

The Effect of Non-condensable Gas on Direct Contact Condensation of Steam/Air Mixture

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Abstract

A series of experiments have been carried out to investigate the effects of non-condensable gas on the direct contact film condensation of vapor mixture under an adiabatic wall condition. The average heat transfer coefficient of the direct contact condensation was obtained at the atmospheric pressure with four main parameters; air-mass fraction, mixture velocity, film Reynolds number, and the degree of water film subcooling having an influence on the condensation heat transfer coefficient. With the analysis of 88 experiments, a correlation of the average Nusselt number for direct contact film condensation of steam/air mixture at an adiabatic vertical wall was proposed as functions of film Reynolds number, mixture Reynolds number, air mass fraction, and Jacob number. The average heat transfer coefficient for steam/air mixture condensation decreased significantly while air mass fraction increased. The average heat transfer coefficients also decreased as the Jacob number increased, and were scarcely affected by the film Reynolds number below a mixture Reynolds number of about 245,000.

Key Words : direct contact condensation, film condensation, non-condensable gas, steam/air mixture

1. Introduction

The surface-type heat exchanger transfers heat through the wall. The surface type heat exchanger capability is reduced because of the thermal resistance of the wall. Many researchers,

therefore, have studied how to reduce this thermal resistance, but there are some restrictions due to the structure of the heat exchanger.

The direct contact heat exchanger can exchange heat without this temperature drop through the wall. Of course, this type of heat exchanger

cannot be adapted to all heat exchangers, due to the problems of the mixing of the two fluids. This type of heat exchanger is used for cooling towers for supplying cooling water, desalination facilities, and the recovery of waste heat in industrial processes. Direct contact heat exchangers have several advantages over surface-type heat exchangers. There are no corrosion or fouling problems that occur in surface-type heat exchangers. There is no thermal resistance of the wall. And they can transfer heat at lower temperature differences. Another advantage is the lower pressure drop compared with tubular-type heat exchangers.

In the nuclear industry, this direct contact condensation plays a very important role in reducing the pressure history of the containment in the case of postulated severe accidents. Also, we can find several cases of direct contact condensation in a nuclear power plant, including the direct vessel injection.

As the direct contact condensation heat transfer occurs, the heat transfer mechanism can be classified into two main mechanisms; the latent heat transfer when the steam is condensed into water film, and the sensible heat transfer by the temperature differences between mixture and water film. Because the convection heat transfer coefficient between the mixture and the liquid film is small, the sensible heat transfer is very small compared with the latent heat transfer. So, the latent heat transfer plays the main role in condensation heat transfer.

Rohsenow et al. [1,2] analyzed the condensation of pure steam and considered the effect of the film subcooling and vapor shear stress at the liquid-vapor interface. They obtained the results that the heat transfer coefficients increased, as the vapor shear stress increased. Kang et al. [3] and Park [4] have studied the effect of wave structures on condensation heat transfer coefficients.

Kim et al. [5] carried out experiments on the countercurrent stratified flow of steam and subcooled water in a rectangular channel. They obtained a correlation which showed the results that the heat transfer coefficients in the smooth interface were more affected by liquid Reynolds number. Their later experiments [6] showed that the interfacial shear stress increased with the increase of steam and liquid flow rates, and the effect of vapor flow rates were more significant. Lim et al. [7] studied condensation of steam on subcooled water in a concurrent horizontal channel. They obtained the local heat transfer coefficients by measuring the velocity profile of pure steam and calculating the condensing mass flow rate. Their experimental correlation showed that the film Reynolds number is more dominant than the vapor Reynolds number for the heat transfer coefficients.

In comparison to the condensation of pure steam, the mixture condensation has additional thermal resistance due to the concentration gradient of non-condensable gas at the interface. This concentration gradient causes the change of the temperature at the interface, resulting in additional heat resistance. The concentration gradient is changed by the air mass fraction of the mixture, the flow pattern of mixture and water film, and the wave structure at the interface. Chan et al. [8] considered the effect of non-condensable gas on the condensation in the stratified horizontal concurrent flow. Their results showed that a reduction in heat transfer rate was found to be as high as 40 percent, with the inlet air mass fraction varying from 0 to 3.4 percent.

