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Effect of submergence on boiling incipience in a vertical thermosiphon reboiler

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Abstract

An analysis of the incipience of nucleate boiling has been developed as a modification and extension of the previous analyses for the wall superheat including the effect of submergence. The minimum degree of wall superheat required for the onset of fully developed boiling of liquid was related to the thermophysical properties of test liquids. The model prediction was made with the experimental data available in literature for nine organic liquids including water. The results predicted from the theoretical model were found to be consistent with the data. An effort was also made to obtain a unified correlation for nine different fluids, together covering wide ranges of heat flux, submergence and inlet liquid subcooling. It was observed that the majority of the data points lie within a maximum error of ±22% and mean absolute deviation of 16 percent.
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Keywords: Incipient boiling; Submergence; Wall superheat; Thermosiphon reboiler; Heat flux

1. Introduction

The vertical tube closed loop thermosiphon reboiler is a simple but effective heat transfer device. It is a one-pass heat exchanger in which boiling occurs inside vertical tubes. When vapor forms in the tubes of such an exchanger, the specific volume is increased and the weight of liquid in the return line causes circulation of liquid through the reboiler. The subcooled liquid entering the tube gets heated by single-phase convection and moves upwards. Depending upon wall temperature condition, subcooled boiling may set-in at the surface. When the liquid temperature attains saturation value; saturated boiling begins with the existence of net vapor, which increases resulting in bubbly to mist flow. Thus the heat transfer to liquid in the reboiler tube generates a changing two-phase flow with various flow regimes spread along the test section. The point at which the two-phase begins is known as incipient point of boiling (IPB), which corresponds to the conditions of minimum degree of wall superheat or heat flux required for the formation and detachment of first vapor bubble from the heated surface. Therefore, information on the conditions required for the onset of nucleate boiling are of paramount importance in the design of two phase flow heat transfer equipments.

The point of onset of nucleation along the tube length and its required wall superheat depend upon a number of operating parameters. Numerous studies on predicting wall superheat through semi-empirical approach for forced convection boiling systems have been studied extensively during the last few years. These studies include the effects of various physical parameters, the size and geometry of nucleation sites, surface roughness, role of surface tension and wettability [1–5]. The widely accepted approach for the prediction of incipient boiling is based on the Gibb's equilibrium theory of bubble in the uniformly superheated liquid and the one-dimensional steady or transient heat conduction equation. It was postulated that in the liquid film adjacent to the heating surface the superheated layer δ^*, must attain a threshold value so that the critical bubble nuclei with radius r_c can further grow to the point of detachment. Zuber [6] was probably the first to analyze the inter-relationship between the local heat flux, the superheated layer and the diameter of the surface cavity. He employed

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Nomenclature

a, b, C_4	constants	
C	heat capacity	$\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$
C_1	constant, $(1+\cos\theta)$	
h_{fg}	latent heat of vaporization	$\text{J}\cdot\text{kg}^{-1}$
k	thermal conductivity	$\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$
K_{sub}	subcooling number $(1+\frac{\rho_L \Delta T_{\text{sub}}}{\rho_v T_s})$	
L	total heated length	m
P	pressure, bar	$\text{N}\cdot\text{m}^{-2}$
Pe_B	Peclet number for boiling $\left(\frac{q\rho_L C_L}{\rho_v h_{\text{fg}} k_L}\sqrt{\frac{\sigma}{g(\rho_L-\rho_v)}}\right)$	
Pr	Prandtl number	
q	heat flux	$\text{W}\cdot\text{m}^{-2}$
r	radius	m
R	universal gas constant	$\text{kJ}\cdot\text{K}^{-1}\cdot\text{k}^{-1}\cdot\text{mol}^{-1}$
Re	Reynolds number	
$r_{\max}$	maximum cavity radius	m
$r_{\tan}$	cavity radius based on the tangency criterion for incipience	m
S	submergence	%
T	temperature	°C (K)
ΔT_s	degree of superheat $(T_w - T_s)$	°C (K)
ΔT_{sub}	degree of subcooling $(T_s - T_L)$	°C (K)
y	distance perpendicular to the heated wall	m
y_{lam}	thickness of the laminar sublayer	m
Z	distance along the test section	m

Greek symbols

δ^*	superheated layer thickness	m
ρ	density	$\text{kg}\cdot\text{m}^{-3}$
θ	contact angle	
μ	dynamic viscosity	$\text{Ns}\cdot\text{m}^{-2}$
ν	kinematic viscosity	$\text{m}^2\cdot\text{s}^{-1}$
σ	surface tension	$\text{N}\cdot\text{m}^{-1}$

Subscripts

pred	predicted
v	vapor
i	incipience
s	saturation
w	wall
exp	experimental
B	boiling
L	liquid
b	bubble
c	cavity, critical condition
Sub	subcooling
NB	non-boiling
OB	onset of boiling
avg	average
MAD	mean absolute deviation

Superscript

n, n_4, n_5, n_6	exponent

the data of the Clark et al. [7] and Griffith and Wallis [8] and carried out a series of calculations and found that the superheated layer thickness is approximately equal to the diameter of the active cavity, i.e., $\delta^*/r_c = 2$. Hsu [9] developed an analytical expression for the size range of active nucleation sites for constant heat flux at the wall.

Kenning [12] developed an empirical correlation for the wall superheat required to initiate nucleate boiling, incorporating the Bergles and Rohsenow [11] assumption in his study. His analysis predicts wall superheat that is considerably lower than those predicted by the former analysis. Unal [16,17] considered the effect of pressure on the boiling incipience under subcooled flow boiling of water in a vertical tube. Further Unal [17] determined the incipient point of boiling for subcooled nucleate flow boiling of water with high-speed photography. Yin and Abdelmessih [18] investigated the phenomenon of liquid superheat during incipient boiling in a uniformly heated forced convection channel. Butterworth and Shock [20] found that the Davis and Anderson [14] equation did not predict the subcooled boiling incipience satisfactorily for higher value of r_c for some fluids and conditions. Wu et al. [23] studied the surfactant effects on boiling incipience. Agarwal [24], Ali and Alam [25], Kamil [26], and Kamil et al. [27,28], experimentally obtained the boiling and non-boiling zones (Z_{OB}/L) for heating surface and wall superheat for incipient boiling for different liquids in a vertical tube thermosiphon reboiler with wide range of submergence. A dimensionless correlation relating the values of heat flux, inlet liquid subcooling and submergence was proposed for Z_{OB}/L and for wall superheat relating the heat flux with thermophysical properties of test liquids. Shamsuzzoha et al. [29] have developed a correlation for prediction of wall superheat including the effect of submergence for the thermosiphon reboiler.

There are two different criteria for the incipient point of boiling, one is $r_{\tan}$, and other is $r_{\max}$. The incipience in forced convection systems was studied extensively by a number of workers [3,11,13,14,22] and they predicted incipience based on the point of tangency $(r_{\tan})$.

The validity of this criterion was proven in many practical applications. Incipience based on $r_{\tan}$ criteria for natural convection system was studied by Agarwal [24], Ali and Alam [25], Kamil [26] and Kamil et al. [27–29]. Hino and Ueda [21] and Sudo et al. [19] used a different hypothesis in forced convection study. They predicted incipience based on the maximum cavity radius $(r_{\max})$ available for nucleation on the heated surface. A summary of previous incipience investigations is given in Table 1.

