Proceeding of the Korean Nuclear Society Autumn Meeting
Taejon, Korea, October 2000

Interfacial Condensation Heat Transfer for Countercurrent Steam-Water Wavy Flow in a Horizontal Circular Pipe

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ABSTRACT

An experimental study of interfacial condensation heat transfer has been performed for countercurrent steam-water wavy flow in a horizontal circular pipe. A total of 105 local interfacial condensation heat transfer coefficients have been obtained for various combinations of test parameters. Two empirical Nusselt number correlations were developed and parametric effects of steam and water flow rates and the degree of water subcooling on the condensation heat transfer were examined. For the wavy interface condition, the local Nusselt number is more strongly sensitive to the steam Reynolds number than water Reynolds number as opposed to the case of smooth interface condition. Comparisons of the present circular pipe data with existing correlations showed that existing correlations developed for rectangular channels are not directly applicable to a horizontal circular pipe flow.

1. INTRODUCTION

The direct-contact condensation heat transfer in a steam-water stratified flow is important in safety analysis of nuclear reactor systems. In the postulated loss-of-coolant-accident (LOCA) due to a break of cold leg in a pressurized water reactor (PWR), emergency core cooling (ECC) water would be injected into the reactor vessel through the cold legs to prevent overheating of the reactor core. Therefore, The subcooled ECC water is brought into direct contact with escaping steam from the reactor core resulting in a countercurrent flow of steam and subcooled ECC water. In addition, the modeling of direct-contact condensation in a countercurrent steam-water stratified flow is important in analysis of water hammer at steam generator feedwater line. The system behavior in each case is highly dependent upon the local condensation rates.

Although a number of studies have already been carried out for more than 30 years both experimentally and theoretically on concurrent (Linehan et al., 1970; Lim et al., 1984) and countercurrent (Segev et al., 1981; Kim et al., 1985; Ruile, 1995) stratified flow, most of them were performed using a rectangular channel having large aspect ratio and performed under very thin water layer thickness. But most of the nuclear piping systems have a circular geometry rather than rectangular. Therefore, due to the great difference in flow channel geometry which can affect the interfacial condensation heat transfer phenomenon, it is doubtful whether existing correlations can be applicable to the nuclear piping systems directly.

The main purpose of present work is to evaluate the condensation heat transfer for countercurrent steam-water stratified-wavy flow in a horizontal circular pipe and to investigate the parametric effects of steam and water flow rate, and degree of water subcooling on condensation heat transfer.

2. EXPERIMENTS

A series of experiments were performed and a total 105 experimental data of local interfacial condensation heat transfer coefficient have been obtained at atmospheric pressure, varying the following test parameters: (1) inlet water flow rate, (2) inlet steam flow rate, and (3) degree of inlet water subcooling.

2.1 Experimental Apparatus

A schematic diagram of the experimental apparatus is shown in Fig. 1. The experimental loop is designed to simulate the countercurrent steam-water stratified pipe flow. The main components of the system are; (a) the test section, (b) the steam supply system, (c) the water supply system, and (d) the data acquisition system.

The test section consists of five transparent Pyrex pipes to enable visual observation of the flow pattern. The test section has inner diameter of 84 mm and length of 2 m. Each pipe is connected in series by flange where pitot-tube and thermocouple are attached to the traversing digital caliper as shown in Fig. 2. To prevent the undesired contact of steam with water in the reservoirs, flow guide is mounted in the reservoirs.

Water is supplied by adjustable centrifugal pump from water surge tanks to the test section. The water surge tanks are used to provide a steady flow rate and a constant temperature of water that flows into the test section. The water flow rate is controlled by ball valve and measured by turbine flow meter. These components make it possible to supply steady flow of constant temperature water into the test section.

Steam supplied by 200 kW electric steam boiler, passes through two steam-water separators and a flow rate measurement unit into the test section. The steam flow rate is controlled by ball valve and measured by vortex flow meter. The two steam-water separators assure dry saturated or slightly superheated steam supply.

2.2 Test Parameters and Test Procedure

The controllable test parameters were (1) water flow rate, (2) steam flow rate, and (3) inlet water subcooling. A total of 105 data have been obtained for various combinations of inlet water and steam flow rates at three different inlet water temperatures of 20, 40, and 55 °C under atmospheric pressure as summarized in Table 1. The ranges of water and steam flow rates are 0.083~0.217 kg/s and 0.013 ~ 0.02 kg/s, respectively. The water Prandtl number and Jakob number varied from 2.07 to 4.15 and from 43 to 180, respectively.

