Study of local boiling phenomenon on different metallic surfaces in non-circular flow passages under laminar flow conditions

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Received 15 April 2006; revised received 1 November 2006; accepted 15 November 2006

The thermal diffusivities of the metals are found to be an influencing parameter in explaining the different threshold temperatures required for the local boiling phenomenon to occur on different metallic surfaces and the Fourier number is used in the correlation to account for the thermal diffusivities. The effect of the degree of sub-cooling on local boiling is also quantified by modifying the Jakob number by replacing the term giving the difference between the temperature of the boiling liquid film and saturation temperature of the liquid, with the ratio of excess temperature to the degree of sub-cooling with an index of 0.2. The proposed final correlation comprising of local boiling coefficient, non-boiling coefficient, Fourier number and modified Jakob number is

\[ K_{lb} = 0.11(K_{nb})_{0.25}(Fo)_{0.25}(Ja_{mod})_{2.5}. \]

Keywords: Local boiling coefficient, Non-boiling coefficient, Surface temperature, Laminar flow

IPC Code: G01F1/684

The local boiling phenomenon was first observed in the jackets of all liquid cooled aircraft engines and rocket motors and has received considerable attention as far back as in 1950’s owing to the development of atomic reactors for power generation. Since then several workers studied the phenomenon of local boiling and different correlations were proposed. Venkateswarlu\textsuperscript{1} and Chiranjivi\textsuperscript{2} studied the local boiling phenomenon in square and rectangular flow passages with different metallic surfaces and coolants. Chiranjivi\textsuperscript{2} correlated his data considering that local boiling is superimposed on convective heat transfer and his data confirmed that different threshold values are required for different metal surfaces, the threshold value being defined as the limiting temperature excess below which no effect of local boiling is visible, which was originally observed by Colburn and Sutton\textsuperscript{3}. Westwater\textsuperscript{4} in his paper discusses various aspects of surface characteristics, bubble formation, growth and frequency and points out that the thermal properties of the solid may also be important. Leppert and Pitts\textsuperscript{5} showed the variation of heat flux during forced convection and with different liquid sub-cooling temperatures and it was clear that the increased sub-cooling requires larger temperature excess for the boiling to initiate, but the effect was not quantified. Ralph and Webb\textsuperscript{6} reviewed the heat transfer surface characteristics and their effect on boiling heat transfer coefficient. Bergez\textsuperscript{7} recorded simultaneously ebullition activity and wall temperatures measured with a thermochronic liquid crystal and concluded that the existing boiling heat transfer models are not applicable and he proposed a new expression for waiting time correlating with thermal diffusivity of the wall. Mukherjee and Dhir\textsuperscript{8} studied lateral merger of vapour bubbles during nucleate pool boiling and concluded that the bubble merger process increased the over all heat transfer by trapping a liquid layer between bubble bases and by drawing cooler liquid towards the wall during contractions. It is evident that the temperatures at the surface and beneath the surface are fluctuating with time, depending on the frequency of bubble formation, their merger and release. This is confirmed by the experimental investigations of Buchholz \textit{et al}\textsuperscript{9} using micro thermocouples. In the present work, the local boiling phenomenon on different metallic surfaces in different non-circular flow passages under laminar flow conditions is studied and an empirical correlation is presented for estimating the heat transfer coefficients under local boiling conditions.

**Experimental Procedure**

The schematic diagram (Fig. 1) shows the general arrangement of the equipment. A large reservoir is used for the liquid to be circulated through the test...
section by means of a centrifugal pump. The test section is an isosceles triangular channel with a base of 7.62 cm and sides of 10.16 cm each. The heat transfer surface is a metallic plate of 6.35 mm thickness, 60.96 cm length and 12.7 cm width. In the present investigation, the test plates are made up of zinc, tin, brass and gunmetal. When these plates are fixed at the bottom of the test section, the cross section of the closed channel is an isosceles triangle. The top surface of each of the plates used as a heat transfer surface is machined to provide a uniformly smooth surface throughout. The plate is fixed at the bottom of the test section, with a thick packing in between, with a minimum number of bolts. To measure the temperature of the test plate, holes 5.1 cm long and 1.6 mm diameter are drilled across the plate through the mid-plane thickness of the plate, i.e., at 3.2 mm distance from the surface. Four such holes equidistant are drilled on each side of the plate and a total of eight copper-constantan thermocouples are inserted in eight holes (Fig. 2).

