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Analysis of Incipience of Nucleate Boiling in a Reboiler Tube

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An analysis has been carried out to predict the boiling incipience with natural circulation flow in a reboiler tube. An equation has been proposed to estimate the wall superheat for different types of liquids covering a wide range of thermo physical properties. Incipience in liquid films is influenced both by turbulent eddies and liquid submergence. This approach utilizes physical parameters of commonly accepted incipience models and takes into account the effects of turbulent eddies and submergence. The results predicted from the theoretical analysis were found to be consistent with the experimental data, having average absolute relative error of 15.42 percent. © 2006 American Institute of Chemical Engineers *AIChE J,* 53: 39–50, 2007
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Introduction

Nucleate boiling of liquids in vertical tube closed loop thermosiphon reboilers with natural circulation is of common occurrence in a variety of process equipments such as reboilers, vaporizers, and evaporators used in process industries. In most of these applications, it is important to predict the operating conditions that trigger nucleate boiling at the wall. Generally, a subcooled liquid entering the tube gets heated by single phase convection and moves upwards. Depending upon wall temperature conditions, subcooled boiling may set in at the surface. When the liquid temperature attains saturation value, saturated boiling begins with generation of vapor, which increases along the tube length, resulting in bubbly to mist flow. Thus, the heat transfer to liquids in the reboiler tube generates a changing two-phase flow with various flow regimes spread along the tube length. The point at which the two-phase begins is known as the incipient point of boiling (IPB). It corresponds to the conditions of minimum degree of superheat or heat flux required for the formation and detachment of the first vapor bubble from the heated surface. Since IBP divides the tube in two distinct regions, non-boiling single phase and boiling two phases with entirely different modes of heat transfer, its prediction is very important in the design of two phase heat transfer equipment.

Incipience in forced convection systems has been studied extensively by a number of investigators, such as Collier,[1] Hsu,[2] Bergles and Rohsenow,[3] Sato and Matsmura,[4] Davis and Anderson,[5] Frost and Dzakowic,[6] Unal,[7] Yin and Abdelmessih,[8] Sudo et al.,[9] and Hino and Ueda,[10] among others. The predicted incipience is either based on the point of tangency between the liquid temperature profile in the vicinity of the heated surface and the superheat temperature profile required for mechanical equilibrium of a vapor bubble growing at a surface cavity or based on the maximum cavity radius available for nucleation on the heated surface. Han and Griffith[11] proposed an analysis similar to Hsu[2] for the nucleate pool boiling. Unal[7] considered the effect of pressure on the boiling incipience under subcooled flow boiling of water in a vertical tube. Celata et al.[12] carried

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out an experimental work to determine the flow pattern map in vertical heated pipes under steady state and transient conditions using Freon-12 in forced convective flow. From the analysis of the experimental measurements, they obtained a map for annular and intermittent flow regimes. Chen, Klausner, and Mei[13] proposed a simplified model for predicting vapor bubble growth in heterogeneous boiling. Wang et al.[14] studied a new type of dynamic instability in a forced convection up flow boiling system at the boiling onset oscillation. Rashidnia[15] studied bubble dynamics on a heated surface. Reed and Mudawar[16] examined the effectiveness of the zero-angle cavities created by curved contact with a flat surface in promoting vapor embryo capture, low temperature boiling incipience, and reduction or elimination of the incipience temperature overshoot with highly wetting liquids. Agarwal,[17] Ali and Alam,[18] Kamil,[19] and Kamil et al.[20,21] experimentally determined the boiling and non-boiling zones for the heating surface and superheat for incipient boiling in a vertical tube thermosiphon reboiler with wide ranges of submergence. A dimensionless correlation relating the values of heat flux, inlet liquid subcooling, and submergence was proposed for predicting Z_{OB}/L and wall superheat relating the heat flux with thermophysical properties of test liquids as given in Table 1. Shim et al.[22] developed an analytical model for fully developed turbulent flow and heat transfer in finned annuli using a modified mixing-length turbulence model. They had extended the model to predict the conditions at the onset of nucleate boiling using the criterion of Davis and Anderson.[5] Again, these predictions agreed well in magnitude and trend with experimental data. Recently, Kamil et al.[23] investigated the effect of liquid submergence on incipience of nucleate boiling in a vertical thermosiphon reboiler.

There are two different criteria for the incipient point of boiling—one is r_{tan}, and the other is r_{max}. The incipience in forced convection systems was studied extensively by a number of workers as given in Table 1, and they predicted incipience based on the point of tangency (r_{tan}). The validity of the criterion was proven in many practical applications. Incipience based on r_{tan} criteria for natural convection system was studied by Agarwal,[17] Ali and Alam,[18] Kamil,[19] and Kamil et al.[20–24] Sudo et al.[9] and Hino and Ueda[10] used a different hypothesis in their forced convection study. They predicted incipience based on the maximum cavity radius (r_{max}) available for nucleation on the heated surface. Thus, it is evident from the literature that in the region of incipient boiling, the thickness of the superheated layer changes wavily and the heat is not transferred by pure heat conduction alone.

Therefore, the present study focuses on the prediction of boiling incipience in a vertical thermosiphon reboiler including the effect of submergence and turbulent eddies. In the present work, experimental data of Agarwal,[17] Ali and Alam,[18] and Kamil[19] are compared with the predicted values from existing incipience models, to evaluate their applicability to a vertical thermosiphon reboiler. A semi-empirical model has been developed including the effect of submergence and turbulent eddies to predict the onset of nucleate boiling for the systems, namely, acetone, methanol, ethyl acetate, ethanol, benzene, propanol, distilled water, toluene, and ethylene glycol.

