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Self-induced thermocapillary convection in film boiling heat transfer from a vertical surface in saturated conditions and viscous regimen

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ABSTRACT

Self-induced thermocapillary convection and its significance with regard to film boiling heat transfer from a vertical saturated surface is discussed. Utilizing a simplified geometrical model, an analytical expression (multiplier factor) for the vapor film thickness and heat transfer coefficient corrected by thermocapillary phenomena was derived. The above equation is a new theoretical viewpoint for the enhancement in the heat transfer coefficient observed in the experimental data in the viscous regime and agree qualitatively with available experimental measurements made on R113 coolant.

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1. Introduction

Whereas a lot of research has been done on nucleate boiling of various fluids (Fujita, 1992), and film boiling (Gian et al., 2009; Arias and Reventos, in press-a; Sarma et al., 1994), and special situations as electrical and magnetohydrodynamics aspects (Verplaetsen and Berghmans, 1999; Arias, in press-b). As regards the thermocapillary convection under imposed temperature gradients, in nucleate boiling we have the works from Zell (1991), Vogel (1994), and the recent works of Bennacer et al. (2004), Buffone et al. (2004), Sefiane and Ward (2007) among others. On the other hand, experimental investigations of Buffone et al. (2005) demonstrated the thermocapillary convection self-induced from vertical interface where an interesting results is that the convection is driven by the surface-tension gradient generated by the self-induced temperature difference. A similar situation is proposed in the present work, where we have a vertical interface and thermocapillary convection driven by evaporation is self-induced, as opposed to thermocapillary convection under imposed temperature gradients. Despite mentioned above, film boiling has received much less attention than thermocapillary-nucleate boiling and not works theoretical and/or experimental have been reported as far as the author know.

Then, as mentioned above, yet the thermocapillary-film-type phenomenon appears susceptible to attack from the theoretical viewpoint.

Here, an attempt is made to relate the peculiar behavior of the heat transfer in vertical geometries with the thermocapillary phenomenon, where nowadays physical models is still missing.

1.1. Pool film boiling

At or near boiling crisis a film boiling occurs when the heat surface is blanketed with vapor film and the heat transfer coefficient rapidly decreases, the film hinders heat transfer, and resulting heat flux is usually small compared with the values observed during nucleate boiling . In most industrial process for vaporizing liquids, film boiling is avoided. However, when large temperature differences are encountered, such as when an ordinary liquid contacts a very hot solid or when vapor is used to heat liquids with low boiling points, film boiling may occur.

2. Theoretical background

On the presence of a thermal gradient, there are two driving forces. In one hand

- Thermal expansion.

On the other hand

- Temperature surface-tension coefficient.

The former effect-thermal convection, scaled by the dimensional Grashof number is proportional to the fourth power layer thickness; hence, for small thickness the thermocapillary convection dominates. In the present paper, where the film boiling can be considered as very small, the thickness of a film boiling in first approximation can be calculated as

$$\zeta \approx \frac{\kappa_v}{h} \tag{1}$$

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Nomenclature

b	*-in* growth coefficient
g	gravity constant
Gr	Grashof number: $Gr = \alpha g l^3 \Delta / \nu^2$
h	heat transfer coefficient
H	liquid depth
i	co-ordinate parallel to the boundary layer
k	wave number
l	characteristic length
m	wave number
n	wave frequency
p	pressure
R	Radius of curvature of the interface
t	time
T	temperature
u	velocity
x	co-ordinate parallel to the wall
z	co-ordinate normal to the wall $\Delta T_{sat} = T_W - T_{sat}$
$\nabla_i T$	thermal gradient in the parallel direction to the interface vapor–liquid.
Greeks	
α	thermal dilatation coefficient $\alpha = (1/\rho)\partial\rho/\partial T$
β	expansion term : $\beta = 1 - \frac{\rho_v}{\rho_l}$
κ	thermal conductivity
η	dynamical viscosity
λ	wave length
ν	kinetic viscosity
η	distance perpendicular to liquid–vapor interface
ω	angular frequency
ρ	density
σ	surface tension
σ'	temperature surface-tension coefficient $\sigma' = \frac{\partial\sigma}{\partial T}$
ζ	vapor film thickness
Δ	temperature differentia
Subscripts	
d	the "most dangerous" wave length defined by Eq. (27)
f	film
v	vapor
l	liquid
vl	vapor–liquid interface
s	the "smallest" wave defined by Eq. (26)
sat	saturation
sub	subcooled
w	wall
0	initial-reference value

where ζ and κ_v are the vapor film thickness and thermal conductivity, respectively and the h is the heat transfer coefficient. Here, we have in mind thin films above $h \approx 300\,\mathrm{W/m^2\,K}$, i.e., vapor film thickness of the order of $\zeta \leqslant 10^{-5}$ m, in this case thermocapillary convection can be largely dominant as it will be demonstrated.

3. Basic assumptions

We have a heat surface which is blanketed with a vapor film with constant density ρ_v; surface tension σ_f; and velocity u_v, the film is separated of the rest of the saturated liquid (upper region) by the boundary layer, which is at saturation temperature and the upper region (saturated liquid) with constant density ρ_l, surface tension σ_l, and velocity u_l.

The actual shape of the liquid–vapor interface, and the co-ordinates in the model used in the analysis are shown in Fig. 1. Additionally, radiative heat transfer is neglected, because it is negligible comparing to the conductive heat transfer in temperature less to 1200 K. Furthermore it is supposed that the velocity field of the film is parallel to the axis "*x*".

For the momentum equation

$$\eta \frac{\partial^2 u}{\partial z^2} = \frac{\partial p}{\partial x} \tag{2}$$

where η is the dynamic viscosity, u the velocity and p the pressure. If the Eq. (2) is discomposed for the vapor and liquid, in this case we have

$$\begin{aligned} \eta_v \frac{\partial^2 u_v}{\partial z^2} &= \frac{\partial p_v}{\partial x} \\ \eta_l \frac{\partial^2 u_l}{\partial z^2} &= \frac{\partial p_l}{\partial x} \end{aligned} \tag{3}$$

with the boundary conditions (in the heater surface):

$$u_v = 0; \quad z = 0 \tag{4}$$

and in the same manner if our system is bounded by walls both at $z = 0$ and $z = H$, "*the rigid case*", we have

$$u_l = 0; \quad z = H \tag{5}$$

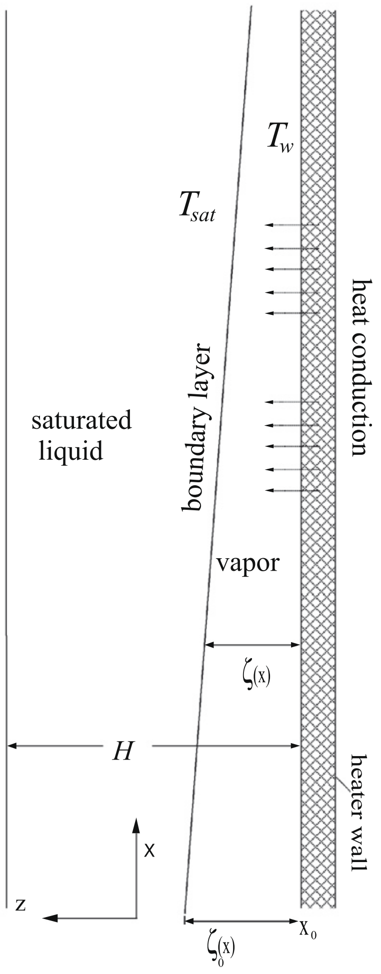

Fig. 1. Physical model and co-ordinates system.

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