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R32 Partial Condensation Inside A Brazed Plate Heat Exchanger

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ABSTRACT

It is well known that when a superheated gas reaches a cold surface, the condensation starts immediately only if the wall temperature is lower than the dew temperature of the refrigerant; in this case, the heat transfer phenomenon can be also affected by the superheating temperature. This paper presents the experimental measurements of the heat transfer coefficient carried out during partial condensation of superheated R32 refrigerant in a commercial brazed plate heat exchanger prototype. The present work aims at highlighting the effects of the superheating during the partial condensation of R32 by imposing 5, 10, 15, and 25 K of vapour superheating at the inlet of the brazed plate heat exchanger prototype. The experimental measurements were carried out by varying the specific mass velocity between 15 and 40 kg m⁻² s⁻¹ and the outlet vapour quality between 0.0 and 0.65. The experimental data were collected at around 36.5 °C saturation temperature (saturation pressure of 2.27 MPa). The present data is used to validate a new step-by-step model for the calculation of the heat transfer coefficient, which accounts for the different superheating inlet temperatures.

1. INTRODUCTION

Superheated vapour condensation occurs in every inverse cycle machine where the refrigerant vapour, coming from the compressor, at the inlet of the condenser exhibits some degrees of superheating, usually between 10 and 40 °C but sometimes even greater than 50 °C. Even if this heat transfer process is largely used in the condensers of refrigerating and heat pump systems, the physical phenomenon appears to be an open research issue; moreover studies of the condensation of superheated vapour are rare in the open literature.

Direct condensation of superheated vapour on a cold surface instantly occurs when the surface temperature is lower than the dew temperature of the refrigerant but several authors suggest different interpretations of the basic physics of this phenomenon.

The superheated vapour condensation has been analytically studied since the early 1960s, Minkowycz and Sparrow (1966, 1969) investigated the effect of superheating on steam condensation heat transfer in laminar and forced convection on a flat plate. The analyses conducted by the authors reveal that the effect of the superheating is to increase the heat transfer but, only for small saturation to wall temperature difference, this enhancement is large enough to be of marked practical interest.

Shang and Wang (1997) analytically investigated the effect of superheating on heat and mass transfer of laminar film condensation of superheated steam on a vertical isothermal plate. The authors obtained practically simple correlations for predicting the heat transfer and mass flow rate during condensation of saturated steam.

Mitrovic (2000) studied the effects of vapour superheat, condensate subcooling, gravity interfacial shear, and vapour convection on the basis of the Nusselt theory (1919). The results show that the vapour superheat affects the condensation kinetics in cooperation with heat transfer in both phases; under comparable conditions, the condensate film is thinner and the heat transfer coefficient larger for superheated than for saturated vapour.

Experimental investigations have been conducted by Longo (2008, 2009) who presented the heat transfer coefficients and pressure drops for saturated vapour (vapour quality at inlet 0.95-1, at outlet 0.00-0.06) and superheated vapour (10°C superheating) condensation of R134a and R410A inside a brazed plate heat exchanger

(BPHE). The heat transfer coefficients of superheated vapour are 8-10% higher than those of saturated vapour under the same refrigerant mass velocity both for R134a and R410A.

More recently, Longo (2011) studied the effect of vapour superheating on hydrocarbons: R600a (Isobutane), R290 (Propane) and R1270 (Propylene) condensation inside a brazed plate heat exchanger. Again, the superheated vapour condensation heat transfer coefficients at $\Delta T_{\text{sup}}=10$ K are higher by 5-10 % as compared to those measured during saturated vapour condensation.

As reported in this brief introduction, even if the laminar and forced convective condensation heat transfer on vertical plate has been analytically studied; there is a lack of experimental works concerning superheated vapour condensation inside brazed plate heat exchangers.

