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Heat transfer coefficient correlations of water subatmospheric vaporization in a channel of a smooth plate heat exchanger, based on Vaschy-Buckingham theorem

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Abstract:

Water as an environmentally friendly, safe and cheap natural refrigerant (R-718) requires vaporization conditions at sub-atmospheric pressures (~1 kPa). At these pressure ranges, studies on the specific vaporization of water are scarce and insufficient. For this reason, the engineering of heat exchangers using R-718 as working fluid remains empirical. The aim of this paper is thus to provide correlations to estimate water pool boiling and falling film evaporation heat transfer coefficient under subatmospheric pressure. Correlations developed are believed to be more widely used by providing dimensionless numbers consistent with phase-change phenomena occurring at sub and at-mospheric pressure. To develop these correlations, heat transfer phenomena during water vaporization at pressures ranging from 0.7 to 1.7 kPa in a scale-lab size (0.2 m wide x 0.5 m high) vertical smooth plate evaporator channel are studied. Filling ratio vary from 1/2 to 1/10 of the total channel height, canal thicknesses are set to 2 mm, 4 mm or 6 mm. Thus, raw experimental data from 139 tests performed under various operating conditions of a thermosyphon loop mimicking a sorption system are exploited. Each test consists of more than 1200 values. A graphical and statistical comparison between the heat transfer coefficient (HTC) obtained from the experimental database and the HTC estimated by empirical correlations from the literature is made, calculating the percentage of predicted data within an error band of $\pm 30\%$ (τ_{30}), the mean absolute error (MAE), the mean relative error (MRE) and the correlation coefficient (r). Two methodical correlations for predicting the HTC of pool boiling and liquid film evaporation, are developed. They predict 86% (with MAE = 19%, MRE = 3%, $r = 0.79$) and 83% (with MAE = 18%, MRE = 2%, $r = 0.84$) of the experimental data in τ_{30} respectively. To extend their validity domains, the proposed correlations are evaluated using independent experimental data from the literature. Giving satisfactory results, it constitutes a first step towards the development of tools used for the sizing and the optimization of the design of two-phase heat exchangers.

Keywords: water vaporization, subatmospheric pressure, heat transfer coefficient, correlation, plate-type heat exchanger.

NOMENCLATURE

Latin letters

A	cross section [m ²]
c _p	thermal mass capacity [J. kg ⁻¹ . K ⁻¹]
d _b	bubble diameter [m]
D _h	hydraulic diameter [m]
e	thickness [m]
g	gravity acceleration [m. s ⁻²]
G	mass velocity [kg. m ⁻² . s ⁻¹]
h	heat transfer coefficient [W. m ⁻² . K ⁻¹]
H _l	height of the liquid column [m]
L _c	capillary length = $\sqrt{\sigma/[(\rho_l - \rho_v)g]}$ [m]
P	pressure [Pa]
P _r	pressure ratio [-]
r	correlation coefficient [-]
Q̇	heat flux [W]
q̇	heat flux density [W. m ⁻²]
q _r	heat flux ratio
S	exchange area [m ²]
T	temperature [K]
MAE	Mean Absolute Error [%]
MRE	Mean Relative Error [%]

Greek Letters

Δh _{lv}	enthalpy of liquid-vapour change of state [J. kg ⁻¹]
ΔT	temperature difference [K]
τ ₃₀	percentage of predicted data within ±30% [%]
λ	thermal conductivity [W. m ⁻¹ . K ⁻¹]
μ	dynamic viscosity [N. m ⁻² . s]
σ	surface tension [N. m ⁻¹]
δ	characteristic length [m]
ρ	density [kg. m ⁻³]
φ	instrumental error (-)

subscripts

acq	relative to the acquisition system
b	boiling
cool	cooling
crit	critical
exp	experimental
f	film
fs	secondary fluid
in	inlet
instr	relative to the instrument
ir	intermittent region
l	liquid
out	outlet
p	plate
sat	saturation
pbr	pooling boiling region
pred	predicted

v	vapor
vr	vapor region
w	wall
z	zone

Dimensionless numbers

Bo	Bond Number [–]
Ca	Capillary number [–]
Co	confinement number [–]
F*	Form factor [–]
μ^*	dynamic viscosities ratio [–]
Nu	Nusselt number [–]
Pr	Prandtl number [–]
ρ^*	density ratio [–]
ρ_v^*	vapor density ratio [–]
We	Weber number [–]

43

44 1. Introduction

45 In a context where regulations on synthetic refrigerants are becoming more and more
 46 restrictive due to their negative impact on the environment, it is urgent to think about the
 47 development of compact, efficient and “clean” refrigeration technologies. Thus, through the
 48 improvement of components such as the evaporator or the compressor, systems using natural
 49 refrigerants like water (R-718), represent a promising alternative. R-718 is an efficient fluid
 50 (high vaporization enthalpy and heat capacity), available, non-toxic, non-flammable,
 51 inexpensive, inert for the environment (ODP: Ozone Depletion Potential and GWP: Global
 52 Warning Potential are zero) and easy to handle (minimizes safety precautions) [1] [2]. Although
 53 the use of water as working fluid is widespread in sorption systems and is gaining ground in
 54 mechanical vapor compression systems equipped with turbo compressors [3], systems using
 55 water as refrigerant still shows poor performances partly due to the poor performance of the
 56 heat exchanger [4]. These poor performances could be mostly explained by the fact that using
 57 R-718 as refrigerant requires vaporization conditions at sub-atmospheric pressures (~1 kPa).

58 Indeed, the evaporator is the essential component of thermal machines since the maximum
 59 permissible heat exchange rate depends on its pressure conditions (thermophysical properties
 60 of the refrigerant) and on the topography of its heat exchange surfaces. However, at
 61 subatmospheric pressure ranges, the specificity of the thermophysical properties of R-718 and
 62 in particular its low vapour density ($\rho_v = 0.009 \text{ kg. m}^{-3}$ and $\rho_l = 999.68 \text{ kg. m}^{-3}$ at 1.2 kPa
 63 versus $\rho_v = 0.598 \text{ kg. m}^{-3}$ and $\rho_l = 958.34 \text{ kg. m}^{-3}$ at 101,325 kPa), lead to atypical
 64 boiling behaviours such as the appearance of bubbles of centimetric size often referenced in the
 65 literature [5] [6]. For example, at 1.2 kPa, the typical working pressure of chillers using R-718,
 66 mushroom-shaped bubbles of 15 cm in diameter can be observed under pool boiling conditions,
 67 in contrast to the spherical or semi-spherical bubbles of a few millimeters that can be observed
 68 at atmospheric pressure [4]. Due to this atypical behavior, correlations and design usually used
 69 to realize heat exchangers of thermal systems do not allow a proper sizing of R-718 evaporators.
 70 The needs to improve the efficiency of said machines using R-718, makes it essential to size
 71 correctly, optimize and reduce the size of heat exchangers, especially the evaporator.

72 In the evaporator, the refrigerant exchange heat from a source, through a process of
 73 vaporization (boiling or evaporation). This exchanger can be made of tubes and shells, fins or
 74 plates. Among the latter, smooth plate heat exchangers can be good candidates. The use of

75 vertical flat plate and falling film heat exchangers is especially attractive for sorption machines
76 and potentially increases their compactness, reduces their weight and price compared to
77 conventional vertical or horizontal tube configurations [4] [7]. The interest of these types of
78 exchangers is also due to their unique advantages such as flexible thermal design (plates can be
79 simply added or removed to meet different thermal services or process requirements), low
80 manufacturing cost, ease of inspection and cleaning to maintain strict hygienic conditions,
81 reduced load, better heat transfer performance due to fluid distribution over the plates,
82 compactness, etc. [8] [9] [10]. Plate heat exchangers are also suitable for standardization of the
83 manufacturing process [5]. Also, their deep channels, large ports and possible laser welding
84 allow for vacuum vaporization [18]. This is consistent with the fact that the use R-718, with its
85 high boiling temperature at atmospheric pressure ($T = 100\text{ }^{\circ}\text{C}$, $P = 101.325\text{ kPa}$), requires to
86 operate at sub-atmospheric pressures close to its triple point ($T = 0.01\text{ }^{\circ}\text{C}$, $P = 0.61\text{ kPa}$) [4]
87 [11] in the part containing the evaporator. But, at these low pressures ranges, in addition to the
88 observation of centimeter size mushroom bubbles, since the pressure at the free surface (P_{sat})
89 is small, the pressure induced by the liquid column ($P_{\text{static}} = \rho_l g H_1$) can be of the same order
90 of magnitude as the system pressure [5] [12]. Several experimental studies ([4] [13] [12] [14]
91 [15]) confirmed the non-negligible influence of the liquid height (hydrostatic pressure) on the
92 heat transfers during low-pressure pool boiling. The filling level (H_1) can therefore no longer
93 be neglected in the analysis of heat transfer phenomena, as is commonly practiced for higher
94 pressures. This is all the more important since the presence of this hydrostatic pressure causes
95 the vapor bubbles to form in an inhomogeneous medium in pressure or subcooling [4]. Thus,
96 the thermophysical properties of the refrigerant present strong variations [6], leading to
97 phenomena of a certain complexity regarding thermal couplings and two-phase flows.

98 As a first approach towards the understanding of these complex phenomena and in order to
99 design compact evaporators for sorption coolers and optimize their size, Giraud et al. [5] [16]
100 [17] conducted experimental studies on the vaporization of water at subatmospheric pressure in
101 a vertical smooth plate evaporator (VSPE) (scale-lab size: 0.2 m wide x 0.5 m high). Filling
102 ratio (H_1) vary from 1/2 to 1/10 of the total channel height, the thicknesses (e) between two
103 succeeding plates are set to 2 mm, 4 mm or 6 mm. The authors [5] mainly highlighted three
104 different working zones: a boiling zone, a liquid film evaporation zone and a vapor zone. Their
105 results are in perfect agreement with the observations of Chang et al. [13]. The liquid film
106 evaporation zone is due to the free surface moving due to boiling as well as the spread of the
107 encapsulated liquid contain between the bubbles and the free surface. This liquid is spread over
108 the top wall plate and forms the liquid film that evaporates immediately [5] [13]. The liquid
109 film evaporation zone is the seat of more than 70% of the heat transfer in the VSPE [17].
110 Studying the effect of the channel thicknesses on the VSPE performance, the authors also

111 observed, for a confinement number $Co = \frac{1}{D_h} \cdot \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} = 0.69$ ($e = 2\text{ mm}$), phenomena
112 such as droplet entrainment by the steam flow [16]. They confirmed experimentally the impact
113 of this observation on the heat transfer: higher performances are observed for a confinement
114 number $Co = 0.69$ compared to a confinement number $Co = 0.35$ ($e = 4\text{ mm}$) and $Co =$
115 0.23 ($e = 6\text{ mm}$) when droplets entrainment is observed, i.e. for high rate of steam production
116 obtained at high entrainment pressures and low filling rate. Then, as a first tool toward the
117 design of compact subatmospheric VSPE, Giraud et al [17] developed dimensionless numbers
118 depending on the measured cooling capacity, the thermophysical properties of the fluid and the
119 operating conditions. They showed that, using these dimensionless number, similar evaporator
120 behavior could be observed for different boiling phenomena kinetics since they show that, based
121 on these numbers, the evaporator behavior is similar when the evaporator is included in a
122 thermosyphon loop than when it is included in an ad-sorption system.

123 Despite the efforts made, efforts still should be made to develop more universal correlations
124 involving in particular a form factor and exchange coefficients. However, the knowledge on
125 the vaporization of water at subatmospheric pressure remains scarce and insufficient and, given
126 the great difference between the characteristics of subatmospheric vaporization and those
127 observed at atmospheric pressure [4] [6], the dimensionless numbers usually used at high
128 pressure are not sufficient to properly predict phenomena observed at subatmospheric pressure.
129 Empirical correlations of the literature developed, verified or adapted to low pressure
130 conditions, are then limited mainly because they do not consider all the independent physical
131 quantities that govern the confined vaporization at subatmospheric pressure such as the form
132 factor (hydrostatic pressure), the inhomogeneous pressure condition of the medium (variation
133 of the thermophysical properties of the refrigerant), the boundary conditions, etc. Moreover,
134 the dimensional analysis adopted for the elaboration of the said correlations, in general, does
135 not verify the independence of the physical quantities despite the necessity to ensure that no
136 physical quantity, apart from the variable of interest, depends on the variation of the other
137 quantities [18]. This independence is a key factor for dimensional analysis modeling since it
138 allows an easily transferred results from one scale to another (similarity).

139 Referring to the observations of literature, the understanding of heat transfer phenomena
140 during pool boiling and liquid film evaporation at sub-atmospheric pressure thus seems
141 fundamental for optimizing design and performances of compact heat plates exchangers. For
142 this purpose, heat exchanger designers inevitably have a great need for methodical HTC
143 correlations based on rigorous dimensional analysis and offering higher accuracy than existing
144 correlation. This is what the paper attend to do: offering correlations with dimensionless
145 numbers requires for the proper design of subatmospheric water heat exchanger. It is believed,
146 that although the obtention of the coefficient may remain empirical, the used of the chosen
147 dimensionless numbers lead to universal correlations since it includes the independent physical
148 properties influencing phase change phenomena at low and high pressure.

149 The approach starts with a presentation of the HTC correlations available in the open
150 literature. Subsequently, the raw experimental database and the experimental set up used by
151 Giraud et al. [16] to determine the heat transfer coefficients during pool boiling and liquid film
152 evaporation of water at sub-atmospheric pressures is described. The constant and the exponents
153 of the general or Dittus-Boelter type and other relevant correlations from the literature [19] is
154 fitted to the database obtained from the experimental results of Giraud et al. [16] for graphical
155 and statistical comparison between the experimental HTC and the HTC estimated. Then, based
156 on the scientific knowledge generated by previous studies, new methodical correlations for
157 predicting HTC during pool boiling and evaporation of the liquid film falling from water in a
158 VSPE is developed. These correlations are based on a rigorous dimensional analysis approach,
159 following Vaschy-Buckingham theorem described by Delaplace et al. [18]. The news proposed
160 correlations consider, among others, the form factor (canal thickness and filling ratio), the
161 thermophysical properties of refrigerant (hydrostatic pressure and saturation pressure), the heat
162 flux density and the parietal superheating. The obtained results are presented and discussed.
163 Then, in order to extend the range of validity and test the robustness of the developed
164 correlations, these models are fitted and evaluated using independent experimental data from
165 the literature.

166

167 **2. HTC correlations of literature**

168 In this section, different relevant correlations commonly used to predict HTC during water
169 vaporization are presented and discussed. There are many correlations in the literature for

170 estimating the HTC during water vaporization, but many of them are valid only for typical heat
 171 transfer surface topography and/or operating conditions (high pressure). The main parameters
 172 affecting nucleate boiling are heat flux, saturation pressure, thermophysical properties of the
 173 working fluid and characteristics of the boiling surface and material etc. [20]. Thus, Dittus-
 174 Boelter [19], Labuntsov [11], Kruzylin [11], Stephan-Abdelsalam [11], Chang (pool boiling
 175 region and intermittent region) [13], Mc Nelly [21] and Yu [20] correlations consist of
 176 parameters that influence water vaporization.

177 The general or Dittus-Boelter type correlation (Eq. 1) [19] which relates the Nusselt number
 178 (Nu) to the Reynolds (Re) and Prandtl (Pr) numbers, is frequently used to estimate the HTC in
 179 heat exchangers. Several authors use this ready-made correlation and adjust the coefficients (C)
 180 and exponents (a, b) to be close to their experimental results.

181

$$\frac{h \delta}{\lambda_l} = C \cdot \left(\frac{\rho_l U \delta}{\mu_l} \right)^a \cdot \left(\frac{\mu_l c_p}{\lambda_l} \right)^b \quad (1)$$

182

183 As a first approach toward a generalization of correlations, efforts are made by [13] [11]
 184 [21] and [20] in developing empirical models of HTC or testing existing correlations in pool
 185 boiling at low pressure conditions. However, from these studies, only those in [13] could be
 186 apply to heat exchange in the context of a plate evaporator.

