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Can the heat transfer coefficients for single-phase flow and for convective flow boiling be equivalent?

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During the past few decades, heat transfer during convective flow boiling inside pipes has been widely studied with the goal of unveiling the physics of the process. Different heat transfer mechanisms have been suggested based on different assumptions. This fact has resulted in a large number of models including different dimensionless numbers and in some cases up to a dozen of adjusted parameters. Here, we show that the convective flow boiling heat transfer coefficient is equivalent to the one for single-phase flow when the influence of the vapour velocity is taken into account. *Published by AIP Publishing.* https://doi.org/10.1063/1.5018659

In the case of flow boiling inside pipes, it is possible to distinguish two clear regimes, namely, nucleate boiling and convective flow boiling. At high heat fluxes, nucleate boiling is dominant and bubbles produced at the wall attribute to the control of the heat transfer. At low heat fluxes, e.g., related to conventional refrigeration applications, convective flow boiling is dominant and the heat transfer coefficient is observed to be directly dependent on the mass flux, i.e., the mass flow of refrigerant per cross area of the pipe, and the thermodynamic quality, i.e., the ratio of the mass vapour flow to the total mass flow. Although research on this area can be referred to the early 40s, no agreement has been achieved on which are the dominant mechanisms controlling the heat transfer from the wall to the working fluid. This fact has limited the development of accurate models. The lack of understanding can be appreciated in a large number of models developed during the past few decades,[1–3] based on different mechanistic approaches and with an increasing number of different dimensionless groups and adjusted parameters that can reach more than a dozen. This quest for unveiling the physics of the process has recently pushed the research towards the study of heat transfer at high spatial and temporal resolutions.[4]

The complexity and challenge for predicting the heat transfer during convective flow boiling contrast with the simplicity of the single-phase heat transfer coefficient in pipes. The equation attributed to that proposed by Dittus-Boelter and McAdams,[5] following the equation proposed by Nusselt in 1910 (as cited in Ref. 6) based on similarity theory, contains only 2 dimensionless groups and 3 adjusted parameters

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

where h is the heat transfer coefficient, D the diameter of the pipe, k the thermal conductivity of the fluid, $Re = GD/\mu$ the Reynolds number (with G being the mass flux and μ the dynamic viscosity), and $Pr = c_P\mu/k$ the Prandtl number (with c_p being the specific heat and k the fluid thermal conductivity). The exponent m is suggested to be 0.3 and 0.4 for cooling and for heating, respectively, $n = 0.8$, and the scaling constant $C = 0.023$. The model is based on two functional forms representing the hydrodynamic and thermodynamic effects $f_1(.)$ and $f_2(.)$, respectively. Several other models were suggested later not only based on larger experimental databases[7,8] but also based on these two dimensionless groups. Experimental and numerical studies of heat transfer in single-phase flow inside pipes[9–11] have shown that the thermal resistance is mainly concentrated in the conductive sublayer, while beyond this sublayer, a rapid diffusion of the heat into the bulk flow is observed.

Considering the case of convective flow boiling, the heat transfer process occurs from the pipe wall to the flowing refrigerant which is flowing forming a liquid film in contact with the wall surrounding a vapour core, as shown in Fig. 1. The heat transfer can be assumed to be controlled by a series of thermal resistances responsible for determining the net heat transfer exchange. Very close to the wall, there is a thin conductive sublayer resistance that is followed by the convective bulk film resistance, from where the heat is transported across the liquid-vapour interface to the bulk vapour. Most models in the literature have hypothesised a dominant thermal resistance across the total liquid film, i.e., summing up the conductive sublayer, the bulk film, and the interface resistance into one equivalent resistance, relating the dominant heat transfer mechanism to the liquid film thickness. Most models for convective flow boiling share a similar structure, i.e., $Nu_{CB} = Nu_{1\phi}f(.)$, with $f(.)$ being a correction function whose functional form has been searched for decades. Typically, the performance of the models has been assessed by their ability in predicting the overall heat transfer coefficient. Due to the limitation of such models for reproducing experimental results, corrections have been added, namely, modifying $f(.)$, for instance, including the influence of the vapour phase on the liquid film, entrainment, deposition of droplets, and the like. In other cases, models have been tried to include corrections to the single-phase heat transfer model, i.e., assuming only liquid, by multiplying by diverse dimensionless groups. In other cases, the mechanism

