.4 and nearer to 1 at high quality. The frictional pressure gradients given by the laminar annular flow solutions were lower than the correlations at lower quality and in fair agreement with the correlations at high quality. The frictional pressure gradient could not be directly observed and its evaluation from measurements required the nondissipative momentum or acceleration pressure gradient estimation. Methods used to estimate the nondissipative pressure gradient required void fraction and quality together with equations that related these and whose accuracy was difficult to quantify. Void fraction and quality could be readily found from the laminar annular flow solutions. They found significant differences between these and values from approximate equations.

Heo et al. [67] investigated the CO_2 condensation pressure drop and heat transfer coefficient in a multiport microchannel with a hydraulic diameter (d_h) of 1.5 mm with variation of the condensation temperature (T) from −5 to 5°C and of the mass flux (G) from 400 to 1000 kg/(m^2.s). The researchers found that the pressure drop and heat transfer coefficient increased with the increase of mass flux and the decrease of condensation temperature. The gradient of the pressure drop with respect to vapor quality (x) was significant with the mass flux (G) increase. For the pressure drop, they found that the Mishima and Hibiki model [37] showed mean deviation of 29.1%.

Ganapathy et al. [68] presented a numerical model for the simulation of fluid flow characteristics and condensation heat transfer in a single microchannel. The researchers based their model on the volume of fluid approach that governed the hydrodynamics of the two-phase flow. They governed the condensation characteristics using the phenomena physics and excluded any empirical expressions in the formulation. They modified the conventional governing equations for conservation of volume fraction and energy to include source terms, which accounted for the mass transfer at the liquid–vapor interface and the associated release of latent heat, respectively. They modeled a microchannel having characteristic dimension of 100 μm using a two-dimensional computational domain. The working fluid was R134a and t he channel wall was maintained at a constant heat flux (q) ranging from 200 to 800 kW/m^2. The vapor mass flux (G_g) at the channel inlet ranged from 245 to 615 kg/(m^2.s). They assessed the predictive accuracy of their numerical model by comparing the Nusselt number and two-phase frictional pressure drop with available empirical correlations in the literature. They obtained a reasonably good agreement for both parameters with a mean absolute error (MAE) of 8.1% for two-phase frictional pressure drop against a recent universal predictive approach by Kim and Mudawar [57], and 16.6% for Nusselt number against the Dobson and Chato correlation [69].

Heo et al. [70] presented comparison of condensation pressure drop and heat transfer of carbon dioxide (CO_2) in three various microchannels. The channels were rectangular, and the numbers of ports were 7, 19, and 23. The hydraulic diameters (d_h) were 1.5, 0.68, and 0.78 mm for the 7, 19, and 23 ports, respectively. The mass flux (G) range was from 400 to 800 kg/(m^2.s), and the test temperature ranged from −5 to 5°C. The researchers found that the highest pressure drop in the microchannel of 23 ports too. The Mishima and Hibiki model [37] had a mean deviation of ±30.1% for the frictional pressure drop.

Murphy [71] investigated heat transfer and pressure drop during condensation of propane (R290) flowing through minichannels because condensation studies of hydrocarbons are important for applications in the petrochemical industry. For accurate design of heat transfer equipment for use in hydrocarbon processing, insights into the mechanisms of propane condensation are required. The researcher designed and fabricated an experimental facility to measure the frictional pressure drop and heat transfer coefficients during condensation of propane in vertical plain tubes with an inner diameter of 1.93 mm. He took measurements across the vapor-liquid dome in nominal quality increments of 0.25 for two saturation temperatures (47°C and 74°C) and four mass flux conditions (75-150 kg/(m^2.s)). He compared the data to the predictions of relevant correlations in the literature. Also, he used the data from his study to develop models for the frictional pressure drop and heat transfer coefficient based on the measurements and the underlying condensation mechanisms.

Mikielewicz et al. [72] presented investigations of flow condensation with the use of the HFE7100 and HFE 7000 as a working fluids and their own condensation model inside tubes with account of non-adiabatic effects. Their model would be confronted with their own data for a new fluid HFE7000 and HFE 7100. One of the objectives of their study was to add data of HFE7100 and HFE7000 for minichannels because of the lack in published studies. This data was greatly interesting because of the very various thermo physical properties of such fluids compared to other substances commonly tested in minichannels. Another reason for understanding the behavior of two phase flow of the working fluids HFE7100 and HFE7000 was due to increased concerns of ozone depletion(ODP) and GWP (global warming potential), as increased knowledge of the performance of this fluids might contribute to HCFC and HFC refrigerants and might use in many other perspective ecological application like organic Rankine cycles. The researchers used a 2.23 mm circular vertical minichannel to measured both two-phase pressure losses of the fluids HFE7100 and HFE7000. They found satisfactory consistency of discussed model with their own experimental data for condensation. Their presented model could be suggested for a wider use amongst engineers, but further validation with experimental data would add value to its robustness.