To measure the water velocity and temperature near the interface as much as possible, a very small diameter pitot-tubes (O.D.=1.6 mm) and thermocouples (O.D.=0.2 mm) were used. The pitot tubes and thermocouples were attached to a digital vernier caliper-mounted traversing system such that both pitot tubes and thermocouples (whose tip was bent at 90°) could be raised vertically by an increment of 1 mm.

The experimental procedure is as follows: (a) set the inlet water temperature to the desired value, (b) set the inlet water and steam flow rates to the predetermined values and wait until the flow becomes steady, (c) measuring the water velocity and temperature profile from bottom of water layer to the steam-water interface with small steps at five axial positions, (d) evaluating the local water bulk temperature and the local heat transfer coefficient, (f) repeating the above procedure for different experimental conditions such as inlet water temperature, water and steam flow rates.

3. ANALYSIS

The control volume describing the heat transfer process in a countercurrent steam-water stratified pipe flow is shown in Fig. 3. To evaluate the condensation heat transfer coefficient, following assumptions are made for the establishment of energy balance and evaluation of water bulk temperature in the test section.

(a) The properties of steam along the test section are constant. Actually, the temperature drop between the steam inlet and the outlet is less than 1 °C.

(b) The heat transfer from the steam and the water side wall to the atmosphere is negligible. Actually, it was shown that the steam condensation rate at the pipe wall is much less than the total steam condensation rate at the steam-water interface from the result of Chun et al (1999).

(c) The water temperature profile across the cross-section (in z-direction) at each vertical point is uniform. It was shown that the temperature difference along z-direction is negligible from the result of Chun et al (1999).

(d) The water velocity profile across the cross-section at each vertical point has the 1/7 power velocity distribution. The assumption of laminar flow profile instead of the 1/7 power velocity profile does not change the water bulk temperature.

Based on the mass and energy balance on the control volume, the interfacial condensation heat transfer coefficient at any location x from the water inlet can be expressed as:

$$h(x) = \frac{i_{fg}}{S_i (T_g - T_f(x))} \frac{d\dot{m}_f}{dx} \tag{1}$$

Thus, the interfacial condensation heat transfer coefficient is deduced from the measured water bulk temperature, water layer thickness, and the rate of increase in local water flow rate along the axial direction of test section.

By using the relations of mass and energy balance on the control volume, the mass flow rate of water, at a any position x, can be expressed as:

$$\dot{m}_f(x) = \frac{\dot{m}_{f\,in}(i_g - i_{f\,in}) + \int_0^x q''_{atm} S_g dx}{i_g - i_f(x)} \tag{2}$$

In Eq. (2), the local water flow rate is evaluated from the local water bulk temperature and the wall condensation can be neglected in the calculation of interfacial condensation heat transfer coefficient in the present work by assumption (b). The local water bulk temperature is defined as follows:

$$T_f(x) = \frac{\int \boldsymbol{r}_f(x,y,z) T_f(x,y,z) V_f(x,y,z) dydz}{\int \boldsymbol{r}_f(x,y,z) V_f(x,y,z) dydz} \tag{3}$$

To evaluate the local water bulk temperature, the water layer is divided into several rectangles under assumption (c) and the 1/7 power velocity profile was used. The nodalization and coordinate of water layer is shown Fig. 4.

The definitions of dimensionless parameters in the present work are as follows:

$$\mathrm{Nu} = \frac{h\, D_{h,f}}{k_f}, \qquad \overline{\mathrm{Nu}} = \frac{\bar{h}\, \overline{D_{h,f}}}{\overline{k_f}}$$

$$\mathrm{Re}_f = \frac{\rho_f V_f D_{h,f}}{\mu_f}, \qquad \mathrm{Re}_g = \frac{\rho_g V_g D_{h,g}}{\mu_g}$$

$$\mathrm{Pr} = \frac{C_{p,f}\, \mu_f}{k_f}, \qquad \mathrm{Ja} = \frac{\rho_f C_{p,f} (T_g - T_f)}{\rho_g i_{fg}}$$

$$D_{h,f} = \frac{4A_{p,f}}{S_i + S_f}, \qquad D_{h,g} = \frac{4A_{p,g}}{S_i + S_g} \tag{4}$$

4. RESULTS AND DISCUSSION

4.1 Local Water Temperature and Velocity Profiles

The water layer thickness in the present experiments varied from 12.5 to 27.5 mm which was much greater than that of previous works with a wide rectangular channel. The flow pattern was determined by visual observations as shown in Fig. 5.

