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MEASUREMENT OF SUBCOOLED BOILING PRESSURE DROP AND LOCAL HEAT TRANSFER COEFFICIENT IN A HORIZONTAL CHANNEL UNDER LPLF CONDITIONS

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Abstract

Horizontal flow is commonly encountered in boiler tubes, refrigerating equipments and nuclear reactor fuel channels of pressurized heavy water reactors (PHWR). Study of horizontal flow under low pressure and low flow (LPLF) conditions is important in understanding the nuclear core behavior during situations like LOCA (Loss of coolant accidents). In the present work, experimental measurements of local heat transfer coefficient and pressure drop are carried out in a horizontal channel under LPLF conditions of sub-cooled boiling. Infrared thermography is used for the measurement of local wall temperature to estimate the heat transfer coefficient in single phase and two phase flows with water as the working medium at atmospheric pressure. Correlation for single phase diabatic pressure drop ratio (diabatic to adiabatic) as a function of viscosity ratio (wall temperature to fluid temperature) is presented. Correlation for pressure drop under sub-cooled boiling conditions as a function of *Bo* (Boiling number) and *Ja* (Jacob number) is obtained. Correlation for single phase heat transfer coefficient in the developing region is presented as a function of *z/d* (ratio of axial length of the test section to diameter). Correlation for two-phase heat transfer coefficient under sub-cooled boiling condition is presented as a function of *Bo*, *Ja* and *Pr* (Prandtl number). Correlation between heat transfer coefficient and friction factor is obtained by applying Reynolds analogy.

1. Introduction

Horizontal flow is common in boiler tubes, refrigerating equipments and nuclear reactor fuel channels of PHWRs. In PHWR, under loss of coolant accident conditions, the core fuel channels encounter very low mass velocities, flow stagnation and flow reversal conditions. Understanding the local heat transfer coefficient and pressure drop under low pressure and low flow conditions is very important to ensure the safety of fuel channels of PHWR in accident conditions similar to LOCA.

1.1 Local heat transfer coefficient

Boiling heat transfer is employed in many industrial processes because of its effectiveness in cooling heat transfer equipments. There is a great demand for two phase flow heat exchangers because of high heat transfer coefficient associated with boiling. The two phase heat transfer coefficient, unlike single phase heat transfer coefficient, is a function of thermodynamic 'quality' which varies along the length of the channel during flow boiling. Saitoh *et al*. [1] carried out experiments on horizontal small diameter tubes using R134a as the working fluid. They conducted experiments for a mass flux of 150 – 450 $kg/m^2.s$ for thermodynamic qualities varying from 0 – 0.2. Local heat transfer coefficient was measured using thermocouples. Choi *et al*. [2] conducted experimental studies to measure boiling heat transfer coefficient of R-22, R-134 and CO_2 in horizontal smooth minichannels of 1.5 *mm* and 3 *mm* diameter using thermocouples. Mass flux and heat flux ranges covered in their study are 200-600 $kg/m^2.s$ and 10 – 40 kW/m^2 up to an exit quality of unity.

Knowledge of spatial variation of temperature is essential for the calculation of local heat transfer coefficient. The limitation associated with the use of thermocouples for surface temperature measurement can be overcome by using thermo-graphic techniques. The first investigation for the determination of local heat transfer coefficient using thermo-graphic technique is reported by Hapke *et al*. [3]. They reported a model for the prediction of the initial point of boiling and wall superheat on the basis of experimental results. Diaz and Schmidt [4] investigated flow boiling phenomena in channels using water, hexane, ethanol and octane as working fluids. Infrared imaging technique is used for capturing outer surface temperature of heater pipe. Boye *et al.* [5] based on the experiments done using thermal camera developed a model for heat transfer during convective boiling in mini channels.

Heat transfer coefficient in sub-cooled flow boiling is higher compared to that of single phase flow. With the increase in quality (in the sub-cooled region), heat transfer coefficient increases and attains a maximum value at $x = 0$. Kandlikar [6] carried out investigations in flow boiling and developed a flow boiling map to depict the relationship among heat transfer coefficient, quality, heat flux and mass flux for different fluids in sub-cooled and saturated flow boiling regions. Bao *et al.* [7] conducted experiments for determining heat transfer coefficient for R11 and HCFC123 in smooth copper tube with inner diameter of 1.95 *mm*. They reported 'local variation' of sub-cooled heat transfer coefficient with quality under different heat flux, mass flux and system pressure. Results show that the heat transfer coefficient is independent of mass flux and increases with the increase in heat flux.

Correlations available in the literature for calculating sub-cooled flow boiling heat transfer coefficient are obtained using experimental data employing thermocouples for the measurement of surface temperatures. No correlation is presently available in the literature that predicts the 'local' variation of heat transfer coefficient using the surface temperature data which is truly local and continuous. Hence, there is a need for performing experiments in sub-cooled boiling region in horizontal channels to study the influence of mass flux, heat flux, inlet sub-cooling, diameter, quality and length on the local heat transfer coefficient.

