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Dominant dimensionless groups controlling heat transfer coefficient during flow condensation inside pipes

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ABSTRACT

A study of the dominant dimensionless groups describing flow condensation at high mass fluxes inside circular pipes was performed by gathering experimental data from the literature covering a wide range of pipe diameters and fluid properties. A new model is presented based on an equivalent Reynolds number defined as the sum of the superficial liquid and vapour Reynolds numbers and an equivalent Prandtl number given as the sum of the liquid and vapour Prandtl number weighted with the thermodynamic quality. This simple model is able to capture the heat transfer coefficient of channels from a hydraulic diameter of 67 μm up to pipes of 14.45 mm in diameter for several fluids.

1. Introduction

In spite of the extensive work in flow condensation in small diameter tubes, the general characteristics of the heat transfer phenomena and which are the dominant mechanisms remain an elusive question. A large number of models has been suggested in the past decades which have been tested against large experimental data bases. A detailed discussion of most of the available models and prediction capabilities has been reviewed by several authors [1–5]. In general, there is no agreement about the dimensionless groups that flow condensation models should include, and different combinations of dimensionless groups have been suggested. In some cases more than 10 dimensionless groups and adjusted parameters were considered. This fact contrast with the case for single-phase heat transfer coefficient in pipes. The equation attributed to Dittus-Boelter and McAdams [6], following the equation proposed by Nusselt in 1910 (as cited in [7]) based on similarity theory, contains only 2 dimensionless groups and 3 adjusted parameters,

$$Nu = \frac{hD}{k} = f_1(Re)f_2(Pr) = CRe^n Pr^m \quad (1)$$

where h is the heat transfer coeficient, D the diameter of the pipe, k the thermal conductivity of the fluid, $Re = GD/\mu$ the Reynolds number (with G the mass flux and μ the dynamic viscosity), $Pr = c_P\mu/k$ the Prandtl number (with c_p 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.

The goal of this work is to identify which dimensionless groups can describe the heat transfer coefficient during flow condensation in pipes. This work focuses on high mass fluxes in order to avoid effects related to flow stratification. A major step was done in this work in collecting data from different experiments including different pipe diameters and fluid properties. The main feature of the data base is that for two selected fluids, the data set was created containing different pipe diameters but limited to a given pressure. In this way the effect of the changes in fluid properties can be neglected limiting the data set to the study of the hydrodynamic effects, i.e. $f_1()$. Another data base was created for considering the effect of the fluid properties, i.e. $f_2()$. This approach has allowed us to explore the dominant dimensionless groups and their effect on the heat transfer coefficient in a novel way.

1.1. Literature review

An overview of some selected models is shown in Table 1. A complete discussion of most of the available models can be found in [1–5].

One of the first correlations available in the literature regarding flow condensation inside pipes was developed by Crosser [8] considering that the condensate forms an annular ring surrounding a turbulent vapour core. The vapour velocity and the viscosity of the liquid phase was used for defining an equivalent superficial vapour Reynolds number. Crosser mentioned that previous

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Nomenclature

$Nu = hD/k$ Nusselt number
$Pr_L = c_{PL}\mu_L/k_L$ liquid Prandtl number
$Pr_V = c_{PV}\mu_V/k_V$ vapour Prandtl number
$Pr_{2\phi} = Pr_L(1-x) + Pr_V\, x$ two-phase Reynolds number
$Re_L = G(1-x)D/\mu_L$ superficial liquid Reynolds number
$Re_V = GxD/\mu_V$ superficial vapour Reynolds number
$Re_{2\phi} = Re_L + Re_V$ superficial two-phase Reynolds number
$Re_{L0} = GD/\mu_L$ liquid Reynolds number
$Re_{V0} = GD/\mu_V$ vapour Reynolds number
$Re_V^* = GxD/\mu_L$ pseudo vapour Reynolds number
x thermodynamic quality
D hydraulic diameter
h heat transfer coefficient
L liquid
V vapour

Table 1
Some previous flow condensation heat transfer correlations.

