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Condensation heat transfer in square, triangular, and semi-circular mini-channels

Melanie Derby [a], Hee Joon Lee [a,b], Yoav Peles [a], Michael K. Jensen [a,*]

[a] Department of Mechanical, Aerospace, and Nuclear Engineering, Rensselaer Polytechnic Institute, Troy, NY 12180, United States
[b] Department of Mechanical Engineering, Kookmin University, Seoul 136-702, South Korea

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ABSTRACT

Condensation heat transfer coefficients in mini-channels were measured with smaller measurement uncertainties than previously obtained using three specially designed copper test sections. Single-phase experiments validated the approach. Data are reported for R134a in 1 mm square, triangular, and semi-circular multiple parallel minichannels cooled on three sides. A parametric study was conducted over a range of conditions for mass flux, average quality, saturation pressure, and heat flux. Mass flux and quality were determined to have significant effects on the condensation process, even at lower mass fluxes, while saturation pressure, heat flux, and channel shape had no significant effects. The lack of shape effects were attributed to the three-sided cooling boundary conditions. Because there was no significant surface tension enhancement, the macro-scale Shah (2009) [26] correlation best predicted the data, with a mean average error (MAE) of 20–30% for all geometries.

1. Introduction

Lightweight and compact condensers for vapor compression cycles have a variety of applications from electronics' cooling to transportation. However, as reported in the literature, uncertainties in measured condensation heat transfer coefficients can be as high as ±20% to ±40% due to the challenges of measuring condensation heat fluxes and wall temperatures at the micro- and mini-scales. Large uncertainties encourage condenser over-design. Typical approaches to measure condensation heat transfer coefficients in the micro- and mini-scale utilize fluid-to-fluid heat exchangers, which can have high experimental uncertainties at the low heat duties encountered in micro- and mini-scale condensation.

Several approaches to measure condensation heat transfer coefficients at the macro-scale have been later applied to the mini- and micro-scale. Akers et al. [1] and Dobson and Chato [2] used fluid-to-fluid heat exchangers, where one fluid was condensing and the other was the coolant. The heat transfer rate was obtained through a coolant-side energy balance. To obtain condensation heat transfer coefficients, thermocouples or thermistors attached to the tube wall measured temperature. This approach was also utilized by Cavallini et al. [3] in a multiple channel, 1.4 mm diameter condenser. An energy balance on the fluid-to-fluid heat exchanger measured heat duty while sensors in two obstructed channels measured wall temperatures. A large cooling water temperature difference was necessary for the temperature measurement's uncertainty to be small compared to the measured temperature difference, which may not have been achieved.

Condensation heat transfer rates can also be found by conducting an energy balance before and after the test section on a pre- and post-heater. Wang et al. [4] constructed an air-cooled condensation test section, with 2 kW pre- and post-heaters, used in conjunction with an air-cooled test section to determine heat transfer rates. Similarly, Agarwal and Garimella [5] determined test section heat transfer rates utilizing 80 W pre- and post-heaters with parallel channels on the order of 100 μm hydraulic diameter in a copper wafer. Several challenges were encountered in their design (e.g., large quality changes, pressure drops, and significant axial conduction), which required a complex numerical conjugate heat transfer analysis to determine condensation heat transfer coefficients.

Another common macro-scale condensation heat transfer coefficient measurement approach applied to mini- and micro-scale condensation is the Wilson plot method, reviewed by Fernandez-Seara et al. [6]. The Wilson plot is a graphical method that determines heat transfer coefficients without direct measurement of wall temperatures. The Wilson plot and modified Wilson plot have several limitations at the mini- and micro-scale. A functional form of the coolant heat transfer coefficient must be assumed, and the condensation heat transfer resistance must be the dominant resistance. Garimella and Bandhauer [7] determined in an uncertainty analysis that a 1.6 condensation-to-coolant resistance ratio (better suited to the macro-scale than the mini- or micro-scale) was needed to obtain the heat transfer coefficient within ±15%. However, the modified Wilson plot has been used by some researchers, such as Webb and Ermis [8], to determine condensation heat

* Corresponding author. Address: JEC 2049 2nd Fl., Rensselaer Polytechnic Institute, 110 8th St., Troy, NY 12180, United States. Tel.: +1 518 276 2843; fax: +1 518 276 6025.

E-mail address: JenseM@rpi.edu (M.K. Jensen).

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Nomenclature

$A_{0,i}$	polynomial curve fitting constant, intercept (°C)
$A_{1,i}$	polynomial curve fitting constant (°C/m)
$A_{2,i}$	polynomial curve fitting constant (°C/m^2)
A_{header}	area of the header (m^2)
$C_{0,i}$	y-axis intercept (°C)
$C_{1,i}$	slope of the line (°C/m)
D_h	hydraulic diameter (m)
dT/dy_i	segment-averaged temperature gradient (°C/m)
h_i	segment-averaged heat transfer coefficient (W/m^2 K)
h_{inlet}	inlet header heat transfer coefficient (W/m^2 K)
h_{outlet}	outlet header heat transfer coefficient (W/m^2 K)
i	the index representing measuring segments 1, 2, and 3
k	thermal conductivity (W/m K)
$L_{seg,i}$	length of the measuring segment (m)
MAE	mean average error
$\dot{Q}_g$	total heat transfer rate from the gradient method (W)
$\dot{Q}_{header,inlet}$	inlet header heat transfer rate (W)
$\dot{Q}_{header,outlet}$	outlet header heat transfer rate (W)
q''_i	channel heat flux (W/m^2)
$\dot{Q}_i$	heat transfer rate in each measuring segment (W)
$T_{f,i}$	segment-averaged fluid temperature (°C)
T_i	temperature in copper block (°C)
$T_{w,i}$	segment-averaged wall temperature (°C)
W	width (m)
w_g	uncertainty in the temperature gradient (°C/m)
w_{Ti}	uncertainty in temperature (°C)
x	quality
Y	thermocouple's distance from the bottom of the channel (m)
y_i	position of the ith thermocouple (m)
$\bar{y}$	average thermocouple position in measuring segment (m)

