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Calculation of Convective Boiling in a Vertical Tube at Sub-atmospheric Pressures

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Abstract: The convective heat transfer coefficient and the wall temperature in a uniform heated tube with upward flow regarding the pressure drop along the tube are calculated. Heat transfer capacity of the flow is implicit function of its velocity and its heat transfer coefficient; hence studying flow boiling heat transfer is prerequisite of calculating heat capacity of the flow. While studying two-phase flow in the vertical tube at sub-atmospheric pressures, we should note that the pressure drop effect on the flow and fluid properties cannot be neglected, because the pressure drop is comparable to operating pressures. Two-phase flow pressure drop and two-phase properties such as void fraction are also investigated in this study. In order to imply the pressure drop effect on the flow along the tube, its length is divided into small cells and the flow and fluid properties in each cell are assumed to be uniform. A model based on Kandlikar observation which is based on two main thermal regions and three sub regions is proposed to calculate the heat transfer coefficient. A computer program has been written to divide the tube length to small equal length cells and calculate pressure drop of each cell and update the flow properties in them. It also uses the appropriate heat transfer correlation to calculate heat transfer coefficient. The heat transfer coefficient is used to calculate the tube wall temperature distribution. The wall temperature distribution at 14.7 and 22.7 kPa, are numerically calculated and compared to experimental results.

Key word: water flow boiling, two-phase pressure drop, upward vertical tube, sub-atmospheric pressure

INTRODUCTION

The heat transfer phenomenon associated with liquid–vapor phase-change in a vertical heating tube is of great importance in evaporators throughout the process industry for mechanical, chemical, petrochemical and hydrometallurgical operation. The technology of industrial energy conservation includes, among other means, the use of heat pumps. Two common types of heat pumps are mechanical heat pumps and absorption heat cycles. The latter can be further divided into absorption heat pumps and absorption heat transformers (AHTs). An AHT may be constructed so as the necessary pressure differences of its components are maintained by hydrostatic legs. In such an AHT, it is possible to circulate the working medium pair without electrical pumps, and this process has been termed a Natural Circulation Loop (NCL) (Kockum, H., A. Jernqvist, 1997). The present paper is part of a project in which Two-Phase Natural Circulation Loop (TPNCL) using water as a working medium is studied (Ghobadi, M., 2009).

When the heat flux from the heating surface to the fluid, increases above a certain value, the convective heat transfer is not sufficient to prevent the wall temperature from rising above the saturation temperature of the flowing liquid. The elevated wall temperature superheats the liquid in contact with the wall and flow boiling is initiated in a certain length called onset of nucleate boiling (ONB).

During flow boiling heat is transferred from the heated surface to the liquid by different mechanisms (Tong, L.S., 1975):

Heat transport by the latent heat of evaporation. This mechanism is important at high pressures but negligible at normal pressure (Jamialahmadi, M., R. Blochl, 1991). Heat transport by continuous evaporation at the root and condensation at the top of the bubbles, while the bubbles are still attached to the heat transfer surface. This mechanism is also very important for fouling of heat transfer surface when the process fluid is contaminated (Jamialahmadi, M., R. Blochl, 1989). Unsteady state heat transfer is the result of bubble agitation

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of thermal boundary layer adjacent to the heat transfer surface. Heat transfer by single phase convection between the bubble growth zones. Over the last decades, extensive experimental and theoretical research efforts have been devoted to understanding the fundamental aspects of subcooled and saturated flow boiling. The result of these investigations have been documented and discussed in several review papers, e.g. McAdams *et al.* (1949), Kandlikar (1991), Steiner and Taborek (1992), Thome (1996), and Prodanovic *et al,* (2002). Steiner and Taborek (1992) summarized the various available correlations of saturated flow boiling. Thome (1996) reviewed the research on pool boiling and flow boiling and addressed several key points on flow boiling. Kandlikar (1991) performed a historical review on flow boiling heat transfer concepts. McAdams *et al.* (1949) and Prodanovic (2002) presented reliable formula for flow boiling regions.

