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Condensation heat transfer coefficient for rectangular multiport microchannels at high ambient temperature

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Abstract

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We experimentally compute the local heat transfer coefficient of blend refrigerant R-410A condensing inside horizontal rectangular multiport aluminium microchannels with hydraulic diameters equal to 0.52 mm and 1.26 mm. The refrigerant flows at near-critical pressure and the cooling air flows at high temperatures proper of hot climates. The experiments are conducted in a bespoke experimental facility and micro-foil sensors are used to measure the local condensation heat flux. The heat transfer coefficient is found to increase with the mass flow rate per unit area and the vapour quality and to decrease with the ambient temperature. Correlations available in the literature do not predict our experimental data satisfactorily because of our extreme operating conditions of high pressure and high cooling air temperature. A novel correlation is therefore obtained to successfully compute the Nusselt number for the condensing annular flow regime in our high pressure and high temperature conditions.

1 Introduction

Microchannels are increasingly being utilized in condensers. The rapid development in aluminium extrusion and brazing processes have led to the wide use of this type of condensers. Microchannels produced by these manufacturing technologies have a non-circular multiport structure, with hydraulic diameters of the order of one millimetre. When microchannel condenser with rectangular cross sections are compared with traditional condensers with round cross sections, the former are usually lighter in weight and more compact, and a smaller amount of fluid can be used to dissipate the same heat as in conventional condensers. This helps reduce the energy loss and is beneficial to

26 the environment. Furthermore, these microchannel condensers are successfully used with
27 blend refrigerants because of their capability to operate under high pressure, which is
28 typical of these refrigerants. Therefore, the technology of these microchannel condensers
29 has paved the way for modern industrial applications, such as electronic cooling for space
30 technology and hot climate air conditioning.

31 Several studies have focused on measuring and modelling the condensation heat trans-
32 fer coefficient. Most of these correlations have been developed for flows through large and
33 circular tubes. However, the behaviour of condensing two-phase flow inside non-circular
34 channels, especially with a small hydraulic diameter, is significantly different from those
35 in large circular channels. The condition of near-critical pressure has a significant im-
36 pact on the condensation heat transfer because the fluid properties in the liquid-vapour
37 dome are considerably different from those at low pressures. Moreover, the coolant type
38 and the cooling conditions targeted to several industrial applications further render the
39 correlations problem specific. It is therefore expected that research works focused on
40 heat transfer in non-circular microchannels with hydraulic diameter smaller than one
41 millimetre are very limited in the literature.

42 Researchers, such as [Yang and Webb \(1996a,b, 1997\)](#), [Webb and Ermis \(2001\)](#), and
43 [Zhang and Webb \(2001\)](#), have addressed the challenge of measuring and correlating the
44 heat transfer coefficient in horizontal aluminium non-circular multiport channels with
45 hydraulic diameter close to or less than a millimetre for both smooth and internally micro-
46 finned channels. Several correlations have been proposed for the heat transfer coefficient,
47 where the wall-shear stress and the surface tension were included in the formulation.

48 Investigations on the heat transfer in microchannels with blend refrigerants are even
49 more limited. [Kim et al. \(2003\)](#) conducted an experimental study similar to that of
50 [Yang and Webb \(1996a,b, 1997\)](#) in flat aluminium multichannels with hydraulic diameters
51 of about 1.5 millimetres. Tubes with internal microfins were also tested. The working
52 fluid was R-410A and the results were compared with those of R-22. They found that, at
53 low mass flow rates per unit area, the heat transfer coefficients of R-410A in micro-finned
54 channels were higher than those in smooth channels, and that the heat transfer coefficient
55 decreased with an increase of mass flow rates per unit area. The heat transfer coefficients
56 of R-410A were slightly higher than those of R-22 for smooth channels and the opposite
57 was true for micro-finned channels. Their correlations predicted the experimental data
58 within $\pm 30\%$.

59 [Agarwal et al. \(2010\)](#) conducted analytical and experimental work for determining
60 the condensation heat transfer coefficients of R-134a in six non-circular horizontal mi-
61 crochannels with hydraulic diameters smaller than a millimetre. The thermal amplifica-
62 tion technique developed by [Garimella and Bandhauer \(2001\)](#) was used to measure the
63 heat transfer in small increments of vapour quality across the liquid-vapour dome. It was
64 recommended that the annular flow model be utilized for the squared, parallel-shaped,
65 and rectangular channels, while the mist flow model was used for channels with sharp
66 corners. When comparing these measured data with other correlations, the significant
67 deviation was attributed to the large diameters for which these models were developed.

68 [Cavallini et al. \(2005\)](#) experimentally measured the heat transfer coefficient and the
69 pressure drop during the condensation of R-134a and R-410A inside multiple parallel
70 1.4 mm hydraulic diameter channels. The experimental data were compared with dif-
71 ferent models from the literature and good agreement was obtained with the data of

72 R-134a, while the same correlations overpredicted the data of R-410A. The discrepancy
73 was attributed to the effect of their smaller diameter compared to those used in other
74 correlations.

75 The analytical study of Wang and Rose (2005a) showed that the sharpness in the
76 corners of the non-circular channels has a major influence in the condensation process
77 and the liquid film generation. Wang and Rose (2005b) provided a theoretical model
78 for the heat transfer coefficient during condensation in microchannels with squared and
79 triangular cross sections in the hydraulic diameter range of 0.5 – 5 mm. The mass flow
80 rate per unit area was in the range of 100 – 1300 kg/m²s for R-134a, R-22, and R-410A.
81 The model was proposed for the annular regime and it was based on the assumption of
82 laminar film condensation flow on the internal walls of the microchannel. The model ac-
83 counted for the effect of surface tension, interfacial shear stress, and gravity. A significant
84 enhancement of the heat transfer coefficient occurred near the channel entrance due to
85 the surface tension. A general agreement was obtained when the theoretical results were
86 compared with relevant experimental data.

87 Wu et al. (2009) theoretically developed a three dimensional simulation model for the
88 condensation heat transfer in the annular regime in rectangular channels. The circum-
89 ference of the inner tube surface was covered with the liquid film and two regions could
90 be distinguished: the thin film region and the meniscus region. The thickness of the
91 condensed film, the wall temperature, and the heat transfer coefficient were computed
92 and the difference between the film and the meniscus condensations in the annular flow
93 regime was addressed.

94 Garimella et al. (2016) conducted experimental work on near-critical heat transfer
95 with refrigerants R-404A and R-410A in single horizontal round tubes of diameters equal
96 to 3.1, 6.2, and 9.4 mm, and in multiport tubes of diameters equal to 0.76 and 1.52 mm.
97 They used the facility employed earlier by Garimella and Bandhauer (2001) with some
98 modifications to operate at near-critical pressure. They measured the local heat transfer
99 coefficient in small differences of quality for mass flow rate per unit area ranging from 200
100 to 800 kg/m²s. They found that the existing equations failed to predict the experimental
101 pressure drop and the heat transfer coefficient. Therefore, they used a multi-regime heat
102 transfer model (wavy, annular, and annular/wavy regimes) and found that the annular
103 regime was relevant for microchannel flows. They also proposed an experimental formula
104 based on the Martinelli parameter.

105 Garimella et al. (2015) experimentally investigated the heat transfer of blend refriger-
106 ants R-404A and R-410A in horizontal circular tubes of diameters ranging between
107 0.76 and 9.4 mm at reduced pressure of $p_r=1, 1.1, \text{ and } 1.2$. The heat transfer coefficient
108 was computed using the overall resistance method. It was found that the spikes of the
109 heat transfer coefficient occurred because of the sharp deviation of the thermodynamic
110 properties at critical temperature. Also, the temperature change had much more effect
111 on the heat transfer coefficient than on the individual change of mass flow rate per unit
112 area. The experimental data were compared with the models for CO₂ flow in similar
113 conditions and the results were in poor agreement. They also proposed a new correlation
114 for the gas-coolant heat transfer coefficient at supercritical pressure that predicted the
115 experimental data successfully.

116 We conclude that there is a dearth of reliable models for the condensation heat transfer
117 coefficient of blend refrigerants flowing in non-circular microchannels with small diame-

118 ters, operating at near-critical pressure and cooled by air at high ambient temperature.
119 The main objective of this study is therefore, for the first time, to investigate the heat
120 transfer performance of the refrigerant R-410A flowing through horizontal rectangular
121 multiport channels during air-cooled condensation at near-critical pressure and high am-
122 bient temperature. A bespoke experimental facility was designed and built for this pur-
123 pose. The latest technology of the micro-foil heat flux sensor technique was utilized to
124 measure the condensation heat transfer through the microchannel condenser.

125 Section 2 describes the laboratory apparatus (§2.1), the reduction of the experimental
126 data (§2.2), the heat transfer measurements in single-phase flow conditions (§2.3), and
127 the uncertainty analysis (§2.4). Section 3 presents the results on the dependence of the
128 condensation heat transfer coefficient on the parameters of the system (§3.1), the predic-
129 tion of our experimental data via existing correlations (§3.2), and our novel correlation
130 for the heat transfer coefficient (§3.3). Section 4 discusses the conclusions of our work.

131 2 Experimental apparatus and procedures

132 Figure 1 shows the schematic of the experimental apparatus. The sub-cooled liquid
133 refrigerant R-410A, circulated by a variable speed gear pump, flowed into a pre-heater
134 evaporator where it was heated by electrical heaters with a total capacity of 1300 W. The
135 capacity of the heaters was controlled by a variable transformer to achieve the required
136 saturation condition. The refrigerant vapour, generated at the evaporator, entered the
137 test section where it was cooled by an air stream. The condensation process then occurred
138 and the refrigerant phase changed along the channel. The phase of the refrigerant before
139 and after the test section was visualised using a pair of sight glasses. The condensed
140 refrigerant from the test section entered a water-cooled sub-cooler to guarantee that all
141 the refrigerant returned to the initial condition of liquid phase. The sub-cooled liquid
142 refrigerant flowed back to the pump through a Coriolis-effect mass flow meter and a liquid
143 receiver.

144 The system operating pressure was controlled by an accumulator and a regulating
145 valve, utilizing the nitrogen pressure to stabilise the refrigerant system pressure to the
146 desired value. The amount of circulated refrigerant was controlled by regulating the speed
147 of the pump and by adjusting the flow control valve. A filter dryer was fixed in the section
148 line to remove any possible moisture from the refrigerant during the refrigerant charging
149 process and the channel replacement. The temperature and pressure before and after the
150 evaporator were measured by thermocouples and pressure transducers, respectively. The
151 pressure drop across the test section was measured by a differential pressure transducer.