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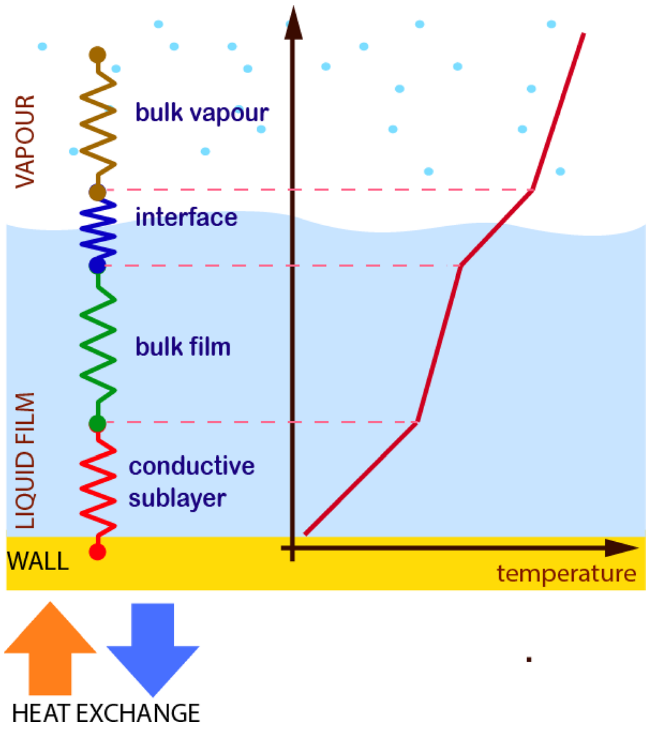

FIG. 1. Heat transfer resistance concept during convective flow boiling.

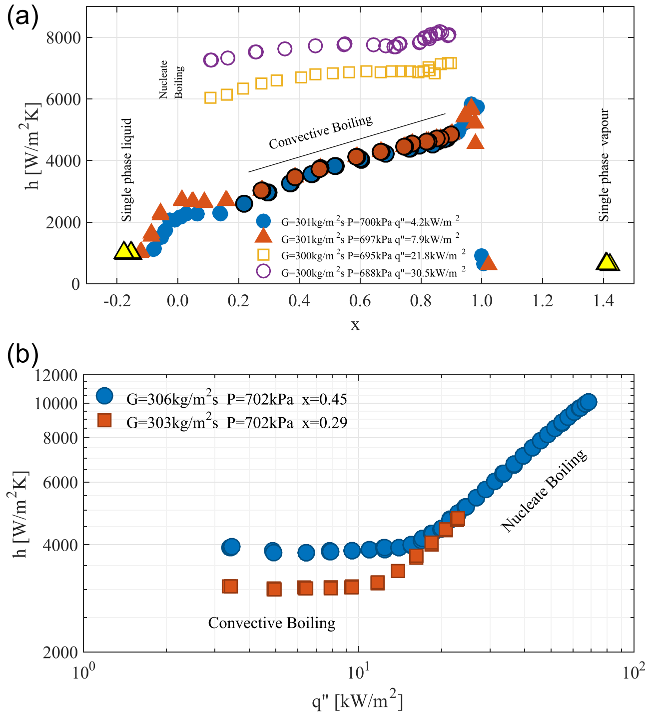

FIG. 2. (a) Typical heat transfer coefficient measurements showing nucleate boiling and convective flow boiling regimes. (b) Convective flow boiling to nucleate boiling transition.

has been attributed to a dominant phase change process occurring at the liquid-vapour interface, i.e., a dominant interface thermal resistance also referred to as a thin film evaporation process.

In this letter, we show that an equivalent heat transfer mechanism controls the heat transfer coefficient both during convective flow boiling and during single-phase flow. This implies that when the influence of the vapour phase is taken into account, the same model can predict the heat transfer coefficient accurately. Furthermore, it will be shown that the same mechanism is valid in other two-phase flow systems such as flow condensation in pipes and two-phase non-boiling flows (e.g., air-water).

