Minichannel condensation in downward, upward and horizontal configuration

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Abstract. An experimental investigation of condensation of R134a inside a single square cross section minichannel when varying the channel orientation is presented. Local heat transfer coefficients are measured in horizontal, vertical downflow and vertical upflow configurations. In the literature the number of local heat transfer coefficient values measured during condensation inside non-circular minichannels is rather limited and the effect of channel orientation during condensation is not much investigated. Some studies have been performed in inclined smooth tubes of larger diameters, where it was shown that the heat transfer coefficient is strongly affected by the liquid and vapour distributions. But minichannels may display a different behaviour because of the relative importance of shear stress, gravity and surface tension. The action of these forces may depend on operating conditions and orientation. In the present study, the channel is obtained from a copper rod and has a square cross section with 1.18 mm side length. Each corner has a curvature radius equal to 0.15 mm, which leads to a hydraulic diameter equal to 1.23 mm. Tests have been performed with R134a at 40°C saturation temperature, at mass velocity ranging between 100 and 790 kg m\(^{-2}\) s\(^{-1}\). From the experimental results, the effect of the channel inclination when varying mass velocity and vapour quality is investigated.

1. Introduction
While the condensation process has been extensively studied in conventional size channels, a complete understanding of the heat transfer mechanism during condensation inside minichannels has not been yet achieved. Minichannels allow high heat transfer rate in reduced size heat exchangers; such devices have been employed in a wide range of applications, from the air conditioning in residential buildings, to the electronics and aerospace industry. Some environmental benefits are also achieved using minichannels: the reduced charge of refrigerant minimize the problems of release of potentially hazardous fluids in the atmosphere and allow using natural fluids such as hydrocarbons. Furthermore, small diameter channels can be used with high pressure fluids, such as carbon dioxide in transcritical cycle equipments.

In the literature the number of local heat transfer coefficient values measured during condensation inside non-circular minichannels is rather limited and the effect of channel orientation during condensation is not much investigated. Some studies have been performed in inclined smooth tubes of larger diameters, where it was shown that the heat transfer coefficient is strongly affected by the liquid and vapour distributions. But minichannels may display a different behavior because of the relative importance of shear stress, gravity and surface tension at the corners and the interaction between the
liquid and vapour phases. The action of these forces may depend on operating conditions and orientation. At some operating conditions, the heat transfer coefficients could be dependent on the effect of surface tension: the liquid is pulled towards the corners leading to a thinner liquid film on the flat sides and therefore to a lower thermal resistance on these parts of the channels. This effect may enhance the heat transfer in the presence of corners, as compared to the case of circular minitubes, at low mass velocity, when the relative importance of shear stress diminishes.

In the open literature only few studies consider the effect of inclination. Furthermore, most of those refer to tubes with larger diameter as compared to minichannels.

Lips and Meyer [1] presented an experimental research on convective condensation of R134a at 40°C saturation temperature using a 8.38 mm inner diameter channel. The inclination varies between vertical downward to vertical upward and the mass velocity \( G \) ranges from 200 to 600 kg m\(^{-2}\) s\(^{-1}\). The heat transfer coefficient is found to be dependent on the distribution of the two phases inside the tube; the flow pattern results from the balance between gravitational force, shear stress and surface capillary force. The authors show that for mass velocity \( G = 200 \) kg m\(^{-2}\) s\(^{-1}\) an optimum inclination angle of 15° in downward flow is found, which leads to an increase of 20% of heat transfer coefficient compared to the horizontal position. At high mass velocities, the shear stress is the dominant force and there is no effect of inclination on the heat transfer coefficient.

Both theoretical and experimental results are presented by Wang and Du [2] for condensation of water inside circular channels with inner hydraulic diameter ranging between 1.94 and 4.98 mm. They performed tests at different inclination (0°, 17°, 34°, 45°) and at low mass fluxes (up to 100 kg m\(^{-2}\) s\(^{-1}\)). The results show that the effect of inclination on heat transfer coefficient was poor in the smallest tubes and strong in the tube with bigger diameters.