This work addresses the effect of the superheating by investigating the partial condensation of R32 inside a BPHE prototype at four different superheating values (5, 10, 15, and 25 K) at the inlet of the test section. A new model for the calculation of the heat transfer coefficient has been implemented in a step-by-step procedure recently proposed by Mancin et al. (2012), which permits to simulate the phase change process accounting for the different superheating at the inlet of the plate heat exchanger.

2. EXPERIMENTAL TEST SETUP AND OPERATING CONDITIONS

The experimental set up is located at the two-phase heat transfer laboratory at the Dipartimento di Ingegneria Industriale (building of Fisica Tecnica) of the University of Padova. This test rig has been upgraded to allow two-phase heat transfer measurements on BPHEs. As it appears from Figure 1, the experimental facility consists of three loops: the refrigerant loop, the cooling water loop and the hot water loop. In the first loop the refrigerant passes through two tube-in-tube heat exchangers, the boiler and the superheater, fed with hot water. Superheated vapour enters the BPHE at a known mass velocity, temperature and pressure and then it is condensed against the cold water, which flows countercurrently. The refrigerant can be completely condensed and subcooled or it can be only partially condensed. The subcooled liquid or the two-phase mixture leaves the test section and goes to a post-condenser, a brazed plate type condenser, where it is eventually fully condensed and further subcooled. The subcooled liquid is passed through a drier and then sent back to the boiler by a magnetically coupled gear pump. A bladder accumulator connected to a nitrogen bottle and a pressure regulator is installed in the refrigerant loop.

Figure 1 Schematic of the experimental test facility.

The specific enthalpy at the inlet of the test section is determined from the measured values of pressure and temperature. The heat transfer rate in the test section can be varied by adjusting the temperature and flow rate of the cooling water. Thus mass velocity and vapour quality change can be independently controlled.

The boiler and superheater are both coaxial tube-in-tube heat exchangers, in which the refrigerant flowing in the tube side is vaporized and then heated by hot water flowing in the annulus side. In the hot water circuit, an electrical heater supplies the water with the power transferred to the refrigerant. Two secondary water circuits control the temperature of the cooling water entering the test section. An electrical heater provides hot water while a chiller

provides cold water. The cold and hot circuit flows are mixed together in a storage tank, where the water temperature is maintained constant. Thermostatic control of the water is accomplished by adjusting the electrical power supplied to the heater.

Table 1 Operating test conditions

<i>Parameter</i>	<i>Mean Value or Range</i>
Refrigerant	R32
Dew temperature [°C]	36.5
Saturation pressure [MPa]	2.27
Outlet Vapour Quality, x_{out}	0.0 - 0.65
Superheating [K]	5 - 10 - 15 - 25
Refrigerant mass velocity [$\text{kg m}^{-2} \text{s}^{-1}$]	15 - 40
Water mass velocity [$\text{kg m}^{-2} \text{s}^{-1}$]	140 - 170
Water temperature gain [K]	2.5-10.9

The tests reported in the present paper aim at investigating the effect of the superheating temperature on the heat transfer coefficient at different mass velocities and different outlet vapour quality. As listed in Table 1, the experimental measurements have been carried out imposing 5, 10, 15 and 25 K of superheating at the inlet of the test section and by varying the refrigerant mass velocity from 15 to 40 $\text{kg m}^{-2} \text{s}^{-1}$ and the outlet vapour quality from 0.0 to 0.65. Data have been collected at 36.5 °C of saturation temperature, which can be considered as a typical condensation level for heat pump applications.

The tested prototype consists of six plates; therefore, the refrigerant flows into two channels while the water flows into three; there are four thermal plates, which present a macro-scale herringbone corrugation with an inclination angle $\beta=65^\circ$. The most important geometrical characteristics of the heat exchanger are listed in Table 2 where a sketch of the BPHE is also drawn.

Table 2 Geometrical characteristics of the BPHE prototype.

