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Conditions of Onset of Boiling in a Vertical Thermosiphon Reboiler

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An analysis of the incipience of nucleate boiling for a vertical thermosiphon reboiler was developed as a modification and extension of earlier analysis. The maximum superheat attained around the onset of boiling was taken from the wall-temperature distributions and correlated with heat flux, submergence, and physical properties of the test liquids. The results predicted from theoretical analysis are consistent with the experimental data that are available in the literature. All of the data for the eight fluids tested, each having different thermophysical properties, were correlated with a single correlation having an average absolute relative error of 21.35%.

Nomenclature

$a1, b1$	=	constant used in equation of Agrawal[13]
$a2, b2$	=	constant used in equation of Ali and Alam[15]
$a3, b3$	=	constant used in equation of Kamil et al.[17]
C_1	=	constant used in equation of Davis and Anderson[5] ($C_1 = 1$, $\theta = 90$ deg)
C_P	=	heat capacity, kJ/kg K
h_{fg}	=	latent heat of vaporization, J/(kgK)
k	=	thermal conductivity, W/m K
N	=	number of experimental data
P	=	pressure, psia
Pr_L	=	Prandtl number of fluid
q	=	heat flux, W/m^2
R	=	gas constant, Nm/kg K
r	=	radius, m
r_{max}	=	maximum cavity radius, m
r_{tan}	=	cavity radius based on the tangency criterion, m
S	=	submergence, %
T	=	temperature, °C
ΔT_s	=	degree of superheat ($T_w - T_s$), °C
ΔT_{sub}	=	degree of subcooling ($T_s - T_L$), °C
y	=	distance perpendicular to the heated wall, m
Z	=	distance along the test section, m
δ^*	=	superheated layer thickness, m
μ	=	viscosity, Ns/m^2
ρ	=	density, kg/m^3
σ	=	surface tension, N/m
ψ	=	exponent used in Eq. (11)

Subscripts

avg	=	average
B	=	boiling
b	=	bubble
c	=	cavity, critical condition
exp	=	experimental
L	=	liquid
OB	=	onset of boiling
OSB	=	onset of surface boiling
pred	=	predicted
s	=	saturation
sub	=	subcooling
v	=	vapor
w	=	wall

I. Introduction

VERTICAL-TUBE closed-loop thermosiphon reboilers, as used in petroleum, chemical, petrochemical, and nuclear power plants, are usually the most cost-effective systems. The heat-transfer coefficients at the onset of boiling are very high because of the nature of nucleate boiling and from a single-phase fluid to a two-phase mixture. The point at which the two-phase region begins is known as the incipient point of boiling, which 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. Therefore, information on the conditions required for the onset of nucleate boiling would be useful in the design of two-phase heat-transfer equipment. The operating parameters influence nucleation along the tube length and required superheat. Although the number of studies addressing boiling incipience in natural-circulation loops is limited, incipience in forced convection systems has been studied extensively during the last few years.

A number of researchers[1–8] predicted incipience based on the point of tangency between the liquid-temperature profile in the vicinity of the heated surface and the superheated-temperature profile required for mechanical equlibrium of a vapor bubble growing in a surface cavity. The validity of this criterion was established in many practical applications. However, a few investigators[9,10] predicted incipience based on maximum cavity radius available for nucleation on the heated surface. Han and Griffith[11] proposed an analysis similar to that of Hsu[2] for nucleate pool boiling. Marsh and Mudawar[12] performed an experimental study to develop a fundamental understanding of boiling incipience in wavy free-falling-turbulent-liquid films. Agarwal,[13] Ali,[14] Ali and Alam,[15] Kamil,[16] and Kamil et al.[17,18] experimentally obtained the boiling and non-boiling zones for a heated surface and superheat for incipient boiling in a vertical-tube thermosiphon reboiler over a wide range of submergence. Shamsuzzoha et al.[19–21] have developed a generalized correlation for prediction of superheat including the effect of submergence for different fluids. An analytical model has been formulated by Shim et al.[22] for fully developed turbulent flow and heat transfer in finned annuli using a modified mixing-length turbulence model. The model has been extended to predict the conditions at the

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Table 1 Summary of boiling incipience investigations

Authors	Flow geometry	Heater material	Fluid	Mean velocity, ms^{-1}	Pressure, bar (psia)	Subcooling, °C	Incipience formula
Sato and Matsumura[4]	Vertical channel	Stainless steel	Water	0.6–4.1	1.0 (14.7)	3–70	$q = \frac{k_L \rho_v h_{fg}}{8\sigma T_s}(T_w - T_s)^2$ Tangency criterion, $r = k_L(T_w - T_s)/2q$
Bergles and Rohsenow[3]	Horizontal annulus	Stainless steel	Water	3.3–17.4	Up to 2.6 (38.0)	32–90	$q = 15.60 P^{1.156}(T_w - T_s)^{2.30/P^{0.023}}$ (P in psia) Graphical solution for water over a pressure range of 15–2000 psia, based on tangency criterion
Han and Griffith[11]	Pool boiling on horizontal surface	Gold finished with 600 grit emery paper	Water	——	1.0 (14.7)	7.0	$q = \frac{k_L \rho_v h_{fg}}{12\sigma T_s}(T_w - T_s)^2$ Tangency criterion, $y = 1.5r$, at the point of tangency
Davis and Anderson[5]	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[6]	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 Abdelmessih[8]	Vertical tube	Stainless steel	Freon-11	0.08–0.4	Up to 2 (30.0)	1.20	For increasing heat flux: $q = \frac{1}{[7 - q/6500]} 2 \frac{k_L \rho_v h_{fg}}{2\sigma T_s}(T_w - T_s)^2$ 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.
Agrawal[13]	Vertical tube	Stainless steel grit	Water, acetone, ethyl acetate, propanol, toluene	——	Atmospheric pressure	0.9–73.0	$(T_w - T_s)^2 = (a_1 - b_1 q)^2 \frac{2\sigma T_s}{k_L h_{fg} \rho_v} q$
Hino and Ueda[10]	Vertical annulus	Stainless steel finished with 4/0 emery cloth	Freon-113	0.1–1.0	1.47 (22.0)	10–30	For $r_{tan} < r_{max}$: $q = \frac{k_L \rho_v h_{fg}}{8\sigma T_s}(T_w - T_s)^2$ For $r_{tan} > r_{max}$: $q = \frac{k_L}{r_{max}}(T_w - T_s) - \frac{2\sigma T_s k_L}{\rho_v h_{fg} r_{max}^2}$
Sudo et al.[9]	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[12]	Falling film over vertical cylinder	Stainless steel with 600 grit paper	Water FC-72	1.29–1.79 1.03–1.65	0.7, 1.05	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[15]	Vertical tube	Stainless steel tube	Water, acetone, ethanol, ethylene glycol	——	Atmospheric pressure	0.2–45.5	$(T_w - T_s)^2 = (a_2 - b_2 q)^2 \frac{2\sigma T_s}{k_L h_{fg} \rho_v} q$
Kamil, Ali, and Alam[17]	Vertical tube	Stainless steel tube	Water, methanol, benzene, toluene, ethylene glycol	——	Atmospheric pressure	0.5–11.6	$(T_w - T_s)^2 = (a_3 - b_3 q)^2 \frac{2\sigma T_s}{k_L h_{fg} \rho_v} q$
Shamsuzzoha, Kamil, and Alam (present study)	Authors performed analysis using experimental data from prior studies[13,14,16]						$(T_w - T_s) = Pr_L \left[\frac{8\sigma T_s q}{k_L \rho_v h_{fg}}\right]^{\frac{1}{2}} (S)^{0.25253}$