Experimentation

The experimental rig was in the form of a U-shaped circulation system made of two long vertical stainless steel tubes whose lower ends were connected by a small horizontal tube, while a vapor liquid separator and condenser vessel were fitted to the upper end, as shown in Figure 1. The stainless steel tube, which served as the test section, was 25.56 mm I.D. and 28.55 mm O.D. Out of a total length of 2100 mm, a section of 1900 mm was tapped between two thick copper clamps, which were designed to provide electrical connection to the tube with almost negligible contact resistance. In order to monitor the heat transfer surface temperature 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 intervals over the remaining length. A copper-constantan thermocouple was used to measure the inlet liquid temperature. The temperature of the boiling liquid, before entry to the vapor liquid separator, was measured by another traversing thermocouple probe. The condensed liquid was drained from the bottom of the condenser vessel through a vertical tube 50 mm I.D. and 300 mm long. A level indicator was attached with this tube to indicate the liquid level in it. The cooling water to the condensers was circulated by means of a centrifugal pump from a small storage tank to which freshwater supply was maintained. The entire set-up was thoroughly lagged to make the heat losses negligible, less than ±2.5%. 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 cold liquid head (submergence) could be varied independently at 75, 50, and 30%. The experimental data were generated by varying the heat fluxes at atmospheric pressure. The range of parameters covered is given in Table 2 for all the systems. Other details of the reboiler and cooling system, along with its operating procedure, are described in detail elsewhere in the literature.[19–21]

Analysis and the Wall Superheat Equation

The basic assumptions made in the present theoretical analysis are:

- The potentially active cavities are of conical shape, and the bubble nucleus that forms at such a surface has the shape of a truncated sphere.
- Bubbles grow and detach from the nucleation sites only when the superheated liquid layer is thick enough so that a net heat flux into the developing bubble is realized.
- The equation for the superheated vapor temperature profile T_b can be obtained from the most extensively used Clausius-Clapeyron relation. Assuming a constant value for $T/h_{fg}\rho_v$ within the wall superheat range associated with incipience, gives the following relationship:

$$T_b = T_s + \frac{2\sigma T_s}{h_{fg}\ \rho_v\ r_b} \tag{1}$$

- The liquid temperature within the thermal layer has a turbulent film temperature profile.

Table 1. Summary of Important Boiling Incipience Investigations

Authors (year)	Flow Geometry	Heater Material	Fluid	Mean Velocity (m s^{-1})	Pressure [bar (psia)]	Sub cooling (°C)	Incipience Formula
Sato and Matsumura (1964)	Vertical channel	Stainless steel	Water	0.6–4.1	1.0 (14.7)	3–70	• Tangency criterion, $r = k_L\,(T_w - T_s)/2q$ $q = \frac{k_L\,\rho_v h_{fg}}{8\sigma\,T_s}(T_w - T_s)^2$
Bergles and Rohsenow (1964)	Horizontal annutus	Stainless steel	Water	3.3–17.4	Up to 2.6 (38.0)	32–90	$q = 15.60\,P^{1.156}\,(T_w - T_s)^{230/P^{0.023}}$ (P in psia) • Graphical solution for water over a pressure range of 15-2000 psia based on tangency criterion
Han and Griffith (1965)	Pool boiling on horizontal surface	Gold finished with 600 grit emery paper	Water	– –	1.0 (14.7)	7	$q = \frac{k_L\,\rho_v h_{fg}}{12\sigma\,T_s}(T_w - T_s)^2$ • Tangency criterion, $y = 1.5r$ at the point of tangency
Davis and Anderson (1966)	Authors performed analysis using experimental data from prior studies.						$q = \frac{k_L\,\rho_v h_{fg}}{8C_1\sigma\,T_s}(T_w - T_s)^2$ • Tangency criterion, $y = r$ at the point of tangency • $C_1 = 1$ for hemispherical bubble nucleus
Frost and Dzakowic (1967)	Authors performed analysis using experimental data from prior studies.						$q = \frac{k_L\rho_v h_{fg}}{8\sigma\,T_s}(T_w - T_s)^2\frac{1}{Pr_L^2}$ • Tangency criterion, $y = Pr_L{}^2 r$ at the point of tangency.
Yin and Abdel-messih (1977)	Vertical tube	Stainle-ss steel	Freon-11	0.08–0.4	Up to 2.0 (30.0)	1.20	• For increasing heat flux: $q = \frac{1}{\left[7 - \frac{q}{6500}\right]}2\frac{k_L\rho_v h_{fg}}{2\sigma T_s}(T_w - T_s)^2$ • For decreasing heat flux: $q = \frac{k_L\rho_v h_{fg}}{5\sigma T_s}(T_w - T_s)^2$ • Tangency criterion, y/r at the point of tangency correlated empirically.
Hino and Ueda (1985)	Vertical annulus	S.S finished with 4/0 emery cloth	Freon-113	0.1–1.0	1.47 (22.0)	10–30	• For $r_{\tan} < r_{\max}$: $q = \frac{k_L\rho_v h_{fg}}{8\sigma T_s}(T_w - T_s)^2$ • For $r_{\tan} > r_{\max}$: $q = \frac{k_L}{r_{\max}}(T_w - T_s) - \frac{2\sigma\,T_s k_L}{\rho_v h_{fg} r_{\max}^2}$

(*continued*)

Table 1. (Continued)

Authors (year)	Flow Geometry	Heater Material	Fluid	Mean Velocity (m s^{-1})	Pressure [bar (psla)]	Sub cooling (°C)	Incipience Formula
Sudo et al. (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 $r_{\tan} > r_{\max}$: $q = \frac{k_L}{r_{\max}}(T_w - T_s) - \frac{2\sigma T_s k_L}{\rho_v h_{fg} r_{\max}^2}$
Kamil, Shamsuzzoha, and Alam (2005)	Authors performed analysis using experimental data from prior studies						$(T_w - T_s) = \left[\frac{8\sigma\, T_s q}{k_L\, \rho_v h_{fg}(1+\frac{2\sigma}{r_c P_s})[1-\frac{RT_s}{h_{fg}}\ln(1+\frac{2\sigma}{r_c P_s})]^2}\right]^{1/2}$ $x\ S^{0.67079}$
Kamil, Hakeem, and Shamsuzzoha (Present Study)	Authors performed analysis using experimental data from prior studies for acetone, methanol, ethyl acetate, ethanol, benzene, propanol, water, toluene, and ethylene glycol.						$(T_w - T_s) = (1.1043)\left[\frac{8\sigma T_s q}{k_L \rho_v h_f g}\right]^{1/2}(S)^{0.59867}$

Assuming a constant wall flux, the temperature gradient and mean liquid temperature in the turbulent boundary layer as given by Collier[1] can be written as:

$$\frac{dT_L}{dy} = -\frac{q}{\rho C_p(\alpha + \varepsilon_H)} \tag{2}$$

Thus, finally, after simplification, we get:

$$\frac{dT_L}{dy} = -\frac{q}{k_L}\frac{1}{\left[1+\frac{Pr_L \varepsilon_m}{Pr_t \nu_L}\right]} \tag{3}$$

After integrating Eq. 3, it is possible to evaluate T_L as given below:

$$T_L = T_w - \frac{q}{k_L}\int_0^y \frac{1}{\left[1+\frac{Pr_L \varepsilon_m}{Pr_t \nu_L}\right]}dy \tag{4}$$