187 Zajaczkowski et al. [11] conducted experimental studies on the influence of subatmospheric
 188 pressure and heat flux density on water nucleate boiling. Experiments were conducted above a
 189 copper plate in an insulated glass cylinder. Pressure ranges between 1 to 10 kPa, and heat fluxes
 190 between 10 and 45 kW/m². As a main result, they showed that accurate approximations of the
 191 experimental heat transfer coefficient are obtained using the Labuntsov correlation (Eq. 2)
 192 (Mean Absolute Deviation (MAD 0.12-0.89), the Kruzylin correlation (Eq.3) (MAD 0.25-0.35)
 193 and the Stephan-Abdelsalam correlation (Eq. 4) (MAD 0.16). Thus, Labuntsov correlation (Eq.
 194 2) and Kruzylin correlation (Eq. 3) directly determines the heat transfer coefficient as a function
 195 of some thermophysical properties of the fluid, the saturation temperature and the heat flux.
 196 Stephan-Abdelsalam correlates (Eq. 4) the Nusselt number as a function of dimensionless
 197 numbers that take into account the heat flux and several thermophysical properties of the fluid.

198

$$h = 0.075 \cdot \dot{q}^{0.67} \cdot \left(1 + 10 \cdot \left(\frac{\rho_v}{\rho_l - \rho_v} \right)^{0.67} \right) \cdot \left(\frac{k_l^2}{v \cdot \sigma \cdot T_{sat}} \right)^{0.33} \quad (2)$$

199

$$h = 0.0777 \cdot \left(\frac{\rho_v \Delta h_{lv}}{\rho_l - \rho_v} \right)^{0.33} \cdot \left(\frac{\rho_l}{\sigma} \right)^{0.33} \cdot \left(\frac{k_l^{0.75} \cdot \dot{q}^{0.7}}{\mu_l^{0.45} \cdot c_{p,l}^{0.12} \cdot T_{sat}^{0.37}} \right) \quad (3)$$

200

$$h = 0.23 \cdot \frac{k_l}{d_b} \cdot \left(\frac{\dot{q} d_b}{k_l T_{sat}} \right)^{0.674} \cdot \left(\frac{\rho_v}{\rho_l} \right)^{0.297} \cdot \left(\frac{\Delta h_{lv} d_b^2}{\left(\frac{k_l}{\rho_l c_{p,l}} \right)^2} \right)^{0.371} \cdot \left(\frac{\left(\frac{k_l}{\rho_l c_{p,l}} \right)^2 \rho_l}{\sigma d_b} \right)^{0.35} \cdot \left(\frac{\rho_l - \rho_v}{\rho_l} \right)^{-1.73} \quad (4)$$

201

202 Stephan-Abdelsalam correlation (Eq. 4) is one of the most used correlations in the
203 literature. However, in this equation, bubble diameter (d_b) is sometimes very difficult to predict.
204 The authors recommend using the Fritz correlation (Eq. 5) for prediction of the bubble diameter
205 as the following [22].

206

$$d_b = 0.0146 \cdot \theta \cdot \left(\frac{2 \sigma}{g(\rho_l - \rho_v)} \right)^{0.5} \quad (5)$$

207

208 In which the contact angle θ is specified equal to 45° for water and 35° for other liquids
209 studied by the author [22]. In their calculations, Zajaczkowski et al. [11] set a contact angle θ
210 equal to 35° .

211 Chang et al. [13] studied the thermal performance and behavior of an evaporator
212 consisting of a pair of parallel stainless steel plates of $138 \times 20.8 \text{ mm}^2$ each with 10 mm
213 channels. The evaporator is inserted in a two-phase thermosiphon loop. The pressure varied
214 between 2.1 kPa and 66.2 kPa. In their experiments, the authors observed three distinct flow
215 regions with different heat transfer properties. They developed a set of three HTC correlations
216 for each region : pool boiling region (Eq. 6), intermittent region (Eq. 7) and vapour region Eq.
217 (8). Correlations for the pool boiling region (Eq. 6) and the intermittent region (Eq. 7) will be
218 used. They allow the HTC of pool boiling and liquid film evaporation to be calculated from the
219 heat flux and reduced pressure.

220

$$h_{pbr} = (1.951 \cdot 10^6 \cdot \dot{q}^{-0.26}) \left(\frac{P_{sat}}{P_{crit}} \right)^{(0.508 - 5.91 \cdot 10^{-6} \cdot q)} \quad (6)$$

221

$$h_{ir} = (2 \cdot 10^6 \cdot \dot{q}^{-0.325}) \cdot \left(\frac{P_{sat}}{P_{crit}} \right)^{(0.449 - 4.72 \cdot 10^{-6} \cdot q)} \quad (7)$$

222

$$h_{vr} = (2.06 \cdot 10^6 \cdot \dot{q}^{-0.412}) \cdot \left(\frac{P_{sat}}{P_{crit}} \right)^{(0.441 - 7.08 \cdot 10^{-6} \cdot q)} \quad (8)$$

223

224 Baki [21] performed a review on experimental boiling data at subatmospheric pressures
225 for three different fluids, among them water, to compare them to four known correlations. His
226 results showed poor accuracy between experimental and predicted data for three of the
227 correlations. Only Mc Nelly correlation (Eq. 9) that determines the HTC from the heat flux,
228 some thermophysical parameters of the fluid and the working pressure, gives satisfactory results
229 (mean error of 40% and correlation coefficient (r) of 0.91).

230

231

$$h = 0.225 \cdot \left(\frac{q c_p}{\Delta h_{lv}} \right)^{0.69} \cdot \left(\frac{P \lambda}{\sigma} \right)^{0.31} \cdot \left(\frac{\rho_l}{\rho_v} - 1 \right)^{0.32} \quad (9)$$

232

233 Yu et al. [20] experimentally estimated water boiling heat transfer coefficient obtained on
 234 a horizontal copper bar surface for a pressure range of 1.8-3.3 kPa and a heat flux range of
 235 4000 – 10000 W · m⁻². Based on their experimental data, the authors obtained, by regression,
 236 the following correlation (Eq.10). The correlation developed is a pressure and heat flux
 237 corrected version of the convective boiling correlation [14].

238

$$h = 0.226 \cdot P_r^{1.497} \cdot q_r^{1.023} \cdot \left(\frac{g \beta \Delta T L_c^3}{\mu_l^2} \right)^{-0.891} \cdot \left(\frac{\mu_l c_{pl}}{\lambda_l} \right)^{0.578} \quad (10)$$

239 3. Test bench

240 Giraud et al. [16] performed an experimental study on a prototype refrigeration system
 241 developed at Lafset to simulate the operating conditions that could occur in machines using
 242 water as refrigerant (Fig.1). The test bench is a closed-loop thermosiphon, assisted by a pump.
 243 It consists mainly of two heat exchangers: an oversized condenser and a smooth plate
 244 evaporator. To form the classic circuit of a refrigerating machine, a liquid supply line and a
 245 steam exhaust line connect these two main components. Type K thermocouples (named T + a
 246 number), pressure sensors (named P + a number) and flowmeters (F1 and F2) are installed in
 247 the refrigerant circuit and in the two auxiliary circuits. The saturation temperatures inside the
 248 evaporator and the condenser are regulated by means of two heating/cooling devices (B1 and
 249 B2). A third heating device (B3) is used to adjust the fluid temperature at the inlet of the
 250 expansion valve to match the temperature of the refrigerant at the outlet of the condenser in a
 251 chiller. The condenser is a cylindrical steel vessel (304L stainless steel) with an inside diameter
 252 of 300 mm and a height of 720 mm insulated with the outside. An oversized copper tube coil
 253 heat exchanger is placed inside the container to condense the steam along the outer surface of
 254 the tube and store the refrigerant in liquid state. The condenser is used here to set driving
 255 pressures and simulate an absorber or compressor that would exist in a real chiller.

256 The entire test bench is made up with ISO-K component to allow high gas tightness. To
 257 limit the introduction of incondensable gas, before filling the condenser with water, the entire
 258 experimental setup is vacuumed during at least 12 hours and the condenser is then filled with
 259 an amount of water that was previously boiled. The condenser and then the entire experimental
 260 setup are regularly expunged of a part of the vapor phase, until the pressure measured is equal
 261 to the water saturation pressure at the measured temperature during at least one week.

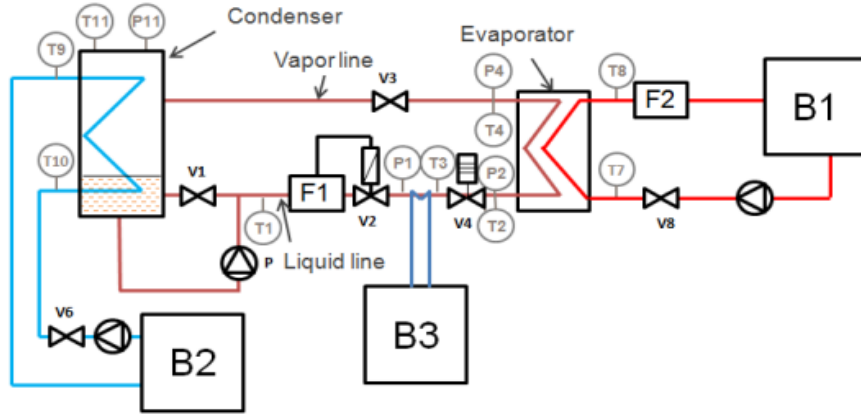


Figure 1 : Drawing of the pump-assisted thermosiphon loop used [16].

262

263 3.1. Evaporator

264 The present study focuses on the vaporization phenomena occurring in the evaporator
 265 (Fig.2a) of the test bench (Fig.1). This heat exchanger consists of three smooth plates 500 mm
 266 high and 200 mm wide. Two of these plates are made of steel (304L stainless steel), the third
 267 plate is made of polymethyl methacrylate (PMMA). The plates are placed in parallel so that
 268 they constitute vertical channels in which a primary fluid or refrigerant and a secondary fluid
 269 (hot source) circulate. These fluids, circulate in co-current from bottom to top. The central plate,
 270 separating the two channels, ensures heat exchange between the two fluids. The channel formed
 271 between the two steel plates in which the secondary fluid flows is fixed at 1 mm. The thickness
 272 of the refrigerant circulation channel between the outer PMMA plate and the central plate can
 273 be set at 2, 4 or 6 mm according to the needs of the experiment, using an O-ring and the spacer.
 274 The refrigerant is fed through three capillary tubes with an inner diameter of 2 mm each, located
 275 at the bottom of the plate. This creates a water column, the height of which is deliberately set
 276 according to the needs of the experiment by means of monitoring.

277 The central plate is 6 mm thick and is the seat of 44 K type thermocouples (22 pairs
 278 facing each other - Fig.2b). A detailed configuration of a pair of thermocouples is shown in
 279 Fig. 3. The thermocouples in each pair are recessed at intervals $l_1 = 4$ mm to measure the
 280 temperature distribution in the evaporator center plate (Fig.2a). The depths of the slots are
 281 identical and equal to $l_2 = 1$ mm (distance between the wall and the thermocouple $T_{1,i}$ and
 282 $T_{2,i}$). In this figure, $T_{1,i}$ represents the temperature measured by the thermocouple (i) on the
 283 refrigerant (primary fluid) side and $T_{2,i}$ the temperature measured by a thermocouple (i) on the
 284 secondary fluid side.

285 Due to the driving pressure imposed in the evaporator by the condenser (pressure set by
 286 the thermostatic bath B2) and the evaporator (pressure set by the thermostatic bath B1),
 287 vaporization (boiling) of the refrigerant occurs. The PMMA plate on the refrigerant side being
 288 transparent, the visualization of boiling phenomena occurring in the channel concerned is
 289 allowed and recorded by a high-speed camera (Phantom VEO410). Acquisitions are performed
 290 at 3000 images per second with a resolution of 1280 x 800 and an exposure time of 191.8 μ s.

291

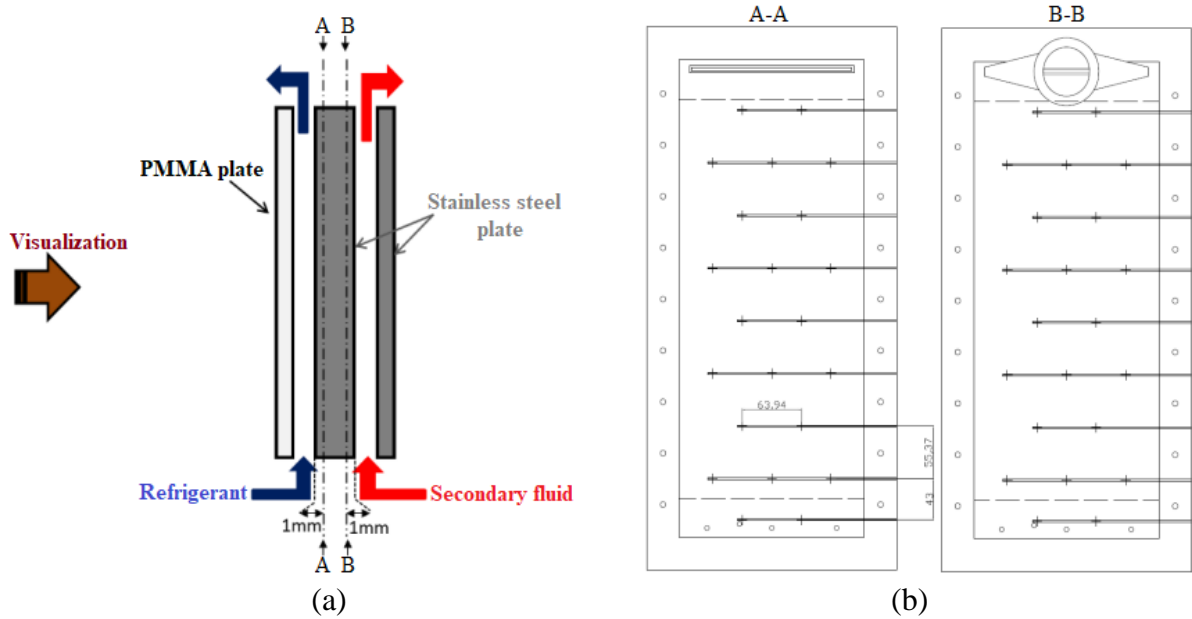


Figure 2 : (a) Vertical smooth plate evaporator (VSPE); (b) Location of thermocouples located in the center plate.

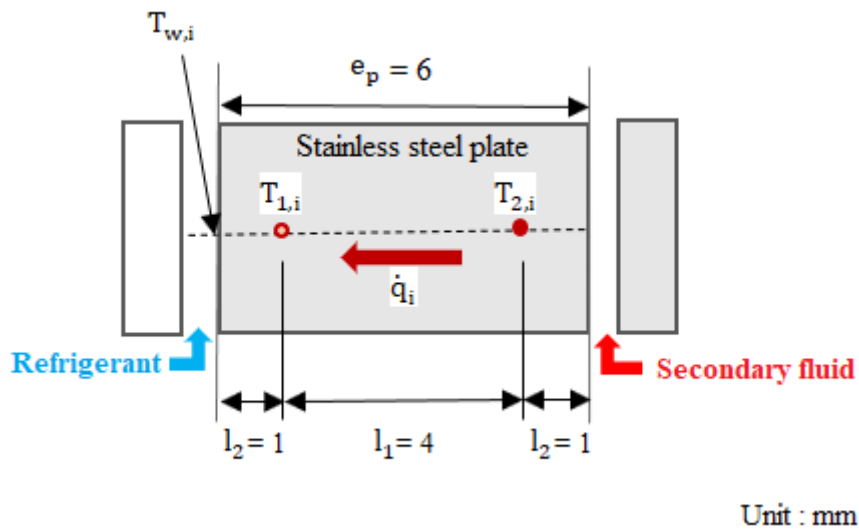


Figure 3: Configuration of a pair of thermocouples

292

293

Further information about the experimental procedure is presented by [16].

294

295 3.2. Experimental Uncertainties

296

297

A summary of the uncertainties is reported in Table 1. All instruments are connected to an acquisition system. The maximum error of the acquisition system is 0.02%. The uncertainties of the data are estimated by assuming that the instrumental error was uniformly distributed and could be estimated by Eq.11 as in [23] cited by [16].

298

299

$$u(a) = \frac{\varphi_{instr} - \varphi_{acq}}{\sqrt{3}} \quad (11)$$

300

301 **Table 1** : Specification of the different measuring devices and standard data uncertainties
 302 associated.