Table 1
Summary of boiling incipience investigations

Authors [reference] (year)	Flow geometry	Heater material	Fluid	Mean velocity [m·s^{-1}]	Pressure [bar (psia)]	Subcooling [°C]	Incipience formula
Sato and Matsumura [13] (1963)	Vertical channel	Stainless steel	Water	0.6–4.1	1.0 (14.7)	3–70	$q=\frac{k_L\rho_v h_{\mathrm{fg}}}{8\sigma T_s}(T_w-T_s)^2$ • Tangency criterion, $r=k_L(T_w-T_s)/2q$
Bergles and Rohsenow [11] (1964)	Horizontal annulus	Stainless steel	Water	3.3–17.4	Up to 2.6 (38.0)	32–90	$q=15.60P^{1.156}(T_w-T_s)^{2.30/P^{0.023}}$ (P in psia) • Graphical solution for water over a pressure range of 15–2000 psia based on tangency criterion
Han and Griffith [10] (1965)	Pool boiling on a horizontal surface	Gold finished with 600 grit emery paper	Water	– –	1.0 (14.7)	7	$q=\frac{k_L\rho_v h_{\mathrm{fg}}}{12\sigma T_s}(T_w-T_s)^2$ • Tangency criterion, $y=1.5r$ at the point of tangency
Davis and Anderson [14] (1966)	Authors performed analysis using experimental data from prior studies						$q=\frac{k_L\rho_v h_{\mathrm{fg}}}{8C1\sigma T_s}(T_w-T_s)^2$ • Tangency criterion, $y=r$ at the point of tangency • $C_1=1$ for hemispherical bubble nucleus
Frost and Dzakowic [15] (1967)	Authors performed analysis using experimental data from prior studies						$q=\frac{k_L\rho_v h_{\mathrm{fg}}}{8\sigma T_s}(T_w-T_s)^2\frac{1}{Pr_L^2}$ • Tangency criterion, $y=Pr_L^2 r$ at the point of tangency.
Yin and Abdelmessih [3] (1977)	Vertical tube	Stainless steel	Freon-11	0.08–0.4	Up to 2.0 (30.0)	1.20	For increasing heat flux: $q=\frac{1}{[7-q/6500]}2\frac{k_L\rho_v h_{\mathrm{fg}}}{2\sigma T_s}(T_w-T_s)^2$ • For decreasing heat flux: $q=\frac{k_L\rho_v h_{\mathrm{fg}}}{5\sigma T_s}(T_w-T_s)^2$ • Tangency criterion, y/r at the point of tangency correlated empirically.
Agrawal [24] (1980)	Vertical tube	Stainless Steel tube	Water, Acetone, Ethyl acetate, Propanol, Toluene	–	Atm. Pr.	0.9–73	$(T_w-T_s)^2=(a-bq)^2\frac{2\sigma T_s}{k_L h_{\mathrm{fg}}\rho_v}q$ $\left(\frac{Z_{\mathrm{OB}}}{L}\times 100\right)=C_4(Pe_B)^{n4}(K_{\mathrm{sub}})^{n5}(S)^{n6}$
Hino and Ueda [21] (1985)	Vertical annulus	Stainless steel finished with 4/0 emery cloth	Freon-113	0.1–1.0	1.47 (22.0)	10–30	• For $r_{\tan}<r_{\max}$: $q=\frac{k_L\rho_v h_{\mathrm{fg}}}{8\sigma T_s}(T_w-T_s)^2$ • For $r_{\tan}>r_{\max}$: $q=\frac{k_L}{r_{\max}}(T_w-T_s)-\frac{2\sigma T_s k_L}{\rho_v h_{\mathrm{fg}} r_{\max}^2}$
Cerza and Sernas [32] (1985)	Falling film over vertical cylinder	Brass	Water	0.32–1.03	1.0 (14.7)	0	• Numerical solution for boiling incipience in laminar developing films based on the tangency criterion.

(*continued on next page*)

Table 1 (*Continued from Table 1*)

Authors [reference] (year)	Flow geometry	Heater material	Fluid	Mean velocity [m·s^{-1}]	Pressure [bar (psia)]	Subcooling [°C]	Incipience formula
Sudo et al. [19] (1986)	Vertical channel	Inconel 600	Water	0.7–1.5	1.2 (17.0)	28–35	• For $r_{\tan} < r_{\max}$: $q = \frac{k_L \rho_v h_{fg}}{8\sigma T_s}(T_w - T_s)^2$ • For $r_{\tan} > r_{\max}$: $q = \frac{k_L}{r_{\max}}(T_w - T_s) - \frac{2\sigma T_s k_L}{\rho_v h_{fg} r_{\max}^2}$
Marsh and Mudawar [22] (1989)	Falling film over vertical cylinder	Stainless steel finished with 600 grit paper	Water, FC-72	1.29–1.79 1.03–1.65	0.7 (10.2) 1.05 (15)	0 9–21	$q = \frac{1}{3.5}\frac{k_L \rho_v h_{fg}}{8\sigma T_s}(T_w - T_s)^2$ • Modified tangency criterion which accounts for turbulence and waviness in water films • The correlation does not apply for highly wetting fluids.
Ali and Alam [25] (1991)	Vertical tube	Stainless steel tube	Water, Acetone, Ethanol, Ethylene-glycol	– – –	Atm. Pr.	0.2–45.5	$(T_w - T_s)^2 = (a - bq)^2 \frac{2\sigma T_s}{k_L h_{fg} \rho_v} q$ $\left(\frac{Z_{OB}}{L} \times 100\right) = 0.0405(Pe_B)^{0.308}$ $\times K_{sub}^{0.356} S^{0.692} \left(\frac{\nu_L}{\nu_v}\right)^{0.216}$
Kamil et al. [27] (1993)	Vertical tube	Stainless steel	Water, Methanol, Benzene, Toluene, Ethylene-glycol	– –	Atm. Pr.	0.5–11.6	$(T_w - T_s)^2 = (a - bq)^2 \frac{2\sigma T_s}{k_L h_{fg} \rho_v} q$ $\left(\frac{Z_{OB}}{L} \times 100\right) = 4.003(Pe_B)^{0.2739}(K_{sub})^{0.39}$ $\times (S)^{0.4522} \left(\frac{\nu_L}{\nu_V}\right)^{0.8656}$
Kureta et al. [30] (1998)	Small diameter vertical tube	Nickel or Stainless steel	Water	0.1–1.0	Atm. Pr.	70–90	$q = 3.0 \times 10^{-4}(\Delta T_S)^{2.6}$
Shamsuzzoha, Kamil and Alam (Present study)	Authors performed analysis using experimental data from prior studies.						• For $r_{\tan} < r_{\max}$: $(T_w - T_s)$ $= \left[\frac{8\sigma T_s q}{k_L \rho_v h_{fg}(1+\frac{2\sigma}{r_c P_s})[1-\frac{RT_s}{h_{fg}}\ln(1+\frac{2\sigma}{r_c P_s})]^2}\right]^{1/2}$ $\times (S)^{0.67079}$

Thus from the literature it is evident that the effect of submergence was not considered in the prediction of degree of wall superheat for the case of vertical thermosiphon reboiler.

Therefore, in the present study, a theoretical analysis was carried out to develop an analytical equation for the incipient point of boiling including the effect of submergence. The model prediction was made with the experimental data for different fluids available in literature and it was also compared with existing incipience models.

2. Experimentation

The experimental reboiler was made of two vertical tubes in a U shaped circulation loop, made of two long vertical and one short horizontal Stainless Steel tubes as shown in Fig. 1. One of the tubes was electrically heated and served as a test section, the upper end of which was connected to a glass tube of matching ID and then to a vapor–liquid separator. The test section was a Stainless Steel tube of 25.50 mm ID, 28.85 mm OD and 1900 mm long. In order to maintain the heat transfer surface temperatures along the tube length, 21 Copper-constantan thermocouples were spot welded on the outer surface of the tube at intervals of 50 mm up to a length of 200 mm from the bottom end and 100 mm over the remaining length, inlet liquid and exit temperature were measured by separate thermocouples. The entire setup was thoroughly lagged with asbestos rope and glass wool and finally covered by a thin aluminum sheet to reduce the heat losses, which were less than ±2.5 percent.