The radius of the first cavity at which boiling occurs can be determined by equating the slopes of T_L and T_b, as suggested by Bergles and Rohsenow[3]:

$$\frac{dT_b}{dr_b} = \frac{dT_L}{dy} \tag{5}$$

Taking the first derivative of Eq. 1:

$$\frac{dT_b}{dr_b} = -\frac{2\sigma T_s}{h_{fg}\ \rho_v\ {r_b}^2} \tag{6}$$

according to Davis and Anderson,[5] the tangency radius is:

$$r_{\tan} = \left[\frac{2\sigma T_s k_L}{q\ h_{fg}\ \rho_v}\right]^{1/2} \tag{7}$$

Equating Eq. 3 and Eq. 6, we get:

$$-\frac{q}{k_L}\frac{1}{\left[1+\frac{Pr_L\ \varepsilon_m}{Pr_t\ \nu_L}\right]} = -\frac{2\sigma T_s}{h_{fg}\ \rho_v\ r_{\tan}^2} \tag{8}$$

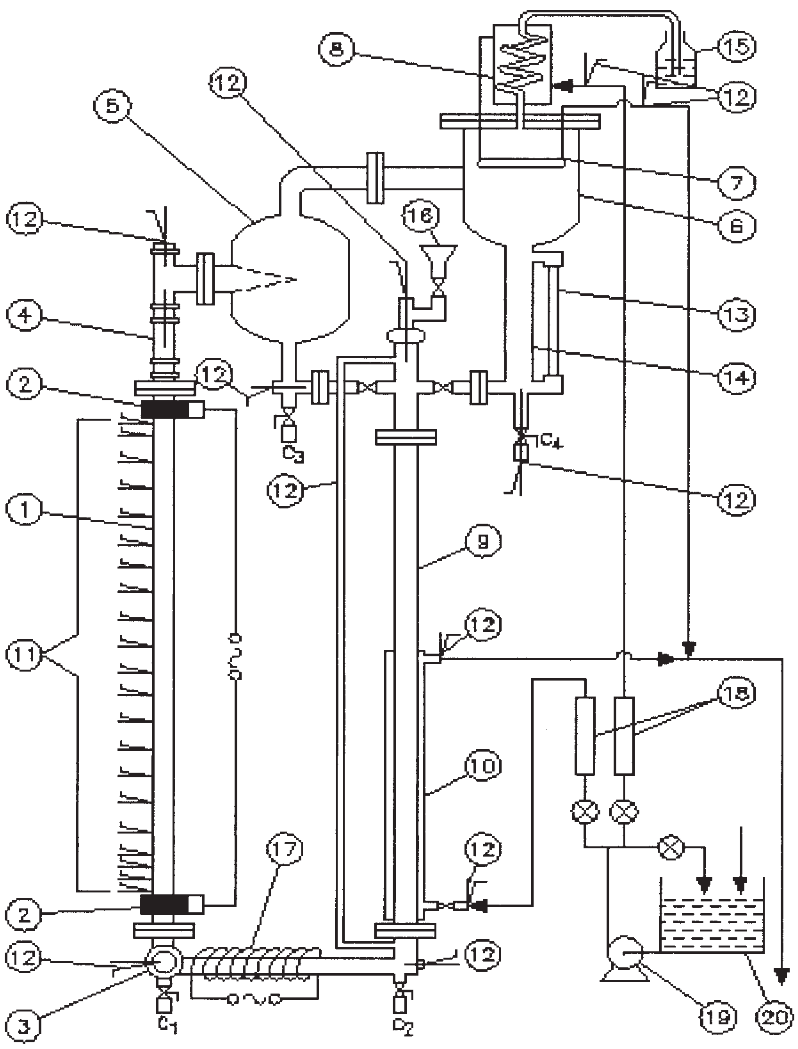
Thus, after simplification and using the tangency criterion, we get:

$$r_{\tan} = \left[\frac{2\sigma T_s k_L}{q\ h_{fg}\ \rho_v}\right]^{1/2}\left[1+\frac{Pr_L \varepsilon_m}{Pr_t \nu_L}\bigg|_{r_{\tan}}\right]^{1/2} \tag{9}$$

Equation 9 for the tangency radius is similar to Eq. 7 except for the turbulent eddy diffusivity term. Thus, from Eq. 9, it is clear that $r_{\tan}$ will be greater for turbulent films than predicted by Eq. 7 as reported in the literature. Because

$$\left[1+\frac{Pr_L\ \varepsilon_m}{Pr_t\ \nu_L}\bigg|_{r_{\tan}}\right]^{1/2} \geq 1 \tag{10}$$

at the point of tangency, the temperature T_b and T_L, in addition to their slopes, must be equal. Thus, equating Eqs. 1



1	Test Section	7	Spiral coil	14	Condenser down flow pipe
2	Copper Clamps	8	Secondary condenser	15	Bubbler
3	View port for inlet liquid	9	Liquid down-flow pipe	16	Feeding funnel
4	Glass tube section	10	Cooling jacket	17	Auxiliary heater
5	Vapor-liquid separator	11	Wall thermocouples	18	Rotameters
6	Primary condenser	12	Liquid thermocouple probes	19	Centrifugal pump
		13	Liquid level indicator	20	Cold water tank

Figure 1. Experimental setup.

and 4, we get:

$$T_w - \frac{q}{k_L}\int_0^y \frac{1}{\left[1+\frac{Pr_L\varepsilon_m}{Pr_t\nu_L}\right]}\,dy = T_s + \frac{2\sigma T_s}{h_{fg}\,\rho_v\,r_{\tan}}, \quad (11)$$

or

$$T_w - T_s = \frac{q\,r_{\tan}}{k_L}\int_0^1 \frac{1}{\left[1+\frac{Pr_L\varepsilon_m}{Pr_t\nu_L}\right]}\,d\left(\frac{y}{r_{\tan}}\right) + \frac{2\sigma T_s}{h_{fg}\,\rho_v\,r_{\tan}} \quad (12)$$