Most of the previous researches on condensation were interested in the heat transfer through the wall with condensation. This study was carried out to analyze the direct contact condensation from the mixture to the water film on the adiabatic wall without heat discharging

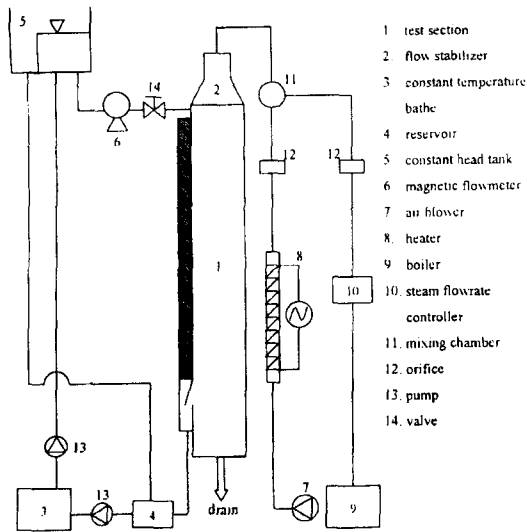


Fig. 1. Schematic Diagram of Experimental Facilities

through the condensing wall. In the present study, experiments for condensation of the steam/air mixture into the water film on the nearly vertical wall (87°) were carried out. The air mass fraction, the velocity of the mixture, the water film flow rate, and the film subcooling at the inlet were selected as the experimental parameters affecting the direct contact condensation heat transfer coefficients. The average Nusselt numbers were correlated with the water film Reynolds number, mixture Reynolds number, air mass fraction of mixture and Jacob number.

2. Experiment

Experimental apparatus is composed of a test section and facilities supplying water film, air, and steam. A schematic diagram is shown in Fig. 1. Fig. 2 shows the side view of the test section. The total length of the test section is 1,750 mm, and the size of the part at which condensation heat transfer occurs is $150W \times 100D \times 1510Lmm^3$. Experiments were carried out on condensation of

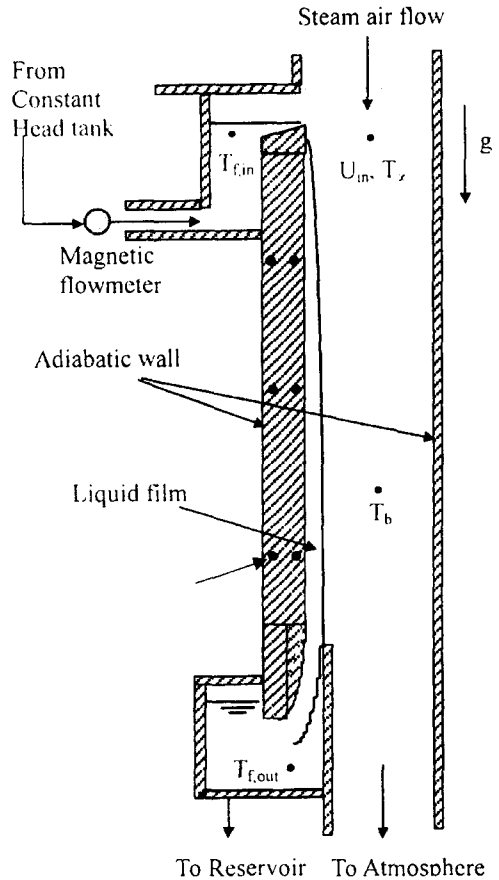


Fig. 2. Side View of the Test Section

external mixture flow for a vertical flat plate. The length of the condensing surface and the size of the cross section of the test duct were designed so that the boundary layers at the opposite wall did not interact. All sides of the test section are insulated to simulate the adiabatic condition. The adiabatic wall condition was confirmed by measuring temperatures at the back of the adiabatic wall of the test section. The temperatures of the water film were measured at the inlet and the outlet of the test section, and the temperature of the steam/air mixture was measured at the outlet of the test section.