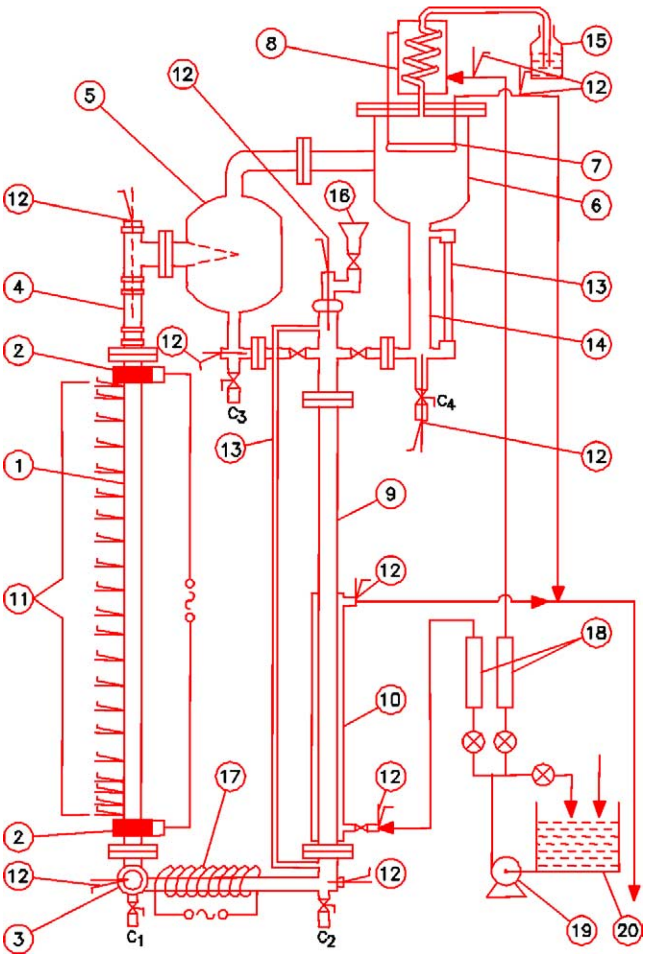

Fig. 1. Schematic diagram of the experimental set-up. 1. Test section. 2. Copper clamps. 3. View port for intel liquid. 4. Glass tube section. 5. Vapor–liquid separator. 6. Primary condenser. 7. Spiral coil. 8. Secondary condenser. 9. Liquid down-flow pipe. 10. Cooling jacket. 11. Wall thermocouple. 12. Liquid thermocouple probes. 13. Liquid level indicator. 14. Condenser downflow pipe. 15. Bubbler. 16. Feeding funnel. 17. Auxiliary heater. 18. Rotameters. 19. Centrifugal pump. 20. Cold water tank.

The maximum liquid head used in the present study corresponded to the liquid level equal to the top end of the reboiler tube. This condition has been termed as 100% submergence. The experimental data were generated for four different levels of submergence and various heat flux. Other details of reboiler and cooling system along with its operating procedure are described in detail elsewhere in literature [26–28].

3. Theoretical analysis

The basic assumptions made in the present theoretical analysis are:

(i) The potentially active cavities are of conical shape and the bubble nucleus which forms at such surface cavities has the shape of a truncated sphere.

(ii) Gibb's equation of bubble coexisting with a surrounding uniformly superheated liquid can be used as a reasonable first approximation. That is

$$(T_b - T_s) = \frac{R T_b T_s}{h_{fg}} \ln\left(1 + \frac{2\sigma}{r_b P_s}\right) \tag{1}$$

Here T_b is not constant over the bubble surface because temperature gradients are present in the wall region.

(iii) The liquid temperature within the thermal layer has a linear profile, which is not significantly altered by the presence of a neighboring bubble.

(iv) A bubble nucleus will grow if the temperature of the fluid at a distance from the wall equal to the bubble height is greater than the wall superheat required for bubble equilibrium.

Fig. 2 shows three possible bubble nucleus shapes, which can exist at the mouth of the cavity. The nucleus was assumed to be formed by the residual vapor from the preceding bubble, which was trapped in the cavity as suggested by Bankhoff [33]. Bergles and Rohsenow [11] considered the case II giving an argument that once a bubble passed this condition of minimum radius it would continue

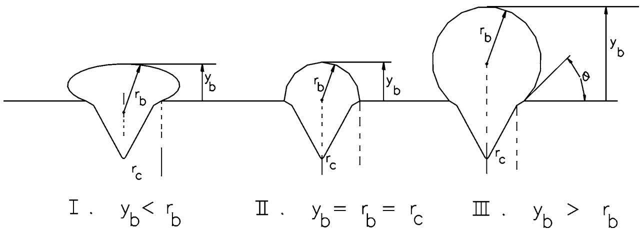

Fig. 2. Possible bubble models.

to grow. In a non-uniform temperature field it is possible for the bubble nucleus to assume some other equilibrium state, say case I or case III in Fig. 2. Whether, a variation of surface tension with temperature results in a stable non-spherical bubble, case I, or whether equilibrium is actually a steady state condition with heat and mass transfer into the base of the bubble offset by loss at the top of it, is open to speculation. Our discussion is concerned with case III, which is truncated spherical bubble. It reduces to the hemispherical bubble, i.e., case II when the bubble contact angle is 90 degrees. It is possible that the nucleus will not grow much beyond the hemispherical shape even if the wall superheat is sufficient, for shear forces acting on the bubble can sweep it from the wall. The hemispherical bubble will have greater stability.

It follows from assumption (iii) that cavities of only a narrow size range are involved at the onset of nucleate boiling. It is expected that the population density of bubbles on the surface just prior to the onset of nucleate boiling will be so low that they will not greatly affect the temperature profile in the fluid. Because the bubble nuclei are very small, they are within the laminar sub layer and that the thermal conductivity of liquid is also constant. The heat transport occurs by conduction through the liquid.

$$T_{L(y)} = T_w - \frac{qy}{k_L} \tag{2}$$

If δ^* represents the superheated layer thickness.

$$T_{L(\delta^*)} = T_s = T_w - \frac{q\delta^*}{k_L} \tag{3}$$

Sato and Matsumura [13] and Kenning [12] have suggested alternatives to assumption (iv) which causes the prediction of lower values of wall superheat required to initiate nucleate boiling, arguing that a more suitable isothermal streamline is one nearer the wall since the bubble disturbs the flow. One useful criterion for the initiation of nucleate boiling was suggested by Bergles and Rohsenow [11]. These authors suggested that at a distance $y = r_b$, $T_L = T_b$ (where $T_b > T_s$), as the condition for the bubble to grow. It was then postulated that the minimum wall superheat $(T_w - T_s)$ required to initiate boiling is determined by the point of tangency of Eq. (1) with Eq. (2), i.e.,

$$\frac{\mathrm{d}T_b}{\mathrm{d}r_b} = \frac{\mathrm{d}T_L}{\mathrm{d}y} \tag{4}$$

The relation between the bubble nucleus height y_b, the bubble nucleus radius r_b and the cavity radius r_c can be obtained by considering the case of a truncated sphere with the angle towards the truncated section being θ, Fig. 2 (case III).

$$y_b = r_b(1 + \cos\theta) = C_1 r_b \tag{5}$$

$$r_c = r_b \sin\theta \tag{6}$$

Griffith and Wallis [8] measured bubble contact angles for water on clean surfaces and they reported typical contact angles for contaminated metal surfaces, which ranged from about 30–90 degrees. These results indicate that engineering surfaces are usually partially wetted by water and organic liquids.

