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A Study of the Evaporation Heat Transfer in Advanced Reactor Containment

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Abstract

In advanced nuclear reactors, the passive containment cooling has been suggested to enhance the safety. The passive cooling has two mechanisms, air natural convection and water cooling with evaporation. To confirm the coolability of PCCS, many works have been performed experimentally and numerically. In this study, the water cooling test was performed to obtain the evaporative heat transfer coefficients in a scaled down segment type PCCS facility which have same configuration with AP600 prototype containment. Air-steam mixture temperature and velocity, relative humidity and wall heat flux are measured. The local steam mass flow rates through the vertical plate part of the facility are calculated from the measured data to obtain evaporative heat transfer coefficients. The measured evaporative heat transfer coefficients are compared with an analytical model which use a mass transfer coefficients. From the comparison, the predicted coefficients show good agreement with experimental data, however, some discrepancies exist when the effect of wave motion is not considered. Finally, a new correlation on evaporative heat transfer coefficients are developed using the experimental values.

I. Introduction

Since TMI and Chernobyl accidents, many efforts have been performed to develope advanced reactors which have inherent safety with passive safety features. Such passive features should be operated by natural physical forces such as natural convection or gravity during accidents. For the containment which is the last barrier against radioactivity release to the public, the passive containment cooling system(PCCS) has been suggested in AP600 to remove the released energy during postulated accidents. PCCS can be operated by the natural circulation of air and the evaporative heat transfer through poured water from the top of the containment. In order to conform the coolability of PCCS, the evaporative heat transfer under vertically countercurrent natural circulated air flow should be studied in such closed gap. The basic studies on evaporation to obtain the phenomenological understanding and appropriate correlations of heat transfer coefficient have been performed in the fields of mechanical and chemical engineering.

W.W. Baumann and F. Thiele[1] analytically studied the mass transfer under multicomponent liquid film. T.S. Chen and C.F. Yuh[2] studied natural convective heat and mass transfer characteristics by the thermal and species diffusion along a vertical tube. C.J. Chang, T.F. Lin and W.H. Yan[3] investigated the effects of tube length and system pressure on heat and mass transfer. W.M. Yan, T.F. Lin and Y.L. Tsay[4] performed the experimental and numerical study on the evaporative cooling of falling liquid film along a vertical channel. They showed that the more effective evaporation is occurred under small flow rate and high liquid temperature. K.R. Chun and R. A. Seban[5] proposed the evaporative heat transfer coefficients on liquid film along the outside of vertical tube under laminar and turbulent flow conditions. In their works, under the laminar flow condition, the heat transfer coefficient was obtained with Nusselt's solution multiplied by the wave factor of Zauzuli. Anders and Thore Berntsson[6] obtained heat transfer coefficient correlations under turbulent falling film with their own experimental data as well as others. In their experiments, the evaporation mass was about 80% of total incoming liquid mass flow.

In nuclear industries, some experiments on passive cooling were performed by Westinghouse[7] and Ambrosini[8]. Westinghouse performed the integrated cooling tests with both of air and water cooling for PCCS and local phenomenological studies such as air flow resistance and water distribution on heated plate. In their experiments, the heat transfer rates were measured with condensation rates inside the vessel. The water film was formed by sprayed cold water at top of the plate. Based on their studies, Westinghouse proposed that the Colburn correlation on mass transfer can be used to predict the evaporation cooling. However, their experimental data showed some uncertainties at a given air Reynolds number for the water flow rates. It seems that this is due to the fact that the length required to reach to the thermal equilibrium state between water and heated plate was not considered, which means that the heat transferred from heated wall to water film is spent to heat up the water and thus the evaporation length is reduced. Ambrosini performed experimental studies on air cooling and water cooling with 2m length vertical duct. In this study, the evaporation heat flux is obtained from simple energy balance considering applied electrical power, sensible heat transfer spent to heat up water film, convection heat transfer from water film surface to air bulk and heat loss to environments. The water film temperature is controlled to be ambient conditions or heated plate temperature to remove the sensible heat spent to heat up the water. Their experimental results were compared with the semi-empirical correlations obtained by Kreith[9] in a short duct. Both the study of Westinghouse and Ambrosini was performed for the averaged heat transfer using observed data at inlet and outlet conditions.