Figures 6 and 7 show typical profiles of local water temperature and velocity for countercurrent steam-water flow. To compare the heat transfer characteristics according to the flow pattern, the experimental data has been obtained for smooth interface condition as well as wavy interface condition.

Figure 6 shows the profiles of local water temperature and velocity for smooth interface condition. The local velocity of water layer reaches a high velocity region at upper water layer and remains more or less constant. The local temperatures of the water layer close to the bottom (i.e., $y/\delta < 0.6$) is just slightly higher than the inlet water temperature and remains fairly constant, and in the higher water layer region (i.e., $y/\delta > 0.8$) the water layer temperature tends to rise sharply to the saturation temperature. If there were a full turbulent thermal mixing in the water layer, the temperature profile should have been almost uniform (i.e., a straight vertical line). The curves in Fig. 6(a) implies that the thick water layer (12.5~27.5 mm) used in the present work prevented from the occurrence of full turbulent thermal mixing. That is, the turbulence generated by the interfacial shear stress did not propagated into the lower water velocity region and effective thermal mixing was restricted within the upper water layer close to steam-water interface.

As opposed to the smooth interface case, for wavy interface condition, the local velocity of water layer reaches maximum at lower water layer region and significantly slowed down as it moves toward the interface due to enhanced shear stress of countercurrent steam flow. The local temperature of water layer is not constant in the water layer but rather it rises more or less gradually from the bottom to the interface as shown in Fig. 7.

Finally, from Figs. 6 and 7, it can be seen that there is an appreciable difference in heat transfer characteristics between the smooth and the wavy interface.

4.2 Parametric Effect of Flow Rates and Water Subcooling

Figures 8 and 9 show the effects of water and steam flow rates and inlet water subcooling on the interfacial condensation heat transfer. As shown in Fig. 8, for a given steam Reynolds number (or steam flow rate), the Nusselt number (or h) increases with the water Reynolds number (or water flow rate). Also, for a given water Reynolds number, the Nusselt number increases with the steam Reynolds number. This is due to the fact that the increase in water flow rate increases initial turbulence and the increase in steam flow rate also increases interfacial shear stress. However, the local Nusselt number in the wavy flow is more sensitive to the steam Reynolds number than the water Reynolds number as apposed to the case of smooth interface condition.

Although it is difficult to evaluate the effect of water subcooling directly from Figs. 8 and 9 due to the lack of experimental data, the effect of water subcooling is, in general, appeciably significant. That is, for the same steam and water Reynolds number, the heat transfer coefficient increases as the water bulk temperature decreases (i.e., as the water subcooling increases).

4.3 Development of Empirical Correlation

Based on the present experimental data, two types of empirical correlations for countercurrent steam-water wavy flow have been developed by least-square-fit method.

The local Nusselt number can be considered to be principally a function of the local steam and water Reynolds numbers, and possibly water Prandtl number, since the thermal resistance lies principally on the water side. The correlation is as follows:

$$\mathrm{Nu} = 1.4\times10^{-6}\ \mathrm{Re}_f^{\ 0.63}\ \mathrm{Re}_g^{\ 1.15}\ \mathrm{Pr}^{1.44} \tag{5}$$

In addition, to show the effect of water subcooling more explicitly, Jacob number can be used instead of Prandtl number. The result is as follows:

$$\mathrm{Nu} = 1.2\times10^{-7}\ \mathrm{Re}_f^{\ 0.59}\ \mathrm{Re}_g^{\ 1.2}\ \mathrm{Ja}^{0.82} \tag{6}$$

From Eqs. (5) and (6), it can be seen that the local Nusselt number is more strongly sensitive to the steam Reynolds number than water Reynolds. However, the effect of water subcooling is similar both for smooth (Chun et al., 1999) and wavy interfaces. The comparison between measured Nusselt number and calculated Nusselt number is shown in Figs. 10 and 11. Most of data (about 90%) agree well with the calculated values within ± 30%.

4.4 Comparison of the Present Data and Existing Correlation

The comparisons of the present experimental data with the existing correlations of Kim, H. J. (1983) and Lim et al. (1985) for wavy interface are shown in Figs. 12 and 13. Kim carried out experiments for countercurrent steam-water flow in 4 ~ 87° inclined rectangular channel having the aspect ratio of 5 and Lim et al. performed for horizontal concurrent steam-water flow in rectangular channel having the aspect ratio of 5.8. To maintain the consistency in the comparison, the dimensionless numbers defined in Eq. (4) are converted to follow Kim's and Lim's definitions, respectively.