Sakamatapan and Wongwises [73] continued the authors' previous work on the condensation of R134a flowing inside a multiport minichannel [74]. The researchers investigated experimentally the pressure drop's characteristics during condensation for R134a flowing inside a multiport minichannel. Two kinds of multiport minichannels having 14 channels, one with a hydraulic diameter (d_h) of 1.1 mm and another with 8 channels with a hydraulic diameter (d_h) of 1.2 mm, were designed as a counter flow tube in a tube heat exchanger. They observed the pressure drop characteristics under mass flux (G) range of 345-685 kg/(m^2.s), heat flux (q) of 15-25 kW/m^2, and saturation temperature (T_{sat}) of 35-45°C. They found that the total pressure drop was dominated by the frictional pressure drop. The frictional pressure gradient increased with the augmentation of mass flux and vapor quality, but the increase of the saturation temperature and channel size led to the frictional pressure gradient decrease. On the other hand, the heat flux had an insignificant influence on the frictional pressure gradient. They conducted the miniscale correlations to predict the frictional pressure gradient, and found that only the multiport minichannel correlations gave a reasonable result. Using the equivalent Reynolds number (Re_{eq}) concept, they proposed a new two-phase friction factor correlation to predict the frictional pressure gradient during condensation. Their correlation was

$$f_{tp} = 6977\,\mathrm{Re}_{eq}^{-0.337}\,x^{-0.031}\left(\frac{\rho_l}{\rho_g}\right)^{6.510}\left(\frac{\mu_l}{\mu_g}\right)^{-11.883} \tag{60}$$

$$\mathrm{Re}_{tp} = \frac{G_{tp}d}{\mu_l} \tag{61}$$

$$G_{tp} = G\left[(1-x) + x\sqrt{\frac{\rho_l}{\rho_g}}\right] \tag{62}$$

López-Belchí et al. [75] studied condensing two-phase flow pressure drop inside a minichannel tube with 1.16 mm inner hydraulic diameter with R1234yf, R134a and R32. According to the available data, most of the models checked capture the trend correctly. The researchers observed the pressure drop characteristics under mass flux (G) range of 350-940 kg/(m^2.s), and saturation temperature (T_{sat}) of 20-55°C. They analyzed experimental data to show the effect of saturation temperature, mass fluxes, vapor quality and fluid properties in pressure drop. Finally, they presented a new correlation model with a mean absolute relative deviation (MARD) value of 8.32% reducing the best correlation MARD by more than 34%. The equations of their model are given below.

$$\left(\frac{dP}{dz}\right)_{tp} = \phi_l^2 \left(\frac{dP}{dz}\right)_l \tag{63}$$

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{64}$$

$$X^2 = \frac{(dP/dz)_l}{(dP/dz)_g} \tag{65}$$

$$\left(\frac{dP}{dz}\right)_l = \frac{2 f_l G^2 (1-x)^2}{\rho_l d} \tag{66}$$

$$\left(\frac{dP}{dz}\right)_g = \frac{2 f_g G^2 x^2}{\rho_g d} \tag{67}$$

$$f_k = \frac{16}{\mathrm{Re}_k} \quad \mathrm{Re}_k < 2000 \tag{68}$$

$$f_k = \frac{1}{16}\left[\log\left(\frac{150.39}{\mathrm{Re}_k^{0.98865}} - \frac{152.66}{\mathrm{Re}_k}\right)\right]^{-2} \quad \mathrm{Re}_k \geq 3000 \tag{69}$$

$$f_k = 0.25(1.1525\mathrm{Re}_k + 895).10^{-5} \quad 2000 \leq \mathrm{Re}_k < 3000 \tag{70}$$

where subscript k denotes l or g for liquid and vapor phases, respectively.

$$\mathrm{Re}_l = \frac{G(1-x)d}{\mu_l} \tag{71}$$

$$\mathrm{Re}_g = \frac{Gxd}{\mu_g} \tag{72}$$

Eq. (69) for the friction factor in turbulent region was confirmed by Fang et al. [76, 77] and Brkic [78] to be the most accurate single-phase friction factor equation flow in smooth tubes.

Eq. (70) for the transition zone was obtained by linear interpolation (Xu and Fang [79]).

The correlation for C was adjusted for best fitting experimental data.

$$C = 4.6468.10^{-6}\left(\frac{P_{sat}}{P_{cr}}\right)^{5.5866} \mathrm{Re}_{l}^{0.4387} \left(\frac{\rho_l}{\rho_g}\right)^{5.7189} X^{0.4243} \tag{73}$$

The experimental tests developed covered the range of, P_{red} = 0.183-0.603, Re_l = 528-8200, ρ_l/ρ_g = 7.03-32.92, X = 0.05-2.53 in a multi-port mini-channel tube with square ports and a hydraulic diameter of 1.16 mm.

