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An Experimental Investigation of the Interfacial Condensation Heat Transfer in Steam/Water Countercurrent Stratified Flow in a Horizontal Pipe

In-Cheol Chu, Seon-Oh Yu, and Moon-Hyun Chun

Korea Advanced Institute of Science and Technology

Byong-Sup Kim, Yang-Seok Kim, In-Hwan Kim, and Sang-Won Lee

Korea Electric Power Research Institute

Abstract

An interfacial condensation heat transfer phenomenon in a steam/water countercurrent stratified flow in a nearly horizontal pipe has been experimentally investigated. The present study has been focused on the measurement of the temperature and velocity distributions within the water layer. In particular, the water layer thickness used in the present work is large enough so that the turbulent mixing is limited and the thermal stratification is established. As a result, the thermal resistance of the water layer to the condensation heat transfer is increased significantly. An empirical correlation of the interfacial condensation heat transfer has been developed. The present correlation agrees with the data within ±15%.

1. Introduction

The interfacial condensation heat transfer in steam/water stratified flow has been studied by a number of investigators [1-5] in the past 28 years. However, most of the experimental studies were conducted using a rectangular channel where the ratio of the width to the height was about five. Thus, a very shallow water layer condition was established, i.e., about 10 mm in Lim et al.'s study and varied values from 1mm to 7mm in the case of other investigators. This enabled to obtain a homogeneous turbulent mixing in the water layer. The condensation heat transfer coefficient was deduced from the local steam condensation rate by measuring the steam mass flow rate [3, 4] and also from the local enthalpy increase rate of water by measuring the water temperature at the bottom [1, 2]. The former method can be used for any water layer condition (thick, shallow, laminar or turbulent). The latter method, on the other hand, can be used only for water films or for highly turbulent water flows where homogeneous mixing is guaranteed. But both methods give no information about turbulent transport characteristics of the water flow.

It should be noted, however, that most of the piping used in the Nuclear Power Plant is horizontal and circular. Therefore, the flow geometry used by previous investigators is not directly applicable to the nuclear piping system. When the area of the flow cross-section of a wide rectangular channel and that of a circular channel are the same, the water layer thickness of the circular channel will be much greater than that of the wide rectangular channel for given flow rates of steam and water. In particular, when the water layer thickness is increased, the heat, mass and momentum transport characteristics, such as the turbulent intensity and/or the distribution of turbulent mixing which plays a major role in the interfacial condensation heat transfer, is changed. That is, the difference in flow geometry will change the overall interfacial condensation heat transfer characteristics. In fact, this was proved by Ruile's study [5] where the water layer was about 40 mm thick in the rectangular channel. Ruile [5] found that thermal stratification and limited turbulent mixing length in the water layer handicapped the vertical transport mechanism and degraded interfacial heat and mass transfer property. In the present experimental study, the interfacial condensation heat transfer in nearly horizontal pipe and the water flow characteristics have been investigated.

2. Experiments

The present experiment has been designed to simulate a steam/water countercurrent stratified flow in the nearly horizontal pipe. As shown in Fig. 1, the experimental set-up consists of a test section, steam and water supply systems. The test section has a downward inclination of 0.2° from the water inlet. The test section is consituted of four transparent tempered glass pipes, and these pipes are connected by flanges where traversible pitot tube and thermocouple are installed at the bottom of the flange as shown in Fig. 2. The total length and inner diameter of the channel are approximately 2.1 m and 0.083 m, respectively. The steam flows into the test section from the 200 kW electric steam boiler through the steam/water separator and a set of steam mass flow measurement system. The separator is used to ensure the supply of dry saturated or slightly superheated steam. The volumetric flow rate of steam is measured by a vortex flowmeter. The density of steam is estimated by the temperature and the pressure measured by thermocouples and an absolute pressure transducer. The volumetric flow rate of water flowing through the test section is measured by calibrated rotameters.