In order to minimise the end effects at the entrance and exit of the test section and to establish uniform velocity distribution within the test section, four calming sections, two on each side of the test section, are fixed with a thick heat insulating material in between them. These calming sections have the same geometrical shape and internal dimensions as that of the test section and when fixed they are flush with the test section. To measure the inlet and outlet temperatures of the liquid passing through the test section, two thermometers (mercury in glass) of 1/10°C accuracy are placed, one at the entrance and the other at the exit of the coolant. The hot liquid from the test section passes through the downstream calming section and is cooled in a single pass counter current heat exchanger before it is circulated to the reservoir. The coolant to be heated is forced by a centrifugal pump through the first calming section, test section, the second calming section and the heat exchanger in that order and then returned to the reservoir.

Results and Discussion

Local boiling phenomenon when applied to the laminar flow forced convective heat transfer is expected to improve the heat transfer rates by several folds. In order to study this phenomenon on metallic surfaces of different thermal properties, the non-boiling data using the heat transfer surfaces of zinc, tin, brass and gunmetal with water as coolant through an isosceles triangular channel are taken. The method of correlation for the experimental laminar flow heat transfer data in non-circular channels of different geometries presented in the literature has been adopted in the present case. The ‘‘J’’ factors and Reynolds numbers based on an equivalent diameter of the test section are computed for all the runs taken and the ‘‘J’’ factors are plotted against Reynolds numbers on a log-log graph (Fig. 3). All the data are

0.0762 m X 0.1016 m X 0.1016 m

0.6096 m

0.5937 m

0.0079 m

0.0127 m

0.127 m

0.127 m

0.194 m

0.0127 m

0.0127 m

0.0079 m

0.5937 m

0.6096 m

Elevation

Sectional end view

Fig. 1 — Schematic diagram of the experimental setup

Fig. 2 — Test section with plate

Fig. 3 — Triangular channel-non boiling

◊ Zinc plate, □ Tin plate, △ Brass plate, × Gun metal plate
in the formation of a single straight line with the slope of $-0.91$, the intercept being 11.5.

Hence the equation correlating the heat transfer data obtained under developing laminar flow non-boiling conditions on different metallic heat transfer surfaces made of zinc, tin, brass and gunmetal which form the heated bottom plate of the isosceles triangular channel with water as coolant is given by

$$J = 11.5(Re)^{-0.91} \quad \ldots(1)$$

The regression coefficient is 0.98 and the standard error of estimate is 0.0446. The 95% confidence limits for the intercept are 8.03 and 16.428, for the slope are $-0.865$ and $-0.955$ and for the regression coefficient are 0.97 and 0.99.

In correlating the local boiling data, an attempt is made to arrive at an unified correlation for the heat transfer rates under local boiling conditions which can account the following parameters that are expected to have influence on the phenomenon.

1. Apparent temperature excess, $\Delta T_{XA}(T_s - T_b)$,
2. Degree of sub-cooling, $\Delta T_{SC}(T_b - T_{av})$,
3. Geometric configuration of the flow passage,
4. Different physical properties of the metals used for heat transfer surfaces, and
5. Physical properties of liquids used as coolants.

In this process along with the experimental local boiling data obtained in the present investigation, the extensive local boiling experimental data on different metallic surfaces made of copper, aluminum, brass, stainless steel and lead available in the reference 2 are used.

The non-boiling coefficient obtained in the present triangular channel is $K_{nb}=J. Re^{0.91}=11.5$. This non-boiling coefficient $K_{nb}$ will increase several folds under local boiling conditions and is designated as local boiling coefficient $K_{lb}$. This increase in heat transfer coefficient is because of the superimposition of the heat content released by the condensation of the bubbles emerging out of local boiling, over that of the usual forced convective heat transfer and also because of the improved free convective currents due to the movement of the bubbles across the cross section of the flow passage.