The general purpose of the present study is to use the best set of correlations to calculate wall temperature along uniform heated vertical tube. Pressure drop effects have been taken into account to reach precise results. Since we encountered different thermal regions with different heat transfer mechanisms; different correlation has been applied for each region until the optimum results was reached. A new model based on Kandlikar flow boiling thermal regions is proposed to calculate the heat transfer in boiling vertical two-phase flow at sub-atmospheric pressures. The tube wall temperature distribution is calculated based on the heat transfer coefficient and has been validated by the experimental results (Saffari, H., 2003). The simulated model is shown in Fig. 1.

2. Flow Boiling Concepts:

As it can be observed in Fig. 1(a), three main regions are initiated inside an evaporation tube: single-phase liquid region (pure liquid), subcooled flow boiling region and saturated two-phase flow boiling region (Steiner, D., J. Taborek, 1992). The subcooled flow boiling, which is perhaps the most complex convective process encountered in the applications, occurs when the fluid changes from the single–phase flow to two-phase flow and nucleate boiling is initiated (ONB). Two circumstances must take place in the subcooled boiling: the temperature of the surface in contact with the heated liquid has to exceed the saturation temperature of the liquid at local pressure, and at the same location, the temperature of the bulk liquid remains below saturation (Teresa, J.L., Z. Jose, 2007). The subcooled flow boiling covers the region going from the ONB to the location where the thermodynamic quality is zero, corresponding to the saturated liquid state.

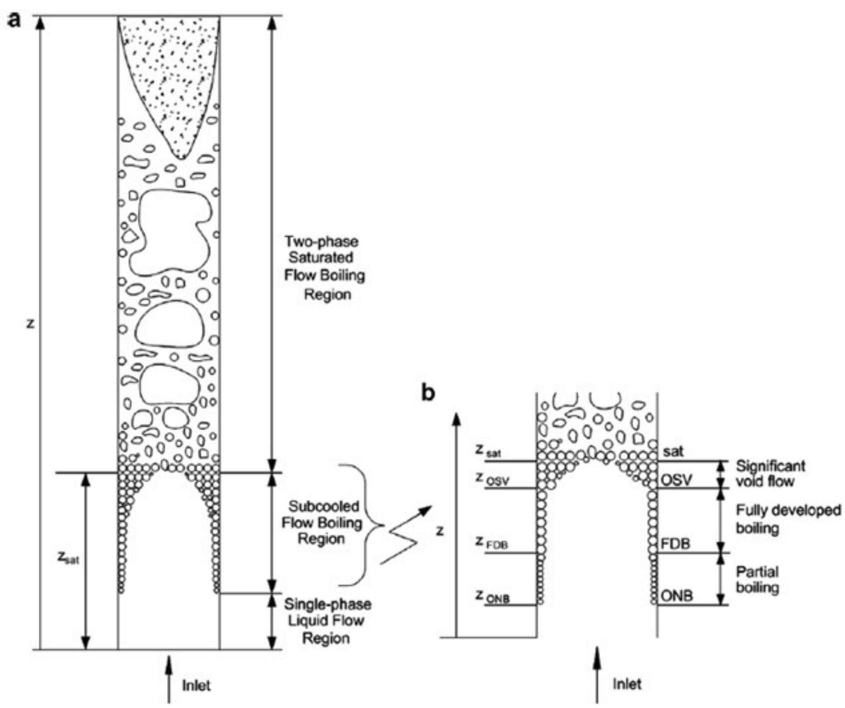

Fig. 1: (a) Upward flow boiling flow regimes. (b) Subcooled flow boiling in detail (Teresa, J.L., Z. Jose, 2007)

Three sub-regions, indicated in Fig. 1(b), depict the distinct regions and locations of subcooled flow boiling proposed by Kandlikar (Steiner, D., J. Taborek, 1992). Three sub-regions initiated in the subcooled region consist of partial boiling, fully developed boiling and significant void fraction. As noted before, the ONB is reached when the wall temperature (T_w) exceeds the saturation temperature (T_s) of the liquid in the local pressure, bringing about the nucleation process. In the region downstream, the partial boiling region, the convection and evaporation heat transfer mechanisms are both notable. This occurs until the convective contribution becomes insignificant. The region is called fully developed boiling (FDB), with nucleation as dominant mechanism. This region extends up to some points, designed as the onset of significant void flow (OSV) or net vapor generation (NGV), where the convective effects could become important again, due to the dramatic increase in the amount of vapor. The significant void flow continues to the onset of saturated point, where saturated flow boiling is initiated. As different heat transfer mechanisms are dominant in each region, a variety of heat transfer coefficients for the mentioned regions are expected to exist.