Thome and Cioncolini [80] presented unified modeling suite convective boiling and condensation for annular flow in macro- and micro-channels. The researchers presented first unified suite of methods, illustrating in particular, the most recent updates. The annular flow suite included models to predict the entrained liquid fraction, void fraction, the wall shear stress and pressure gradient, and a turbulence model for momentum and heat transfer inside the annular liquid film. In particular, the turbulence model allowed prediction of the local liquid film thicknesses and the local heat transfer coefficients during convective evaporation and condensation. The benefit of a unified modeling suite was that all the included prediction methods were consistently formulated and were proven to work well together, and provided a platform for continued advancement based on the other models in the suite. The annular flow in convective condensation was established almost immediately at the channel inlet and persisted over most of the condensation process until the condensate flooded the channel.

Mikielewicz et al. [81] presented experimental investigations on pressure drop during the condensation in flow of HFE7000 in vertical minichannel of 2.23 mm inner diameter. The researchers scrutinized a new working fluid HFE7000, which had a strongly differing properties in comparison to the other fluids that were commonly used for studies in the minichannels. They observed the pressure drop characteristics under mass flux (G) range of 240-850 kg/(m^2.s), heat flux (q) of 47.2-368.7 kW/m^2, and saturation temperature (T_{sat}) of 35-93°C over the mass quality (x) range of 0-1. They compared their experimental results with the in-house developed model for two-phase flow pressure drop with inclusion of non-adiabatic effects. The comparisons results showed satisfactory agreement.

Kim and Mudawar [82] presented a review of databases and predictive methods for pressure drop in adiabatic, condensing and boiling mini/micro-channel flows. Their study addressed the limited validity of most published methods to a few working fluids and narrow ranges of operating conditions by discussing the development of two consolidated mini/micro-channel databases. The first database was for adiabatic and condensing flows, and consisted of 7115 frictional pressure gradient data points from 36 sources, and the second database for boiling flow, and consisted of 2378 data points from 16 sources. These researchers used these consolidated databases to assess the accuracy of previous models and correlations as well as to develop 'universal' correlations, which were applicable to a large number of fluids and very broad ranges of operating conditions.

Illán-Gómez et al. [83] studied condensing two-phase flow pressure drop gradient and heat transfer coefficient (HTC) inside a mini-channel multiport tube with R1234yf and R134a. The researchers used many models available in the literature to compare predictions of these two fluids. They analyzed experimental data to get the effect of saturation temperature, mass flux, vapor quality and fluid properties. HTC values of R1234yf seemed to be lower than R134a under similar conditions. They proposed a readjusted HTC model. Also, two-phase flow pressure drops were lower in the case of the new refrigerant R1234yf.

Ramírez-Rivera et al. [84] measured experimentally two-phase flow pressure drop of R134a and R32 in condensation and evaporation fluid flow in a multiport extruded aluminium tube (MPEs) with d_h = 0.715 and 1.16 mm. Their experimental conditions range was: mass flux (G) 200-1229 kg/(m^2.s), saturation temperatures (T_s) (5, 7.5, 12.5, 30, 35, 40, 45, 50, 55)°C, heat flux (q) 2.55-70 kW/m^2. The researchers developed two experimental facilities at the Technical University of Cartagena, Spain to study boiling and condensing flow phenomena. They compared their experimental data with some well-known correlations that were developed for macro/mini-channel tubes. Classic macro-channel correlations such as Friedel [6] and Müller-Steinhagen and Heck [85] predicted satisfactorily well their experimental pressure drop data. However, the Souza and Pimenta correlation [86] estimated their experimental pressure gradient data very well with multiport tubes of d_h = 1.16 mm, but failed to predict their experimental data in the tube of d_h = 0.715 mm with R134a and R32. Also, they tested seven correlations specially developed for mini/micro-channels. They found that Cavallini et al. [26] and Zhang and Webb [45] predicted with reasonable accuracy their experimental two-phase flow pressure drop data.

Goss et al. [87] investigated experimentally the pressure drop during the convective condensation of R-134a inside eight round (d = 0.77 mm) horizontal and parallel microchannels. The researchers quantified all pressure drop contributions, including the ones related to expansion, contraction, flow direction change, acceleration, and friction, for microchannel arrangement. Their test conditions included the mass flux (G), vapor quality (x), heat flux (q), and pressure (P), ranging from 230 to 445 kg/(m^2.s), 0.55 to 1, 17 to 53 kW/m^2, and 7.3 to 9.7 bar, respectively. The frictional pressure drop roughly was corresponding to 95% of the net pressure drop. They evaluated the effect of temperature, heat flux, and mass flux on the pressure drop. Their results showed that the pressure drop increased with an increase in mass flux (G) and a decrease in saturation temperature (T_s), whereas it is not affected as much by the heat flux (q). They compared their experimental results with correlations and semi-empirical models described in the literature. Correlations based upon the adiabatic two-phase flows within bore pipes could reasonably predict the pressure drop for condensing microchannel flows. The Cavallini et al. model [26] presented the best prediction performance.