Saffari and Naziri [3] presented a theoretical and numerical analysis of heat transfer during stratified condensation inside inclined tubes. The results are presented for three different refrigerants: R141b, R11 and R134a. The inclination angle is shown to have a significant effect on condensation heat transfer coefficient and the optimum inclination angle was found in the range 30°-50° from the horizontal position in upflow configuration.

An experimental analysis was conducted by Lyulin et al. [4] to investigate the heat transfer coefficient during convective condensation of pure ethanol vapour inside a smooth tube with an inner diameter of 4.8 mm and a length of 200 mm. During the tests mass velocity varied from 0.24 to 2.04 kg m\(^{-2}\) s\(^{-1}\) and the saturation temperature was fixed at 58°C. The study focused both on the difference between the wall to saturation temperature and the influence of inclination angles. The results show that the heat transfer coefficient reduces with the increase of the wall to saturation temperature difference; the dependency of the heat transfer coefficient on inclination has a maximum in the range 15°-35° due to the complex gravity drainage mechanism of the condensed liquid.

Da Riva and Del Col [5] simulated annular condensation inside a circular cross section minichannel with 0.96 mm inner diameter assuming horizontal orientation, vertical orientation downflow and zero-gravity conditions. Using R134a as working fluid, at \( G = 100 \) kg m\(^{-2}\) s\(^{-1}\) much higher heat transfer coefficients are obtained in the horizontal configuration as compared to the vertical one. At \( G = 800 \) kg m\(^{-2}\) s\(^{-1}\) all the simulation cases display almost identical result and the shear stress in found to be the dominant force.

Bortolin et al. [6] presented a number of steady-state simulations of condensation of R134a at mass fluxes \( G = 400 \) kg m\(^{-2}\) s\(^{-1}\) and \( G = 800 \) kg m\(^{-2}\) s\(^{-1}\) inside a 1 mm square cross section minichannel. The gravity force has a minor effect on the liquid-vapour interface distribution in the square minichannel; the authors found that at these mass velocities, the heat transfer mechanism is dominated by shear stress and surface tension.

In the present study, an experimental investigation of condensation of R134a inside a single square cross section minichannel when varying the channel orientation is presented. Local heat transfer coefficients are measured in horizontal configuration, vertical downflow and vertical upflow configurations. The channel has a square cross section with 1.18 mm side length. Each corner has a curvature radius equal to 0.15 mm, which leads to a hydraulic diameter equal to 1.23 mm.
Tests have been performed with R134a at 40°C saturation temperature, at mass velocity ranging between 100 and 790 kg m$^{-2}$ s$^{-1}$. From the experimental results, the effect of the channel inclination when varying mass velocity and vapour quality is investigated.

2. Experimental apparatus

A schematic representation of the test rig for the experimental runs reported in this work is depicted in Fig. 1. After passing through the test section, the working refrigerant is subcooled in the post condenser by using brine at about 5°C which flows inside an auxiliary loop served by a thermal bath. The subcooled refrigerant is sent through a filter dryer into an independently controlled gear pump; a Coriolis-effect mass flow meter is used to measure mass flow. The evaporator is a tube-in-tube heat exchanger in which the fluid is heated by using hot water flowing in a closed auxiliary loop provided with PID-controlled electrical heaters so that the refrigerant enters the test section as superheated vapour. The test section consists of a pre-sector used to achieve the desired conditions at the inlet of the actual measuring sector. Both the pre-sector and the measuring sector are counter-flow heat exchangers where the refrigerant is cooled by distilled water provided by a second thermal bath at desired temperature.

![Figure 1. Experimental test rig: PS (pre-sector); MF (mechanical filter); FD (filter dryer); PV (pressure vessel); CFM (Coriolis-effect mass flow meter); TV (valve); P (pressure transducer); DP (differential pressure transducer); T (thermocouple).](image)

The test section is designed to achieve an improved precision in the evaluation of condensation heat transfer coefficients. The minichannel is obtained from a copper rod soldered together with stainless steel rods, which are used as adiabatic sectors to achieve a good thermal separation between the pre-sector and the measuring sector. Copper and steel rods have been internally holed by electroerosion to obtain a square cross section with 1.18 mm side length (Fig. 2, top). Each corner has a curvature radius equal to 0.15 mm, which leads to a hydraulic diameter of 1.23 mm. The measuring sector is 225 mm long. These dimensions have been checked from the measurements of the laminar flow friction factor in Del Col et al. [7]. On the outer tube surface of the adiabatic stainless steel sectors, the temperature of the refrigerant at the inlet and at the outlet of the measuring sector can be measured with good accuracy; the pressure taps are also placed in the adiabatic segments and they are connected to digital strain gauge pressure (absolute and differential) transducers through warmed pressure lines (Fig. 2, bottom).

