International Journal of Heat and Mass Transfer 106 (2017) 280–288

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

A simplified energy dissipation based model of heat transfer for subcooled flow boiling

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ARTICLE INFO

Article history:
Received 23 July 2016
Received in revised form 22 October 2016
Accepted 22 October 2016

Keywords:
Subcooled flow boiling
Heat transfer
Modeling

ABSTRACT

In the paper a model is presented based on energetic considerations for subcooled flow boiling heat transfer. The model is the extension of authors own model developed earlier for saturated flow boiling and condensation. In the former version of the model we used the heat transfer coefficient for the liquid single-phase as a reference level, due to the lack of the appropriate model for heat transfer coefficient for the subcooled flow boiling. That issue was a fundamental weakness of the that approach. The purpose of present investigation is to fulfil this drawback. Now the reference heat transfer coefficient for the saturated flow boiling in terms of the value taking into account the subcooled flow conditions. The wall heat flux is based on partitioning and constitutes of two principal components, namely the convective heat flux and partial evaporation heat flux of the liquid replacing the detached bubble. Both terms are accordingly modeled. The convective heat flux is regarding vapour bubbles travelling longitudinally and the liquid moving radially – liquid pumping. The results of calculations have been compared with some experimental data from literature showing a good consistency.

1. Introduction

The subcooled flow boiling for a long time is perceived as one of the most effective ways of removal of large heat fluxes due to a large temperature difference and presence of boiling in the flow. The phenomenon found application in various areas of technology where efficient cooling is required. An example of such application is nuclear reactor cooling, medical applications where cooling of neutron generators used in treatment of tumors is necessary, testing of materials, cooling of electronic equipment or cooling of gas turbine nozzles. Understanding of the physics of local boiling in subcooled liquids flowing inside heated channels is still unsatisfactory. A number of papers in the literature are devoted to this issue but the complexity of the process makes the analysis of the issue very challenging. Several modeling approaches have been developed to predict the heat transfer rate during subcooled flow boiling. Such models can be generally divided into two categories, namely purely empirical correlations for heat flux calculations or the formulas based on mechanistic models. The empirical approaches express the wall heat flux or partitioning of the wall heat flux. Non-consistent empirical correlations for heat transfer coefficient are used for expressing a particular wall heat flux partitioning. Non-consistency partially stem from the fact that empirical correlations are generally limited to particular flow conditions. Hence empirical correlations do not include modeling of the heat transfer mechanisms. The alternative are the mechanistic models which are capable of determining the particular heat flux components individually. Usually two main aspects of the problem are studied, firstly, the inception of subcooled boiling and its distance from the inlet of the channel and, secondly, heat transfer from the wall to fluid. Hence empirical correlations for wall heat flux partitioning can only provide information regarding how the wall heat flux is to be partitioned. They cannot be used for the prediction of the wall heat flux itself. The mechanistic models, on the other hand, which are based on the relevant heat transfer mechanisms occurring during the boiling process, have the capability for individual determination of each of the relevant heat flux components. Hence the mechanistic models can be used for both the prediction of the wall heat flux and the partitioning of the wall heat flux between the liquid and vapor phases. An excellent review of literature on the topic of empirical correlations for heat flux, empirical correlation for partitioning of wall heat flux and mechanistic models for prediction of wall heat flux and partitioning can be found in Warrier and Dhir [1].

The objective of the present work is to devise a model for calculation of the convective part of heat transfer coefficient in

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2016.10.077
0017-9310/

Nomenclature

A	projection area, m^2
a	thermal diffusivity $[m/s^2]$
B	blowing parameter [−]
Bo	Boiling number, $B = q/(G\ h_{lv})$
c	specific heat [J/(kg K)]
C	constant
D_h	hydraulic diameter [m]
E	enhancement factor
f	departure frequency. [1/s]
F	reduction factor, enhancement factor
g	gravitational acceleration $[m^2/s]$
G	mass velocity, $[kg/(m^2s)]$
h_{lv}	latent heat [J/kg]
$\dot{m}$	mass flow rate, kg/s
p	pressure $[N/m^2]$
P	empirical correction, perimeter
q	heat flux $[W/m^2]$
R	radius [m]
Re	Reynolds number, $Re = G\ D_h/\mu_l$
S	suppression factor,
T	temperature [°C]
t	time [s]
u	velocity [m/s]
w	superficial velocity, [m/s]
V	volume $[m^3]$
x	quality [–]
z	wall normal coordinate [m]