transfer coefficients at the mini-scale in parallel channels of diameter 0.44–1.56 mm.

Garimella and Bandhauer [7] and Bandhauer et al. [9] further adapted the Wilson plot method for the mini-scale by creating a "thermal amplification" loop. The test section consisted of a counter-flow fluid-to-fluid heat exchanger; however, there were competing objectives in the design of the test section, as the desire for low quality changes and, thus, heat duties, conflicted with the need for the condensing resistance to dominate in the Wilson plot method. The "thermal amplification" loop separated the cooling portion into two loops: one high mass flow rate, low thermal resistance loop for the Wilson plot method, and a second lower mass flow rate and larger temperature change loop for heat duty measurement.

Matkovic et al. [10] measured the coolant water and wall temperature profiles along a fluid-to-fluid heat exchanger. The wall temperature measurements were fit to a polynomial form, and the local derivative of the coolant temperature profile was used to determine condensation heat flux while studying condensation of R134a in a 0.96 mm channel. However, this approach was extremely sensitive to temperature measurements.

Shin and Kim [11,12] determined condensation heat transfer rates by comparing an air-cooled test section to an electrically heated test section. Refrigerant flowed through one tube, while an identical copper tube with fins contained a *DC* resistive heater; heat transfer rate was assumed to be the same in both tubes when comparable surface temperatures were obtained by varying the *DC* power. Although this method was able to measure heat transfer in single channels, two practical barriers to implementing this method existed: measuring extremely low mass flow rates and constructing two identical finned copper test sections.

Baird et al. [13] used thermoelectric coolers to determine condensation heat flux. R11 and HFC123 in single tubes of diameter 0.92 mm and 1.95 mm were cooled by ten thermoelectric coolers, or *TEC*s, were used to create ten "quasi-local" energy balances. For a *TEC*, the cold side cooling rate was obtained through an equation with material properties provided by the manufacturer. "Quasi-local" heat transfer coefficients were found in this novel approach. However, recent work by Derby et al. [14] utilized *TEC*s as heat flux sensors for single-phase and condensation of R134a, and showed several shortcomings of TECs as heat flux sensors. Despite rigorously calibrating the *TEC*s in a special calibration apparatus, experimental uncertainties in single phase Nusselt number of more than 100% were found due to inconsistencies in the devices themselves. Also, hot-side and cold-side thermal resistance were a possible explanation of for the unpredictable performance of the cooling output of the *TEC*s. In conclusion, *TEC*s are useful for cooling applications, but need further consideration before implementing as condensation heat flux sensors.

Measuring condensing heat fluxes and wall temperatures presents a challenge, as there is no direct electrical means to measure the heat flux, as in boiling. Additionally, the small heat duties encountered at the micro- and mini-scale pose a problem for macro-scale fluid-to-fluid heat exchanger techniques, as the measured heat fluxes are very sensitive to temperature measurements. The thermal amplification loop, developed by Garimella and Bandhauer [7] and Bandhauer et al. [9], was a rigorous approach to measuring condensation heat transfer coefficients, yet uncertainties were ±21%, on average, with uncertainties as high as ±40%. Clearly, new condensation heat flux and wall temperature need to be developed.

Although measurement of mini- and micro-scale condensation heat fluxes and wall temperatures (and, thus, heat transfer coefficients) has been difficult, researchers have identified several important parameters. Of prime importance is the increase of condensation heat transfer coefficients with decreasing channel diameter, which makes mini- and micro-scale condensation attractive. Shin and Kim [12] tested three circular and three rectangular channels, with hydraulic diameters ranging from 0.493 mm to around 1 mm, and found condensation heat transfer coefficient to be proportional to the hydraulic diameter, $D_h^{-0.54}$ for circular channels and $D_h^{-0.45}$ for rectangular channels. However, Baird et al. [13] found little effect of diameter between a 0.92 mm and 1.95 mm tube. In the micro-channel diameter range, Agarwal and Garimella [5] studied channels with hydraulic diameters of around 100–160 μm, and reported heat transfer coefficients of 20–70 kW/m^2 K—higher than seen in other literature using larger channels.

In addition to channel diameter, several studies investigated the dependency of the condensation heat transfer coefficient on the channel shape, which modified surface tension forces. Shin and Kim [12] found the length-averaged condensation heat transfer coefficients to be higher in rectangular channels at lower mass fluxes, and higher in circular channels at higher mass fluxes. Wang and Rose [15] argued that in non-circular channels condensate gathers at the corners, thins the liquid film and lowers thermal resistance compared to the uniform film of a circular channel. Eventually, however, the corners in a non-circular channel flood, hindering the benefit gained from the thin liquid film. Therefore,