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New model for heat transfer calculation during film condensation inside pipes

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ABSTRACT

In this paper a new model is presented for heat transfer calculation during film condensation inside pipes. This new model has been verified by comparison with available experimental data of a total of 22 different fluids, including water, various refrigerants and organic substances, which condense inside horizontal, vertical and inclined tubes. The model is valid for a range of internal diameters ranging from 2 mm to 50 mm, reduced pressure values ranging from 0.0008 to 0.91, *Pr* values for the liquid portion of the condensate from 1 to 18, values of Reynolds number for the liquid portion between 68 and 84827, and for the portion of the steam between 900 and 594373, steam quality from 0.01 to 0.99 and mass flux rates in the ranges of 3–850 kg/(m^2 s). The mean deviation found for the data analyzed for vertical and inclined tubes was 13.0%, while for the horizontal tube data the mean deviation was 11.8%. In all cases, the agreement of the proposed model for horizontal, vertical and inclined tubes is good enough to be considered satisfactory for practical design.

1. Introduction

Although there currently are many papers addressing the issue of flow condensation in small diameter tubes, the general characteristics of this heat transfer phenomenon and its dominant mechanisms make this still a difficult issue reaching problem. In the last five decades, a significant number of models have been suggested, which have been subjected to tests with experimental values existing in databases.

A detailed analysis of most of the available models and their prediction capabilities has been provided by several authors [1–6]. However, there is still discrepancy about the dimensionless groups that must govern condensation models, which is why different combinations of variables and dimensionless groups have been suggested.

In some cases more than 10 dimensionless groups and adjusted parameters are considered, however, this fact contrasts with a common basis, and that is that all the existing studies establish as a base relation the expression for the determination of the coefficient of heat transfer in single-phase; the overwhelming majority of cases uses the equation attributed to Dittus-Boelter, which is a follow-up to the model proposed by Nusselt in 1910 based on the similarity theory, which contains only 2 dimensionless groups and 3 adjusted values, as shown in Eq. (1):

$$Nu_L = \frac{hd}{k} = f_1(Re)f_2(Pr) = cRe^m Pr^n \quad (1)$$

In Eq. (1) h is the single-phase heat transfer coefficient, d is the inner diameter of pipe, k is the fluid thermal conductivity, $Re = Gd/\mu$ is the Reynolds number, (with G being the mass flux and μ the dynamic viscosity); $Pr = \mu c_p/k$ is the Prandtl number. The exponent n is suggested to be 0.3 and 0.4 for cooling and heating respectively, while $m = 0.8$ and $c = 0.023$. The model is based on two functional forms that represent the hydrodynamic and thermodynamic effects.

At the present time, in the available and acquaintance literature, only was reported one model that allows evaluating the condensation heat transfer inside of pipes with any geometric orientation (horizontal, vertical and inclined), this model is the Shah's Equation [7,8].

One of the objectives of the present work is to development one model that can best describe the coefficient of heat transfer during the condensation of flow in tubes with any geometric orientation (horizontal, vertical and inclined) and that it allows obtaining