3. Governing Equations:

Calculation of pressure, temperature and vapor quality along the vertical tube is performed from the well-known equations of conservation of mass and energy and pressure drop written below:

$$G = \dot{m}/A = \rho.u = Const \tag{1}$$

$$\frac{W}{A}\frac{di}{ds} = \frac{4q}{D} \tag{2}$$

$$\frac{dp}{dz} = \frac{dp_a}{dz} + \frac{dp_f}{dz} + \frac{dp_g}{dz} \tag{3}$$

The length of the tube is accounted for by the coordinate z. As the mass flow rate $(\dot{m})$, and cross-sectional area of the tube (A) are fixed by design, the mass flux (G), Eq. (1), is constant. In the Energy Eq. (2) the heat flux is also constant and shows that a uniform heat source supply the energy. Three terms in Eq. (3) stand for acceleration, gravitational and frictional respectively.

3.1. the Liquid Single-phase Region:

As indicated, water enters the duct below its boiling point as compressed liquid. The pressure drop terms can be calculated with the following equations (Qi, S.L., P. Zhang, 2006). As the acceleration term is too small in contrast to the gravitational and frictional terms, it can be neglected in the liquid flow.

$$\frac{dp_g}{dz} = \rho_f g z \tag{4}$$

$$\frac{dp_f}{dz} = f\,\frac{L}{D}\frac{u^2}{2} \tag{5}$$

$$f = \frac{0.25}{\left\{\log\left[\left(\frac{e}{D}\right)+\left(\frac{5.74}{\mathrm{Re}^{0.9}}\right)\right]\right\}^2} \tag{6}$$

The heat transfer coefficient in laminar flow is calculated by Petokhov (2003) formula; hence the wall temperature can be calculated by using the following correlations:

$$Nu = 1.31\left[\frac{1}{\mathrm{Re}\,\mathrm{Pr}}\frac{x}{D}\right]^{-1/3}\left[1 + 2\,\frac{1}{\mathrm{Re}\,\mathrm{Pr}}\frac{x}{D}\right] \tag{7}$$

$$T_b = \frac{q\pi Dlz}{c_p} + T_i \tag{8}$$

$$T_w = \frac{q}{h_l} + T_b \tag{9}$$

The most important research content for the flow boiling in the tube is locating the onset of nucleate boiling (ONB), because the ONB marks the boundary between the single-phase and two-phase heat transfer region. Using Thome formula (Collier, J.G. and J.R. Thome, 1996), and with the help of Ishi correlation (Mishima, K., M. Ishii, 1984) to calculate wall temperature at
ONB, the location can be determined:

$$Z_{ONB} = \frac{Gc_{pf}D}{4}\left[\frac{\left(\Delta T_{SUB}\right)_i + \left(\Delta T_{SAT}\right)_{ONB}}{q} - \frac{1}{h_{fo}}\right] \tag{10}$$

$$\left(\Delta T_{SAT}\right)_{ONB} = \frac{4\sigma T_g h_{fo}}{\rho_{fg}.K_f.i_{fg}}\left[1 + \sqrt{1 + \frac{\left(\Delta T_{SAT}\right)_{ONB}.K_f.h_f.\rho_{fg}}{\sigma.T_s h_{fo}}}\right] \tag{11}$$

3.2. The Flow Boiling Region:

Flow boiling pressure drop calculation is done by calculating each term of pressure drop in Eq. (3). Void fraction determined by using Zivi (1964).