The coolant path consists of grooves obtained by machining the copper rod: this particular coolant channel is externally covered by a film of epoxy resin that has been holed in some places to insert the
thermocouples for the measurement of the coolant temperature profile during condensation. This geometry allows the accurate measurement of the water temperature along the channel. Besides, other thermocouples have been inserted in the copper fins to measure the wall temperature without passing through the coolant. On the whole, the measuring sector has been provided with more than sixty T-type thermocouples. The local heat flux is obtained from the coolant temperature profile while the local heat transfer coefficient is determined by knowing the saturation and wall temperatures. After the construction, the test section was inserted in a glass cylinder to reduce heat dissipation and then installed in a mechanical structure which allows varying the channel orientation (Fig. 3).

More details about test section geometry and calibration can be found in Del Col et al. [7] where also the experimental technique has been validated by single phase heat transfer tests.

Figure 2. Square cross section minichannel (top); pressure tap in the stainless steel and coolant channel in the copper rod (bottom).

Figure 3. Test section at 90° channel orientation for condensation tests in upflow conditions.

3. Condensation tests

During test runs, when the experimental apparatus is working in steady state conditions, measurements of temperature, mass flow and pressure (absolute and differential) are recorded for 50 s and averaged and then reduced in a MATLAB® Version R2009a environment, calculating fluids properties with NIST Refprop Version 9.0 [8].

The vapour quality at the inlet of the measuring section \(x_{\text{in}}\) is obtained from the specific enthalpy of the refrigerant \(h_{\text{in}}\) at the same position. The parameter \(h_{\text{in}}\) depends on the inlet conditions of the refrigerant in the measuring sector; \(h_{\text{in}}\) is known from the local pressure and temperature when the working fluid enters the measuring sector as superheated vapour. A suitable equation to interpolate the water temperatures \(T_w\) along the axial position \(z\) is established. The equation parameters are calculated by means of the least square method. Three possible interpolating equations are considered in the following order of preference to minimize the uncertainty of heat transfer coefficients: a second order polynomial, an exponential equation with three parameters in the form \(y = a + b e^{-x/c}\) and a third order polynomial. A higher number of parameters is not required for the present interpolations. The best fitting equation is chosen according to the adjusted \(R^2\) square \((R^2_{\text{adj}})\) method as the one with the minimum number of parameters which guarantees that all the thermocouples are fitted within the experimental uncertainty and that at least 68% of the thermocouples are fitted within 0.03 K, as the experimental uncertainty is normally distributed. If all these conditions are satisfied by the second order polynomial and by the exponential, as the number of parameters is the same, the former is
preferable because it means a lower experimental uncertainty on the heat transfer coefficients. Beside these conditions, the chosen equation should lead to heat transfer coefficients that are insensitive to the method of interpolation, that is to say that the variation in heat transfer coefficients using the chosen equation and the next admissible equation should be within the experimental uncertainty. If the third order polynomial interpolates the data in the best way, the sensitivity analysis is performed using the fourth order polynomial. In statistics, the coefficients of determination $R^2$ and adjusted $R^2$ are utilized to assess the fitting procedure (Rawlings et al. [9]). Unlike $R^2$, $R_{adj}^2$ increases with the number of parameters of the fitting equation only if the new term improves significantly the model, thus, let $f_1$ and $f_2$ be two admissible functions that are one after the other in the order of preference, $f_1$ would be the chosen polynomial degree for the fitting equation in this paper if

$$
|R_{adj}^2 (f_1) - R_{adj}^2 (f_2)| \leq 0.0035
$$

The slope of the water temperature profile along the measuring sector is used to calculate the local heat transfer rate per unit length $q'(z)$:

$$
q'(z) = -\dot{m}_w c_{p,w} \frac{dT_w(z)}{dz}
$$

where $z$ is the axial position coordinate along the tube oriented with the refrigerant flow, $dT_w/dz$ is the derivative of the equation interpolating the water temperature along $z$, $\dot{m}_w$ is the water mass flow rate and $c_{p,w}$ is the specific heat of the water.