Substituting Eq. 9 for $r_{\tan}$ in Eq. 12 yields

$$T_w - T_s = \left[\frac{8\sigma T_s\,a}{h_{fg}\,\rho_v\,k_L}\right]^{1/2} \times \left[\frac{1}{2}+\frac{1}{2}\int_0^1 \frac{\left[1+Pr_L\varepsilon_m t\,\nu_L\Big|_{r_{\tan}}\right]}{\left[1+\frac{Pr_L\,\varepsilon_m}{Pr_t\,\nu_L}\right]}\,d\left(\frac{y}{r_{\tan}}\right)\right] \Big/ \left[\left[1+\frac{Pr_L\,\varepsilon_m}{Pr_t\,\nu_L}\Big|_{r_{\tan}}\right]^{1/2}\right] \quad (13)$$

or

Table 2. Range of Experimental Parameters

Systems	Submergence (Percent)	ΔT_{sub} (°C)	Heat Flux (W/m^2)
Acetone	44–100	0.9–26.7	3548–14500
Methanol	30–100	1.0–3.7	4105–21305
Ethyl acetate	28–97	2.5–44.5	3548–18800
Ethanol	30–100	1.1–21.6	3800–21884
Benzene	30–100	0.7–3.6	4106–29225
Propanol	39–100	1.2–54.2	3342–21765
Water	30–100	0.5–4.6	5730–43373
Toluene	30–100	1.9–8.7	4106–32085
Ethylene glycol	30–100	3.25–15.8	15115–33654

$$T_w - T_s = \left[\frac{8\sigma T_s q}{h_{fg}\,\rho_v\,k_L}\right]^{1/2}\psi_1 \tag{14}$$

where the turbulent boundary layer profile multiplier Ψ_t is defined as:

$$\psi_1 \cong \left[\frac{1}{2}+\frac{1}{2}\int_0^1 \frac{\left[1+\left.\frac{Pr_L\,\varepsilon_m}{Pr_t\,\nu_L}\right|_{r_{\tan}}\right]}{\left[1+\frac{Pr_L\,\varepsilon_m}{Pr_t\,\nu_L}\right]}\, d\left(\frac{y}{r_{\tan}}\right)\right] \Bigg/ \left[\left[1+\left.\frac{Pr_L\,\varepsilon_m}{Pr_t\,\nu_L}\right|_{r_{\tan}}\right]^{1/2}\right] \tag{15}$$

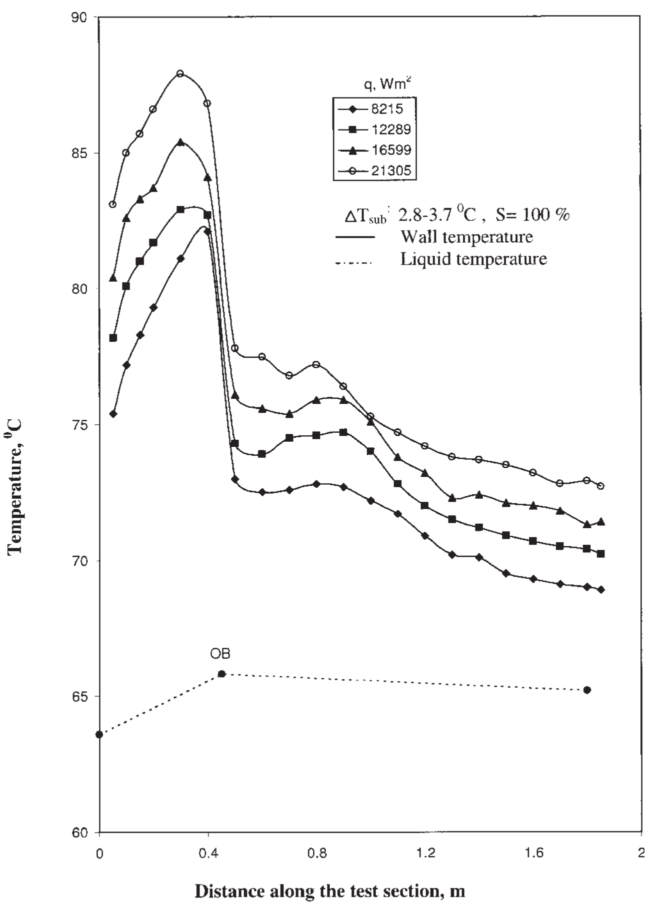


Figure 2. Wall temperature profiles along the test section with heat flux as a parameter for methanol.

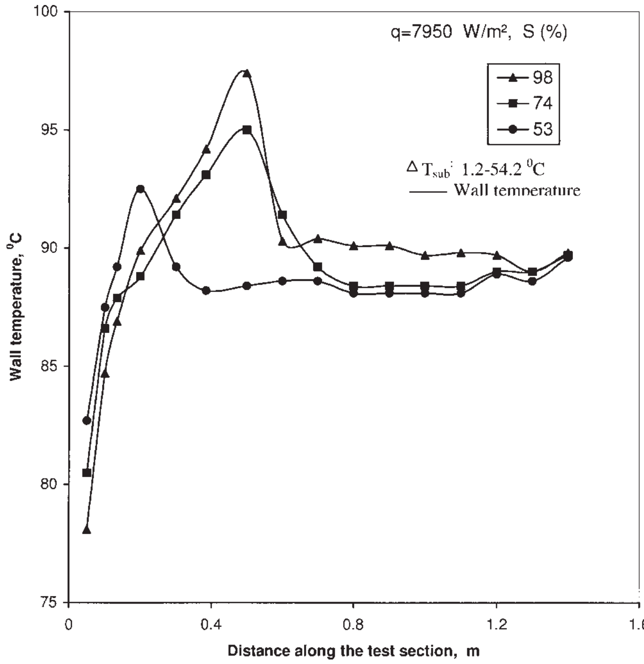


Figure 3. Wall temperature profiles along the test section with submergence as a parameter for propanol.

Thus, Eq. 14 is a general expression for incipience in turbulent boundary layers including flow in closed channels. The above equation assumes bubble growth based on the mean liquid temperature within the thermal boundary layer. Due to subcooled liquid continuously moved from the film to the wall, turbulent eddies may suppress the growth of bubbles at potential nucleation sites. This may happen despite the high mean liquid superheat available at the wall. These transient phenomena create a quenching effect on the nucleation process, which is further complicated by liquid submergence. A few workers, such as Agarwal,[17] Ali and Alam,[18]

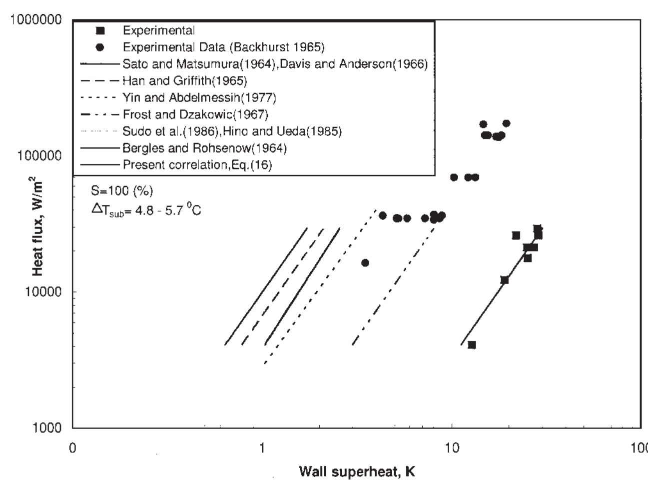


Figure 4. Heat flux versus degree of superheat at boiling incipience for benzene.