$$\frac{dp_a}{dz} = \frac{G}{A}\frac{w_o\rho_{fg}}{\rho_g A}\frac{dx}{dz} = \frac{w_o\rho_{fg}q}{\rho_g A.h_{fg}} \tag{11}$$

$$\frac{dp_g}{dz} = \overline{\rho}g \tag{12}$$

$$\overline{\rho} = \left(1-\alpha\right)\rho_f + \alpha.\rho_f \tag{13}$$

$$\alpha = \frac{1}{1 + \frac{1-x}{x}\left(\frac{\rho_g}{\rho_f}\right)^{2/3}} \tag{14}$$

And the frictional term is calculated by Friedel (1979) correlation.

$$\left(\frac{dp}{dz}\right)_{tp} = \left(\frac{dp}{dz}\right)_{fo} \phi_{lo}^{\ 2} \tag{15}$$

$$\phi_{lo}^{\ 2} = E + \frac{3.24FH}{\left(\frac{w_{go}^{\ 2}}{gD\rho^2}\right)^{0.045}\left(\frac{w_{go}^{\ 2}D}{\bar{\rho}\sigma}\right)^{0.035}} \tag{16}$$

$$E = (1-x)^2 + x^2\frac{\rho_f}{\rho_g}\frac{f_{go}}{f_{fo}} \tag{17}$$

$$F = x^{0.78}(1-x)^{0.24} \tag{18}$$

$$H = \left(\frac{\rho_f}{\rho_g}\right)^{0.91}\left(\frac{\mu_g}{\mu_f}\right)^{0.19}\left(1-\frac{\mu_g}{\mu_f}\right)^{0.7} \tag{19}$$

Attention is now focused on three regions shown in Fig. 1. (b): partial boiling from ONB to FDB, fully developed boiling from FDB to OSV and significant void flow or net vapor generation from OSV to saturation. To change from one correlation to another in the subcooled flow boiling region, determination of the ONB, FDB, OSV and saturated boiling is previously required. The length of the tube at which the water temperature reaches the ONB, FDB, OSV and saturation state will be designed by $z_{ONB,}$ z_{FDB}, z_{osv} and z_{sat}, respectively. ONB location, z_{ONB} was calculated by Thome correlation. FDB location, the location of transition from partial to fully developed boiling, is obtained by the method proposed by Shah (1977). Transition is assumed when $(T_{sat}-T_b)/(T_w-T_{sat})=2$, which is in excellent agreement with Prodanovic (2002) observations.

The point of OSV is predicted from a correlation by Saha and Zuber (1977) as made by Kandlikar (1991):

$$\Delta T_{sub,OSV} = \begin{cases} 0.0022(qD/k), for: \mathrm{Re.Pr} \le 7000 \\ 153.8(qG/c_p), for: \mathrm{Re.Pr} > 7000 \end{cases} \tag{20}$$

The saturation point is obtained when the thermodynamic steam quality is x=0. The proposed correlation for each region is shown in Table 1.

The Prodanovic correlation is based on modified Jakob number $Ja^* = c_p\Delta T_{sub}/i_{fg}$ and is obtained by substitution of $Ja^* = c_p\Delta T_{sub}/i_{fg}$ with ΔT_{sub} .

The nonequiulibrium quality x in the Kandlikar equation is defined as $x = (i - i_{f,sat})/i_{fg}$. The nonequilibrium quality at the location where $T_w = T_{sat}$, is denoted by $\dot{x}$, can be calculated by $\dot{x} = -qc_p/h_f i_{fg}$.

