Condensation of Carbon Dioxide in Microchannels
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ABSTRACT

Heat transfer coefficients and pressure drops during condensation of carbon dioxide (CO₂) are measured in small quality increments in rectangular microchannels of $0.10 \leq D_h \leq 0.16$ mm and aspect ratios from 1 to 4. Channels are fabricated on a copper substrate by electroforming copper onto a mask patterned by X-ray lithography, and sealed by diffusion bonding. The test section is cooled by chilled water circulating at a high flow rate to ensure that the thermal resistance on the condensation side dominates. A conjugate heat transfer analysis, in conjunction with local pressure drop profiles allows driving temperature differences, heat transfer rates, and condensation heat transfer coefficients to be determined accurately. Condensation heat transfer coefficients and pressure drops are measured for $G = 400, 600$ and $800 \text{ kg m}^{-2} \text{ s}^{-1}$, for $0 < x < 1$, and saturation temperatures of 15, 20 and 25°C ($P_r = 0.69, 0.78$ and 0.87). The data are used to evaluate the applicability of correlations developed for larger hydraulic diameters and different fluids for predicting condensation heat transfer and pressure drop of CO₂.

KEYWORDS

Carbon dioxide, condensation, heat transfer, pressure drop, experimental

NOMENCLATURE

AAD	Absolute average deviation	(%)
AD	Average deviation	(%)
AR	Aspect ratio (W/H)	(-)
C	Constant correlated by Chisholm [1]	(-)
Co	Confinement number $\sqrt{\sigma/(g(\rho_L - \rho_v))}/D_h$	(-)
D	Diameter (circular channels)	(mm)
D_h	Hydraulic diameter (non-circular channels)	(mm)
G	Mass flux	(kg m ⁻² s ⁻¹)
h	Heat transfer coefficient	(kW m ⁻² K ⁻¹)
i	Specific enthalpy	(kJ kg ⁻¹)
L	Length	(m)
$\dot{m}$	Mass flow rate	(kg s ⁻¹)
N	Number of data points	(-)
P	Pressure	(kPa)
P_r	Reduced pressure (P/P_{crit})	(-)
$\dot{Q}$	Heat duty	(kW)

Re	Reynolds number	(-)
W	Width	(m)
x	Quality	(-)
X	Martinelli parameter	(-)

Greek Symbols

α	Void fraction	(-)
ρ	Density	(kg m ⁻³)

Subscripts and Superscripts

amb	Ambient
avg	Average
cond	Condensation
crit	Critical
exp	Experimental
f	Friction
in	Inlet
L	Liquid
out	Outlet
pre	Pre-heater
post	Post-heater

sat Saturated

V Vapor

1. INTRODUCTION

The need for smaller, more efficient and environmentally friendly thermal energy systems continues to drive research on two-phase flow condensation in mini- and microchannels. As channel diameter is decreased, higher heat transfer coefficients and heat fluxes can be obtained, leading to a reduction in heat exchanger size, material use and inventory of refrigerant. Additionally, small channels can more effectively contain the higher working pressures characteristic of new environmentally friendly refrigerants including carbon dioxide.

A growing body of research characterizing two-phase flow regimes, flow transitions, pressure drop and heat transfer for condensing flows in mini- and microchannels is now available. Still, the exact definitions of, and demarcation between macro-, mini- and microchannels remain nebulous. Rather than defining a hard cutoff between channel classifications, it can best be concluded that the difference between mini- and microchannels and macrochannels assumes importance when forces or phenomena generally not significant at the macro scale take on increasing importance. For two-phase flow, this is usually the increasing importance of surface tension forces and the decreasing influence of gravitational forces as channel size decreases. Kew and Cornwell [2] relate the magnitudes of surface tension and gravitational forces through the Confinement number (microchannel effects exist when $Co > 0.5$), while Serizawa *et al.* [3] relate the same forces through the Laplace constant (L), proposing microchannel effects exist when $L > D_h$. Both relations imply that microchannel effects are relevant in larger channels for fluids with larger surface tension. However, defining microchannels in this manner is complicated by the general inconsistency in usage of terminology between academic literature and industry. For consistency throughout this paper, channels with $D_h < 1$ mm are referred to as microchannels, with minichannels encompassing a range of $1 \leq D_h \leq 5$ mm.

In an effort to better understand the increased role of surface tension and other effects at this scale, the authors' research group has conducted studies on flow visualization, heat transfer and pressure drop

during condensation of a variety of refrigerants, over a range of operating conditions in channels with D_h from 0.1 to 5 mm, with a focus on synthetic refrigerants including R-134a, R-404A and R-410A. Coleman and Garimella observed and classified the significant flow regime transitions in conventional round channels and mini- and microchannels of varying geometries using air-water mixtures [4] and condensing HFC refrigerants [5-7]. With an understanding of the important flow mechanisms of condensing refrigerants, Garimella *et al.* developed experimentally validated multi-regime pressure drop models for circular [7] and noncircular [8] mini- and microchannels. The development of the novel thermal amplification technique [9] enabled determination of condensation heat transfer coefficients at small quality increments with high accuracy, and led to the development of heat transfer models for circular [10] and non-circular mini- and microchannels [11]. Extensive studies on condensation and supercritical gas cooling heat transfer and pressure drop with newer “high pressure” refrigerants have been conducted at reduced pressures (P_r) from 0.8 to 1.2 in channels with $0.76 \leq D_h \leq 9.40$ mm [12-15]. A technique for determining heat transfer and pressure drop in extremely small channels ($0.10 \leq D_h \leq 0.16$ mm) was developed for condensing flows of R-134a [16].

The present work extends the studies of Agarwal and Garimella [16] and Fronk and Garimella [17] to consider condensation of carbon dioxide (CO_2) at $0.10 \leq D_h \leq 0.16$ mm and aspect ratios (AR) from 1 to 4. As a low cost, non-toxic, low global warming potential (GWP) fluid, CO_2 has garnered much attention recently as a replacement for synthetic refrigerants [18, 19]. With a low critical temperature (31.1°C), CO_2 based heat pump systems typically reject heat through non-isothermal supercritical gas cooling rather than a typical condensation process. However, CO_2 condensation is of interest in applications such as cascaded vapor compression systems. The investigation of CO_2 condensation represents several notable differences. At the temperatures of interest (0 to 25°C), the saturation pressure ranges from 3500 to 6435 kPa, significantly higher than the saturation pressures of interest for synthetic refrigerants. Thus, it is necessary to design test channels, pumps, auxiliary heat exchangers and all other experimental equipment to safely contain these high pressures. Due to the low critical temperature, CO_2 condensation at the

saturation temperatures of interest (0 to 25°C) occurs at much higher reduced pressures (0.47 to 0.87) than those of synthetic refrigerants. In contrast, the reduced pressures of R-134a and R-404A at the same saturation temperatures range from 0.07 to 0.16, and from 0.16 to 0.33, respectively. Fig. 1 shows the variation of reduced pressure of R-134a, R-404A and CO₂ as a function of saturation temperature.