onset of nucleate boiling using the criterion of Davis and Anderson[5]; these predictions agreed well in magnitude and trend with experimental data. Hapke et al.[23] investigated the onset of nucleate boiling (ONB) and the heat-transfer characteristics during flow boiling in a minichannel. They used the thermographic measuring method and measured the axial distribution of the external wall temperature. A summary of previous incipience investigations is presented in Table 1.

Thus, it seems that the available literature does not include the effect of submergence in the prediction of the degree of superheat in natural-circulation systems. Therefore, a semiempirical model has been developed for predicting superheat, including the effect of submergence, using the experimental data available in the literature.[13,14,16] The proposed model has been compared with existing incipience equations.

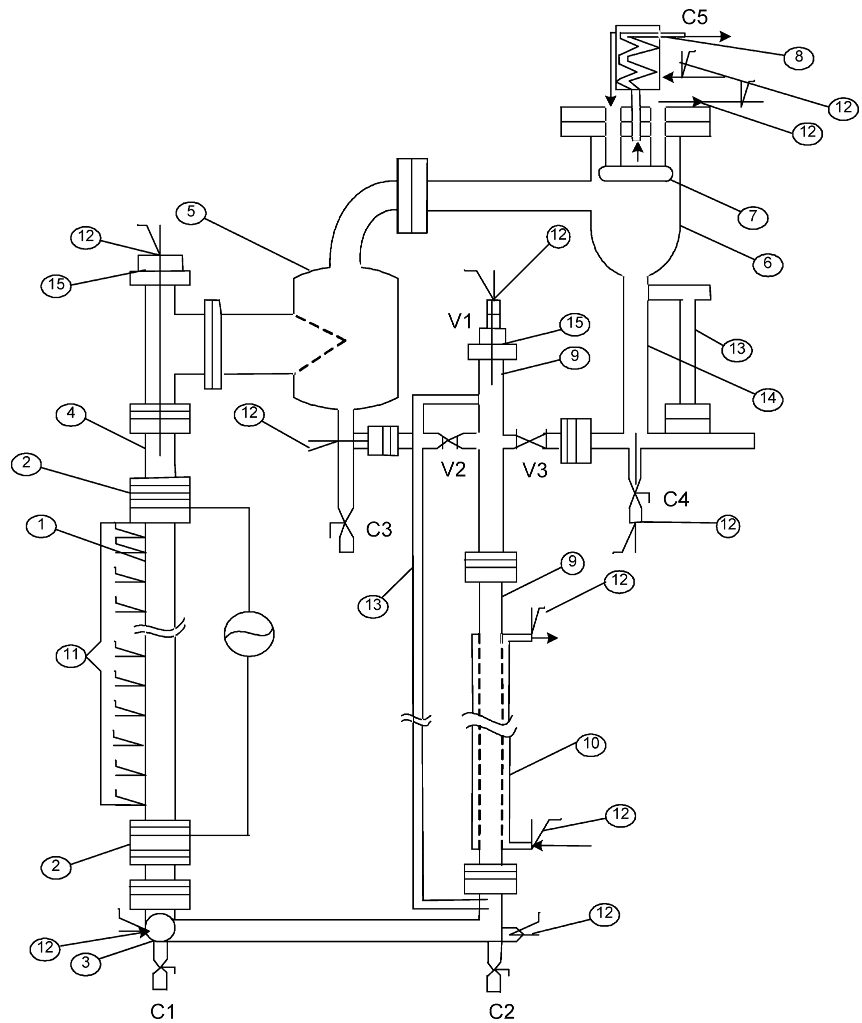

Fig. 1 Schematic diagram of experimental setup from Ali and Alam,[15] Kamil,[16] and Kamil et al[17,18]: 1, test section; 2, copper clamps; 3, viewport for inlet liquid; 4, glass tube section; 5, vapor–liquid separator; 6, primary condenser; 7, spiral coil; 8, secondary condenser; 9, liquid down-flow pipe; 10, cooling jacket; 11, wall thermocouples; 12, liquid-temperature probes; 13, liquid-level indicator; 14, condenser down-flow pipe; 15, removable screwed cap; V1–V3, control valves; and C1–C5, drain cock valves.