152 2.1 Test section

153 The test section, shown in Fig. 2, was composed of two parts: the air duct and the mi-
154 crochannel tubing assembly. The cross-flow air stream flowing over the microchannel was
155 supplied by the air duct and extracted through the duct by a centrifugal fan. The inlet
156 temperature of the cooling air was controlled by a duct heater positioned upstream of
157 the fan and by an integrated temperature control system to guarantee a low level of tem-
158 perature fluctuations. The tubing assembly was composed of the rectangular aluminium
159 microchannels, mounted horizontally in the air duct and connected to the refrigerant loop

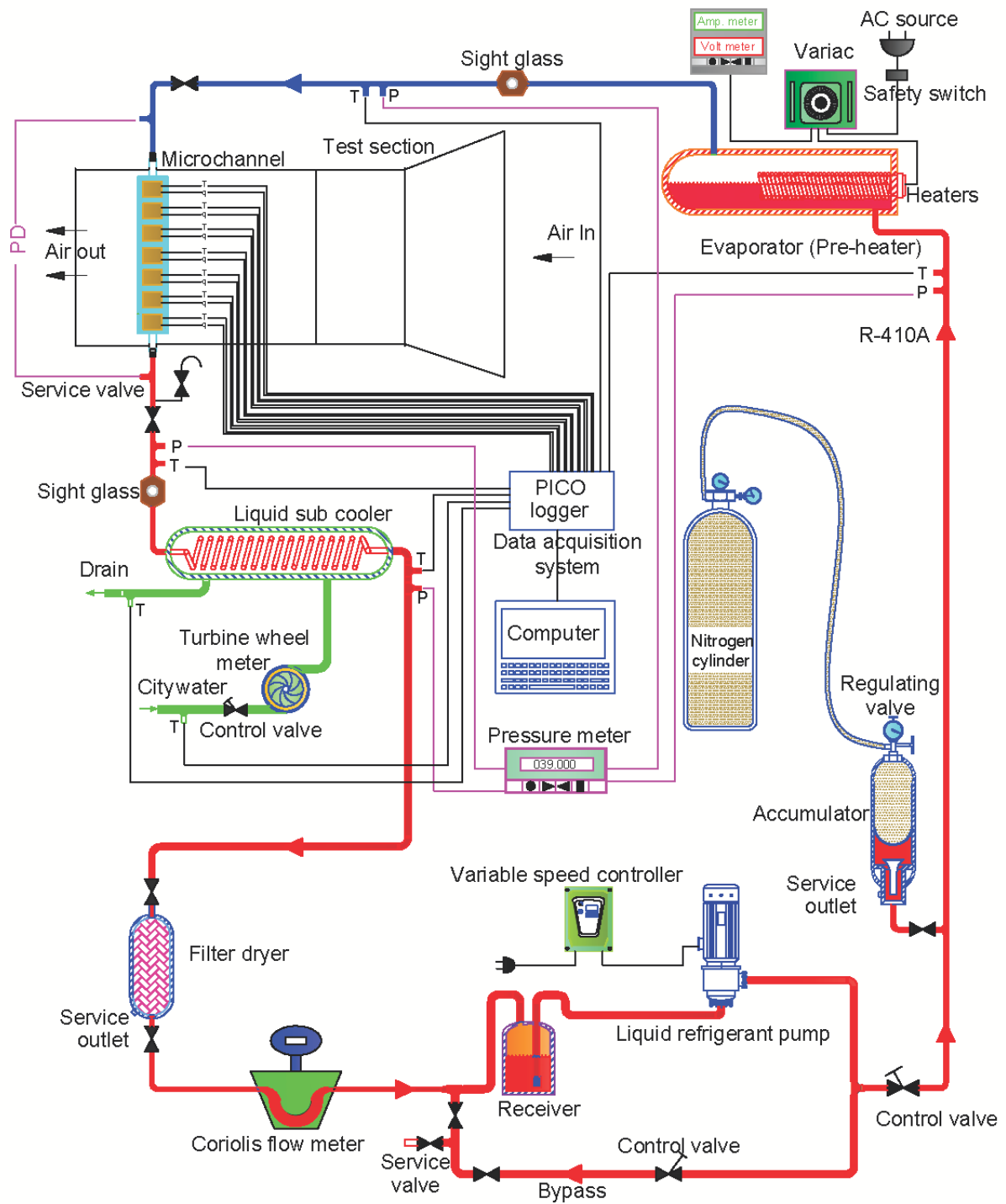


Figure 1: Schematic diagram of the test facility.

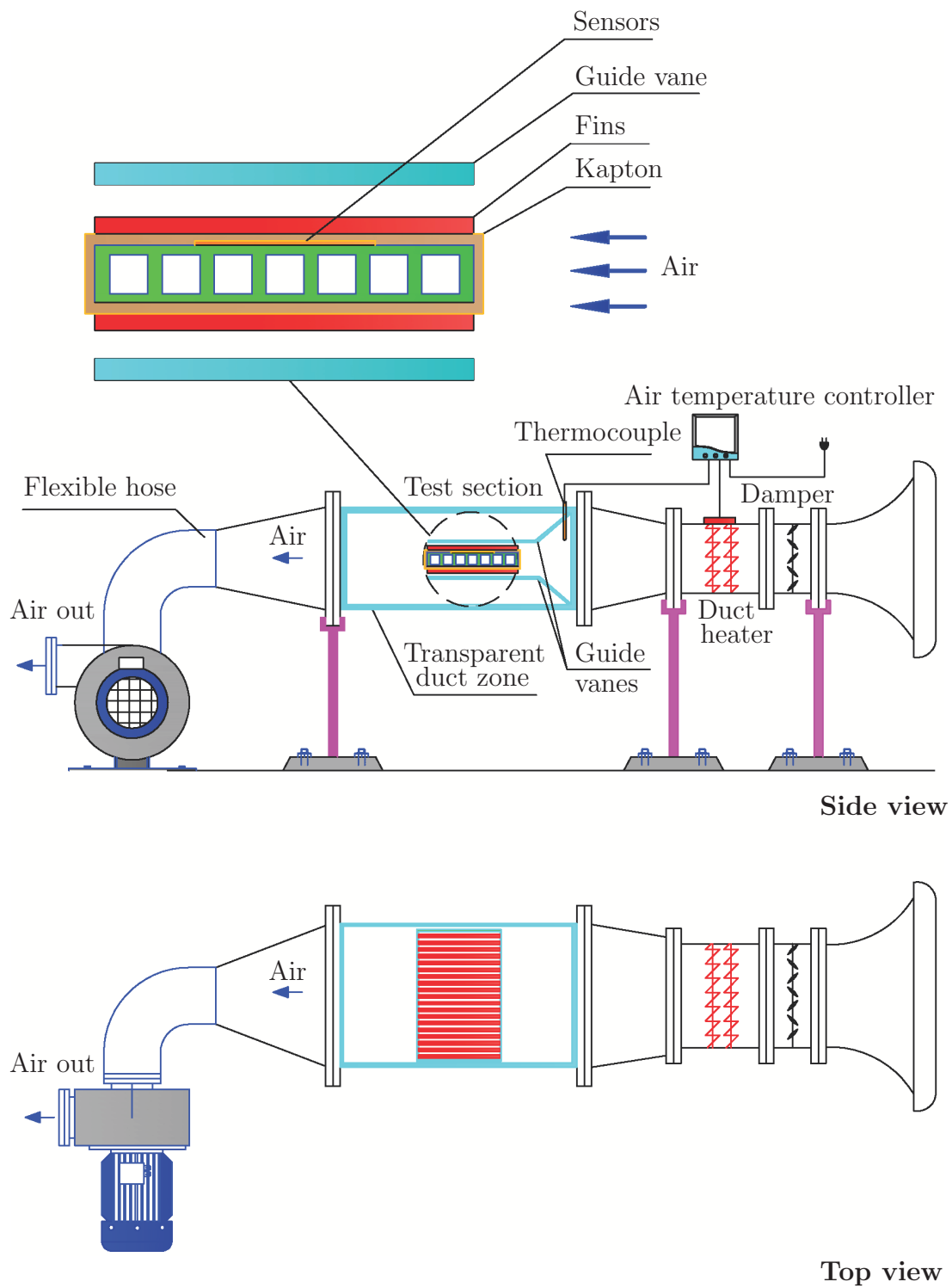


Figure 2: Schematic of the test section.

160 with a pair of adapters. A sketch of the cross sections of the channels is shown in Fig. 3.
 161 Table 1 provides the dimensions of the channels.

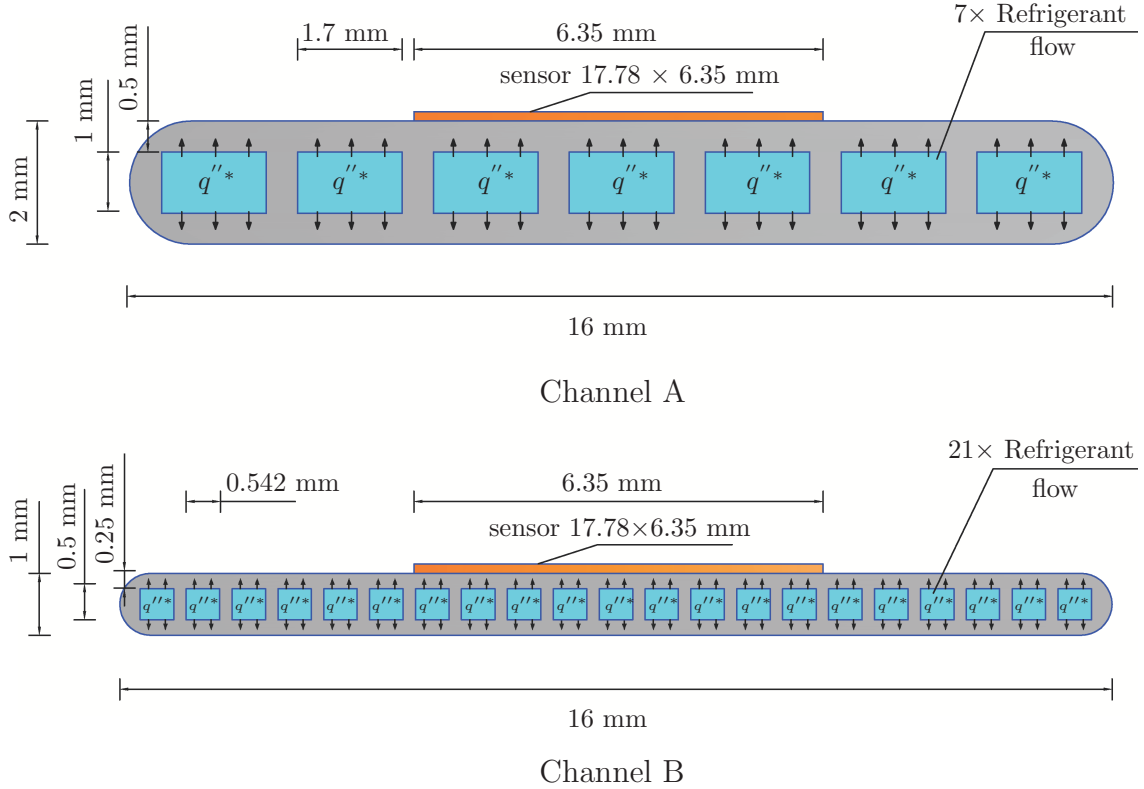


Figure 3: Schematic of the channel cross sections.