[Fig. 1 to be inserted approximately here]

Additionally, CO₂ condensation at high reduced pressures results in thermophysical properties substantially different from the conventional synthetic refrigerants and air/water mixtures that have typically been investigated. Table 1 shows a comparison of liquid and vapor densities, viscosities and surface tension for air/water, R-134a and CO₂ at typically investigated conditions. CO₂ has a lower surface tension, viscosity and liquid/vapor density ratio than both air/water and R-134a, which impacts the relative superficial liquid and vapor velocities, and the interfacial shear between vapor and liquid in the annular flow regime. These combined effects may impact heat transfer and pressure drop in ways that conventional models fail to predict accurately.

[Table 1 to be inserted approximately here]

2. PRIOR WORK

Internal condensation heat transfer and pressure drop are intimately related to the local flow patterns and transition mechanisms, which in turn are functions of quality, void fraction, mass flux and channel geometry. Dobson and Chato [20] state that internal condensation in large horizontal tubes can be divided into gravity controlled and vapor shear controlled regions. The gravity controlled region is characterized by formation of condensate around the top circumference, with liquid pooling and flowing axially along the bottom. Heat transfer in this regime is generally modeled through a Nusselt type analysis [21, 22]. However, as shown by Coleman and Garimella [6] in a visualization study of condensing R-134a, the size (as seen on a G versus x flow regime map) of the stratified and wavy flow regimes decreases with decreasing diameters, and is completely absent at small diameters ($D_h < 1$ mm) at the mass fluxes of interest. Thus, for the hydraulic diameters and mass fluxes in the present study, the

flow is expected to be primarily annular or intermittent, with heat transfer and pressure drop dominated by vapor shear.

Shear dominated condensation in large diameter tubes has received much attention in the past, resulting in theoretical, semi-empirical and purely empirical correlations. An example of a correlation that has enjoyed widespread use is the purely empirical Shah [23] correlation and its subsequent modification [24]. The correlation is simple to apply and yields good predictions for a range of tube sizes and fluids. In recent years, efforts have been made to develop and refine accurate heat transfer and pressure drop models that are valid over a range of tube sizes, mass fluxes and fluids, including experimental and analytical studies by Dobson and Chato [20], Cavallini *et al.* [25] and Thome *et al.* [26]. In general, these studies have focused on round tubes with $D_h > 3$ mm.

Accurate measurement of condensation heat transfer and pressure drop in microchannels at temperatures relevant for heat rejection applications is needed to assess the applicability of these and other models at small D_h . Obtaining accurate, repeatable results is made difficult by the small condensation heat duties, low mass flow rates, and high heat transfer coefficients characteristic of microchannels and the inability to easily apply and measure a constant heat flux. These difficulties are compounded when it is desired to measure heat transfer coefficients in small quality increments at high reduced pressures; where the associated i_{fg} and condensation heat duties are very small. Whereas evaporation experiments can accurately measure and control electric resistance heaters to determine evaporation heat duty, no similar device exists for condensation. Some researchers [27] have used thermoelectric coolers (TEC) to provide a fixed cooling load, with operation being controlled by an electrical power input, similar to electrical heaters. However, it has been shown that inconsistencies and high uncertainties in TEC performance make them unreliable for use as actual heat flux measurement devices [28]. Thus, the condensing refrigerant must be coupled with circulating water or another coolant. However, it is desirable to maintain a high coolant mass flow rate and minimal difference in temperature ($T_{sat} - T_{coolant}$) across the test section to limit coolant-side thermal resistance, which results in high uncertainty in

measured heat duty. Other methods to determine the condensation heat duty more accurately include the use of thermopiles as heat flux sensors [29] and the thermal amplification technique proposed by Garimella and Bandhauer [9].

An extensive review of two-phase flow regime mapping, void fraction, heat transfer and pressure drop and experimental techniques for condensation in mini- and micro channels appears in [30-32] and is not replicated here. However, as noted by Agarwal and Garimella [16] extremely limited research exists on condensation in channels with $D_h < 0.20$ mm. They concluded that there were no reliable models for predicting condensation heat transfer and pressure drop or flow regime at the small channel sizes of interest here ($0.10 < D_h < 0.16$ mm).

Adiabatic N₂/water visualization studies by Kawahara *et al.* [33] ($D_h = 0.10$ mm, rectangular), Chung and Kawaji [34] ($0.05 < D_h < 0.50$ mm, rectangular) and Chung *et al.* [35] ($D = 0.096$ mm, circular and $D_h = 0.10$ mm, rectangular) attempted to capture the effects of channel diameter and mass flux on flow regime transition and void fraction in channels of size comparable to those investigated in the present study. They found that channels with $D_h > 0.10$ mm exhibited similar flow regimes as minichannels (i.e., bubbly, slug, slug-annular and annular flow, with the absence of any stratified or wavy flow regimes), while channels with $D_h \leq 0.10$ mm exhibited only slug flow, which they attributed to the greater viscous and surface tension effects. Furthermore, they found negligible difference in flow behavior between rectangular and circular channels. In the present study, caution should be used when attempting to extrapolate this flow behavior in the channels of interest due to the higher order of magnitude of surface tension of water compared to that of R-134a or CO₂.

Agarwal and Garimella [16] developed a technique for accurately measuring the heat transfer coefficient and pressure drop during condensation of R-134a at small quality increments in rectangular microchannel test sections ($0.10 < D_h < 0.16$ mm, $AR = 1$ to 4.) The test section design, experimental techniques and data reduction methods used in their work form the basis for those used in the present study and are

discussed in greater detail in the following sections. They found that condensation heat transfer and pressure drop correlations developed for macro and mini channels underpredicted the data consistently. The correlation of Lockhart and Martinelli [36] with $C = 12$ (laminar film and turbulent vapor core) surprisingly made the most accurate prediction of pressure drop despite being developed for gas oil mixtures in much larger tubes.