II. Experimentation

The experimental reboiler was made of two vertical tubes joined in a U shape with the upper ends connected to a vapor–liquid separator and total condenser vessels, forming a thermosiphon loop, as schematically shown in Fig. 1. One of the vertical tubes, which served as the test section, was electrically heated. The test liquid boiled in this tube, flowed upward through a glass section, and entered into the separator. The liquid drained down the bottom of the separator, while vapors went to a water-cooled condenser. The condensate joined the separator liquid near the top end of the other jacketed vertical tube (down-flow pipe) through which the total liquid circulated back to the test section through a viewing action. The test section was an electrically heated stainless steel tube of i.d. 25.56 mm, o.d. 28.85 mm, and length 1900 mm. The stabilized power was supplied through a low-voltage, high-current transformer. The energy input to the test section was measured by a calibrated precision-type voltmeter and ammeter. Twenty-one copper constantan thermocouples were spot-welded onto the outer surface of the tube at intervals of 50 mm up to a length of 200 mm from the bottom and 100 mm over the remaining length in order to monitor the heat-transfer surface temperatures. The test section was electrically isolated from the rest of the setup by specially designed flanges and an upper glass-tube section. The lower end of the flanges was connected to a viewport through which the liquid coming out of the down-flow pipe could be visually observed to ensure the complete absence of any air or vapor bubbles before its entry into the test section. A copper constantan thermocouple probe was provided in the viewport to measure the inlet liquid temperature. The visual observation of the boiling liquid emerging out of the test section was made through the upper glass tube section. Another thermocouple probe was inserted into the exit line leading to the vapor–liquid separator. Provisions were also made to measure the flow rates and temperatures in and around the condenser and at other strategic locations in the reboiler loop to ensure reliable computation of circulation rates through the heat balance. A glass tube level indicator was provided with the down-flow pipe to indicate the liquid head (submergence) for the reboiler. The entire setup was thoroughly lagged to reduce the heat losses to a negligibly small value. After the assembly and initial testing of the experimental setup, some forced-convection data were collected using water to check the heat balance and standardization of the apparatus. The heat-transfer surface was then stabilized for the reproducibility of the experimental data. This was done by boiling distilled water, under conditions of full submergence and nearly zero inlet degree of subcooling, for several hours, followed by aging, to obtain stable wall-nucleating characteristics. A few runs were also conducted to check the overall heat balance under conditions of boiling. Care was also taken that, once the tube wall was stabilized, it remained fully submerged, because the dry test surface always entraps a very thin film of air. This air, on heating, takes the shape of tiny air bubbles, which leave the surface on further heating. Thus, microconvection sets in near the heat-transfer surface, in addition to the convections due to density difference. The test liquid was also boiled off at the start of every experiment to drive out the dissolved air completely, which was indicated by the disappearance of the air bubbles in the bubbler. In an experimental run, the desired heat flux was impressed and the submergence adjusted by draining/adding the requisite amount of test liquid. The cooling-water rate was regulated to give a maximum temperature rise consistent with no loss of vapor due to inadequate condensation. With the layout designed for closed-system operation and with a stabilized power supply, the unit, once charged and started, could be run continuously for a sufficient time. When the steady-state condition was established, the readings of electrical input, wall thermocouples, liquid thermocouple probes, and cooling water were recorded. The maximum liquid head used in the present study corresponded to a liquid level equal to the top end of the reboiler tube. This condition has been termed 100% submergence. The cold liquid head could be varied independently by maintaining the submergence at 75, 50, and 30%. The experimental data were generated with increasing heat flux and 1.0-atm pressure. The detailed description of the experimental setup has been discussed earlier.[13–18]

III. Mathematical Analysis

The assumptions included in the present model are as follows:

1) 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.

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

3) 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.

4) The equation for the superheated vapor temperature profile T_b can be obtained from the Clausius–Clapeyron relation, which is also used by other investigators[1,2,4,12,19]:

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

The three different possible bubble nucleus shapes that can exist at the mouth of a cavity are shown by Davis and Anderson[5] and Hsu.[2] These nuclei are assumed to be formed by the residual vapor from the preceding bubble. Bergles and Rohsenow[3] considered the case of a hemispherical bubble shape, giving an argument that, once a bubble passed the hemispherical condition of minimum radius, it would continue to grow. Our discussion is concerned with a truncated spherical bubble. It reduces to the hemispherical bubble when the bubble contact angle is 90 deg, as shown by Davis and Anderson.[5] Only the cavities in a narrow size range are involved at the onset of nucleate boiling, and the population density of bubbles on the surface just prior to the onset of nucleate boiling are 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 sublayer and the thermal conductivity of liquid is also constant. The heat transport occurs by conduction through the liquid only within the laminar sublayer:

$$T_L = T_w - qy/k_L \tag{2}$$

or

$$\frac{\mathrm{d}T_L}{\mathrm{d}y} = -\frac{q}{k_L} \tag{3}$$

Taking the first derivative of Eq. (1) gives

$$\frac{\mathrm{d}T_b}{\mathrm{d}r_b} = -\frac{2\sigma T_s}{h_{\mathrm{fg}}\rho_v r_b^2} \tag{4}$$

One useful criterion for the initiation of nucleate boiling was proposed by Bergles and Rohsenow.[3] In their analysis of flow-surface boiling, these authors suggested that, at a distance $y = r_b$, $T_L = T_b$ (where $T_b > T_s$) is the condition for a 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); that is,

$$\frac{\mathrm{d}T_b}{\mathrm{d}r_b} = \frac{\mathrm{d}T_L}{\mathrm{d}y} \tag{5}$$

Substituting Eqs. (3) and (4) into Eq. (5) and solving for r_b at the incipient point, we get

$$r_b = \sqrt{\frac{2\sigma T_s k_L}{q h_{\mathrm{fg}}\rho_v}} \tag{6}$$

At the incipient point, in addition to their slopes being equal, the temperatures T_b and T_L must also be equal at $y = r_b$ $(r_b = r_{\tan})$. Equating Eqs. (1) and (2),

$$T_w - \frac{qy}{k_L} = T_s + \frac{2\sigma T_s}{h_{\mathrm{fg}}\rho_v r_b} \tag{7}$$

or

$$(T_w - T_s) = \frac{2\sigma T_s}{h_{\mathrm{fg}}\rho_v r_b} + \frac{q r_b}{k_L} \tag{8}$$