Figures 12 and 13 show that there is a large disagreement between the present experimental data and the correlations of Kim's and Lim's. The main reason for this discrepancy can be attributable to the differences in the water layer thickness and the flow channel geometry. However, it is difficult to quantify the effects of water layer thickness and flow channel geometry, because there are other differences in experimental range and conditions such as flow direction and inclination.

4.5 Uncertainty Analysis

The uncertainty analysis for the local heat transfer coefficients has been carried out by an error propagation method. The uncertainty of interfacial heat transfer coefficients (s_h) is computed by Root-Sum-Square (R.S.S.) method of bias limit (s_B) and precision limit (s_P) as follows:

$$s_h = \sqrt{s_B^2 + s_P^2} \qquad (7)$$

The uncertainty of the interfacial condensation heat transfer coefficients is in the range of 12.02 ~ 45.68 % and slightly decreases as the heat transfer coefficient increase. The uncertainty is mainly due to the error of water bulk temperature that results from the error produced in the determination of water layer thickness. The detailed results are summarized in Table 2.

5. CONCLUSIONS

The interfacial condensation heat transfer for countercurrent steam-water wavy flow in a horizontal circular pipe has been experimentally investigated. The main conclusions of the present work are as follows:

(a) In case of wavy flow, comparing with stratified smooth flow, it can be seen that the turbulence generated by the interfacial shear stress propagated into lower region of water layer and the thermal mixing in the water layer was more efficient.

(b) The effects of water and steam flow rates, and inlet water subcooling were examined. The local Nusselt number is more strongly sensitive to the steam Reynolds number than water Reynolds number in contrast to countercurrent stratified smooth flow.

(c) From the total 105 data of local interfacial condensation heat transfer coefficient, two Nusselt number correlations were developed and most of data (about 90%) agree with the correlations within ± 30%.

(d) Comparisons of the present experimental data with two existing correlations of Kim, H. J. (1983) and Lim et al. (1985) showed that there is a great difference between the present data and two existing correlations. The main reason for this discrepancy can be attributable to the differences in the experimental conditions and the water layer thickness due to the geometry of flow channel.

(e) The uncertainty of local interfacial condensation heat transfer coefficient is in the range of 12.02 ~ 45.68 % and the uncertainty is mainly due to the error of water bulk temperature that results from the error produced in the determination of water layer thickness.

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Table 1 Test Matrix of the Present Experiment

Water Inlet Temperature (°C)	Water Flow Rate (LPM)	Steam Flow Rate (kg/s)	Re_f	Re_g	No. of Data
25	5 ~ 13 (5 Cases)	0.013 ~ 0.019 (6 Cases)	4000 ~ 10000	12000 ~ 19000	41
45	5 ~ 13 (5 Cases)	0.015 ~ 0.018 (5 Cases)	4000 ~ 13000	14000 ~ 21000	31
55	5 ~ 13 (5 Cases)	0.015 ~ 0.02 (4 Cases)	4000 ~ 14000	17000 ~ 23000	33
Total Number of Experimental Data					**105**

Table 2 Uncertainty of Local Heat Transfer Coefficients

	Parameter	Bias Limit	Precision Limit
Independent Parameter	Water Layer Thickness	1.0 mm	0.5 mm
	Inlet Water Flow Rate	0.5 %	0.5 %
	Local Water Velocity	5 %	0.5 %
	Local Water & Steam Temperature	2.2 °C	0.1 °C
Dependent Parameter	Water Bulk temperature	0.84 ~ 2.15 °C	0.37 ~ 1.35 °C
	Local Water Flow Rate	0.51 ~ 0.72 %	0.51 ~ 0.55 %
	Local Heat Transfer Coefficient, h	10.4 ~ 39.9 %	6.03 ~22.2 %
Uncertainty of Local Heat Transfer Coefficient, h (Root Sum Square of Bias Limit and Precision Limit)		**12.02 ~ 45.68 %**	