Devices	Type	Accuracy of the device	Range	Data uncertainties
Thermometer	T-type thermocouple	0.1 K	- 10 to 85 °C	0.06 K
Refrigerant flow meter	Thermal mass flow meter	$\pm 1\%$ of full scale	0.04 - 2 kg h ⁻¹	0.01 kg h ⁻¹
Coolant flow meter	Electromagnetic flow meter	$\pm 0.5\%$ of reading	0.017 - 0.034 kg s ⁻¹	0.00006 kg s ⁻¹
Pressure measurement	Pressure transmitter with ceramic sensor	$\pm 0.075\%$ of reading	0 - 5 kPa	0.0009 kPa

303

304 3.3. Experimental program and data reduction

305

306 In order to estimate the impact of the thermophysical properties of the fluid, the heat
 307 flow, the parietal superheating, the filling height of the channel and the geometrical parameters
 308 on the HTC during the confined vaporization of water at subatmospheric pressure in a smooth
 309 plate heat exchanger, the raw experimental data obtained on the test bench (Fig.1) by Giraud et
 310 al. [16] is exploited. These data are determined as a function of the thickness e of the refrigerant
 311 vaporization channel. The database consists of 43 points for $e = 2$ mm, 48 points for $e = 4$ mm
 312 and 48 points for $e = 6$ mm. A total of 139 tests with 1200 values collected for each of them
 313 during at least 20 minutes once steady-state is obtained are exploited. Data obtained are then
 314 used to develop dimensionless number since they allow to vary internal measurements
 315 representing the causes (dimensionless number) and measuring their consequences on the heat
 316 transfer phenomena occurring in the smooth plate evaporator (Fig.2).

317 The experimental conditions covered are summarized in table 2.

318

Table 2 : Domain covered by the experimental test

Parameters	Field
Filling ratio	[10 – 50] % (equivalent in [5 – 25] cm)
Temperature set point at the condenser (bath B ₂ - T _{fs})	[2 – 15] °C (equivalent in an operating pressure of [0.7 – 1.7] kPa)
Temperature set point at the thermostatic bath B3	[30 – 45] °C (equivalent in a mass vapor quality inlet of [1.7 – 3.8] %)
Temperature set point at the evaporator (bath B ₁ - T _{in,fs})	[10 – 25] °C (equivalent in a driving pressure of [0.3 – 2.1] kPa)
Channel thick (refrigerant side - e)	2 mm, 4 mm, 6 mm

319

320 Pressures, temperatures, flow rates etc. are recorded by a LabVIEW program, once the
321 desired experimental conditions are ensured and steady-stated conditions are reached.
322 Experimental data relevant to this study are:

- 323 - inlet temperature of the secondary fluid $T_{in,fs}$
- 324 - cooling capacity \dot{Q}_{cool} which is calculated by a balance at the secondary fluid level,
- 325 - saturation pressures detected at the inlet and outlet of the evaporator refrigerant ($P_{in,ref}$ and
326 $P_{out,ref}$),
- 327 - temperatures measured by the thermocouples on the secondary fluid side T_2 and on the
328 refrigerant side T_1 etc.

329 Based on this data, a local heat flux is estimated assuming one directional heat flux and
330 quasi-steady state conditions. This local heat flux is the heat flux estimated at the area where a
331 pair of thermocouples is located. It is determined by the derivative of the one-dimensional heat
332 conduction (Eq. 12).

333

$$\dot{q}_i = \lambda_p \frac{T_{2,i} - T_{1,i}}{l_1} \quad (12)$$

334 The average heat flow from the secondary fluid to the refrigerant for each zone (boiling
335 zone, liquid film evaporation zone and vapor zone) is then determined by the Eq. 13.

$$\dot{q}_z = \frac{S}{l_p H_z} \sum_{i=1}^n \left(\lambda_p \frac{T_{2,i} - T_{1,i}}{l_1} \right) \quad (13)$$

336

337 Where $S = l_p * 0.05537m^2$ is the area of the mesh around a thermocouple (elementary
338 area), $l_p = 0.2$ m is the width of the plate, n is the maximum number of thermocouples in the
339 considered zone, $\lambda_p = 16.3$ W. m⁻¹. K⁻¹ is the thermal conductivity of stainless steel, assumed
340 to be constant in the temperature range studied and H_z is the height of the considered zone. This
341 considered zone is based on the previous observation of Giraud et al. [5], those identify four
342 different working zones: a zone at the entrance of the channel where the thermal equilibrium is
343 reached between the refrigerant and the secondary fluid and in which the incoming bubbles
344 collapse, a boiling zone, a liquid film evaporation zone and a vapour zone (Fig. 4). Thus, heat
345 flows are averaged by zone (i.e. a mean heat flow is estimated for the boiling area, one for the
346 falling film area and one for the vapor area).

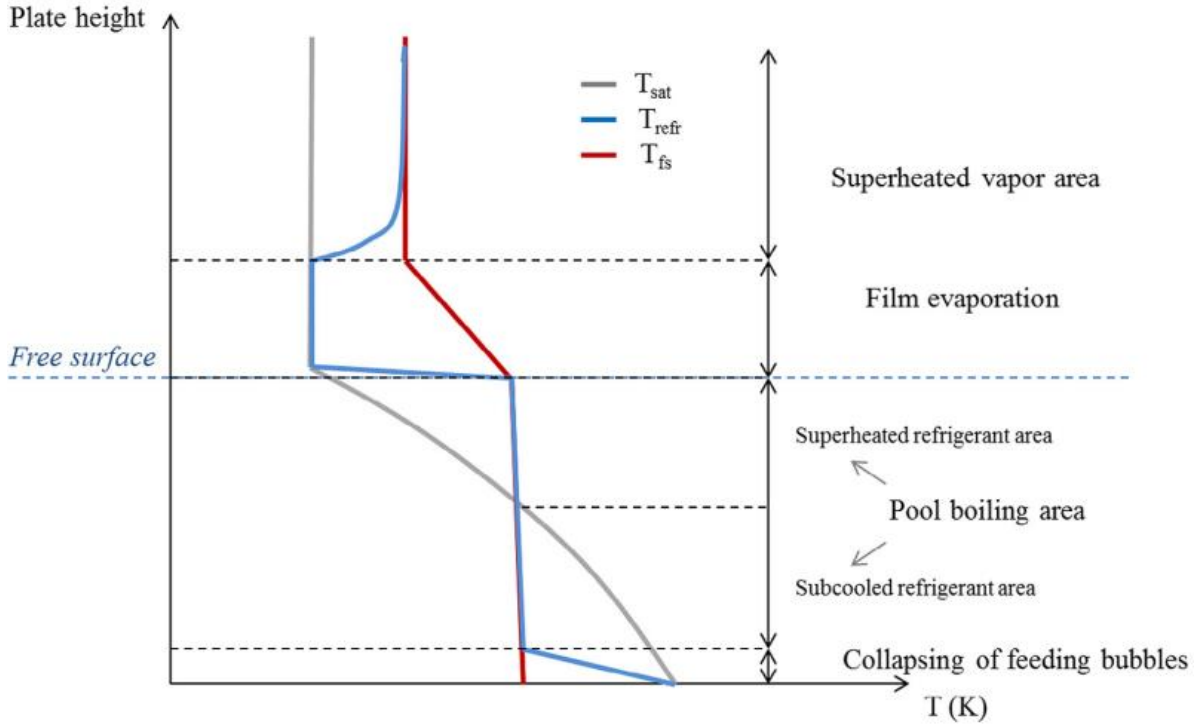


Figure 4: Schematic diagram of the temperature development of the secondary fluid and refrigerant as a function of evaporator height [17]

347

348 The local temperature at the plate wall $T_{w,i}$ in front of a thermocouple (i) on the refrigerant
 349 side (Fig. 3) is calculated by extrapolation by Eq. (14).

$$T_{w,i} = T_{1,i} - \frac{\dot{q}_i \cdot l_2}{\lambda_p} \quad (14)$$

350 Thus, the average wall temperature for the considered area on the refrigerant side $T_{w,z}$ is
 351 then calculated by the Eq. 15.

352

$$T_{w,z} = \frac{1}{n} \sum_{i=1}^n \left(T_{1,i} - \frac{l_2 \cdot (T_{2,i} - T_{1,i})}{l_1} \right) \quad (15)$$

353

354 Thereafter, only heat transfer phenomena in the boiling zone and the evaporation zone
 355 of the liquid film are considered.

356 When vaporizing water at low pressure in the VSPE, for a filling height H_1 , the periodic
 357 appearance of bubbles is observed at a distance H_b below the free liquid surface [5] such that
 358 $H_1 > H_b$. And H_b varies according to the operating conditions of each test point. So, the
 359 saturation temperature in the boiling zone $T_{sat,z} = T_{sat,b}$ depends on the hydrostatic pressure
 360 ($P_{static} = \rho_l g d_s$) and the free surface pressure ($P_{sat} = P_{out,ref}$). This saturation temperature is
 361 determined by Eq. 16.

$$T_{sat,b} = T_{sat}(P_{out,ref} + \rho_l g H_b) \quad (16)$$

362

363 For the liquid film zone, the saturation temperature $T_{sat,z} = T_{sat,f}$ depends on the free
364 surface pressure ($P_{sat} = P_{out,ref}$) and is determined by Eq. 17.

$$T_{sat,f} = T_{sat}(P_{out,ref}) \quad (17)$$

365

366 Thus, the heat transfer coefficient for each zone is determined by Eq. 18.

$$h_z = \frac{\dot{q}_z}{T_{w,z} - T_{sat,z}} \quad (18)$$

367

368 The liquid column height (H_l) related to the feed mass flow rate, is calculated by the
369 Eq.19.

$$H_l = \frac{P_{in,ref} - P_{out,ref}}{\rho_l g} \quad (19)$$

370

371 The distance between the nucleation site and the free interface (H_b) and the projection
372 height of the droplets forming the liquid film, are determined manually by the software ImageJ.
373 The thermophysical properties of the fluid are calculated according to the saturation
374 temperatures of each medium using Coolprop software.

374

375 4. Test of HTC correlations of literature

376 In order to evaluate the ability of these correlations to predict HTC in the experimental
377 configuration used in this paper, i.e. for confined pool boiling and film liquid evaporation at
378 subatmospheric pressures, a comparison is made between the HTC obtained following the data
379 reduction introduced in 3.4 and the HTC predicted by correlations from the literature. The
380 comparison conducted is: a graphical and statistical comparison conducted by calculating the
381 Mean Absolute Percentage Error (MAE) (Eq. 20), the Mean Relative Percentage Error (MRE)
382 (Eq. 21) and the correlation coefficient (r) (Eq. 22). The MAE is used to check the accuracy of
383 the predictions and the MRE is used to check whether a correlation is over or under predicted
384 average. The percentage of predicted data within a range of $\pm 30\%$, noted τ_{30} , is also calculated,
385 to provide a quantitative indication of the correlations performance. This value, although it
386 seems high, is consistent with the very complex phenomena of subatmospheric water boiling
387 [16]. The correlation coefficient, always ranging between -1 and 1, measures the affinity
388 between the experimental and predicted data:

- 389 - The closer r is to zero, the weaker the linear relationship.
- 390 - Positive values of r indicate a positive correlation when the values of the two variables tend
391 to increase together.
- 392 - Negative values of r indicate a negative correlation when the values of one variable tend to
393 increase and the values of the other variable decrease.

394 - The values 1 and -1 each represent "perfect" correlations, positive and negative respectively.
 395 Thus, the relationship is said to be linear.

$$MAE = \frac{100}{N} \sum_{i=1}^N \frac{|h_{pred} - h_{exp}|}{h_{exp}} \quad (20)$$

396

$$MRE = \frac{100}{N} \sum_{i=1}^N \left(\frac{h_{pred} - h_{exp}}{h_{exp}} \right) \quad (21)$$

397

$$r = \frac{cov(h_{exp}, h_{pred})}{\sqrt{var(h_{exp}) \cdot var(h_{pred})}} \quad (22)$$

398

399 To remain consistent, the constant (C) and coefficient (a and b) of the correlation
 400 Eq. (1) are fitted to the experimental database of Giraud et al. [16] (Table 3), minimising the
 401 sum of the differences between the experimental and theoretical values of the Nusselt number
 402 (Nu). The parameters constituting Eq. 1 are calculated as a function of the saturation
 403 temperatures of each medium considered. The definition of the carateritic length (δ) and
 404 velocity (U) depends on the area under consideration (table 7).

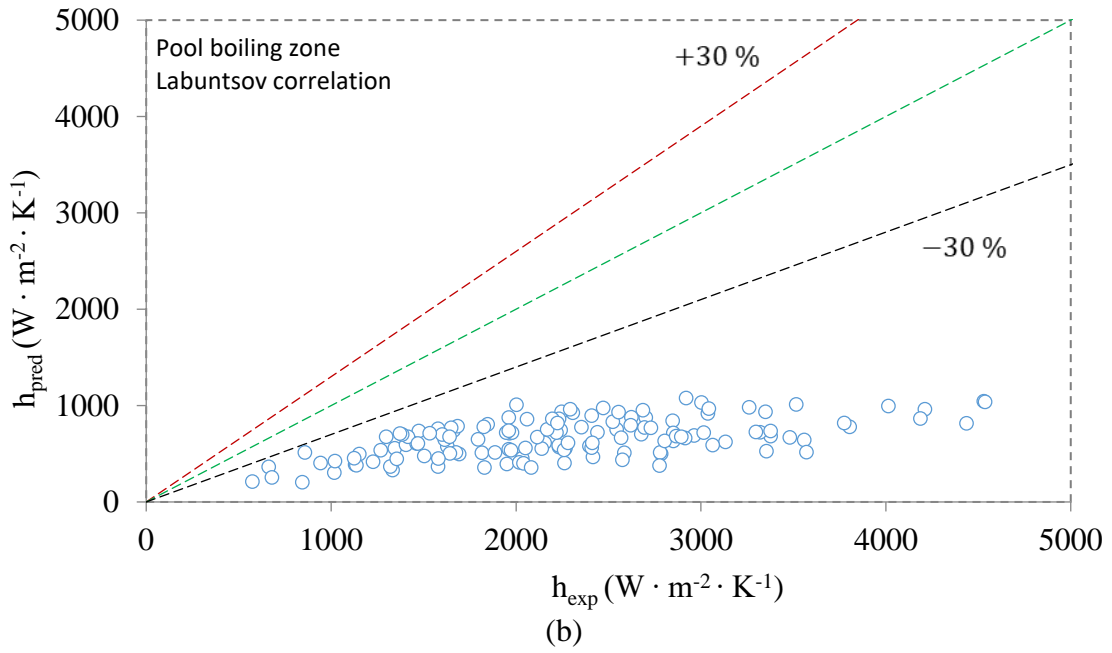
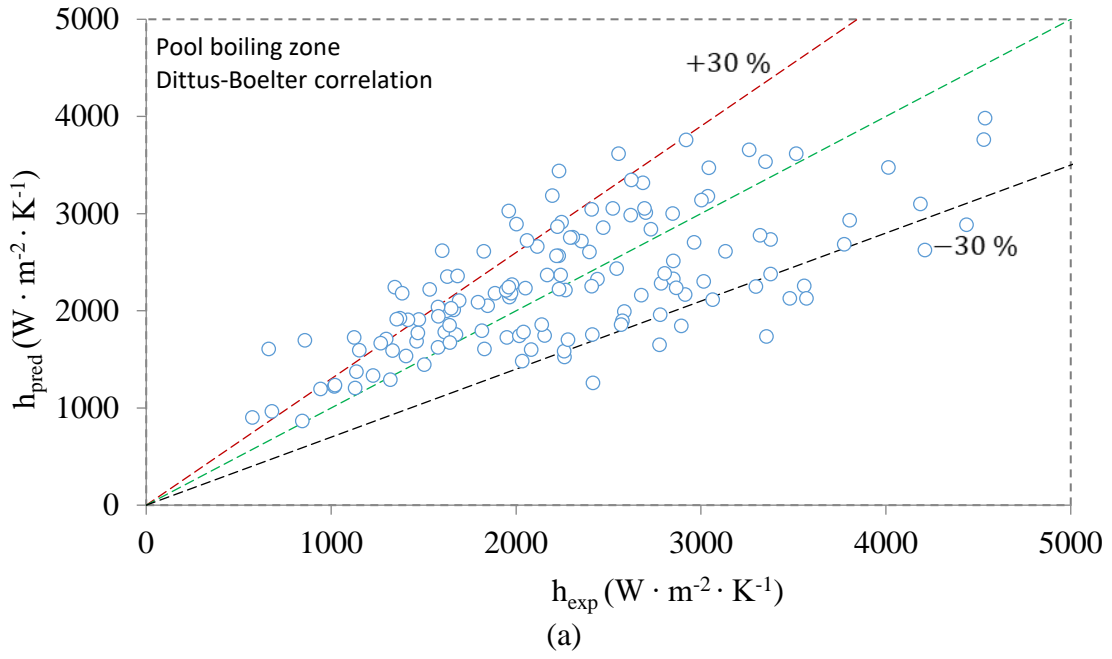
Table 3 : Constants and coefficients fitted to experimental database of Giraud et al. [16]