Table 1: Proposed correlation for each thermal region

Regions	Correlations	Equation	
Single-phase liquid	Petokhov (2003)	$Nu = 1.31\left[\frac{1}{\mathrm{Re}\,\mathrm{Pr}}\frac{x}{D}\right]^{-1/3}\left[1+2\frac{1}{\mathrm{Re}\,\mathrm{Pr}}\frac{x}{D}\right]$	(21)
Partial Flow Boiling	Prodanovic (2002)	$\frac{h_{tp}}{h_l} = \exp(14.542)\,Bo^{0.729}\,\dot{Ja}^{-.0354}\left(\frac{\rho_g}{\rho_f}\right)^{1.811}\mathrm{Pr}^{7.032}$	(22)
		$Ja^{*} = c_p\Delta T_{sub}/i_{fg}$	(23)
Fully Developed Boiling	Kandlikar (1991)	$\frac{h_{tp}}{h_l} = \left[\frac{1}{f_1(Bo)} + \frac{x}{\dot{x}}\right]^{-1}$	(24)
		$f_1(Bo) = \begin{cases} 230Bo^{0.5}, for: Bo > 3\times10^{-5} \\ 1+46Bo^{0.5}, for: Bo \leq 3\times10^{-5}\end{cases}$	(25)
Net Vapor Generation		$x = \left(i - i_{f,sat}\right)/i_{fg}$	(26)
		$\dot{x} = -qc_p/h_f i_{fg}$	(27)
Saturate Flow Boiling	McAdams (1949)	$q = 4.77\Delta T_{sat}^{3.86}$	(28)

4. Method:

As mentioned before, the integral procedure has been used to study the flow in the tube length. The initiated program divides the tube length to small equal cells where the flow properties' except temperature can be assumed to be constant in each cell. Starting from the first cell of the tube at the beginning of the flow tube, flow properties are updated due to the cell's pressure and pressure drop is calculated in each of them; hence by calculating the pressure drop of each cell, the pressure of the next cell is obtained. This procedure is repeated until the end of the tube. The algorithm used for the procedure is shown in Fig. 2.

RESULTS AND DISCUSSIONS

The heat Transfer and the wall temperature of a uniform heated tube at two different sub-atmospheric initial conditions have been numerically analyzed. The wall temperature distribution results have been verified by experimental observations (Saffari, H., 2003). Table 2 shows conditions used in numerical analyses according to (Saffari, H., 2003). The tube diameter is 5.4 millimeter and its height is 1.3 meter.

Table 2: Properties values used in the numerical analysis

Conditions	P(kPa)	G $\left(kg/m^2s\right)$	T_i (C)	q $\left(Kw/m^2\right)$
Condition I	22.7	71.7	49	21.7
Condition II	14.7	100.9	31	8.4