The temperature of the water film was

Table 1 Experimental Parameters and Ranges

$V_{mix,in}$	ω_a	$T_{f,in}$	$Re_{f,in}$
2.0m/s	0.0	25°C	4,000 - 6,000
		50°C	1,500 - 9,000
	0.2	25°C	1,500 - 9,000
		50°C	2,500 - 10,000
	0.4	25°C	1,500 - 9,000
		50°C	2,500 - 10,000
0.7	25°C	1,500 - 9,000	
	50°C	2,500 - 10,000	
3.2m/s	0.7	25°C	1,500 - 9,000
		50°C	2,500 - 10,000
4.0m/s	0.7	25°C	1,500 - 9,000
		50°C	2,500 - 10,000
5.4m/s	0.7	25°C	1,500 - 9,000
		50°C	2,500 - 10,000

controlled by a constant temperature bath. The temperature of the water film is maintained at the experimental condition with a maximum standard deviation of 1.5 percent. The subcooled water film is supplied to the test section from the constant head tank, and it falls along the nearly vertical wall by gravity.

The steam flow rate, the air flow rate, and the temperatures of steam and air to be supplied can be adjusted according to each given experimental condition; the air mass fraction of the mixture and the mixture velocity. Steam is supplied from a boiler, which has a capacity of 300 kg/hr and operated at about 7 kg_i/cm². The suspended droplets were filtered through the steam separator, and the steam was depressurized to 0.2 - 0.5 kg_i/cm². The supplied steam flow rate was controlled within a 10% error in all tests, and the standard deviations of the temperature were within 1.6%. Air is supplied from a blower. The temperature of the air is controlled with a heater which has a maximum power of 7.5 kW, and it is controlled within 1 - 2°C deviations. After the

steam and the air is mixed in the mixing chamber for uniform mixing, the mixture flows into the test section through the flow stabilizer. Experimental conditions that were conducted in this study are listed in Table 1.

The mechanism of transferring heat from the mixture to the film is shown in Fig. 3. In the direct contact condensation heat transfer, the total heat gained by the water film is calculated as follows:

$$q_{tot,1} = \dot{m}_{f,out} C_{p,f,out} T_{f,out} - \dot{m}_{f,in} C_{p,f,in} T_{f,in} \quad (1)$$

where q_{tot} is the total heat transfer rate, \dot{m} the mass flow rate, C_p the specific heat, and T the temperature. Subscript f denotes a water film. The total heat transferred to the water film from the mixture can be calculated by:

$$q_{tot,2} = \int_0^L \int_0^W q'' dz dx = \int_0^L \left(m'' h_{fg} + k \frac{dT}{dy} \right) W dx \quad (2)$$

where q'' is the heat flux, m'' the condensing mass flux, h_{fg} the latent heat, W the width of the test section, and L the length of the test section. The sensible heat transfer by the temperature difference between the interface and the bulk mixture is negligible in comparison to the latent heat transfer due to the low convective heat transfer coefficient. So the right hand side of Eq. (2) can be simplified as follows by the area average:

$$q_{tot,2} = \bar{m}'' A \bar{h}_{fg} \quad (3)$$

Here A is the area where the condensation heat transfer occurs. So $\bar{m}'' A$ is the total condensation rate during the heat transfer, and is equal to the increased mass flow rate of condensate. Then Eq. (3) can be re-expressed as follows:

$$q_{tot,2} = \bar{h}_{fg} (\dot{m}_{f,out} - \dot{m}_{f,in}) \quad (4)$$

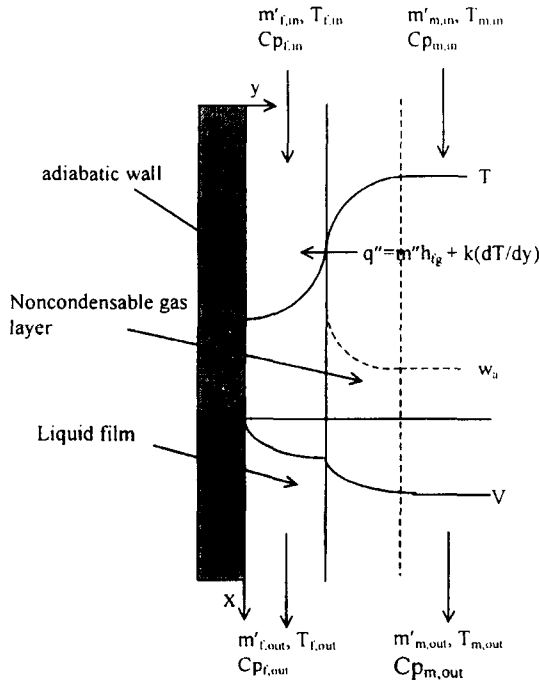


Fig. 3. Model for the Analysis of the Heat Balance

By the heat balance, the total heat gained by the water film must be same to the total heat transferred to the water film. Using Eqs. (1) and (4), the mass flow rate of the water film at the outlet can be calculated from the inlet mass flow rate:

$$\dot{m}_{f,out} = \dot{m}_{f,in} \frac{1 - C_{p_f} T_{f,in} / \bar{h}_{fg}}{1 - C_{p_f} T_{f,out} / \bar{h}_{fg}} \quad (5)$$

Substituting Eq. (5) into Eq. (1), total heat transfer rate can be expressed as:

$$\begin{aligned} q_{tot} &= \dot{m}_{f,in} C_{p_f} \frac{T_{f,out} - T_{f,in}}{1 - C_{p_f} T_{f,out} / \bar{h}_{fg}} \\ &= \rho_f \dot{Q}_{f,in} C_{p_f} \frac{T_{f,out} - T_{f,in}}{1 - C_{p_f} T_{f,out} / \bar{h}_{fg}} \end{aligned} \quad (6)$$

where the specific heat (C_p) and density (ρ_f) of water film was evaluated at the average temperature of the inlet and the outlet of the water film. The latent heat was evaluated at the saturation temperature of the mixture, and the measured values of $\dot{Q}_{f,in}$ (volumetric flow rate of water film), $T_{f,in}$, and $T_{f,out}$ were used.

Using the method of log mean temperature difference (LMTD), the average heat transfer coefficient is defined by:

$$\bar{h} = q_{tot} / (A \Delta T_{lm}) \quad (7)$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \quad (8)$$

$$\begin{aligned} \Delta T_1 &= T_{m,in} - T_{f,in} \\ \Delta T_2 &= T_{m,out} - T_{f,out} \end{aligned} \quad (9)$$

where the subscripts m and f designate the mixture and water film, respectively. The temperatures $T_{m,in}$ and $T_{m,out}$ are assumed to be the saturation temperature. In this experiment, the temperature of the mixture was superheated by 3 - 10°C, but this is neglected because the heat transfer rate by the sensible cooling of the superheated steam to the saturation temperature is small compared with the total heat transfer rate. Even if the superheated steam is cooled to the saturation temperature, its ratio was at maximum 6.2%.

The average Nusselt number is defined by:

$$\overline{Nu} = \bar{h}L/k_f \quad (10)$$

where k_f is the thermal conductivity of the water film.

3. Uncertainty Analysis

An uncertainty analysis was carried out for the average heat transfer coefficients obtained by the present experiments in a 95 percent confidence interval, as follows [9]:

$$\frac{U_{\bar{h}}}{\bar{h}} = \sqrt{\left\{ \left(\frac{1}{\bar{h}} \frac{\partial \bar{h}}{\partial q_{tot}} U_{q_{tot}} \right)^2 + \left(\frac{1}{\bar{h}} \frac{\partial \bar{h}}{\partial \Delta T_{lm}} U_{\Delta T_{lm}} \right)^2 \right\}} = \sqrt{\left(\frac{U_{q_{tot}}}{q_{tot}} \right)^2 + \left(\frac{U_{\Delta T_{lm}}}{\Delta T_{lm}} \right)^2} \quad (11)$$