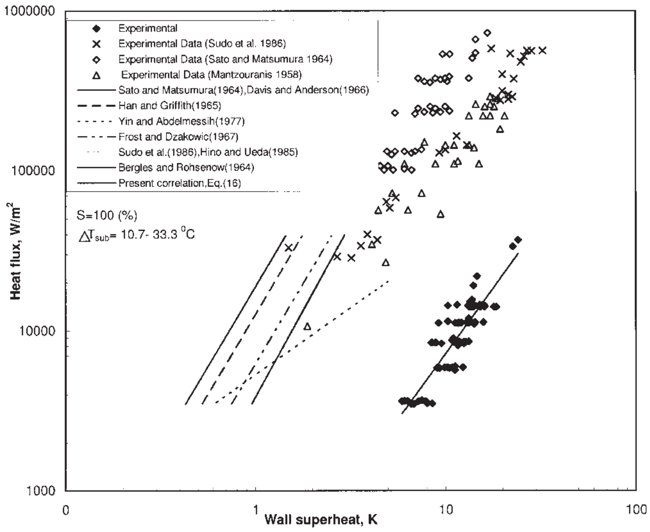


Figure 5. Heat flux vs. degree of superheat at boiling incipience for water.

Kamil,[19] and Kamil et al.,[19–21] have investigated the effect of inlet liquid subcooling and submergence on heat transfer, circulation rate, and boiling incipience in a vertical thermosiphon reboiler. In a natural circulation reboiler, the induced flow is established due to the differential head 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. Therefore, the rate of circulation depends upon liquid submergence, heat flux, inlet liquid subcooling, and frictional resistance. At a given submergence, the liquid head in the cold leg remains unchanged, while increase in the heat flux shifts the boiling incipience towards the 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 than that of higher submergence value. The detailed description of the effect of submergence on induced flow has been discussed earlier by Kamil[19] and Kamil et al.,[25] among others. Yin and Abdelmessih[8] have also investigated the effect of velocity on δ^*/r_c. Thus, the submergence has an important effect on boiling incipience in the case of a nat-

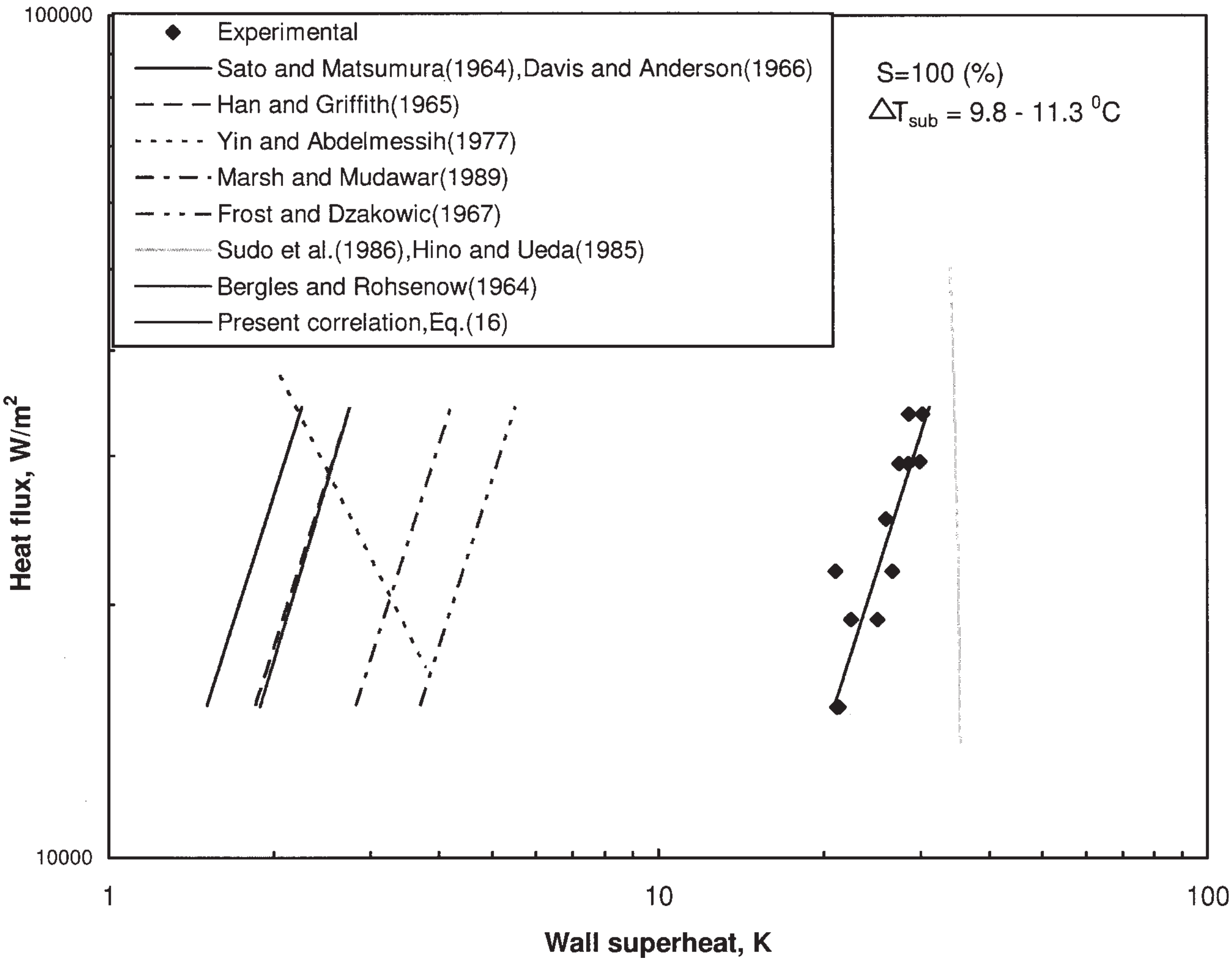


Figure 6. Heat flux versus degree of superheat at boiling incipience for ethylene glycol.