1.2 Pressure drop under sub-cooled boiling conditions

Pressure drop is important in designing heat-removal systems utilizing high-heat-flux sub-cooled boiling. Tong *et al.* [8] reported an experimental investigation on pressure drop across small-diameter vertical tubes in highly sub-cooled flow boiling. The effect of five parameters namely mass flux, inlet temperature, exit pressure, tube internal diameter and length-to-diameter ratio on both single-phase and two-phase pressure drop is studied. The range of parameters covered in the study are: inside diameters 1.05 to 2.44 *mm*, mass fluxes from 25,000 to 45,000 $kg/m^2\ s$, exit pressures from 4 to 16 *bar*, and inlet temperatures from 22 to 66 *°C*. Pressure drop correlations are presented for predicting both single-phase and sub-cooled boiling pressure drop in small-diameter tubes under different heat-flux conditions. Hahne *et al*. [9] obtained pressure drop data on sub-cooled flow boiling of refrigerant R12 and R134a using a 20 *mm* diameter and 500 *mm* long vertical test section made of copper tube. Range of parameters considered were pressure: 8-20 *bar*; mass fluxes: 750 – 3000 kg/m^2s; sub-cooling: 2 - 47.6 *K*; and heat fluxes: 0 – 207500 Wm^{-2}. A correlation as a function Boiling number (*Bo*) and Jakob number (*Ja*) was suggested using the reduced pressure drop (sum of frictional and accelerational component) data. Hoffman and Wong [10] developed a semi-empirical model for the prediction of the pressure drop in sub-cooled convective boiling of water flows in both horizontal and vertical tubes subject to uniform high heat fluxes. The basic equations used for the pressure gradient due to friction, flow acceleration and gravity based on the separated flow model were modified for sub-cooled boiling regime. The correlations available in the literature are, in general, developed using data from vertical test sections. Hence, there is a need to study the pressure drop at LPLF conditions under sub-cooled boiling condition in a horizontal channel.

Jordan and Leppert [11] experimentally showed that the friction factor for sub-cooled boiling heat transfer agrees reasonably well with the Reynolds analogy given by

$$St = \frac{f}{8} \tag{1}$$

where f is the friction factor and St is the Stanton number. This is based on the assumption that, during sub-cooled boiling, the high degree of liquid agitation due to the bubbles activity virtually destroys the laminar sub-layer, which is the substantial contributor to the thermal resistance during single phase heat transfer [11]. The deviation from the analogy found for the data is attributed to the entrance effect seen in all of the experimental runs having higher wall temperature observed at the entrance.

Based on the literature survey, the major objectives of the present work are as follows

- Measurement of local heat transfer coefficient using infrared thermography in sub-cooled boiling conditions under low pressure and low flow conditions in small diameter stainless steel tubes (5.5 *mm*, 7.5 *mm* and 9.5 *mm*) and suggest a correlation for *Nu*.
- Measurement of pressure drop under sub-cooled boiling conditions and establish a functional relationship between friction factor and Nusselt number using Reynolds analogy.

2. Experimental setup and procedure

Experimental setup used for the measurement of local heat transfer coefficient and pressure drop in sub-cooled boiling conditions is shown in Figure 1. The system consists of a gear pump (1H.P, 0-360 *g/s*) and a speed controller for varying the flow rate of gear pump and a manually controlled valve on the pump delivery side for regulating the water flow through the test section. Circular stainless steel (SS304) tubes of outside diameter 6.0 *mm*, 8.0 *mm* with 0.55 *m* length and outside diameter 12.5 *mm*, 10.0 *mm* with 1.0 *m* lengths are used as test sections for validation experiments. The wall thickness of the test sections is 0.25 *mm*. The length chosen for the test sections are sufficiently higher than the hydrodynamic and thermal developing length (L/d = 40-80). The test section wall is heated directly using a 20 *kW* (40 *V*, 500 *A*) capacity AC transformer. The voltage and hence the power applied to the test section is varied using a variac. Fluid temperatures at the inlet and the outlet of the test section are measured by thermocouples. The temperature distribution on the heated test section is measured by 'Thermoteknix' Ti200 infrared camera. The surface of heater element facing the camera is painted black using a thin coat of black board paint which provides an emissivity of unity. The uncertainty in the temperature measurement is not more than $\pm 0.5°C$. Mass flow rate of the water flowing through the test section is measured using an electromagnetic flow meter (Rosemount make, accuracy ± 0.25%). Differential pressure transmitters (Rosemount make, accuracy ± 0.1% of the range) of ranges 6.22 *kPa* and 62.2 *kPa* are used to measure the test section pressure drop.