And the uncertainties for q_{tot} and ΔT_{lm} can be calculated by:

$$\frac{U_{q_{tot}}}{q_{tot}} = \left[\left(\frac{1}{q_{tot}} \frac{\partial q_{tot}}{\partial \dot{Q}_{f,in}} U_{\dot{Q}_{f,in}} \right)^2 + \left(\frac{1}{q_{tot}} \frac{\partial q_{tot}}{\partial T_{f,out}} U_{T_{f,out}} \right)^2 + \left(\frac{1}{q_{tot}} \frac{\partial q_{tot}}{\partial T_{f,in}} U_{T_{f,in}} \right)^2 + \left(\frac{1}{q_{tot}} \frac{\partial q_{tot}}{\partial C_{p_f}} U_{C_{p_f}} \right)^2 + \left(\frac{1}{q_{tot}} \frac{\partial q_{tot}}{\partial h_{fg}} U_{h_{fg}} \right)^2 + \left(\frac{1}{q_{tot}} \frac{\partial q_{tot}}{\partial \rho_f} U_{\rho_f} \right)^2 \right]^{1/2} \quad (12)$$

$$\frac{U_{\Delta T_{lm}}}{\Delta T_{lm}} = \left[\left(\frac{1}{\Delta T_{lm}} \frac{\partial \Delta T_{lm}}{\partial \Delta T_1} U_{\Delta T_1} \right)^2 + \left(\frac{1}{\Delta T_{lm}} \frac{\partial \Delta T_{lm}}{\partial \Delta T_2} U_{\Delta T_2} \right)^2 \right]^{1/2} \quad (13)$$

With these analyses, the uncertainties for the average heat transfer coefficients were 18.2 percent maximum and 2.8 percent minimum. So the average heat transfer coefficients of the present experiments can be said to have the confidence of 81.8 percent minimum in a 95 percent confidence interval.

The uncertainty increased with increasing film Reynolds number and the inlet temperature of the

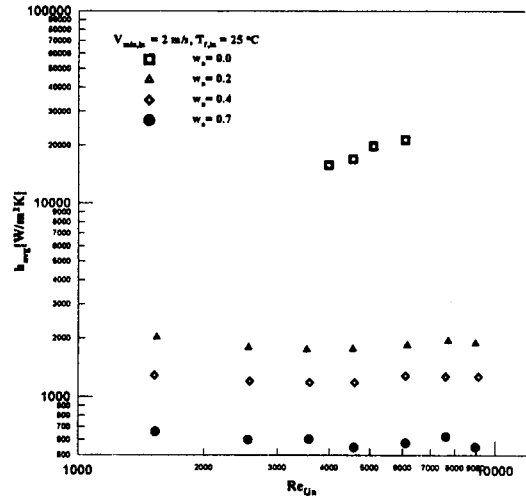


Fig. 4. Effect of Air Mass Fraction and Film Reynolds Number on Averaged Heat Transfer Coefficient

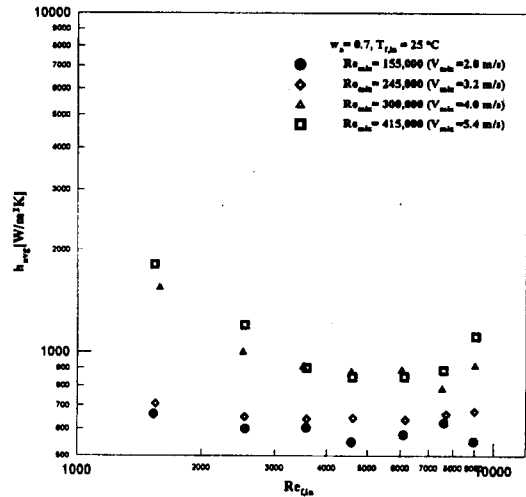


Fig. 5. Effect of Mixture Velocity and Film Reynolds Number on Averaged Heat Transfer Coefficient

water film. But the uncertainty has the tendency to increase with the decreasing velocity of the mixture. In spite of the fact that these uncertainties were affected by the various parameters, the

temperatures of the inlet and outlet of the water film were proven to be the main source of these uncertainties.

4. Results and Discussion

4.1. The Effect of the Air Mass Fraction of the Steam-air Mixture

Fig. 4 shows the average heat transfer coefficients as the air mass fraction of the mixture varies. The air mass fractions varied from 0.0 to 0.7, while the inlet velocity of the mixture and the inlet temperature of the water film was fixed at 2 m/s and 25 °C respectively.