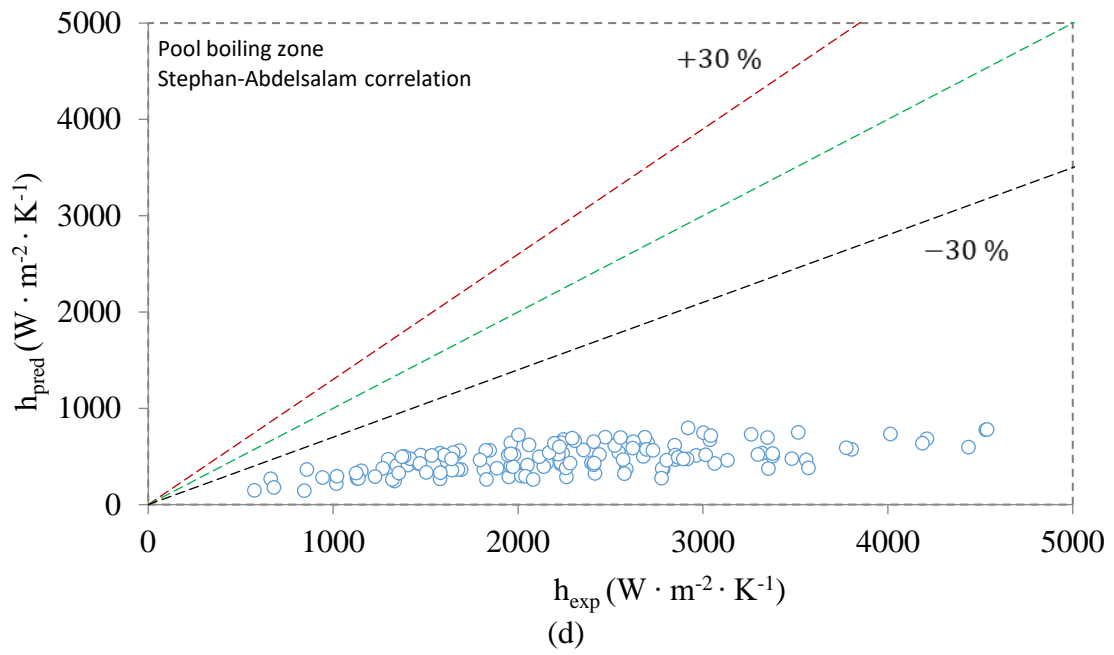
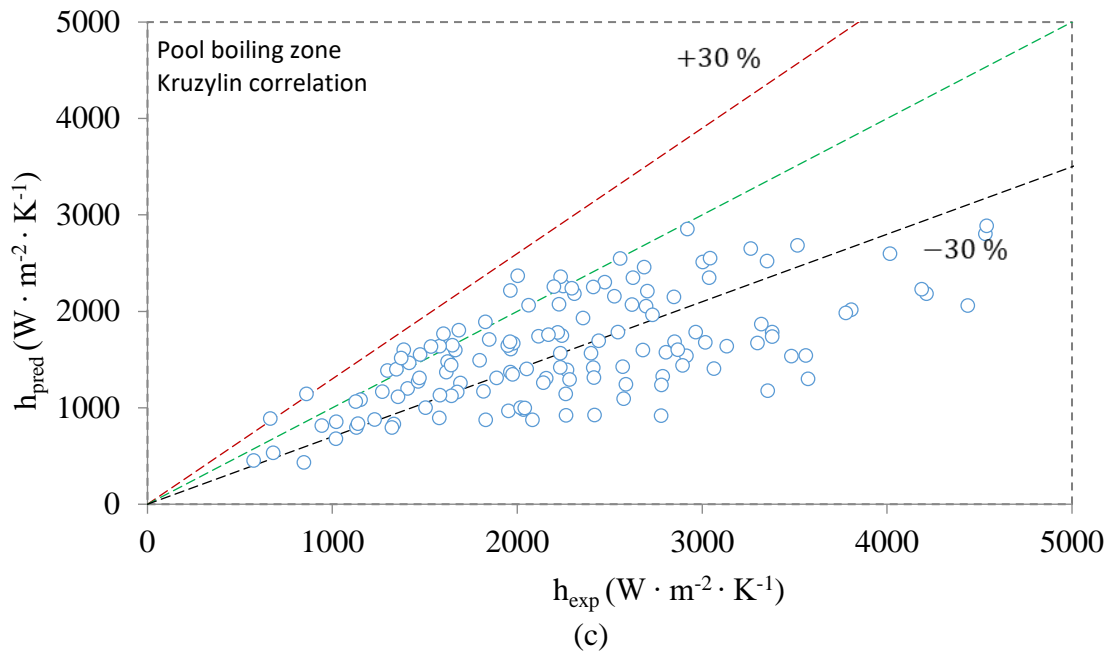
Constants and coefficients fitted			
C	a	b	Zones
38.405	0.484	-1.976	Pooling boiling
0.067	0.742	-1.541	Liquid film evaporation

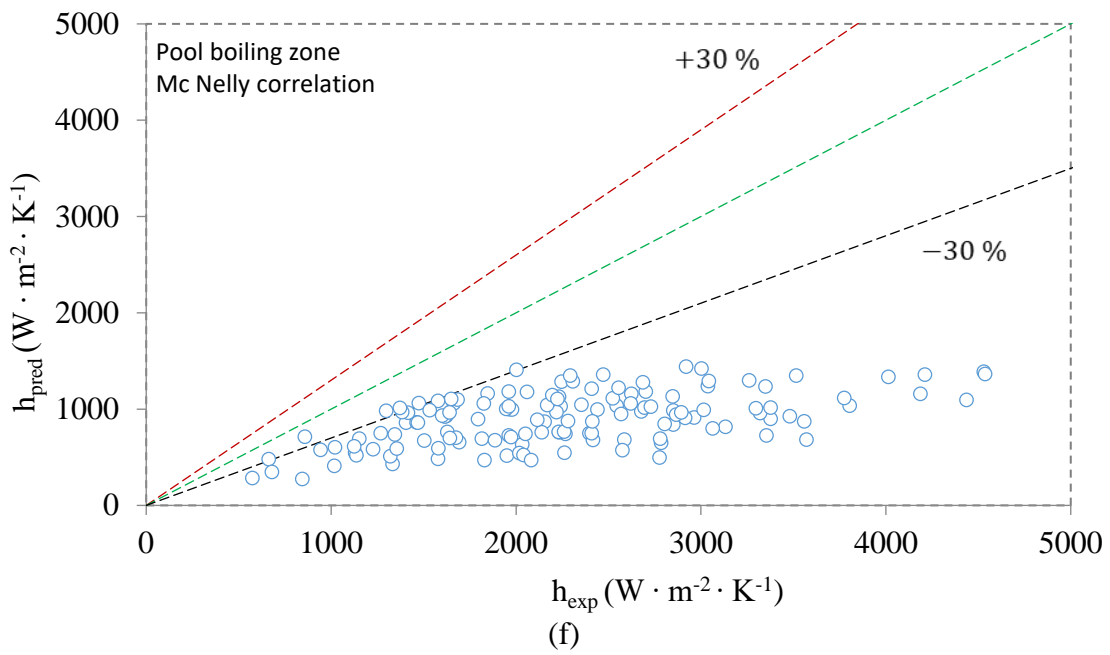
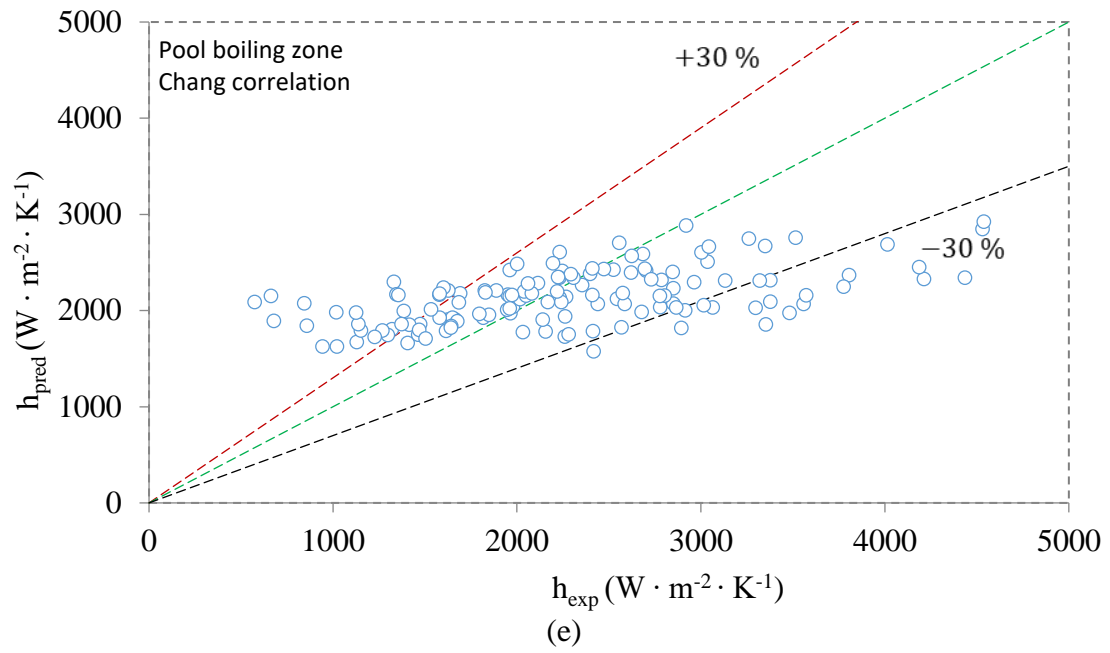
405

406 4.1. Pool boiling zone

407 The results of the comparisons between the experimental HTC of pool boiling and those
 408 predicted by the literature correlations are shown to the Fig. 5 (graphical comparison) and
 409 Table 4 (statistical comparison).







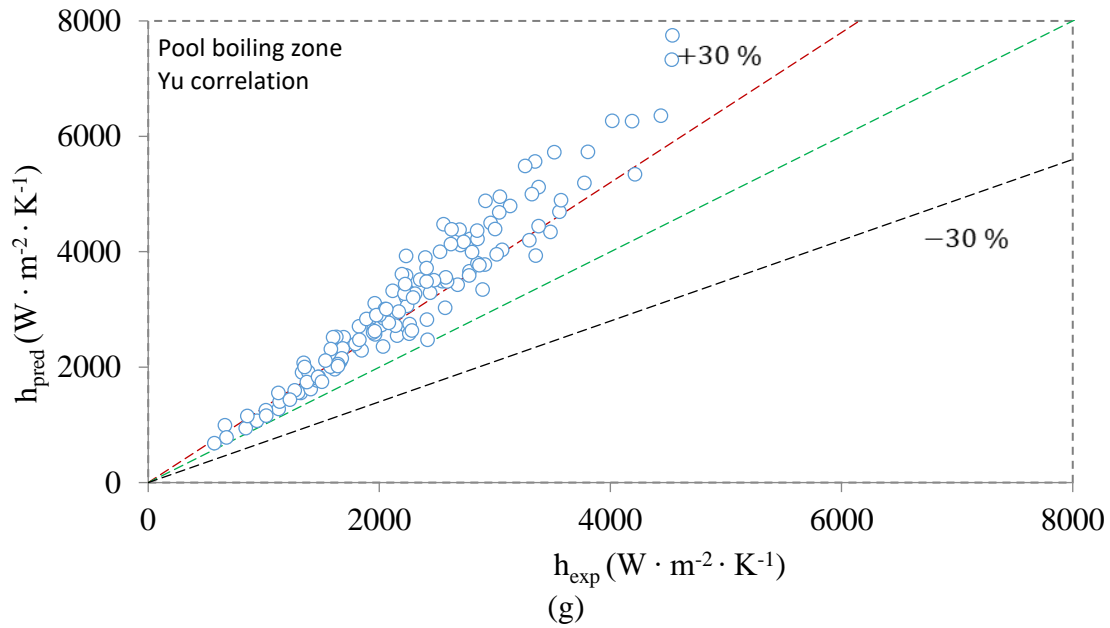


Figure 5 : Comparison between experimental HTC (h_{exp}) of pool boiling zone and HTC predicted (h_{pred}) by correlations of: (a) Dittus-Boelter [19], (b) Labuntsov [11], (c) Kruzylin [11], (d) Stephan-Abdelsalam [11], (e) Chang [13], (f) Mc Nelly [21] and (g) Yu [20]

410

Table 4 : Statistical comparison between the HTC experimental of pool boiling zone and the HTC predicted by the correlations in the literature.

Correlations	Pool boiling zone			
	MAE [%]	MRE [%]	τ_{30} [%]	r [-]
Dittus-Boelter [19]	24	7	73	0.67
Kruzylin [11]	28	-26	54	0.64
Labuntsov [11]	69	-69	0	0.58
Stephan-Abdelsalam [11]	77	-77	0	0.60
Chang [13]	29	8	69	0.57
Mc Nelly [21]	57	-57	3	0.55
Yu [20]	39	39	27	0.96

411

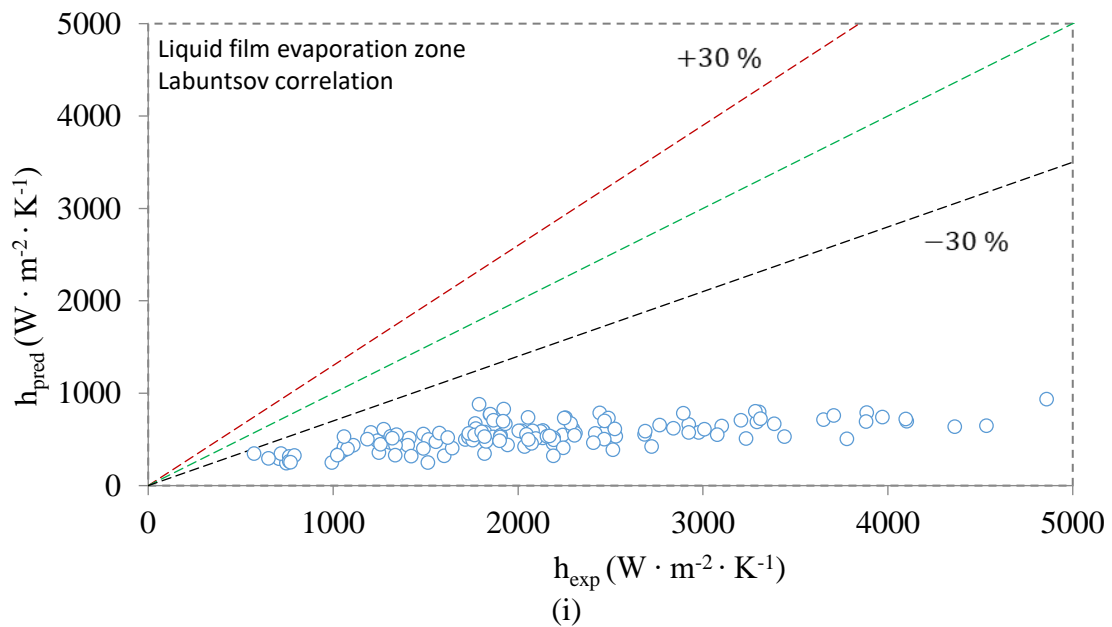
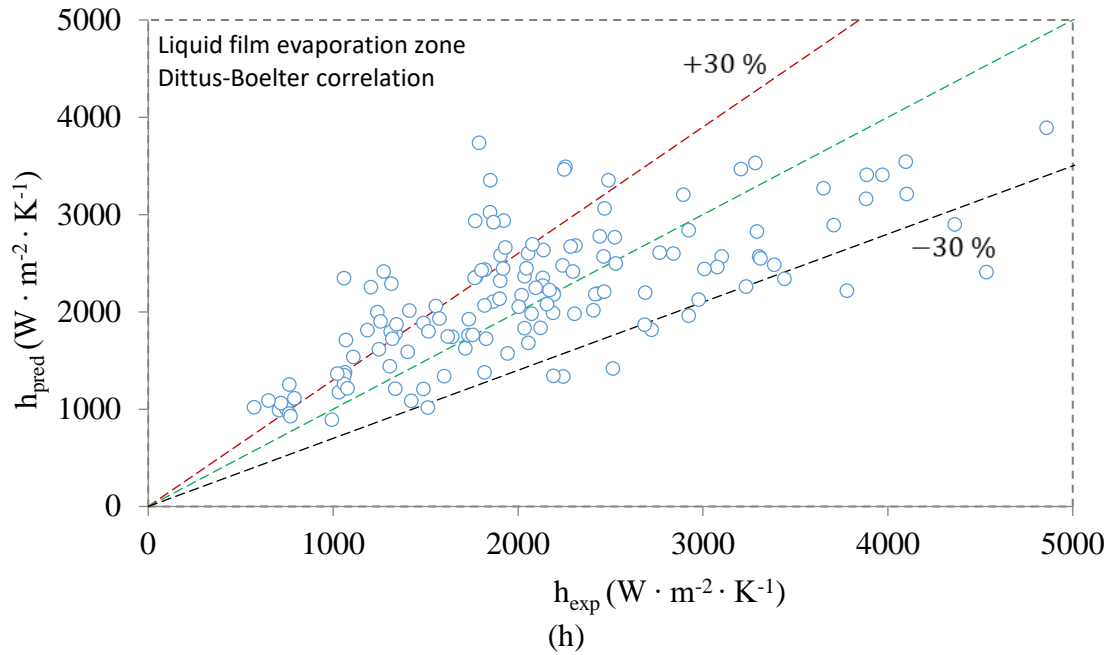
412 Table 4 indicates that the HTC experimental database for the pool boiling area did not
 413 accurately match the majority of the correlations considered.