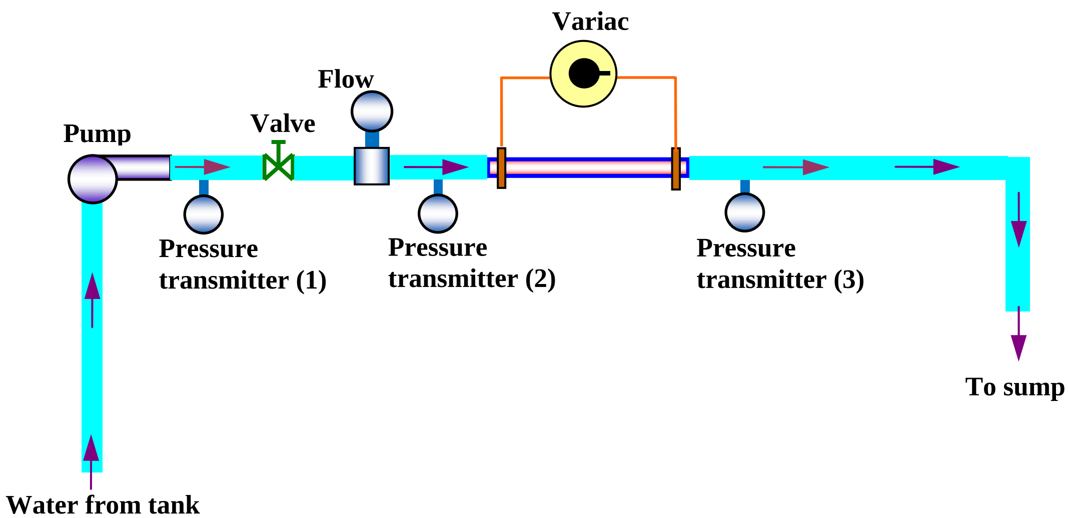

Figure 1 Schematic of the set-up for the measurement of local heat transfer coefficient

3. Results and discussions

Adiabatic single phase friction factor measurements are performed for 5.5 *mm* diameter test section of 0.55 *m* length to validate the experimental setup. The adiabatic friction factor is compared with that of Colebrook's correlation. Surface roughness of 4.5 μm is assumed. The predicted friction factor value has a maximum deviation of ± 6 %. The uncertainty in the measurement of friction factor is around 5% using the methodology described by Moffat [12].

3.1 Pressure drop measurements

Pressure drop is measured under diabatic single phase and sub-cooled boiling conditions for the following range of parameters: mass flux (450 – 935 $kg/m^2.s$), heat flux (170 – 530 kW/m^2), inside diameter (5.5 *mm*, 7.5 *mm* and 9.5 *mm*), length (0.55 *m*, 0.75 *m* and 1.0 *m*) and inlet temperature (28°*C*, 50°*C* and 70°*C*). A typical variation of the pressure drop with heat flux is shown in Figure 2. For a given mass flux as the heat flux is increased, the pressure drop initially decreases due to the decrease in fluid viscosity in the near-wall region and the pressure drop increases later. As the wall super-heat $T_w - T_{sat}$ reaches a critical value, boiling occurs and a thin bubbly layer builds up on the heated surface [8] resulting in the increase of the pressure drop. Measured pressure drop, for which the wall super-heat is less than the critical value of nucleation suggested by Davies and Anderson [13] as given by equation 2, is categorized as single phase pressure drop. If the test section wall super-heat exceeds the nucleation criteria, then the measured pressure drop is considered as two-phase pressure drop.

$$T_w - T_{sat} = \sqrt{\frac{8\sigma T_{sat} q''}{\rho_v h_{fg} k_l}} \qquad (2)$$

Single-phase diabatic pressure drop data is correlated as a function of friction factor ratio (f/f_a) and wall-to-bulk viscosity ratio ($\bar{\mu}_w/\bar{\mu}_l$) [8]. Diabatic friction factor is normalized with the adiabatic friction factor estimated at the fluid average temperature using Blasius friction factor correlation

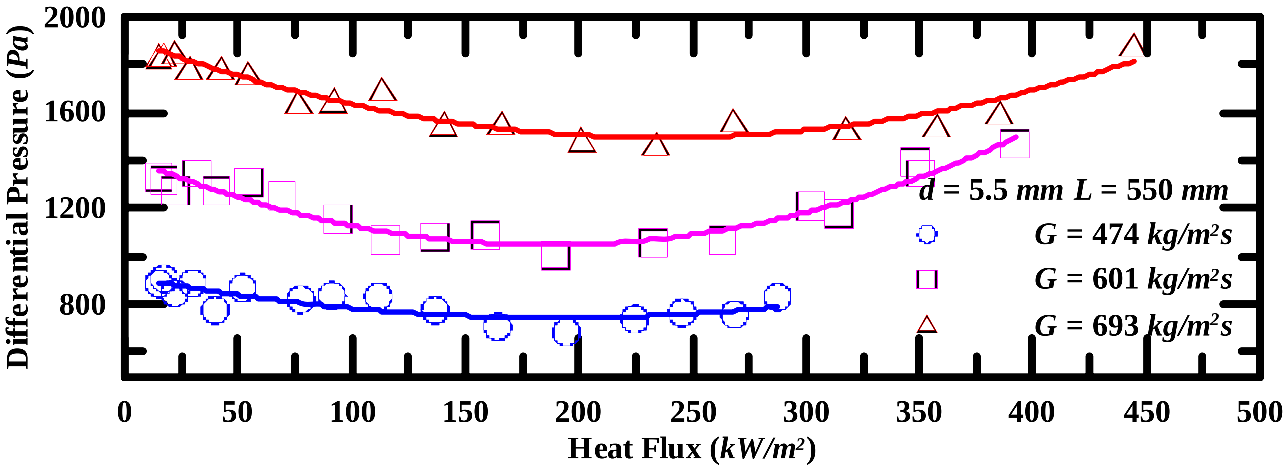

Figure 2 Variation of the pressure drop with the heat flux and mass flux.

