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On the scaling of convective boiling heat transfer coefficient

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ABSTRACT

Flow boiling has been widely used during the last century, but the key mechanisms controlling the heat transfer process still remain elusive. In the particular case of convective flow boiling inside a heated pipe, most of the existing correlations have been proposed assuming an enhancement factor to the liquid Reynolds number. In this work, we show that during convective boiling heat transfer, the vapor Reynolds number plays a dominant role which has been overlooked. By comparing experimental data in this work and from the literature, we show that the enhancement factor to the liquid Reynolds number does not appropriately correlate the heat transfer coefficient. Further, it can be shown that most of the proposed enhancement factors can be re-written in terms of the explicit contribution of the Re_V. In particular, at high qualities, the influence of Re_L becomes negligible.

1. Introduction

In the case of flow boiling inside a pipe, early research has suggested two regimes [1] namely saturated nucleate boiling and two-phase forced convection heat transfer. It was assumed that the nucleate boiling may occur near the inlet or at low flow rates. This is because it is assumed that the vapor generated during the forced convection suppresses the conditions for nucleate boiling. Moreover, the dominant mechanism of the heat transfer during flow boiling was assumed to be forced convection[2]. These two regions were suggested to be limited at high heat fluxes by the boiling crisis which is referred to as the departure of nucleate boiling, DNB, and dryout respectively [1]. Now, these two regions are widely-known as nucleate flow boiling and convective flow boiling [3].

In the case of convective boiling, early research suggested that the heat transfer from the heated wall to the liquid is controlled by the liquid film with a linear temperature profile across the film [1]. Later, several experimental and numerical research on the single-phase flow heat transfer [4–6] suggested that the thermal resistance of the conductive sublayer plays a dominant role in controlling the heat transfer rates. Beyond the conductive sublayer, rapid diffusion of heat takes place. As the conductive sublayer thickness can be smaller than the thickness of the liquid film, the early assumption of controlling the heat transfer by the liquid film thickness remains unclear. Assuming that the conductive sublayer is controlling the heat transfer process, then the heat transfer coefficient of convective flow boiling, flow condensation and single-phase flow can be considered to be equivalent [7–9]. The convective flow boiling is dominant at low heat fluxes and the heat transfer rate depends on the mass flux and the vapor quality.

Nucleate flow boiling is dominant at high heat fluxes and high working pressures. The mechanism responsible for the heat transfer is attributed to the bubbles produced at the wall. Contrary to early assumptions both regimes can be found from low to high thermodynamic qualities. The transition between these two regimes has also motivated research in order to determine if the transition is triggered sharply or if there is a region where both mechanisms interact. One particular approach for dealing with the convective flow boiling to the nucleate flow boiling transition is to consider an asymptotic model

$$h_{2\phi} = \left(h_{CB}^n + h_{NB}^n\right)^{1/n} \tag{1}$$

where if $n \to \infty$ the transition is sharp, while low values of n implies an overlapping of both models[10]. For example, Steiner and Taborek [11] consider a value of n between 3 and 4. Although from a practical perspective models have tried to combine in one expression both regimes, the underlying physics of each regime is quite different and thus the quest of a general expression has hindered the quest of improving the understanding of the physics of each regime.

In this work, we focus particularly on the convective flow boiling regime. This regime can be found in systems operating at pres-

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Nomenclature

D	Channel diameter (m)
AVG	Average error
MAE	Mean absolute error
C_o	Convection number
f	Enhancement factor
h	Heat transfer coefficient $(Wm^{-2}K^{-1})$
G	Mass flux $(kgm^{-2}s^{-1})$
k	Thermal conductivity of liquid $(Wm^{-1}K^{-1})$
Nu	Nusselt number $(\frac{hD}{k})$
Pr	Prandtl number
Pr_L	Liquid phase Prandtl number
Pr_V	Vapor phase Prandtl number
Re_L	Liquid Reynolds number
T_{sub}	Inlet subcooling (K)
Re_V	Vapor Reynolds number
$Re_{2\phi}$	Two-phase Reynolds number
Nu_{pred}	Predicted Nusselt number
$Nu_{\exp}$	Experimental Nusselt number
x	Vapor quality
μ_l	Viscosity of liquid $(kgm^{-1}s^{-1})$
μ_v	Viscosity of vapor $(kgm^{-1}s^{-1})$
ϕ_{Ltt}	Two-phase flow multiplier
χ_{tt}	Martinelli parameter
ρ_l	Density of liquid (kgm^{-3})
ρ_v	Density of vapor (kgm^{-3})
Bo	Boiling number
q''	Heat Flux (Wm^{-2})
h_{lv}	Latent heat of vaporization $(Jkg^{-1}K^{-1})$
h_l	Specific enthalpy of liquid $(Jkg^{-1}K^{-1})$
g	Acceleration due to gravity (ms^{-2})
Fr	Froude number
S	Suppression coefficient
A	Cross section area (m^2)
$T_{w,\,i}$	Inner wall temperature (K)
T_f	Fluid temperature (K)

sures far from critical pressure and/or high mass fluxes where the transition to nucleate flow boiling can occur at quite high heat fluxes. Since the past few decades, a large number of analytical, numerical and experimental studies have been carried out to identify the mechanisms controlling the heat transfer and to provide models to predict the heat transfer rate. Most works have suggested that the heat transfer coefficient during convective flow boiling can be expressed equivalent to the all liquid-phase heat transfer coefficient times a correction function in terms of the Nusselt number i.e. $Nu = Nu_{1\phi} \times f(\cdot)$. Here $Nu_{1\phi}$ is the single-phase Nusselt number and $f(\cdot)$ is the correction function. One of the simplest expressions for $Nu_{1\phi}$ is given by the equation attributed to Dittus-Boelter and McAdams [12], following the equation proposed by Nusselt in 1910 (as cited in [13]) based on similarity theory, which contains only 2 dimensionless groups (Re = Reynolds number and Pr = Prandtl number) and 3 adjusted parameters,

$$Nu_{1\phi} = \frac{hD}{k} = f_1(Re) f_2(Pr) = CRe^n Pr^m \tag{2}$$

where h is the heat transfer coefficient, D is the diameter of the channel, k is the fluid thermal conductivity and C, n, m are the adjusted parameters. The parameters C and n are 0.023 and 0.8 respectively, whereas m is suggested to be 0.3 for cooling and 0.4 for heating. Then, most models have suggested expressions for convective flow boiling of the form