Tube type	Type A	Type B
Number of channels	7	21
Hydraulic diameter (mm)	1.26	0.52
Channel width (mm)	16	16
Channel length (m)	0.49	0.45

Table 1: Dimensions of the microchannels.

162 Micro-foil heat flux sensors were utilized to measure the local heat flux and the outer
 163 surface temperature simultaneously during the condensation process along the channel.
 164 A significant feature of the condensation process inside the microchannels is the low
 165 mass flow rate per unit area corresponding to a high heat transfer coefficient, which
 166 renders these measurements challenging. The specifications of these micro-foil sensors
 167 are presented in Table 2. Five micro-foil sensors were mounted to the channel with the
 168 1.26 mm hydraulic diameter, while seven micro-foil sensors were mounted to the channel
 169 with the 0.52 mm hydraulic diameter. A data logger employing a pico-software was used
 170 to record the sensor readings. The micro-foil sensors were fixed on the outer channel
 171 surface at regular intervals along the channel length. These sensors were mounted by
 172 wrapping sticky Kapton strips around the channel. To replicate the geometry of the

173 compact air condenser, straight aluminium fins were placed on the upper and lower outer
 174 surface of the channel. They were fixed using a very thin film of thermal paste that
 175 introduced a negligible thermal resistance because of its very high thermal conductivity.

Model	Dimension (mm)	Thermo-couples type	Nominal sensitivity $\left(\frac{\mu\text{V}}{\text{W/m}^2}\right)$	Maximum heat flux $< 60^\circ\text{C}$ (W/m^2)	Time constant (s)	Maximum operating temp. $(^\circ\text{C})$
27036-1 RdF	$6.35 \times 17.78 \times 0.076$	T	0.032	568000	0.05	260

Table 2: Specifications of the micro-foil sensors.

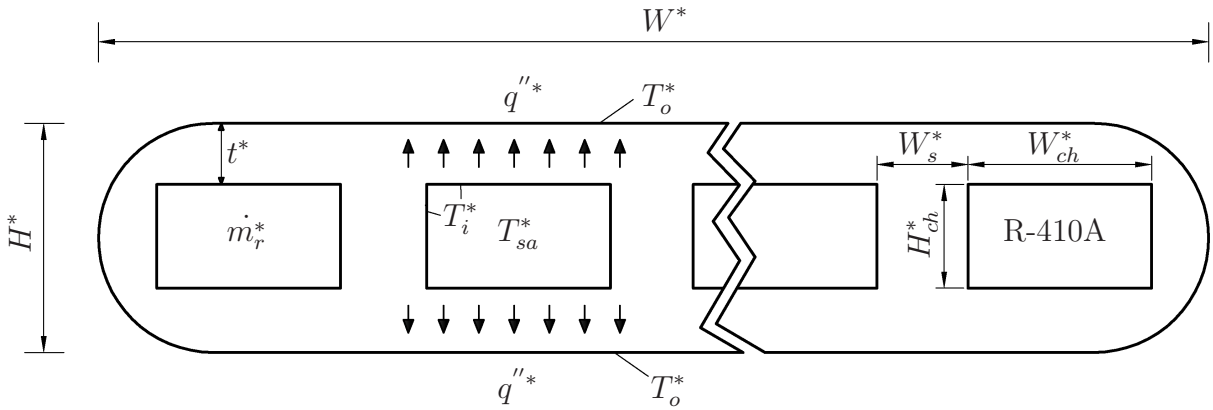


Figure 4: Schematic of channel cross section showing the system parameters used to determine the heat transfer coefficient.

176 2.2 Data reduction

177 Figure 4 shows the key parameters used to determine the heat transfer coefficient. The
 178 local heat flux during the condensation of R-410A was measured directly by the micro-foil
 179 heat flux sensors and it was assumed uniform at the outer surface area for each interval
 180 along the channel. The main assumptions of the approach were steadiness and one-
 181 dimensionality of the heat transfer conduction along the vertical direction, homogeneity
 182 and isotropy of the thermal conductivity k_{ch}^* of aluminium, and uniformity of the heat
 183 transfer coefficient h^* and of the saturation temperature T_{sa}^* of the refrigerant in cross-
 184 sectional planes. Dimensional quantities are henceforth indicated by the superscript $*$.

185 Thanks to these assumptions and to the symmetry of the channel with respect to the
 186 horizontal middle line, each side wall of height H_{ch}^* separating the microchannels could

187 be treated as two rectangular symmetrical fins. The length of each fin was equal to $H_{ch}^*/2$
 188 and the common fin tip was adiabatic because of the symmetry. The computation of the
 189 convection coefficient could thus be carried out by analyzing one half of the channel. From
 190 the assumption of one-dimensionality it follows that the temperature of the fin base was
 191 equal to the temperature T_i of the top (or bottom) internal surface of each microchannel.
 192 As discussed by [Qu and Mudawar \(2003\)](#) and [Kim and Mudawar \(2010\)](#), the heat transfer
 193 from both sides of each half microchannel was $\dot{Q}_s^* = \eta_{fin} h^* \Delta L^* H_{ch}^* (T_{sa}^* - T_i^*)$, where ΔL^*
 194 is the length of each measuring interval along the multiport microchannel, η_{fin} is the fin
 195 efficiency:

$$\eta_{fin} = \frac{\tanh(m^* H_{ch}^*/2)}{m^* H_{ch}^*/2}, \quad (1)$$

196 $m^* = \sqrt{2h^*/(k_{ch}^* W_s^*)}$, and W_s^* is the width of the wall between two adjacent microchan-
 197 nels. The heat transfer through the microchannel top (or bottom) internal surface
 198 was $\dot{Q}_b^* = h^* \Delta L^* W_{ch}^* (T_{sa}^* - T_i^*)$, where W_{ch}^* is the width of each microchannel. The
 199 heat transfer to the exterior from one half of the microchannel was $\dot{Q}_{ch}^* = \dot{Q}_s^* + \dot{Q}_b^* =$
 200 $q''^* \Delta L^* (W_s^* + W_{ch}^*) = h^* \Delta L^* (T_{sa}^* - T_i^*) (\eta_{fin} H_{ch}^* + W_{ch}^*)$, where q''^* is the external heat
 201 transfer flux per unit area that was measured directly by the heat-flux sensors and was
 202 assumed uniform for each interval along the channel. The fin efficiency η_{fin} was com-
 203 puted to be higher than 0.95 and thus assumed to be equal to unity. This proves that
 204 the temperature gradient along the side walls could be neglected and the temperature of
 205 the side walls could be assumed uniform and equal to T_i^* .

206 The inner surface temperature T_i^* was calculated by the one-dimensional heat conduc-
 207 tion balance,

$$T_i^* = T_o^* + \frac{q''^* t^*}{k_{ch}^*}, \quad (2)$$

208 where T_o^* is the measured outer surface temperature and t^* is the thickness of the alu-
 209 minium layer separating the microchannels and the exterior.

210 The local heat transfer coefficient of condensing R-410A was determined as follows:

$$h^* = \frac{q''^* (W_s^* + W_{ch}^*)}{(T_{sa}^* - T_i^*) (H_{ch}^* + W_{ch}^*)} = \frac{2W^*}{NP^*} \frac{q''^*}{(T_{sa}^* - T_i^*)}, \quad (3)$$

211 where $W^* = N(W_s^* + W_{ch}^*)$ is the multiport microchannel width and $P^* = 2(H_{ch}^* + W_{ch}^*)$ is
 212 the wetted perimeter of each microchannel. Note that the first expression in (3) coincides
 213 with equation (2) in [Kim and Mudawar \(2010\)](#) when $\eta_{fin} = 1$. The factor 2 multiplying
 214 H_{ch}^* in their equation (2) is absent in (3) because their microchannel height coincides with
 215 half of ours because of the symmetry of our system. The assumption of the temperature
 216 of the fin base being equal to T_i^* is further supported by the maximum Biot number of the
 217 half wall separating two microchannels being $Bi = h^* W_s^*/(2k_{ch}^*) = 0.01$, thus sufficiently
 218 small for the one-dimensionality approximation to be valid.

219 The local Nusselt number of condensing R-410A was:

$$Nu = \frac{h^* D_h^*}{k_l^*}, \quad (4)$$

220 where $D_h^* = 2W_{ch}^* H_{ch}^*/(W_{ch}^* + H_{ch}^*)$ is the hydraulic diameter of the channels and k_l^*
 221 is the thermal conductivity of the liquid phase. Another quantity of interest was the

222 vapour quality, i.e., the mass fraction of vapour in the saturated mixture, denoted by x .
 223 The vapour quality at the inlet of the test section, x_{in} , was determined from the energy
 224 balance of the evaporator as:

$$x_{in} = \frac{1}{h_{fg}^*} \left(h_{e,i}^* + \frac{\dot{Q}_e^* - \dot{Q}_{loss}^*}{\dot{m}_r^*} - h_{e,l}^* \right), \quad (5)$$

225 where $\dot{Q}_e^* = I^*V^*$ is the electrical-heater capacity of the evaporator, I^* is the total
 226 electrical current of all working heaters, V^* is the voltage, \dot{Q}_{loss}^* is the heat loss to the
 227 environment from the evaporator, \dot{m}_r^* is the refrigerant mass flow rate, h_{fg}^* is the latent
 228 heat of condensation at the inlet of the test section at its saturation pressure, $h_{e,l}^*$ is the
 229 saturated liquid enthalpy at the evaporator pressure, and $h_{e,i}^*$ is the liquid enthalpy at
 230 the evaporator inlet. The evaporator efficiency was determined when the fluid in the
 231 entire rig was in the liquid phase and it could be used when the two-phase flow condition
 232 occurred. The evaporator efficiency was defined as:

$$\eta_e = \frac{\dot{Q}_e^* - \dot{Q}_{loss}^*}{\dot{Q}_e^*} = \frac{\dot{m}_r^*(h_{e,o}^* - h_{e,i}^*)}{\dot{Q}_e^*}, \quad (6)$$

233 where $h_{e,o}^*$ is the liquid enthalpy at the evaporator outlet. Equation (5) becomes:

$$x_{in} = \frac{1}{h_{fg}^*} \left(h_{e,i}^* + \frac{\dot{Q}_e^* \eta_e}{\dot{m}_r^*} - h_{e,l}^* \right). \quad (7)$$

234 The change of vapour quality at each interval along the test section was calculated from
 235 the energy balance at different test section intervals as:

$$\Delta x = \frac{2W^* q''^*}{\dot{m}_r^* h_{fg}^*} \Delta L^*. \quad (8)$$

236 We also computed G^* , the mass flow rate per unit area, by dividing the mass flow rate by
 237 the cross-sectional area of the channels, and the reduced pressure p_r as the ratio of the
 238 refrigerant condensation pressure to its critical pressure. The thermodynamic properties
 239 of the refrigerant R-410A were calculated using the database of NIST REFPROP version
 240 9.1 (Lemmon et al., 2013) and the tabulated values in the handbook by ASHRAE (2017).