In the case of the pure steam condensation ($w_a=0.0$), all of the supplied steam condensed before reaching the outlet of water film in the range of $Re_{f,in}$ larger than 6,000, and the temperature of the water film reached saturation temperature in the range of $Re_{f,in}$ smaller than 4,000. In these cases, it is impossible to know the heat transferring area and to calculate the average heat transfer coefficients by the LMTD method.

The air, the non-condensable gas, in the steam/air mixture reduces the average heat transfer coefficients significantly. This is due to the concentration gradient at the interface at which heat transfer occurs. The condensation of the steam in the mixture increases the air mass fraction at the interface. The increased air concentration interferes with the condensation of the steam into the water film. This phenomenon leads to the increase of the thermal resistance at the interface and to the decrease of the heat transfer by condensation.

In the cases of the condensation of the pure steam, the temperature at the interface is almost the saturation temperature, and so it can be

assumed that there are no thermal resistances at the vapor side. This means that the thermal resistance exists only at the inside of the water film. So the effect of the film Reynolds number of the water film is dominant for the case of pure steam condensation.

4.2. The Effect of the Mixture Velocity

Fig. 5 shows the variation of the average heat transfer coefficients with the change of the mixture inlet velocity. The increase of the mixture velocity causes greater diffusion of the non-condensable gas at the interface into the bulk mixture. It also increases the diffusion of the steam at the bulk mixture into the interface. This increases the steam mass fraction and the temperature at the interface, in comparison with the case of lower mixture velocity. As the mixture velocity increases, turbulence will be induced in the vicinity of the interface at the water film side, and this will increase the heat transfer from the interface to the water film. For these reasons, the average heat transfer coefficients increase as the mixture velocity increases.

As shown in Fig. 5, the average heat transfer coefficients increase significantly between the inlet mixture velocity of 3.2 m/s and 4.0 m/s. The Reynolds numbers of the mixture are 245,000 and 300,000 respectively at those velocities. This range of the Reynolds number of the mixture seems to be the transient region which induces turbulence in the vicinity of the interface in the vertical concurrent flow. The difference of the average heat transfer coefficient between the two cases, i.e., a mixture Reynolds number of 245,000 and 300,000, becomes small as the film Reynolds number increases. As the film Reynolds number increases, the film thickness increases. So the effect of the induced turbulence in the vicinity

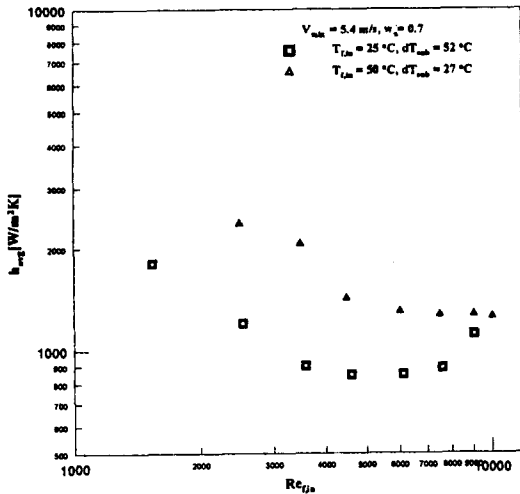


Fig. 6. Effect of Film Subcooling and Film Reynolds Number on Averaged Heat Transfer Coefficient

of the interface at the water film side becomes relatively small.

As the temperature increases at the interface, the motion of the water film play an important role on the heat transfer as in the case of the pure steam condensation. On the other hand, as the mixture velocity decreases, the thermal resistance at the mixture side becomes large and the effect of the motion of the water film for the average heat transfer coefficients become relatively small compared with the cases of high mixture Reynolds number. So the average heat transfer coefficients are scarcely affected by the film Reynolds number below the mixture Reynolds of number about 245,000.

4.3. The Effect of the Degree of the Water Film Subcooling

Fig. 6 shows the influence of the degree of the water film subcooling on the average heat

transfer coefficients for the case of fixed inlet velocity and air mass fraction of the mixture, 5.4 m/s and 0.7 respectively. As shown in Fig. 6, the average heat transfer coefficients increase as the subcooling of the water film becomes small.