Substituting the value of r_b from Eq. (6) into Eq. (8) and solving for $(T_W - T_S)$, we get

$$(T_w - T_s) = \left[\frac{8\sigma T_s q}{k_L \rho_v h_{\mathrm{fg}}}\right]^{\frac{1}{2}} \tag{9}$$

Equation (9) is a general expression for flow boiling as suggested by earlier workers.[1,4,5,9,10] But Frost and Dzakowic[6] have extended this treatment to cover other liquids. In their study, nucleation was assumed to occur when the liquid temperature, T_L, was matched to the temperature for bubble equilibrium, T_b, at a distance nr_c, where $n = Pr_L^2$, rather than r_c, as assumed in earlier work.[1,4,5,9,10] But no justification for this assumption was presented by Frost and Dzakowic.[6] They proposed the equation

$$(T_w - T_s) = Pr_L \left[\frac{8\sigma T_s q}{k_L \rho_v h_{\mathrm{fg}}}\right]^{\frac{1}{2}} \tag{10}$$

with the assumption that nucleation occurs at a surface when the liquid temperature T_L just exceeds the vapor bubble temperature T_b at a distance $y = Pr_L^2 r_{\tan}$. The above equation has been fairly successful in correlating incipience data for a large number of fluids.

Perhaps most of the studies pertaining to boiling incipience are for flow boiling, whereas the present analysis is for a natural-circulation thermosiphon reboiler. Some workers[13–21] have investigated the effect of inlet liquid subcooling and submergence on heat transfer, circulation rate, and boiling incipience in a vertical thermosiphon reboiler. In the 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 the liquid level corresponding to the top of the test section, termed 100% 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, whereas an increase in the heat flux shifts the point of boiling incipience toward the tube inlet. As submergence is lowered, the liquid head decreases, whereas the vapor fraction increases due to the enhanced effect of saturated boiling in the tube. However, the differential head that causes circulation becomes smaller than that at higher values of submergence. A detailed description of the effect of submergence on induced flow has been discussed by Kamil et al.[24] Thus from the above, it is clear that submergence has an important effect on boiling incipience in the case of a natural-circulation reboiler. Yin and Abdelmessih[8] have also investigated the effect of velocity on δ^*/r_c. Therefore, it is important to include the effect of submergence in the prediction of the degree of superheat. Thus after the effect of submergence was incorporated, Eq. (10) was modified as

$$(T_w - T_s) = Pr_L \left[\frac{8\sigma T_s q}{k_L \rho_v h_{\mathrm{fg}}}\right]^{\frac{1}{2}} (S)^{\psi} \tag{11}$$

Equation (11) is a general expression for incipient boiling, and several interesting observations can be made from this equation, involving only the superheat, which is easy to measure directly. The right-hand side as a whole can be evaluated with reasonable accuracy from the measurable quantities.

IV. Results and Discussion

A. Error Analysis

The wall superheat was calculated from the measured values of electrical input to the test section and various temperatures measured at approximate locations with the help of thermocouples. The measurements involved include voltage, input current, temperature, and tube dimensions. The measured values are subject 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 Moffat[25] and was found to be 2.10%.

B. Wall-Temperature Profiles Along the Heated Test Section

The boiling process showing its effective influence on the wall-temperature profile under certain unique conditions of wall superheat and heat flux for a given liquid only. It was also clear that there exists a point at which the bubbles start appearing at the surface, though the liquid temperature 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, which is characteristic of single-phase convection, as illustrated in Fig. 2. These profiles were made using the experimental data of Kamil.[16] The wall superheat required for incipient boiling in a thermosiphon reboiler is much higher than that for pool boiling at the same heat flux and degree of subcooling. This may be attributed to the dominant role of convective heat transfer, which suppresses the nucleate boiling. In fact the nucleation of bubbles must have started on attainment of the required minimum superheat even before the point mentioned above had 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. The position of incipient boiling shifts upstream with increasing heat flux and the same trend is also seen with increasing submergence. Incipient boiling at a given location is readily recognized either by a sharp drop in wall temperature over the entire downstream section, as shown in Fig. 2, or by drastic increases in the heat-transfer coefficient. The latter can be determined from the change of slope on the plot of heat flux vs superheat. This is observed to happen when the liquid temperature has attained its saturation value, enabling the bubble growth to reach its maximum size. This is the onset of saturated boiling. The effects of heat flux and submergence on the general nature of wall-temperature profiles and the onset of boiling are essentially similar for the different fluids. However, the maximum values of wall superheat and locations of boiling incipience for different systems, even under identical conditions, need not be the same, as discussed in the literature.[13–18]

Figure 3 shows the variation of wall temperature for acetone with heat flux as a parameter along the test section. These profiles were made using the experimental data of Agarwal[13] to explain the phenomenon of boiling incipience. The behavior, as observed in Fig. 3, is the same for other test liquids. The values of wall temperature, the locations of peak values, and the lengths of various zones are different for different test liquids. Similar plots have been also shown by other researchers.[8,10]

Figure 4 shows a plot of wall temperature T_w vs tube length Z with liquid submergence as a parameter for methanol. The typical observed variation of wall temperature indicates that there exist different regimes of heat transfer in a reboiler tube. These profiles were