241 2.3 Verification of heat transfer in single-phase flow conditions

242 Single-phase heat transfer tests were run to verify the thermal performance of the ex-
 243 perimental apparatus and instrumentation. The apparatus was operated with the liquid-
 244 phase at a system pressure in the range of 30 – 35 bar, a refrigerant temperature in the
 245 range of 35 – 40°C, and a mass flow rate per unit area in the range of 400 – 800 kg/m²s.
 246 As the heat was extracted by the flowing air, the refrigerant remained in the liquid phase.
 247 The energy balance across the test section was applied between the refrigerant side and
 248 the average of the sensor measurements of the heat flux. The energy balance index was
 249 defined as:

$$e = \frac{(1/N) \sum_{i=1}^N q_i''^*}{\dot{m}_r^* (h_{l,o}^* - h_{l,i}^*) / A_s^*}, \quad (9)$$

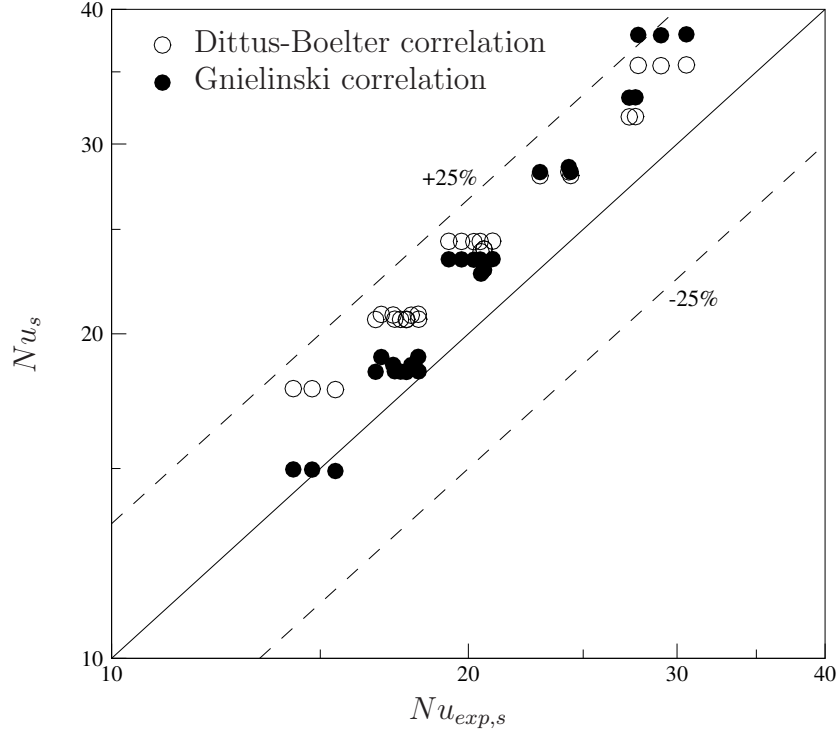


Figure 5: Comparison between our measured single-phase Nusselt numbers $Nu_{exp,s}$ and those computed by the Dittus-Boelter correlation (Dittus and Boelter, 1930) and by the Gnielinski correlation (Gnielinski, 1976).

250 where $h_{l,i}^*$ and $h_{l,o}^*$ are the liquid enthalpies of the flow before and after the test section,
 251 respectively, $q_i''^*$ is the heat flux for each sensor, N is the number of heat flux sensors,
 252 and A_s^* is the outer surface area of the channel. The values of e were in the range of
 253 $0.87 - 0.96$.

254 A comparison between our measured single-phase Nusselt numbers and
 255 those calculated by the Dittus-Boelter correlation, $Nu_s = 0.023Re^{0.8}Pr^{0.3}$
 256 (Dittus and Boelter, 1930), and by the Gnielinski correlation, $Nu_s = (f/8)(Re -$
 257 $1000)Pr / [1 + 12.7(f/8)^{0.5}(Pr^{2/3} - 1)]$ (Gnielinski, 1976) (where Pr is Prandtl number
 258 and Re is the single-phase Reynolds number based on D_h^* and the mean velocity),
 259 was carried out as a further verification of the experimental procedures. The friction
 260 coefficient f in the Gnielinski correlation was computed by the first Petukhov correlation,
 261 $f = (0.79 \ln Re - 1.64)^{-2}$ (Petukhov, 1970). Figure 5 shows that our data agree
 262 better with those computed via the Gnielinski correlation than with those obtained
 263 via the Dittus-Boelter correlation, which confirms the discussion on page 319 in
 264 Kays and Crawford (1993).

265 2.4 Experimental uncertainty

266 The experimental uncertainties of the measured parameters were obtained from the cal-
 267 ibration of the measuring instruments provided by the manufactures. As an important
 268 check of the experimental facility, the measured temperature of the refrigerant at the
 269 inlet of the test section was compared with the saturation temperature obtained from the

270 saturation pressure. The disagreement was below 0.17°C. The uncertainty of the heat
 271 transfer coefficient, U_h^* , was dominated by the uncertainties of the heat flux, U_q^* , and of
 272 the temperature difference, $U_{(T_{sa}-T_i)^*}$. It was determined using the method proposed by
 273 Moffat (1988) as:

$$U_h^* = \sqrt{\left[\frac{2W^*}{NP^* (T_{sa}^* - T_i^*)} U_q^* \right]^2 + \left[\frac{2W^*}{NP^* (T_{sa}^* - T_i^*)^2} U_{(T_{sa}-T_i)^*} \right]^2}. \quad (10)$$

274 The estimated uncertainties of the experimental parameters are presented in Table 3. The
 275 dimensions of the cross section of the multiport microchannels were measured using an
 276 optical microscope. The channel width was measured by a micrometer with a maximum-
 error uncertainty of 0.001 mm.

Parameter	Uncertainty (%)
Temperature	±0.35
$T_{sa}^* - T_i^*$	±0.48
Pressure	±0.025
Pressure difference	±0.015
Refrigerant mass flow rate	±0.03
Heat flux	±1.7
Heat transfer coefficient	±14

Table 3: Uncertainties of the measured quantities.

277 The mean absolute percentage error between the experimentally measured Nusselt
 278 numbers Nu_{exp} and the Nusselt numbers Nu_{pred} predicted by empirical correlations is
 279 defined as
 280

$$E(\%) = \frac{100\%}{M} \sum_{i=1}^M \frac{|Nu_{i,pred} - Nu_{i,exp}|}{Nu_{i,exp}}. \quad (11)$$

281 3 Results

282 Tests were conducted on the refrigerant flowing through the microchannels at the two
 283 reduced pressures of 0.7 and 0.8 over a range of mass flow rate per unit area of 200 –
 284 800 kg/m²s with a vapour quality ranging between 0.1 and 0.8 along the channel. The
 285 condensation process was achieved by utilizing cooling air at the two ambient tempera-
 286 tures of 35°C and 45°C.

287 3.1 Condensation heat transfer coefficient

288 Figures 6 and 7 illustrate the variation of the local heat transfer coefficient h^* with the
 289 vapour quality x for different mass flow rates per unit area for the two reduced pressures,
 290 the two ambient air temperatures, and the two channels. The heat transfer coefficient
 291 increases with the mass flow rate for all configurations and this change is more significant
 292 at high vapour qualities and at high mass flow rates. For small mass flow rates, the heat
 293 transfer coefficient changes only slightly as the vapour quality varies considerably.

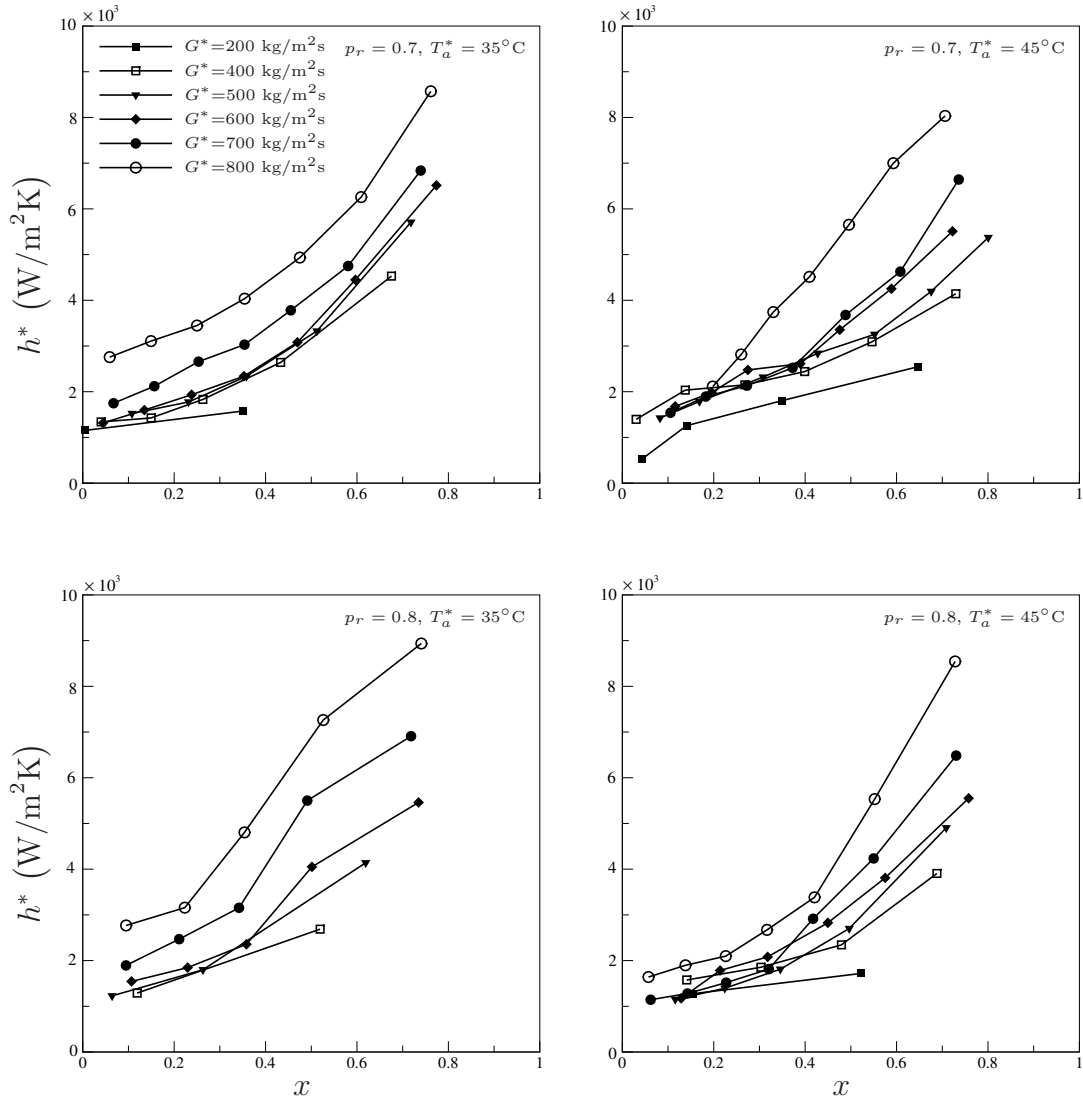


Figure 6: Local condensation heat transfer coefficient h^* of R-410A as a function of the vapour quality x for the two reduced pressures and the two ambient air temperatures for different mass flow rates per unit area. The hydraulic diameter is $D_h^* = 0.52$ mm.

