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# Simple and general correlation for heat transfer during flow condensation inside plain pipes

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ABSTRACT

This work proposes a new general and simple model to determine the local flow condensation heat transfer coefficient inside plain pipes. The model considers two regimes corresponding to high mass fluxes and/or high thermodynamic qualities and low mass fluxes and/or low thermodynamic qualities. For each region, a new model is suggested which resembles the single-phase heat transfer coefficient model but defining an equivalent Reynolds number in terms of the sum of the superficial liquid and vapour Reynolds numbers. The models consider that the superficial vapour Reynolds number plays a major role in controlling the heat transfer coefficient. The model is able to predict the heat transfer coefficient from channels with a hydraulic diameter of 67 μm up to pipes with a hydraulic diameter of 20 mm for several fluids. No noticeable effect of the diameter of the channel, shape or fluid properties on the heat transfer coefficient has been observed for the studied cases.



## 1. Introduction

Condensers based on mini/micro-channels are of great relevance and interest in connection with the growing demand of heat exchangers for several applications ranging from condensers of cooling equipment and air conditioning systems, horizontal tubular evaporators of water-desalinating thermal units, heaters of power systems, heat pipes, etc. Furthermore, in some applications heat exchangers are the main component in cryogenic processes like separation units and liquefaction of natural gas plants, and the design and performance of the unit can affect some other major equipments like compressors and drivers [1].

During the last decades a large number of models of flow condensation inside pipes has been proposed and tested against experimental data bases as reviewed in the literature by several authors, e.g. [2–5]. In spite of the extensive work, the general characteristics of the heat transfer phenomena and which are the dominant mechanisms remain an elusive question. This fact is observed in the different dimensionless groups considered in the models suggested under different assumptions. In some cases more than 10 dimensionless groups and adjusted parameters are needed for predicting the experimental data.

The goal of this work is to present a condensation heat transfer coefficient model for flow inside plain pipes. The approach is based on identifying the dominant dimensionless groups and their effect on the heat transfer coefficient of experimental data gathered from the literature. In this work, it will be shown that the heat transfer coefficient shows a distinctive behaviour at high mass fluxes and/or high thermodynamic qualities and low mass fluxes and/or low thermodynamic qualities, fact that is considered for proposing a model for each region.

### 1.1. Literature review

Extensive reviews of heat transfer models for flow condensation inside pipes can be found in [2–5]. In this section, the focus will be on highlighting the difference in the selected dimensionless groups considered in some selected models. Tables 1 and 2 summarise the models discussed in this section. A common reference for most heat transfer models is the single-phase heat transfer coefficient in pipes. The equation attributed to Dittus-Boelter and McAdams [6], following the equation proposed by Nusselt (1910) based on similarity theory (as cited in [7]), contains only 2 dimensionless groups and 3 adjusted parameters,

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

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 $G$ the mass flux and $\mu$ the dynamic viscosity), $Pr = c_P\mu/k$ the Prandtl number (with $c_p$ the specific heat). The exponent $m$ is

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**Nomenclature**

$c_P$ heat capacity [J kg$^{-1}$ K$^2$]
$G$ mass flux [kg m$^{-2}$ s$^{-1}$]
$P$ pressure [kPa]
$Nu = hD/k$ Nusselt number [–]
$Pr = c_P\mu/k$ Prandtl 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 [–]
$\rho$ density [kg m$^{-3}$]
$\mu$ viscosity [Pa s]
$x$ quality [–]
$L$ liquid
$V$ vapor

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 or influence of the fluid properties $f_1(.)$ and $f_2(.)$ respectively. Several models were suggested later based on larger experimental data bases for example for taking into account the change of the fluid properties with temperature [8,9], effect neglected in the discussed model, or the shape of the pipe [10].