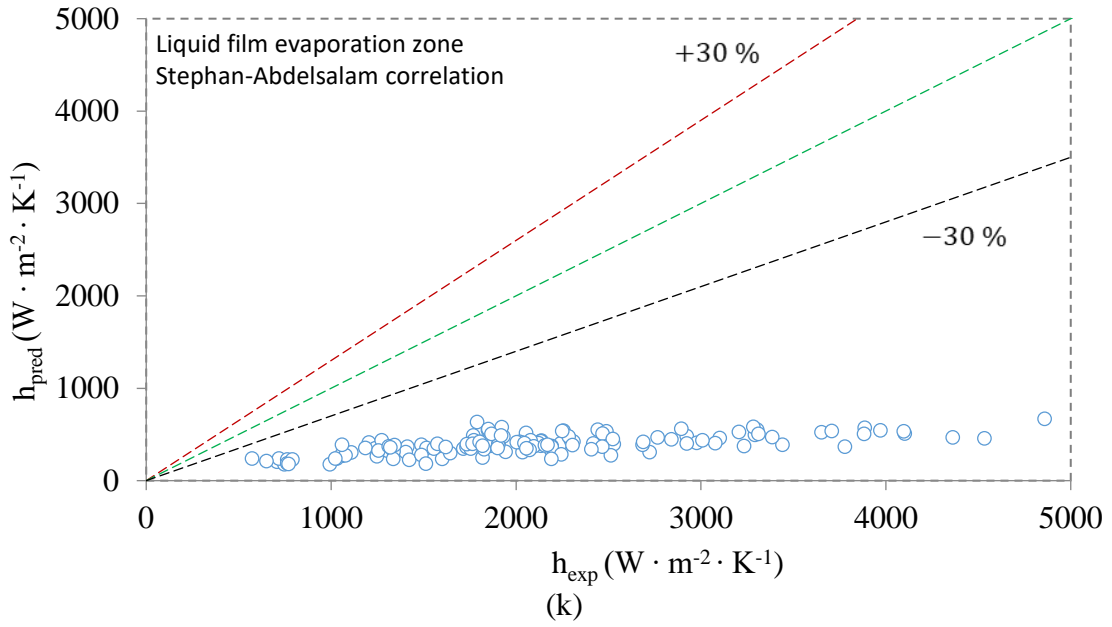
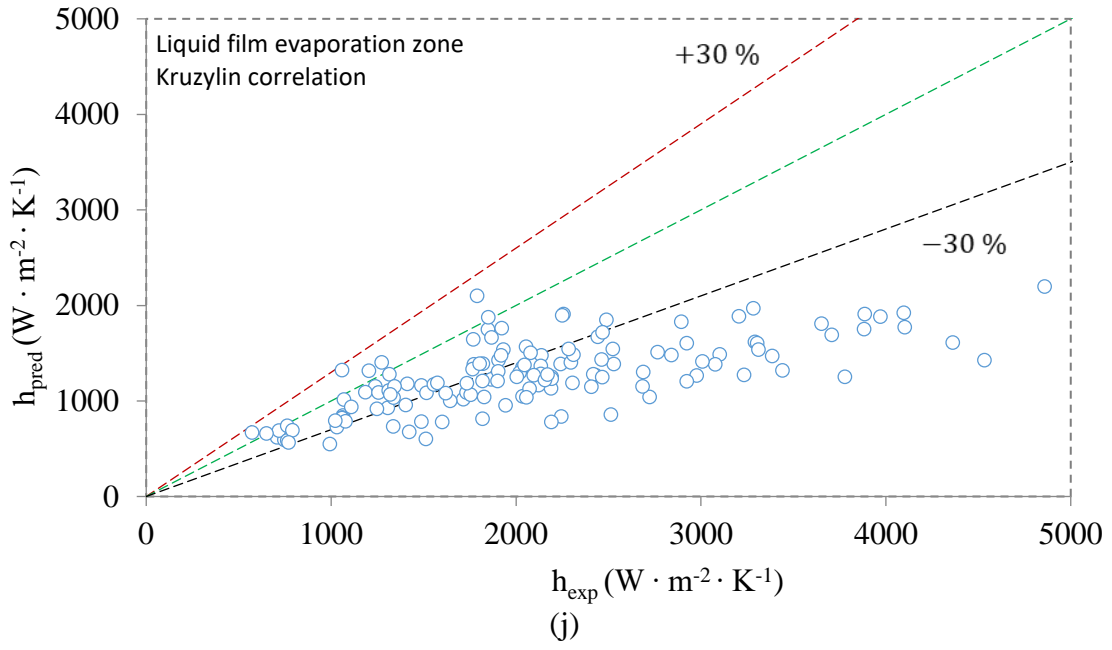
414 Labuntsov (Eq.2) and Stephan-Abdelsalam (Eq. 4) correlations show the worst results
 415 by predicting 0% of the experimental database within an error band of $\pm 30\%$. Mc Nelly (Eq. 9)
 416 and Yu (Eq. 10) correlations, also poorly predict 3%, and 27% of the experimental database
 417 within an error band of $\pm 30\%$, respectively. All these above-mentioned correlations give very
 418 poor precision and underestimate the experimental database. The Dittus-Boelter (Eq. 1),
 419 Kruzylin (Eq. 3) and Chang (Eq. 6) correlations, however, provided more accurate results.
 420 Dittus-Boelter correlation shows the best performance by satisfactory predicts 73 % of the
 421 database within an error band of $\pm 30\%$ with an accuracy of 24%. For the pool boiling zone, this
 422 correlation over-estimates the experimental values by 7%. Kruzylin correlation correctly
 423 predicts 54 % of the database within a $\pm 30\%$ error band with an accuracy of 28%. The
 424 correlation (Eq. 3) underpredicts 26% of the experimental data. Chang correlation offers a

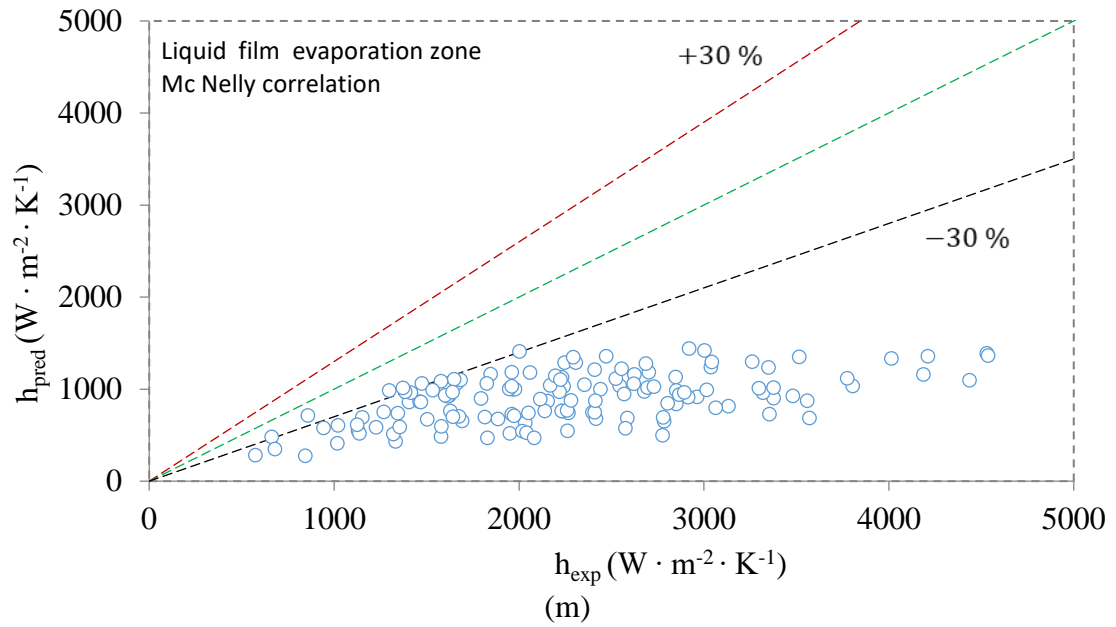
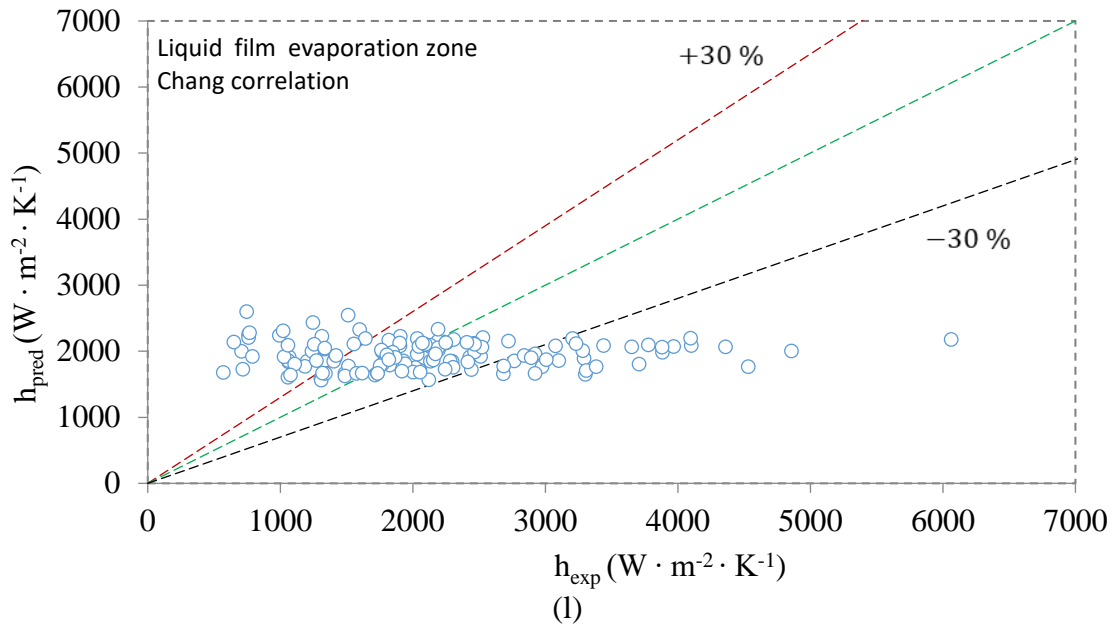
425 satisfactory result and correctly predicts 69% of the database within an error band of $\pm 30\%$ with
426 an accuracy of 29 %. It over-estimates the experimental values by 8 %.

427 4.2. Liquid film evaporation zone

428 For the liquid film evaporation zone, the comparison between the experimental HTC
429 and those estimated by correlations from the literature are shown in the Fig. 6 (graphical
430 comparison) and Table 5 (statistical comparison).







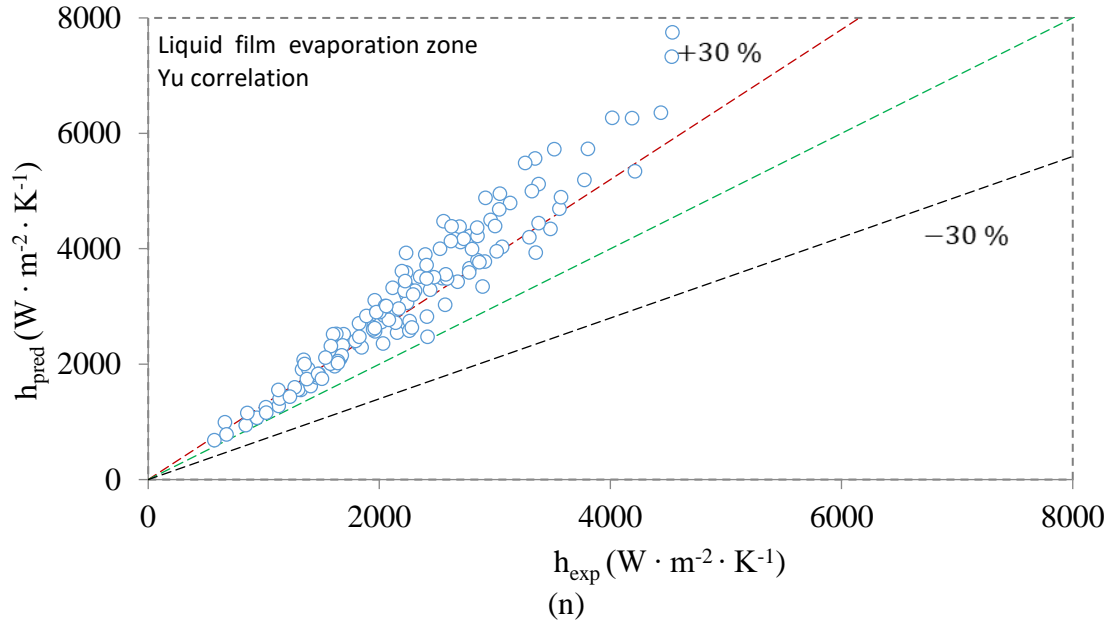


Figure 6 : Comparison between experimental HTC (h_{exp}) of liquid film evaporation zone and HTC predicted (h_{pred}) by correlations of: (h) Dittus-Boelter [19], (i) Labuntsov [11], (j) Kruzylin [11], (k) Stephan-Abdelsalam [11], (l) Chang [13], (m) Mc Nelly [21] and (n) Yu [20]

431

Table 5 : Statistical comparison between the experimental HTC of liquid film evaporation zone and the HTC predicted by the correlations from the literature

Liquid film evaporation zone				
Correlations	MAE [%]	MRE [%]	τ_{30} [%]	r
Dittus-Boelter [19]	27	12	68	0.72
Kruzylin [11]	36	- 34	35	0.72
Labuntsov [11]	73	- 73	0	0.67
Stephan-Abdelsalam [11]	80	- 80	0	0.69
Chang [13]	40	13	54	0.50
Mc Nelly [21]	61	- 61	1	0.64
Yu [20]	31	31	45	0.97

432

433 Table 5 shows that correlations from the literature do not accurately predict the
 434 experimental HTC database for the liquid film evaporation zone, despite their validity at low-
 435 pressure conditions and/or the amount and diversity of involved parameters in their
 436 development.