It is observed from the present local boiling data obtained with zinc plate that when $\Delta T_{XA}$ is increased, the local boiling coefficient $K_{lb}$ increases correspondingly. Hence, a correlation between apparent temperatures excess $\Delta T_{XA}$ and the local boiling coefficient $K_{lb}$ is tried on a log-log graph as suggested earlier. A plot of $K_{lb}$ versus $\Delta T_{XA}$ has been constructed (Fig. 4) with horizontal datum line of constant non-boiling coefficient at 11.5, representing the limit of non-boiling region and above this line exists the region of local boiling in which coefficient $K_{lb}$ increases as $\Delta T_{XA}$ is increased. The line representing the trend of the increased local-boiling coefficients when intersects the $K_{nb}$ line, the abscissa of the point of intersection denotes the threshold temperature excess, $\Delta T_{xo}$, which is the limiting temperature excess that is required for local boiling to take place. In the case of water and zinc metal surface the $K_{lb}$ varies as $(\Delta T_{XA})^{1.0}$ and the similar trend is observed with tin-water, brass-water and gunmetal-water systems, $K_{lb}$ varies as $(\Delta T_{XA})^{0.59}$, $(\Delta T_{XA})^{0.73}$ and $(\Delta T_{XA})^{0.84}$, respectively. And to facilitate quick reference and comparison the data of these four systems are summarized in Fig. 5 with straight-line diagrams representing each surface.

In order to evaluate the influence of the degree of sub-cooling on heat transfer rates under local boiling conditions, the boiling coefficients are plotted on a log-log graph against $\Delta T_{SC}$ for the zinc-water system (Fig. 6). It is observed from the graph that the $K_{lb}$ values sharply fall with the increasing $\Delta T_{SC}$. The trend is same for all the data of the present investigation as well as for the data taken from the reference. Hence, it is established that boiling coefficient increases with $\Delta T_{XA}$ and decreases when $\Delta T_{SC}$ is increased showing the pronounced effect of the degree of sub-cooling. It is evident that the increase or the decrease of $K_{lb}$ is
dependant on the ratio $\Delta T_{xa}/\Delta T_{sc}$. The data of boiling coefficients for the zinc-water system when plotted against $\Delta T_{xa}/\Delta T_{sc}$, are well represented by a straight line with a positive slope of 0.57 (Fig. 7). Similarly, all the data of boiling coefficients when plotted against $\Delta T_{xa}/\Delta T_{sc}$, are well represented by straight lines with positive slopes varying from 0.45 to 0.57. The apparent variation in the slopes of straight lines representing the boiling coefficient data with respect to different metallic heat transfer surface-water systems in log-log plots of $K_{ib}$ versus $\Delta T_{xa}$ has been brought to an average slope of 0.5.

The laminar flow heat transfer rates under non-boiling conditions, in the absence of any phase change, are different for different non-circular channels because of their geometric and dynamic similarities and separate, well established empirical and analytical expressions are available in the literature. In the present investigation also, the non-boiling coefficient, $K_{nb}$ values are different for triangular, rectangular and semicircular channels and are equal to 11.5, 22.5 and 15, respectively in the laminar flow region and Prandtl numbers varying from 4 to 70. When local boiling is taking place on different metallic surfaces, the mode of heat transfer is different unlike in the non-boiling conditions, because the heat is being transferred by two distinctive modes, one by the forced convection associated with natural convection and the other by the heat released in the main stream of coolant by the ebullition of bubbles emanating from the local boiling occurring in the thin layer adjacent to the heat transfer surface. The amount of heat transported to the mid stream of the liquid coolant by the condensing bubbles is super imposed over the convective heat transfer attained in it. Further, the quantity of heat super imposed is dependent on the properties of the metallic heat transfer surface which give rise to the number of growth centres of bubbles, their intensity and frequency.

It follows from this reasoning and analysis that the ratio of $K_{ib}/K_{nb}$ quantitatively accounts for the superimposed heat transfer by local boiling over the laminar flow convective heat transfer in any channel and on any metal heat transfer surface-water system and it should vary as $\Delta T_{xa}/\Delta T_{sc}$. In order to verify the
same, a plot of $K_{lb}/K_{nb}$ versus corresponding ratio $\Delta T_{XA}/\Delta T_{SC}$ for the local boiling data obtained in the semi-circular channel, $K_{nb} = 15$, and triangular channel $K_{nb} = 11.5$ on the same gunmetal surface is presented on a log-log graph (Fig. 8). It can be seen from the plot that all the data of $K_{lb}/K_{nb}$ in both the channels with gun metal heat transfer surface, almost fall on a single straight line with a slope of 0.5, confirming that the flow passage geometry has no effect on the contribution part of heat transfer by local boiling.