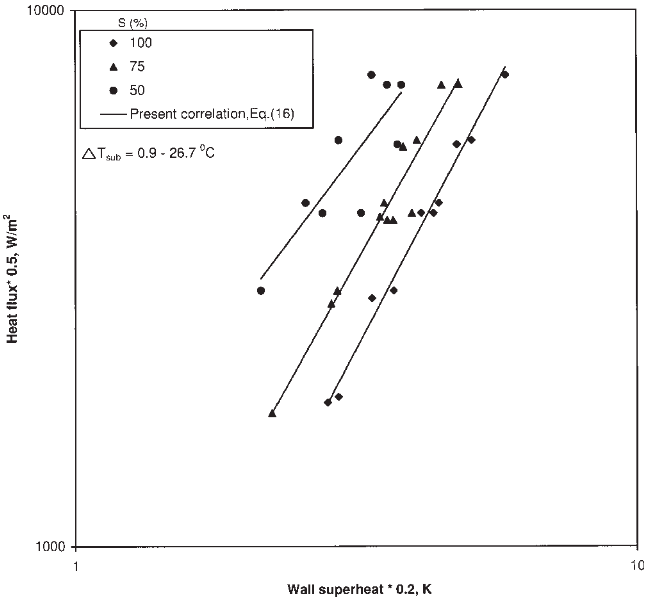


Figure 7. Heat flux versus degree of superheat at boiling incipience for acetone.

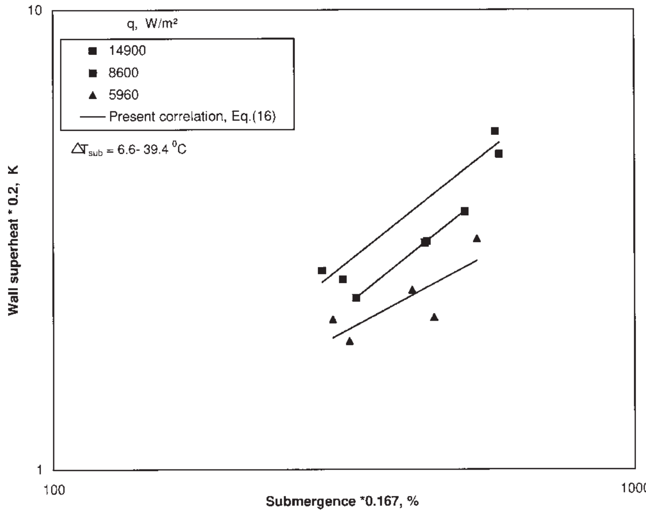


Figure 9. Heat flux versus degree of superheat at boiling incipience for ethyl acetate.

ural circulation reboiler. Hence, after incorporating the effect of submergence as suggested by Kamil et al.,[23] the above equation was modified as:

$$T_w - T_s = \psi_1 \left[\frac{8\sigma T_s q}{k_L\, h_{fg}\, \rho_v}\right]^{1/2} S^n \tag{16}$$

Equation 16 is the general expression for boiling incipience in the turbulent boundary layer for a vertical thermosiphon reboiler. The above equation involves only the superheat that is easy to measure directly, and the right-hand side as a whole can be evaluated with reasonable accuracy from the measurable quantities.

Results and Discussion

Wall and liquid temperature profiles along the heated test section

Figure 2 shows the variation for wall temperature profiles along the test section with heat flux as a parameter for methanol using data from the literature.[19] The typical behavior, as observed in the above figure, remains the same for other test liquids. However, the values of wall temperature, location of peak values, and the lengths of various zones are different. The variation of liquid temperature along the tube length has been shown corresponding to the lowermost heat flux. The liquid temperature increases linearly with distance along the

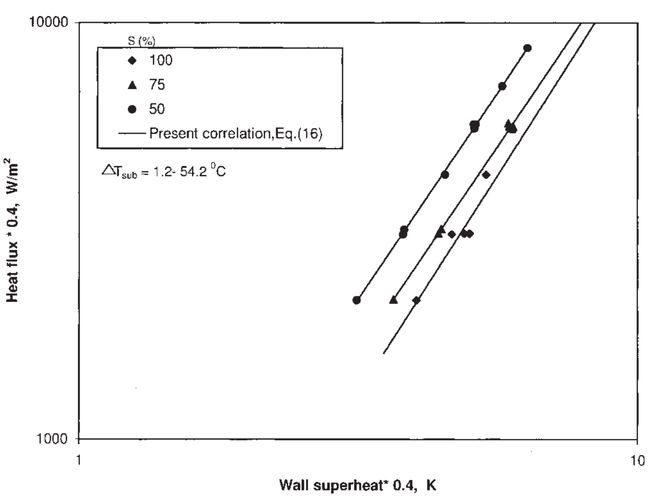


Figure 8. Heat flux versus degree of superheat at boiling incipience for propanol.

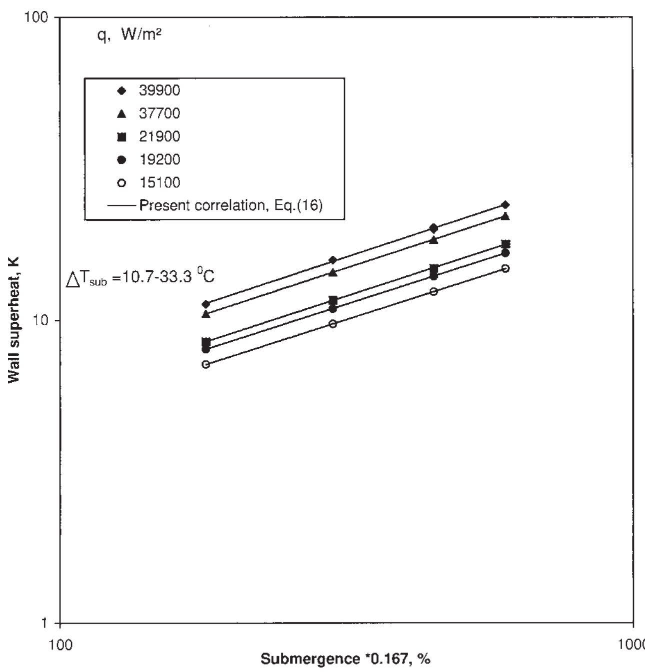


Figure 10. Degree of superheat versus submergence for water.