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Nomenclature

c	Dittus-Boelter constant (0.023)
Cl	curve fitting routine, defined in Eq. (21)
C_p	specific heat, J·kg^{-1}·K^{-1}
d	equivalent inner tube diameter, m
G	mass flux, kg·m^{-2}·s^{-1}
g	gravitational acceleration, m·s^{-2}
h_{fg}	latent heat of vaporization, J·kg^{-1}. K^{-1}
h	single-phase heat transfer coefficient, kg·m^{-1}·K^{-1}·s^{-1}
h_T	two-phase heat transfer coefficient, kg·m^{-2}·s^{-3}·K^{-1}
$\bar{h}$	mean heat transfer coefficient, kg·m^{-2}·s^{-3}·K^{-1}
h_C	heat-transfer coefficient assuming all mass to be flowing as liquid, kg·m^{-2}·s^{-3}·K^{-1}
h_{med}	experimental measured value, kg·m^{-2}·s^{-3}·K^{-1}
h_{cam}	heat transfer coefficient determined with Eq. (13)
Ja	Jakob number
J_g	dimensionless velocity
k	fluid thermal conductivity, W·m^{-1}·K^{-1}
L_C	total length of pipe in which condensation occurred
N	numbers of experimental points
Nu	Nusselt number
Nu_E	Nusselt number for experimental data
Nu_L	Nusselt number for single-phase used in Eq. (13)
Nu_T	Nusselt number for two-phase
Nu_{vert}	Nusselt number determined with Eq. (15)
P	fluid pressure, kg·m^{-1}·s^{-2}
Pc	fluid critical pressure, kg·m^{-1}·s^{-2}
Pr	Prandtl number
Pr_L	Prandtl number for single-phase
p_r	reduced pressure
Re	Reynolds number
Re_{eq}	equivalent Reynolds number for two-phase
Re_L	liquid Reynolds number
Re_V	vapor Reynolds number
ΔT	temperature difference across the condensate film
T_{sat}	saturation temperature, °C
T_P	wall temperature, °C
x	thermodynamic vapor quality
X_{tt}	dimensionless Martinelli parameter
y	axial distance from the point where condensation started
Y	coefficient used in Eq. (14)
Z	dimensionless Shah parameter
Greek symbols	
β	thermal expansion coefficient, K^{-1}
μ	dynamic viscosity, kg·m^{-1}·s^{-1}
ρ	density, kg·m^{-3}
υ	liquid kinematic viscosity, m^2·s^{-1}
Subscripts	
L	liquid
Superscript	
m	Dittus-Boelter constant for Re_L
n	Dittus-Boelter constant for Pr_L

one better adjustment index from the experimental available data, than the obtained with the Shah's Equation.

The present work focuses on high mass flows although they include low mass flows in order that the new model allows taking into account the effects related with stratification. An important step was taken in the collection of experimental data provided by various authors, including different tube diameters and dissimilar fluid properties.

In this paper, the results of these efforts are presented, combined in a correlation, which shows good agreement with the data of 22 different types of fluids provided by 39 sources, including water, refrigerants and a variety of organic substances; tubes diameter ranging from 2 to 50 mm; flow rates from 3 to 850 kg/(m^2 s), and reduced pressures $p_r = P/P_c$ ranging from 0.0008 to 0.91.

In this paper only data for macrochannels were presented, which include canals with bigger or similar diameters to 2 mm. While presently there is great interest in microchannels, however, the present investigation is a part of the doctoral education accomplished by the main author, and he is he goes focused to macrochannels. The need to make the study separately of both cases, because as surface tension effects become important in microchannels.

2. Literature review

Table 1 shows a summary of selected models available in the current technical literature for condensation heat transfer calculations. A complete discussion of most of the available models can be found in [1–6].

Carpenter and Colburn [9] developed one of the first correlations available in the known literature to determine the heat transfer by condensation inside tubes. The condensate forms an annular ring surrounding a turbulent steam core. The vapor velocity and viscosity of the liquid phase were used to define an equivalent surface vapor Reynolds number.

Carpenter and Colburn mentioned that in previous research works several authors had already identified the direct effect of steam velocity on the heat transfer coefficient, among them Goodykoontz and Dorsch [10,11]. The experimental data obtained by Carpenter and Colburn showed a good agreement with its model, for which $m = 0.2$ for low Reynolds numbers ($Re < 10^3$) and $m \approx 0.8$ for large Re numbers ($Re > 10^3$). Subsequently in the work of Rosson [13], a model was proposed based on the idea that the steam core could be replaced with a liquid flow that produces the same interfacial cut-off stress between liquid and vapor. An equivalent Reynolds number was defined, Re_{eq}, and replaced in the single–phase Eq. (1) to its homologous term.

The new way to determine the heat transfer coefficient of local condensation was given by Eq. (2):

$$Nu_L = \frac{hd}{k} = cRe_{eq}^m Pr_L^n \tag{2}$$

with the equivalent Reynolds number defined by Eq. (3):

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

In Eq. (3), μ_L is the liquid dynamic viscosity, x is the steam quality, ρ_L is the liquid density and ρ_V is the steam density. Eq. (2) contains 3 adjusted parameters, which will be dependent on the value of Re_{eq}, where, for $Re_{eq} \geqslant 5 \times 10^4$, $m = 4/5$ and $c = 0.026$, and for $Re_{eq} \leqslant 5 \times 10^4$, $m = 1/3$ and $c = 5.03$.