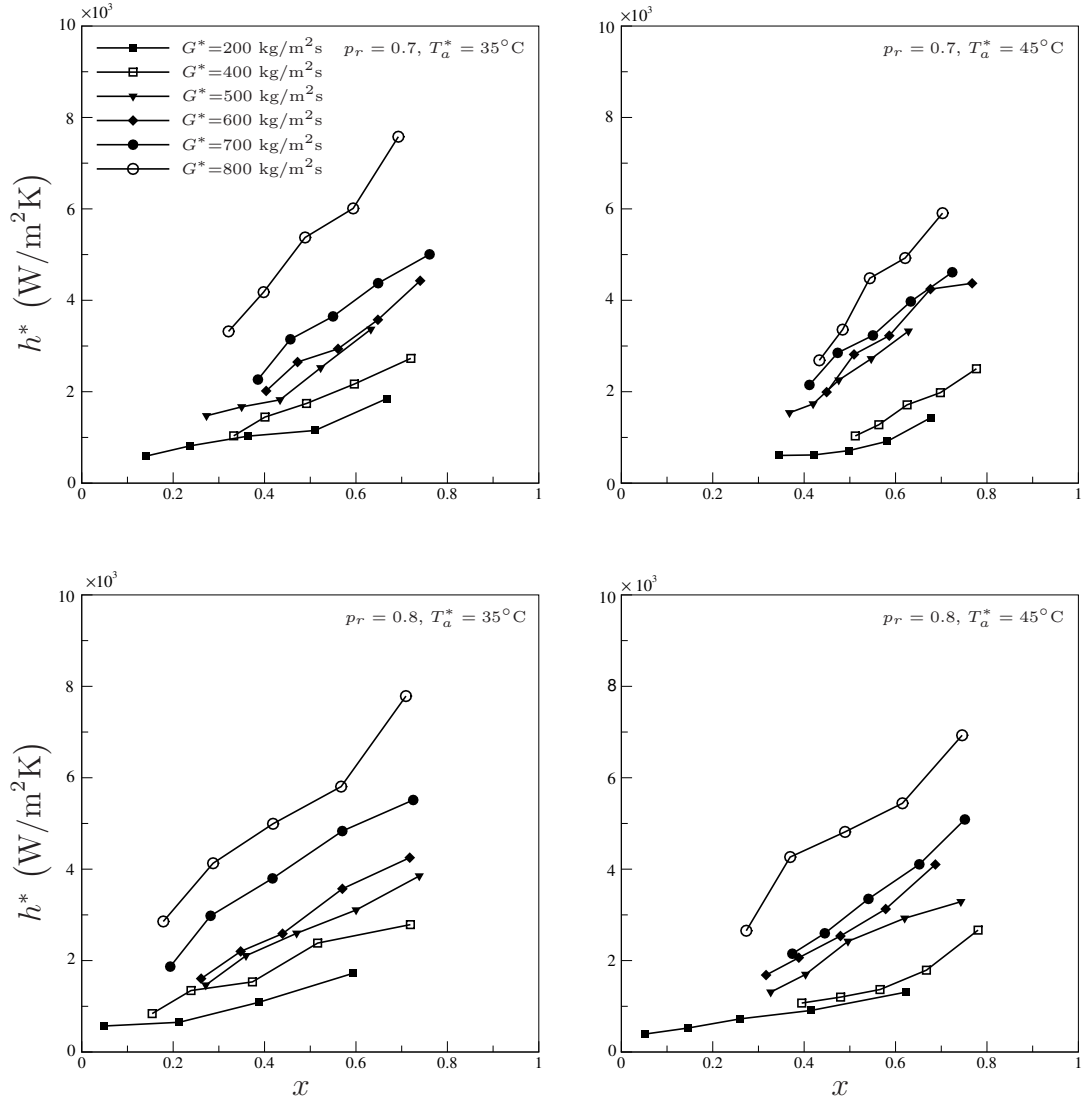


Figure 7: Local condensation heat transfer coefficient h^* of R-410A as a function of the vapour quality x for the two reduced pressures and the two ambient air temperatures for different mass flow rates per unit area. The hydraulic diameter is $D_h^* = 1.26$ mm.

294 When the condensation process evolves along the channel, the vapour quality sub-
295 stantially decreases from the inlet value. The annular flow regime often occurs due to
296 the channel wall temperature being colder than the fluid temperature, thus creating a
297 liquid film that covers the perimeter of the channel while the vapour flows in the core.
298 The thickness of the liquid film is thus smallest at the inlet where the vapour quality is
299 high. The liquid film thickens as the refrigerant flows along the channel during the con-
300 densation process. As a consequence, the heat transfer coefficient is high at high vapour
301 quality because the liquid film that forms on the internal channel surface is thin as the
302 refrigerant is still mostly in the vapour phase. The heat transfer coefficient is large when
303 the vapour quality is large also because of the high specific volume of the vapour phase,
304 which leads to a high vapour core velocity. The heat transfer coefficient in the channel
305 with the small hydraulic diameter, shown in Fig. 6, is higher than in the larger channel,
306 shown in Fig. 7, particularly at high mass flow rates and vapour qualities. As expected,
307 less scatter in the heat transfer coefficient values is found for the larger diameter as the
308 uncertainty of the measurements is lower for that channel. We also observe that at low
309 vapour qualities the heat transfer coefficient of the refrigerant in the small channel varies
310 only slightly although the mass flow rate increases by four times. The slope of the heat
311 transfer coefficient curves becomes sharper for the channel with the small hydraulic di-
312 ameter, particularly at high mass flow rates. For small vapour qualities, $x < 0.3$, the
313 small-diameter heat transfer coefficient data collapse for most of the mass flow rates.
314 This is likely to be due to the thickness of the liquid film in the annular regime remain-
315 ing constant. When the reduced pressure increases at constant cooling air temperature,
316 the latent heat of condensation decreases because of the thermodynamic properties of
317 the working fluid. The temperature difference between the inner wall and the saturated
318 fluid increases due to increase of the saturation temperature at approximately constant
319 wall temperature. Therefore, there is no clear evidence of the relation between the heat
320 transfer coefficient and reduced pressure, as shown in Fig. 8.

321 Figure 9 shows the effect of the ambient air temperature on the local condensation
322 heat transfer coefficient. When the reduced pressure, the mass flow rate per unit area,
323 and the range of the temperature difference between the inner wall and the saturated
324 fluid are constant, the heat transfer coefficient is slightly higher at lower ambient air
325 temperature. Figures 10 and 11 show that the temperature difference between the inner
326 wall temperature and the saturation temperature increases along the channel due to
327 the decrease of the wall temperature. The saturation temperature of the refrigerant is
328 approximately constant along the channel and only very slightly affected by the pressure
329 drop. When the flow is annular, the liquid film thickness increases along the channel due
330 to the condensation process, which leads to a reduction of the wall temperature. The
331 increase of the cooling effect of the wall temperature is linked with the decrease of the
332 condensation heat transfer coefficient. This decrease is due to the development of the
333 liquid layer on the inner perimeter of the multiport channel. The heat transfer coefficient
334 is high when most of the fluid is in the gaseous phase and decreases when the phase
335 changes from gaseous to liquid.

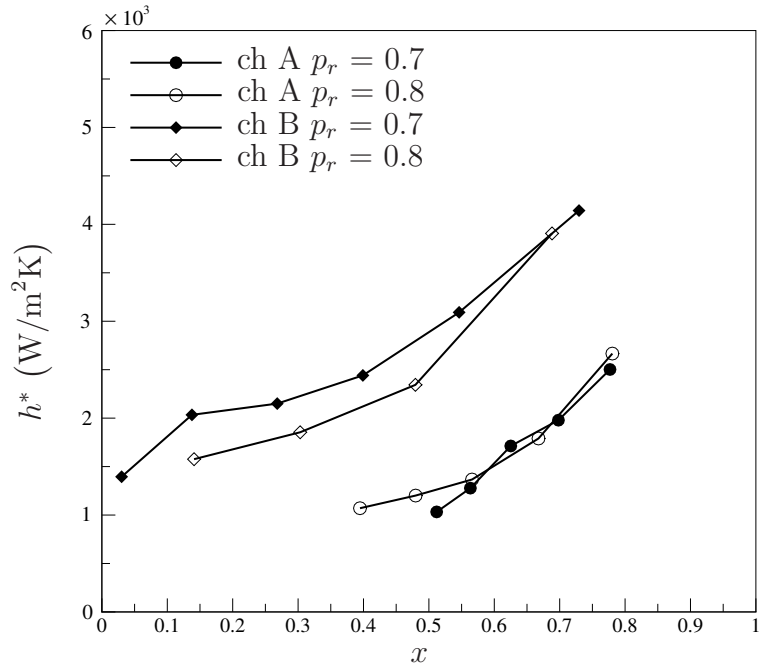


Figure 8: Local condensation heat transfer coefficient h^* as a function of the vapour quality x at two reduced pressures, $p_r = 0.7$ and 0.8 , with $T_a^* = 45^\circ\text{C}$ and $G^* = 400 \text{ kg/m}^2\text{s}$. The range of $T_{sa}^* - T_i^*$ is $2 - 12^\circ\text{C}$.