To investigate the heat transfer process, the test section consists of a 5 mm ID stainless steel pipe heated with the Joule effect. The facility is equipped with a conditioning section to heat up the working fluid (R134a) to the desired local thermodynamic quality where the heat transfer coefficient is determined by 4 thermocouples installed at the outer wall of the pipe and one inner thermocouple for determining the fluid temperature. The details of the experimental facility, experimental procedures, calibration tests, and uncertainty analysis are presented in the supplementary material.

Figure 2 shows the heat transfer coefficient for R134a at high heat fluxes and low heat fluxes. At high heat fluxes, it is possible to see a dependency on the heat flux corresponding to the nucleate boiling regime. At low heat fluxes, it is possible to observe the dependence of the heat transfer coefficient on the heat flux, corresponding to the convective boiling regime, except at low qualities, $x < 0.2$, where the nucleate boiling regime looks to be dominant. In the same figure, the heat transfer coefficient corresponding to the single-phase liquid and vapour case is shown. The figure also shows the transition from the convective boiling to the nucleate boiling regimes in terms of the heat flux. This work is limited to the convective boiling regime. It is worth noticing that no suitable model is available for determining the transition between these two regimes. For simplicity, in this work, convective boiling is limited to mass fluxes above 200 kg/m^2 s for preventing flow stratification and relatively low heat fluxes for avoiding vapour generation at the wall.

By assuming that the dominant thermal resistance is mainly concentrated in the conductive sublayer also for the convective flow boiling case, the same physical model should describe the heat transfer coefficient during single-phase flow and convective flow boiling. This implies that the role of the vapour is limited to increasing the flow velocity compared to the all-liquid phase case. This implies that the heat transfer coefficient has to scale with the velocity of the liquid-vapour mixture and thus in terms a two-phase flow Reynolds number[12]

$$Re_{2\phi} = Re_L + Re_V = \frac{GxD}{\mu_V} + \frac{G(1-x)D}{\mu_L} \quad \text{for} \quad 0 < x < 1, \qquad (2)$$

with x being the thermodynamic quality. For $x < 0$, $Re_{2\phi} = Re_{L0} = GD/\mu_L$, while for $x > 1$, $Re_{2\phi} = Re_{V0} = GD/\mu_V$.

Following Eq. (1), the convective flow boiling and the single-phase flow heat transfer coefficients from Fig. 2 are shown in Fig. 3 in terms of $Re_{2\phi}$. The data shown are limited to a thermodynamic quality range $0.2 < x < 0.9$. The Nusselt number Nu is scaled by $Pr_{2\phi} = Pr_L(1-x) + Pr_V x$ which provides a transition from the two-phase flow to the all-liquid and to the all-vapour case.[12] The exponent n was selected to be 0.4 for heating while 0.3 for cooling, following Eq. (1). As the heat transfer mechanism during single-phase

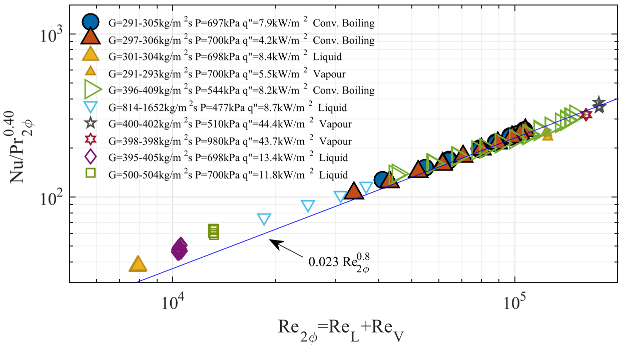

FIG. 3. Dependency of the experimental heat transfer coefficient of single-phase flow and convective flow boiling in terms of $Re_{2\phi}$.