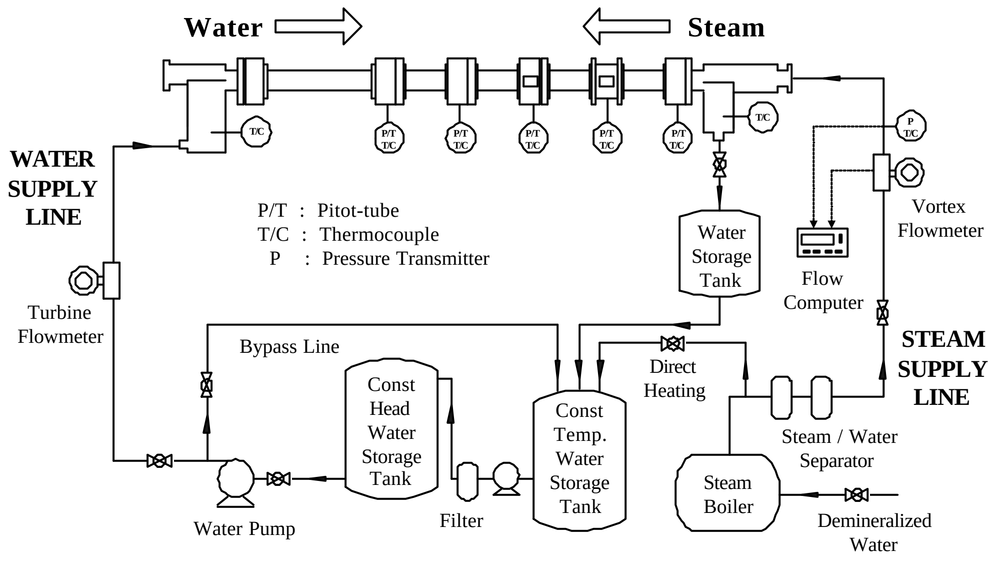


Fig. 1 Schematic Diagram of Experimental Apparatus

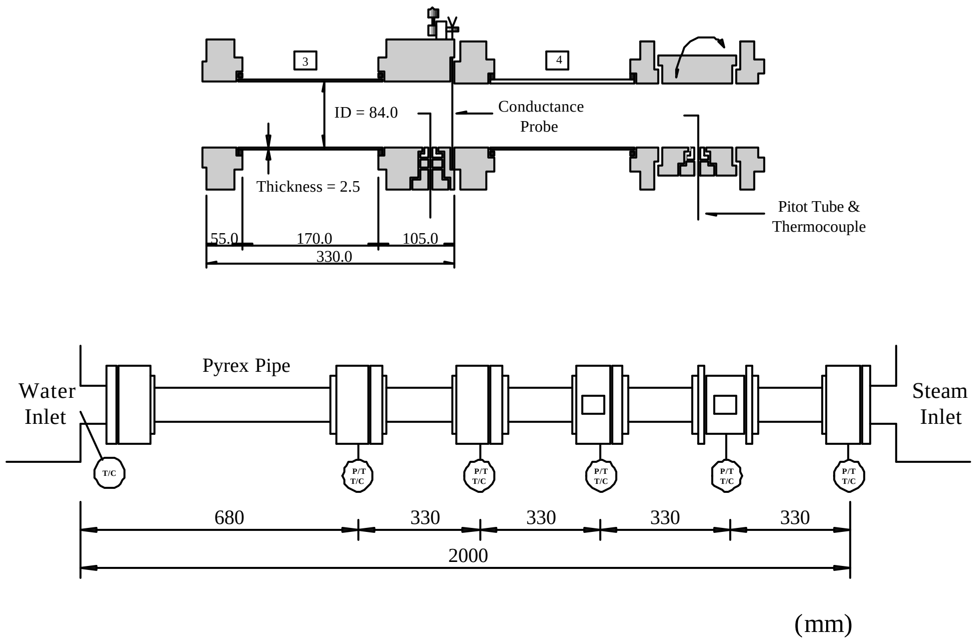


Fig. 2 Schematic Diagram of Test Section

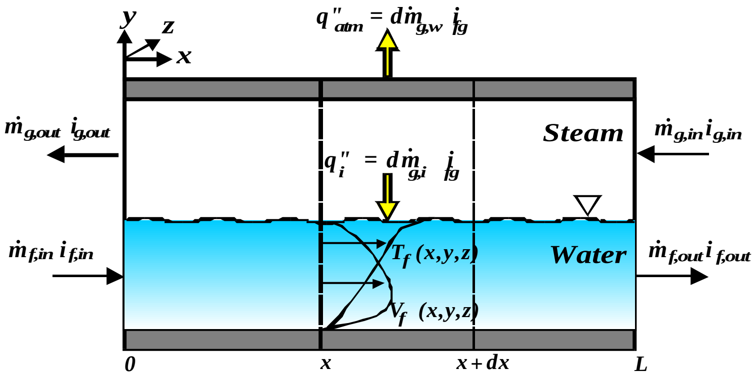