$$Nu_{2\phi} = CRe_L^n Pr^m f(\cdot) \tag{3}$$

with $Re_L = GD(1-x)/\mu_l$. For example, Chen 1966 [14] suggested a correction function f defined as the ratio of two-phase Reynolds number to the single-phase Reynolds number as

$$f = \left(\frac{Re_{2\phi}}{Re_L}\right)^{0.8} = \phi_{Ltt}^{0.89} \tag{4}$$

where ϕ_{Ltt} is the two-phase pressure drop factor defined like

$$\phi_{Ltt}^2 = 1 + \frac{C}{\chi_{tt}} + \frac{1}{\chi_{tt}^2} \tag{5}$$

where χ_{tt} is the Martinelli parameter given as $\chi_{tt} = ((1-x)/x)^{0.9}(\rho_v/\rho_l)^{0.5}(\mu_l/\mu_v)^{0.1}$. Although the model is suggesting an enhancing factor with respect to the liquid single-phase flow heat transfer coefficient, it is interesting to note that the model can be written like

$$Re_L^{0.8} f = Re_L^{0.8}\left(1 + \frac{C}{\chi_{tt}} + \frac{1}{\chi_{tt}^2}\right)^{0.89/2} \tag{6}$$

$$\begin{aligned} &= Re_L^{0.8} + Re_V^{0.8} C^{0.445}((1-x)/x)^{0.4}(\rho_l/\rho_v)^{0.22}(\mu_v/\mu_l)^{0.8445} \\ &\quad + Re_V^{0.8}(\rho_l/\rho_v)^{0.445}(\mu_v/\mu_l)^{0.889} \\ &= Re_L^{0.8} + Re_V^{0.8}\phi \end{aligned} \tag{7}$$

with

$$\begin{aligned} \phi = {} & C^{0.445}((1-x)/x)^{0.4}(\rho_l/\rho_v)^{0.22}(\mu_v/\mu_l)^{0.8445} \\ & + (\rho_l/\rho_v)^{0.445}(\mu_v/\mu_l)^{0.889} \end{aligned} \tag{8}$$

$$Re_L = \frac{GD(1-x)}{\mu_l} \tag{9}$$

$$Re_V = \frac{GDx}{\mu_v} \tag{10}$$

Fig. 1 shows that the term ϕ is greater than 1 and $Re_V^{0.8}$ dominates over $Re_L^{0.8}$ in a wide range of vapor quality and pressure. Therefore, the model proposed by Chen 1966 [14] is in fact considering a strong dependency on the Re_V.

In a similar manner, Gungor and Winterton [15] proposed the enhancement factor f as:

$$f = 1 + 24000 Bo^{1.16} + 1.37\chi_{tt}^{-0.86} \tag{11}$$

where the $Bo = \frac{q''}{h_{lv}G}$ is the boiling number. For low heat fluxes the second term becomes negligible. Then it is possible to see that, following the same approach as in the previous model, $Re_L^{0.8}(1 + 1.37\chi_{tt}^{-0.86})$ leads to:

$$Re_L^{0.8} f = Re_L^{0.8}(1 + \chi_{tt}^{-0.86}) \approx Re_L^{0.8} + Re_V^{0.8}\left(\frac{\rho_l}{\rho_v}\right)^{0.43}\left(\frac{\mu_v}{\mu_l}\right)^{0.886} \tag{12}$$

where

$$\psi = \left(\frac{\rho_l}{\rho_v}\right)^{0.43}\left(\frac{\mu_v}{\mu_l}\right)^{0.886} \tag{13}$$

contains only fluid properties. As $Re_V\psi \gg Re_L$, this model also suggests a strong dependency on Re_V.

Some models have considered additive concepts including convective flow boiling and nucleate flow boiling. For example, Kandlikar [3] suggested a dependency of the heat transfer coefficient on the convection number $C_o = \left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}$ and Froude number multiplier $Fr_{lo} = \frac{G^2}{\rho_l^2 gD}$ with C_1, C_2, C_5 empirical constants.

$$\frac{h_{CB,2\phi}}{h_l} = C_1 C_o^{C_2}(25Fr_{lo})^{C_5} \tag{14}$$

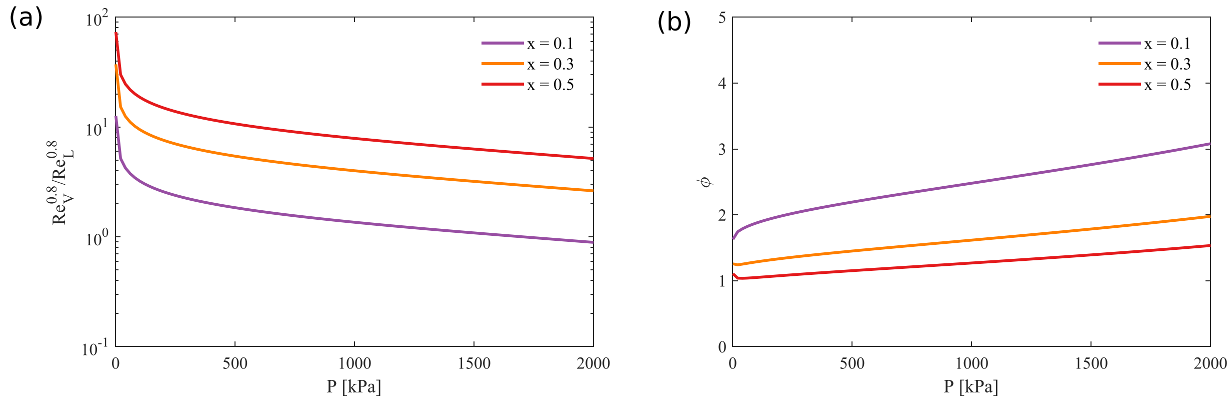

Fig. 1. (a) The vapor phase Reynolds (Re_V) number dominates over liquid phase Reynolds number (Re_L) over a wide range of system pressure. (b) Because the ϕ value is close to 1, it suggests a dominant influence of Re_V.

Table 1
Summary of experiments on flow boiling in horizontal tubes with R134a and diameter between 3 and 15 mm. (NB: Nucleate boiling, CB: Convective boiling) .