($f_a = 0.316\ Re^{-0.25}$). Correlation developed separately for each diameter predicts the data with a maximum deviation of ± 20% and root mean square deviation (rms) of 8.5%. The combined correlation for all the three diameters and lengths predicts experimental data with a maximum deviation of ± 40% and rms deviation of 21% . The combined correlation is given in equation 3.

$$\frac{f}{f_a} = 0.9185\left(\frac{\overline{\mu}_w}{\overline{\mu}_l}\right)^{0.16} \tag{3}$$

The average fluid temperature along the tube length is taken as the mean of the measured inlet and exit temperature and the average wall temperature is estimated using the equation

$$\overline{T}_w = \overline{T}_l + \frac{q''}{\overline{h}} \tag{4}$$

where $\overline{h}$ is the average heat transfer coefficient and is obtained from the correlation given by [14]

$$Nu = 0.0157\ Pr^{0.4}\ Re^{0.85} \tag{5}$$

Results show that the friction fraction ratio decreases with the decreasing wall-to-bulk viscosity ratio; that is, with the increasing temperature difference $\overline{T}_w - \overline{T}_l$. Higher temperature difference represents the higher heat flux applied to the test section, resulting in a lower pressure drop.

A correlation for diabatic pressure drop under sub-cooled boiling conditions in terms of Boiling number (*Bo*) and Jakob number (*Ja*) is suggested by Hahne *et al*. [9] for water in a 20 *mm* diameter test section as given by

$$\frac{\Delta P_{sb}}{\Delta P_{1ph}} = 1 + 32500 Bo^{1.6} Ja^{-1.2} \tag{6}$$

ΔP_{1ph} is the adiabatic single phase pressure drop calculated using Blasius friction factor equation. The *Ja* number is estimated based on the fluid inlet temperature. Measured pressure drop data is tested with the above correlation. The root mean square (rms) deviation is 26% with a maximum deviation of ± 60%. A new correlation is attempted incorporating diameter effect into Hahne's correlation [9] and is given by.

$$\frac{\Delta P_{sb}}{\Delta P_{1ph}} = 1 + 32500 Bo^{1.6} Ja^{-1.2}\left(\frac{d}{20}\right)^{0.54} \tag{7}$$

where d is the diameter in *mm.* This modified correlation predicts the present experimental data within an rms deviation of 17% and a maximum deviation of ± 30%.

3.2 Validation of infrared technique for measurement of local heat transfer coefficient

The wall temperature measurement technique using thermal camera is validated by conducting single phase heat transfer measurement. The local heat transfer coefficient is obtained using the equation given by

$$h = \frac{q''}{\left(T_w(z) - T_l(z)\right)} \tag{8}$$

where q'' is the applied heat flux , $T_w(z)$ is local wall temperature and $T_l(z)$ is local fluid temperature which is obtained by linear interpolation between inlet and exit fluid temperatures.

Variation of single phase local heat transfer coefficient with axial length and thermodynamic quality is shown in Figure 4 for a test section of 5.5 *mm* diameter and 550 *mm* length. Comparison of the experimental results with the heat transfer coefficients estimated at local bulk fluid temperature using Dittus Boelter correlation [14] and Gnielinski correlation [15] is also shown in Figure 3. The test is carried out for *Re* in the range of 5000-12400 and the deviation is found to be within ± 15%. The variation of wall temperature and fluid temperature along the axial length is shown in Figure 4. Due to the initial thermal developing region, temperature difference increases till z/d = 20. Beyond z/d = 20, the temperature difference is constant showing the fully developed flow. A correlation for heat transfer coefficient in the thermally developing single phase region is developed and is given by equation 9. The correlation fits the data within 20% deviation with an rms deviation of 7.41 %.

$$\frac{h_{exp}}{h_{DB}} = 0.0012\left(\frac{z}{d}\right)^2 - 0.0501\left(\frac{z}{d}\right) + 1.697 \tag{9}$$