Work on CO₂ condensation has been extremely limited, regardless of channel size. Large diameter tube studies were conducted by Iqbal and Bansal [37] and Kang *et al.* [38] in horizontal round tubes with $D = 6.52$, and 5.15 , respectively. Huai and Koyama [39] investigated pressure drop and heat transfer during CO₂ condensation in a multi-port extruded aluminum test section with 10 circular ports with $D = 1.31$ mm. They conducted experiments at high reduced pressure ($0.88 < P_r < 0.99$) with $25.4 \leq T_{\text{sat}} \leq 30.5^\circ\text{C}$ and $123 \leq G \leq 315 \text{ kg m}^{-2} \text{ s}^{-1}$. They reported heat transfer coefficients between 2 and 6 kW m⁻² K⁻¹; however, their data showed unusual fluctuations as quality varied from 1 to 0, which they attributed to poor thermal contact between the test section and chilled copper block. Park and Hrnjak [40] conducted an experimental investigation of CO₂ condensation at reduced pressures of 0.23 and 0.31 ($T_{\text{sat}} = -25^\circ\text{C}$ and -15°C) at a wall subcooling from 2 to 4°C in circular channels with $D = 0.89$ mm and $200 \leq G \leq 800 \text{ kg m}^{-2} \text{ s}^{-1}$. They measured heat transfer coefficients from 3 to over 10 kW m⁻² K⁻¹, with increasing heat transfer coefficients at higher qualities, higher mass fluxes and lower reduced pressures. The effect of the different wall subcooling was weak and well within the reported experimental uncertainty. They found that heat transfer was best predicted by the correlation of Akers *et al.* [41], with an average absolute difference < 20%. For condensation pressure drop, they found that the McAdams *et al.* [42] correlation, based on a homogenous flow model, provided reasonably accurate results. Kim *et al.* [43] studied CO₂ condensation in smooth tubes with $D = 3.48$ mm and microfinned tubes with a 3.51 mm melt-down diameter (*i.e.*, the inner diameter of a theoretical smooth tube with the same cross-sectional area as that of the microfin tube) at the same reduced pressures as the Park and Hrnjak [40] study. Heo *et al.* [44, 45] investigated CO₂ condensation in multiport rectangular tubes for hydraulic diameters ranging from 0.68 to

1.5 mm, and saturation temperatures of -5 to 5°C, while Zhang *et al.* [46] explored CO₂ condensation in a minichannel ($D_H = 0.9$ mm) condenser. They both found the condensation model of Thome *et al.* [26] to predict heat transfer best. Recently, Shah [47] published an investigation comparing the published CO₂ condensation data with multiple correlations from the literature. For mass fluxes below 300 kg m⁻² s⁻¹, he found that the empirical Shah [24] correlation had acceptable predictive ability. However, no correlation was able to predict data for mass flux greater than 300 kg m⁻² s⁻¹ (the range of the present study), suggesting the need for additional research. Investigations on CO₂ condensation at hydraulic diameters similar to those under consideration in the present study ($D_h < 0.20$ mm) are not available.

3. EXPERIMENTAL APPROACH

Heat transfer and pressure drop experiments for CO₂ in microchannels with $0.10 \leq D_h \leq 0.16$ mm and $1 \leq AR \leq 4$ were conducted. The experimental methodology, microchannel test section design, and data analysis procedures used in the present study were described in detail in [48] and [16]. Thus, only a brief overview of the methodology, and the modifications necessary for CO₂ condensation experiments, is presented here.

Multiple parallel rectangular channels were formed on a copper substrate by electroforming copper onto a mask patterned by X-ray lithography. Fig. 2a shows the layout of seven separate test sections on a single copper wafer. Each test section has a channel length of 40 mm, a depth of 0.10 mm, and a 4 mm inlet and outlet header to allow for the requisite refrigerant distribution. The aspect ratio of the test sections varies from $AR = 1$ to 4. The number of channels in parallel varies from 15 to 40, depending on the aspect ratio. Multiple parallel channels were fabricated so that condensation heat duties and refrigerant mass flow rates were large enough to yield accurate heat transfer coefficients at fine quality increments. The uncertainty in the width and depth of the channels is ± 0.5 μ m, with a surface roughness of ~ 10 -15 nm as validated by SEM images of the channels.

The etched sheets were then diffusion bonded to a 6-mm copper wafer to form closed channels. The resulting bond strength was conservatively estimated to be 10 MPa by the manufacturer. After bonding,

the top wafer was thinned to 1 mm to reduce the thermal resistance between the condensing refrigerant and the heat sink. Copper inlet and outlet tubes with a 3.175 mm OD were soldered to mating holes drilled in the channels to allow for connection to the experimental facility. A hydrostatic pressure test up to 10.5 MPa was performed on the refrigerant test section to ensure that it would safely work at CO₂ saturation pressures. A completed refrigerant channel assembly is shown in Fig. 2b.

Upper and lower coolant blocks were soldered to the assembled refrigerant channels as shown in Fig. 2c. Water flows countercurrent to the refrigerant through five parallel 15-mm long channels in each block ($D = 0.794$ mm). The completed assembly was then installed in the experimental loop, shown schematically in Fig. 3.

[Fig. 2 to be inserted approximately here]

[Fig. 3 to be inserted approximately here]

Temperature measurements at the inlet and outlet of each component shown are obtained using NIST traceable microprobes (ThermoWorks, Inc.) with accuracy of $\pm 0.1^\circ\text{C}$. Pressure is measured with Rosemount absolute pressure transducers with a set span of 0 to 10,000 kPa and an accuracy of $\pm 0.065\%$ of span at the inlet and outlet of the pre-and post- heater assemblies.

Starting at state point **1** in Fig. 3, the CO₂ is a subcooled liquid. By measuring the temperature and pressure, the specific enthalpy is determined using an equation of state proposed by Span and Wagner [49]. Between state point **1** and **2**, a specified amount of heat is added with an 80 W electrical resistance heater controlled by Variac device. The electrical input to the heater ($\dot{Q}_{\text{pre}}$) is measured using an Ohio Semitronics watt transducer (GW5-103E), with an accuracy of $\pm 0.2\%$ of reading. The specific enthalpy of state point **2** is then calculated from Eq. (1), where ($\dot{Q}_{\text{amb}}$) is the heat gain from the surroundings due to natural convection and radiation, calculated based on the physical geometry of the heater assembly, insulation properties and the local fluid temperature as described in Agarwal [48].

$$\dot{Q}_{pre} = \dot{m}(i_2 - i_1) - \dot{Q}_{amb} \quad (1)$$

The thermodynamic quality at state point **2** is determined as a function of the measured local pressure and calculated enthalpy. This specifies the quality at the test section inlet. From state point **2** to **3**, the CO₂ flows through a microchannel test section. The test section is coupled with a cooling water stream, flowing at a sufficiently high rate to maintain a nearly isothermal wall temperature. The condensation duty ($\dot{Q}_{cond}$) is rejected from the CO₂ and transferred to the coolant in the test section.