Reference	D [mm]	L [mm]	Pressure [kPa]	G [kg m^{-2} s^{-1}]	q" kW m^{-2}	Note
Greco and Vanoli [16]	6	6000	303–739	360	10.9–20.8	CB
Saitoh et al. [17]	3.1	3235	350–488	150–450	5–29	NB, CB
Mastrullo et al. [18]	6	780	216–350	200–350	10	CB
da Silva Lima et al. [19]	13.84		350–572	300–500	7.5–17.5	CB
Del Col [20]	8	1000	792–1160	200–600	14–30	NB
Grauso et al. [21]	6	780	263–445	146–520	5–20.4	CB
Manavela Chiapero et al. [22]	5	2000	838	298–497	10.5–20	NB, CB
Kundu et al. [23]	9.52	1200	361–402	100–400	3–10.5	CB
Xu et al. [24]	4.065	1200	538–676	185–410	18–28.0	NB

As $h_l = 0.023Re_L^{0.8}Pr_l^{0.4}(k_l/D)$, and considering $C_2 = -0.9$ and $C_5 = 0$ for $Fr_{lo} > 0.04$ as reported in Kandlikar [3] , the product $h_lC_o^{C_2}$ can be written as:

$$h_lC_o^{-0.9} \approx Re_L^{0.8}\left(\left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\right)^{-0.9} \quad (15)$$

$$\approx Re_V^{0.8}\left(\frac{\rho_l}{\rho_v}\right)^{0.45}\left(\frac{\mu_v}{\mu_l}\right)^{0.8} \quad (16)$$

with

$$\psi^* = \left(\frac{\rho_l}{\rho_v}\right)^{0.45}\left(\frac{\mu_v}{\mu_l}\right)^{0.8} \quad (17)$$

containing only fluid properties and equivalent to the correction factor ψ shown in the previous case. In this case, the model is also suggesting a strong dependency on the Re_V.

In summary most models, although they have been based on the assumption of an enhancement of the liquid Reynolds number (Re_L), the underlying models are suggesting a strong role of the vapor Reynolds number (Re_V) in the convective boiling heat transfer.

Regarding experimental studies in flow boiling in horizontal pipes, several researchers have investigated the phenomenon in the past decade considering R134a as working fluid. Table 1 provides a brief summary of these studies for pipe diameters ranging from 3 to 15 mm. Fig. 2 shows the corresponding experimental matrix.

The studies show that the flow boiling heat transfer coefficients can either show a dependency on the local vapor quality and mass flux or an independence on the local vapor quality. The former case is attributed to dominant convective boiling, while the latter to dominant nucleate boiling. The studies of Greco and Vanoli [16] in a horizontal tube of diameter 6 mm with R134a as working fluid found convective boiling dominance. The studies of Saitoh et al.

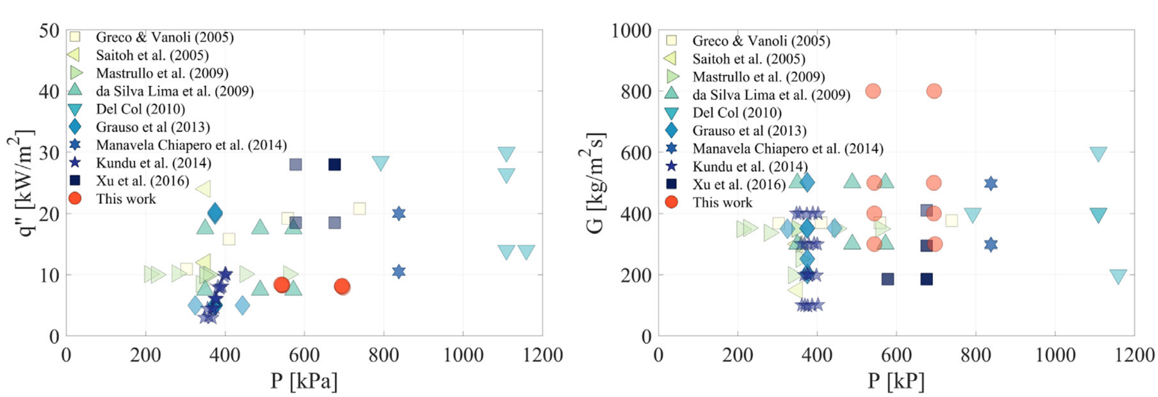

Fig. 2. The experimental matrix of the cases discussed in the Introduction.

[17] in multiple horizontal tubes with I.D. 0.51, 1.12, and 3.1 mm suggest that nucleate boiling is dominant at very low qualities and forced convective evaporation is dominant at high qualities. The transition from nucleate boiling to convective boiling was also observed by reducing the heat flux for a same mass flux. Mastrullo et al. [18] observed a strong dependency of the heat transfer coefficients on the mass flux. In addition, almost independency on the fluid pressure is observed. The studies of Da Silva Lima et al. [19] suggest that, at high vapor quality the mass flux influence the heat transfer coefficient, while at low vapor quality the mass flux influence is negligible suggesting a nucleate boiling dominance. Grauso et al. [21] investigated the flow boiling heat transfer in a horizontal tube of diameter 6 mm showing a convective boiling dominance. The studies of Manavela et al. [22] suggested that as the mass flux increases, the effect of heat flux decreases. Similarly, the studies of Deng et al. [25] and Kundu et al. [23] found convective boiling dominance while Xu et al. [24] observed dominant nucleate boiling.

By observing the existing studies as shown in the experimental matrix, it is possible to conclude that at low heat flux and low pressure, the heat transfer coefficient shows a dependency on the local vapor quality and mass flux. This is attributed to the dominant convective boiling mechanism. However, at high heat flux and high pressure the heat transfer mechanism is attributed to dominant nucleate boiling. In this case the heat transfer coefficient becomes independent of the local vapor quality. Thus, to model the heat transfer coefficients, most of the studies used the additive concept of convective boiling and nucleate boiling heat transfer. The total heat transfer coefficient is obtained by a weighted summation of heat transfers e.g. [3,14,15,26] due to the above-said two mechanisms.

In this work we show that, instead of an enhancement in the liquid Reynolds number, the factor (f) represents the contribution of vapor Reynolds number, and thus the vapor Reynolds number is a key term in modeling the convective boiling heat transfer coefficients.