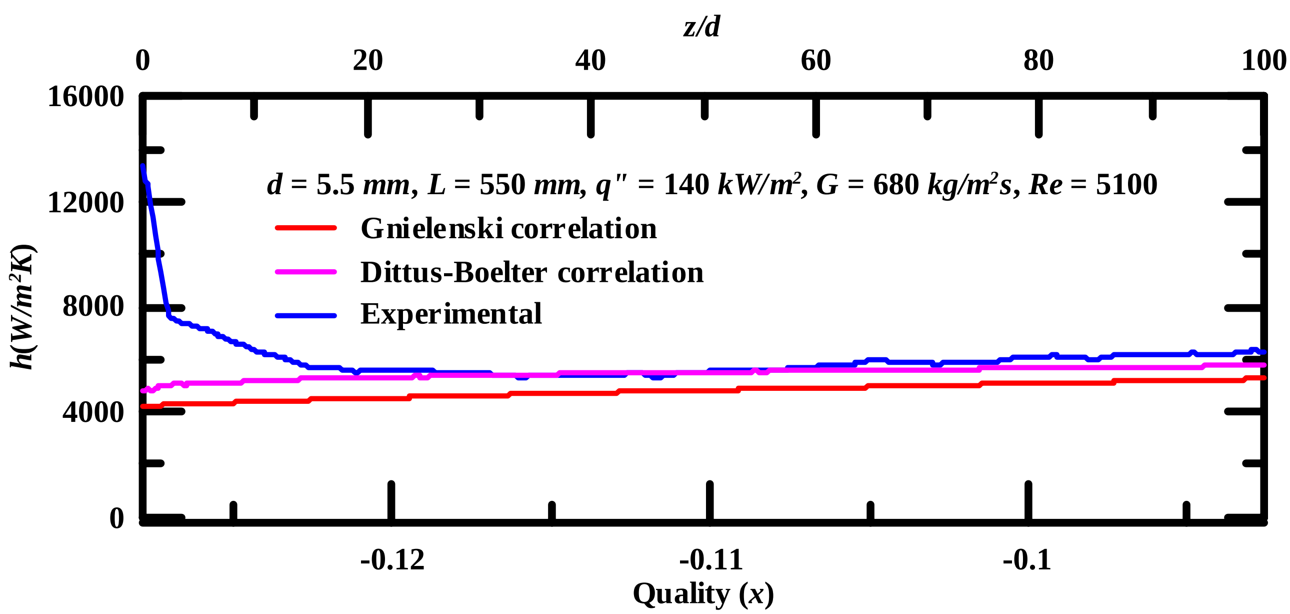

Figure 3 Comparison of experimental and estimated heat transfer coefficient for a circular pipe of 5.5 mm diameter and length 550 mm at *Re* = 5100

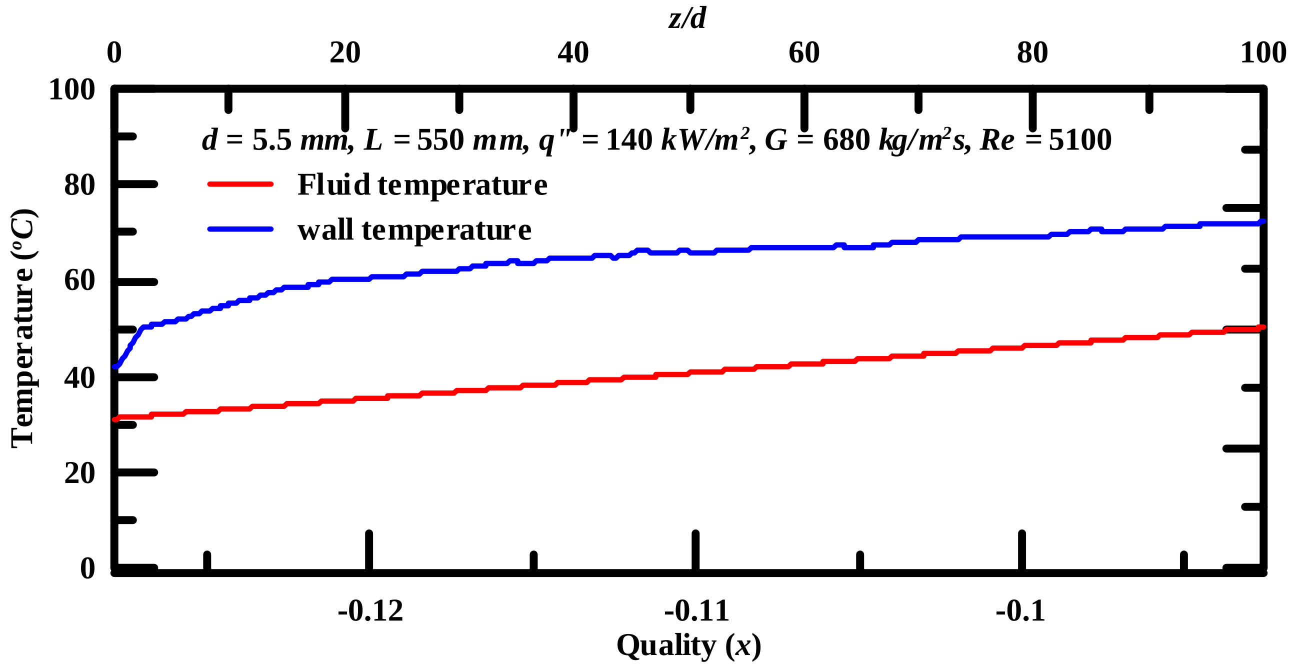

Figure 4 Variation of outer wall temperature and fluid temperature along axial length of a circular pipe of 5.5 *mm* diameter and length 550 *mm* at *Re* = 5100