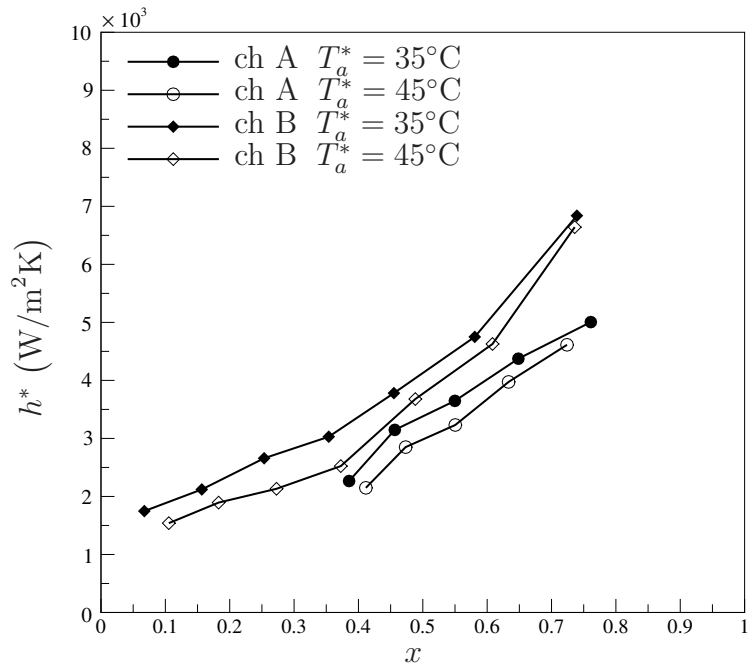


Figure 9: Local condensation heat transfer coefficient h^* as a function of vapour quality x at two ambient air temperatures $T_a^* = 35^\circ\text{C}$ and $T_a^* = 45^\circ\text{C}$ with $p_r = 0.7$ and $G^* = 700 \text{ kg/m}^2\text{s}$.

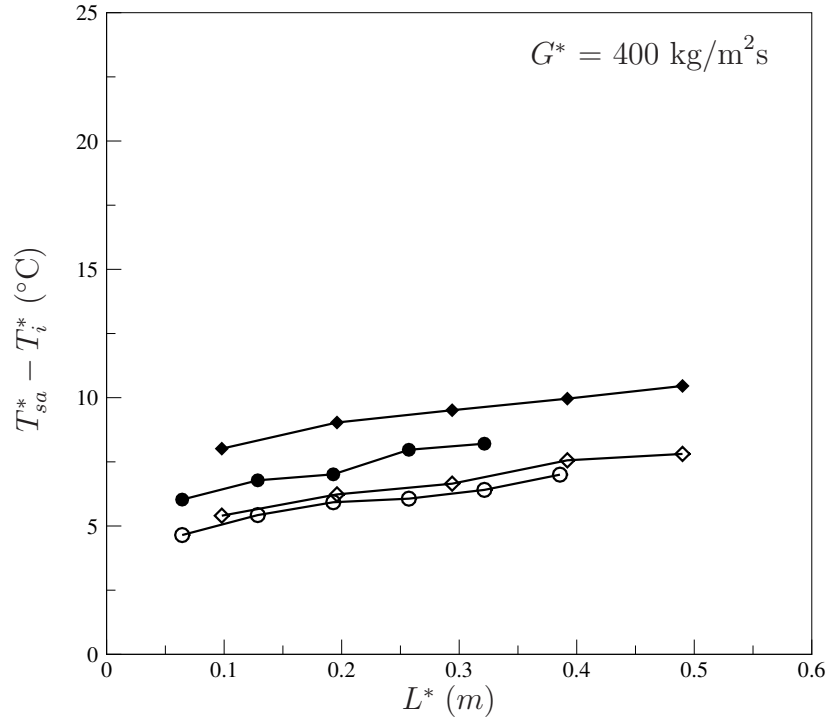
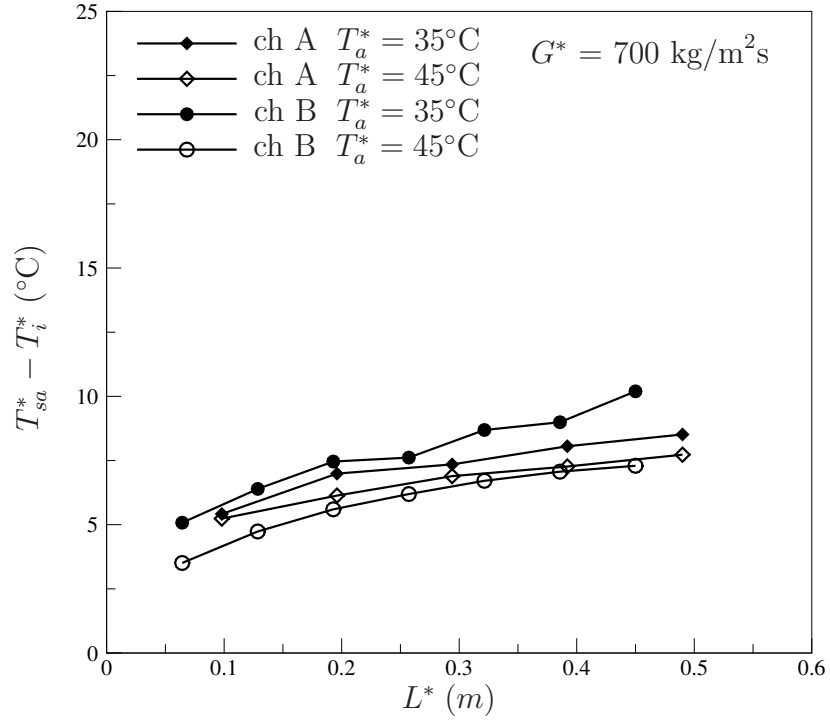


Figure 10: Temperature difference between the inner wall and the saturated liquid as a function of the channel thermal length at two ambient air temperatures $T_a^* = 35^{\circ}\text{C}$ and $T_a^* = 45^{\circ}\text{C}$ with $p_r = 0.7$.

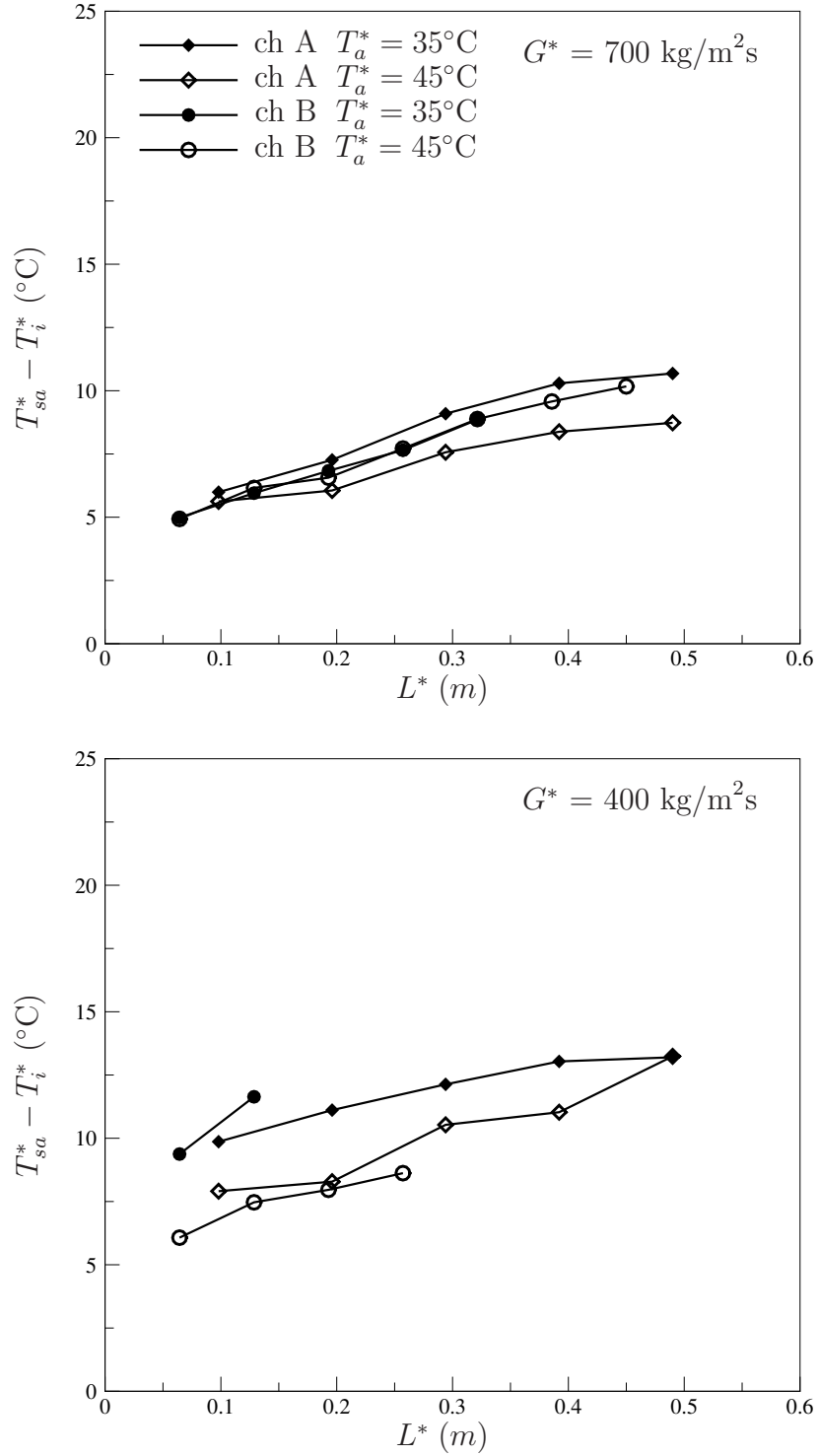


Figure 11: Temperature difference between the inner wall and the saturated liquid as a function of the channel thermal length at two ambient air temperatures $T_a^* = 35^{\circ}\text{C}$ and $T_a^* = 45^{\circ}\text{C}$ with $p_r = 0.8$.