It is convenient to express r_b in wall superheat equation (1) in terms of bubble height or distance from the wall to the top of the bubble, y_b, as defined in Eq. (5).

It then follows that:

$$(T_b - T_s) = \frac{RT_bT_s}{h_{\mathrm{fg}}} \ln\left(1 + \frac{2\sigma C_1}{y_b P_s}\right) \tag{7}$$

First derivative of above equation is

$$\frac{\mathrm{d}T_b}{\mathrm{d}y_b} = -\frac{2\sigma C_1 RT_s^2\left[1 - \frac{RT_s}{h_{\mathrm{fg}}}\ln\left(1 + \frac{2\sigma C_1}{y_b P_s}\right)\right]^{-2}}{y_b^2 P_s h_{\mathrm{fg}}\left(1 + \frac{2\sigma C_1}{y_b P_s}\right)} \tag{8}$$

and from Eq. (2)

$$\frac{\mathrm{d}T_L}{\mathrm{d}y} = -\frac{q}{k_L} \tag{9}$$

Substituting Eqs. (8) and (9) in to Eq. (4) gives

$$\frac{\mathrm{q}}{k_L} = \frac{2\sigma C_1 RT_s^2\left[1 - \frac{RT_s}{h_{\mathrm{fg}}}\ln\left(1 + \frac{2\sigma C_1}{y_b P_s}\right)\right]^{-2}}{y_b^2 P_s h_{\mathrm{fg}}\left(1 + \frac{2\sigma C_1}{y_b P_s}\right)} \tag{10}$$

Rather than imposing specific relations between y_b and δ^* as in previous studies, combining Eqs. (3) and (10) and solving for wall superheat, we get

$$(T_w - T_s)^2 = \frac{2\sigma T_s q\left(\frac{\delta^*}{r_c}\right)^2}{k_L \rho_v h_{\mathrm{fg}}\left(1 + \frac{2\sigma}{y_b P_s}\right)\left[1 - \frac{RT_s}{h_{\mathrm{fg}}}\ln\left(1 + \frac{2\sigma}{y_b P_s}\right)\right]^2} \tag{11}$$

Zuber's [6] appears to have first analyzed the inter-relations between the local heat flux, the superheat layer and the diameter of the surface cavity. He found that the super heated layer thickness (δ^*) is approximately equal to the diameter of the active cavity. This relation has been subsequently used by several other investigators [13,15,19,21] in incipient boiling studies. Eq. (11) can be further simplified without sacrifice of accuracy by assuming that $\theta = 90°$, which causes $C_1 = 1$, $y_b = r_b = r_c$ from Eqs. (5) and (6), and corresponds to a hemispherical bubble, at the onset of post-critical growth. A number of workers [24–29,31] have investigated the effect of inlet liquid subcooling and submergence on heat transfer, circulation rate and boiling incipience in a natural circulation vertical thermosiphon reboiler. In case of a natural circulation reboiler, the induced flow rate is established due to the differential head existing between the cold and hot legs. The hydrostatic head in the cold leg (down flow pipe) of a thermosiphon reboiler depends upon the liquid submergence, the maximum value of which equals to the liquid level corresponding to the top end of the test section, termed as 100 percent submergence ($S = 100\%$). The rate of circulation, therefore, depends upon liquid submergence, heat flux, inlet liquid subcooling, vapor fraction and frictional resistance. At a given submergence, the liquid head in the cold leg remains unchanged while increase in heat flux shift the points of boiling incipience towards tube inlet. As the submergence is lowered, the liquid head gets decreased while the vapor fraction increases due to the enhanced effect of saturated boiling in the tube. However, the differential head causing circulation becomes smaller then that at higher value of submergence. The detailed description of the effect of submergence on induced flow has been discussed earlier by Kamil et al. [34]. Thus from the above, it is clear that submergence has an important effect on boiling incipience in case of natural circulation reboiler. Yin and Abdelmessih [3] has also investigated the effect of velocity on δ^*/r_c. Therefore, it is important to include the effect of submergence in the prediction of degree of wall superheat. Thus after incorporating the effect of submergence, the above equation was modified as:

$$(T_w - T_s) = \left[\frac{8\sigma T_s q}{k_L \rho_v h_{fg}\left(1 + \frac{2\sigma}{r_c P_s}\right)\left[1 - \frac{RT_s}{h_{fg}} \ln\left(1 + \frac{2\sigma}{r_c P_s}\right)\right]^2}\right]^{1/2} (S)^n \tag{12}$$

This equation is the general expression of incipient boiling and several interesting observations can be made from it, which involves only the wall superheat that is easy to measure directly and right-hand side as a whole can be evaluated with reasonable accuracy from the measurable quantities. Some justified simplifications as used by other investigators for systems of low surface tension or high pressure, it is possible to write that

$$1 \gg \frac{2\sigma C_1}{y_b P_s}$$

This simplification reduces the Eq. (11) into different investigator's results as:

(a) $\theta = 90°$ (consequently $y_b = r_b = r_c$) and $\delta^* = 2r_c$ at the onset of nucleate boiling, which reduces to Sato and Matsumura [13] and Davis and Anderson [14] correlation, respectively. (b) If $\theta = 90°$ and $\delta^* = \sqrt{6} r_c$, reduces to an equation which was derived by Han and Griffith [10] for pool boiling. (c) Yin and Abdelmessih [3] developed correlations for two incipience condition for increasing and decreasing heat flux, $\theta = 90°$, and from experimental observations they have correlated the values of δ^*/r_c with heat flux. (d) $\theta = 90°$ and $\delta^*/r_c = a - bq$ resulting the equation developed by Agrawal [24], Ali and Alam [25] and Kamil et al. [27],

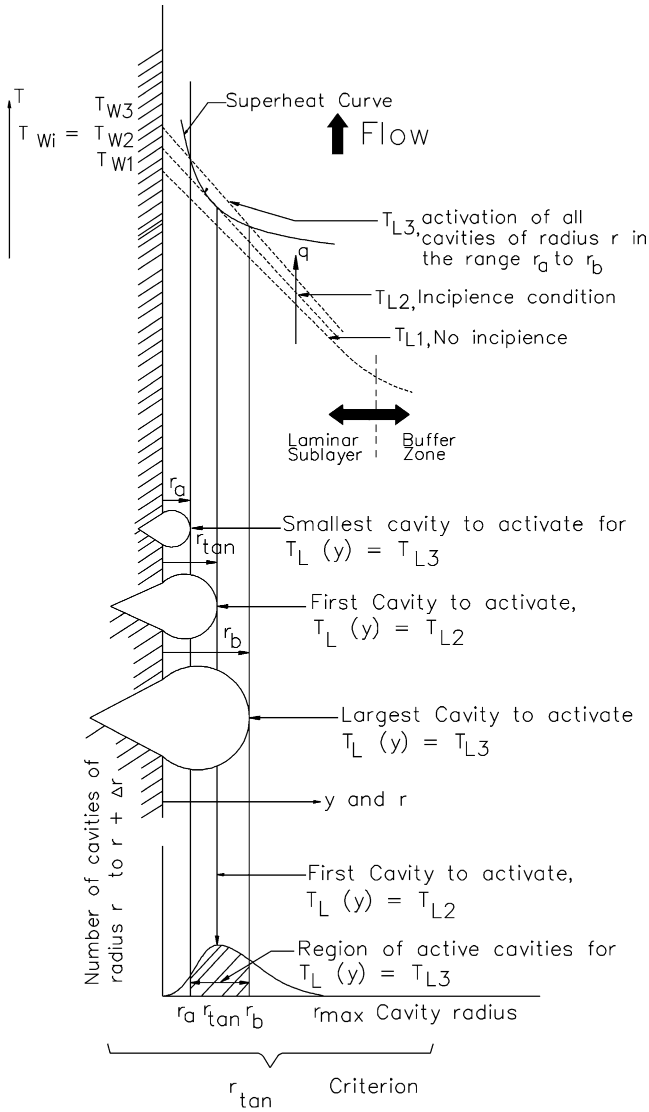

Fig. 3. Schematic of incipience conditions based on the r_{tan} criterion.

where they have proposed different values of a and b for different organic liquids and water.