<i>Geometrical Characteristics</i>	<i>Value</i>
Plate Length, L [mm]	526
Plate Width, W [mm]	111
Number of Plates [-]	6
Number of Refrigerant Channels [-]	2
Number of Water Channels [-]	3
Number of Heat Transfer Plates [-]	4
Plate Thermal Conductivity [$\text{W m}^{-1} \text{K}^{-1}$]	16

3. HEAT TRANSFER DATA REDUCTION

In the heat exchanger the refrigerant condenses by flowing in counter-current with the cold water; the heat flow rate exchanged in the BPHE is obtained from a thermal balance on the cooling water side as given by:

$$q = \dot{m}_w \cdot c_{p,w} \cdot (t_{w,out} - t_{w,in}) \quad (1)$$

The vapour quality exiting the test section (x_{out}) can be calculated from the heat balance as described by Eq. (1). In particular, the enthalpy at the exit of the heat exchanger, $h_{ref,out}$:

$$h_{ref,out} = h_{ref,in} - \frac{q}{\dot{m}_{ref}} \quad (2)$$

where $h_{ref,in}$ is the enthalpy of the superheated vapour at the inlet of the BPHE obtained from the measured values of pressure $p_{ref,in}$ and temperature $t_{ref,in}$. The thermodynamic state at the outlet of the brazed plate heat exchanger is

completely defined by the values of the enthalpy of the two-phase mixture $h_{ref,out}$ and the outlet refrigerant pressure $p_{ref,out}$. The latter is obtained from the measured values of refrigerant inlet pressure $p_{ref,in}$ and pressure drop Δp_{ref} , as:

$$p_{ref,out} = p_{ref,in} - \Delta p_{ref} \quad (3)$$

Thus, the vapour quality is estimated by the following equation:

$$x_{out} = \frac{h_{ref,out} - h_L}{h_{LG}} \quad (4)$$

where h_L and h_{LG} are the saturated liquid enthalpy and the isobaric latent heat, respectively, both estimated at the outlet refrigerant pressure $p_{ref,out}$. All the thermodynamic and transport properties of the refrigerants have been estimated using RefProp 8.0 (2007). Considering the condensation heat transfer coefficient, if the wall temperature at the refrigerant inlet is lower than the local dew temperature, then vapour condensation from superheated vapour occurs. Therefore, the overall condensation heat transfer coefficient can be estimated considering the entire exchanged heat flow rate, as follows:

$$K = \frac{q}{A \cdot (\Delta t_{ML})_{CC} \cdot F_t} \quad (5)$$

A is effective the heat transfer area of the embossed herringbone while $(\Delta t_{ML})_{CC}$ is the logarithmic mean temperature difference, which is calculated using the measured values of $p_{ref,in}$, $t_{ref,out}$, $t_{w,in}$, $t_{w,out}$ as:

$$\begin{aligned} \Delta_1 &= t_{DEW}(p_{ref,in}) - t_{w,out} & \Delta_2 &= t_{ref,out} - t_{w,in} \\ (\Delta t_{ML})_{CC} &= \frac{\Delta_1 - \Delta_2}{\ln\left(\frac{\Delta_1}{\Delta_2}\right)} \end{aligned} \quad (6)$$

Finally, F_t is the Correction Factor, which accounts for the thermal effects of the first and last channel of the brazed plate prototype. In BPHEs, the two outer plates serve as ends plates and ideally do not transfer any heat. This means that the channel at both ends of the BPHE are heated or cooled by one side only, and the fluid experiences a smaller temperature change than in the other channels. Therefore, the ends plates have a significant influence on the overall thermal performance when the number of total channels is not sufficiently large (Wang et al., 2007). A procedure to calculate the Correction Factor F_t has been suggested by Mancin et al. (2012). Considering the BPHE prototype, it consists of five channels; the refrigerant flows in two of those whereas the other three are fed with water. By flowing in the two external channels, the water is in contact with an adiabatic wall (end plate) and it exchanges heat with the refrigerant only through the internal plate. Therefore, the thermal performance of the two external channels is different from that of the third channel where the water exchanges heat with the refrigerant on both sides. In particular, in the hypothesis that each channel is fed with the same water flow rate, the central water channel presents a double heat transfer area and, consequently, the water experiences a higher temperature gain. The values of the overall condensation heat transfer coefficient K and of the correction factor F_t can be iteratively calculated by applying the following procedure. The iteration starts by estimating the overall condensation heat transfer coefficient K with the correction factor equal to 1; from this value, in the hypothesis that the K can be considered constant in all the channels, the values of the efficiency of the external and internal channels ($R=0$) are given by:

$$\begin{aligned} \varepsilon_{EXT} &= 1 - e^{-NTU_{EXT}} \\ \varepsilon_{INT} &= 1 - e^{-NTU_{INT}} \end{aligned} \quad (7)$$

where NTU_{EXT} and the NTU_{INT} are calculated:

$$NTU_{EXT} = \frac{K \cdot A_{lp}}{\dot{C}_{min}}$$

$$NTU_{INT} = \frac{K \cdot 2 \cdot A_{lp}}{\dot{C}_{min}} \quad (8)$$

where $\dot{C}_{min} = \dot{m}_w \cdot c_{p,w}$; the total heat flow rate transferred in the BPHE is given by the following general equation:

$$q = (n \cdot \varepsilon_{INT} + 2 \cdot \varepsilon_{EXT}) \cdot [\dot{C}_{min} \cdot (t_{sat} - t_{w,in})] \quad (9)$$

where n is the number of the internal water channels (one for the present prototype). On the other hand, considering the energy balance at the heat exchanger the same heat flow rate has to be equal to:

$$q = \dot{m}_w \cdot c_{p,w} \cdot (t_{w,out} - t_{w,in}) \quad (10)$$

By combining Eqs. (9) and (10), it is possible to calculate the mean outlet water temperature and, then, the ideal logarithmic mean temperature difference as defined by eq. (6).

Finally, the correction factor F_t is given by:

$$F_t = \frac{q}{K \cdot A \cdot (\Delta t_{ML})_{CC}} \quad (11)$$

The calculated values of the Correction Factor F_t are shown in Figure 2 as a function of the outlet vapour quality; as one can see, those values are always greater than 0.95. Even if it is not highlighted in the diagram, the results show that at low refrigerant mass velocity (i.e. 15 and 20 kg m⁻² s⁻¹), the Correction Factor F_t is higher than 0.96 and there is not any noticeable effect of the superheating. At higher mass velocity (i.e. 40 kg m⁻² s⁻¹), the Correction factor F_t is lower and it decreases when increasing the outlet vapour quality and the superheating of refrigerant at the inlet of the BPHE.

The procedure continues by calculating a new value of the overall condensation heat transfer coefficient K and then it restarts till the difference between two successive values of K is lower than a fixed value (10⁻³ W m⁻² K⁻¹).

From those results, the condensation heat transfer coefficient can be calculated by applying the well-known Wilson plot method (Fernández-Seara et al., 2007):

$$HTC_{Exp} = \left(\frac{1}{K} - \frac{1}{\alpha_w} - \frac{s}{\lambda_p} \right)^{-1} \quad (12)$$

where α_w is the water side heat transfer coefficient, s is the plate thickness and λ_p its thermal conductivity that is equal to 16 W m⁻¹ K⁻¹. The single phase heat transfer coefficients for water (α_w) are calculated with equations suggested by the manufacturer and validated by present authors with single phase tests. The deviations between the experimental values measured by the present authors and those calculated with manufacturer equations are within $\pm 10\%$ in the Reynolds number range $10 < Re_{LO} < 10000$. Furthermore, two models from the open literature have been selected for the estimation of the single phase heat transfer coefficient: one suggested by Martin (2010) and one recently proposed by Dović et al. (2009). The Martin (2010) model was able to predict the experimental measurements within $\pm 10\%$ whereas the values calculated using the model suggested by Dović et al. (2009) are within $\pm 20\%$ as compared to the experimental ones. In the end, the correlation proposed by the manufacturer showed the best agreement with the experimental single phase heat transfer results and it has been used in the present calculations.