2. Method

Experimental setup and procedure

The experimental facility is a closed loop containing a heated section, a pump, a conditioner, a main tank and R134a as working fluid. The pressure of the fluid in the test section is controlled by the saturation conditions in the main tank. A shell and tube type heat-exchanger is used to control the inlet temperature of the working fluid. The flow rates are measured by a Coriolis mass flow meter installed at the inlet of the test section. The test section is a stainless steel tube of length 2035 mm and I.D. 5 mm. The test section (Fig. 3) consists of 5 subsections of equal lengths which can be independently heated with the Joule effect. To convert the AC to DC power supply to the section, a controller and rectifier circuit is used. The total electrical input power is calculated from measured voltage and current in the heated section. Moreover, appropriate insulation is used at the outer surface of the test section to reduce the heat losses [27]. The influence of the axial heat conduction in the heated wall was studied in [27].

To measure the temperatures at different locations, 10 thermocouples are installed at the outside bottom wall and 7 at the outside top wall. Moreover, at two locations (at 1117 mm and 1917 mm) from the inlet, thermocouples are installed on the top, bottom, and side walls along-with in-flow internal thermocouples. all the variables (the temperatures, absolute pressures, pressure differences, mass flow rate) are acquired at a frequency of 10 Hz and are logged with a National Instrument data acquisition system.

Measurements and accuracy of measurements

T-type thermocouples of 0.5 mm diameter are used to measure the temperatures with 0.1K of accuracy. The saturation temperature T_{sat} is calculated based on the equilibrium properties of the fluid with the software REFROP version 9.1 [28].

Absolute pressure transducers are used to measure the inlet and outlet pressures with an accuracy of 0.04% at full scale of 2500kPa. A differential pressure transducer is used to measure the pressure drop across the test section with an accuracy of 0.075% at full scale (50kPa). The error in the heat flux (q'') is associated with the errors in the voltage and current measurements. The vapor quality is calculated by using a heat balance along the test section as:

$$x(z) = \frac{\int_{z_0}^{z} q''\pi D_i dz - G\ A\ cp_l\ T_{sub}}{G\ A\ h_{lv}} \tag{18}$$

here $x(z)$ is the vapor quality at location z [m] along the heated section, G [kg m^{-2} s^{-1}] is the mass flux, h_{lv} Jkg^{-1}K^{-1} is the enthalpy of vaporization and T_{sub} [K] the inlet subcooling, cp_l [J kg^{-1} K^{-1}] is the liquid phase heat capacity of the fluid, and A [m^2] is the cross section area of the pipe.

The local heat transfer coefficient measurements are done at the location 1917 mm from the inlet by applying the Newton equation as:

$$h = \frac{q''}{T_{w,i} - T_f} \tag{19}$$

where h is the heat transfer coefficient, T_f is the fluid temperature measured with the in-flow thermocouple, $T_{w,\,i}$ is the inner wall temperature and q'' is the heat flux. It is worth noting that the inner wall temperature $T_{w,\,i}$ is calculated by solving the 1-D steady state heat conduction equation along the radial direction of the test section by assuming a uniform heat generation. Moreover, the measured outer wall temperature $T_{w,\,o}$ is considered as a boundary condition. The outer wall temperature $T_{w,\,o}$ is the average temperature measured at four positions (top, bottom and two side walls) at the above-mentioned location from the inlet. The mutual measurement difference between the thermocouples was less than 0.4$^\circ C$. The mean measurement uncertainty of the heat transfer coefficients is about 10% at high heat fluxes, but at low heat fluxes it can reach up to 30%.

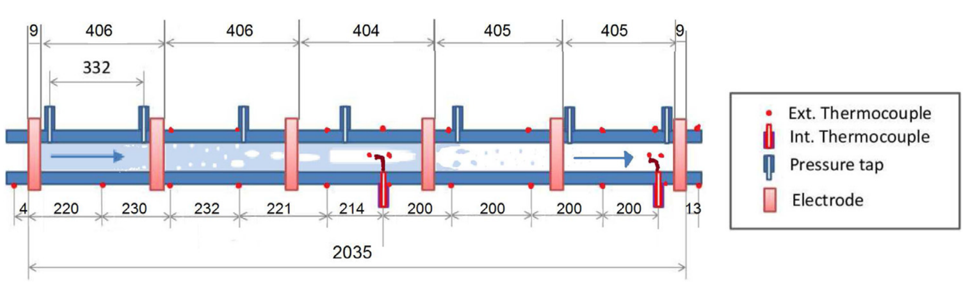

Fig. 3. Sketch of the test section.

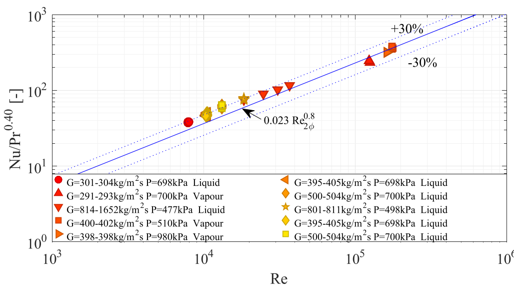

Fig. 4. Single-phase liquid and vapour heat transfer coefficient measurement and prediction by Dittus-Boelter correlation.

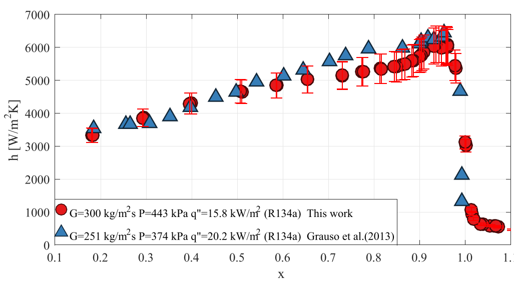

Fig. 5. Comparison of the two-phase flow heat transfer coefficient to a similar case form the literature.

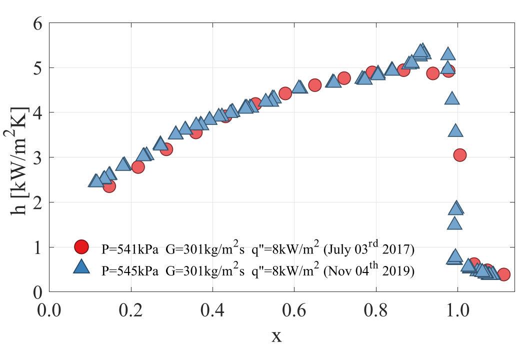

Fig. 6. Test of repeatability of the heat transfer coefficient.