Table 1 presents a summary of the aforementioned previous studies on condensation pressure drop in microchannels and minichannels.

Author	*D*	Fluids	Orientation/ Conditions	Range/ Applicability	Techniques, Basis, Observations
Koyama et al. [8]	1.114 mm (8 channels) 0.807 mm (19 channels)	R-134a	Horizontal	G = 100-700 kg/(m^2.s) x = 0-100%	The experimental data of frictional pressure drop agreed with the correlation of Mishima and Hibiki [9], while the correlations of Chisholm and Laird [10], Soliman et al. [11], and Haraguchi et al. [12], overpredicted.
Garimella [13]	0.4-4.91 mm	R-134a	Horizontal	G = 150-750 kg/(m^2.s) x = 0-100%	His flow regime-based models yield substantially better pressure drop predictions than the traditionally used correlations that were primarily based on air-water flows for large diameter tubes.
Garimella et al. [14]	0.5-4.91 mm	R-134a	Horizontal Considering the intermittent only, intermittent/discrete wavy annular transition, and annular only flow	G = 150-750 kg/(m^2.s) x = 0-100%	Presenting a multiple flow-regime model for pressure drop during condensation.
Haui and Koyama [25]	1.31 mm	CO_2	Horizontal	P = 6.48-7.3 MPa T_{inlet} = 21.63- 31.33°C q = 1.10-8.12 kW/m^2 G = 123.2-315.2 kg/(m^2.s) x = 0-100%	The pressure drop was very small along the test section. The existing model failed to predict the experimental data.
Cavallini et al. [26]	0.4-3 mm	high pressure (R410A), medium (R134a) and low pressure (R236ea)	Horizontal		No model is able to predict the frictional pressure gradient of the high pressure fluid R410A, several models accurately predict the medium pressure fluid R134a and a few satisfactorily estimate

Author	*D*	Fluids	Orientation/ Conditions	Range/ Applicability	Techniques, Basis, Observations
					the low pressure refrigerant R236ea.
Chowdhury et al. [27]	0.7 mm	R-134a	Horizontal	$AR = 7$ $T_{sat} = 30\,^{o}C$ $x = 20\text{-}80\%$ $G = 130, 200\ kg/(m^2.s)$	A unique process for fabrication of the microchannel involving milling and electroplating steps was adopted to maintain the channel geometry close to design values.
Garimella [28]	1 mm	R-134a		$G = 300\ kg/(m^2.s)$ $x = 50\%$ $P = 1500\ kPa$	Comparing different techniques for predicting the pressure drop during condensation.
Agarwal and Garimella [29]	0.42- 0.8 mm	R-134a	Horizontal	$G = 150\text{-}750\ kg/(m^2.s)$ $x = 0\text{-}100\%$	Their resulting model predicted 80% of the data within ±25%. The tube shape effect on pressure drop was demonstrated.
Cavallini et al. [30]	0.96 mm	R134a	Horizontal		Presenting a model for calculation of the frictional pressure gradient during condensation or adiabatic liquid-gas flow inside minichannels with different surface roughness.
Cavallini et al. [32]	0.96 mm	R134a and R32	Horizontal		Presenting modification of the friction factor in the proposed model previously developed by Cavallini et al. [30] to take into consideration also effects due to wall roughness.
Park and Hrnjak [33]	0.89 mm	CO_2	Horizontal	$T_{sat} = -15, -25\,^{o}C$ $G = 200\text{-}800\ kg/(m^2.s)$	Many correlations could predict their measured values of pressure drop relatively well such as the

Author	***D***	**Fluids**	**Orientation/ Conditions**	**Range/ Applicability**	**Techniques, Basis, Observations**
					Mishima and Hibiki model [37].
Agarwal and Garimella [38]	100-200 µm	R134a	Horizontal	G = 200-800 kg/(m^2.s) x = 0-100% T_{sat} = 30, 40, 50, 60 °C	The pressure drop increased with increasing vapor quality, increasing mass flux and decreasing saturation temperature.
Song et al. [39]	1.5 mm x 1.0 mm	FC72 and steam	Horizontal	q = 130-170 kW/m^2 for steam q = 10-30 kW/m^2 for FC72	Presenting preliminary results from a new research program for making accurate heat transfer and pressure drop measurements during condensation in microchannels.
Keinath and Garimella [40]	0.5-3 mm	R404a	Horizontal	G = 200-800 kg/(m^2.s) x = 5-95% T_{sat} = 30-60 °C	Presenting a novel and accurate methodology for the quantitative investigation of two-phase flow regimes and flow parameters during condensation in minichannels.
Fronk and Garimella [41]	100-200 µm	CO_2	Horizontal	G = 600 kg/(m^2.s) x = 0-100% T_{sat} = 15, 20 °C	Using the collected data to evaluate the applicability of correlations developed for larger hydraulic diameters and various fluids for predicting condensation heat transfer and pressure drop of CO_2.
Kuo and Pan [42]	135 µm	steam	Horizontal	2.10×10^{-6}-9.11×10^{-6} kg/s of steam for the uniform cross-section microchannel 2.10×10^{-6}-5.93×10^{-6} kg/s of steam for the converging microchannel	Their experimental data agreed well with the obtained correlations, with the maximum mean absolute errors of 6.4% for the two-phase frictional multiplier.