Fig. 3 Control Volume of Countercurrent Steam-Water Stratified Flow

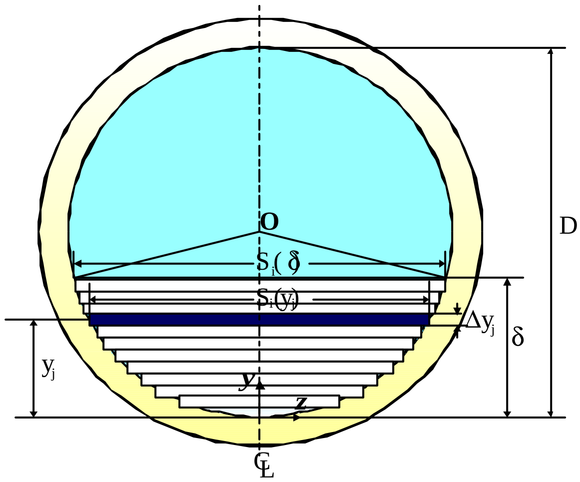
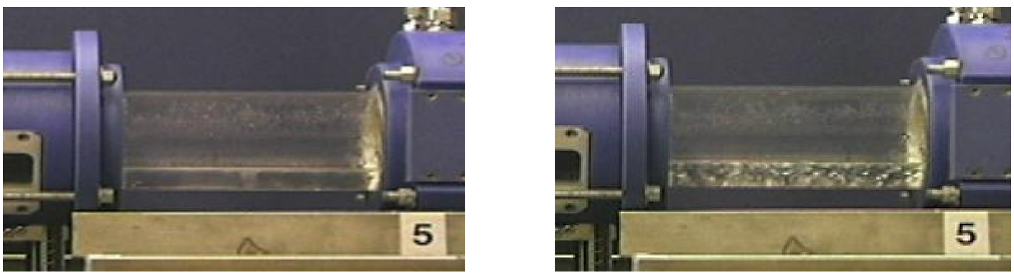


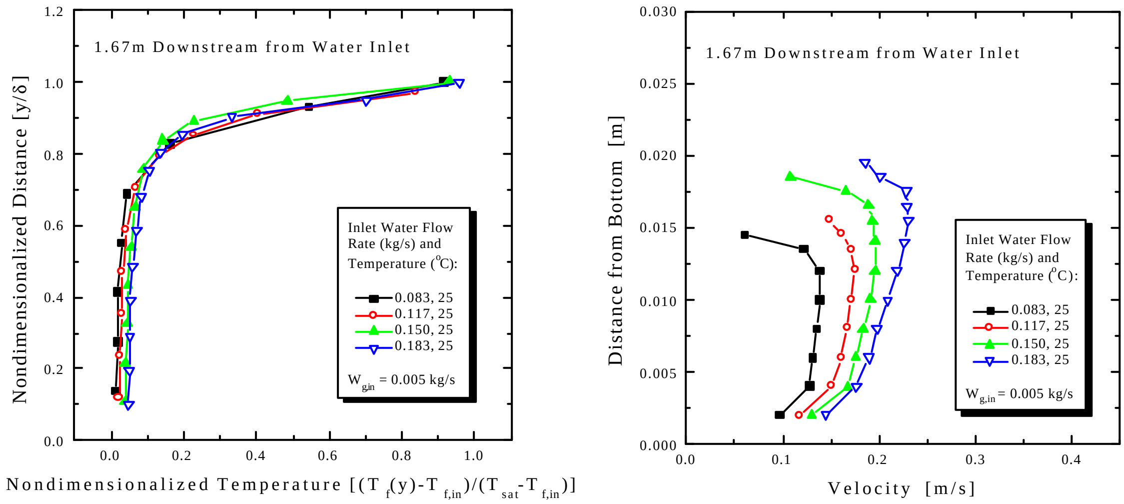
Fig. 4 Nodalization for the Calculation of Water Bulk Temperature