Authors	Correlations
Garimella et al. (2016)	$Nu_{pred} = 0.0133 Re_l^{4/5} Pr_l^{1/3} \left[1 + \left(\frac{x}{1-x} \right)^{0.80} \left(\frac{\rho_l^*}{\rho_g^*} \right)^{0.88} \right],$ <p>where $Re_l = G^*(1-x)D_h^*/\mu_l^*$, $Pr_l = \mu_l^* c_{pl}^*/k_l^*$, G^* is the mass flow rate per unit cross-sectional area, c_{pl}^* is the specific heat of the liquid-phase, ρ_l^* and ρ_g^* are the liquid and vapour densities, respectively, and μ_l^* and μ_g^* are the liquid and vapour viscosities, respectively.</p>
Shah (2016)	$Nu_{pred} = Nu_{lo} \left[1 + 1.128x^{0.817} \left(\frac{\rho_l^*}{\rho_g^*} \right)^{0.3685} \left(\frac{\mu_l^*}{\mu_g^*} \right)^{0.2363} \left(1 - \frac{\mu_g^*}{\mu_l^*} \right)^{2.144} Pr^{-0.1} \right]$ $Nu_{lo} = 0.023 \left(\frac{G^* D_h^*}{\mu_l^*} \right)^{0.8} Pr_l^{0.4}$
Koyama et al. (2003)	$Nu_{pred} = 0.0152 (1 + 0.6Pr_l^{0.8}) \frac{\phi_g Re_l^{0.77}}{X_{tt}}$ $\phi_g^2 = 1 + 21[1 - \exp(-0.319D_h^*)]X_{tt} + X_{tt}^2$ $X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g^*}{\rho_l^*} \right)^{0.5} \left(\frac{\mu_l^*}{\mu_g^*} \right)^{0.1}$
Jige et al. (2016)	$Nu_{pred} = (Nu_{An,F}^3 + Nu_{An,S}^3)^{1/3}$ $Nu_{An,F} = \frac{\phi_g}{1-x} \sqrt{f_g \frac{\rho_l^*}{\rho_g^*}} Re_l^{0.5} (0.6 + 0.06Re_l^{0.4} Pr_l^{0.3})$ $Nu_{An,S} = 0.51 \left[\frac{\rho_l^* h_{fg}^* \sigma^* D_h^*}{\mu_l^* k_l^* (T_{sa}^* - T_i^*)} \right]^{0.25}$ $\phi_g = \sqrt{x^{1.8} + (1-x)^{1.8} \frac{\rho_g^* f_l}{\rho_l^* f_g} + 0.65x^{0.68}(1-x)^{1.43} \left(\frac{\mu_l^*}{\mu_g^*} \right)^{1.25} \left(\frac{\rho_g^*}{\rho_l^*} \right)^{0.75}}$ $f_g = \begin{cases} C_1/(G^* D_h^*/\mu_g^*), & \text{for } G^* D_h^*/\mu_g^* \leq 1500 \\ 0.046/(G^* D_h^*/\mu_g^*)^{0.2}, & \text{for } G^* D_h^*/\mu_g^* > 1500 \end{cases}$ $f_l = \begin{cases} C_1/(G^* D_h^*/\mu_l^*), & \text{for } G^* D_h^*/\mu_l^* \leq 1500 \\ 0.046/(G^* D_h^*/\mu_l^*)^{0.2}, & \text{for } G^* D_h^*/\mu_l^* > 1500 \end{cases}$ $C_1 = 24 (1 - 1.355\mathcal{A} + 1.947\mathcal{A}^2 - 1.701\mathcal{A}^3 + 0.956\mathcal{A}^4 - 0.254\mathcal{A}^5),$ <p>where \mathcal{A} is the aspect ratio of the microchannels.</p>

Table 4: Correlations of Nusselt numbers.

3.2 Comparison with existing heat transfer correlations

Various approaches for predicting the heat transfer coefficients during condensation have been presented in the literature. Three main categories can be identified: the two-phase multiplier approach, the boundary layer approach, and the shear force approach. The present experimental data were compared with the predictions obtained by different correlations, i.e., those by Garimella et al. (2016) and Shah (2016) based on the multiplier approach, the one by Koyama et al. (2003) based on the boundary layer approach, and the one by Jige et al. (2016) based on the shear force approach. Most of the correlations overpredict our heat transfer coefficients because either the channel diameters used for those correlations were larger and circular or they do not account for flow phenomena that are specific to microchannels in our conditions of near-critical pressure and high coolant air temperature. Table 4 summarizes the correlations used to predict our experimental data.

Garimella et al. (2016) utilized a correlation similar to the one proposed by Cavallini and Zecchin (1974), but the regression analysis was carried out using their own experimental data. Garimella et al. (2016)'s correlation predicts our experimental data with $E = 50\%$ for channels A and B, as shown in Fig. 12. Although Garimella et al. (2016) also used R-410A, with nearly the same reduced pressure, their predicted values correlate poorly with our measured data. This is arguably because their range of diameters was much larger than ours, their tube had a round cross section while our channels were rectangular, and they used water at low temperature as coolant, while our cooling fluid was air at high temperature. In addition, their correlation does not account for the effect of reduced pressure. The comparison with Shah (2016)'s correlation, shown in Fig. 13, is not satisfactorily as it highly overpredicts our Nusselt number data. Shah (2016) used the correlations of Shah (1979) and Cavallini et al. (2006), but they employed the methodology by Shah (2013) to calculate the heat transfer coefficient.

Figure 14 shows the comparison between our local experimental Nusselt number Nu_{exp} and that predicted by Koyama et al. (2003). Their correlation gives the best overall agreement, especially at high mass flow rates and for the flow through the large channel. For low mass flow rates the agreement is not as satisfactory since their correlation includes the effect of the wavy annular flow that is unlikely to occur in our case, especially in the channel with the small hydraulic diameter. Part of the data is predicted within $E = 30\%$. Figure 15 shows the comparison between our experimental data and the prediction by Jige et al. (2016)'s correlation. Their correlation leads to similar agreement to that given by Koyama et al. (2003)'s correlation, i.e., within $E = 30\%$ for most of our Nusselt number data. Koyama et al. (2003) and Jige et al. (2016) used the methodology by Haraguchi et al. (1994), which combines the effects of the annular regime and gravity.

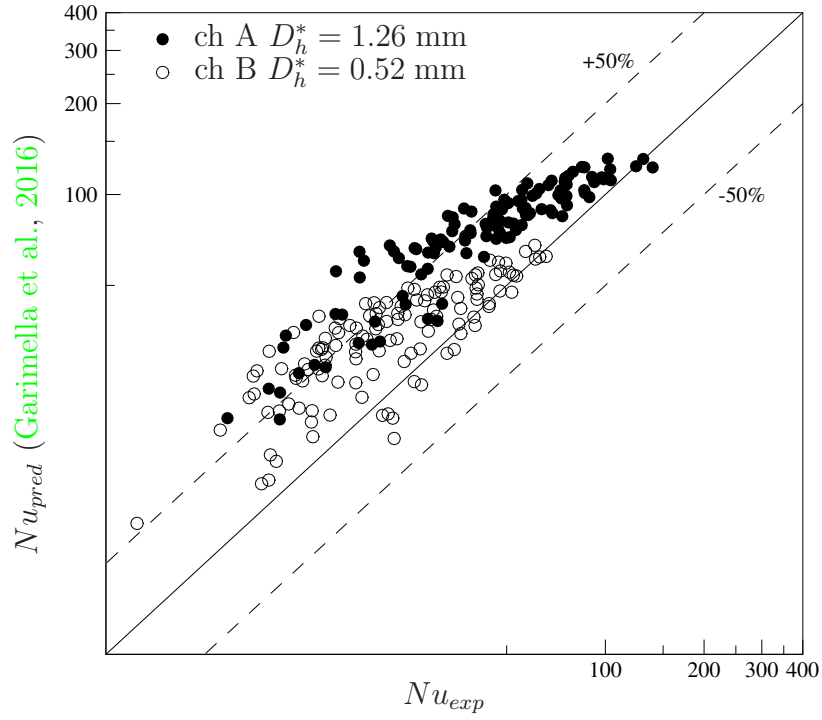


Figure 12: Comparison between the experimentally measured Nusselt number Nu_{exp} and the Nusselt number Nu_{pred} predicted by [Garimella et al. \(2016\)](#)'s correlation.

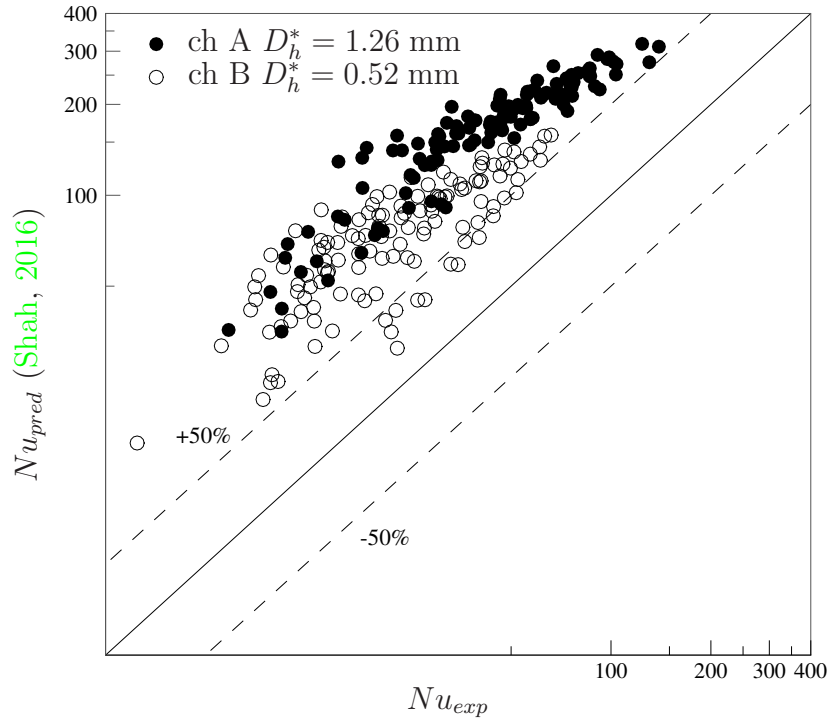


Figure 13: Comparison between the experimentally measured Nusselt number Nu_{exp} and the Nusselt number Nu_{pred} predicted by [Shah \(2016\)](#)'s correlation.

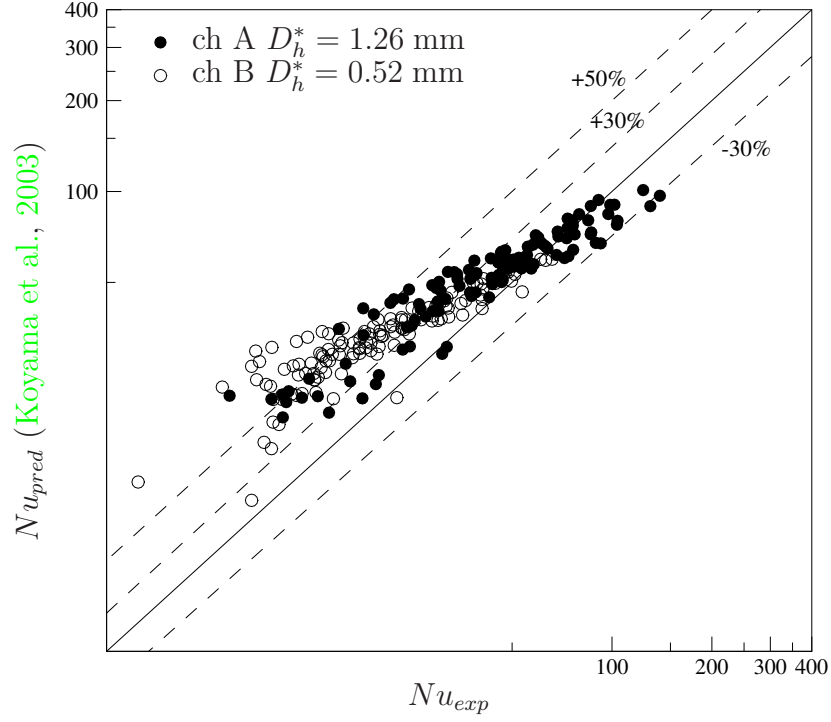


Figure 14: Comparison between the experimentally measured Nusselt number Nu_{exp} and the Nusselt number Nu_{pred} predicted by Koyama et al. (2003)'s correlation.