The refrigerant mass flow rate is measured by means of a Coriolis effect mass flow meter (accuracy of $\pm 0.2\%$ of the measured value) while water and refrigerant temperatures were measured by means of calibrated T-type thermocouples (accuracy ± 0.05 K). The water temperature gain is measured with a calibrated 4-junction T-type thermopile with an accuracy of ± 0.03 K. The refrigerant pressure at the inlet of the test section is obtained from an absolute pressure transducer, which reports an accuracy of $\pm 0.055\%$ of the full scale. From the error propagation

analysis applied to the Wilson plot method, the uncertainty of the condensation heat transfer coefficient depends on the uncertainties of the overall condensation heat transfer coefficient (between 1.7% and 9.1%) and the water-side heat transfer coefficient ($\pm 10\%$). For the present test conditions, the uncertainty of the condensation heat transfer coefficient varies between 3.8% and 9.4%. The outlet vapour quality is measured with an uncertainty of ± 0.03 .

Figure 2 Correction Factor F_t as a function of the outlet vapour quality.

4. HEAT TRANSFER RESULTS

Figures 3a and 3b report the ratio between the experimental heat transfer coefficient and the Nusselt (1919) heat transfer coefficient for laminar film condensation on a vertical surface from a stagnant vapour plotted against the thermodynamic mean vapour quality (3a) and the saturation to wall mean temperature difference (3b), respectively. The gravity driven heat transfer coefficient, as suggested by Nusselt (1919), is given by:

$$\text{HTC}_{Nu} = 0.943 \cdot \left[\frac{\rho_L \cdot (\rho_L - \rho_G) \cdot g \cdot h_{LG} \cdot \lambda_L^3}{\mu_L \cdot L \cdot (t_{sat} - t_{wall})} \right]^{0.25} \quad (13)$$

The mean vapour quality is defined as the mean value between the thermodynamic vapour quality of the superheated gas at the inlet of the BPHE and the outlet vapour quality. The saturation to wall mean temperature difference can be calculated from the values of the overall and condensation heat transfer coefficients as defined by the following equation:

$$(t_{sat} - t_{wall}) = \frac{K \cdot (\Delta t_{ML})_{CC} \cdot F_t}{\text{HTC}_{Exp}} \quad (14)$$

At $15 \text{ kg m}^{-2} \text{ s}^{-1}$, the ratio of experimental to Nusselt (1919) HTC appears to be independent on both the mean vapour quality and the saturation to wall temperature difference; furthermore, there is only a little effect of the superheating since data at $\Delta T_{sup}=15 \text{ K}$ show slightly higher ratios than those at $\Delta T_{sup}=5 \text{ K}$. At $20 \text{ kg m}^{-2} \text{ s}^{-1}$, the ratio of HTCs increases with the superheating passing from around 1.35 at $\Delta T_{sup}=5 \text{ K}$ to around 1.6 at $\Delta T_{sup}=25 \text{ K}$, moreover, it also increases with the mean vapour quality whereas it decreases when increasing the saturation to wall mean temperature difference.

Finally, when increasing refrigerant mass velocity to $40 \text{ kg m}^{-2} \text{ s}^{-1}$, the ratio on the y-axis increases with the mean vapour quality and decreases with the saturation to wall mean temperature difference. Considering the effect of the superheating at $40 \text{ kg m}^{-2} \text{ s}^{-1}$, at low mean vapour quality the ratio of HTCs seems to be independent on superheating, while at high mean vapour quality, the ratio of HTCs increases with the superheating. Similar results

have been found by other authors, among them: Minkowycz and Sparrow (1966 and 1969), Mitrovic (2000) and Longo (2011).