3.3 Measurement of two phase heat transfer coefficient

Experiments are carried out at atmospheric pressure conditions to determine sub-cooled boiling local heat transfer coefficient under various conditions of mass flux and heat flux. Typical variation of heat transfer coefficient along the length of a test section with a constant heat flux boundary condition is shown in Figure 5. The wall temperature along the length of the test section from the inlet side increases at a comparatively higher rate up to z/d = 20 and more or less remains constant after z/d = 40 as seen in Figure 6. This initial temperature rise may be attributed to the effect of the thermal developing length. The liquid in this region is single phase as the wall temperature is below the required value for incipience of nucleate boiling and the heat transfer coefficient has a decreasing trend due to the increase in $T_w(z)$- $T_l(z)$. Beyond z/d = 40, the temperature difference, $T_w(z)$ - $T_l(z)$ decreases and hence the heat transfer coefficient increases continuously along the length of the test section. The wall temperature required for the onset of nucleation is shown in Figure 6 (estimated using equation 2) and is about 109 ^oC. Under fully developed conditions with fluid in single phase, the heat transfer coefficient approaches to the value predicted by the standard correlations of Dittus Boelter [14] or Gnielinski [15]. This is seen in Figure 5 where the ratio h_{exp}/h_{DB} approaches to 1 at approximately z/d = 40. Beyond this point, the heat transfer coefficient starts increasing due to the onset of nucleate boiling. The wall temperature measured shown in Figure 6 indicates the onset of boiling condition at an approximate value of z/d = 35. This observation suggests that the increase in heat transfer coefficient beyond the region z/d = 40 is due to the occurrence of nucleate boiling and this region can be identified as the sub-cooled boiling region as shown in Figure 5.

3.3.1 Effect of heat flux on sub-cooled heat transfer coefficient

Figure 7 shows the variation of heat transfer coefficient with thermodynamic quality for 5.5 *mm* diameter test section for different heat fluxes. In all these experiments, the mass flux is maintained reasonably constant (within ± 1%). For a given quality, the wall temperature increases with the

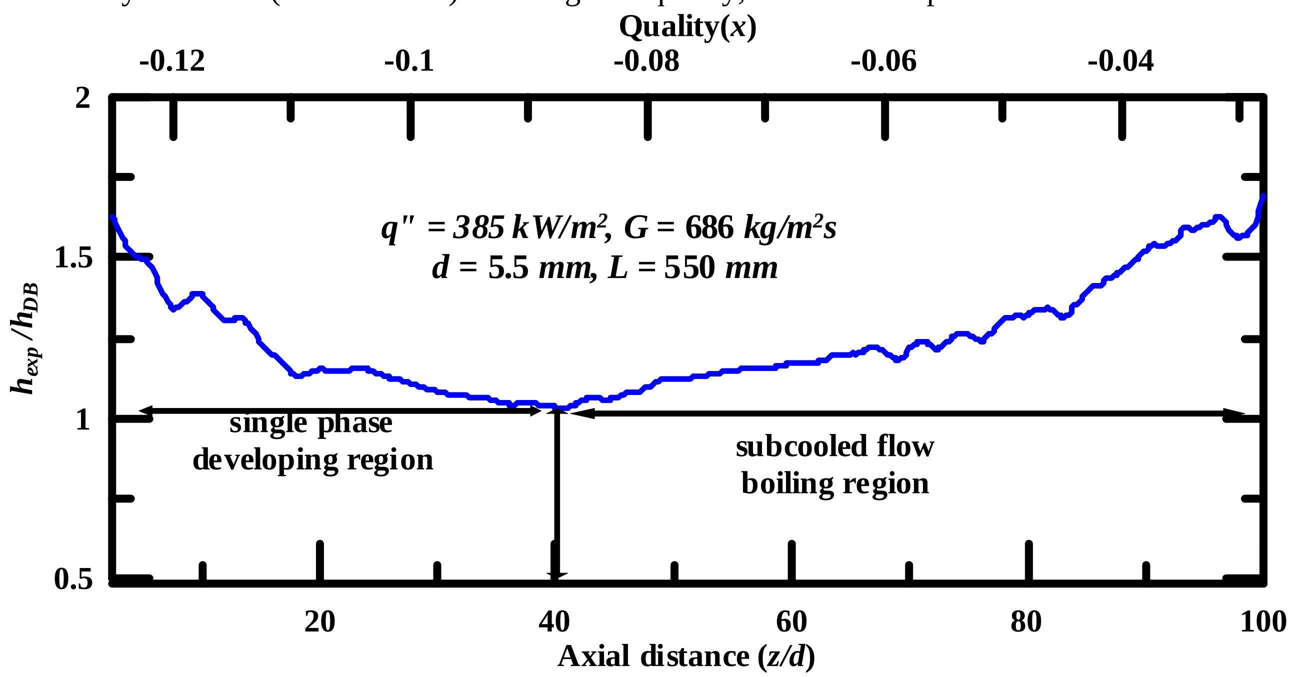

Figure 5 Variation of heat transfer coefficient along the length of the test section with single phase developing region