Author	D	Fluids	Orientation/ Conditions	Range/ Applicability	Techniques, Basis, Observations
Goss et al. [44]	0.8 mm	R134a	Horizontal	G = 57-125 kg/(m^2.s) P = 6.8-11.2 bar	The pressure drop correlation proposed by Zang and Webb [45] gave the best results, with a deviation of 30%.
Keinath and Garimella [46]	0.5-3.0 mm	R404A			The Garimella et al. [14] model tends to overpredict the pressure drop data. The poorest agreement is for the 3-mm tube data.
Bohdal et al. [48]	0.1-3.30 mm	R134a	Horizontal	G = 0-1200 kg/(m^2.s) x = 0-100% T_{sat} = 30-40 °C	The pressure drop in two-phase flow during R134a condensation is dependent on: the agent type, process parameters and the structure of two-phase flow.
Bohdal et al. [49]	0.31-3.30 mm	R134a and R404A	Horizontal	G = 100-1300 kg/(m^2.s) x = 0-100% T_{sat} = 20-40 °C	The pressure drop during the condensation of the R134a and R404A refrigerants is described in a satisfactory manner with Friedel [6] and Garimella [13] correlations.
Bohdal et al. [50]	0.31-3.30 mm	R134a, R404a and R407C	Horizontal	G = 0-1300 kg/(m^2.s) x = 0-100% T_{sat} = 20-50 °C	The pressure drop during the condensation of the R134a, R404a and R407C refrigerants is described in a satisfactory manner with Friedel [6] and Garimella [13] correlations.
Alshqirate et al. [51]	0.6, 1.0, and 1.6 mm	CO_2		Re_d = 2000-15000	Using the dimensional analysis technique to develop correlations for Nusselt numbers and pressure drops.
Kim and Mudawar [52]	1 mm	FC-72 and water (counter flow)	Horizontal	For FC-72, G = 248-367 kg/(m^2.s) T_{sat} = 57.8-62.3 °C x = 23-100%	Their new model accurately captured the pressure drop and heat transfer coefficient data in both magnitude and

Author	***D***	**Fluids**	**Orientation/ Conditions**	**Range/ Applicability**	**Techniques, Basis, Observations**
				For water, mass flow rate = 3-6 g/s	trend, evidenced by mean absolute error values of 3.6% and 9.3%, respectively.
Kim et al. [53]	1 mm	FC-72 and water (counter flow)	Horizontal	For FC-72, G = 248-367 kg/(m^2.s) T_{sat} = 57.2-62.3 ^{o}C q = 4.3-32.1 kW/m^2 For water, G = 69-138 kg/(m^2.s)	The homogenous flow model provides far more accurate predictions of pressure drop than the separated flow models. Among the separated flow models, Kim et al. [53] achieve better predictions with those for adiabatic and mini/micro-channels than those for flow boiling and macro-channels.
Rose and Wang [54]		R134a	laminar annular flow		The momentum pressure gradient is not small in comparison with the friction pressure gradient. The friction pressure gradient in the annular flow case is appreciably smaller than given by the earlier correlations.
Fronk and Garimella [55]	1.435 mm	Ammonia (NH_3)	Horizontal	G = 75-150 kg/(m^2.s) T_{sat} = 30, 40, 50, 60 ^{o}C (corresponding to P_r = 0.10-0.23)	The coupled influences of ammonia properties and microscale geometry were outside the applicable range of most condensation pressure drop and heat transfer models. Additional reliable data pressure drop and heat transfer for smaller tube diameters and with working fluids like ammonia were necessary.
Charun [56]	1.4, 1.6, 1.94, 2.3 and 3.3 mm	R404A	Horizontal	G = 97-902 kg/(m^2.s) x = 0-100%	The pressure drop during the R404A refrigerant condensation is satisfactorily described by the Friedel [6]