437 Labuntsov (Eq. 2) and Stephan-Abdelsalam (Eq. 4) correlations show the worst results by
 438 predicting 0 % of the experimental database within a $\pm 30\%$ error band. As for the Kruzylin
 439 (Eq. 3), Mc Nelly (Eq. 9) and Yu (Eq. 10) correlations, they also poorly perform by predicting
 440 35%, 1% and 45% respectively of the experimental database within a $\pm 30\%$ error band.
 441 However, Dittus-Boelter (Eq. 1) and Chang (Eq. 6) correlations provided satisfactory results.
 442 Dittus-Boelter correlation (Eq. 1) correctly predicts 68 % of the database within an error band
 443 of $\pm 30\%$ with an accuracy of 27%. It over-estimates the experimental values by 12%. Chang
 444 correlation (Eq. 7) correctly predicts 54% of the database within a $\pm 30\%$ error band with an

445 accuracy of 40%. The values calculated with the correlation (Eq. 7) are increased by 13%
446 compared to the experimental values.

447 The poor results of the correlations from the literature in accurately predicting the heat
448 transfer coefficient may be due to the complexity of the coupled mechanisms involved during
449 confined vaporization at subatmospheric pressure. Furthermore, with the exception of the
450 correlations (Eq. 6) and (Eq. 7), the other correlations were originally designed to predict free
451 boiling heat transfer coefficients and at higher pressures, although their validity at low pressures
452 has been verified by [11] [21] and [20]. All these correlations are also developed for higher
453 heat flux densities than those observed in this study. From a physical point of view, the
454 observations of Chang et al. [13] are in perfect agreement with those observed by Giraud et
455 al. [5] [16]. Therefore, the good performance of the correlations (Eq. 6) and (Eq. 7) was
456 expected.

457 However, all these correlations in the literature (Eq. 1 to Eq. 10, except Eq. 5) do not take
458 into account all the relevant parameters that influence the HTC at low pressure. For example,
459 the liquid height is absent of these equations whereas it has been shown that this parameter
460 strongly influence the heat transfer at low pressure [13] [4] [14] [24] [15]. Also, given the
461 centimetric size of the bubbles during the vaporization of water at subatmospheric pressure [4]
462 , a parameter influencing their rise and their growth would obviously be determining. It should
463 also be noted that the dimensional analysis adopted for the development of the correlations (Eq.
464 1 to Eq. 10, except Eq. 5), does not verify the physical independence of the physical quantities.
465 For the correlations of Labuntsov (Eq. 2), Kruzylin (Eq. 3), Stephan-Abdelsalam (Eq. 4), Chang
466 (Eq. 6 and Eq. 7), Mc Nelly (Eq.9) and Yu (Eq. 10), the thermophysical properties of the fluid
467 and the heat flux density (\dot{q}), depend on the saturation conditions of the medium (P_{sat} , T_{sat}). Eq.
468 5 also shows a dependence between the bubble diameter (d_b) and the thermophysical properties
469 of the fluid. However, it is necessary to ensure that no physical quantity, apart from the variable
470 of interest, depends on the variation of the other quantities [18]. This independence is a key
471 factor for modeling by dimensional analysis since it makes possible to easily transpose results
472 from one scale to another (similarity).

473 5. New pool boiling and liquid film evaporation HTC predictive correlations

474

475 5.1. Dimensional analysis

476 Based on the scientific knowledge generated by the literature, the rigorous dimensional
477 analysis approach based on the Vaschy-Buckingham theorem requires the definition of the
478 independent physical quantities governing the heat transfer phenomena occurring in the
479 evaporator (Fig.2a). These parameters, which are to be considered during boiling and
480 evaporation at subatmospheric pressures in the evaporator, are the thermophysical properties of
481 the liquid and vapour, the thermodynamic conditions (pressure and temperature) and boundary
482 conditions.

483 The HTC (h) on the refrigerant side, the target variable or variable of interest, depends
484 mainly on : the pressure (P_{sat}), the vapor mass velocity (G_v), the heat flow (\dot{q}), the enthalpy of
485 change of state from liquid to vapour (Δh_{lv}), the difference between the refrigerant side wall
486 temperature T_w and the temperature of the saturated liquid medium T_{sat} noted (ΔT) (maximum
487 temperature difference for a bubble to develop), the densities of the liquid and vapour (ρ_l and
488 ρ_v), the vapour density at liquid temperature (ρ_b), the specific heat (or mass heat capacity) of
489 the liquid (c_{pl}), the thermal conductivity of the liquid (λ_l), the dynamic viscosities of the liquid

490 and vapor (μ_l and μ_v), the surface tension at the liquid-vapour interface (σ), the thickness of
 491 the plate containment (e), the liquid column height (H_l), and the acceleration of gravity (g).

492 The crucial point for a rigorous dimensional analysis is the verification of the physical
 493 independence of the listed physical quantities. It is then necessary to make sure that no physical
 494 quantity, except the variable of interest h , is influenced by the variation of the other quantities
 495 [18]. All the thermophysical properties of the refrigerant, listed here, except the vapour density
 496 at liquid temperature (ρ_b), are physically dependent on the saturation conditions of the medium
 497 (P_{sat}, T_{sat}). The vapor mass velocity is defined by Eq. 23 which shows that G_v , Δh_{lv} , and $\dot{q}(\Delta T)$
 498 do not have to be listed simultaneously. It is then sufficient to take one of these parameters (for
 499 example: G_v).

$$G_v = \frac{\dot{q}_z}{\Delta h_{lv}} \quad (23)$$

500

501 Thus, the problem can be set as follows:

$$h = h(H_l, \mu_v, \rho_v, \rho_b, \sigma, g, \lambda_l, G_v, e, \mu_l, \rho_l, c_{pl}) \quad (24)$$

502

503 The mathematical model of the physical law studied (Eq. 24) indicates that the HTC (h)
 504 depends on 12 independent dimensional parameters. Thus, the Vaschy-Buckingham theorem
 505 allows us to express Eq. 24 in dimensionless form by Eq. 25 (see details in Appendix A).

506

$$Nu = Nu(F^*, \mu^*, \rho^*, \rho_v^*, Bo, Ca, Pr, We) \quad (25)$$

507

508 Thus, Eq. 25 shows that the Nusselt number (Nu) depends on 8 dimensionless groups.
 509 Since the heat transfer phenomena in the pool boiling zone and the evaporation zone of the
 510 liquid film observed (Fig. 3), are considered, the thermophysical properties of the fluid vary
 511 according to each medium as they are calculated according to the saturation temperatures
 512 defined by Eq. 16 and Eq.17. Although the physical meaning and/or definition of the
 513 dimensionless numbers introduced in Eq. 25 may depend on the area considered (pool boiling
 514 or liquid film evaporation), the formulation for each area remains identical. The definition of
 515 these and their meaning are described in Table 6.

Table 6 : Dimensionless numbers

Notations	Formulas	Physical definitions and meaning
Nu	$\frac{h \delta}{\lambda_l}$	Nusselt number compares the effective thermal energy transport through the boundary layer between the refrigerant and the core plate wall (convective transfer) to purely diffusive conduction (conduction in the medium).
F*	$\frac{e}{H_l}$	The form factor is the ratio between the plate confining thickness (e) and the water column height (H_l). It allows to find the right filling ratio and channel thickness for better system performance and is thus of great importance for the evaporator sizing.

ρ_v^*	$\frac{\rho_b}{\rho_v}$	The ratio between the density of vapour at liquid temperature (ρ_b) and the density of vapour at saturation temperature (ρ_v) is a determining parameter that influences the rise and growth of bubbles at low pressure.
F^{**}	$\frac{e}{\delta}$	The shape factor is the ratio between the containment thickness of the plate (e) and the characteristic length (δ).
Ca	$\frac{U \mu_l}{\sigma}$	Capillary number compares the force of viscosity to the force of interfacial tension (e.g., two-phase microchannel flow).
We	$\frac{\rho_l U^2 \delta}{\sigma}$	Weber number characterizes the flow of fluids at the interface of a multiphase system. It corresponds to the ratio of inertial forces to surface tension forces.

516

517 The definition of the characteristic length (δ) and velocity (U) depends on the area
518 considered. The physical quantities that constitute them are different whether the pool boiling
519 zone or the falling film zone is considered. These numbers are defined in table 7.

Table 7 : characteristic lengths (δ) and velocities (U)

Notations	Formulas	Physical definitions and meaning	Zones
$\delta = \delta_b$	$\sqrt{\frac{\sigma}{\rho_l g}}$	δ_b is the modified capillary length. It is the characteristic distance below which capillary phenomena are predominant over gravitational phenomena. The modified capillary length characterizes the relationship between the surface tension forces and the gravitational forces acting on the interface.	Pool boiling
$U = U_b$	$\frac{\dot{q}_b}{\rho_v \Delta h_{lv}}$	U_b is the superficial velocity of the vapour bubble.	Pool boiling
$\delta = \delta_f$	$\left(\frac{\mu_l^2}{\rho_l^2 g}\right)^{\frac{1}{3}}$	δ_f is the visco-gravity length.	Liquid film evaporation
$U = U_f$	$\frac{\dot{q}_f}{\rho_v \Delta h_{lv}}$	U_f is the characteristic average evaporation velocity of the liquid film, determined on the basis of the total heat flux at the wall.	Liquid film evaporation

520

521 Thus, for the pool boiling zone, F^{**} (Table 6) is the Bond number (Bo). It represents the
522 ratio of buoyancy forces to surface tension forces. It is commonly used in confined boiling
523 studies and can be seen as an indicator of bubble "squeezing" [25].

524 For the falling film zone, F^{**} (Table 6) is the shape factor between the containment
525 thickness of the plate (e) and visco-gravity length (δ_f) (noted Bo_f) since the flow of the liquid
526 film falling on the vertical smooth plate (Fig.2) is induced by the gravity force (gravity flow).
527 The thickness of the liquid film necessarily depends on the visco-gravity length and the flow

528 velocity. The HTC during evaporation of a liquid film running down a wall depends on the
529 thickness of the film, which depends on the flow regime.

530 **5.2. Development of news HTC correlations**

531 The monomial form (Eq. 26) was preferred to simplify the identification of the
532 unknowns. Indeed, the relatively general form, i.e., the one based on a sum of monomials where
533 each monomial is a product of internal measurements at different powers, more accurate, makes
534 the identification of the unknowns very complex and requires a lot of experimental data.
535 Moreover, in the search for the process relationship via a monomial form, it is necessary to rely
536 on an experimental program where each of the dimensionless numbers varies within a
537 sufficiently wide range. Otherwise, the value of the exponents $a_{i \in [1;8]}$ is not guaranteed and
538 may bias the value of the constant C [18].

$$539 \quad Nu = C \cdot (F^*)^{a_1} \cdot (\mu^*)^{a_2} \cdot (\rho^*)^{a_3} \cdot (\rho_v^*)^{a_4} \cdot (Bo)^{a_5} \cdot (Ca)^{a_6} \cdot (Pr)^{a_7} \cdot (We)^{a_8} \quad (26)$$

540 For the range of operating points tested in this study, the dimensionless numbers
541 μ^* , ρ^* and Pr show small variations that bias the values of coefficients and the constant C of
542 Eq. 26. As a result, their elimination from the list of thermophysical properties is possible and
543 this does not change the appearance of the scatterplot cloud formed by Eq. 26. This elimination
544 is in line with the suggestion of Kew and Corwell (1997) quoted by [26], who encourages to
545 eliminate any non-dimensional group that does not improve the accuracy of the prediction
546 model. However, there is no evidence that the non-dimensional group eliminated do not
547 improve the accuracy of the model prediction in another range of operating conditions and/or
548 with another fluid. Thus, in this paper Eq. 26 is simplified by becoming the Eq. 27 but it is
549 highly recommended to use Eq. 26 for higher variation of μ^* , ρ^* and Pr.

$$550 \quad Nu = C \cdot (F^*)^{a_1} \cdot (\rho_v^*)^{a_2} \cdot (F^{**})^{a_3} \cdot (Ca)^{a_4} \cdot (We)^{a_5} \quad (27)$$

550

551 Based on the regression analysis of the experimental database, using the least squares
552 method (minimizing the sum of the differences between the experimental and theoretical values
553 of the Nusselt number Nu), following HTC correlations is obtained:

554 - **Pool boiling zone**

$$555 \quad \frac{h \delta_b}{\lambda_l} = 2.664 \cdot 10^{-6} \cdot \left(\frac{e}{H_l}\right)^{0.410} \cdot \left(\frac{\rho_b}{\rho_v}\right)^{0.157} \cdot \left(\frac{e}{\delta_b}\right)^{-0.382} \cdot \left(\frac{U_b \mu_l}{\sigma}\right)^{-2.812} \cdot \left(\frac{\rho_l U_b^2 \delta_s}{\sigma}\right)^{1.630} \quad (28)$$

555

556 - **Liquid film evaporation zone**

$$557 \quad \frac{h \delta_f}{\lambda_l} = 1.195 \cdot 10^{-8} \cdot \left(\frac{e}{H_l}\right)^{-0.411} \cdot \left(\frac{\rho_b}{\rho_v}\right)^{0.231} \cdot \left(\frac{e}{\delta_f}\right)^{0.767} \cdot \left(\frac{U_f \mu_l}{\sigma}\right)^{-3.247} \cdot \left(\frac{\rho_l U_v^2 \delta_f}{\sigma}\right)^{1.929} \quad (29)$$

557

558 The domain of correlations application is the important factor that limits its
559 performance. The correlations Eq. (28) and Eq. (29) are developed according to the range
560 covered by the experimental test (Table 1). Thus, for working pressures ranging from 0.7 to
561 1.8 kPa, and for dimensionless numbers which vary according to the ranges listed in Table 8.
562 The heat fluxes and superheats that allowed the determination of the experimental HTC (Eq.
563 18) are between 1352-14507 W. m⁻² and 1.12-7.14 K respectively.