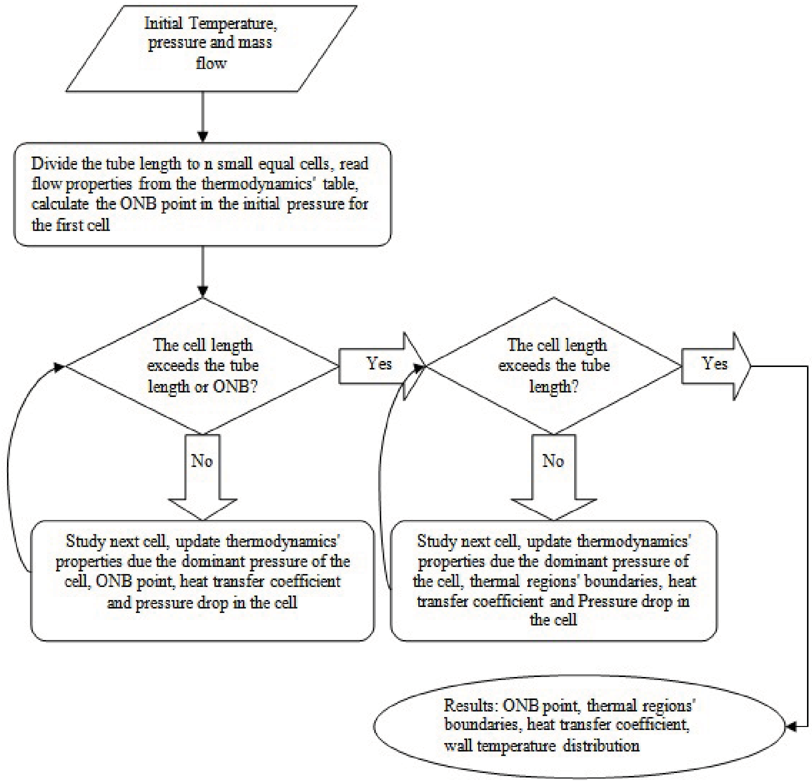

Fig. 2: Program algorithm

As shown in Fig. 3, the pressure drop along the tube is fairly comparable to the initial pressure and it even exceeds the pressure at the end of the tube in condition II. The main fraction of pressure drop at the ONB point is the gravitational term; hence as the void fraction increases dramatically at this point, the total pressure drop and the pressure slope decrease apparently. Pressure drop rises along the tube, as we go through it. Passing through the flow boiling the frictional fraction become dominant and the total pressure drop increases through the tube. This increase happens because initiation of bubbles leads to increase of the flow roughness.

Fig. 4, Shows the experimental validation of numerical analysis for wall temperature distribution. As we can see in Fig. 4(a) the deviation of numerical results from experimental results is less than 5% and the rise of two diagrams is well coped with each others. This error rises up to 10% in the second condition at the pressure near the vacuum condition (see Fig. 4(b)). Rising through the tube, the temperature of the liquid increases as it is heated, at the same time that the saturation point is falling. The saturation point of the liquid falls because of the decreasing hydrostatic pressure while going up in the tube. As we expected, the wall temperature increases whereas the difference T_{wall}-T_{water} shortens until saturation is reached. At this point the wall temperature decreases abruptly due to the discontinuity in the increased value of the heat transfer coefficient. Downstream shows a small peak at the location where the transition for the Kandlikar correlation from nucleates to the convective region takes place.

Fig. 5 depicts the heat transfer coefficient in two conditions. As it is obvious, heat transfer coefficient jumps at ONB point. At this point nucleation heat transfer mechanism becomes active. After the jump the heat transfer coefficient rises with a sharp slope. The heat transfer increases due to the nucleation process that is enhanced until reaching OSV. From this point, convectional and nucleation mechanisms both become important and the heat transfer increases with a constant slope approximately.

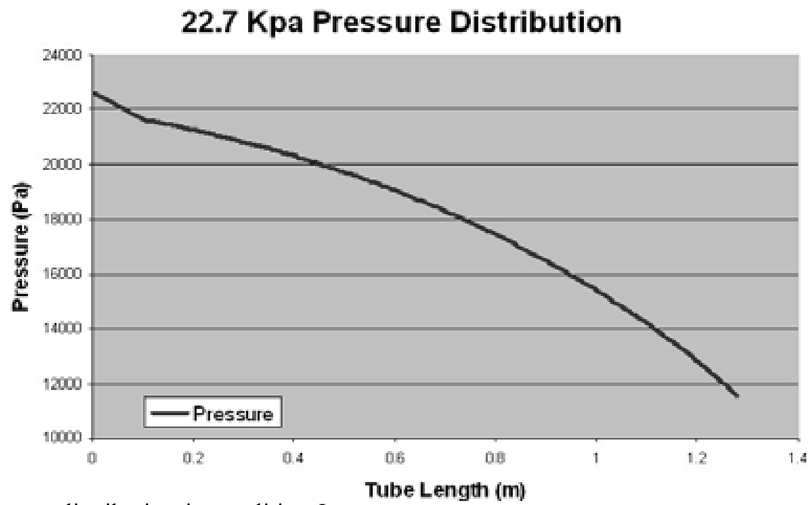

Fig. 3: (a) Pressure distribution in condition I.