As the water film is subcooled more, the temperature difference between the bulk mixture and the water film becomes larger. As a result of this, for the case of the inlet temperature of the water film at 25°C, the total heat transfer rate is larger than that of the case at 50°C. The log mean temperature difference, the denominator in the definition of the heat transfer coefficient, becomes large when the degree of subcooling increases. The interface temperature increases because the thermal diffusion rate in the liquid film is lower than the condensation heat transfer at the interface. The increased interface temperature prevents the condensation at the interface. In these experiments, the increasing rate of total heat transfer due to the condensation is smaller than that of the log mean temperature difference while the degree of the water film subcooling increases. So the average heat transfer coefficients become small, when the degree of the water film subcooling increases.

4.4. The Correlation for the Average Nusselt Number

The correlation for the average Nusselt number is suggested with the variables; film Reynolds number ($Re_{f, in}$), mixture Reynolds number (Re_{mix}), air mass fraction of mixture (w_a), and Jacob number (Ja), which is the dimensionless number of the degree of water film subcooling. The suggested form of the correlation is as follows:

$$\overline{Nu} = a Re_{f, in}^b Re_{mix}^c w_a^d Ja^e \quad (14)$$

where:

$$Re_{f,in} = \frac{4\Gamma_{f,in}}{\mu_{f,in}}, \quad \Gamma_{f,in} = \frac{\dot{m}_{f,in}}{W}$$

$$Re_{mix} = \frac{\rho_{mix} V_{mix,in} L}{\mu_{mix}}$$

$$w_a = \frac{\rho_{air}}{\rho_{mix}}$$

$$Ja = \frac{\rho_f C_{p_f} \Delta T_{sub}}{\rho_{mix} h_{fg}}$$

Here W and L are the width and the length of the test section, ΔT_{sub} is the degree of subcooling of the liquid film, and $V_{mix,in}$ is the velocity of the mixture. The density of the mixture is an intensive property, which was calculated by the followed relation.

$$\rho_{mix} = w_a \rho_{air} + (1 - w_a) \rho_{steam}$$

A transport property of mixture like viscosity (μ_{mix}) was calculated by the method of Wilke [10].

Because the tendency of the heat transfer coefficient by the mixture Reynolds number is significantly different in each region below, the correlated equations for the averaged Nusselt number (Eq. (10)) of the experimental results are suggested below for two regions of mixture Reynolds number:

$$i) 150,000 < Re_{mix} < 249,000$$

$$\overline{Nu} = 37.41 Re_{f,in}^{-0.012} Re_{mix}^{0.530} w_a^{-1.038} Ja^{-0.592} \quad (15)$$

$$ii) 286,000 < Re_{mix} < 417,000$$

$$\overline{Nu} = 724.16 Re_{f,in}^{-0.359} Re_{mix}^{0.530} w_a^{-1.038} Ja^{-0.592} \quad (16)$$

The water film Reynolds number, mixture Reynolds number, and the air mass fraction used in Eq. (15) and Eq. (16) are evaluated at the inlet condition, and the Jacob number is evaluated with values at the average temperature of the inlet and the outlet. The best fitting correlation covers most of experimental data within $\pm 20\%$, as shown in Fig. 7.

5. Conclusions

The effect of air mass fraction, Reynolds number of the steam/air mixture, degree of water film subcooling, and film Reynolds number of water film on the average heat transfer coefficients are analyzed under the condition that the steam/air mixture condenses into the water film on the adiabatic vertical wall.

The average heat transfer coefficients decrease as the air mass fraction of the mixture increases, due to air concentration resistance near the interface. The film Reynolds number has no influence on the average heat transfer coefficients below a mixture Reynolds number of 249,000. This means that the thermal resistance in the steam/air mixture side is relatively large with