Table 8 : Dimensionless numbers range of variation

(a) Pool boiling	
Dimensionless numbers	Validity range
Nusselt number, Nu	$2.751 \leq Nu \leq 21.118$
Form factor, F^*	$0.007 \leq F^* \leq 0.106$
Vapor density ratio, ρ_v^*	$1.259 \leq \rho_v^* \leq 2.507$
Capillary number, Ca	$0.001 \leq Ca \leq 0.013$
Bond number, Bo	$0.722 \leq Bo \leq 2.192$
Weber number, We	$0.164 \leq We \leq 16.948$
(b) Liquid film evaporation	
Nusselt number, Nu	$0.064 \leq Nu \leq 0.530$
Form factor, F^*	$0.007 \leq F^* \leq 0.106$
Vapor density ratio, ρ_v^*	$1.332 \leq \rho_v^* \leq 2.524$
Capillary number, Ca	$0.001 \leq Ca \leq 0.016$
Modified Bond number, Bo_f	$31.111 \leq Bo_f \leq 117.917$
Weber number, We	$0.002 \leq We \leq 0.479$

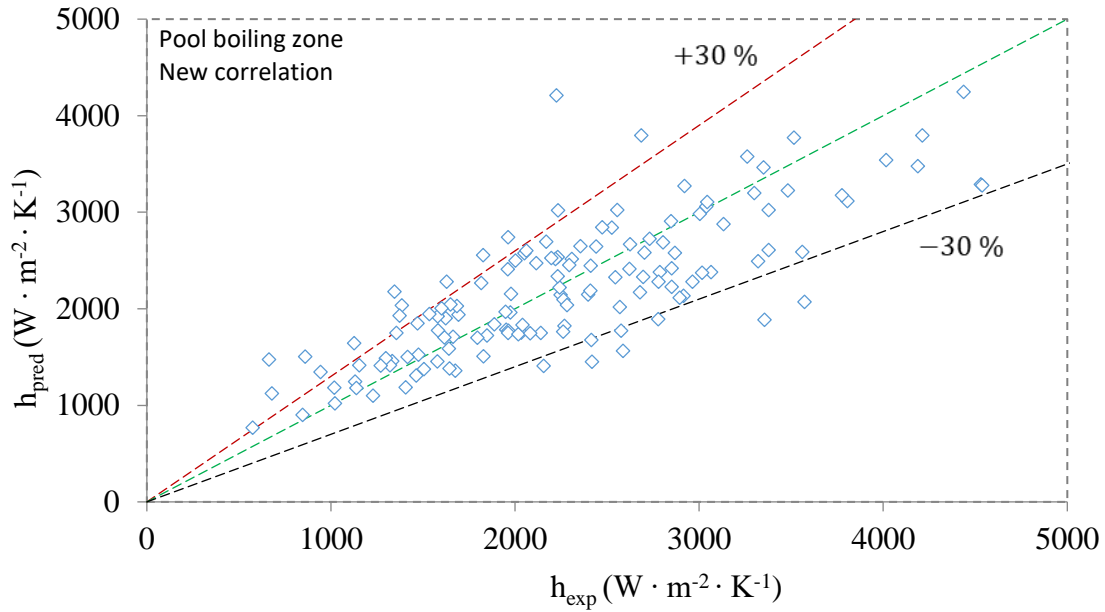
564

565 Eq. (28) and (29) show opposite sign exponents for the dimensionless group $\frac{e}{H_1}$ in one hand
566 (positive sign for the pool boiling zone, negative sign for the liquid film evaporation zone) and
567 $\frac{e}{\delta_{f/b}}$ on the other hand (negative sign for the pool boiling zone, positive sign for the liquid film
568 evaporation zone). Although the obtained values of the exponents have no physical meaning
569 since they are obtained by a mathematical fitting, the obtained opposite sign could be expected.
570 Indeed, for a given thickness of the narrow channel, low H_1 leads to relative high bubble
571 frequency. Since the bubble frequency is high, the liquid film created by these bubbles is often
572 regenerated leading to thicker falling film. On the contrary, high H_1 leads to low bubble event
573 and the falling film could get thinner before being regenerated again. Thus, the estimated HTC
574 is expected to be higher in the pool boiling zone for low H_1 than for high H_1 values, whereas in
575 the falling film evaporation zone, the reverse trend is expected: a higher HTC is expected for
576 high H_1 than low H_1 . Regarding the dimensionless group $\frac{e}{\delta_{f/b}}$, the physical meaning is different
577 regarding the considered zone. Indeed, in the pool boiling zone, δ_b refers to the capillary length
578 whereas in the liquid film zone δ_f refers to the visco-gravity length. Nevertheless, the obtained
579 sign could be explained by the fact that, for a given running conditions, a lower e leads to a
580 warmer environment fostering boiling in on hand whereas it increases friction and refrains the
581 falling of the liquid film in the other hand.

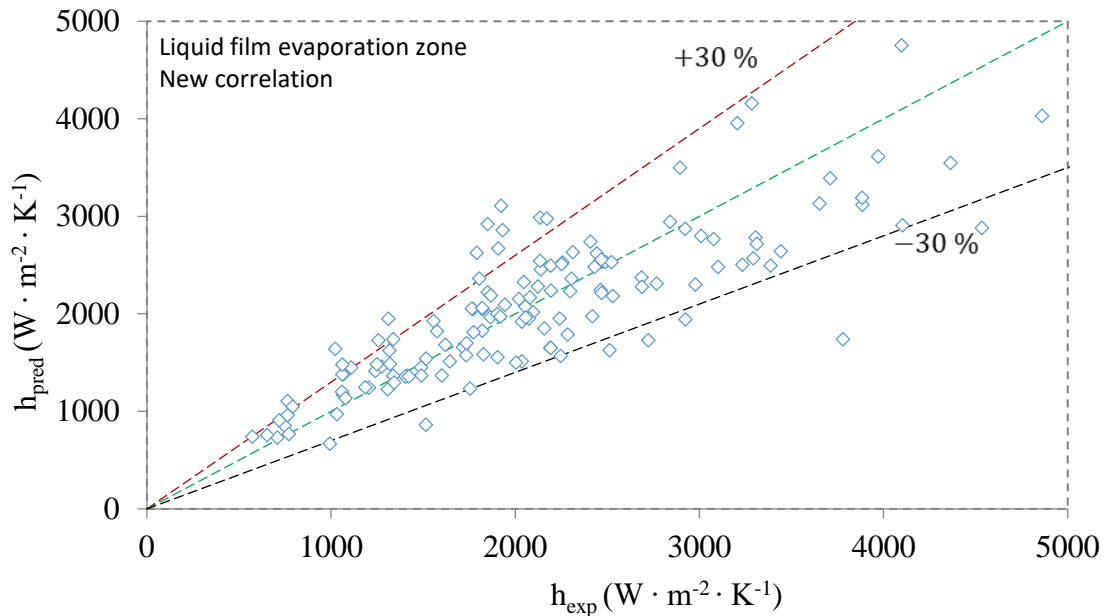
582

583 6. Evaluation and validation of the developed correlations

584 Fig. 7 a and b reflect the graphical comparisons between the experimental HTC and the
585 HTC calculated by the correlations (Eq. 28) and (Eq. 29) (named thereafter “new correlation”).



(a)



(b)

Figure 7 : Comparison between the experimental HTC (h_{exp}) and the HTC calculated (h_{pred}) by the new correlations: (a) pool boiling zone, (b) liquid film evaporation zone.

586 The quantitative indications of the performance of the developed correlations (Eq. 28
 587 and Eq. 29) are the following: they predict 86 % (MRE = 3 % and MAE = 19 %) and 83 %
 588 (MRE = 2 % and MAE = 18 %), respectively, of the experimental database within an error band
 589 of $\pm 30\%$. The correlation coefficients of Eq. 28 and Eq. 29 are respectively equal to 0.79 and
 590 0.84 and the shape of the scatterplots (Fig. 7 a and b) show that the experimental and predicted
 591 heat transfer coefficients are strongly linearly correlated. Thus, they very satisfactory results.
 592 From a physical point of view, the new models involve different dimensionless numbers of
 593 great importance for the sizing of two-phase heat exchangers (Table 6). For example, the form

594 factor (e/H_1) should help in finding the right filling ratio and channel thickness to obtain the
 595 best system performance. They are then made up of dimensionless numbers that have a physical
 596 meaning and relevance to the phenomenon being studied. However, a predictive correlation
 597 must not only have reasonable statistical similarity to the experimental results, but it must also
 598 capture trends in the experimental data. Thus, Fig. 8 represents the variations of the calculated
 599 HTC (h_{pred}) by the newly proposed correlations as a function of the heat fluxes (\dot{q}), at fixed pressures (P_{sat}).
 600

601

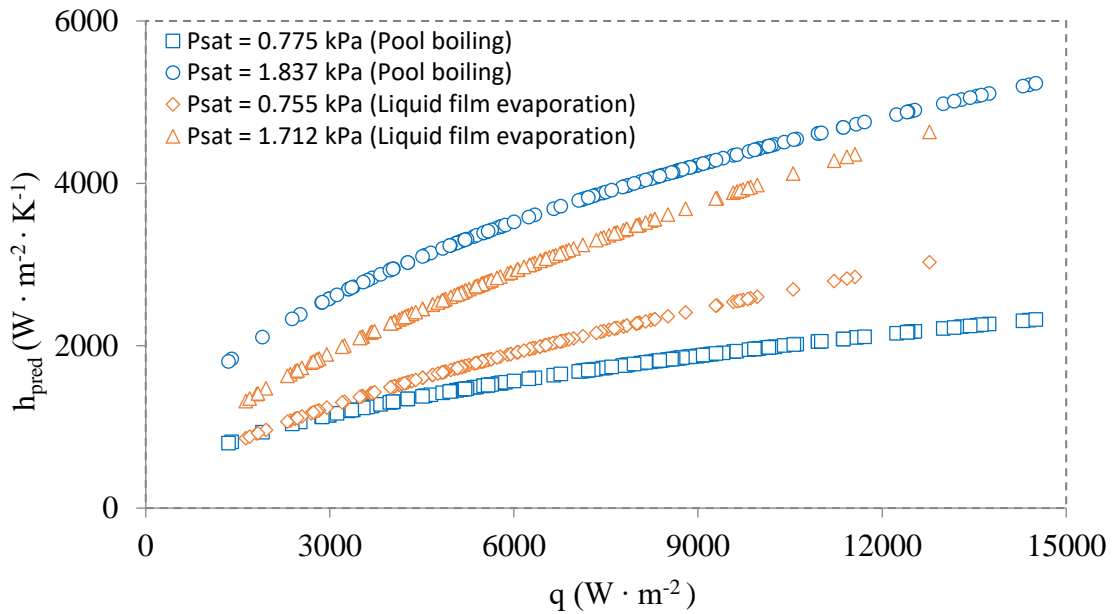


Figure 8 : Variations of HTC predicted by the proposed new correlations as a function of heat flux

602

603 According to Fig. 8, at each given P_{sat} , h_{pred} increases with heat flux (q). That is, when
 604 q and/or P_{sat} increase, h_{pred} increases accordingly. In other words, at subatmospheric pressure,
 605 when the heat flux is constant, the water pool boiling or liquid film evaporation HTC increases
 606 with pressure. This trend is consistent with studies in the literature done at subatmospheric or
 607 atmospheric pressure [13] [11] [20] [27]. For given conditions (constant P_{sat}), these said trends
 608 are easily analyzed with (Eq. 18) which allows to determine the experimental heat transfer
 609 coefficients. Thus, the proposed correlations correctly capture the relationships between HTC
 610 (h), working pressure (P_{sat}) and heat flux (\dot{q}) verified extensively in the literature, for different
 611 pressure and/or temperature conditions.

612 The constants and coefficients of the correlations (Eq. 28) and (Eq. 29) are specific to
 613 the evaporator studied (Fig.2) and to the domain covered by the experimental program
 614 described in subsection 2.3. In order to extend their areas of validity, they are evaluated by
 615 fitting their constants and coefficients with different experimental data obtained at low pressure
 616 by other authors (Table 9).

617

618

619

620

Table 9 : Constants and coefficients fitted to experimental data from the literature

Data base	Constants and coefficients fitted						Areas
	C	a ₁	a ₂	a ₃	a ₄	a ₅	
Zajaczkowski et al. [11]	$1.839 \cdot 10^{-3}$	0,111	-0.798	-0.405	-1.543	0.988	Pooling boiling region
Chang et al. [13]	$2.299 \cdot 10^{-3}$	0.111	-0.798	-0.405	-1.543	0.988	Pooling boiling region
Chang et al. [13]	$4.818 \cdot 10^{-6}$	-0.385	-0.871	0.756	-2.002	1.406	Intermittent region

621

622

623

The experimental data from Zajaczkowski et al. [11] and Chang et al. [13] are compared with the corresponding values estimated by the new correlations (Table 9), as shown in Fig.9.

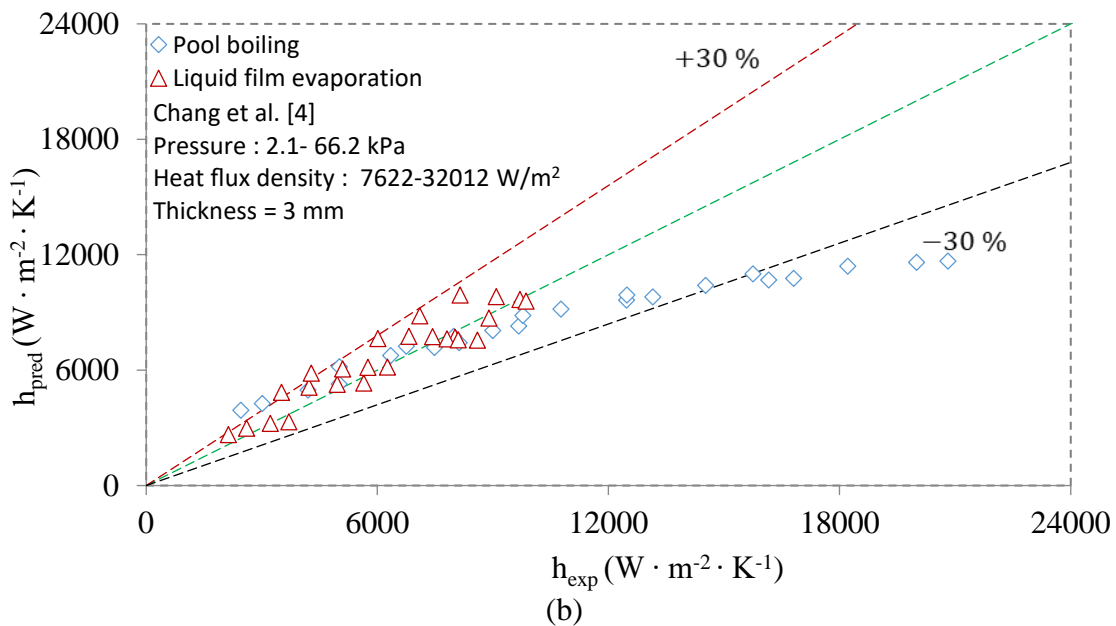
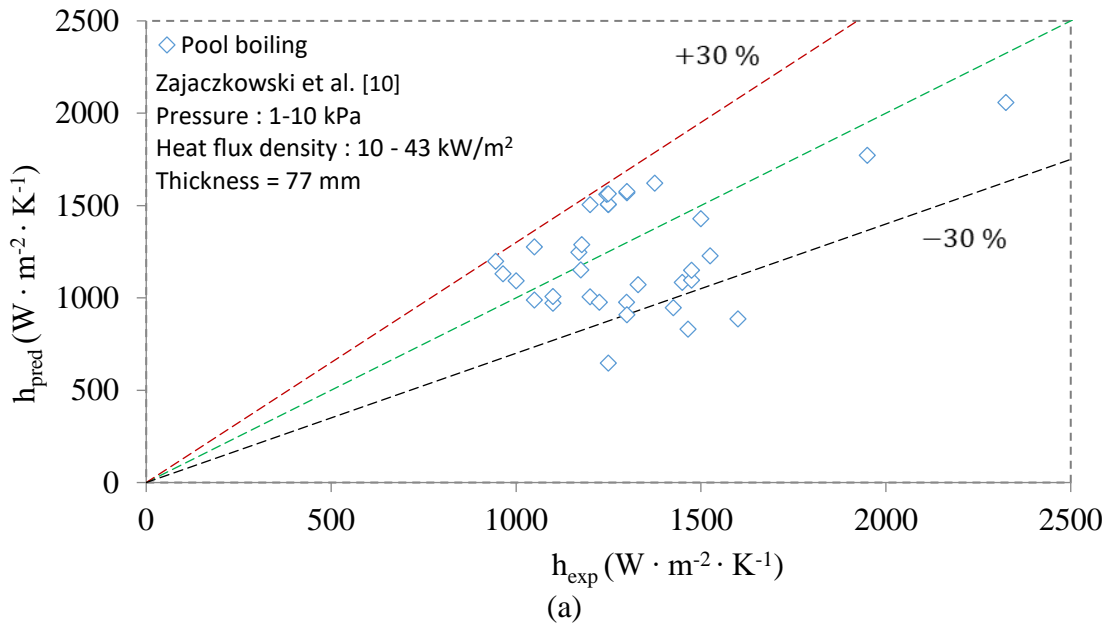


Figure 9 : Comparison between experimental HTC (h_{exp}) from the literature and HTC calculated (h_{pred}) by the new correlations

624 According to Fig.9, the developed pool boiling correlation predicts 71% (with MRE =-1%,
 625 MAE=5%) and 85% (with MRE = -2%, MAE = 4%) of the respective Zajaczkowski et al. [11]
 626 and Chang et al. [13] experimental databases. As for the liquid film evaporation correlation, it
 627 predicts 92% (with MRE =2%, MAE = 2%) of the experimental data from Chang et al. [13]
 628 (Intermittent region). Thus, from these results, it can be inferred that the dimensionless numbers
 629 found in this work are relevant for correlating the pool boiling and liquid film evaporation HTC
 630 at low pressure.

631 It is believed that the proposed correlations could then be used more widely. By keeping
 632 the dimensionless numbers and fitting them with more data one would be able to predict all the
 633 HTC of water at subatmospheric pressure and at higher pressure since these numbers consider
 634 many relevant parameters. As a result, these new correlations and/or the dimensionless numbers
 635 used should allow the design of various two-phase evaporators.