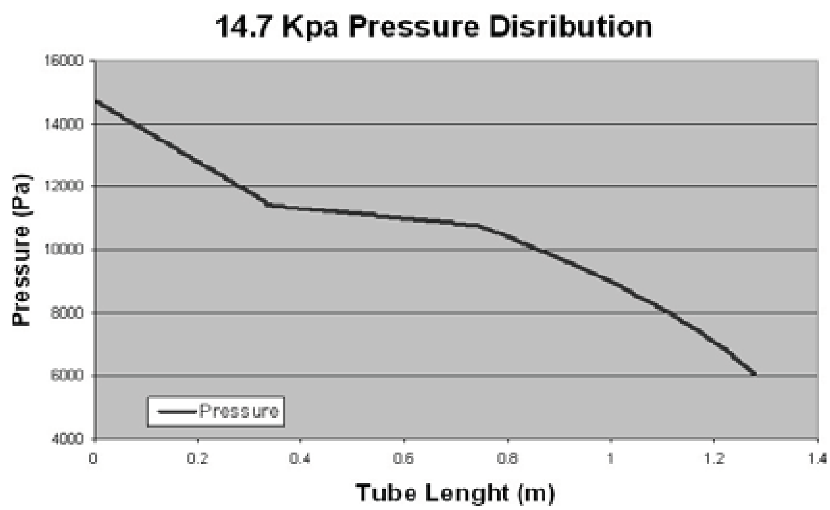

Fig. 3: (b) Pressure distribution in condition II.

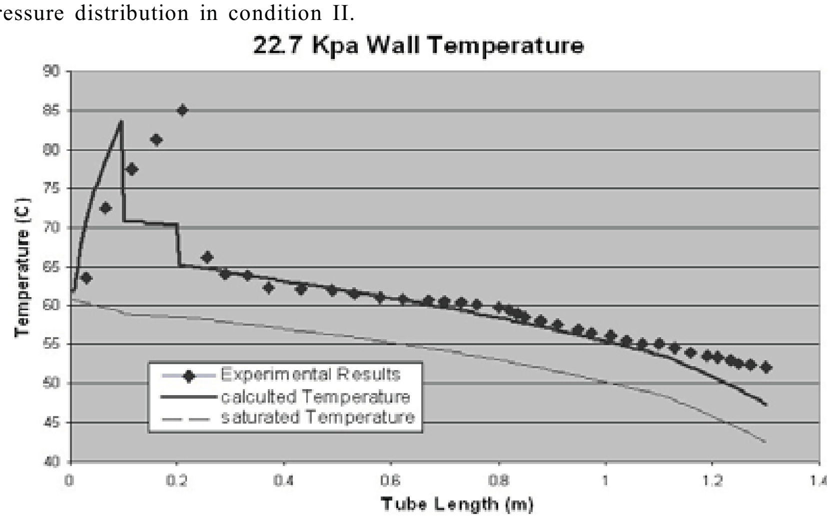

Fig. 4: (a) Wall temperature in condition I.

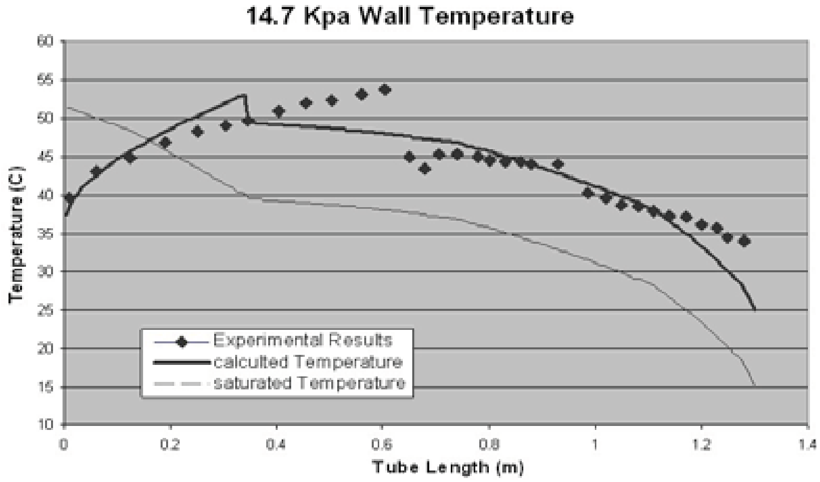

Fig. 4: (b) Wall temperature in condition II.