The local heat transfer coefficient (HTC) inside the minichannel can be defined as in Eq. (3) where $p_i$ is the perimeter of the channel, $T_{sat}$ the refrigerant saturation temperature and $T_{wall}$ the wall temperature:

$$
HTC(z) = \frac{q'(z)}{p_i [T_{sat}(z) - T_{wall}(z)]}
$$

The saturation temperature is derived from the saturation pressure measurements at the inlet and at the outlet of the measuring sector. As stated in Del Col et al. [7], since the difference between saturation temperature and wall temperature is pretty large for typical test conditions and the saturation temperature drop due to pressure drop is small, the local saturation temperature can be obtained from the linear trend between the ends of the measuring sector without making a significant error in the evaluation of heat transfer coefficients.

Finally, the local thermodynamic vapour quality $x(z)$ at any location $z$ can be found from the heat flow rate $q(z)$, the refrigerant mass flow rate $\dot{m}_r$, the latent heat $h_{LG}$ and the inlet vapour quality $x_{in}$:

$$
x(z) = x_{in} - \frac{\int_z^i q'(z)dz}{\dot{m}_r h_{LG}} = x_{in} - \frac{q(z)}{\dot{m}_r \cdot h_{LG}}
$$

The error analysis has been executed according to the method described in Del Col et al. [7]. As the experimental uncertainty associated to the heat transfer rate strongly depends on the axial position, the expanded uncertainty of the heat transfer coefficient is high at the ends of the measuring sector, while it is low in the middle (see uncertainty bands in Fig. 4 and 5). As a consequence, at low mass velocities, the uncertainty is high only at high vapour qualities because the refrigerant exits the test section as subcooled liquid, while at high mass velocities the uncertainty is high both at high and low vapour quality, because at the outlet of the test section, a two-phase flow still occurs. The channel orientation has negligible influence on the error analysis.
4. Experimental results

The experimental conditions adopted during R134a condensation tests at 40°C saturation temperature have been reported in Table 1. The heat transfer coefficients have been measured with mass fluxes between 100 and 790 kg m\(^{-2}\) s\(^{-1}\) at three different channel orientations: horizontal configuration (0°), vertical downward flow (-90°) and vertical upward flow (+90°). In Fig. 4 and 5 experimental heat transfer coefficients are reported respectively at \(G = 390\) kg m\(^{-2}\) s\(^{-1}\) and \(G = 200\) kg m\(^{-2}\) s\(^{-1}\); three different data series for the three tested channel configurations are presented. For these mass velocities, the heat transfer coefficients display the same trend that one would expect for condensation inside a conventional tube: they increase with thermodynamic vapour quality and with mass velocity, implying that condensation must be controlled by shear stress; the influence of gravity is shown to be negligible at these operating conditions. Although not reported here, even at \(G = 790\) kg m\(^{-2}\) s\(^{-1}\) no effect of channel orientation is found from the tests. The experimental heat transfer coefficients found for the test runs with horizontal channel agree with those measured for the same conditions by Del Col et al. [7] within the experimental uncertainty.

Each plot of Fig. 6 and 7 is referred to a mass flux respectively of 135 and 100 kg m\(^{-2}\) s\(^{-1}\) and offers three different curves for the different test channel configurations (0°, 90° downflow and 90° upflow).

The HTCs during condensation of R134a are reported for vapour qualities between 0.2 and 0.9. The effect of channel orientation is relevant at \(G = 100\) kg m\(^{-2}\) s\(^{-1}\) and particularly at \(G = 135\) kg m\(^{-2}\) s\(^{-1}\): the HTC measured in 90° downflow orientation is found to be lower as compared to the horizontal and vertical upflow configurations especially at low vapour qualities.