In order to estimate the contribution of the local boiling on different metallic heat transfer surfaces, the data for copper, aluminum, brass and stainless steel in rectangular channel and the data of the gun metal in semi-circular channel along with the local boiling data obtained on zinc, tin, brass and gun metal in triangular channel are plotted on a log-log graph as $K_{lb}/K_{nb}$ versus $\Delta T_{XA}/\Delta T_{SC}$ (Fig. 9). The data representing the copper surface in rectangular channel line up on the straight line with a slope of 0.5 at the top of entire data while the data representing the stainless steel surface in the same rectangle fall as the lowest and the last straight line with the same slope 0.5. The rest of the data falls between these two limits of copper and stainless steel in the order of their thermal conductivities but the variations of $K_{lb}/K_{nb}$ values are not proportional to their respective thermal conductivities. For example, the data for aluminum heat transfer surface is very close to that of copper, not in proportion to the vast difference in their thermal conductivities of 206 and 377 W/m-K. Hence, some other physical properties of the metal heat transfer surface coupled with their respective thermal conductivities may be influencing the super imposed local boiling heat transfer rates.

It is expected that under local boiling conditions, unsteady state could prevail at the portions of the nucleation centers of bubbles on the heat transfer surface. When these bubbles begin to grow, the liquid micro-layer at the base of these bubbles evaporates
and an enhanced quantity of heat is required for the latent heat of vaporization in addition to the heat that is being transferred by the metallic surface under steady state. This process causes lower temperatures at the bubble growth centers leaving the rest of the area at higher temperatures. It is expected that heat at a higher temperature will rush to the area of unsteady state, which is at a lower temperature, from all directions and this process of heat transfer takes place by thermal diffusion.

The copper metal has the highest thermal diffusivity while the lowest being that of the stainless steel and the thermal diffusivity of aluminum fall close to copper. It is well established that Fourier number \((F_o)\) is a measure for unsteady state heat transfer and is employed in the present investigation to account for the different rates of heat transport by the mechanism of thermal diffusivity. Hence, the \(K_{lb}/K_{nb}\) values for all the metal surfaces at a particular value of \(\Delta T_{XA}/\Delta T_{SC}\) are read from the figure and plotted against their respective Fourier numbers on a log-log graph (Fig. 10). From the figure it is evident that \(K_{lb}/K_{nb}\) is a function of \((F_o)^{0.25}\). The equation correlating all the local boiling data obtained on different metallic surfaces from copper to stainless steel in channels of rectangular, triangular and semi-circular cross sections under laminar flow conditions with water as coolant, is

\[
K_{lb} = 1.82 (K_{nb}) (F_o)^{0.25} (\Delta T_{XA}/\Delta T_{SC})^{0.5} \quad \ldots(2)
\]

In order to arrive at a more comprehensive and unified correlation, the local boiling data for 40% glycerol-water mixture on copper, aluminum and stainless steel in rectangular channel\(^2\) and local boiling data of water, 20, 40 and 60% glycerol-water mixtures obtained on brass heat transfer surface also in rectangular channel\(^1\) are taken for analysis. A thin layer adjacent to the heat transfer surface is super heated under local boiling conditions and this zone may be considered as under local boiling. The rate of heat transfer across the interface of the superheated thin layer and the vapour bubble limits the vapour bubble growth. The quantity of heat required to superheat a unit volume of the liquid coolant to a particular degree, varies with the liquid coolant and similarly the quantity of heat required to produce a unit volume of vapour also varies from coolant to coolant. The relation between the heat flux expended to superheat a unit volume of liquid coolant and the volumetric heat of vaporization for different coolants can be accounted using the Jakob number \([[(\Delta \rho C_p)/(\rho \lambda)]\]. Jakob numbers have been calculated for all the data considering a uniform rise of superheat for all the coolants through \(1^\circ C\) and the liquid film temperature is taken as \(t_b\) to calculate the physical properties.