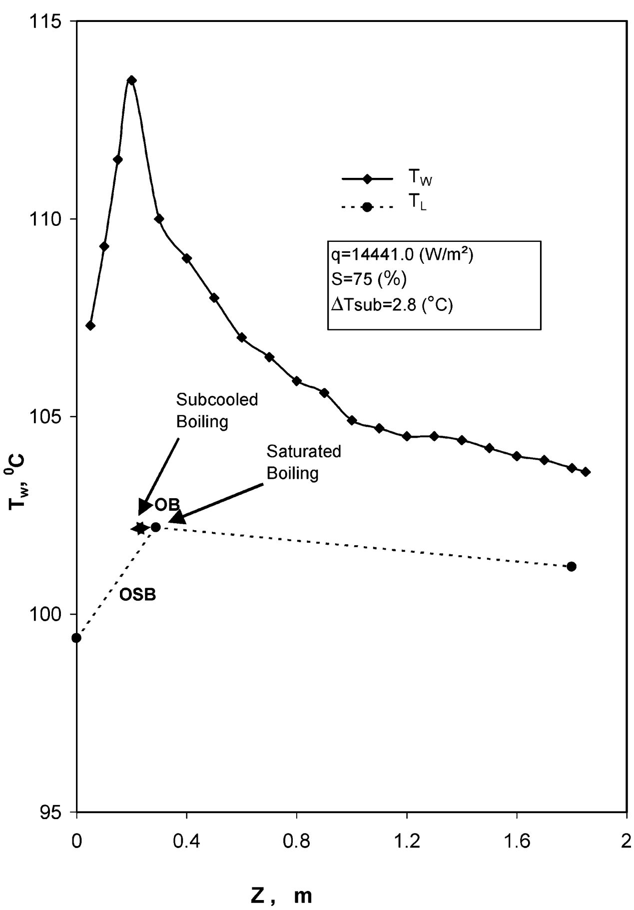

Fig. 2 Wall- and liquid-temperature profiles for water along the tube length, from Kamil.[16]

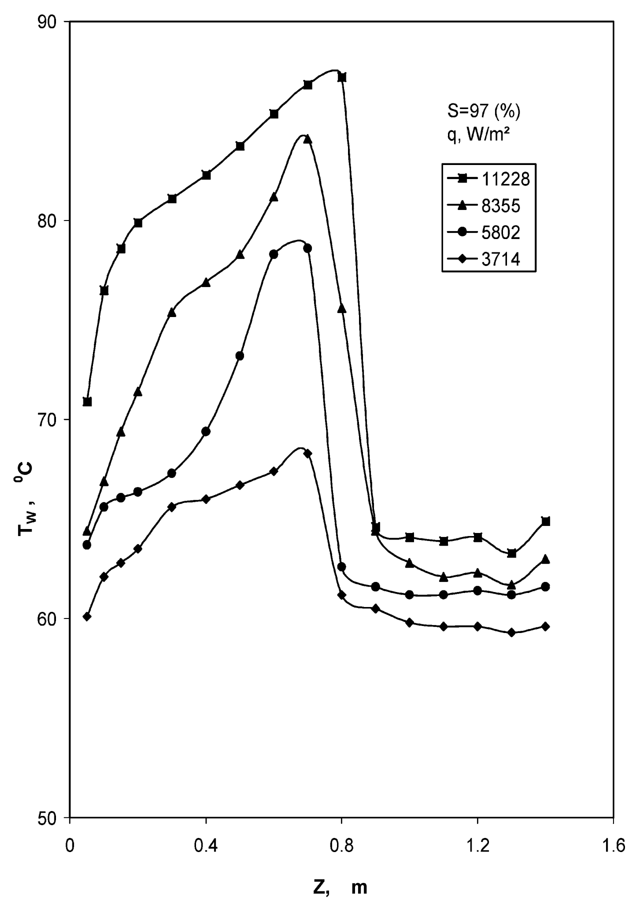

Fig. 3 Wall-temperature profiles for acetone along the test section with heat flux as parameter, from Agarwal.[13]

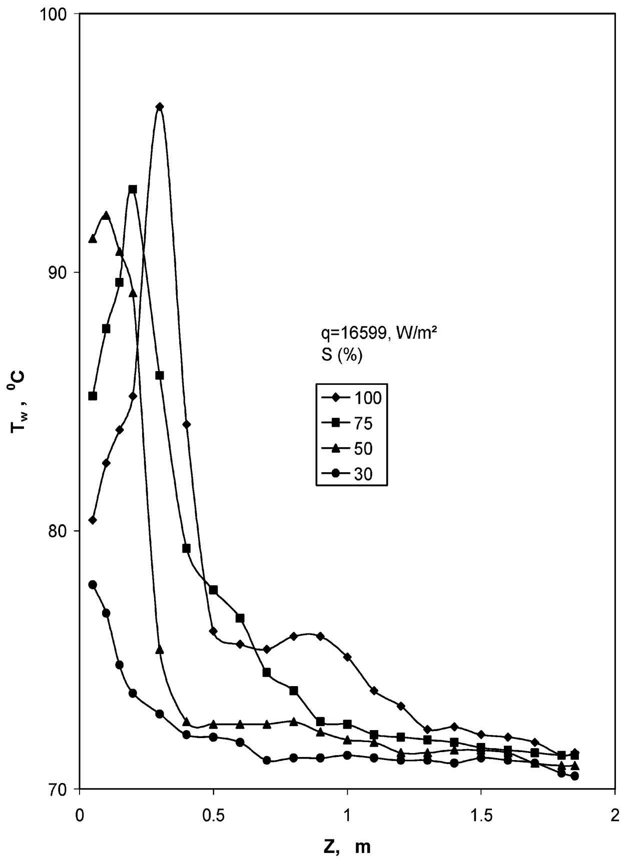

Fig. 4 Wall-temperature profiles for methanol along the test section with submergence as parameter, from Ali.[14]

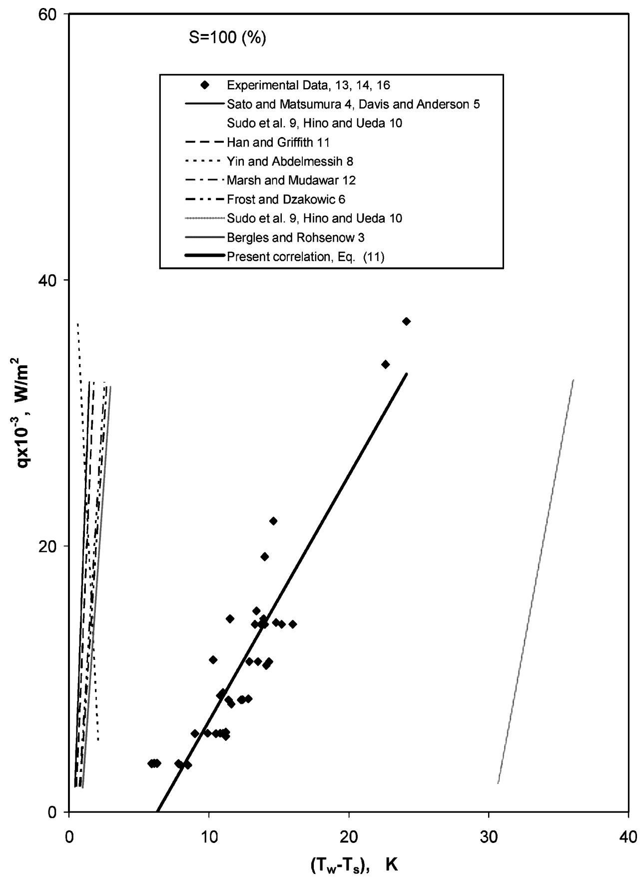

Fig. 5 Heat flux vs degree of superheat at boiling incipience for water.