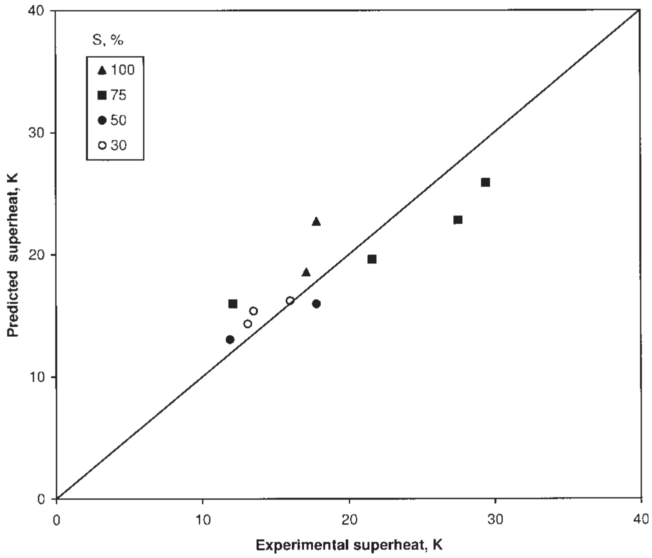


Figure 11. Comparison between experimental and predicted values of superheat for methanol.

tube length until it attains the saturation value, which itself decreases linearly as the liquid moves upwards due to the reduction of the hydrostatic head.

The plots of wall temperature versus tube length with liquid submergence as a parameter have been shown in Figure 3. The location of wall temperature peaks gets 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 values. 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 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 a 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 (Figures 2 and 3). 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 the 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 the saturated boiling regime as exhibited by the slowly decreasing wall and liquid temperature profiles. Further details are given elsewhere by Kamil[19] and Kamil et al.[20,21]

Boiling incipience in a vertical thermosiphon reboiler

From the wall and liquid temperature distributions as discussed earlier, it is observed that there exists a point at which the bubbles start appearing at the surface, though the liquid is still below its saturation value. This may be the onset of subcooled/surface boiling, and its effect is exhibited in deviation of wall temperature curves from straight-line behavior, characteristic of single-phase convection. In fact, the nucleation of bubbles must have started on attainment of the required minimum superheat even before the point mentioned above has been reached. As the liquid moves upward, its temperature rises and the boiling process becomes increasingly effective with additional turbulence at the wall. The wall temperature increases with diminished rate, which eventually becomes zero, showing a maximum wall superheat followed by a severe fall in its value. This is observed to happen when the liquid temperature has attained its saturation value, enabling bubble growth to the maximum size. This marks the onset of saturated boiling (OB). The effect of heat flux and liquid submergence on the general nature of wall temperature distribution and onset of boiling is essentially similar for all the test liquids. However, the maximum values of wall superheat and locations of boiling incipience for different systems, even under identical conditions, were not the same. The superheat for boiling incipience was calculated corresponding to the maximum values of wall temperatures attained for a particular run.

Figure 4 shows the heat flux versus wall superheat plot for benzene at a submergence of 100%. The scatter of data points is mainly due to highly unstable points of maxima followed by a steep fall in the wall temperature from which the values of $(T_w - T_s)$ are taken. In the same plot, the experimental data of Backhurst[26] are also shown. Figure 5 shows the heat flux versus wall superheat plot for water at a submergence of 100%. In the above figure, data of Sato and Matsumura,[4] Sudo et al.,[9] and Mantzouranis[27] for water have been plotted. Similar variation has been shown in Figure 6 for ethylene glycol. For these systems at 100% submergence, around 97% of the data points lie within $\pm 18\%$ of the correlation line, as shown in Figures 4 and 5, respectively. These figures clearly indicate that natural convection data and forced convection incipient boiling data are not in agreement.

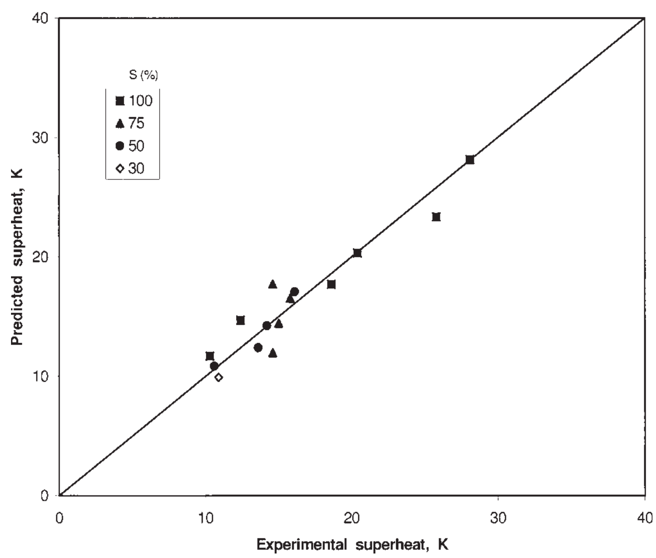


Figure 12. Comparison between experimental and predicted values of superheat for ethanol.

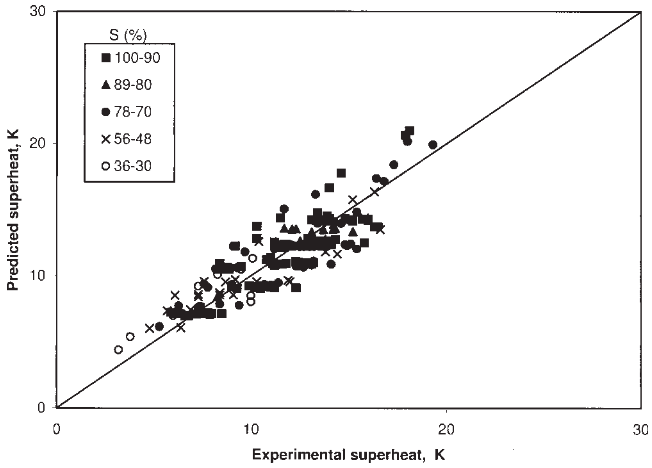


Figure 13. Comparison between experimental and predicted values of superheat for water.

The incipient boiling superheat was also calculated from the correlations of other investigators, and the same is plotted in the above figures. An almost similar trend was observed for other test liquids. At a particular submergence, as the heat flux increases, the wall superheat also increases. At low heat flux, the submergence has less influence on incipient boiling in comparison to high heat flux. Thus, from the above, it is evident that none of the correlations predict the data well, and generally most of the correlations under predict the superheat values for different systems. But the correlations of Sudo et al.[9] and Hino and Ueda[10] for $r_{tan} > r_{max}$ criteria over predicts the superheat value for water and ethylene glycol at different submergences.

Figures 7 and 8 show the plots of heat flux versus wall superheat with submergence as a parameter for acetone and propanol, respectively. These figures also show that at different submergences there are different predicted lines. From the plots, it is clear that the superheat increases almost 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 the above figures. It is, therefore, clear that submergence strongly influences the condition of the onset of nucleate boiling. Figures 9 and 10 show the plots of degree of superheat versus submergence with heat flux as a parameter for ethyl acetate and water, respectively. These figures also show that at different heat fluxes there are different predicted lines. In the above figures, the predicted results agree well with experimental data. Figures 11–14 show the comparison of the experimental wall superheat with predicted superheat (Eq. 16) for methanol, ethanol, water, and toluene, respectively. The majority of data points lie within considerable error limits. The values of exponent "n" and Ψ_t in Eq. 16 with maximum error are given in Table 3 for all the systems.