Greek symbols

α	heat transfer coefficient $[W/(m^2K)]$
λ	thermal conductivity [(W/mK)]
δ	penetration depth, boundary layer thickness, [m]
ε	q_a/q_{ev}, [−]
μ	dynamic viscosity [kgm/s]
ρ	density $[kg/m^3]$
Φ	enhancement factors [−]
σ	surface tension $[kg/s^2]$
τ	shear stress $[N/m^2]$

subscripts

a	agitated
b	bulk
con	convective
ev	evaporative
in	inlet
l	liquid phase
lp	liquid pumping
onb	onset of boiling
OSV	onset of significant voids
O	reference
p	constant pressure
Pb	pool boiling
ref	reference
sat	saturation
S	subcooled
sub	subcooling
TP	two-phase
TPB	two-phase flow boiling
v	vapour
w	wall

subcooled flow boiling developed on the basis of energy dissipation in the flow. The presented approach belongs to the group of mechanistic treatments to determination of the contribution of convective heat transfer in subcooled flow boiling. The resultant model of subcooled flow model is a modification to the saturation flow boiling developed earlier by the authors, presented in detail in [2–4]. In addition the heat flux due to evaporation has been determined.

The beginning of the nucleate boiling starts at the location where vapour can exist in the steady state condition on the heated surface without condensation. The greater the fluid energy (longitudinally) the bubbles can grow until departure from the heated surface and penetration into the liquid (end of wall voidage region in Fig. 1).

The region of the subcooling zone can be either large or small in relation to the fluid properties, mass flux, pressure and heat flux. It is a non-equilibrium region in which the quality and void fraction are assuming positive non-zero values but the liquid temperature is below the saturation temperature. Modeling of such phenomenon represents significant difficulties.

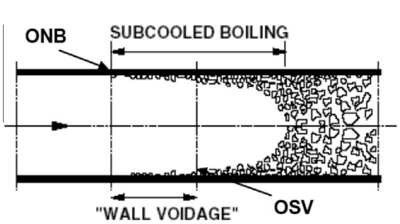

Fig. 1. Incipience of flow boiling.

One of the earliest models for empirical determination of partitioning of wall heat flux was developed by Griffith et al. [5]. Based on visual observations during experiments, they identified two distinct boiling regions, see Fig. 2, namely a highly subcooled region with a low void fraction, i.e. "region I", and a slightly subcooled region with a significant void fraction – "region II". Region I extends over the heated area between the onset of nucleate boiling (ONB) and the onset of significant voids (OSV) locations, while region II begins at OSV and extends until saturated boiling begins in the entire cross-section of the flow. They used the arithmetic superposition of single phase forced convection heat flux, q_l, and fully developed pool boiling heat flux, q_{Pb}:

$$q_w = q_l + q_{Pb} \tag{1}$$

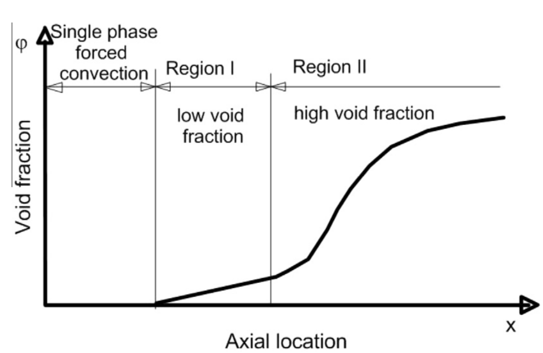

Fig. 2. Variation of void fraction with axial distance.