At state point **3**, the CO₂ is a two-phase mixture of liquid and vapor. From state **3** to **4**, heat is added until the CO₂ is a superheated vapor at state **4**. By measuring the local temperature and pressure, the enthalpy, h_4 , is determined. Heat is added and measured as before, using an 80 W resistance heater and watt transducer. The condensation heat duty is then found from Eq. (2). The calculated ambient heat/loss gains in the preheater and postheater were on the order of 0.5 to 1 W. This corresponded to an average of 1.4 % of the calculated heat duty, with a maximum of 3.5%. Heat loss/gain were minimal as experiments were conducted at saturation temperatures near ambient conditions (15 to 25°C).

$$\dot{Q}_{cond} = (\dot{Q}_{pre} + \dot{Q}_{pre,amb}) + (\dot{Q}_{post} + \dot{Q}_{post,amb}) - \dot{m}(i_4 - i_1) \quad (2)$$

The superheated vapor is condensed to a subcooled liquid state in a shell-and-tube heat exchanger, coupled with a circulating chilled water/glycol solution. The liquid refrigerant is circulated through the test loop by a Micropump gear pump (GAH-X21) capable of a maximum differential pressure of 520 kPa and a maximum system pressure of 34.5 MPa. The refrigerant mass flow rate is measured using a Rheonik Coriolis mass flow meter (RHM015) capable of measuring flow rates from 0.004 to 0.6 kg min⁻¹ with an uncertainty of ±0.2% of rate.

The system pressure is maintained at the desired saturated pressure by adjusting the total CO₂ charge in the system using an attached liquid CO₂ tank and a series of needle valves. For each data point, the heat input to the pre-heater and post-heater, pump speed and refrigerant charge level were adjusted to achieve

the desired test conditions. The system ran until steady state was achieved, after which data were recorded at a rate of 100 Hz for a minimum of 5 minutes.

4. DATA ANALYSIS

A thorough description of the data analysis procedure is presented in [48] and [16]; a representative data point for the 0.30×0.10 mm (15 parallel channels) with $G = 595 \text{ kg m}^{-2} \text{ s}^{-1}$ and $x_{\text{avg}} = 0.61$ at a saturation temperature of 20°C is described here to provide an overview of the analysis.

4.1 Pressure Drop Analysis

The frictional pressure drop through the microchannels was determined from a combination of the measured differential pressure, the calculated minor losses due to expansions, contractions and bends, the frictional pressure drop through the inlet and outlet tubes, and the pressure rise due to deceleration as shown in Eq. (3). Fig. 4 shows a schematic representation of the test section with locations of the important sources of minor pressure “losses” indicated.

$$\Delta P_f = \Delta P_{\text{measured}} - \Delta P_{\text{others,in}} - \Delta P_{\text{others,out}} + \Delta P_{\text{decceleration}} \quad (3)$$

[Fig. 4 to be inserted approximately here]

The measured pressure drop for the representative case is 25.1 kPa. The frictional pressure drop in the inlet (93 Pa) and outlet tubes (83 Pa) was determined from the multiple flow regime model of Garimella *et al.* [7]. Pressure drops from flow contractions from the inlet fitting to the inlet tube (AA: 44 Pa), from inlet header to the microchannels (CC: 392 Pa), and from outlet header to the outlet tube (EE: 40 Pa) were calculated from the homogeneous model as recommended by Hewitt *et al.* [50]. Expansion ΔP s from inlet tube to inlet header (BB: 6 Pa), from the microchannels to the outlet header (DD: 143 Pa), and from the outlet tube to the outlet fitting (FF: 11 Pa) were calculated using the separated flow model [50]. The ΔP s due to tube bends in the inlet and the outlet (6 and 4 Pa, respectively) and changing flow direction in the inlet and the outlet headers (390 and 266 Pa, respectively) were estimated using the homogeneous flow model [50].

The deceleration pressure gain (530 Pa) was found using the separated flow model (Eq. (4)) as described in Carey [51]. The equation was evaluated at the appropriate fluid properties, mass flux, inlet and outlet qualities and inlet and outlet void fractions as determined from the Baroczy model [52].

$$-\Delta P_{deceleration} = \left[\left(\frac{G^2 x_{out}^2}{\rho_v \alpha_{out}} \right) + \frac{G^2 (1 - x_{out})^2}{\rho_L (1 - \alpha_{out})} \right] - \left[\left(\frac{G^2 x_{in}^2}{\rho_v \alpha_{in}} \right) + \frac{G^2 (1 - x_{in})^2}{\rho_L (1 - \alpha_{in})} \right] \quad (4)$$

A conservative uncertainty of $\pm 50\%$ was assigned to each calculated pressure drop. By summing all of the pressure losses and gains, the frictional pressure drop in the channels was determined. For the representative case, the net pressure drop through the test section inlet and outlet was 0.92 kPa and 0.24 kPa respectively, with a 0.53 kPa deceleration pressure gain. The resulting frictional pressure drop was found to be 24.49 kPa, or 97% of the measured ΔP .

4.2 Heat Transfer Coefficient

Agarwal [48] developed a segmented, iterative model on the *Engineering Equation Solver* (EES) [53] software platform to calculate the average condensation heat transfer coefficient in the test section. Due to the high thermal conductivity of the copper test sections, it was necessary to account for axial heat conduction through the channel walls. Thus, the coupled conjugate effects of conduction through the wall and convection within the refrigerant and coolant channels were addressed by dividing the test section into discrete segments, as shown in Fig 5. While this model was originally developed for use with R-134a, it is not dependent on the working fluid, and is therefore used here for CO₂ condensation data analysis with minimal changes.

[Fig. 5 to be inserted approximately here]

The resistance network equations were setup and the refrigerant heat transfer coefficient was computed using the appropriate segment heat transfer areas and heat transfer coefficients. On the refrigerant side, the heat transfer area was calculated assuming the channel walls are rectangular fins with adiabatic tips. For the conditions studied, the calculated fin efficiency was very high ($\eta_{fin} > 99\%$); thus, the refrigerant-

side heat transfer area was essentially the entire channel surface area. Closure to the system of equations was provided by setting the sum of each segment heat duty to the measured condensation heat duty.