(a) Smooth Interface

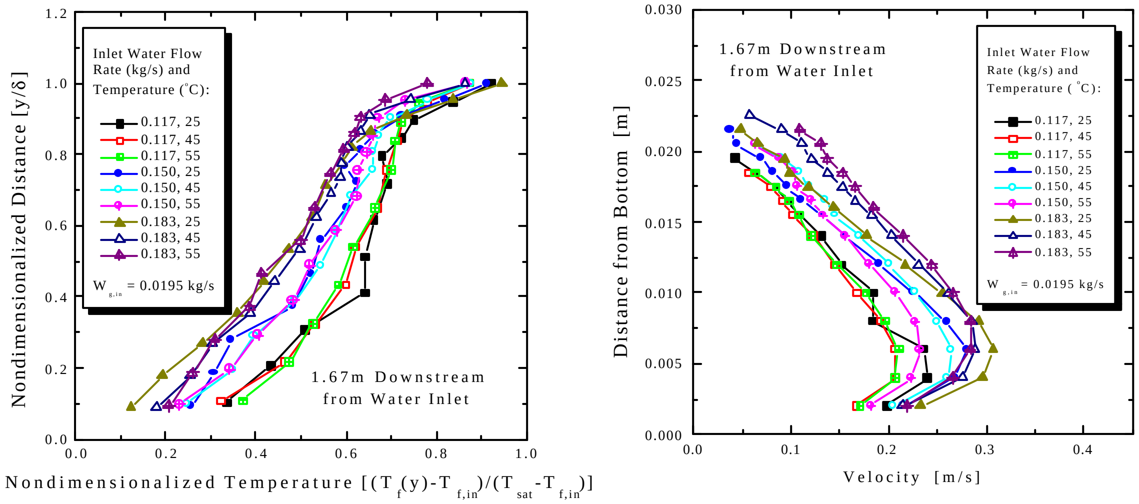
(b) Wavy Interface

Fig. 5 Flow Pattern



(a) (b)

Fig. 6 Local Temperature and Velocity Profiles at Smooth Interface



(a) (b)

Fig. 7 Local Temperature and Velocity Profiles at Wavy Interface

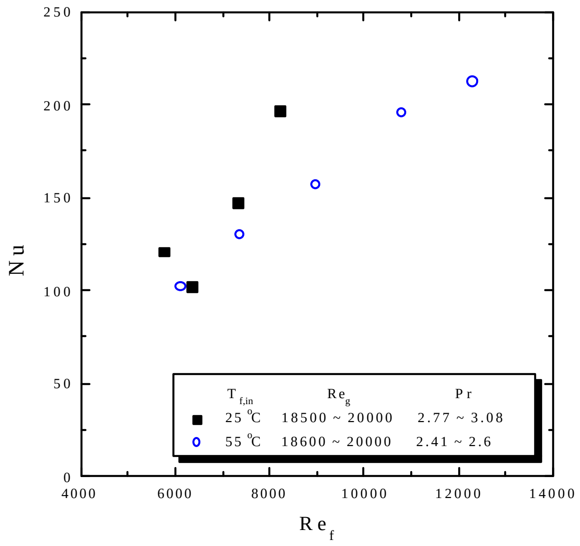


Fig. 8 Effect of Water Flow Rates on the Interfacial Condensation Heat Transfer

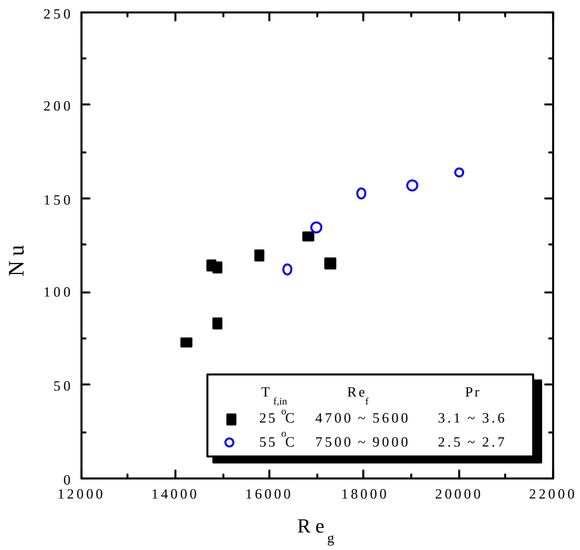


Fig. 9 Effect of Steam Flow Rates on the Interfacial Condensation Heat Transfer

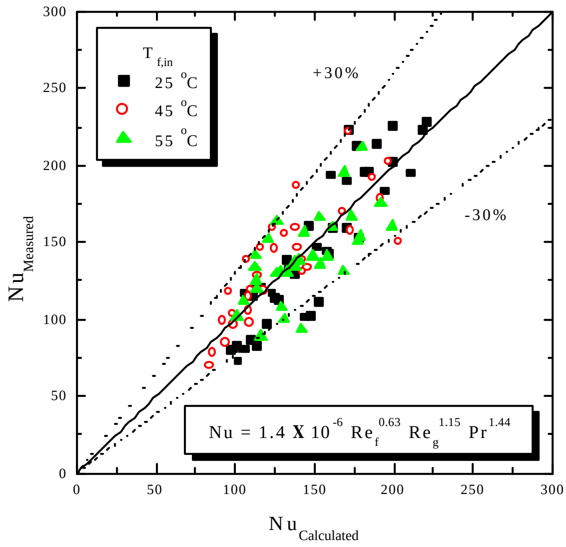


Fig. 10 Comparison of Measured Nusselt Number with the Calculated Value (Eq. 5)