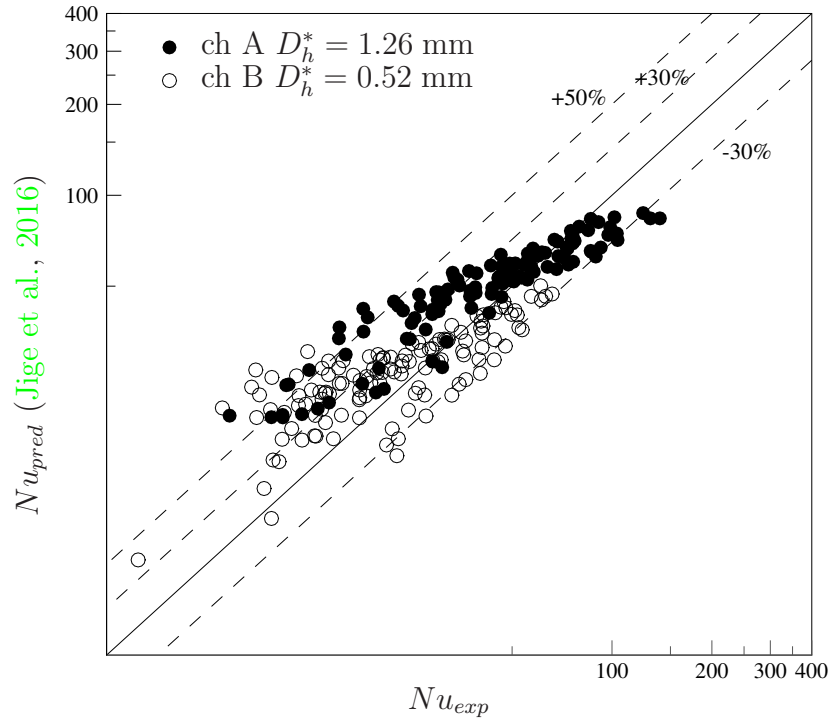


Figure 15: Comparison between the experimentally measured Nusselt number Nu_{exp} and the Nusselt number Nu_{pred} predicted by Jige et al. (2016)'s correlation.

3.3 Novel correlation of the condensation heat transfer coefficient

Most of our data are within the range of dimensionless superficial velocity

$J_G = G^* x [D_h^* g^* \rho_g^* (\rho_l^* - \rho_g^*)]^{-1/2} \geq 2.5$, where g^* is the gravitational acceleration, and therefore, as shown by Cavallini et al. (2002), the annular regime dominated in both channels. For small hydraulic diameters the annular flow is likely to occur because of the significant shear forces. This is because the surface tension dominates over gravity when the diameter is small, as discussed by Nema et al. (2014), who identified the transition criterion to distinguish between the effects of surface tension and gravity forces in microchannel flows. Furthermore, the non-circular cross section helps perpetuate the annular flow regime even at low mass flow rates and for a wide range of vapour qualities. Indeed, the non-circular geometry causes a pressure reduction at the corners due to the curvature of the liquid-vapour-interface, which tends to drive more liquid to the corners and maintain the annular flow regime. This is known as the Gregorig effect (Gregorig, 1962). Also, when the condensation process occurs near the critical pressure, the properties of the liquid and vapour become similar which means that the annular flow occurs for a wide range of vapour quality and mass flow rates.

Only the data that satisfied the annular flow inequality proposed by Cavallini et al. (2002) were considered to obtain an empirical correlation of the Nusselt number as a function of independent dimensionless parameters. The Nusselt number is written as:

$$Nu_{pred} = f(Re_l, Pr_l, p_r, X_{tt}) = A Re_l^n Pr_l^{n_1} p_r^{n_2} X_{tt}^{n_3}, \quad (12)$$

where A , n , n_1 , n_2 , and n_3 are constants found through a non-linear regression analysis. The empirical correlation is:

$$Nu_{pred} = 0.018 Re_l^{0.94} Pr_l^{-0.22} p_r^{1.4} X_{tt}^{-1.04}. \quad (13)$$

Figure 16 shows the comparison between our experimental data and the prediction obtained with our correlation (13). The present correlation successfully predicts the experimental data with $E = 25\%$, more accurately than any other correlation available in the literature.

4 Conclusions

We have experimentally studied the local heat transfer coefficient of the refrigerant R-410A during condensation inside horizontal multiport aluminium microchannels of rectangular cross section with hydraulic diameters $D_h^* = 0.52$ mm and $D_h^* = 1.26$ mm at two reduced pressures, $p_r = 0.7$ and $p_r = 0.8$. The condensation process was accomplished using air as the coolant at two temperatures, $T_a^* = 35^\circ\text{C}$ and $T_a^* = 45^\circ\text{C}$. The experimental technique adopted in this study allowed the direct measurement of the two key parameters needed for calculating the local heat transfer coefficient, i.e., the local heat flux during condensation along the channel and the temperature difference.

The condensation heat transfer coefficient increases with an increase of refrigerant mass flow rate per unit area and vapour quality and it is larger for the flow in the smaller hydraulic diameter. At high mass flow rates per unit area and vapour qualities

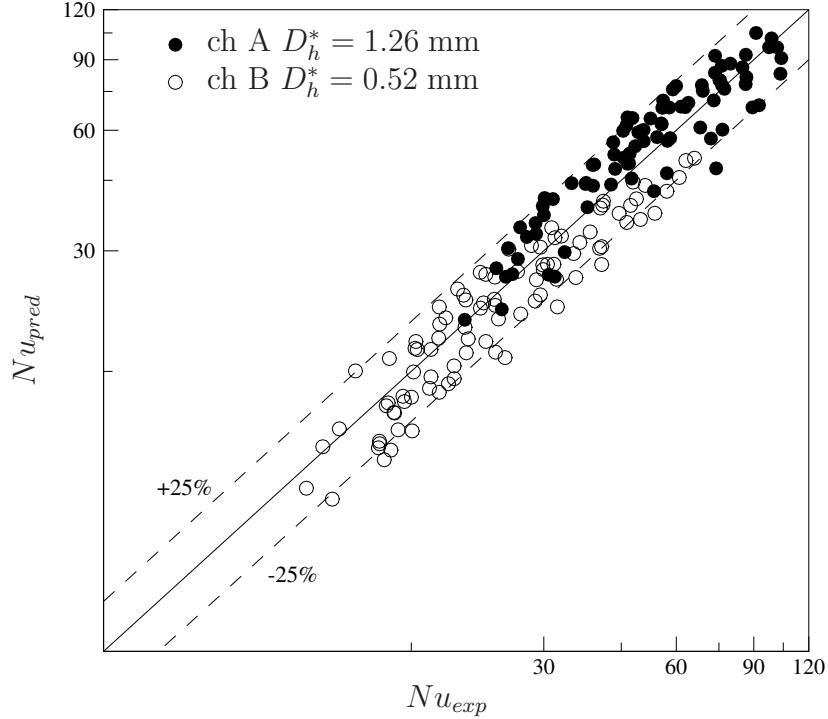


Figure 16: Comparison between the experimentally measured Nusselt number Nu_{exp} and the Nusselt number Nu_{pred} predicted by correlation (13).

411 for a low reduced pressure, the heat transfer coefficient is slightly larger for a lower
 412 ambient air temperature. The prediction of the experimental data through correlations
 413 available in the literature was found to be unsatisfactory mainly because of our extreme
 414 conditions of operation, i.e., high ambient temperature and near-critical pressure. Only
 415 the predictions by the correlations of [Koyama et al. \(2003\)](#) and [Jige et al. \(2016\)](#) offered
 416 a modest agreement with our experimental data, but this was limited to high values of
 417 the vapour quality for the flow in the larger channel. A new empirical correlation for
 418 the local heat transfer coefficient for the annular flow regime was obtained to predict our
 419 experimental data successfully.

420 The next research step is to extend the present analysis to investigate the heat trans-
 421 fer for the refrigerant R-410A at critical conditions. In the facility used in our study,
 422 the critical conditions would be achieved at a pressure of approximately 50 bar. The
 423 comparison between the near-critical and critical heat transfer coefficient would be ex-
 424 tremely beneficial for a comprehensive understanding of condensation heat transfer in
 425 microchannels. Further interesting aspects to be explored are the effect of free-stream
 426 turbulence of the oncoming cooling stream on the microchannel heat transfer and the
 427 flow visualization at high operating pressure during the condensation process. The latter
 428 represents a challenge because of the difficulty in using a transparent material suitable
 429 for the flow visualization that is also able to sustain near-critical or even critical pressure.

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432 is highly acknowledged.

433

434 **Nomenclature**

435

436	A_s^*	outer surface area of the channel, mm ²
437	c_p^*	specific heat, J/kg K
438	D_h^*	microchannel hydraulic diameter, mm
439	e	energy balance index
440	G^*	mass flow per unit cross-sectional area, kg/m ² s
441	g^*	gravitational acceleration, m/s ²
442	h^*	local heat transfer coefficient, W/m ² K
443	H_{ch}^*	internal height of each microchannel, mm
444	h_l^*	saturated liquid enthalpy at the evaporator pressure, J/kg
445	h_e^*	enthalpy, J/kg
446	h_{fg}^*	latent heat, J/kg
447	J_G	scaled superficial velocity
448	k^*	thermal conductivity, W/m K
449	L^*	length of the multiport microchannel, mm
450	\dot{m}^*	mass flow rate, kg/s
451	N	number of microchannels
452	Nu	Nusselt number
453	P^*	wetted perimeter of each microchannel
454	Pr	Prandtl number
455	p^*	pressure, Pa
456	p_r	reduced pressure, $p^*/p_{critical}^*$
457	\dot{Q}^*	heat transfer, W

458	q''^*	local external heat flux measured by the sensors, W/m^2
459	Re	Reynolds number based on D_h^* and bulk velocity
460	T^*	temperature, $^\circ\text{C}$
461	t^*	channel thickness, mm
462	W^*	width of multiport channel, mm
463	W_{ch}^*	internal width of each microchannel, mm
464	W_s^*	width of wall separating adjacent microchannels, mm
465	X	Lockhart-Martinelli parameter
466	x	vapour quality

467 **Greek symbols**

468	ΔL^*	thermal length of each interval along the channel, mm
469	Δx	vapour quality difference
470	μ^*	dynamic viscosity, $\text{kg}/\text{m s}$
471	ρ^*	density, kg/m^3
472	η	fin efficiency
473	ϕ	two-phase pressure drop multiplier

474 **Subscripts**

475	a	air
476	ch	channel
477	e	evaporator
478	exp	experimental
479	g	gas phase
480	l	liquid phase
481	lo	liquid only
482	$loss$	heat loss
483	w	wall
484	i	in/inner
485	o	out/outer

486 *pred* predicted
487 *r* refrigerant
488 *sa* saturation
489 *tt* turbulent liquid-turbulent vapour

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