Figure 3a (from top to bottom) Ratio of experimental to calculated HTC (Eq. 13) versus thermodynamic mean vapour quality as a function superheating when varying refrigerant mass velocity.

Figure 3b (from top to bottom) Ratio of experimental to calculated HTC (Eq. 13) versus saturation to wall mean temperature difference as a function of superheating when varying refrigerant mass velocity.

5. CONDENSATION HEAT TRANSFER MODEL

Recently, Mancin *et al.* (2012) have proposed a new computational procedure based on the measurements carried out during partial condensation of R407C and R410A inside different brazed plate heat exchangers; the condensation heat transfer coefficients were measured by imposing 15 K of superheating at the inlet of the test section and by varying the refrigerant mass flow rate between 15 and 40 kg m⁻² s⁻¹. Since the present R32 data points show that local heat transfer coefficients are strongly affected by inlet superheating, beside mass velocity, vapor quality, temperature difference, the model is here updated as follows. The model combines two components, for gravity and shear dominated condensation, where the gravity dominated mean heat transfer coefficient HTC_{Nu} is given by eq. (13) increased by a factor 1.2 while the shear dominated mean heat transfer coefficient HTC_A is calculated at the mean vapour quality as suggested by Cavallini *et al.* (2006) for shear controlled condensation inside tubes:

$$HTC_A = \alpha_{LO} \cdot \left[1 + 1.128 \cdot x^{0.817} \left(\frac{\rho_L}{\rho_G} \right)^{0.3685} \left(\frac{\mu_L}{\mu_G} \right)^{0.2363} \left(1 - \frac{\mu_G}{\mu_L} \right)^{2.144} \cdot Pr^{-0.1} \right] \quad (15)$$

The heat transfer coefficient α_{LO} of the refrigerant liquid is calculated with the single phase flow equations for the tested plate, using liquid properties with the total flow rate. The two components are combined considering the transitions for condensation in a vertical tube suggested by Shah (2009) and the transition between the ΔT dependent and ΔT independent condensation suggested by Cavallini *et al.* (2006). The heat transfer coefficient is dominated by gravity and calculated as $HTC_{Gr} = 1.2 \cdot HTC_{Nu}$ for $J_G \leq J_{G1}$ (Shah, 2009):

$$J_{G1} = 0.89 - 0.93 \cdot e^{-0.087 \cdot Z^{-1.17}} \quad (16)$$

The heat transfer coefficient is ΔT independent and calculated with equation (15) when (Cavallini *et al.*, 2006) $J_G > J_{G2}$:

$$J_{G2} = \left[\left(\frac{7.5}{4.3 \cdot X_{tt}^{1.11} + 1} \right)^{-3} + 2.6^{-3} \right]^{-0.333} \quad (17)$$

For $J_{G1} < J_G < J_{G2}$ the heat transfer coefficient is calculated as:

$$HTC_{CALC} = HTC_{Gr} + (HTC_A - HTC_{Gr}) \frac{J_G - J_{G1}}{J_{G2} - J_{G1}} \quad (18)$$

The model is applied in a step by step procedure as presented in Mancin *et al.* (2011). The single phase heat transfer coefficients for superheated vapour (α_G) is calculated with the Martin (2010) equation. Since the mechanism of the convective heat transfer of the superheated vapour may be considered to be similar to turbulent heat transfer in a rough tube as suggested by Fujii *et al.* (19) the friction factor to be inserted in the Martin equation is derived from the two phase pressure gradient. The comparison between the model proposed and the experimental results are summarized in Table 3 and plotted in Figure 6. Table 3 lists the relative deviation e_R , the absolute deviation e_A and the standard deviation σ_N for the entire database subdivided as a function of the superheating temperature.

Table 3 Comparison between the calculations and the experimental database. Present model.