Experiment validation

The measured heat transfer coefficients in case of liquid and vapor single-phase are compared against the Dittus-Boelter [13] equation. As it is seen in Fig. 4 the measured data is in good good agreement with the Dittus-Boelter equation, it justifies the reliability of the measurements. The two-phase flow heat transfer coefficients are compared (Fig. 5) against one set of experimental data in the literature [21] with similar working conditions. A test of reproducibility is also presented in Fig. 6. All these comparisons suggest that the measurement uncertainty in the facility is very low.

Experimental Method

The pressure of the fluid was kept constant at the outlet of the test section for each experiment. Extreme care was taken to assure that steady-state conditions are reached before recording the data. The steady-state conditions are declared when the time-averaged variations of the mass flux and pressure reach below ± 6%. The inlet subcooling at the entrance of the test section was kept at least $7°C$. This minimizes the occurrence of subcooled boiling of the liquid before entering the test section. Two-phase flow instabilities [29,30] were avoided by controlling a valve at the inlet of the test section. For each data, about 100s were recorded corresponding to about 1000 points.

Before the experiments, the facility was first heated up to the planned power. Each data was recorded then by decreasing the power to a desired value. By doing so, the jump in the wall temperature observed for the onset of nucleate boiling is avoided. This provides a good repeatability of the experiments. Moreover, the experiments were performed by decreasing the vapor quality from $x > 1$ to $x < 0$. This procedure avoids the jump in the wall temperature that is observed, for example, when the onset of nucleate boiling occurs or changes of the flow pattern [31,32].

In this work, experimental data over a wide range of fluid properties and pipe diameters have been gathered which is summarized in Table 2. The performance of the models are evaluated by $\theta_{\pm 10\%}$, $\theta_{\pm 20\%}$ and $\theta_{\pm 30\%}$. These represent the percentage of data points predicted within ± 10%, ± 20% and ± 30% respectively. In addition, the average and mean absolute errors are defined as

$$AVG = \frac{1}{N}\sum \frac{Nu_{pred} - Nu_{exp}}{Nu_{exp}} \times 100 \tag{20}$$

$$MAE = \frac{1}{N}\sum \frac{|Nu_{pred} - Nu_{exp}|}{Nu_{exp}} \times 100 \tag{21}$$

3. Results and discussion

As this work is focused on dominant convective flow boiling, in order to determine the corresponding working conditions, the dependency of the heat transfer coefficient in terms of the heat flux is presented in Fig. 7. It is possible to distinguish the convective flow boiling region characterized for the independence on the heat flux, and the nucleate flow boiling region independent on the mass flux and proportional to the heat flux. The figure shows that the heat transfer coefficient becomes independent of heat fluxes below $q'' = 10kW/m^2$ as thus convective dominant. For this reason, only heat fluxes below 10 kW/m^2 are considered in the present study.

Fig. 8 shows the heat transfer coefficients in terms of the vapor quality. As the experiment is done at low heat fluxes, it is expected that dry-out of the wall is negligible and the heat transfer will decrease suddenly at $x \approx 1$ as shown in the figure. This trend is also confirming the validity of the experimental setup. The heat transfer coefficient increases monotonously with x as expected for convective flow boiling. The heat transfer coefficient corresponding to the same mass flux but different pressures shows similar trends with a slight shift.

The data points from Fig. 9 corresponding to the vapor quality range $0.2 - 0.8$ are shown in terms of the Re_L, Re_V and $Re_{2\phi} = Re_L + Re_V$. In the figure the Nusselt number is scaled by the weighted Prandtl number $Pr_{2\phi} = Pr_V x + Pr_L(1 - x)$. In the same figure, experimental data corresponding to the heat transfer in all liquid and all vapour case are included. A significant difference between the single-phase flow and the convective flow boiling heat transfer is observed in Fig. 9a. However, in Fig. 9b the difference is minimum. This implies that to follow the analogy of single-phase flow, the Re_V plays a dominant role over Re_L. To include the effects of both Re_L and Re_V on the scaling of Nusselt numbers, an equivalent two-phase Reynolds number ($Re_{2\phi} = Re_L + Re_V$) is used in Fig. 9c. In this case, no noticeable difference between the single-phase flow and convective flow boiling heat transfer is observed. This fact suggests that equivalent heat transfer mechanisms are taking place in these cases.

Fig. 10 shows a comparison between the contribution of Re_L and Re_V on the two-phase Nusselt($Nu_{2\phi}$) number prediction. This two-phase Nusselt number scales the net convective boiling heat transfer rate. For the comparison, the experimental data presented in this work and the data from the literature were selected by con-

Table 2
Convective flow boiling experiments considered in this study to compare with the presented model.

Author(s)	Fluid	Diameter [mm]	G [kg m^{-2} s^{-1}]	P [kPa]	q'' [kW m^{-2}]
Wojtan et al.(2005) [33]	R22	13.84	300-500	581	7.5
da Silva Lima et al.(2009) [34]	R134a	13.00	300	348	7.6
da Silva Lima et al.(2009) [34]	R134a	13.00	300	486	7.6
da Silva Lima et al.(2009) [34]	R134a	13.00	300	569	7.6
Jabardo et al.(2000) [35]	R134a	12.70	300	386	5.0
Jabardo et al.(2000) [35]	R404a	12.70	300	776	5.0
Jabardo et al.(2000) [35]	R22	12.70	300	638	5.0
Kundu et al.(2014) [23]	R134a	9.52	400	386	3.0
Kundu et al.(2014) [23]	R134a	9.52	300	386	3.0
Kundu et al.(2014) [23]	R134a	9.52	400	348	3.0
Grauso et al.(2013) [21]	R134a	6.00	351-501	373	5.0
Saitoh et al.(2005) [17]	R134a	4.00	300	413	12.0
Lu et al.(2013) [36]	R1234yf	3.90	400	435	11.0
Lu et al.(2013) [36]	R134a	3.90	200-400	435	11.0
Kanizawa et al.(2015) [37]	R134a	2.32	500	605	5.0
Li et al.(2012) [38]	R1234yf	2.00	400	508	12.0