made using the experimental data of Ali.[14] The linear rise in the temperature of the liquid as it moves upward through the tube results from sensible heating under uniform heat flux. When the minimum required wall superheat is attained, bubbles start nucleating at the surface, but collapse due to the presence of subcooled liquid cores. 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 a decreasing rate, eventually becoming zero at peak values in Figs. 3 and 4. Once the bulk-liquid temperature attains saturation value, the bubbles generated at the surface grow to their maximum size and detach, resulting in the existence of a vapor phase in the tube. All of the heat supplied gets absorbed as latent heat of vaporization, converting the liquid to vapor. Thus the two-phase flow goes upward through the tube with an increasing quantity of vapor and hence changing flow patterns. This corresponds to the saturated-boiling regime, as exhibited by the slowly decreasing wall-temperature profiles.

As the value of heat flux is raised, the wall temperature also increases to provide a temperature difference adequate for transferring additional heat. In the convective mode of heat transfer, this should be almost in the same ratio as that of heat-flux change. But in nucleate boiling, this is not so, because the increased heat flux enables a larger number of nuclei for bubble generation to become active, and thus enhances the heat-transfer coefficient and requires a small temperature difference. This explains the shifting of the wall-temperature curves with heat flux as observed in Figs. 3 and 4. The shifting of the saturated boiling to a lower level in the tubes as the submergence is reduced from 100% is probably due to the change in circulation rates. The decrease in the value of liquid submergence reduces the driving force for liquid circulation and hence the flow rate of the liquid through the reboiler tube. At a lower rate of liquid circulation, the rate of change of temperature with tube length becomes higher, and the saturation temperature is attained at a much smaller distance from the inlet.

C. Boiling Incipience

The mathematical analysis for the prediction of boiling incipience established in Sec. III, is based on Gibbs's equilibrium theory, which is widely used in the literature.[1–8] In the present study, it was assumed that nucleation occurs when the liquid temperature, T_L, matches the temperature for bubble equilibrium, T_b, at a distance nr_c, where $n = Pr_L^2$, as suggested by Frost and Dzakowic.[6]

Figure 5 shows the heat flux vs superheat curve for water at a submergence of 100%. Almost 94% of the data points lie within ±16% of the correlation line. Figure 5 clearly shows that the predicted values are in good agreement with the experimental data. The ranges of parameters covered in developing and validating the correlation are given in Table 2 for all eight systems. The incipient boiling superheat was also calculated from other investigator correlations and is plotted in the same figure. A similar trend has been shown for other test liquids. From Fig. 5, at constant submergence, as the heat flux increases, the superheat also increases. For low heat flux, the submergence has less effect on incipient boiling than for high heat flux. For a fixed submergence, low superheat is required for the low heat flux, and high superheat is required for higher heat flux for incipient boiling. From Fig. 5 it is clear that none of the correlations predict the data well, and generally most of the correlations underpredict the superheat values for different systems. But the Sudo et al.[9] and Hino and Ueda[10] correlations for $r_{\tan} > r_{\max}$ overpredict the superheat value for the same fluids, namely water and ethylene glycol at different submergence values. A comparison of the proposed model with other investigations is shown in Table 3, giving the average absolute relative error (AARE) of the proposed model and other investigators' correlations. The values of the exponent ψ in Eq. (11) with maximum errors (%) are given in Table 2 for different systems.

Figure 6 shows the heat flux vs incipient-boiling superheat for acetone computed from Eq. (11). The effect due to submergence is evident. The predicted results agree well with experimental data at constant submergence. This figure also shows that at a fixed submergence, there are different predicted lines.

Table 2 Ranges of parameters and maximum percent error for different systems

Systems	Submergence, %	ΔT_{sub}, °C	Heat flux, W/m^2	Exponent ψ, Eq. (11)	Maximum error, %
Acetone	30–100	0.2–45.5	3,548–15,115	0.4971	±23
Methanol	30–100	1.0–3.7	4,105–21,305	0.3860	±18
Ethanol	30–100	1.1–21.6	3,800–21,884	0.1989	±20
Benzene	30–100	0.7–3.6	4,106–29,225	0.2690	±19
Propanol	39–97	1.2–54.2	3,342–21,765	0.0563	±15
Water	30–100	0.2–73.0	3,486–43,373	0.4891	±16
Toluene	30–100	1.9–68.3	2,042–32,085	0.2573	±18
Ethylene glycol	30–100	3.25–15.8	15,115–33,654	0.3619	±20

Table 3 Comparison of proposed model, Eq. (11), with other correlations

	Average absolute relative error (AARE)[a]							
System	Acetone	Methanol	Ethanol	Benzene	Propanol	Water	Toluene	Ethylene glycol
Sato and Matsumura[4]	95.93	95.49	94.97	91.16	93.32	92.60	91.5	90.37
Bergles and Rohsenow[3]	90.67	89.24	89.19	87.99	86.86	84.19	88.73	88.13
Han and Griffith[11]	95.02	94.48	93.84	90.40	91.82	90.94	89.59	88.21
Davis and Anderson[5]	95.93	95.49	94.97	91.16	93.32	92.60	91.50	90.37
Frost and Dzakowic[6]	86.59	77.03	54.00	63.36	22.69	87.05	63.71	76.43
Yin and Abdelmessih[8]	88.87	89.29	87.17	83.04	82.50	82.22	80.78	86.52
Hino and Ueda[10] and Sudo et al.,[9] $r_{tan} < r_{max}$	95.93	95.49	94.97	91.16	93.32	92.60	91.50	90.37
Hino and Ueda[10] and Sudo et al.,[9] $r_{tan} > r_{max}$	56.71	58.43	51.43	36.04	37.36	209.0	25.63	69.83
Marsh and Mudawar[12]	92.39	91.57	90.60	85.35	87.51	86.16	84.09	81.99
Author correlation, Eq. (11)	20.06	15.85	17.25	17.65	12.88	13.02	5.42	17.92