A series of experiments have been carried out by varying the inlet water and steam flow rates. Three different inlet water temperatures of 20, 40, 55°C were used at atmospheric pressure conditions. The water Reynolds number ranged from 1,800 to 14,300, whereas the steam Reynolds number varied from 12,800 to 46,800. The Prandtl number for water varied from 2.50 to 5.60.

In the present study, the condensation rate of steam at the interface is deduced from the measurement of the rate of increase in bulk water temperature due to the steam condensation. To evaluate the rate of increase in bulk water temperature along the flow stream, the vertical(radial) temperature distribution and velocity profiles of water layer at four different positions are simultaneously measured by traversing the pitot tube and the thermocouple from the bottom of the pipe to the steam/water interface (in y-direction). The locations of the four pitot tubes and thermocouples are approximately 0.6 m, 1.1 m, 1.5 m, 2.0 m from the water inlet, respectively. The water layer thickness is determined by visual observation and indication of the subcooling from the thermocouple.

To obtain the interfacial condensation heat transfer coefficient from the measurement, the following assumptions were made:

(1) the steam is saturated and the pressure of the steam is constant along the test section so that thermal properties of the steam along the test section are constant,

(2) the temperature of water in horizontal cross-stream direction (z-direction) at each vertical position (y-direction)

is uniform and the velocity of water follows the 1/7th law profile in horizontal cross-stream direction at each vertical position.

Based on the mass and energy balance equation, the local interfacial condensation heat transfer coefficient can be defined as follows:

$$h(x)=\frac{i_{fg}}{S_i(T_g-T_f)}\frac{dW_f}{dx} \tag{1}$$

Using the mass and energy conservation, the local condensation rate is evaluated from the measured temperatures of the bulk water, condensation rate of steam at the wall, and water layer thickness.

The channel-average heat transfer coefficient is obtained by integrating the local heat transfer coefficient (Eq. 1) over the total channel length $0<x<L$:

$$\bar{h}=\frac{1}{L}\int_0^L h dx \tag{2}$$

The temperature of the bulk water is evaluated by integrating the measured local temperature and velocity profiles over the water layer thickness, and applying the above assumption (2).

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

Because of the difficulty in measuring the velocity near the interface, it is also assumed that the water flow is stagnant just above the interface due to the interfacial shear stress. For example, V_L is considred to be zero just above the interface and the velocity is assumed to decrease linearly just below the interface.

3. Results and Discussion

In the present experiments, a total of 104 data for the interfacial condensation heat transfer coefficient have been obtained. When the experimental data are plotted in the Mandhane flow pattern map, they fall into the transition zone from stratified smooth to stratified wavy. Also, a visual observation shows that there is a smooth or two dimensional wavy interface. The ratio of the water layer thickness to the pipe diameter varied from 0.265 to 0.386 which is much higher than those obtained in the wide rectangular channels.

The steam condensation rate at the wall of the test section and steam reservoir has been evaluated by directly measuring the heat flux to the atmosphere. The results show that the wall condensation rate is negligible compared to the total steam condensation rate.

Figures 3 and 4 show the typical water temperature and velocity profiles measured. As the overall temperature gradient is increased, the turbulent heat transport from the interface to the bulk water is increased. Full turbulent mixing would give almost a straight vertical temperature profile close to the mean temperature of the water layer. But in the present experiments the thick water layer limits the turbulent mixing, i.e., the turbulence generated by the interfacial shear cannot propagate far away from the interface, thus the thermal resistance of water layer to the interfacial condensation heat transfer becomes fairly large. It can also be seen that even for the lowest steam flow rate, the velocity near the interface slows down due to the interfacial shear of the countercurrent steam flow.

In the present experiments, the heat transfer coefficients averaged over the total length of the channel vary from 161.67 to 594.74 W/m^{2}K. The interfacial condensation heat transfer coefficient increases as the water and steam flow rates increase, but it is much more dependent on the water flow rate. Also the heat transfer coefficient has a tendency to increase slightly as the bulk water temperature is increased.