The incipience of nucleate boiling depends on the existence of active cavities as well as a temperature profile that can satisfy the superheat requirement. If a sufficiently wide range of cavity size is involved, the first cavities to become active will be those corresponding to the point of tangency $y = r_{tan}$ of the superheat curve and the temperature profile curve as shown schematically in Fig. 3. The temperature profiles in Fig. 3 are shown as curved purely for illustrative purpose and the analytical treatment are confined to the linear portion of the curve. It is also based on assumption that r_{tan} is less than the superheated sublayer thickness y_{lam} for the equation which deals by r_{tan} criteria. In r_{tan} criteria Eq. (1) can be plotted for the system as all properties are evaluated at the saturation temperature. Nucleation should occur at the heat flux which brings the two curves tangent to each other or $T_L = T_b$ and $dT_b/dr_b = dT_L/dy$ at $y = r_b$. The bubble originating from a cavity of this radius can now grow. Only a small additional increase in wall temperature is necessary to activate a considerable angle of cavity sizes.

4. Results and discussion

4.1. Variation of wall and liquid temperature along the heated tube length

Fig. 4 shows the variation for wall temperature profiles along the test section with heat flux as parameter for methanol. These profiles were made from the experimental data of Kamil [26] to explain the phenomenon of boiling incipience. The wall temperature, T_w rises at a fast rate with Z up to a point beyond which a steep fall sets in followed by a gradual decrease over the remaining portion of the heated tube. The shape of the curve is almost similar for other heat fluxes and approximately the same inlet liquid subcooling. However, the curves at higher heat fluxes get shifted to higher wall temperatures. The typical behavior, as observed above, remains the same for other test liquids; the values of wall temperature, location of peak values and the lengths of various zones are different. Plots of T_w versus Z suggest that the wall temperature distributions are strongly influenced by the liquid submergence. For the lower submergence level it is observed that the initial portions before peak values of wall temperature are absent. However, the effect of heat flux remains similar at this lower submergence also.

The plots of wall temperature T_w versus tube length Z with liquid submergence as a parameter have been shown in Fig. 5. The location of wall temperature peaks get shifted towards the tube inlet and the curves move to lower values of T_w as the liquid submergence is reduced from high to low. For the submergence of 100 and 75 percent, all the three regimes of heat transfer (single phase convection, surface and saturated boiling) as shown in Fig. 5 are present, while at lower submergence of 50 percent the initial portions of curves before peak are absent. The variation of liquid temperature along the tube length has been shown in the Fig. 4 corresponding to the lower most curve of wall temperature only. The liquid temperature increases linearly with the distance along the tube length till it attains the saturation value which itself decreases linearly as the liquid moves upwards due to the reduction of the hydrostatic head [26–29].

The typical variation of wall and liquid temperatures as observed, indicates that there exist different regimes of heat transfer in a reboiler tube. The linear rise in the temperature

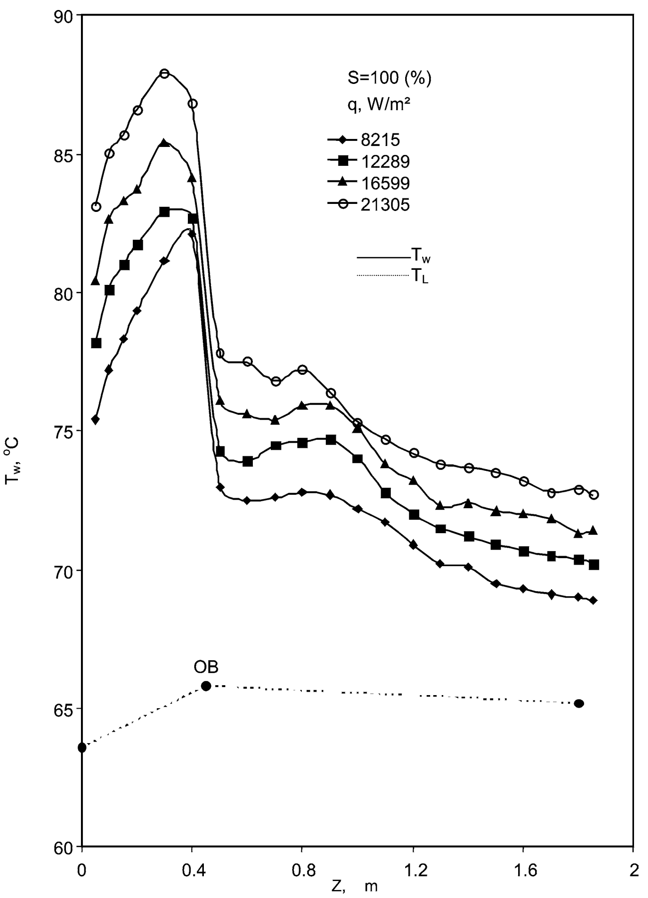

Fig. 4. Wall temperature profiles along the test section with heat flux as parameter for methanol.

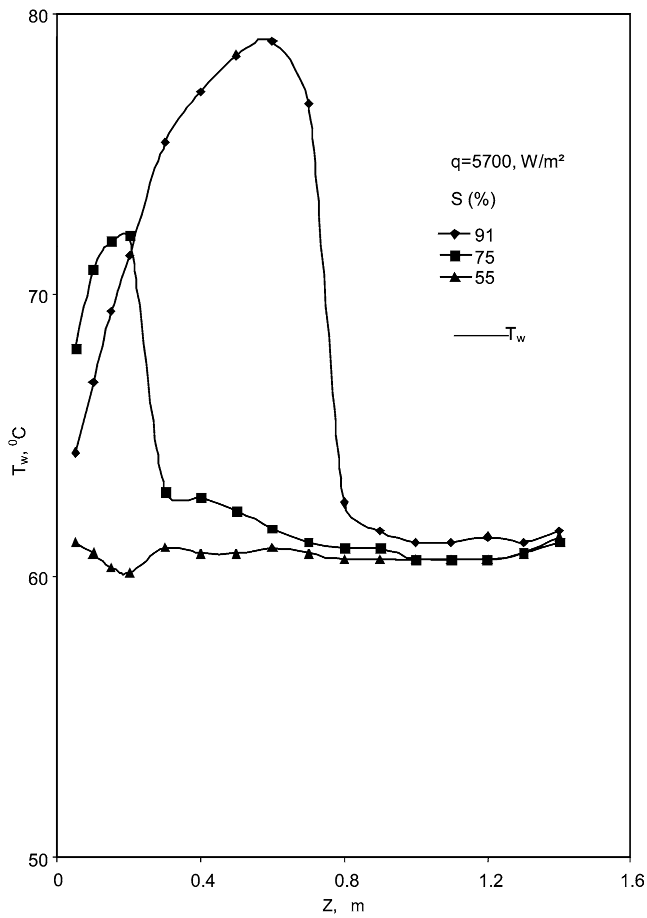

Fig. 5. Wall temperature profiles along the test section with submergence as parameter for acetone.

of liquid as it moves upwards through the tube results from sensible heating under uniform heat flux. When the minimum wall superheat required is attained, the bubbles start nucleating at the surface but collapse there due to the presence of subcooled liquid core. The onset of subcooled boiling thus creates additional turbulence at the surface. This explains why the linearly increasing wall temperature corresponding to convective heat transfer, starts varying at decreasing rate eventually becoming zero at peak values (Figs. 4 and 5). Once the bulk liquid temperature attains saturation value the bubbles generated at the surface grow to their maximum size and get detached resulting in the existence of vapor phase in the tube. All the heat supplied gets absorbed as latent heat of vaporization converting the liquid to vapor. The two phase flow moves upwards through the tube with increasing quantity of vapor and hence changing flow patterns. This corresponds to saturated boiling regime as exhibited by the slowly decreasing wall and liquid temperature profiles.