flow and convective flow boiling is shown to be equivalent, this result can be extended to the case of flow condensation or non-boiling two-phase flows where a similar heat transfer mechanism can be considered. This fact is shown in Fig. 4 with experimental data from the literature for non-boiling two-phase flows (air-water) for slug and annular flow regimes in a 1.95 mm ID pipe,[13] flow condensation of binary mixtures[14] in a 8 mm ID, and flow condensation of a single-component fluid in 1 mm ID pipe[15] and in a 92 μm hydraulic diameter square channel.[16] The plot shows that the experimental data follow the $0.023Re_{2\phi}^{0.8}$ line, implying that the selected cases can be predicted by the traditional single-phase heat transfer coefficient obtained using Eq. (1). The equivalence between convective flow boiling and flow condensation has been reported experimentally by Sun and Hewitt[17] although no model has been able to address it. Furthermore, the search for a unified model capable of predicting the cases shown in Fig. 4 has motivated a large amount of research. Figure 5 shows the dependency of the heat transfer coefficient in terms of $Re_{2\phi}$ for convective boiling, condensation, and non-boiling two-phase flow for a large experimental dataset from the literature. The conditions of the experiments are provided in the supplementary material. The experimental dataset for flow condensation includes pipes with the internal diameter from 14.45 mm down to 0.49 mm, and microchannels of different shapes including triangular, semi-circular, rectangular, and square cross-sections with the hydraulic diameter from 1460 μm down to 67 μm. No noticeably effect of the geometry of the channels is observed, and all the data are well captured quantitatively and qualitatively by $Re_{2\phi}$. Furthermore, no effect or influence of the flow pattern is observed. This fact can be attributed to the fact that the dispersed phase is not interacting with the conductive sublayer, particularly at high mass fluxes and when bubbles are not produced at the wall. The experimental dataset for convective boiling includes pipes with the internal diameter ranging from 13.84 mm down to 2 mm and channels with the hydraulic diameter from 3.63 mm down to 0.78 mm, while for non-boiling two-phase flow, the data correspond to slug and annular flow regimes of air-water mixtures including pipes with the internal diameter ranging from 27.9 mm down to 1.95 mm and channels with the hydraulic diameter of 506 μm and 335 μm.

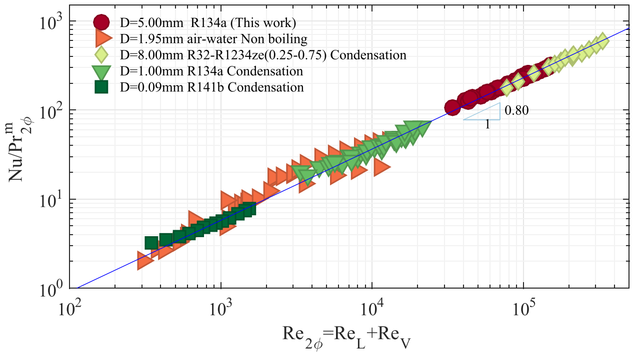

FIG. 4. Selected experiments showing the dependency of the heat transfer coefficient in terms of $Re_{2\phi}$.

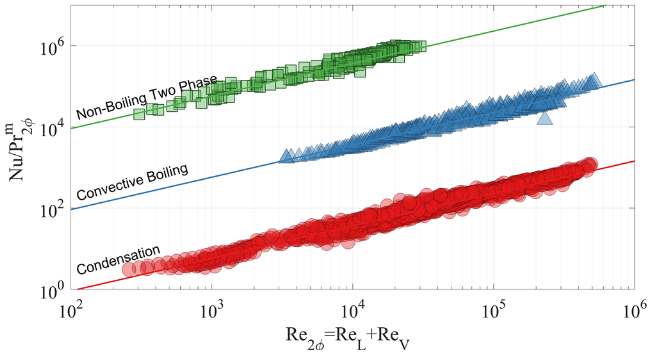

FIG. 5. Experimental data showing the dependency of the heat transfer coefficient in terms of $Re_{2\phi}$ for convective boiling, condensation, and non-boiling two-phase. *The curves have been shifted intentionally for improving visibility*.

From Fig. 5, it is possible to see that $Re_{2\phi}$ is able to capture the trend of the data independent of whether the heat transfer coefficient corresponds to single-phase flow, convective boiling, condensation, or non-boiling two-phase flows.

In summary, the equivalence between the heat transfer coefficient for single-phase flow and convective flow boiling is shown experimentally. Assuming that the vapour phase plays a major role in increasing the velocity of the flow and taking this influence into a two-phase flow Reynolds number, the Nusselt numbers for single-phase flow and convective flow boiling become equivalent. This equivalence is shown to be also valid for the case of non-boiling two-phase flows (air-water) and flow condensation. This implies that the well-known heat transfer coefficient model by Dittus-Boelter can be considered as a unified heat transfer coefficient model when the suggested two-phase flow Reynolds ($Re_{2\phi}$) and Prandtl ($Pr_{2\phi}$) numbers are used instead.

See supplementary material for the details of the heat transfer experiments and experimental database.

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