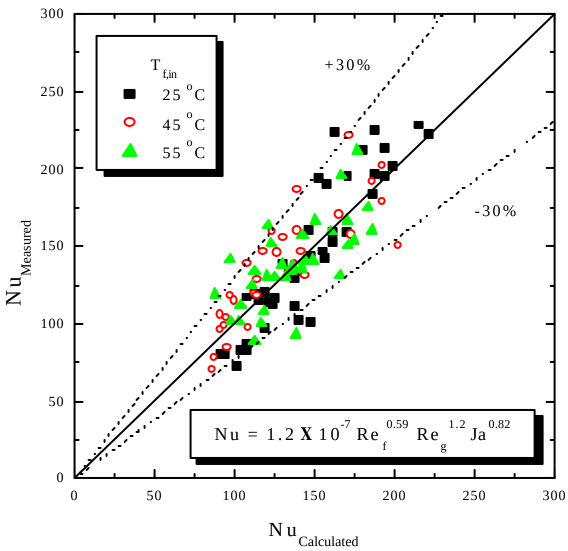


Fig. 11 Comparison of Measured Nusselt Number with the Calculated Value (Eq. 6)

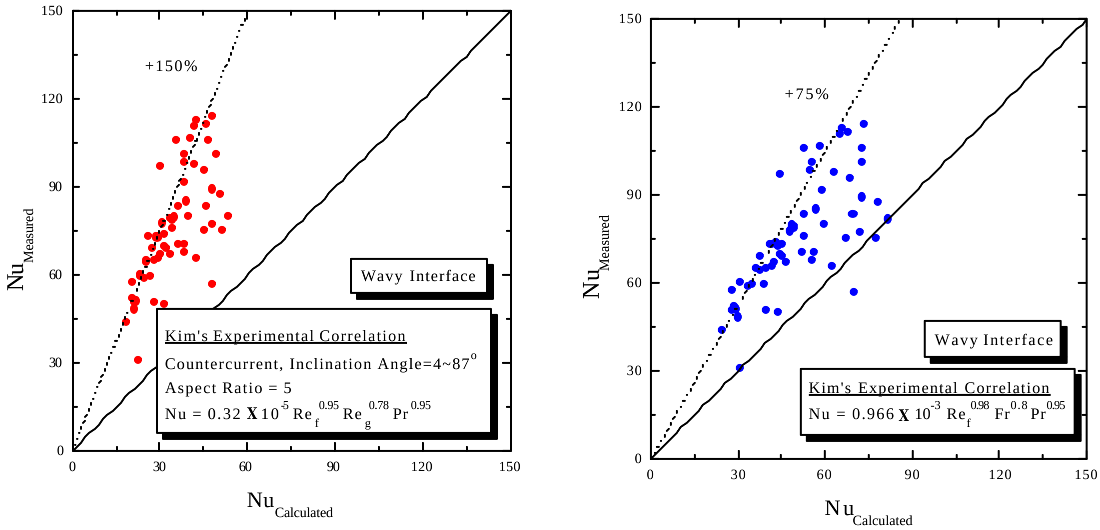


Fig. 12 Comparison of the Present Data with Kim's Correlations

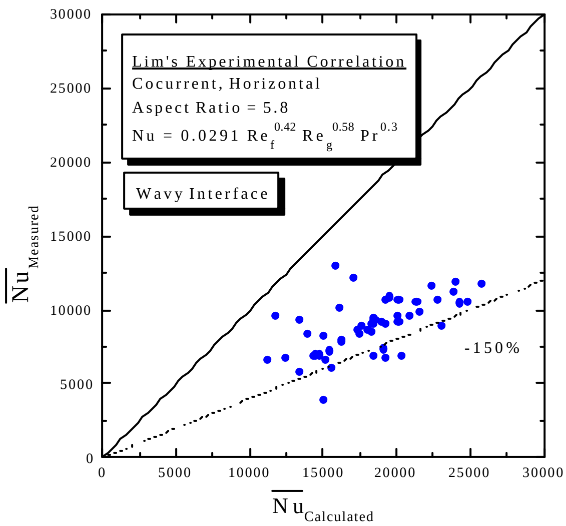


Fig. 11 Comparison of the Present Data with Lim's Correlation