[a]Average absolute relative error: $\mathrm{AARE} = \left[\frac{1}{N}\sum_{i=1}^{N}\left|\frac{\Delta T_{\mathrm{pred}} - \Delta T_{\mathrm{exp}}}{\Delta T_{\mathrm{exp}}}\right| \times 100\right]$ where N is the number of experimental data points.

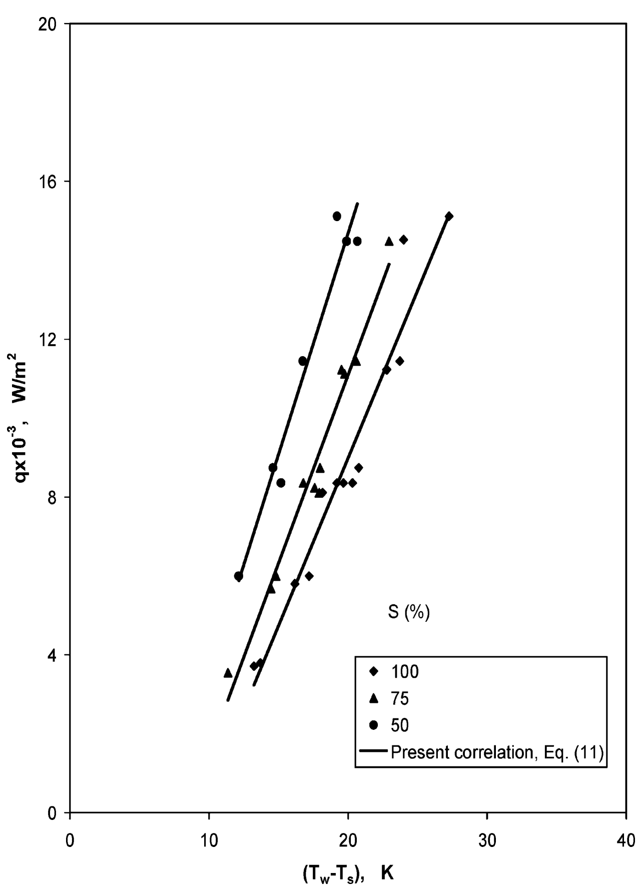

Fig. 6 Heat flux vs degree of superheat at boiling incipience for acetone.

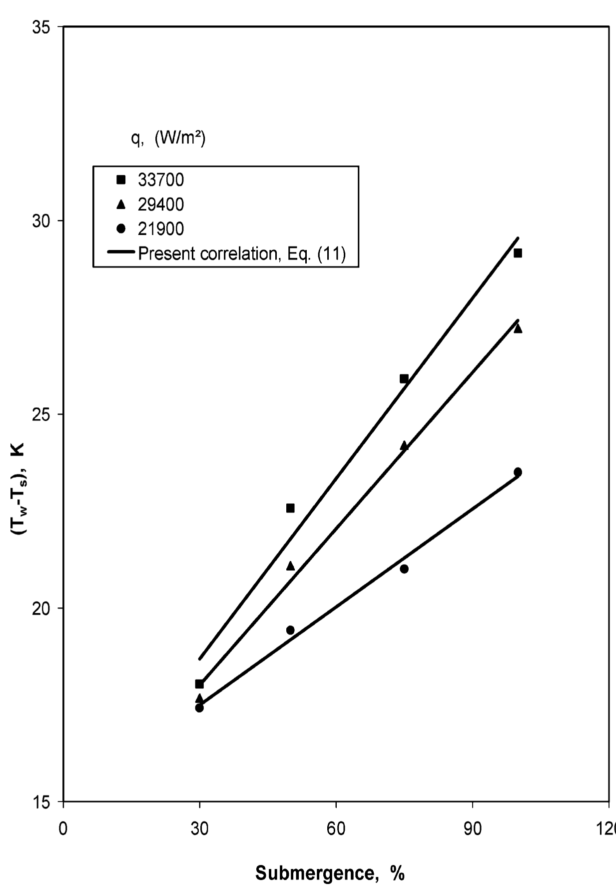

Fig. 7 Degree of superheat vs submergence for ethylene glycol.

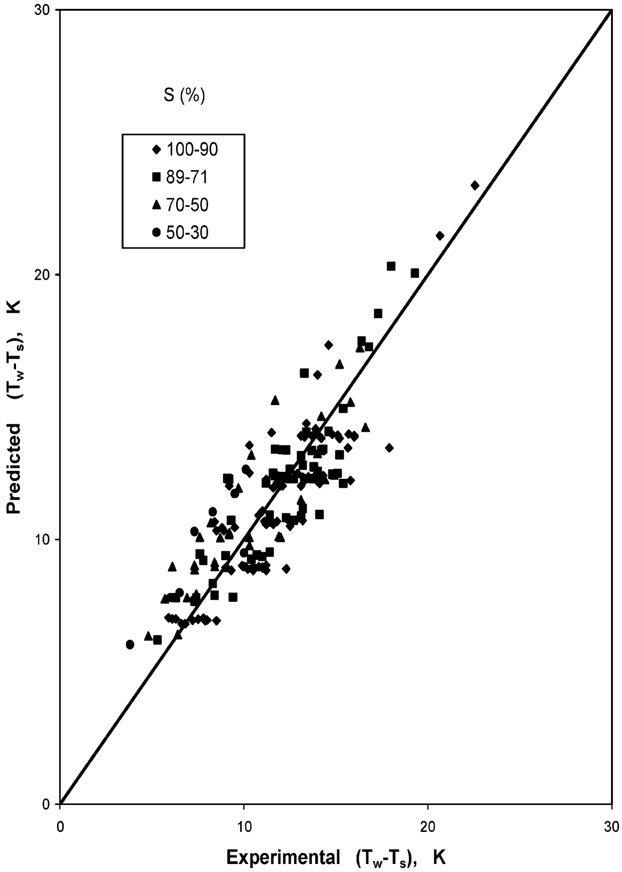

Fig. 8 Comparison between experimental and predicted values of superheat for water.