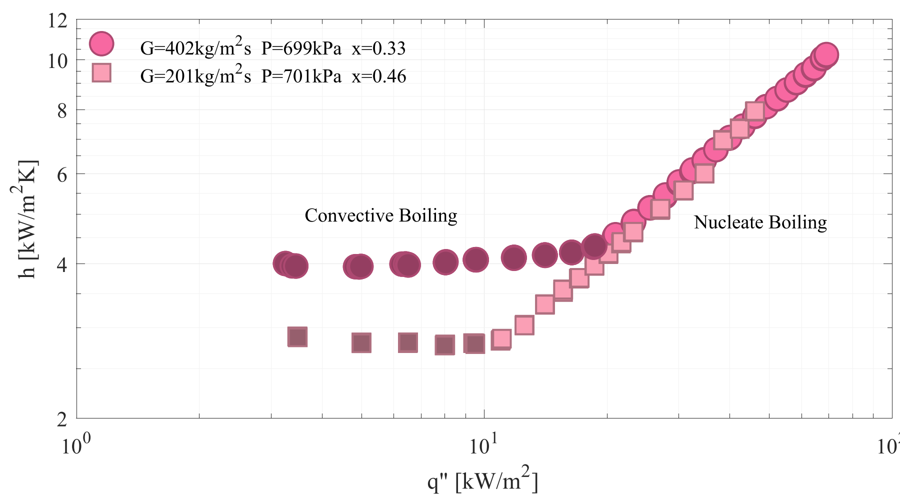

Fig. 7. Heat transfer coefficient measurements showing two distinct regimes, namely nucleate boiling and convective boiling. Nucleate boiling regime is characterized by a sharp dependency of heat transfer coefficient with heat flux. Above $10kW/m^2$ the measurements show the nucleate boiling regime. The convective boiling regime is characterized by an independence of heat transfer coefficient with heat flux. Below $10kW/m^2$ the measurements show the convective boiling regime.

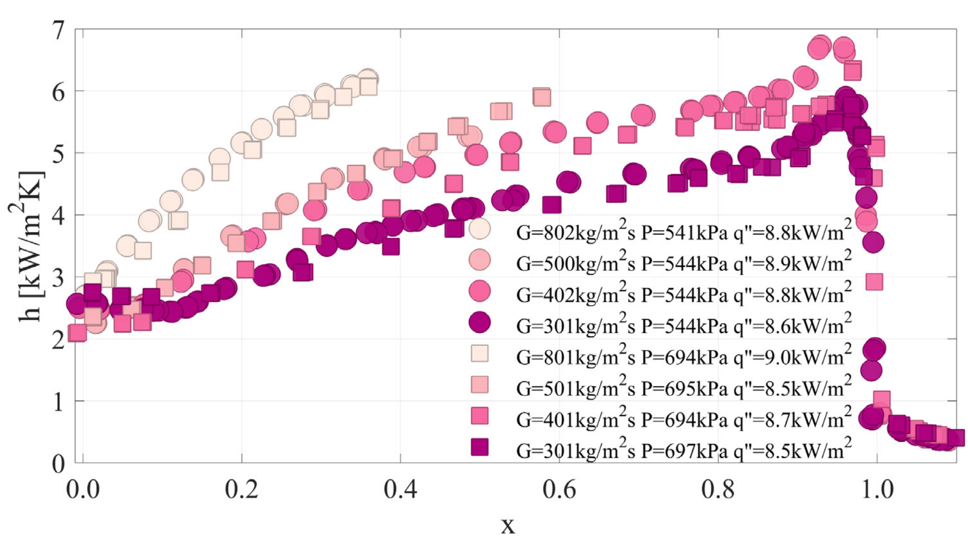

Fig. 8. The measurements showing the variation of convective boiling heat transfer coefficient with vapor quality. The data confirm two well-known properties of convective boiling heat transfer coefficient, namely (i) the heat transfer coefficient decreases with vapor quality and (ii) the heat transfer coefficient increases with mass flux.