For the coolant channels, the heat transfer coefficient was determined using the Churchill correlation [54], with the measured mass flow rate and the properties evaluated at the average local measured temperature. The water-side heat transfer coefficient was assumed to be constant through the test section. The coolant flow rate was high enough such that the $\Delta T_{\text{coolant}} < 0.1^\circ\text{C}$. For the representative case, the velocity through each channel was 8.15 m s^{-1} , with $\text{Re} = 5784$ and resulting $h_{\text{water}} = 40 \text{ kW m}^{-2} \text{ K}^{-1}$. A conservative uncertainty of $\pm 25\%$ was assigned to the calculated value of h_{water} to determine the refrigerant heat transfer coefficient uncertainty.

The conductive thermal resistance in each copper node was evaluated from the thickness and thermal conductivity. The refrigerant temperature at each node was determined from the local saturation pressure. Agarwal [48] developed empirical relations to account for the change in saturation temperature due to the non-linear saturation pressure change through the test section for R-134a, which could be on the order of several $^\circ\text{C}$ for some conditions. Due to the high absolute pressure of CO_2 , expected pressure drops from 10 to 125 kPa represent a change of less than 2% of the total pressure and correspond to a saturation temperature change of less than 0.9°C at the highest pressure drop. Because the expected pressure drops resulted in a small change in CO_2 properties, it was assumed that the pressure decreased linearly through the test section. With all inputs specified, the system of equations was computed iteratively to obtain the average condensation heat transfer coefficient. The resulting refrigerant-side heat transfer coefficient for this representative case was $17.3 \text{ kW m}^{-2} \text{ K}^{-1}$.

An analysis of heat transfer and pressure drop uncertainty was conducted using a propagation of errors approach [55]. A summary of the uncertainties in measured and estimated parameters is presented in Table 2. The uncertainty in the measured differential pressure was $\pm 0.25 \text{ kPa}$ ($25.21 \pm 0.25 \text{ kPa}$), and a

$\pm 50\%$ uncertainty was assigned to each calculated inlet/outlet ΔP and deceleration pressure gain described above. The resulting uncertainty in the frictional pressure drop was 24.58 ± 0.40 kPa or 1.6 %.

[Insert Table 2 approximately here]

Considering the $\pm 0.2\%$ uncertainty in mass flow measurement and ± 0.5 μm uncertainty in channel dimension, the resulting mass flux for the representative case was 807 ± 3.3 $\text{kg m}^{-2} \text{s}^{-1}$ ($\pm 0.55\%$). The uncertainty of the pre-heater inlet temperature ($16.7 \pm 0.1^\circ\text{C}$) and pressure (5755 ± 6.5 kPa), combined with the uncertainty in the heat added by the pre-heater (35.2 ± 0.07 W) and the ambient heat gain with a conservative $\pm 50\%$ uncertainty yielded a test section inlet quality of 0.79 ± 0.004 (0.5%). A similar analysis on the post-heater yielded a test section outlet quality of 0.42 ± 0.003 (0.7%). The calculated condensation heat duty was 14.7 ± 0.22 W (1.5%).

The uncertainty of the channel inlet and outlet pressure resulted in a saturation temperature uncertainty of approximately $\pm 0.05^\circ\text{C}$ (0.25%). The calculated water-side heat transfer coefficient was 42 $\text{kW m}^{-2} \text{K}^{-1}$ with an assigned uncertainty of $\pm 25\%$. Thus, the resulting refrigerant heat transfer coefficient was determined to be 17.4 ± 1.9 $\text{kW m}^{-2} \text{K}^{-1}$, corresponding to an uncertainty of 11.5%.

5. EXPERIMENTAL RESULTS

Four test sections with different hydraulic diameter, aspect ratio and number of parallel channels were investigated:

- 0.10×0.10 mm $\times$ 40 parallel channels ($D_h=0.10$ mm)
- 0.20×0.10 mm $\times$ 18 parallel channels ($D_h=0.13$ mm)
- 0.30×0.10 mm $\times$ 15 parallel channels ($D_h=0.15$ mm)
- 0.40×0.10 mm $\times$ 15 parallel channels ($D_h=0.16$ mm)

Experiments were conducted at nominal mass fluxes of $G = 400, 600$ and 800 $\text{kg m}^{-2} \text{s}^{-1}$ at saturation temperatures of $T_{\text{sat}} = 15, 20$ and 25°C ($P_r = 0.69, 0.78$ and 0.87). An effort was made to maintain a

temperature difference between the saturated refrigerant and coolant water temperature of 4°C for all test conditions. A total of 335 data points were obtained, 79 for $D_h = 0.1$ mm, 83 for $D_h = 0.13$ mm, 88 for $D_h = 0.15$ mm and 85 for $D_h = 0.16$ mm.

It should be emphasized that the present investigation was focused on multiple parallel microchannels. To mitigate the possibility of maldistribution or instabilities, the design of the test section ensured that the mass flux and (and thus pressure drop) through the microchannels is larger (mass flux 1.5 - 2 greater in channels) than through the headers. Furthermore, as the data were obtained at high reduced pressure, the liquid/vapor density ratio in the present study is relatively low (3 to 5) compared to water/air (> 800). This should also help mitigate maldistribution concerns. Finally, maldistribution and instabilities are expected to be more prevalent in boiling flows, which are characterized by energy addition and rapid flow acceleration as bubbles nucleate and are confined in the small geometries. In prior flow visualization studies of condensation in multiple parallel mini- and microchannels, flow instabilities were not observed.

5.1 Pressure Drop Results

The frictional pressure drop through the microchannel test sections was calculated from the measured pressure drop as detailed in Section 4.1. The average uncertainty in frictional pressure drop for all data points was 1.8%. The combined deceleration pressure gain and minor losses for the four test sections was on an average only 2.6% of the measured pressure drop. In the present study, the mass flux through the inlet and outlet header was much smaller than the mass flux through the microchannels, resulting in very low flow velocities. Thus, the frictional pressure drop through the microchannels dominated any losses in the inlet and outlet headers. Additionally, it was desired to maintain a small quality change (and the related void fraction change) through the test section. Therefore, the calculated pressure gain from deceleration of the flow was small (< 1.5% of the measured ΔP) and on the same scale as the calculated minor losses.

Fig. 6 shows the frictional pressure gradient as a function of test section average quality for each test section at saturation temperatures of 15 and 25°C, and mass fluxes of 400, 600 and 800 kg m⁻² s⁻¹. The

pressure gradient was determined by dividing the measured frictional pressure drop by the length of the microchannels, which was 40 mm for all test sections. The results at a saturation temperature of 20°C are not presented here; however, they exhibit the same trends.