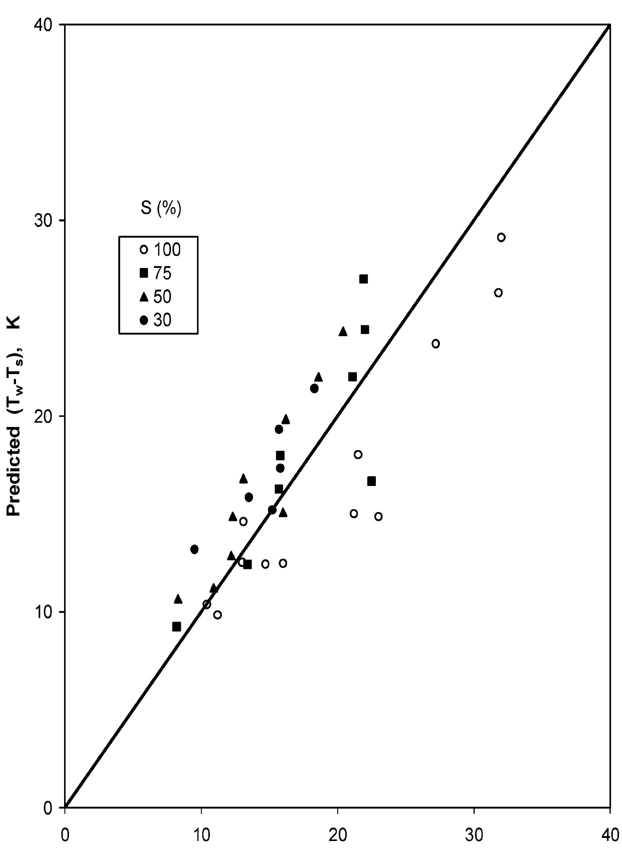

Fig. 9 Comparison between experimental and predicted values of superheat for toluene.

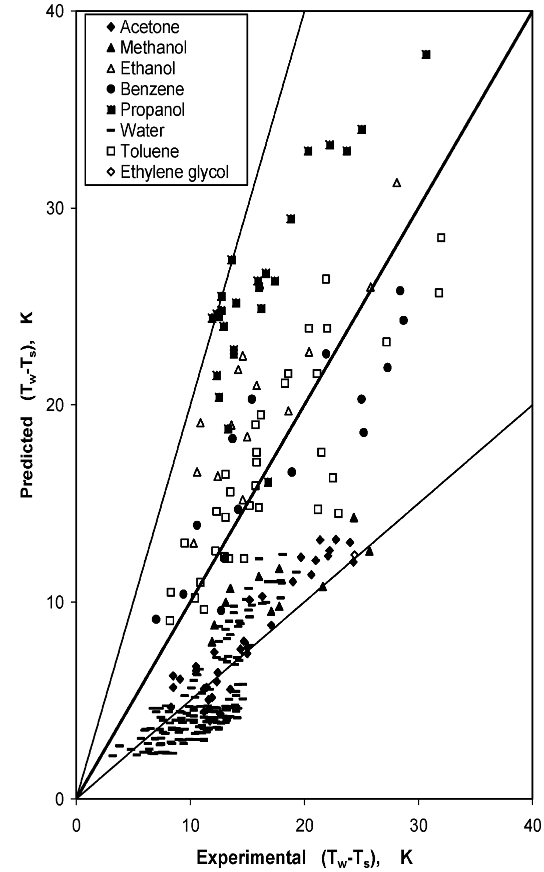

Fig. 10 Comparison of experimental and predicted degree of superheat by proposed correlation, Eq. (12), for all test liquids.

Figure 7 shows the plot of degree of superheat vs submergence for ethylene glycol. From the plot, it is clear that the superheat increases linearly with increased 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 figure. Therefore, it is clear that submergence has a strong effect on the conditions for the onset of nucleate boiling.

Figures 8 and 9 show the comparison of the experimental degree of superheat with predicted superheat by Eq. (11) for water and toluene. The majority of the data points lie within considerable error limits. The maximum errors for different systems are tabulated in Table 2.

An effort was also made to obtain a single correlation of all eight systems together, covering wide ranges of heat flux, submergence, and ΔT_{sub}. The correlation for incipient-boiling superheat was obtained as

$$(T_w - T_s) = Pr_L \left[\frac{8\sigma T_s q}{k_L \rho_v h_{\text{fg}}} \right]^{\frac{1}{2}} (S)^{0.25253} \qquad (12)$$

Figure 10 shows a plot of the comparison of experimental data with the predicted superheat by the proposed correlation covering all of the data for eight different fluids with widely varying properties. The ranges of parameters covered in developing and validating the correlation is given in Table 2. It is observed that 91% of the data points of the present study are within a maximum error of ±28% and an AARE of 21.35%.

V. Conclusions

The key conclusion from the present investigation is that at constant submergence, as the heat flux increases, the superheat required for incipient boiling increases. Based on theoretical analysis, Eq. (11) has been developed, which predicts the superheat required for the onset of boiling. The exponent ψ determined for different fluids and the corresponding maximum percent error are given in Table 2. The data may also be correlated by a single equation, Eq. (12), for all of the fluids, with a maximum error of $\pm 28\%$. Superheat increases linearly with submergence for a constant heat flux. At a low value of heat flux, the role of submergence has less effect than for high heat flux. The comparisons made from various correlations clearly show that submergence is an important parameter in the prediction of ONB in a vertical thermosiphon reboiler.

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