Author	*D*	Fluids	Orientation/ Conditions	Range/ Applicability	Techniques, Basis, Observations
					and Garimella [13] correlations.
Kim and Mudawar [57]	0.0695- 6.22 mm	17 various working fluids (air/CO_2/N_2-water mixtures, N_2–ethanol mixture, R12, R22, R134a, R236ea, R245fa, R404A, R410A, R407C, propane, methane, ammonia, CO_2, and water)		G = 4.0-8528 kg/(m^2.s) Re_{lo} = 3.9-89798 x = 0-100% P_r = 0.0052-0.91	Proposing a new universal approach to predict two-phase frictional pressure drop for adiabatic and condensing mini/micro-channel flows.
Zhang et al. [59]	1.088 and 1.289 mm	R22, R410A and R407C	Horizontal	G = 300-600 kg/(m^2.s) T_{sat} = 30, 40 ^{o}C x = 10-90%	Two phase pressure drop and condensation heat transfer coefficients of R22 and R407C are equivalent but both higher than those of R410A. R410A a s a substitute for R22, has more advantages than R407C in view of the characteristics of condensation pressure drop and heat transfer.
Mikielewicz et al. [60]					Presenting a general method for calculation of two-phase flow pressure drop in flow boiling and flow condensation because flow boiling and flow condensation are often regarded as two opposite or symmetrical phenomena, however their description with a single correlation have yet to be suggested.

Author	***D***	**Fluids**	**Orientation/ Conditions**	**Range/ Applicability**	**Techniques, Basis, Observations**
Son and Oh [61]	1.77 mm	R22, R134a and R410A		G = 450-1050 kg/(m^2.s) T_{sat} = 40 ^{o}C	The condensation pressure drop of R134a is higher than that of R22 and R410A for the same mass flux. Presenting a new pressure drop model where the Chisholm factor (C) is a function of the two-phase Weber number (We_{tp}), and two-phase Reynolds number (Re_{tp}).
Zhang et al. [62]	0.9 mm	CO_2		T_{sat} = -5-15 ^{o}C G = 180, 360 and 540 kg/(m^2.s) x = 20-80%	The measured pressure drop over the condenser increases with the mass flux and the vapor quality, but decreases with the saturation temperature.
Garimella and Fronk [63]	0.1 < d_h < 5 mm	synthetic and natural refrigerants and their azeotropic and zeotropic mixtures			These experiments resulted in flow-regime-based heat transfer and pressure drop models with very good predictive capabilities for such micro-channel geometries.
Wang and Rose [64]		R134a, ammonia (NH_3)	laminar annular flow		Results for pressure gradient given by the annular laminar flow model are generally lower than those given by the correlations.
Liu et al. [65]	1.152 mm (circular) 0.952 mm (square)	R152a	Horizontal	T_{sat} = 40-50 ^{o}C G = 200-800 kg/(m^2.s) x = 10-90%	Channel geometry has little effect on frictional pressure gradients. Koyama et al. [8] underestimates the square and circular microchannels data while Agarwal and Garimella [29] overestimate the square microchannel data. Predictions of Cavallini et al. [30] show

Author	D	Fluids	Orientation/ Conditions	Range/ Applicability	Techniques, Basis, Observations
					large root-mean-square errors for data in both square and circular microchannels.
Wang et al. [66]		R134a	laminar annular flow		The frictional pressure gradients given by the laminar annular flow solutions are in fair agreement with the correlations at high quality and lower than the correlations at lower quality.
Heo et al. [67]	1.5 mm	CO_2	Horizontal	G = 400-1000 kg/(m^2.s) T_{sat} = -5-5 °C	The Mishima and Hibiki model [37] showed mean deviation of 29.1%.
Ganapathy et al. [68]	100 µm	R134a	constant heat flux	G_g = 245-615 kg/(m^2.s) q = 200-800 kW/m^2	Using the volume of fluid approach. The mean absolute error (MAE) is 8.1% for two-phase frictional pressure drop against a recent universal predictive approach by Kim and Mudawar [57].
Heo et al. [70]	1.5, 0.78, and 0.68 mm for the 7, 23, and 19 ports	CO_2	Horizontal	G = 400-800 kg/(m^2.s) T_{sat} = -5-5 °C	The Mishima and Hibiki model [37] has a mean deviation of ±30.1% for the frictional pressure drop.
Murphy [71]	1.93 mm	Propane (R290)	Vertical	T_{sat} = 47°C and 74°C G = 75-150 kg/(m^2.s)	The results and the corresponding correlations contribute to the understanding of condensation of hydrocarbon.
Mikielewicz et al. [72]	2.23 mm	HFE7000 and HFE7100	Vertical		Satisfactory consistency of discussed model with their own experimental data for condensation has been found.