Table 1. Condensation tests matrix.

| \(G\) [kg m\(^{-2}\) s\(^{-1}\)] | Horizontal (0°) | Vertical downflow (-90°) | Vertical upflow (+90°) | Vapour quality range [-] |
|---|---|---|---|---|
| 100 | x | x | x | 0.2-0.8 |
| 135 | x | x | x | 0.2-0.9 |
| 200 | x | x | x | 0.2-0.9 |
| 390 | x | x | x | 0.2-0.9 |
| 790 | x | x | x | 0.25-0.9 |

![Figure 4. Effect of channel orientation on heat transfer coefficient during condensation at mass flux \(G = 390\) kg m\(^{-2}\) s\(^{-1}\).](image1)

![Figure 5. Effect of channel orientation on heat transfer coefficient during condensation at mass flux \(G = 200\) kg m\(^{-2}\) s\(^{-1}\).](image2)
When moving from $G = 135$ kg m$^{-2}$ s$^{-1}$ to $G = 100$ kg m$^{-2}$ s$^{-1}$ it appears that the HTCs in horizontal and upflow do not vary significantly while the coefficients in vertical downflow are higher at the lower mass flux. The present results suggest that more tests should be performed at mass flux between 100 and 200 kg m$^{-2}$ s$^{-1}$ with vertical downflow.

5. Comparison against correlations

Cavallini et al. [10] proposed a method to predict condensation heat transfer coefficients inside horizontal smooth tubes with internal diameter greater than 3 mm. Since no differences between the three configurations ($0^\circ$, $+90^\circ$, $-90^\circ$) have been found at mass flux ranging between 200 and 790 kg m$^{-2}$ s$^{-1}$, the present data is compared against Cavallini et al. [10] model. As depicted in Fig. 8, the correlation by Cavallini et al. [10] predicts the present experimental heat transfer coefficients within 13% at mass flux between 390 and 790 kg m$^{-2}$ s$^{-1}$; on the contrary, at 200 kg m$^{-2}$ s$^{-1}$ the heat transfer coefficients are underpredicted on average by 24%. As reported by Del Col et al. [7] the correlation for macroscale condensation is not expected to predict the heat transfer enhancement due to the surface tension in the square minichannel at 200 kg m$^{-2}$ s$^{-1}$ or lower mass flux.
Recently, Shah [11] provided a correlation to predict heat transfer coefficients during condensation inside circular and rectangular small diameter channels in horizontal configuration. The correlation has been validated with eight fluids over a wide range of parameters that include tube diameters from 0.49 to 5.3 mm, mass fluxes from 50 to 1400 kg m\(^{-2}\) s\(^{-1}\) and reduced pressure from 0.048 to 0.52. At the highest mass flux (\(G = 790\) kg m\(^{-2}\) s\(^{-1}\)) in the present database, the model by Shah [11] has been found to overpredict by 30% the experimental data. At lower mass velocity (100 and 135 kg m\(^{-2}\) s\(^{-1}\)), the model predicts the experimental data in horizontal channel and vertical upflow inside the ±20% band; such data is compared against the model by Shah [11] in Fig. 9. However, it should be pointed out that the Shah [11] model has been validated only for horizontal channels.

6. Conclusions
An accurate experimental technique has been employed in this work to assess the effect of channel orientation on heat transfer coefficients during the condensation of R134a at 40°C saturation temperature inside a square section minichannel, with a hydraulic diameter of 1.23 mm.

Local heat transfer coefficients have been reported at mass velocities between 100 kg m\(^{-2}\) s\(^{-1}\) and 790 kg m\(^{-2}\) s\(^{-1}\) for three different configurations: horizontal, vertical downflow and vertical upflow.

According to the experimental uncertainty, no differences have been noticed for the three configurations at mass velocities from 790 kg m\(^{-2}\) s\(^{-1}\) down to 200 kg m\(^{-2}\) s\(^{-1}\), where the condensation heat transfer is controlled by shear stress and surface tension. On the contrary at 100 and 135 kg m\(^{-2}\) s\(^{-1}\), the heat transfer coefficients in vertical downflow are lower as compared to the ones measured in the horizontal configuration and vertical upflow.

Comparisons are reported with the Cavallini et al. [10] model at high mass flux, and with the model by Shah [11] at low mass flux.

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