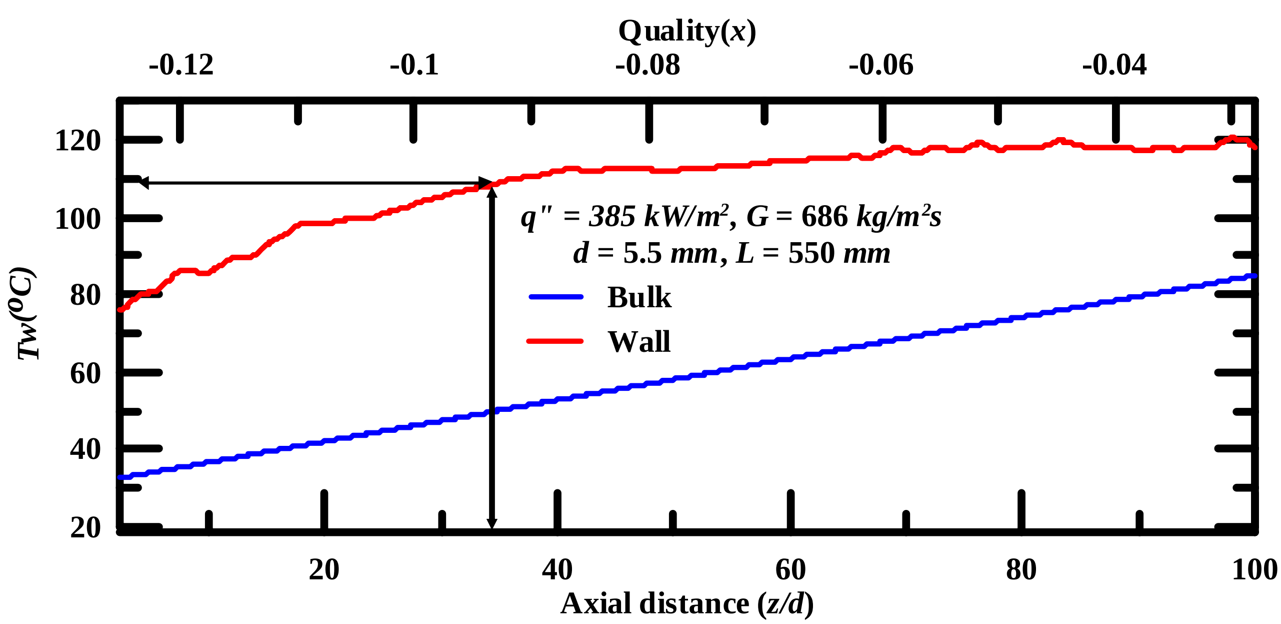

Figure 6 Variation of wall temperature along the length of the test section with single-phase developing region

increase in the heat flux. However, the fluid temperature remains constant for a given quality for different heat fluxes. Increase in wall temperature results in increased amount of nucleation and hence heat transfer coefficient increases with increase in heat flux.

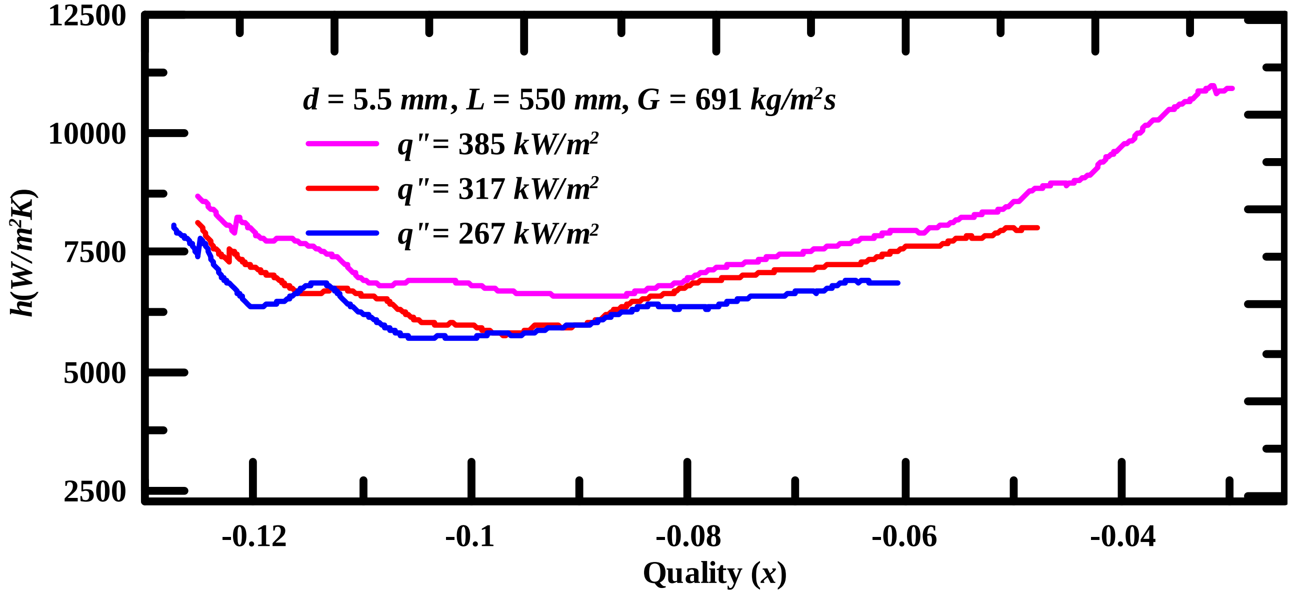

Figure 7 Variation of heat transfer coefficient with quality for different heat fluxes for d = 5.5 *mm* test section

3.3.2 Empirical correlation for local heat transfer coeffient in sub-cooled boiling

Boiling number, Jakob number, Prandtl number and density ratio govern the heat transfer in sub-cooled boiling region. Two phase heat transfer coefficient is correlated with boiling number, Jakob number and Prandtl number and the correlation is given in equation 10. Density ratio is not considered in this correlation because the operating pressure in the present study is nearly atmospheric and constant. Sub-cooled heat transfer coefficient is normalized with single-phase heat transfer coefficient estimated using Dittus Boelter correlation.

$$\frac{h_{exp}}{h_{DB}} = 137.63 Bo^{0.762} Ja^{-0.533} Pr^{0.267} \tag{10}$$

Properties of the fluid are estimated at local bulk fluid temperature. The results show reasonably good agreement with the experimental data with an rms deviation of 8.2%.