636

637 7. Conclusion

638 Raw experimental data, obtained in order to study heat transfer phenomena during the
 639 vaporization of water at pressures ranging from 0.7 to 1.7 kPa in a vertical evaporator channel
 640 with smooth plates of standard size (0.2 m wide x 0.5 m high) included inside a thermosyphon
 641 loop mimicking a sorption system are processed and exploited.

642 A graphical and statistical comparison between the HTC from the experimental database
 643 and the HTC estimated by empirical correlations from the literature is made. Among the
 644 correlations used, only those of Dittus-Boelter (Eq.1), Kruzylin (Eq. 3) and Chang (Eq. 6 and
 645 Eq. 7) gave good statistical results but still, show poor performance (Table 10).

Table 10: Results obtained from the statistical comparison between the experimental HTC and the HTC predicted by Dittus-Boelter (Eq.1), Kruzylin (Eq. 3) and Chang (Eq. 6 and Eq. 7) correlations.

Pool boiling zone				
Correlations	MAE [%]	MRE [%]	τ_{30} [%]	r [-]
Dittus-Boelter [19]	24	7	73	0.67
Kruzylin [11]	28	- 26	54	0.64
Chang [13]	29	8	69	0.57
Liquid film evaporation zone				
Correlations	MAE [%]	MRE [%]	τ_{30} [%]	r
Dittus-Boelter [19]	27	12	68	0.72
Chang [13]	40	13	54	0.50

646

647 Based on a rigorous dimensional analysis, two news methodical correlations to predict the
 648 pool boiling and liquid film evaporation HTC are developed by fitting the experimental
 649 database by mean of the least squares method. Unlike the empirical correlations in the literature,
 650 these new models consider among others the thermophysical properties of the fluid (saturation
 651 pressure, hydrostatic pressure), the heat flow, the parietal superheating, the filling height of the
 652 channel and the geometrical parameters (boundary conditions). They therefore involve various

653 dimensionless numbers of great importance for the dimensioning of two-phase heat exchangers
 654 and the optimization of their design like for example, the form factor (e/H_1), which it is
 655 believed to help finding the correct filling ratio and channel thickness to obtain the best system
 656 performance. Moreover, it is believed that the proposed correlations could then be used more
 657 widely by keeping the dimensionless numbers and fitting them with more data. Currently, the
 658 new correlations predict 86% (with MAE = 19%, MRE = 3%, $r = 0.79$) and 83% (with MAE
 659 = 18%, MRE = 2%, $r = 0.84$) of the experimental data in τ_{30} respectively.

660 To extend their validity domains, the new correlations are evaluated by fitting the
 661 constants and coefficients with experimental data from Zajackowski et al. [11] and Chang et
 662 al. [13] (Table 9). The pool boiling correlation correctly predicts 71% (with MRE = -1%, MAE
 663 = 5%) and 85% (with MRE = -2%, MAE = 4%) of the databases of respective authors. As for
 664 the liquid film evaporation correlation, it predicts 92% (with MRE = 2%, MAE = 2%) of the
 665 experimental data from [13] (Intermittent region). Thus, from these satisfactory results, the new
 666 proposed models constitute a first step towards the development of tools to assist in the sizing
 667 of two-phase heat exchangers and the optimization of their operation.

668 In order to allow a generalization and to better understand the impact of all the
 669 parameters taken into account, experimental test campaigns with a wider range of pressures are
 670 envisaged in order to make the correlations more robust and to allow the insertion of
 671 dimensionless numbers removed from (Eq. 26) due to their small variation. Research on the
 672 decisive effect that non-condensable gases such as air can have on heat transfer by vaporization
 673 at sub-atmospheric pressure also seems necessary. Tests will be also performed with other
 674 fluids.

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 678 with Senegal (Service de Coopération et d'Action Culturelle (SCAC) of the French Embassy in
 679 Dakar). This project focuses on compact plate evaporators using water as refrigerant in order
 680 to properly size and design them.

681 Appendix A: Dimensional Analysis Approach

682 The main step of the dimensional analysis is the choice of fundamental quantities, i.e., the
 683 quantities according to which the dimensions of the independent physical parameters governing
 684 the studied phenomena are expressed. In the international system, the fundamental quantities
 685 are the mass M, the length L and the time T. In heat transfer problems, as in the case of this
 686 study, it is necessary to add a fourth dimension Θ . To go back to the dimensionless numbers
 687 that characterize the physical problem under study, this requires the free choice of the repeated
 688 variables, provided that they are dimensionally independent and that their dimensions cover all
 689 the fundamental dimensions (the determinant of the matrix is non-zero) [18]. The repeated
 690 variables ($e, \mu_l, \rho_l, c_{p_l}$) and the non-repeated variables ($H_1, \mu_v, \rho_v, \rho_b, \Delta h_{lv}, \sigma, g, \lambda_l$) form the
 691 matrix with dimensions D (Table A.1). The repeated variables form the central matrix C (blue
 692 background) and the unrepeated variables form the residual matrix R (grey background).

Table A.1 : Matrix with dimensions D.

	h	H_1	μ_v	ρ_v	ρ_b	G_v	σ	g	λ_l	e	μ_l	ρ_l	c_{p_l}
M	1	0	1	1	1	1	1	0	1	0	1	1	0
L	0	1	-1	-3	-3	-2	0	1	1	1	-1	-3	2
T	-3	0	-1	0	0	-1	-2	-2	-3	0	-1	0	-2

$$\Theta \quad -1 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad -1 \quad 0 \quad 0 \quad 0 \quad 1$$

693

694 In order to find the dimensionless groupings, linear combinations between the rows of the
 695 matrix with dimensions D (Table A.1) to transform the central matrix C into an identity matrix
 696 are made. These operations lead to the matrix with modified dimensions Dm (Table A.2) an
 697 identity matrix (modified central matrix Cm), the dimensionless numbers characterizing the
 698 physical phenomenon can be written, based on the coefficients contained in the columns of the
 699 new residual matrix R (modified residual matrix Rm). Each non-repeated variable represented
 700 by G_i (i.e. each column of the modified residual matrix) forms a dimensionless number π_i (Eq.
 701 A.1) [18].

Table A.2 : Matrix with modified dimensions Dm

h	H_l	μ_v	ρ_v	ρ_b	G_v	σ	g	λ_l	e	μ_l	ρ_l	c_{p_l}
-1	1	0	0	0	-1	-1	-3	0	1	0	0	0
1	0	1	0	0	1	2	2	1	0	1	0	0
0	0	0	1	1	0	-1	-2	0	0	0	1	0
1	0	0	0	0	0	0	0	1	0	0	0	1

702

$$\pi_i = \frac{G_i}{e^{a_{i1}} \mu_l^{a_{i2}} \rho_l^{a_{i3}} c_{pl}^{a_{i4}}} \quad (A.1)$$

703

704 The process relationship (Eq.24) can now be expressed in dimensionless form by the
 705 Eq. (A.2).

706

$$\pi_h = \frac{h e}{\mu_l c_p} = F \left(\begin{array}{l} \pi_{H_l} = \frac{H_l}{e}; \pi_{\mu_v} = \frac{\mu_v}{\mu_l}; \pi_{\rho_v} = \frac{\rho_v}{\rho_l}; \pi_{\rho_b} = \frac{\rho_b}{\rho_l}; \\ \pi_{G_v} = \frac{G_v e}{\mu_l}; \pi_{\sigma} = \frac{\sigma e \rho_l}{\mu_l^2}; \\ \pi_g = \frac{\rho_l^2 g e^3}{\mu_l^2}; \pi_{\lambda_l} = \frac{\lambda_l}{\mu_l c_{p_l}} \end{array} \right) \quad (A.2)$$

707

708 The space of the dimensionless groupings (π_i) obtained Eq. (A.2) is modified in order
 709 to simplify the interpretation of the dimensionless numbers and to try to understand the physical
 710 phenomena involved in the process of boiling and evaporation at subatmospheric pressure in a
 711 PHE. These rearrangements, which consist of raising some to different powers, multiplying or
 712 dividing them, adding or subtracting them, make it possible to go from Eq. (A.2) to (Eq.25).

713 References

714

- [1] K. Amir A. and M. Norbert , "Comparing water (R718) to other refrigerants".ASME
International Mechanical Engineering Congress and Exposition, December 14, 2007,
[https://doi.org/10.1115/IMECE2006-13341.](https://doi.org/10.1115/IMECE2006-13341)

- [2] J. Seiler, R. Volmer, D. Krakau, J. Pöhls, F. Ossenkopp, L. Schnabel and A. Bardow, "Capillary-assisted evaporation of water from finned tubes – Impacts of experimental setups and dynamics," *Applied Thermal Engineering* 165 (2020) 114620. <https://doi.org/10.1016/j.applthermaleng.2019.114620>.
- [3] K. Amir A. and M. Norbert , "Comparing water (R718) to other refrigerants".*ASME International Mechanical Engineering Congress and Exposition, December 14, 2007*, <https://doi.org/10.1115/IMECE2006-13341>.
- [4] F. Giraud, "Vaporization of water at subatmospheric pressure:fundamentals of boiling phenomena and path towards the design of compact evaporators for sorption chillers," Thermique et Energétique, Ph.D. Thesis, L'institut National des Sciences Appliquées de Lyon, Lyon, France, 2015.
- [5] F. Giraud, C. Toublanc, R. Rullière, J. Bonjour and . M. Clause, "Experimental study of water vaporization occurring inside a channel of a smooth plate-type heat exchanger at subatmospheric pressure," *Applied Thermal Engineering* 106 (2016) 180-191. <http://dx.doi.org/10.1016/j.applthermaleng.2016.05.151>.
- [6] S. Michaie, R. Rullière and J. Bonjour, "Towards a more generalized understanding of pool boiling at low pressure : Bubble dynamics for two fluids in states of thermodynamic similarity," *Experimental Thermal and Fluid Science* 101 (2019) 217-230. <https://doi.org/10.1016/j.expthermflusci.2018.10.009>.
- [7] D. Triché, "Étude numérique et expérimentale des transferts couplés de masse et de chaleur dans l'absorbeur d'une machine à absorption ammoniac-eau," Thèse, Mécanique des fluides, Énergétique, Procédés, Université Grenoble Alpes, 2016, 199.
- [8] F. A. Mota, E. P. Carvalho and M. A. Ravagnani, "Chapter 7 : Modeling and Design of Plate Heat Exchanger," in *Heat Transfer Studies and Applications 165 (2012) 199*, *Janeza Trdine 9, 51000 Rijeka, Croatia, InTech*.
- [9] F. Tâboas, M. Vallès, M. Bourouis and A. Coronas, "Assessment of boiling heat transfer and pressure drop correlations of ammonia/water mixture in a plate heat exchanger," *International journal of refrigeration* 35 (2012) 633-644. [doi:10.1016/j.ijrefrig.2011.10.003](https://doi.org/10.1016/j.ijrefrig.2011.10.003).
- [10] Z. H. Ayub, T. S. Khan, S. Salam, K. Nawaz, A. H. Ayub and M. S. Khan, "Literature survey and a universal evaporation correlation for plate type heat exchangers," *International Journal of Refrigeration* 99 (2019) 408-418. <https://doi.org/10.1016/j.ijrefrig.2018.09.008>.
- [11] B. Zajackowski, T. Halon and Z. Krolicki, "Experimental verification of heat transfer coefficient for nucleate boiling at sub-atmospheric pressure and small heat fluxes," *Heat Mass Transfer*, 52 (2016) 205–215. <https://doi.org/10.1007/s00231-015-1549-8>.
- [12] F. Giraud, R. Rullière, C. Toublanc, M. Clause and J. Bonjour, "Experimental evidence of a new regime for boiling of water at subatmospheric pressure," *Experimental*

Thermal and Fluid Science 60 (2015) 45-53.
<http://dx.doi.org/10.1016/j.expthermflusci.2014.07.011>.

- [13] S. W. Chang, D. C. Lo, K. F. Chiang and C. Y. Lin, "Sub-atmospheric boiling heat transfer and thermal performance of two-phase loop thermosyphon," *Experimental Thermal and Fluid Science* 39 (2012) 134-147.
[doi:10.1016/j.expthermflusci.2012.01.017](https://doi.org/10.1016/j.expthermflusci.2012.01.017).
- [14] L. Schnabel, C. Scherr and C. Weber, "Water as refrigerant – Experimental evaluation of boiling characteristics at low temperatures and pressures". *VII Minsk International Seminar "Heat Pipes, Heat Pumps, Refrigerators, Power Sources"*, Minsk, Belarus, September 8-11, 2008.
- [15] F. Giraud, R. Rullière, C. Toublanc, M. Clause and J. Bonjour, "Subatmospheric pressure boiling on a single nucleation site in narrow vertical spaces," *International Journal of Heat and Fluid Flow* 58 (2016) 1-10.
<http://dx.doi.org/10.1016/j.ijheatfluidflow.2015.12.002>.
- [16] F. Giraud and B. Tréméac, "Influences of confinement on subatmospheric water vaporization phenomena in a vertical rectangular channel," *International Journal of Heat and Mass Transfer* 145 (2019) 118725.
- [17] F. Giraud, P. Vallon and B. Tremeac, "Experimental study of water vaporization occurring inside the channel of a smooth-plate type heat exchanger connected to an adsorber and comparison with trends observed in absorption configuration". *International Journal of of refrigeration* 77 (2017) 60-74,
<http://dx.doi.org/10.1016/j.ijrefrig.2017.02.021>.
- [18] G. Delaplace, K. Loubière, F. Ducept and R. Jeantet, *Modélisation en génie des procédés par analyse dimensionnelle. Méthode et exemples résolus*, Paris: Editions Lavoisier TEC et DOC, 2014.
- [19] S. Bengt, "Chapter Two - Advanced heat transfer topics in complex duct flows, *Advances in Heat Transfer*, 49 (2017) 37-89,
<https://doi.org/10.1016/bs.aiht.2017.09.001>".
- [20] L.-H. Yu , S.-X. Xu , G.-Y. Ma and J. Wang, "Experimental research on water boiling heat transfer on horizontal copper rod surface at sub-Atmospheric pressure," *Energies* 8 (2015) 10141-10152; [doi:10.3390/en80910141](https://doi.org/10.3390/en80910141).
- [21] T. Baki , "Boiling pure fluids at sub atmospheric pressures," *International Journal of Mathematics and Computers in Simulation*, 14 (2020) 87-91, DOI: [10.46300/9102.2020.14.12](https://doi.org/10.46300/9102.2020.14.12).
- [22] S. A. Alavi Fazel and S. Roumana, "Pool boiling heat transfer to pure liquids," *Continuum Mechanics, Fluids, Heat*, n° ISBN : 978-960-474-158-8 (2010) 211-216.
- [23] K. Zhang, Y. Hou, W. Tian, Y. Fan, G. Su and S. Qiu, "Experimental investigations on single-phase convection and steam-water two-phase flow boiling in a vertical rod

bundle, *Experimental Thermal and Fluid Science*, 80 (2017) 147–154.
<https://doi.org/10.1016/j.expthermflusci.2016.08.018>".

- [24] F. Giraud, R. Rullière, C. Toublanc, M. Clausse and J. Bonjour, "Experimental evidence of a new regime for boiling of water at subatmospheric pressure," *Experimental Thermal and Fluid Science* 60 (2015) 45-53.
<http://dx.doi.org/10.1016/j.expthermflusci.2014.07.011>.
- [25] J. Bonjour, F. Boulanger, D. Gentile and M. Lallemand, "Etude phénoménologique de l'ébullition en espace confiné à partir d'un site de nucléation isolé," *Rev. Gén. Therm.* 36 (1997) 562-572..
- [26] R. L. Amalfi, F. Vakili-Farahani and J. R. Thome , "Flow boiling and frictional pressure gradients in plate heat exchangers: part 2, comparison of literature methods to database and new prediction methods," *International Journal of Refrigeration* 61 (2016) 185-203. <http://dx.doi.org/10.1016/j.ijrefrig.2015.07.009>.
- [27] I. S. Kiyomura, T. S. Mogaji, L. L. Manetti and M. E. Cardoso, "A predictive model for confined and unconfined nucleate boiling heat transfer coefficient," *Applied Thermal Engineering* 127 (2017) 1274–1284;
<http://dx.doi.org/10.1016/j.applthermaleng.2017.08.135>.

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