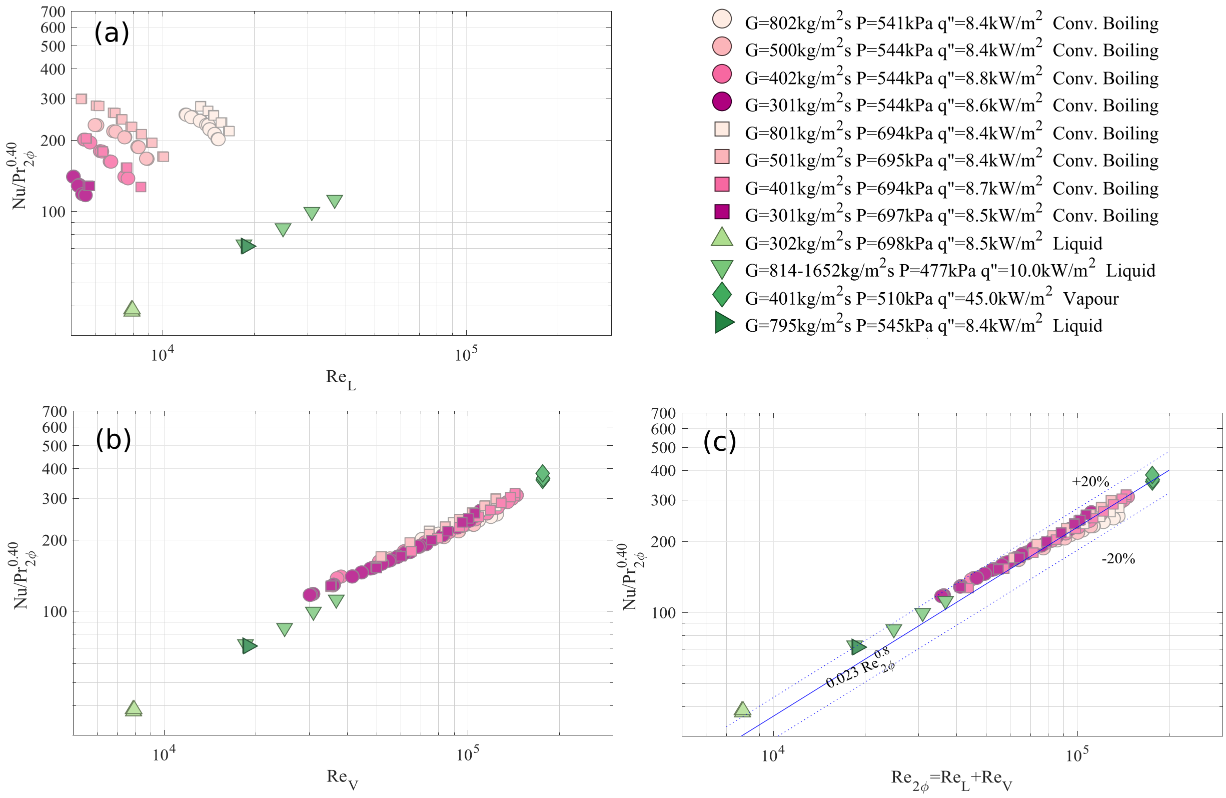

Fig. 9. (a) The influence of Re_L is poor to follow the analogy of single-phase flow. (b) The Re_V shows significant influence to follow the analogy of single-phase flow without any adjusting parameter. (c) By using the combined influence of Re_L and Re_V, the convective boiling heat transfer follows the analogy of single-phase flow heat transfer.

sidering cases related to low pressure, high mass flux and low heat flux. It was also evaluated that the data were showing independence on the heat flux when presented in terms of h-x plots. Furthermore, it was considered mainly experimental data that have included proper experimental validation of the facility in terms of single-phase flow heat transfer coefficient and energy balance. The selected cases are summarized in Table 2. Fig. 10a shows the contribution of the liquid Reynolds number in scaling the experimental two-phase Nusselt numbers which represents the two-phase heat transfer coefficients. It is seen from Fig. 10a that the contribution of Re_L to scale the Nusselt numbers is very low. Particularly at high qualities ($x > 0.5$) the contribution is negligible. However, in Fig. 10b, which shows the contribution of the vapor Reynolds number in scaling the experimental Nusselt numbers, a major contribution of Re_V is observed. Moreover, at high qualities ($x > 0.5$), almost no contribution of Re_L is observed in Fig. 10a, and 100% contribution of Re_V is seen in Fig. 10b. The combined effect of Re_L and Re_V is shown in Fig. 10c, where the comparison between the predicted and experimental Nusselt numbers is plotted with $Re_{2\phi} = Re_L + Re_V$. It is possible to conclude from the mentioned comparisons that during a convective boiling heat transfer, the Re_V plays a key role in controlling the fluid velocity which affects the heat transfer. Also, the effect of Re_V dominates over Re_L in wide range of qualities and other working conditions. Hence, the results suggest that the enhancement factor incorporated in most of the existing literature does not necessarily represent the enhanced influence of Re_L; instead it represents the influence of Re_V.

Fig. 11 shows the comparison of presented method with a few correlations available in the literature as discussed in the introduction. It is evident from the existing research that during convective boiling, the fluid velocity plays a dominant role in controlling the heat transfer rate. In this context, in Fig. 11a, the calculated Nusselt numbers using Chen's correlation [14] show a gradual reduction with vapor quality. This indicates that with the increase in vapor quality, appropriate consideration of the increase in fluid velocity is absent. In particular, at high flow qualities ($x > 0.7$) as a strong influence of vapor phase is expected, the large errors ($\pm$ 20%) suggest that the enhancement of the influence of liquid Reynolds number is insignificant. Note that the Chen's correlation is presented for predicting the convective boiling heat transfer coefficients in a vertical channel. As the data presented in this work are of horizontal channels, the gradual decrease in the predicted Nusselt numbers with qualities may be attributed to the difference in the channel orientation. In Fig. 11b, although the calculated Nusselt numbers (Gungor and Winterton [15]) show a uniform prediction characteristics (defined as the ratio of calculated to experimental Nusselt numbers), overall it under-predicts the heat transfer coefficients by 20%. Particularly, only about 26% data fall within $\pm$ 20% error interval. This suggests that although the nature of gradual increase in the fluid velocity with vapor quality is appropriately considered by the model, the amount of enhancement of the fluid velocity is under-predicted. Fig. 11c shows the model proposed by Kandlikar [3] which gives comparable prediction than the model by Gungor and Winterton Gungor and Winterton [15].

In contrast, in the presented method, the use of vapor Reynolds number remarkably addresses the increment in flow velocity with vapor quality. This is evident from the comparisons (Fig. 11d), as the prediction characteristics are uniform and close to 1 in all qualities ranging from 0.2 to 0.8. Thus, Re_V is a key factor to capture the variation of flow velocity compared to the liquid Reynolds number (Re_L) and enhancement factor (f). Interestingly, the comparison shows that using the two-phase Reynolds number, the convective boiling heat transfer coefficients can be calculated without using any additional adjusting parameter to the well-known Dittus-Boelter equation. This also justifies the presented analogy of convective boiling heat transfer coefficient to the single-phase heat