[Fig. 6 to be inserted approximately here]

Several trends can be discerned from Fig. 6. For all diameters, saturation temperatures and mass fluxes, pressure gradient is a strong function of quality, with increasing gradient observed at higher qualities. This is as expected due to the higher void fractions and associated high velocities of the lower density vapor at high quality. Additionally, all diameters and saturation temperatures show an increase in pressure gradient with increasing mass flux at a fixed quality. Comparison of the top four plots with the bottom four plots in Fig. 6 reveals an increase in pressure gradient with a decrease in saturation temperature/reduced pressure. The liquid/vapor density ratio increases at lower reduced pressure, which yields a larger difference between the vapor and liquid velocities and an increase in interfacial shear stress.

The effects of changing hydraulic diameter and aspect ratio on pressure gradient are less clear, as they are not varied independently. Fig. 7 shows pressure gradient data for each tube plotted at $T_{\text{sat}} = 15$ and 25°C and $G = 600$ and 800 kg m⁻² s⁻¹. For both saturation conditions, the $D_h = 0.10$ mm, $AR = 1$ test section has the highest pressure gradient at an equivalent mass flux and quality. The $D_h = 0.16$ mm, $AR = 4$ test section has the second highest gradient, followed by the $D_h = 0.13$, $AR = 2$ and $D_h = 0.15$ $AR = 3$ test sections. For channels with a fixed aspect ratio, an increasing pressure gradient with decreasing hydraulic diameter would be expected. In experiments with R-134a at the same channel dimensions, Agarwal [48] also noted an aspect ratio effect, with the highest gradients observed at the highest aspect ratio ($AR=4$), followed by $D_h = 0.10$ mm, $AR = 1$, $D_h = 0.13$, $AR = 2$ and $D_h = 0.15$ $AR = 3$ in that order. Agarwal [48] reasoned that the annular film may be less stable at higher aspect ratios, resulting in higher pressure drop and heat transfer. In the absence of a more detailed study in which hydraulic diameter and aspect ratio

effects are studied independently, or detailed flow visualizations are performed to ascertain the impact of aspect ratio on flow regimes and transition mechanisms at channels of this scale, it is difficult to draw further conclusions. However, it is clear that pressure drop models that do not account for aspect ratio will fail to predict the data accurately.

[Fig. 7 to be inserted approximately here]

5.2 Heat Transfer Results

The uncertainty of the calculated heat transfer coefficient is a function of the measured quantities, the uncertainty in channel dimensions and the assumed uncertainty in the water-side heat transfer coefficient. The average uncertainties in condensation heat transfer coefficient for the $D_h = 0.16, 0.15, 0.13$ and 0.10 mm channels are $\pm 16.0, \pm 11.5, \pm 11.3$ and $\pm 18.4\%$, respectively. The primary contribution to the uncertainty is the assumed $\pm 25\%$ uncertainty in water-side heat transfer coefficient. At conditions where high refrigerant-side heat transfer coefficients are expected, the thermal resistance ratio between the water and refrigerant-side decreases, resulting in an increase in uncertainty. Thus, the highest experimental uncertainties are observed at high mass flux, high quality and low saturation temperature, when the heat transfer coefficient is expected to be the highest. The overall uncertainty can be further decreased by evaluating the refrigerant heat transfer coefficient at larger quality differences, which would increase the condensation heat duty and minimize the uncertainty contribution of the heater power measurement, albeit at the cost of lower resolution in vapor quality.

Fig. 8 shows the condensation heat transfer coefficient as a function of test section average quality for each test section at saturation temperatures of 15 and 25°C, and mass fluxes of 400, 600 and 800 kg m⁻² s⁻¹.

[Fig. 8 to be inserted approximately here]

As the two phenomena are related, the data exhibit trends that correspond to those seen for pressure drop in Fig. 6. As the quality increases, the void fraction and the probability that annular flow (with its

characteristic thin liquid layer) is present increases, resulting in increased heat transfer coefficient. Similarly, for the same channel and quality, increasing mass flux results in higher heat transfer coefficients.

At equivalent channel dimensions, mass flux and quality, a trend of decreasing heat transfer coefficient at higher reduced pressures can be discerned as the saturation temperature is increased from 15 to 25°C (P_r from 0.69 to 0.87). This can be attributed to the same reasons for reduced pressure drop at higher reduced pressures, namely the reduction in liquid/vapor density ratio resulting in decreased flow velocities and interfacial shear stress. For the saturation temperatures considered here, the liquid-to-vapor density ratio is 5.1, 4.0, and 2.9 for $T_{\text{sat}} = 15, 20$ and 25°C, respectively. In addition, the enthalpy of condensation decreases as the temperature increases.

As with pressure drop, an influence of aspect ratio is observed. Fig. 9 shows the data for each test section at $T_{\text{sat}} = 15$ and 25°C and $G = 600$ and 800 kg m⁻² s⁻¹ so that the combined effect of changing hydraulic diameter and aspect can be observed.

[Fig. 9 to be inserted approximately here]

Much of the data at equivalent mass flux, saturation temperature and quality for each tube lie within experimental error; however, some general trends can be extracted. For all conditions, over the range of qualities, the highest heat transfer coefficients are observed in the $D_h = 0.10$ mm, $AR=1$ and $D_h = 0.16$ mm, $AR = 4$ test sections. These are the test sections with the smallest hydraulic diameter and largest aspect ratio, respectively. At $T_{\text{sat}} = 15^\circ\text{C}$, for both mass fluxes, the $D_h = 0.13$ mm, $AR = 2$ case has the third highest condensation heat transfer coefficient over the range of qualities studied, followed by the $D_h = 0.15$ mm, $AR = 3$ test section. At $T_{\text{sat}} = 25^\circ\text{C}$ ($P_r = 0.87$), the difference between these two sections is within the experimental error. These trends are again consistent with those observed by Agarwal [48] for experiments with R-134a. He observed the highest heat transfer coefficient in the channels with the highest aspect ratio ($D_h = 0.16$ mm, $AR = 4$). Clearly, aspect ratio has an effect on heat transfer

coefficient. Through detailed analytical modeling of condensation in small channels, Wang and Rose [56] showed a strong dependence of heat transfer coefficient on microchannel shape, in addition to hydraulic diameter. The observed effects imply that there is an optimum combination of hydraulic diameter and aspect ratio to maximize heat transfer coefficient and minimize pressure drop for a given operating condition. Further insights into the effect of changing hydraulic diameter and aspect ratio could be obtained by varying the two independently.