Author	***D***	**Fluids**	**Orientation/ Conditions**	**Range/ Applicability**	**Techniques, Basis, Observations**
					The presented model can be suggested for a wider use amongst engineers, but further validation with experimental data would add value to its robustness.
Sakamatapan and Wongwises [73]	1.1 mm (14 channels), 1.2 mm (8 channels)	R134a	Horizontal	G = 345-685 kg/(m^2.s) q = 15-25 kW/m^2 T_{sat} = 35-45°C	Proposing a new two-phase friction factor correlation using the equivalent Reynolds number (Re_{eq}) concept to predict the frictional pressure gradient during condensation.
López-Belchí et al. [75]	1.16 mm	R1234yf, R134a and R32	Horizontal	G = 350-940 kg/(m^2.s) T_{sat} = 20-55°C	Presenting a new correlation model with a mean absolute relative deviation (MARD) value of 8.32% reducing the best correlation MARD by more than 34%.
Thome and Cioncolini [80]			Annular flow		Presenting unified modeling suite for annular flow, convective boiling and condensation in macro- and micro-channels.
Mikielewicz et al. [81]	2.23 mm	HFE7000	Vertical	G = 240-850 kg/(m^2.s) q = 47.2-368.7 kW/m^2 T_{sat} = 35-93°C x = 0-100%	The comparisons of the experimental results with the in-house developed model for two-phase flow pressure drop with inclusion of non-adiabatic effects show satisfactory agreement.
Kim and Mudawar [82]					Presenting a review of databases and predictive methods for pressure drop in adiabatic, condensing and boiling mini/micro-channel flows.

Author	D	Fluids	Orientation/ Conditions	Range/ Applicability	Techniques, Basis, Observations
Illán-Gómez et al. [83]	1.16 mm	R1234yf, and R134a	Horizontal	G = 350-940 kg/(m^2.s) T_{sat} = 20-55°C q = 4.37-20.52 kW/m^2 for R1234yf q = 5.08-20.75 kW/m^2 for R134a x = 12-87%for R1234yf x = 13-89%for R134a	Pressure drop for R1234yf is by 5-7% lower than for R134a. The existing models are able to predict frictional pressure drop reasonably well.
Ramírez-Rivera et al. [84]	0.715 and 1.16 mm	R134a and R32	Horizontal	For condensing flow G = 200-800 kg/(m^2.s) T_{sat} = 30, 35, 40, 45, 50, 55°C q = 2.55-70 kW/m^2	Friedel [6] and Müller-Steinhagen and Heck [85] predict satisfactorily well the experimental pressure drop data. The Souza and Pimenta correlation [86] estimates the experimental pressure gradient data very well with multiport tubes of d_h = 1.16 mm, but fails to predict the experimental data in the tube of d_h = 0.715 mm with R134a and R32. The Cavallini et al. model [26] presents the best prediction performance. Cavallini et al. [26] and Zhang and Webb [45] predicted with reasonable accuracy the experimental two-phase flow pressure drop data.
Goss et al. [87]	0.77 mm	R134a	Horizontal	G = 230-445 kg/(m^2.s) x = 55-100% q = 17-53 kW/m^2 P = 7.3 to 9.7 bar	The pressure drop increases with an increase in (G) and a decrease in saturation temperature (T_s), whereas it is not influenced as much by the heat flux (q).

Table 1. Summary of previous studies on condensation pressure drop in microchannels and minichannels

3. Recommendations for future studies

Finally, recommendations for future studies will be given. These new points can be expected to be the research focus in the coming years. Studying the condensation pressure drop in microscales can be done using:

1. The experiments with new kinds of non-circular shapes like trapezoidal, elliptical,.., etc. To the best of the authors' knowledge, the study of condensation pressure drop in microscales of elliptic cross-section is not yet tackled in literature. Only recently a new interest has been devoted to the elliptical cross-section, produced by mechanical fabrication in metallic microchannels for practical applications in MEMS. Also, the experimental study of the condensation pressure drop in microscales can be done with using various triangular cross sections such as right isosceles triangular.

2. The experiments with new kinds of environmentally friendly refrigerants such as HDR-14, which is low global warming fluid for replacement of R245fa. The GWP is an index used to compare the potential of gases to produce a greenhouse effect and the reference is CO_2 with a value of 1. HDR-14 has a global warming potential (GWP) of only 7, much lower than the value of 930 of R245fa (both considering a period time horizon of 100 years). Also, HDR-14 has a much lower atmospheric life time (0.1 year) in comparison with the atmospheric life time of R245fa (7.6 years).

3. The experiments with oil in the refrigerant loop (refrigerant/lubricant mixture) at various concentrations. For example, Akhavan-Behabadi et al. [88] utilized polyolester oil (POE) as the lubricant in R600a/POE mixture to study experimentally the heat transfer characteristics of flow condensation.