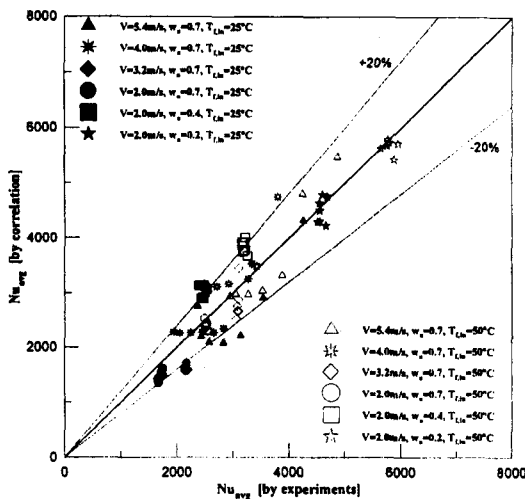


Fig. 7. Comparison Between the Correlation and the Experiments

respect to that in the water film side as the mixture velocity decreases. The average heat transfer coefficients increase as the mixture Reynolds number increases. The average heat transfer coefficients increase as the degree of the water film subcooling decreases. A correlation of the average Nusselt number is obtained within the ranges of these works with film Reynolds number of water film, Reynolds number of mixture, air mass fraction of mixture, and Jacob number, which represents the degree of water film subcooling.

Nomenclature

\dot{m}	Mass flow rate, kg/s
$q_{tot,1}$	Total heat gained by the water film, m
$q_{tot,2}$	Total heat transferred from the mixture to the water film, W
q_{tot}	Total heat, W
L	Length of the test section, m
W	Width of the test section, m
q''	Heat flux, W/m ²
x, y, z	Coordinate
C_p	Specific heat, J/kg K
T	Temperature, K
h_{fg}	Latent heat, J/kg
k	Thermal conductivity, W/mK
m''	Mass flux, kg/m ²
A	Area, m ²
ρ	Density, kg/m ³
\dot{Q}	Volumetric flow rate, m ³ /s
h	Heat transfer coefficient, W/m ² K
Nu	Nusselt number
U	Uncertainty
w_a	Mass fraction, ρ_{air}/ρ_{mix}
Re	Reynolds number
Ja	Jacob number
Γ	Mass flow rate per unit width, kg/m s
μ	Dynamic viscosity, kg/m s

V velocity, m/s

Subscripts

f	Water film
in	Inlet
out	Outlet
m, mix	Mixture
air	Air
sub	Subcool

References

1. W. M. Rohsenow, "Heat Transfer and Temperature Distribution in Laminar-Film Condensation," *J. Heat Transfer*, **78**, pp. 1645-1648 (1956).
2. W. M. Rohsenow, J. H. Webber, and A. T. Ling, "Effect of Vapor Velocity on Laminar and Turbulent-film Condensation," *J. Heat Transfer*, **78**, pp. 1637-1643 (1956).
3. H. C. Kang and M. H. Kim, "Measurement of Three-dimensional Wave Form and Interfacial Area in an Air-Water Stratified Flow," *Nuclear Eng. and Design*, **136**, pp. 347-360 (1992).
4. S. K. Park, *Effects of Wavy Interface on Film Condensation of Steam/Air Mixture on a Vertical Surface*, Ph. D. dissertation, Pohang University of Science and Technology (POSTECH) (1996).
5. H. J. Kim and S. G. Bankoff, "Local Heat Transfer Coefficients for Condensation in Stratified Countercurrent Steam-Water Flows," *J. Heat Transfer*, **105**, pp. 706-712 (1983).
6. H. J. Kim, S. C. Lee, and S. G. Bankoff, "Heat Transfer and Interfacial Drag in Countercurrent Steam-Water Stratified Flow," *Int. J. Multiphase Flow*, **11**, 5, pp. 593-606 (1985).
7. I. S. Lim, R. S. Tankin, and M. C. Yuen, "Condensation Measurement of Horizontal

- Cocurrent Steam/Water Flow," *J. Heat Transfer*, **106**, pp. 425-432 (1984).
8. T. S. Chan and M. C. Yuen, "The Effects of Air on Condensation of Stratified Horizontal Concurrent Steam/Water Flow," *J. Heat Transfer*, **112**, pp. 1092-1095 (1990).
9. H. W. Coleman and W. G. Steele Jr., *Experimentation and Uncertainty Analysis for Engineers*, John Wiley & Sons Inc., New York (1989).
10. R. C. Reid, J. M. Prausnitz, and B. E. Poling, *The properties of gas & liquids*, 4th Ed., McGraw-Hill, pp. 407-410 (1986).