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Heat transfer and bubble formation in pool boiling: Effect of basic surface modifications for heat transfer enhancement

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Abstract

The present project within the joint research program on fundamentals of boiling heat transfer aims at better understanding of the basic processes which produce heat transfer enhancement in nucleate pool boiling. In the experiments, heater surfaces with two kinds of modifications for enhancement in the form of macro cavities with comparatively simple shapes are used in order to link experimental results of bubble formation and heat transfer to the geometric features of the cavities without additional assumptions, and particularly to resolve their overall effect on heat transfer into local convective or evaporative contributions without introducing severe simplifications.

The very accurate results on local heat transfer obtained so far around the circumference of the horizontal test tube with and without macro cavities for enhancement, and their combination with the local events connected to growing, departing and sliding bubbles are suitable to interpret the basic convective and evaporative processes which produce heat transfer enhancement in nucleate pool boiling.

Keywords: Heat transfer; Bubble formation; Pool boiling; Enhancement; Heater surface modification

1. Introduction

1.1. Related previous work of other researchers and of the authors

Within the joint research program on fundamentals of boiling heat transfer presented in this special issue, the project discussed here tries to contribute to better understanding of the basic processes which produce heat transfer enhancement in nucleate pool boiling. Since the first high-performance heater surface configurations were patented in the late 1960's, a great variety of evaporator tubes with enhanced surfaces have been developed, that can be divided in two main groups, one based on integral-fin tubes with modified fins to form reentrant grooves or tunnels ("structured surfaces"), see Fig. 1, and another based on plain tubes with sintered porous metallic matrix bonded to the tube surface ("porous surfaces"), see the example in Fig. 2.

A great number of experimental investigations of enhanced pool boiling heat transfer have been reported in the literature, see e.g. the reviews in [4–8]. Most of those measurements were performed, however, at saturation pressures p_s near atmospheric, and the correlations developed on the basis of the experimental results—also several containing a physical model of enhanced heat transfer fitted to the particular geometric configuration of the heating surface—mainly use the data gained at atmospheric pressure, but claim applicability in a broader sense, see, e.g. [9–20]; very detailed, but highly complex models have been developed recently by Chien and Webb [21] and Liter and Kaviany [22].

A good and comprehensive example of experimental results for pool boiling heat transfer at pressures near atmospheric from different structured or porous surfaces has been taken from Memory et al. [3] in Fig. 3. The data for refrigerant R114 ($CF_2Cl{\cdot}CF_2Cl$) exhibit the typical improvement of heat transfer from *structured* surfaces over plain tubes, with heat transfer coefficients α being higher by a factor of three to five at low heat fluxes q for modified integral-fin tubes (see the dashed lines

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Nomenclature

d	bubble diameter	mm
D	tube diameter	mm
N	number of bubbles or active nucleation sites per analyzed area or time interval, 1/seq, 1/area	
N/A	density of active nucleation sites	cm^{-2}
p	pressure	bar
p^*	reduced pressure, $= p_s p_c^{-1}$	
P_a, R_a	arithmetic mean roughness height acc. to DIN EN ISO 4287 (P_q, P_p, $P_{p,m}$, P_t, P_z (µm) = other standardized roughness parameters)	µm
q	heat flux	$kW \cdot m^{-2}$
t	time	ms
T	temperature	°C
ΔT	superheat	K

Greek symbols

α	heat transfer coefficient	$kW \cdot m^{-2} \cdot K^{-1}$
Δ	difference	
φ	azimuthal angle	°

Characteristic parameters

Nu	Nusselt number
Gr	Grashof number
Pr	Prandtl number

Subscripts

A	at detachment
a, OD	outer diameter
B	bubble
c	in the critical state
cum	cumulated for an extended time interval, typically 500 ms
el	electrical
fpi	fins per inch
K	at the base of fins
lam	laminar
loc	local
m	mean
min	minimum
s	in the saturated state
sim	simultaneously ≡ within time interval of 1 ms between succeeding high speed video frames

for several types of GEWA-finned tubes). The improvement by the *porous* surface (High Flux tube) and the Thermoexcel-HE, Turbo-B tubes, (solid lines) is much better, but their α, q-behaviour is entirely different, thus ending up with basically the same heat transfer coefficients for all kinds of surfaces at high heat fluxes near 100 $kW \cdot m^{-2}$.

Investigations with Propane, Propylene and several refrigerants boiling on structured tubes in a wide pressure range (up to 50% of the pertaining critical pressures p_c) reveal significant deviations from the rather uniform increase of α with q for the structured tubes at atmospheric pressure in Fig. 3. In the case of GEWA-TX tubes, in particular,

(a) α-values are much higher than expected and almost do not increase with q at high reduced pressures p_s/p_c and small heat fluxes, and
(b) at low pressures p_s/p_c and high heat fluxes, α becomes independent of pressure and heat flux,

see the data in Fig. 4 (from [1]) at $p^* = p_s/p_c = 0.147$ (a) and p^* from 0.055 to 0.147 (b). Both effects could be explained using photos of bubble formation: In the first case, stable vapour–liquid interfaces bridge the narrow gaps of 0.23 mm (see Fig. 1) between the tops of neighbouring fins at the top of the (horizontal) tube and trap vapour in the tunnels between the fins, and in the second case, heat transfer is dominated by the restricted release of the great amount of vapour produced in the tunnels at high heat flux, similar to single-phase forced convective heat transfer which is independent of heat flux and pressure, while at the highest pressure in Fig. 4, nucleation exists all over the tube surface and α increases with q and p^*, also at the highest heat flux investigated (for more details, see [6,23]).

Both peculiarities also occurred with Propylene and refrigerant R134a ($CF_3 \cdot CH_2F$) boiling on the TX-tube, while the first effect (a) was not found with another refrigerant (R152a, $CHF_2 \cdot CH_3$) within the same ranges of q and p^* [1,24–26]. The latter also holds for all fluids investigated with the GEWA-YX tube, and the second effect (b) was less pronounced with these tubes, obviously because of the somewhat wider gaps between the fins (0.34 mm) and their different shape (see Fig. 1).

Fig. 5 demonstrates that pressure dependencies of the heat transfer coefficient α_{OD} for various types of structured surfaces may differ significantly from each other, also within the range of intermediate heat fluxes ($q_{OD} = 20\ kW \cdot m^{-2}$), and that enhancement vanishes at high reduced pressures p^*. Fig. 5 from [27] has been supplemented by data from Fig. 6 for a porous surface showing that its superiority to the structured surfaces disappears at high reduced pressures in a similar way as with increasing heat flux at constant pressure for the other porous surface in Fig. 3.

Measurements with Propane boiling on a CuNi-tube (8 mm OD) with aprox. 300 µm thick, plasma sintered Ni-based porous layer are presented in Fig. 6 as example of heat transfer from porous surfaces over an extended range of reduced pressures (and heat fluxes) [28]. The porous layer was manufactured at DLR-Institute of Thermodynamics, University of Stuttgart, and the tube was assembled in Paderborn and tested in the apparatus and in between the experiments of this paper.

The data show a systematic increase of heat transfer coefficient α with heat flux q and reduced pressure p^* which