4. The experiments with new types of working fluids such as refrigerant/lubricant/nanoparticles mixture. For example, Akhavan-Behabadi et al. [88] used R600a/ polyolester oil (POE)/CuO nano-refrigerant to study experimentally the heat transfer characteristics of flow condensation. Nano-refrigerant is a type of nanofluid where a refrigerant is used as the base fluid [89]. Recently, Dalkiliç and co-workers [90, 91] presented a review paper on nanorefrigerants.

5. The experiments with new types of refrigerant blends. For instance, new refrigerant blends, like R-417A, are becoming very important due to the possibility of using them in R-22 systems with only minor changes (drop-in refrigerants) [92]. R-417A is the composition with mass fractions of 46.6% R-125, 50% R-134a, and 3.4% R-600. Also, we can carry out the experiments with new kinds of refrigerant blends like R1234yf and R1234ze(E) as constituents as well as blends of old refrigerants with Hydro-Fluoro-Olefin (HFO) refrigerants. These blends can be azeotropic blends or zeotropic blends like zeotropic mixture R32/R1234ze(E). R32 (CH_2F_2) is flammable and has for this reason not been used pure.

6. Similar to recent work on condensation in macroscales at microgravity conditions [93, 94], studies on condensation pressure drop in microscales at microgravity conditions can

http://dx.doi.org/10.5772/60965

be done. These studies will be important because future manned space missions will be expected to greatly increase the space vehicle's weight, size, and heat dissipation requirements. An effective means to reducing both weight and size is replacing single-phase thermal management systems with two-phase counterparts that capitalize upon both sensible and latent heat of the coolant rather than sensible heat alone. This shift is expected to yield orders of magnitude enhancements in condensation heat transfer coefficients. A major challenge to this shift is the reliable tools lack for accurate prediction of heat transfer coefficient and two-phase pressure drop in reduced gravity.

4. Summary and conclusions

This paper provides a comprehensive, up-to-date review in a chronological order on the research progress made on condensation pressure drop in microscales. Also, studies on condensation pressure drop in microscales are summarized in a table. At the end, some suggestions for future work are presented. Therefore, the present study can not only be used as the starting point for the researcher interested in condensation pressure drop in microscales, but it also includes recommendations for future studies on condensation pressure drop in microscales.

Nomenclature

a ; constant, Eq. (5)

a ; geometry-dependent constant, Eq. (12)

a ; exponent, Eq. (10)

A ; constant, Eq. (10)

AR ; aspect ratio (-)

b ; constant, Eq. (5)

b ; geometry-dependent constant, Eq. (13)

b ; exponent, Eq. (10)

c; constant, Eq. (5)

c ; exponent, Eq. (10)

C ; coefficient in Lockhart-Martinelli parameter (-)

C_p; specific heat, J/(kg.K)

d ; diameter, m

E ; entrainment ratio, Eq. (21)

E ; parameter, Eq. (28)

Eu ; Euler number (-)

f ; Fanning friction factor (-)

F ; parameter, Eq. (19)

F ; parameter, Eq. (29)

G ; mass flux, kg/(m^2.s)

H ; parameter, Eq. (20)

H ; parameter, Eq. (30)

J_g ; dimensionless gas velocity = $xG/[gd_h\rho_g(\rho_l-\rho_g)]^{0.5}$

L; channel length, m

N; total number of unit cells

P; pressure, Pa

Ra ; arithmetical mean deviation of the assessed profile (according to ISO 4287: 1997), μm

Rz ; maximum height of profile (according to ISO 4287: 1997), μm

Re ; Reynolds number (-)

RR ; relative roughness of the tube, Eq. (26)

Su; Suratman number (-)

T ; temperature, ^{o}C

U ; velocity, m/s

W ; parameter, Eq. (17)

We ; Weber number (-)

x ; mass quality (-)

X ; Lockhart-Martinelli parameter (-)

Z ; parameter, Eq. (18)

Greek Symbols

α; void fraction (-)

Δ; difference

ϕ^2 ; two-phase frictional multiplier (-)

μ; dynamic viscosity, kg/m.s

ϱ; density, kg/m^3

σ; surface tension, N/m

ψ; dimensionless surface tension term (-)

Subscripts

bubble; bubble

cond; condenser

cr; critical

d; diameter

eq; equivalent

g; gas

gc ; gas core

go ; gas phase with total mass flow rate

h ; hydraulic

i; inner

in; inlet

l; liquid

lo ; liquid phase with total mass flow rate

m; mean

sat; saturation

slug ; slug

transition transition

tube; tube

UC; unit cell

unit cell unit cell

tp; two-phase

Acknowledgements

Professor Ahmet Selim Dalkilic wishes to thank King Mongkut's University of Technology Thonburi (KMUTT) for providing him with a Post-doctoral fellowship. Professor Somchai Wongwises wishes to thank the "Research Chair Grant" National Science and Technology

Development Agency, the Thailand Research Fund and the National Research University Project for the support.

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