One of the first correlations available in the literature regarding flow condensation inside pipes was developed by Crosser [11] assuming that the condensate form an annular ring surrounding a turbulent vapour core. The model considers the vapour velocity and the viscosity of the liquid phase for defining an equivalent superficial vapour Reynolds number. Crosser has pointed out to previous research work identifying the direct effect of the vapour velocity on the heat transfer coefficient, namely Jakob, Erk and Eck (1932), Schmidt (1937) and Carpenter and Colburn (1951) (as cited in [11]). The experimental data show good agreement with the model and the exponent $n$ of the equivalent Reynolds number was 0.2 at low Reynolds number approaching 0.8 at high Reynolds number. Rosson [12] developed a model for semi-stratified and laminar annular flow similar to the model of Crosser. The model considers that the heat transfer coefficient is a function of the thickness of the liquid boundary layer that depends on the temperature difference across the liquid film. Akers, Deans and Crosser (1958) (as cited in [12]) 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 [13]. 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 $\rho_L$ and $\rho_V$ the liquid and vapour density respectively. The model contains 3 adjusted parameters, $m = 1/3$ and $C = 0.026$ and $n = 0.8$ for $Re_{eq} > 50,000$, and $C = 5.03$ and $n = 1/3$ for $Re_{eq} < 50,000$.

The influence of the vapour flow rate has also been acknowledged by Goodykoontz and Dorsch [14,15] who studied condensation of steam in vertical tubes. The experimental data was correlated in terms of the product of the quality and the square of the total mass flux, although, a general model was not provided.

Cavallini and Zecchin (1974) [16] proposed a similar model to the one from Akers, Deans and Crosser (1958) but considering another value for the scaling constant C probably as a consequence of the different fluids studied. The model was suggested for dominant annular flow regime.

It is possible to see that the first available models were considering a direct influence of the vapour flow rate on the heat transfer coefficient. This influence observed in the experiments was implemented in the models by including the product of the total mass flux and the quality, i.e. $G\,x$. Later models have considered alternative descriptions and the dependency on the vapour velocity was introduced in an indirect manner in some cases by multipliers applied to a single phase Dittus-Boelter heat transfer model.

Shah (1979) [17] suggested a dimensionless correlation for predicting heat transfer coefficient during film condensation inside pipes 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. The reduced pressure $P_R = P/P_C$ was introduced in the model while the dependency on the liquid-vapour density ratio was removed compared with the previous two discussed models. The model was extended to two regimes in [18]. The regime transition is defined in terms of the dimensionless vapour velocity, $J_G$, and Shah's correlating parameter, Z, given as

$$J_G^T = \frac{1}{2.4Z + 0.263} \text{ with } Z = \left(\frac{1}{x} - 1\right)^{0.8} P_R^{0.4} \tag{4}$$

and

$$J_G = \frac{Gx}{(g\ D\ \rho_g(\rho_l - \rho_g))^{0.5}} \tag{5}$$

Tandon et al. (1995) [19] proposed a modification to the Akers, Deans and Crosser (1958) correlations based on condensation experiments for R12 and R22 acknowledging the direct dependency of the heat transfer coefficient on the average vapour mass velocity. It is considered that a high vapour mass velocity results in a higher turbulence of the liquid film increasing the heat transfer coefficient. They also observed a change in slope in the heat transfer coefficient versus the equivalent vapour Reynolds number defined in terms of the liquid viscosity, i.e. [11]. 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 $(h_{LV}/C_P\Delta T)$ where $\Delta T$ is the temperature difference across the condensate film. The model is presented with two branches where the exponent changes from $n = 0.67$ to $n = 0.125$ from the defined shear-controlled flow (annular and semi-annular) to the gravity-controlled flow (wavy flow).

Dobson and Chato (1998) [20] suggested a model considering an annular flow and a wavy flow regime. For the annular flow regime, the model is similar to the single-phase flow heat transfer coefficient equation but multiplied by a term including the Martinelli parameter $X_{tt}$. For the wavy flow regime, the model considers a separate heat transfer contribution by the film condensation in the upper part of the horizontal tube from the forced-convective heat transfer in the bottom pool. The model for film condensation includes the liquid Jacob and the Galileo number. The boundary for