Table 3 lists the comparison of local heat transfer coefficient prediction for the present data by the existing correlations. The estimated heat transfer coefficient using the correlation suggested by Papel [16], Badiuzzaman [17] and Moles and Shaw [18] show very large root mean square (rms) deviation. Correlation by Liu and Winterton [19] predicted the experimental data with an rms deviation of 8.9%. Since few outliers are found above 40% deviation range, the correlation is modified by adjusting the suppression factor S (S=1). This modified Liu and Winterton [19] correlation predicted the experimental data with an rms deviation of 10% with outliers of 2.1% above 20% deviation range.

3.4 Reynolds analogy

Using the present data of sub-cooled boiling pressure drop and heat transfer coefficient, the applicability of Reynolds analogy in predicting the friction factor from heat transfer coefficient is investigated. From the local heat transfer coefficient data (including the single phase and two phase

Table 3 Comparison of predicted sub-cooled boiling heat transfer coefficient using various existing correlations with the experimental data

Correlation	Average error(%)	rms Error	Data Outliers (%)		
			20%	30%	40%
Present correlation	-0.33	8.2	0.55	0.08	-
Liu and Winterton [19]	1.3	8.9	2.9	0.4	0.01
Modified Winterton	-5.1	9.9	2.1	-	-
Papell [16]	-30.4	43.8	62	45	32
Moles and Shaw [18]	-65.7	67.8	99	98	4
Badiuzzaman [17]	-182	184			100

region), average heat transfer coefficient for the test section is obtained and hence the Stanton number St is calculated using the relation given by

$$St = \frac{Nu}{Re\,Pr} \tag{11}$$

A correlation between friction factor f and Stanton number St is obtained using the present data and is given by

$$St = \frac{f}{60.2} + 0.0026 \tag{12}$$

The correlation predicts the present data with a maximum deviation of ± 30% and rms deviation of 18%. The variation from the Reynolds analogy $f = 8 \times St$ may be attributed to the following reasons

- The Prandtl number of the fluid varies from 2.3 - 3.5 instead of the assumption of $Pr = 1$
- The pressure drop across the test section includes the pressure drop in the hydrodynamic developing length of the test section where the pressure gradient may be larger.
- The thermal developing region of the test section has significant effect in the heat transfer coefficient of that section.

4. Conclusions

Sub-cooled boiling pressure drop and heat transfer coefficient are measured in horizontal tubes under low pressure and low flow conditions. Local wall temperature measurement is carried out using thermal image technique and hence the local heat transfer coefficient is estimated.

The correlation for single-phase diabatic friction factor predicts the data with an rms 21%. A correlation developed for two phase sub-cooled boiling pressure drop satisfies the experimental data with an rms deviation of 17%.

An empirical correlation developed for single-phase heat transfer coefficient in the developing region of the heated test section predicts the experimental data with an rms deviation of 7.4%. A correlation for heat transfer coefficient in the sub-cooled boiling region as a function of Bo, Ja and Pr is developed and it predicts the experimental data with an rms deviation of 8.2%.

A correlation similar to Reynolds analogy is developed as a function of friction factor and Stanton number with a maximum deviation of ± 30%. The difference from the Reynolds analogy may be attributed to the variation in Prandtl number and the entrance effect in the data measurement for both the pressure drop and heat transfer coefficient.

5. References

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6. NOMENCLATURE

Bo	Boiling number ($\frac{q''}{Gh_{fg}}$)
Cp	Specific heat ($J/kg.K$)
d	Inner diameter (mm)
f	Friction factor
G	Mass flux (kg/m^2s)
h	Heat transfer coefficient ($W/m^2.K$)
h_{fg}	Latent heat of vaporization (J/kg)
Ja	Jakob number ($\frac{Cp \times \Delta T_{sub}}{h_{fg}}$)
k	Thermal conductivity ($W/m.K$)
L	Length (m)
Nu	Nusselt number (hd/k)
Pr	Prandtl number ($\mu Cp/k$)
q''	Heat flux (W/m^2)
Re	Reynolds number (Gd/μ)
St	Stanton number ($Nu/Re.Pr$)
T	Temperature (°C)
z	Axial distance (m)

Greek symbols

ρ	Density (kg/m^3)
μ	Absolute viscosity ($Pa.s$)
σ	Surface tension (N/m)
ΔP	Pressure drop(Pa)
ΔT_{sub}	Degree of subcooling (T_{sat} - T_i)

subscripts

a	Adiabatic
DB	Dittus Boelter
exp	Experiment
l	Liquid
sat	Saturated
sb	Sub-cooled boiling
w	Wall
1ph	Adiabatic single phase

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