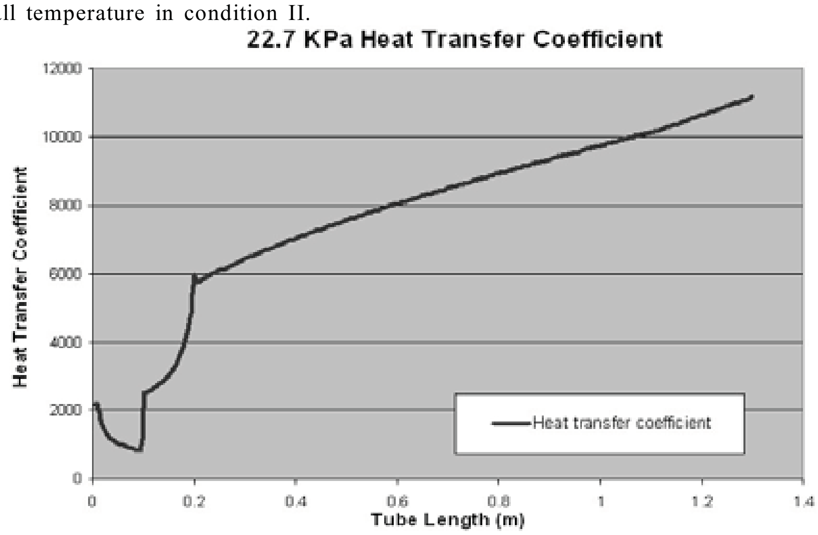

Fig. 5: (a) Heat transfer coefficient in condition I.

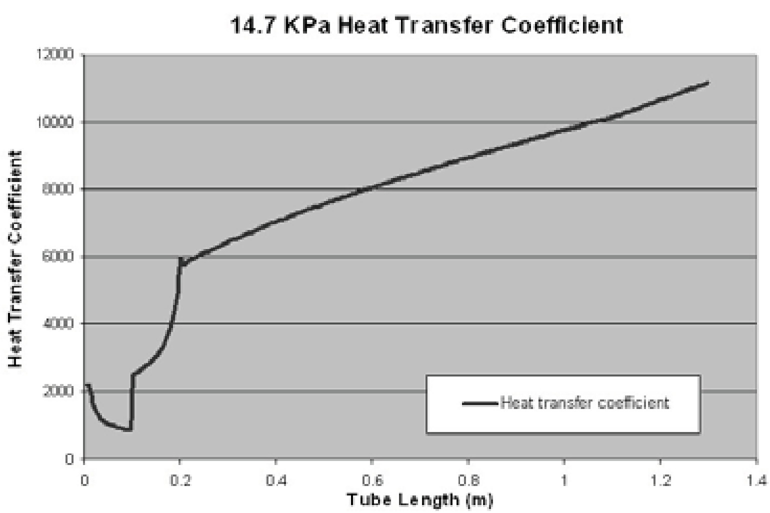

Fig. 5: (b) Heat transfer coefficient in condition II.

Nomenclature:

English symbols

A tube area (m^2)

C_p heat capcity *(T)*

D tube diametere *(m)*

e Roughness *(m)*

G mass velocity $\left(kg/m^2s\right)$

h heat transfer coefficient $\left(W/m^oC\right)$

i enthalpy (J/Kg)

j volumetric flux (superficial velocity) $\left(m/S\right)$

L tube length *(m)*

q heat absorbed from uniform heat source $\left(kW/m^2\right)$

T temperature $\left({}^oC\right)$

u velocity $\left(m/s\right)$

W mass rate of flow $\left(kg/s\right)$

x mass vapor quality

Z height *(m)*

Greek Symbols

α void fraction

β volumetric quality

μ viscosity $\left(Ns/m^2\right)$

σ surface tension $\left(N/m\right)$

Gradient and differences

ΔT_{SAT} wall superheat

ΔT_{SUB} liquid subcooling

Dimensionless number

Bo boiling umber

Nu Nusselt number $\left(hD/k\right)$

Pr Prandtl number $\left(c\mu/k\right)$

Re Reynolds number $\left(GD/\mu\right)$

Subscripts

f fluid

fo fluid only

g vapor (gas)

go vapor only (gas only)

FDB fully Developed Boiling

ONB onset of Nucleate Boiling

OSV onset of Significant Void fraction

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