A plot of \((K_{lb}/K_{nb}) (F_o)^{-0.25}\) at a constant \((\Delta T_{XA}/\Delta T_{SC})\) value versus Jakob numbers is constructed on a log-log graph (Fig. 11) and it is found that \((K_{lb}/K_{nb}) (F_o)^{-0.25}\) varies as \(Ja^{-2.5}\). Hence, the general correlation can be given as (Fig. 12) for \((\Delta T_{XA}/\Delta T_{SC})>0.3\)

\[
(K_{lb}/K_{nb})(F_o)^{-0.25} Ja^{-2.5} = 0.11 (\Delta T_{XA}/\Delta T_{SC})^{0.5} \quad \ldots(3)
\]

The regression coefficient is 0.92 and the standard error of estimate is 0.05477. The 95% confidence limits for the intercept are 0.1089 and 0.113, for the
slope are 0.47 and 0.529 and for the regression coefficient are 0.89 and 0.935.

Here in deriving this empirical equation, the degree of superheat of the thin liquid coolant layer, adjacent to the heat transfer surface giving rise to vapour bubbles, is assumed to be as 1°C for all the data. Actually, this degree of superheat is basically dependent on the two limiting thermal boundary conditions; one is the heat transfer surface temperature \( t_s \) and the other being the average bulk temperature of the coolant \( t_{av} \). These two thermal boundary conditions have already been considered and incorporated into the correlation through the parameters \( \Delta T_{XA} = t_s - t_b \) and \( \Delta T_{SC} = t_b - t_{av} \), and hence the assumption of \( \Delta t = 1°C \) is used for the present correlation and with the mathematical rearrangement of the above equation by including the \( (\Delta T_{XA}/\Delta T_{SC})^{0.5} \) term in the Jakob number instead of \( \Delta t \) an estimate for the degree of superheat as \( \Delta t = (\Delta T_{XA}/\Delta T_{SC})^{0.2} \) can be arrived.

A modified Jakob number, \( Ja_{mod} \), is defined as

\[
Ja_{mod} = [(\Delta T_{XA}/\Delta T_{SC})^{0.2} \rho C_p]/[\rho \lambda]
\]

The entire data, with \( (\Delta T_{XA}/\Delta T_{SC}) > 0.3 \), can be correlated by the modified form of the Eq. (3).

\[
K_{lb} = 0.11 \left( K_{nb}\right)^{0.25} (Ja_{mod})^{2.5} \ldots(4)
\]

**Conclusion**

From the above observations, it can be inferred that, (i) The present equation can handle and correlate data of local boiling obtained on different metallic heat transfer surfaces with different coolants of different physical properties in channels of having different geometrical flow configurations. (ii) This could be a very useful equation for the design of the heat transfer equipment where local boiling phenomenon is expected to occur under laminar flow conditions. This design equation can give probable heat transfer rates (a) for conditions of degree of subcooling employed, (b) for industrially important metals of construction of heat transfer equipment and (c) for varying physical properties of the liquid coolants employed. (iii) The limits of the usage of this equation have to be verified with some more data obtained with pure liquids of varying physical properties.
Nomenclature

\( C_p \) = specific heat of the liquid coolant, J/kg K

\( D_e \) = equivalent diameter, \( 4r_h \), m

\( K_{lb} \) = local boiling coefficient \((J.Re)^m\)

\( K_{nb} \) = non-boiling coefficient \((J.Re)^m\)

\( r_h \) = hydraulic radius, m

\( t_b \) = boiling point of the coolant, °C

\( t_s \) = surface temperature, °C

\( t_{av} \) = average temperature of the liquid, °C

\( v \) = average velocity, m/s

\( \Delta T_{XA} \) = \((t_s-t_b)\), °C

\( \Delta T_{SC} \) = \((t_b-t_{av})\), °C

\( J \) = Colburn \('J'\) factor, \((St(Pr)^{2/3})\)

\( Re \) = Reynolds number, \( D_e G/\mu \)

\( St \) = Stanton number, \( h/(C_p G) \)

\( Pr \) = Prandtl number, \( C_p \mu/k \)

\( F_o \) = Fourier number, \( 4\alpha L/(vD^2) \)

\( Ja \) = Jakob number, \( (\Delta\rho_k C_p)/(\rho_v \lambda) \)

\( Ja_{mod} \) = Jakob number with \( \Delta = (\Delta T_{XA}/\Delta T_{SC})^{0.2} \)

\( \rho_l \) = density of the liquid, kg/m³

\( \rho_v \) = density of vapour, kg/m³

\( \phi_v \) = Viscosity correction factor, \((\mu/\mu_0)^{0.14}\)

\( \mu \) = viscosity, P

\( \lambda \) = latent heat of vapourization, J/kg

References