Based on the experimental data for the countercurrent stratified flow in a nearly horizontal pipe with a smooth and a wavy interface and from the least-square fit of the data, an empirical correlation for the channel average condensation heat transfer coefficient has been developed in terms of bulk flow properties. The correlation obtained is as follows:

$$\mathrm{Nu} = 4.31\times10^{-3}\,\mathrm{Re}_f^{1.04}\,\mathrm{Re}_g^{0.26}\,\mathrm{Pr}^{0.97} \qquad (4)$$

The channel average heat transfer coefficient is calculated from Eq. (2). The average Reynolds numbers and Prandtl number are evaluated by averaging the measured values at each station arithmetically. Definitions of the major parameters used in the present correlation are as follows:

$$\mathrm{Nu} = \frac{hD_{h,f}}{k_f},\ \mathrm{Re}_f = \frac{\rho_f V_f D_{h,f}}{\mu_f},\ \mathrm{Re}_g = \frac{\rho_g V_g D_{h,g}}{\mu_g}, \qquad (5)$$

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

A comparison between the predicted Nusselt numbers and the experimentally measured data is shown in Fig. 5, and it can be seen that the agreement is within ±15%. When the present experimental data is compared with the existing correlation for the inclined wide rectangular channel, it can be seen that the measured Nusselt numbers of the present work are much smaller than the values predicted by the existing correlation as can be seen in Fig. 6. The main reason for this is due to the degradation of the turbulent mixing efficiency in the water layer in the present work. However, it is difficult to quantify the degree of the dependency of the condensation heat transfer on the channel geometry due to the difference in experimental conditions and definitions of the length scales that are used in the correlating parameters.

4. Conclusion

The interfacial condensation heat transfer for the steam/water countercurrent stratified flow in nearly horizontal pipe has been investigated experimentally. In the present experimental work, the water layer thickness is much larger than the depth of the interface region where the turbulent mixing generated by the interfacial shear is efficient. Therefore, the thermal resistance of the water layer to the interfacial condensation heat transfer is increased and the overall condensation efficiency is reduced.

Based on the experimental data, a new empirical correlation that is applicable for the interfacial condensation in the nearly horizontal pipe flow with thick water layer (i.e., Eq. 4) has been developed. This correlation agrees with the experimental data within ±15%.

Acknowledgement

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Nomenclature

A_p = cross-sectional flow area

D_h = hydraulic diameter

h = interfacial condensation heat transfer coefficient

i_f = specific enthalpy of water

i_{fg} = latent heat of condensation

S_f = water-side wall perimeter

S_g = steam-side wall perimeter

S_i = interfacial perimeter

$T_f(x)$ = bulk temperature of water

V_f = velocity of water

W = mass flow rate

x = axial distance from water inlet

y = vertical distance from pipe bottom

z = horizontal cross-stream distance from pipe center line

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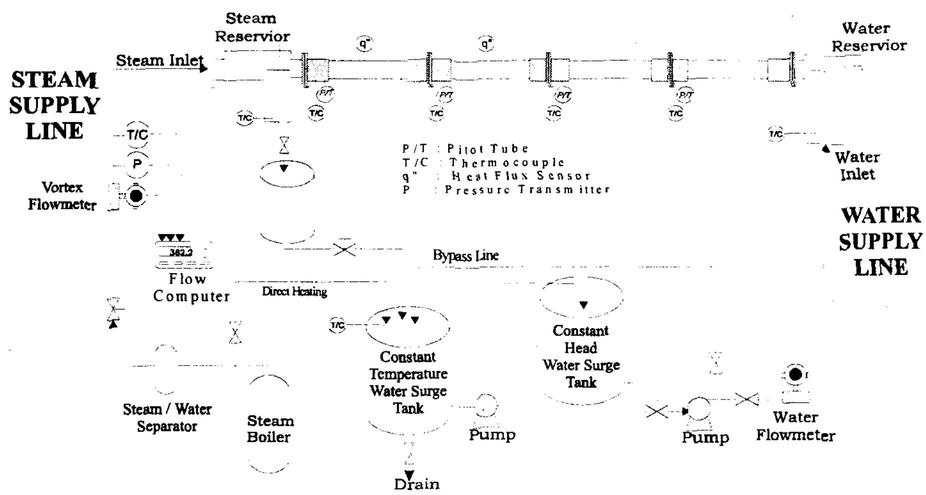