Data	Runs	Relative deviation e_R [%]	Absolute deviation e_A [%]	Standard deviation σ_N [%]
$\Delta T_{sup}=5$ K	30	2.9	4.6	2.94
$\Delta T_{sup}=10$ K	26	-0.6	3.9	2.61
$\Delta T_{sup}=15$ K	33	-4.2	5.4	4.43
$\Delta T_{sup}=25$ K	10	-3.6	3.6	1.62
Overall	99	-1.0	4.6	3.40

It is interesting to point out that the model is able to predict the entire experimental database with a relative deviation $e_R=-1\%$, an absolute deviation $e_A=4.6\%$ and a standard deviation of $\sigma_N=3.4\%$. The model tends to underestimate the experimental measurements at slightly high outlet vapor quality.

Figure 6 Comparison between the experimental database and present model.

6. CONCLUSIONS

This paper investigates the effect of the inlet superheating during condensation of R32 inside a BPHE prototype; the heat transfer coefficients have been measured by imposing four different superheating at the inlet of the test section: 5, 10, 15, and 25 K. The refrigerant mass velocity has been varied between 15 and 40 kg m⁻² s⁻¹ and the outlet vapour quality ranges between 0.0 and 0.65. As it appears from the experimental results, the condensation heat transfer coefficient increases with the superheating particularly at low saturation to wall mean temperature difference. Considering a refrigerant mass velocity of 20 kg m⁻² s⁻¹, and taking as the reference the heat transfer coefficients measured at $\Delta T_{sup}=5$ K, in the present range of operating test conditions, the heat transfer is enhanced between 5% and 10% at $\Delta T_{sup}=10$ K, from 10% and 17% at $\Delta T_{sup}=15$ K while from 20% and 30% at $\Delta T_{sup}=25$ K. The experimental measurements have been compared with the calculated values from a model proposed in the present paper; the model presents a relative deviation $e_R=-1.0\%$, an absolute deviation $e_A=4.6\%$ and a standard deviation of $\sigma_N=3.4\%$.

NOMENCLATURE

A	area	(m ²)	Greek Symbols	
	specific heat flow rate capacity	(W K ⁻¹)		
c_p	specific heat capacity	(J kg ⁻¹ K ⁻¹)	α	heat transfer coefficient (W m ⁻² K ⁻¹)
D_h	Hydraulic diameter	(m)	λ	thermal conductivity (W m ⁻¹ K ⁻¹)
GBF	mass velocity	(kg m ⁻² s ⁻¹)	μ	dynamic viscosity (Pa s)
g	gravitational acceleration	(m s ⁻²)	ρ	density (kg m ⁻³)
h	enthalpy	(J kg ⁻¹)	Subscript	
h_{LG}	latent heat	(J kg ⁻¹)		
HTC	heat transfer coefficient	(W m ⁻² K ⁻¹)		
J_G	$2 GBF \times [g D_h \rho_G (\rho_L - \rho_G)]$	(-)	CALC	calculated
K	overall heat transfer coefficient	(W m ⁻² K ⁻¹)	DEW	dew point
L	plate length	(m)	Exp	experimental
	mass flow rate	(kg s ⁻¹)	G	gas phase

p	pressure	(Pa)	in	inlet
Pr	Prandtl Number	(-)	L	liquid phase
q	heat flow rate	(W)	LO	liquid with the total flow rate
Re	Reynolds Number	(-)	Nu	gravity driven
s	plate thickness	(m)	out	outlet
t	temperature	(°C)	ref	refrigerant
T	temperature	(K)	RED	reduced
x	vapour quality	(-)	sat	saturation
X_{II}	$(\mu_L/\mu_G)^{0.1}(\rho_G/\rho_L)^{0.5}[(1-x)/x]^{0.9}$	(-)	w	water
Z	$[1/(x-1)]^{0.8} p_{RED}^{0.4}$	(-)	wall	wall

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