As the value of heat flux is raised, the wall temperature also increases in order to provide adequate temperature difference for transferring the additional heat. In convective mode, this should be almost in the same ratio as that of heat flux change. But in nucleate boiling it is not so because the increased heat flux enables larger number of nuclei for bubble generation becoming active and thus enhance the heat transfer coefficient and requiring small temperature difference. This explains the shifting of wall temperature curves with heat flux as observed in Figs. 4 and 5. The point, where saturated boiling set in, gets shifted to a lower level in the tube as the submergence is reduced from 100 percent, probably due to the change in circulation rate. The decrease in the value of liquid submergence reduces the driving force for liquid circulation and hence its rate through reboiler tube. At a lower rate of liquid circulation the rate of change in its temperature along the tube length becomes higher and the saturation temperature is attained at much smaller length from the inlet.

4.2. Boiling incipience

Eq. (12) is the general proposed model for $r_{\tan}$ criteria, which can be used for the incipient boiling at wide range of pressure drop, surface tension and cavity sizes. Davis and Anderson [14] suggested that the size of active cavities that are involved at the incipience of nucleate boiling under some normal engineering conditions can be of the order of 1 µm. The validity of their criteria was proven in many practical applications where the fluid is not highly wetting and therefore this value of r_c was taken in the present analysis.

Figs. 6–9 show the heat flux versus wall superheat curve for ethanol, benzene, water, and toluene respectively. These figures clearly show that the predicted values are in good agreement with the experimental data. The range of parameters covered in developing and validating the correlation is given in Table 2 for all systems. Almost a similar trend has been shown for other test liquids at constant submergence. At constant submergence, with increase in heat flux the wall superheat also increases. For low heat flux, the submergence has less importance on incipient boiling in comparison to high heat flux. None of the correlations predict the data well and generally most of the correlations under predict the wall superheat values for different systems. But Hino and Ueda [21], and Sudo et al. [19] correlations for $r_{\tan} > r_{\max}$ criteria over predicts the wall superheat values for some of the liquids (namely water and ethylene glycol) at different submergence. A comparison of the proposed model with other investigators has been shown in Table 3, giving mean absolute deviation (MAD) of the proposed model and other investigators correlation. The value of superheat and MAD has also been calculated from Eq. (12), which clearly exhibit that an important parameter is missing in their correlations, which may be submergence. Therefore, it was necessary to include the effect of submergence in Eq. (12) as discussed earlier. The values of exponent 'n' in Eq. (12) with a maximum error (%) are given in Table 2 for all the systems.

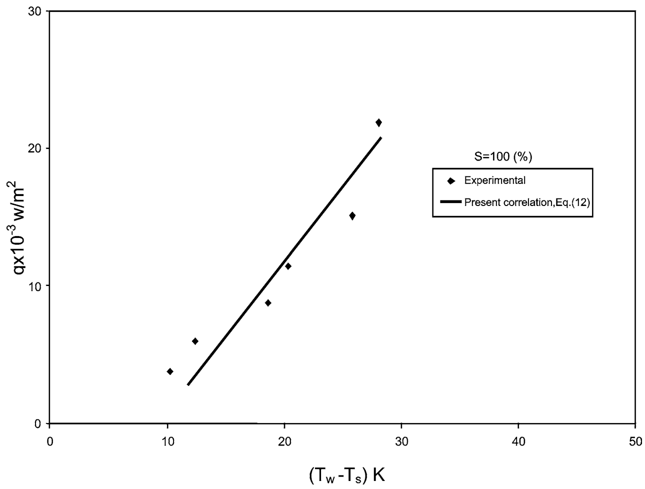

Fig. 6. Heat flux versus degree of superheat at boiling incipience for ethanol.

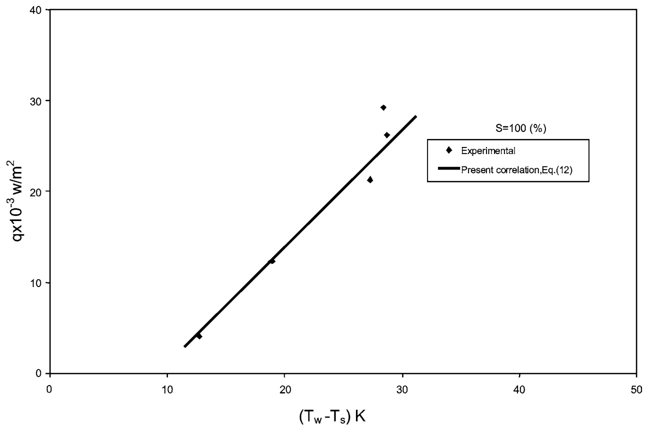

Fig. 7. Heat flux versus degree of superheat at boiling incipience for benzene.

Table 2
Range of parameters and values of exponent and corresponding maximum percent error for different systems in proposed model, Eq. (12)

Systems	Submergence (Percent)	ΔT_{sub} [°C]	Heat Flux [$W \cdot m^{-2}$]	Exponent 'n' Eq. (12)	Maximum Error [%]
Acetone	30–100	0.2–45.5	3548–15115	0.8006	±19
Methanol	30–100	1.0–3.7	4105–21305	0.7997	±20
Ethyl acetate	30–100	2.5–44.5	3548–14500	0.6672	±20
Ethanol	30–100	1.1–21.6	3800–21884	0.7384	±11
Benzene	30–100	0.7–3.6	4106–29225	0.6601	±15
Propanol	40–100	1.2–54.2	3342–21765	0.6754	±16
Water	30–100	0.2–73.0	3486–43373	0.6846	±16
Toluene	30–100	1.9–68.3	2042–32085	0.6293	±15
Ethylene glycol	30–100	3.25–15.8	15115–33654	0.6158	±17

Table 3
Comparison of various correlations with experimental data

Mean Absolute Deviation (MAD)										
Systems		Acetone	Methanol	Ethyl acetate	Ethanol	Benzene	Propanol	Water	Toluene	Ethylene glycol
Sato and Matsumura [13]		95.93	95.49	91.59	94.97	91.16	93.32	92.60	91.5	90.37
Bergles and Rohsenow [11]		90.67	89.24	87.70	89.19	87.99	86.86	84.19	88.73	88.13
Han and Griffith [10]		95.02	94.48	89.70	93.84	90.40	91.82	90.94	89.59	88.21
Davis and Anderson [14]		95.93	95.49	91.59	94.97	91.16	93.32	92.60	91.50	90.37
Frost and Dzakowic [15]		86.59	77.03	47.46	54.00	63.36	22.69	87.05	63.71	76.43
Yin and Abdelmessih [3]	* *	88.87	89.29	78.92	87.17	83.04	82.50	82.22	80.78	86.52
	# #	96.78	96.44	93.35	96.02	93.80	94.72	94.15	93.29	92.39
Hino and Ueda [21],	$r_{tan} < r_{max}$	95.93	95.49	91.59	94.97	91.16	93.32	92.60	91.50	90.37
Sudo et al. [19]	$r_{tan} > r_{max}$	56.71	58.43	37.09	51.43	36.04	37.36	209.0	25.63	69.83
Marsh and Mudawar [22]		92.39	91.57	84.27	90.60	85.35	87.51	86.16	84.09	81.99
Kureta et al. [30]		76.78	74.02	69.91	73.74	71.22	67.63	61.44	72.72	72.81
Author's correlation Eq. (12)	$r_{tan} < r_{max}$	17.31	18.05	18.18	8.24	13.94	13.09	12.01	11.99	15.55