6. COMPARISON WITH THE LITERATURE

Data from the present study were compared with the predictions or several correlations from the literature developed for macro, mini- and microchannels for two-phase pressure drop and condensation heat transfer coefficient. Many of the correlations were developed for significantly different channel shapes and sizes, fluid types and reduced pressures.

6.1 Pressure Drop Correlations

The data are compared with predictions of the homogeneous mixture model, the correlations of Lockhart and Martinelli [36] and Cavallini *et al.* [25] developed for macrochannels, the correlations of Mishima and Hibiki [57] and Lee and Lee [58] developed for air-water flow in mini- and microchannels, the Agarwal *et al.* [8] correlation for non-circular mini- and microchannels, the Andresen *et al.* [13] correlation for pressure drop of high-reduced pressure refrigerants, and the Garimella *et al.* [59] correlation developed from condensing R-134a data in channels of the same hydraulic diameter as the present study. Of the correlations investigated, four were developed from data for a single tube, two for data from multiple parallel channels, one from a mixed dataset, and the last was the homogenous model. None of the available correlations were developed or verified with data encompassing both the diameter range and fluid properties of interest in the present study.

The results predicted by these correlations for all of the data are presented in Fig. 10. It can be seen that other than the Agarwal and Garimella [8] and Garimella *et al.* [59] correlations, the data were generally

underpredicted. Table A.1 in the Appendix shows a breakdown of the average deviation (AD, Eq. (5)) and absolute average deviation (AAD, Eq. (6)) of each correlation for each test section.

$$AD = \left[\frac{\sum_{i=1}^N (\Delta P_{\text{model},i} - \Delta P_{\text{exp},i})}{N} \right] \cdot 100 \quad (5)$$

$$AAD = \left[\frac{\sum_{i=1}^N |\Delta P_{\text{model},i} - \Delta P_{\text{exp},i}|}{N} \right] \cdot 100 \quad (6)$$

Only the Garimella *et al.* [59] model accounted for channel aspect ratio; thus, it is perhaps unsurprising that the data for the test section with the highest aspect ratio ($AR = 4$) were in general predicted least accurately. The pressure drop trend for each test section over the range of conditions was predicted well by homogeneous model, although the absolute value of the pressure drop was systematically under predicted using this model. Triplet *et al.* [60] showed good agreement of air/water pressure drop measurements in circular ($D = 1.1$ and 1.45 mm) and semi-triangular ($D_h = 1.09$ and 1.49 mm) with the homogeneous model in the bubbly and slug flow regimes. However, as shown previously, the very high surface tension of water in air compared to CO_2 may make the uniform phase velocity assumption necessary for the homogeneous model more realistic for air-water mixtures than for the present case.

[Fig. 10 to be inserted approximately here]

Of the three other two-phase multiplier correlations not developed specifically for microchannels, the Lockhart and Martinelli [36] correlation predicts the data the best (AAD = 61.5%), followed by the correlation of Andresen *et al.* [13] (75.7%) and Cavallini *et al.* [25] (AAD = 82.9%). All three correlations underpredict the data. While showing the best agreement, the results of the Lockhart and Martinelli [36] correlation exhibit larger scatter, most probably due to the step change in the C parameter **Error! Reference source not found.** as the liquid or vapor film Reynolds number transitions between

laminar and turbulent. The Andresen *et al.* [13] and Cavallini *et al.* [25] correlations were both developed from data for refrigerants at intermediate and high reduced pressures, and both include terms to account for the effect of surface tension through the Confinement and Weber number, respectively; however, both still significantly underpredict data. The correlations were developed primarily from data for round tubes with diameters significantly larger than the rectangular channels of interest here, where the flow regime transitions are anticipated to be different and surface tension forces are increasingly important.

The two correlations developed for mini- and microchannel using air-water data under predicted the data for all conditions, with the Mishima and Hibiki [57] correlation, developed for high-aspect rectangular channels and round tubes with air/water data, performing the worst. As seen in Eq. (7), they correlate the Chisholm parameter (C) parameter as a function only of hydraulic diameter.

$$C = 21 \cdot \left(1 - e^{-0.319 \cdot D_h}\right) \quad (7)$$

Thus, at the small hydraulic diameters of interest in this study, C tends towards zero, which effectively eliminates any contribution to frictional pressure drop of the liquid and vapor phase coupling, which is unrealistic. The Lee and Lee [58] correlation, also based on high aspect ratio channels with air/water mixtures, shows better agreement, but still significantly underpredicts the data from the present study. The inclusion of surface tension, slug velocity and liquid Reynolds number probably extends the applicability of the correlation to a greater range than that of Mishima and Hibiki [57], although the fluid properties and geometry in the present study are quite different.

The Agarwal *et al.* [8] (AAD = 26.6%) and Garimella *et al.* [59] (AAD = 24.3%) correlations shows the best agreement of any correlations studied. Both of these approaches were developed for condensing refrigerants in small channels. The Agarwal *et al.* [8] correlation is based on flow visualization, heat transfer and pressure drop studies of R-134a at geometries reasonably close to those of interest in the present study ($D_h = 0.424$ mm, $AR = 2$), and captures the data trend well despite being applied to a different fluid at a smaller diameter. The correlation accounts for surface tension through the same

capillary number as Lee and Lee [58], but although it was developed for rectangular channels, it does not explicitly capture the effects of aspect ratio. Thus, like the other correlations, deviation is the highest at the highest aspect ratio.

The Garimella *et al.* [59] model was developed from condensing R-134a data using the same microchannels of varying aspect ratio as those investigated in the present study. The model considers pressure loss mechanisms in the annular and intermittent flow regimes, and uses a technique to provide a smooth transition between the regimes. This technique is referred to as the annular flow factor (AFF) concept, which quantifies the predominance of annular flow in the channels. As this model accounts for small hydraulic diameter and aspect ratio effects, the overall agreement with the data is best. However, the AFF concept relies on an empirical input for closure. Thus, the large difference in properties between R-134a and CO₂ likely explain the observed deviation between the data and the Garimella *et al.* [59] model. Further refinement of the underlying inputs could extend its range of applicability to the fluid properties of interest in the present study.

6.2 Heat Transfer Correlations

Condensation heat transfer data were compared with eight heat transfer correlations; four developed from data for single large tubes, and four for multiple parallel mini- and microchannels. All eight correlations were generally derived for shear-dominated condensation, which is expected to be most significant in the channels of interest. The approaches taken to predict heat transfer can be divided into two categories: two-phase multiplier correlations [20, 23, 29, 61] and those that take a boundary-layer approach [11, 25, 62, 63].