Fig. 1 Schematic diagram of the test facility

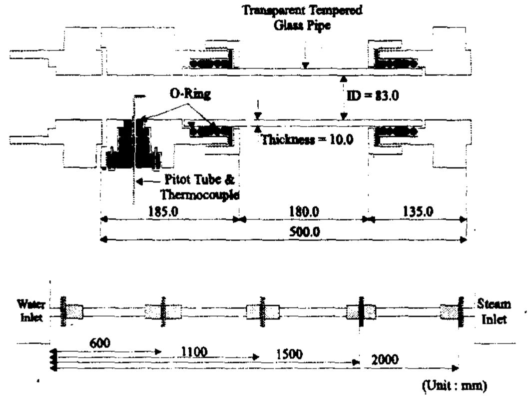

Fig. 2 Schematic diagram of the test section

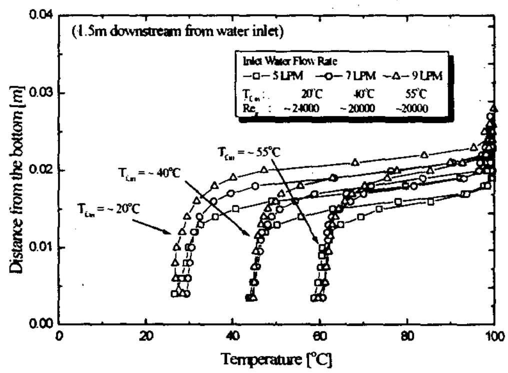

Fig. 3 Local temperature profiles of water flow

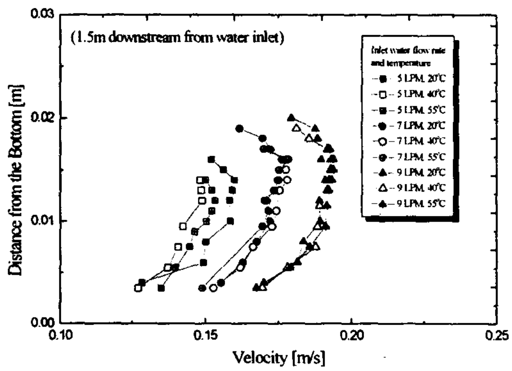

Fig. 4 Local velocity profiles of water flow

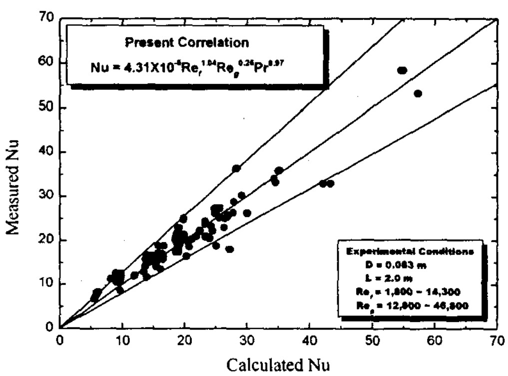

Fig. 5 Comparison between predicted and experimentally measured Nusselt numbers

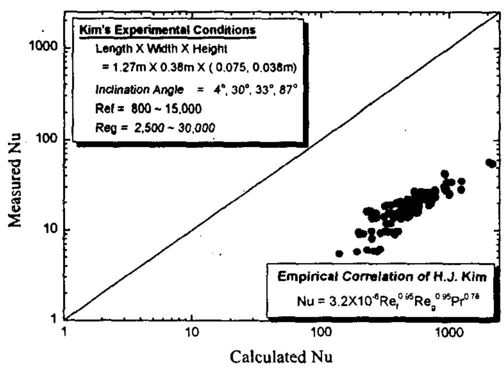

Fig. 6 Comparison of the present data with the H. J. Kim's correlation