* * = Increasing Heat Flux
= Decreasing Heat Flux

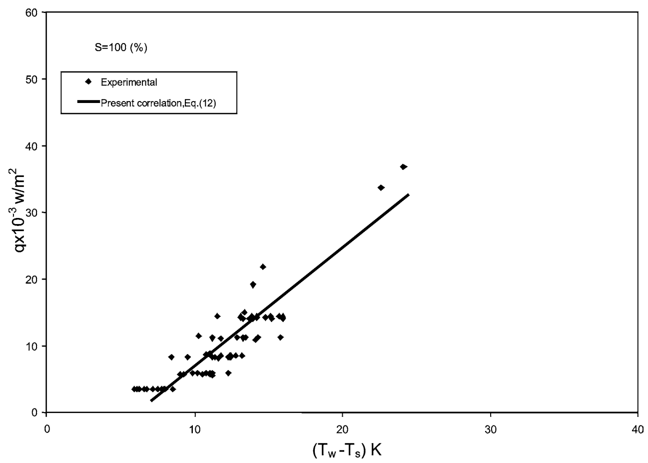

Fig. 8. Heat flux versus degree of superheat at boiling incipience for water.

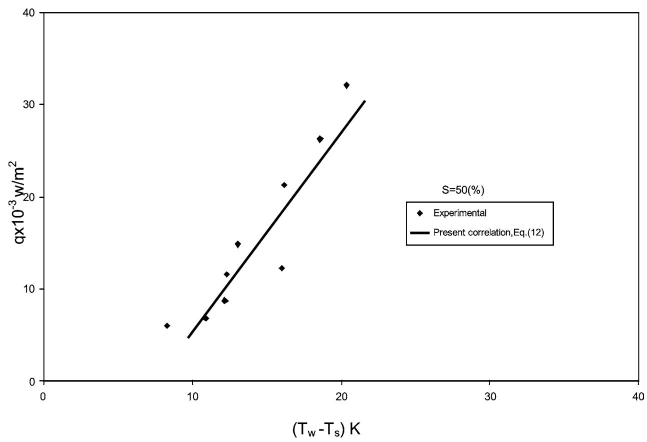

Fig. 9. Heat flux versus degree of superheat at boiling incipience for toluene.

Figs. 10 and 11 show the plot of degree of superheat versus submergence for ethanol and ethylene glycol, respectively. From these plots it is clear that the superheat increases linearly with increase in submergence for a constant heat flux. These lines are almost parallel to each other. As the value of heat flux is decreased, the lines shift to a lower level as exhibited in Figs. 10 and 11, respectively. It is therefore clear that the submergence has a strong effect on the condition of the onset of nucleate boiling.

Figs. 12 and 13 show the heat flux versus superheat plots at different submergences for acetone and toluene, respectively. From these figures it is clear that the predicted results agree well with experimental data at constant submergence.

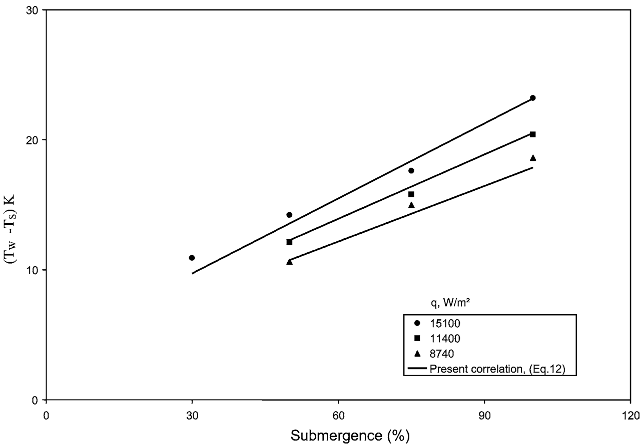

Fig. 10. Degree of superheat versus submergence for ethanol.

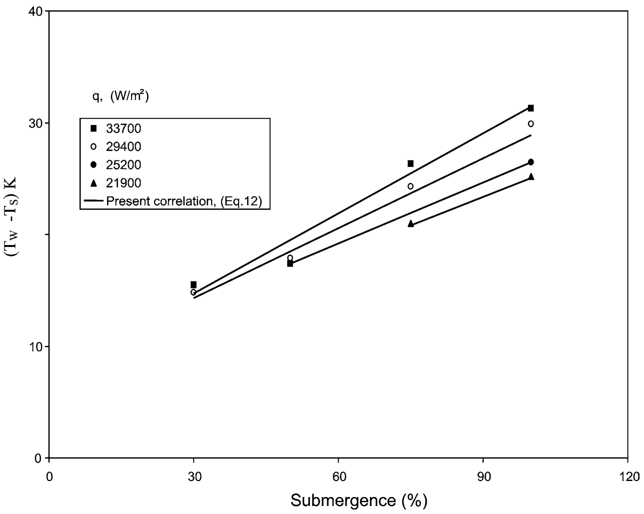

Fig. 11. Degree of superheat versus submergence for ethylene glycol.

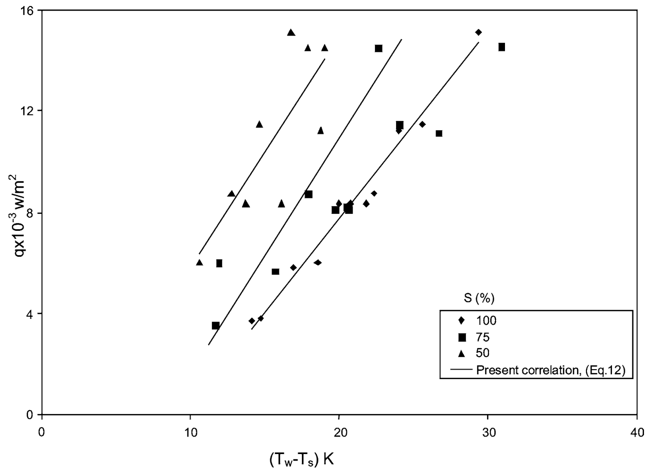

Fig. 12. Heat flux versus degree of superheat at boiling incipience for acetone.

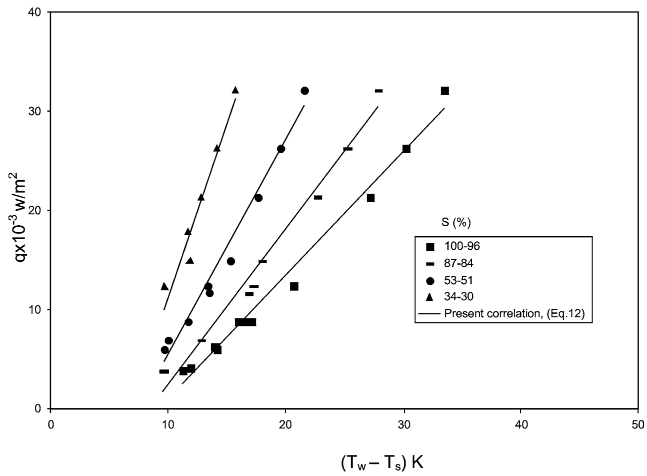

Fig. 13. Heat flux versus degree of superheat at boiling incipience for toluene.

These figures also show that at different values of submergence, there are different predicted lines.