Author(s)	Equation
Crosser [8]	$Nu = CPr_L^{1/3}\left\{G\left[x\left(\frac{\rho_L}{\rho_V}\right)^{0.5}\right]\frac{D}{\mu_L}\right\}^n$ $C = 0.0265$ $n = 0.8$ for $\frac{Gx}{\mu_L} > 60000$ $n = 0.2$ for $\frac{Gx}{\mu_L} < 60000$
Akers et al. (1958)	$Nu = CPr_L^{1/3}\left\{G\left[(1-x) + x\left(\frac{\rho_L}{\rho_V}\right)^{0.5}\right]\frac{D}{\mu_L}\right\}^n$ $C = 0.026$ $n = 0.8$ for $Re_{eq} > 50000$ $C = 5.3$ $n = 1/3$ for $Re_{eq} < 50000$
Cavallini and Zecchin [64]	$Nu = 0.05Re_L^{0.8}Pr_L^{0.33}\left[1 + \frac{x}{(1-x)}\left(\frac{\rho_L}{\rho_V}\right)^{0.5}\right]^{0.8}$
Shah [65]	$Nu = 0.023Re_{L0}^{0.8}Pr_L^{0.4}\left[(1-x)^{0.8} + \frac{3.8x^{0.75}(1-x)^{0.04}}{P_R^{0.38}}\right]^{0.8}$
Tandon et al. [66]	$Nu = CPr_L^{1/3}\left(\frac{h_{LV}}{C_p\Delta T}\right)\left(\frac{Gx}{\mu_L}\right)^n$ $C = 0.084$ $n = 0.67$ for $Re_V^* = \frac{Gx}{\mu_L} > 30000$ $C = 23.1$ $n = 1/8$ for $Re_V^* < 30000$
Dobson and Chato [21]	$Nu = 0.023Re_L^{0.8}Pr_L^{0.4}\left(1 + \frac{2.22}{X_{tt}^{0.89}}\right)$
Cavallini et al. [38]	$Nu = 0.023Pr_L^{0.4}Re_{L0}^{0.8}\left[1 + 1.128x^{0.8170} \times \left(\frac{\rho_L}{\rho_G}\right)^{0.3685}\left(\frac{\mu_L}{\mu_G}\right)^{0.2363}\left(1 - \frac{\mu_G}{\mu_L}\right)^{2.144}Pr_L^{-0.1}\right]$
Bohdal et al. [67]	$Nu = 25.084Re_L^{0.258}Pr_L^{-0.495}P_R^{-0.258}\left(\frac{x}{1-x}\right)^{0.266}$
This work	$Nu = 0.023(Pr_L(1-x) + Pr_Vx)^{0.3}(Re_L + Re_V)^{0.8}G > 200\ \mathrm{kg/m^2\,s}$

research work had already identified the direct effect of the vapour velocity on the heat transfer coefficient, e.g. Jakob et al. (1932), Schmidt (1937) and Carpenter and Colburn (1951) (as cited in [8]). The experimental data from Crosser showed good agreement with his model and the exponent n of the equivalent Reynolds number was 0.2 at low Reynolds number approaching 0.8 at high Reynolds number. Akers et al. (1958) (as cited in [9]) proposed a model based on the idea that the vapour core might be replaced with a liquid flow that produces the same liquid-vapour interfacial shear stress. An equivalent Reynolds number was defined and then replaced in the single phase Sieder-Tate (1937) equation [10]. The local condensation heat transfer coefficient is given as

$$Nu = \frac{hD}{k_L} = CPr_L^m\ Re_{eq}^n \tag{2}$$

with the equivalent Reynolds number defined as

$$Re_{eq} = G\left[(1-x) + x\left(\frac{\rho_L}{\rho_V}\right)^{0.5}\right]\frac{D}{\mu_L} \tag{3}$$

with G the mass flux, x the thermodynamic quality, and ρ_L and ρ_V the liquid and vapour density respectively. The model contains 3 adjusted parameters, $m = 1/3$ and $C = 0.026$ and $n = 0.8$ for $Re_{eq} > 50000$, and $C = 5.03$ and $n = 1/3$ for $Re_{eq} < 50000$.

The influence of the vapour flow rate on the heat transfer coefficient has also been acknowledged by Goodykoontz and Dorsch [11,12] who studied condensation of steam in a vertical tube. The experimental data was correlated in terms of the product of the quality and the square of the total mass flux, although no general model was provided.

Cavallini and Zecchin [64] proposed a similar model to the one from Akers et al. (1958) but considering another value for the scaling constant C probably as a consequence of the different fluids studied. Shah [65] suggested a dimensionless correlation for predicting heat transfer coefficient by considering the similarity between the mechanisms of film condensation and boiling without bubble nucleation. The model results in an expression containing 9 adjustable parameters and 5 dimensionless groups. Compared to the previous two discussed models, the reduced pressure $P_R = P/P_C$ was introduced in the model while the dependency on the liquid-vapour density ratio was removed. Tandon et al. [66] proposed a modification to the Akers et al. (1958) correlations based on condensation experiments for R12 and R22. The direct dependency of the heat transfer coefficient on the average vapour mass velocity was acknowledged by plotting the heat transfer coefficient versus the vapour mass velocity. A high vapour mass velocity results in a higher turbulence of the liquid film increasing the heat transfer coefficient. They observed a change in slope in the heat transfer coefficient versus the vapour Reynolds number defined in terms of the liquid viscosity. The change in the slope is attributed to changes from annular and semi-annular flow to wavy flow. The model introduced the Jakob number defined as