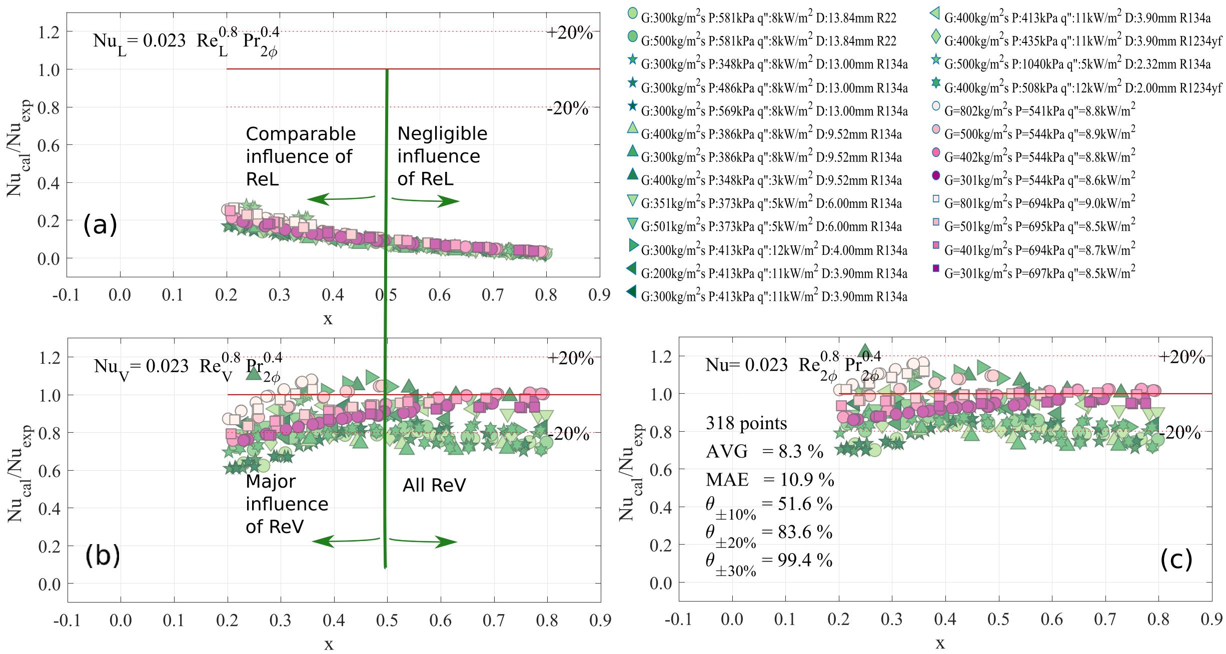

Fig. 10. (a) The maximum contribution of Re_L to predict the Nusselt numbers is found to be about 20%. At high qualities the contribution is negligible. (b) A major contribution of Re_V is observed to scale the Nusselt numbers. At high qualities the Re_V shows almost 100% contribution. (c) By using the combined influence of Re_L and Re_V the Nusselt numbers can be predicted well. This indicates that the Re_V plays a dominant role in defining the fluid velocity and it's influence on the heat transfer is high compared to Re_L.

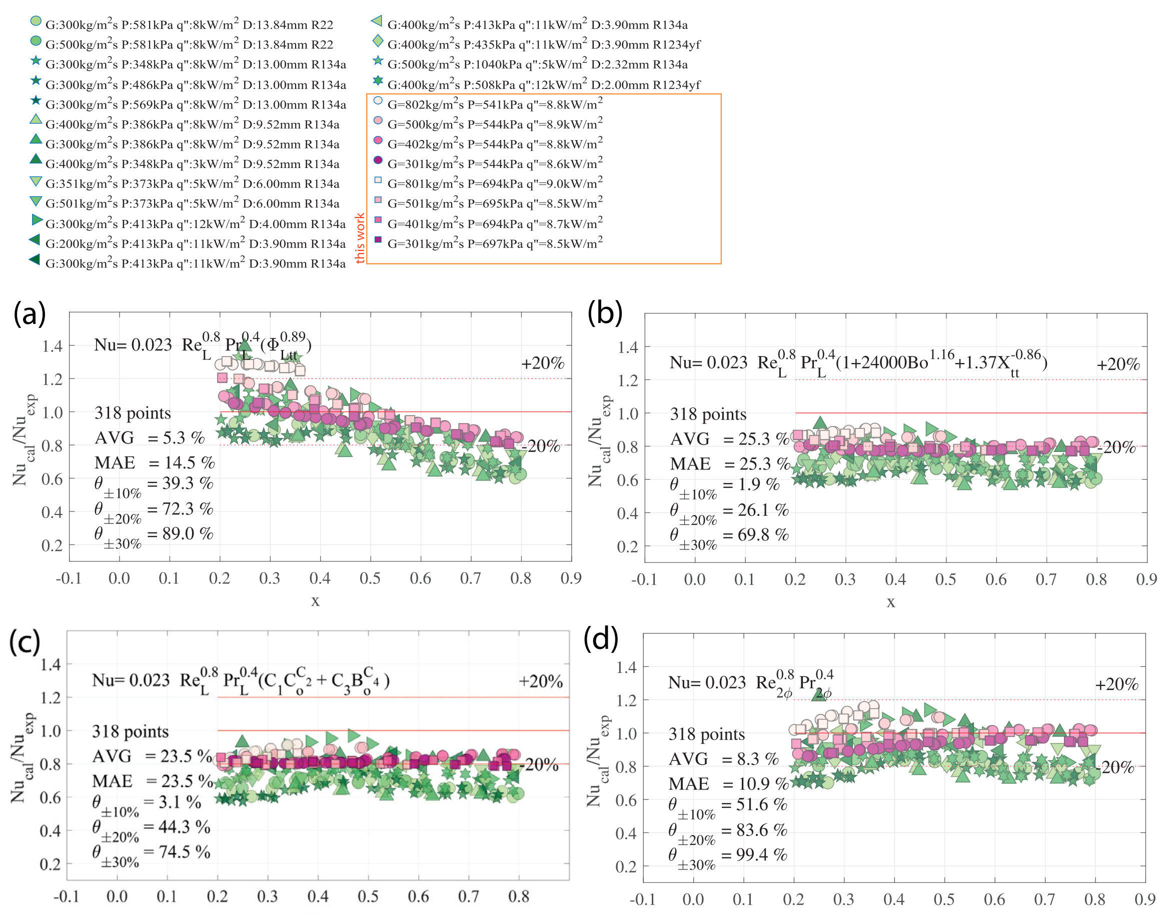

Fig. 11. The presented method is compared with some of the existing correlations proposed by a) Chen [14] (b) Gungor and Winterton [15] (c) Kandlikar [3]. (d) The present scaling method predicts about 84% data within ± 20% and 99% data within ± 30% error interval.

transfer correlation by Dittus-Boelter. Contrary to the approach of increasing the complexity of the models by adding dimensionless groups and adjusted parameters for improving the heat transfer predictions, identifying the dominant dimensionless groups the models can be simplified. Further work is required to identify the working conditions that lead to a transition to nucleate flow boiling.

4. Conclusions

In this work, we show that during convective boiling heat transfer, the vapor Reynolds number plays a dominant role which has been overlooked. It has been shown that, although models have considered an enhancement function for the liquid single-phase flow heat transfer coefficient, the suggested models can be rewritten in terms of vapor Reynolds number. However, in most of the cases, models have not considered an explicit dependency of the vapor Reynolds number. Identifying the dominant dimensionless groups can reduce the complexity of the models and thus avoid the risk of over fitting.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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