Figs. 14–17 show the comparison of the experimental wall superheat with predicted superheat (Eq. (12)) for ethanol, water, toluene and ethylene glycol, respectively. The majority of data points lie within considerable error limits. The maximum error for different systems is tabulated in Table 2.

The absolute deviation is given as:

$$\text{Percent deviation} = \frac{\Delta T_{\text{pred}} - \Delta T_{\text{exp}}}{\Delta T_{\text{exp}}} \times 100 \tag{13}$$

The mean absolute deviation (MAD) is given as:

$$\text{MAD} = \frac{1}{N}\sum_{i=1}^{N}\left|\frac{\Delta T_{\text{pred}} - \Delta T_{\text{exp}}}{\Delta T_{\text{exp}}}\right| \times 100 \tag{14}$$

Where N is the number of experimental data points.

An effort was also made to obtain a unified correlation using the data of all the systems together and the optimum value of parameter n in Eq. (12) was found to be 0.67079.

$$(T_w - T_s) = \left[\frac{8\sigma T_s q}{k_L \rho_v h_{\text{fg}}\left(1+\frac{2\sigma}{r_c P_s}\right)\left[1-\frac{R T_s}{h_{\text{fg}}}\ln\left(1+\frac{2\sigma}{r_c P_s}\right)\right]^2}\right]^{1/2} \times S^{0.67079} \tag{15}$$

Fig. 18 shows the plot of comparison of experimental wall superheat with those predicted by the proposed correlation covering all the data of the nine different liquids with widely varying thermophysical properties. The ranges of parameters covered in developing and validating the correlation is given in Table 2. It was observed that the majority of data points of present study lie within the maximum error of ±22% and mean absolute deviation of 16%.

5. Conclusions

Key conclusions from the study are as follows:

(1) The analytical study shows that the onset of fully developed boiling requires a minimum degree of wall

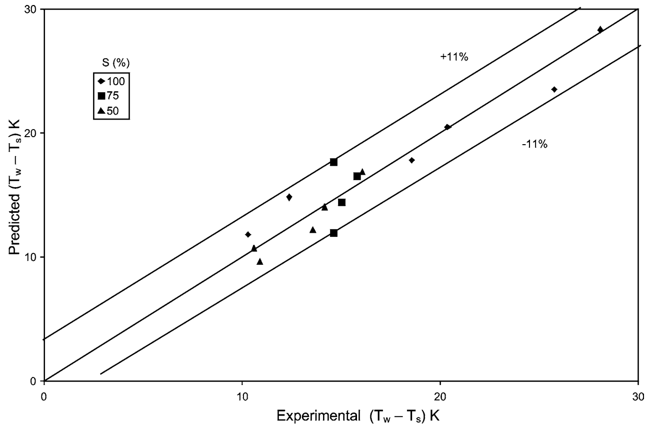

Fig. 14. Comparison between experimental and predicted values of superheat for ethanol.

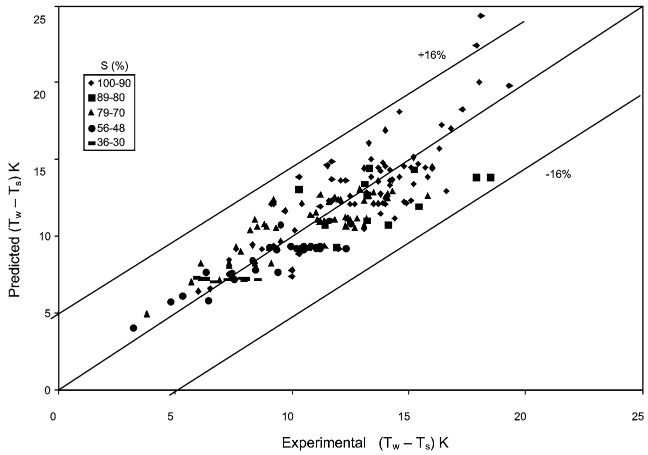

Fig. 15. Comparison between experimental and predicted values of superheat for water.

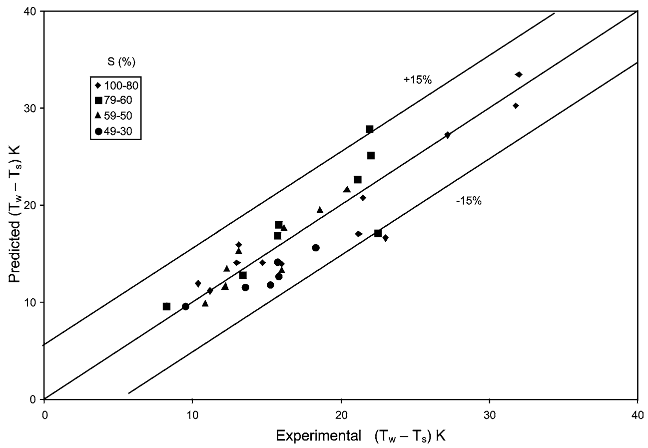

Fig. 16. Comparison between experimental and predicted values of superheat for toluene.

superheat for a given liquid and heat transfer surface. (ii) At constant submergence as the heat flux increases, the wall superheat required for incipient boiling increases. (iii) The wall superheat increases linearly with submergence for a constant heat flux. At low values of heat flux, the role

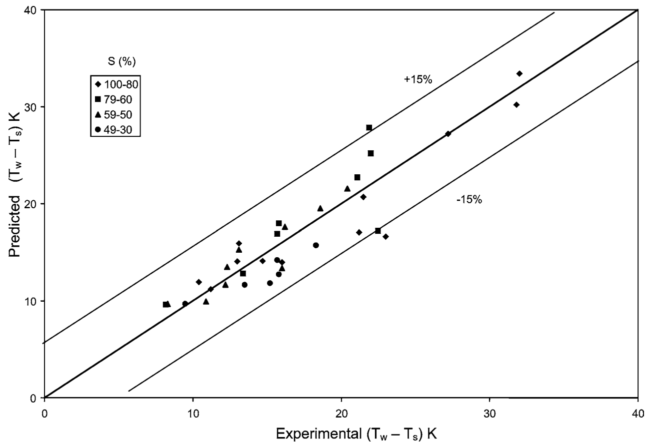

Fig. 17. Comparison between experimental and predicted values of superheat for ethylene glycol.

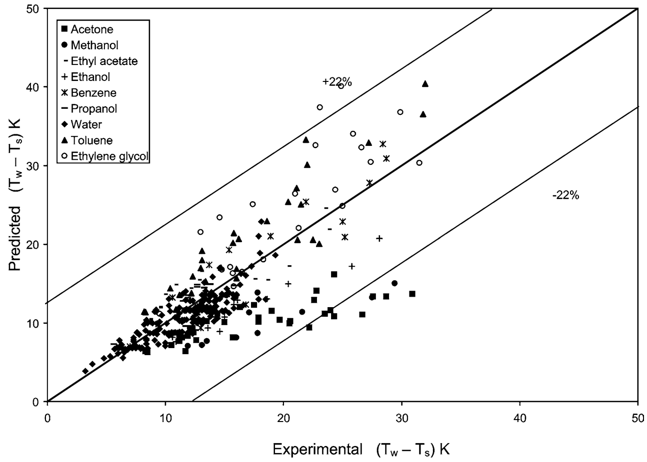

Fig. 18. Comparison of experimental degree of superheat with those predicted by proposed correlation Eq. (15) for all test liquids [24,26,31].

of submergence is less important in comparison to that at high heat flux. (iv) Based on the theoretical analysis, Eq. (12) has been developed, which can predict the wall superheat required for onset of boiling at a given heat flux, submergence, pressure and cavity radius using the thermophysical properties of test liquids. The data available from various sources were correlated by a unified equation for all the liquids with a maximum error of ±22%. (vi) The form of the equation is fairly general and may be extended to other ranges and systems.

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