An effort was also made to obtain a unified correlation for the data of all systems with widely varying thermophysical properties. The correlation for the incipient boiling superheat was obtained by regression as:

$$(T_w - T_s) = (1.1043)\left[\frac{8\sigma T_s q}{k_L\ \rho_v\ h_{fg}}\right]^{1/2} (S)^{0.59867} \tag{17}$$

Figure 15 shows the plot of the comparison of the experimental wall superheat data with those predicted by the proposed correlation (Eq. 17). It was observed that the majority of data points lie within the maximum error of ±18 % and an average absolute relative error of 15.42%.

Experimental uncertainty

In the present study, the measured variables are the wall temperatures, liquid temperatures, and electrical input to the test section. The measurements involved include voltage, current, temperature, and tube dimensions. The measured values are subjected to some uncertainties due to the error of measurement. Taking into account the least count and accuracy of each instrument employed, uncertainty analysis has been carried out using the method suggested by Schultz and Cole.[28] The resolution in the measurement of temperature was 0.1°C over the temperature encountered; the average uncertainty in the measurement of temperature is 0.2 %. Hence, the uncertainty in estimating wall superheat is ±0.4%.

Conclusions

Based on the theoretical analysis, a new semi empirical model has been proposed to predict the wall superheat

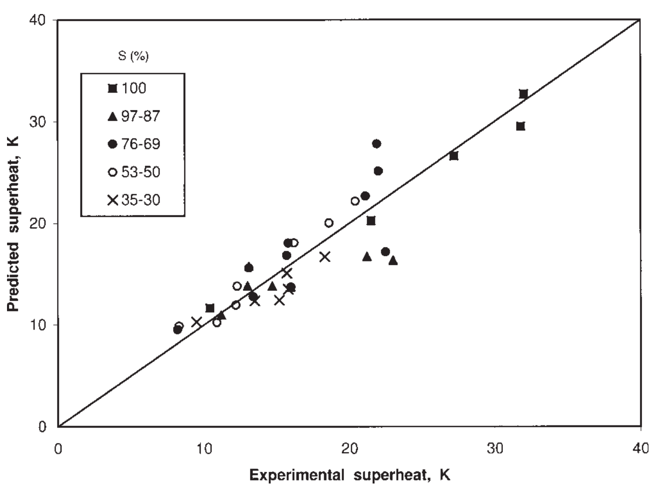


Figure 14. Comparison between experimental and predicted values of superheat for toluene.

Table 3. Values of Exponent, Turbulent Boundary Layer Profile Multiplier, and Maximum Percent Errors for Different Systems

Systems	Exponent "n" Eq. 16	Maximum Error (%)	Ψ_t
Acetone	0.78265	±18	0.9498
Methanol	0.50659	±16	3.0798
Ethyl acetate	1.0053	±12	0.2125
Ethanol	0.70808	±11	1.0078
Benzene	0.52619	±17	1.5517
Propanol	0.45699	±14	2.2095
Water	0.60331	±15	1.0308
Toluene	0.56017	±14	1.1889
Ethylene glycol	0.62556	±18	0.7771

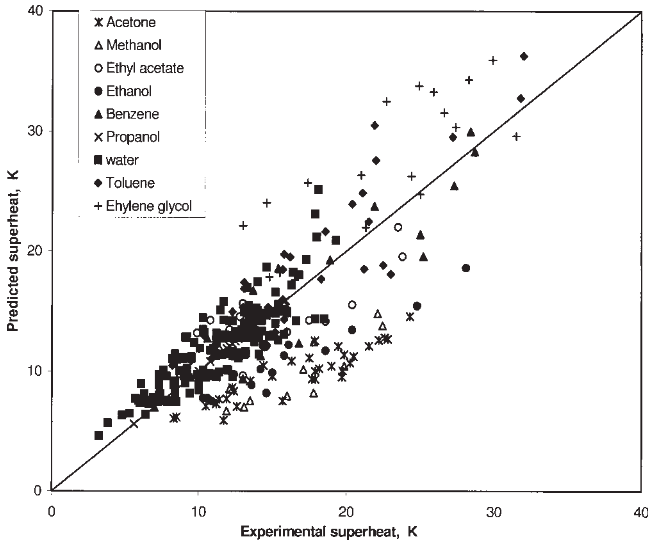


Figure 15. Comparison of experimental with predicted degree of superheat by proposed correlation (Eq. 17) for all test liquids.

required for onset of boiling in a vertical thermosiphon reboiler. A correlation for the minimum degree of wall superheat for a given liquid and heat transfer surface was developed in terms of the thermophysical properties of the liquid, liquid submergence, and turbulent eddies. At constant submergence, as the heat flux increases, the superheat required for incipient boiling increases. The superheat also increases with submergence for a constant heat flux. Thus, incipience is strongly influenced by turbulent eddies and submergence, which together suppress nucleation from surface cavities, requiring higher wall superheat for incipience compared to other forced convection systems.

Notation

C_P = heat capacity, J/kg K
h_{fg} = latent heat of vaporization, J/kg
k = thermal conductivity, W/m K
n = exponent used in Eq. 16
Pr = Prandtl number
Pr_t = turbulent Prandtl number
q = heat flux, W/m^2
r = radius, m
R = gas constant, Nm/kg K
$r_{\max}$ = maximum cavity radius, m
$r_{\tan}$ = cavity radius based on the tangency criterion, m
S = submergence, percent (%)
T = temperature, °C (K)
ΔT_s = degree of superheat ($T_w - T_s$), K
$\Delta T_{\rm sub}$ = degree of subcooling ($T_s - T_L$), K
y = distance perpendicular to the heated wall, m
Z = distance along the test section, m

Greek letters

δ^* = superheated layer thickness, m
ρ = density, kg/m^3
σ = surface tension, N/m
ε_H = eddy heat diffusivity, m^2/s
ε_m = eddy momentum diffusivity, m^2/s
Ψ_t = turbulent boundary layer profile multiplier
α = thermal diffusivity of liquid ($k_L/\rho C_P$), m^2/s)
μ = dynamic viscosity, kg/m s
ν = kinematic viscosity, m^2/s

Subscripts

B = boiling
b = bubble
c = cavity, critical condition
L = liquid
OB = onset of boiling
s = saturation
sub = subcooling
v = vapor
w = wall

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