The predictions of these correlations are compared with the data from the present study in Fig. 11, with a breakdown of the average and absolute average deviation shown in Table A.2 in the Appendix.

[Fig. 11 to be inserted approximately here]

Unsurprisingly, the correlations based on the more accurate pressure drop models (Lockhart and Martinelli [36] and Agarwal and Garimella [8]) provide the best results, with Traviss *et al.* [62] (AAD = 18%), Dobson and Chato [20] (AAD = 19%) and Agarwal *et al.* [11] (AAD=20%) correlations predicting the heat transfer the best. Of these three models, Agarwal *et al.* [11] predicts the data with the least scatter.

Of the two other macro channel correlations, the Shah [23] overpredicts much of the data (AAD = 29%), while the Cavallini *et al.* [25] correlation underpredicts nearly all of the data (AAD = 41%). The pressure drop correlation used in this model also did not yield good predictions, significantly underpredicting all of the data. However, the overall trends in the data appear to be predicted best by the Cavallini *et al.* [25] correlation, indicating that some of the flow mechanisms are being captured properly.

Of the mini- and microchannel correlations, the Wang *et al.* [61] was best (AAD = 26%), followed by the Cavallini *et al.* [63] (AAD=38%) and the Koyama *et al.* [64] (AAD=51%), respectively. The order of agreement of these correlations is matched by that of the pressure drop model used. Clearly, accurately modeling two-phase pressure drop allows for successful implementation of the heat and momentum transfer analogy and yields accurate heat transfer coefficient predictions.

7. CONCLUSIONS

Heat transfer coefficients and pressure drops during the condensation of carbon dioxide in rectangular channels with $0.10 \leq D_h \leq 0.16$ mm and $1 \leq AR \leq 4$ were accurately measured at reduced pressures of 0.69, 0.78 and 0.87 ($T_{\text{sat}} = 15, 20$ and 25°C) and mass fluxes of 400, 600 and $800 \text{ kg m}^{-1} \text{ s}^{-1}$. A segmented model developed by Agarwal [48] for R-134a, which accounted for the conjugate conduction and convection heat transfer effects in the copper test section, was modified to obtain the CO_2 heat transfer coefficients at fine vapor quality increments. By accounting for minor pressure losses/gains due to expansions, contractions and bends in the test section inlet and outlet headers, the frictional pressure drop in the microchannels was determined. The uncertainties in heat transfer coefficients and frictional pressure drops were quantified using a propagation of errors approach.

The data showed an expected trend of increasing heat transfer coefficient and pressure drop at lower reduced pressure and increasing quality. Hydraulic diameter and aspect ratio were not independently varied; therefore, their effects were not explicitly documented. However, it was found that the pressure gradients and heat transfer coefficients followed similar trends, with the highest h and ΔP for the $100 \times 100 \mu\text{m}$ ($D_h = 0.1 \text{ mm}$, $AR = 1$) channel, followed by $D_h = 0.16 \text{ mm}/AR = 4$, $D_h = 0.13 \text{ mm}/AR = 2$, and $D_h = 0.15 \text{ mm}/AR = 3$, in descending order. With further investigations that isolate the effects of these two variables, it may be possible to develop an optimal combination of hydraulic diameter and aspect ratio that yields low pressure drops and high heat transfer coefficients.

The data were compared with the predictions of several heat transfer and pressure drop models developed for a range of working fluids, operating conditions and channel geometries. None of the correlations were developed for condensation of such high pressure refrigerants at comparable hydraulic diameters. Additionally, few of the correlations explicitly accounted for channel aspect ratio. Of the pressure drop models, the best results were obtained from the Garimella *et al.* [59] model (AAD = 24%) developed for condensing R-134a in channels with similar diameter and aspect ratio to those in the present study. The heat transfer correlations [11, 20, 61, 62] based on the more accurate pressure drop correlations showed the best agreement with the data.

Further studies should focus on developing techniques for accurately measuring heat transfer coefficients at low mass fluxes, extending the range of available data. Flow visualization of CO_2 and other refrigerants under various operating conditions would allow verification and quantification of the local void fraction and transition mechanisms between annular and intermittent flow, leading to more reliable and applicable flow-regime based models for both pressure drop and heat transfer coefficient.

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Fig. 1. Comparisons of Refrigerant Reduced Pressures

Fig. 2. (a) Microchannel Test Section Fabrication Pattern, (b) Completed Microchannel Test Section, and (c) Microchannel Test Section with Coolant Blocks

Fig. 3. Experimental Facility Schematic

Fig. 4. Refrigerant Flow Path Schematic for Pressure Drop Analysis

Fig. 5. Heat Transfer Nodes

Fig. 6. Frictional Pressure Gradient Experimental Results

Fig. 7. Effect of Channel Geometry on Frictional Pressure Gradient

Fig. 8. Condensation Heat Transfer Coefficient Experimental Results

Fig. 9. Effect of Channel Geometry on Condensation Heat Transfer Coefficient

Fig. 10. Comparison of Correlations with Frictional Pressure Gradient Data

Fig. 11. Comparison of Correlations with Condensation Heat Transfer Coefficient Data

Table 1. Comparison of Properties

Fluid	T (°C)	P_{sat} (kPa)	P_r (-)	σ (N m⁻¹)	μ_L (kg m⁻¹ s⁻¹)	μ_V (kg m⁻¹ s⁻¹)	ρ_L (kg m⁻³)	ρ_V (kg m⁻³)
Air/Water	25	101	-	7.20×10^{-2}	8.90×10^{-4}	1.85×10^{-5}	997	1.18
R-134a	50	1683	0.41	3.72×10^{-3}	1.24×10^{-4}	1.39×10^{-5}	1053	87.5
CO ₂	20	5729	0.78	1.19×10^{-3}	6.80×10^{-5}	1.81×10^{-5}	773	194

Table 2. Summary of Uncertainty of Measured and Estimated Parameters

	Parameter	Uncertainty
Measured Parameters	Mass flow rate	$\pm 0.2\%$
	CO ₂ temperature	$\pm 0.1^{\circ}\text{C}$
	Ambient temperature	$\pm 2.0^{\circ}\text{C}$
	Channel dimensions	$\pm 0.5\ \mu\text{m}$
	Absolute pressure	$\pm 6.5\ \text{kPa}$
	Differential pressure	$\pm 0.25\ \text{kPa}$
	Pre/post heater power	$\pm 0.2\%$
Calculated/Estimated Parameters	Ambient heat losses	$\pm 50\%$
	Water-side heat transfer coefficient	$\pm 25\%$
	Minor losses	$\pm 50\%$