In this study, the water cooling test was performed to obtain the evaporative heat transfer coefficients in PCCS facility which is a scaled down segment, but has the shape with the AP600 prototype containment having closed one-side heated plate. In experiments, air-steam mixture temperature, velocity, relative humidity and wall heat flux are measured at three points according to the vertical plate 0.8m to 2m from the bottom. And, the local heat transfer coefficients are calculated and averaged through the vertical plate of the facility from the measured data. For the evaluation of the measured heat transfer coefficients and physical understanding of the phenomena, a simple analytical model for the local evaporative heat transfer is also established. The analytical evaporation model is based on the heat and mass transfer analogy. Also the effects of the complex phenomena such as interfacial shear caused by count-current flow, wavy effect of water film are evaluated in the analytical model. From the comparison, the predicted coefficients show good agreement at low water film temperature even some discrepancies at high water film temperature. Finally, a new correlation on evaporative heat transfer coefficients are developed using the experimental values.

2. Experimental Apparatus

Figure 1 shows the schematic diagram of experimental facility. The test section is a 1/26 length scaled segment of AP600 PCCS geometry. The facility is composed of preheating system, water distribution system, heating plate and air baffle, water drain system and air blowing system as shown in Fig. 1. The preheating system having the power of 50kW controls the temperature of inlet water which is supplied to the test section through the water distributor. The water distribution system has a flow meter and a water distributor. For the uniform wetting, the water distributor has many holes which eject the water to heated surface. The stainless steel heated plate is 0.3m wide and 2m long with total power of 28kW. The lengths of curved and vertical part are 0.8m and 1.2m, respectively. In experiments, all measurements are implemented at the vertical section. The air baffle is made with an acril plate sealed with teflon seal and silicon sealant to protect air and steam loss. The gap size between the heating plate and baffle is 4cm. The air blower is connected to the inlet of air duct to control the air flow. For the uniform air distribution, honeycombs and grid are set in the air inlet.

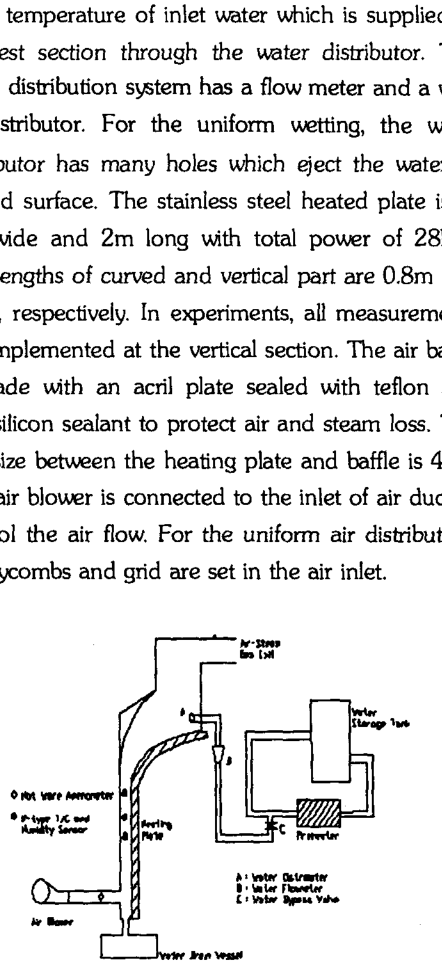

Fig. 1. Schematic Diagram of PCCS Facility

The experimental conditions performed in this study and other experimental conditions of Westinghouse and Ambrosini are shown in Table 1.

3. Measurements of Evaporative Heat Transfer Coefficients

To estimate the evaporation rate, air-steam mixture temperature, relative humidity and air inlet velocity are measured. The relative humidity is measured by THT-B120 humidity sensor. This sensor generates voltage signals of 0~1V as the humidity varies. The air-steam mixture temperatures are measured by k-type thermocouples of 0.5mm diameter. The relative humidities and air-steam mixture temperatures are measured at three points along the test sections. The air inlet velocity is measured with the hot wire anemometer, model 6201 of KANOMAX.

In these experiments air mass flow rates and water inlet temperatures are varied with the range of 50,000~350,000 for Re_{air} and 20~25℃, respectively. The inlet water temperature is preheated to keep constant initial conditions. The preheated water temperatures are 40, 50℃ which are correspond to Pr_l of 4.0, 3.7, respectively. To compensate the water temperature reduction by the evaporation cooling, the stainless steel plate is heated. To minimize the water temperature variation, the heat flux is controlled to equal the outlet water temperature to the inlet water temperature. Therefore, it can be assumed that the water temperature is axially uniform through the test section.

In the air baffle, the heat transfer from water film surface to air-steam mixture has three mechanisms ;

Table 1. Comparisons of Experimental Conditions Between this Study and Others

	L/D_h	Heat Flux (kw/m^2)	film flow rate (kg/ms)	Air Velocity (m/s)	Air Temperature(℃)	Relative Humidity
Westinghouse	18	5~28	0.02~0.083	up to 12	up to 40	up to 60
Ambrosini	11.67	up to 30	0.013~0.17	up to 10	25~50	45~85
This Study	17	up to 28	up to 0.75	up to 7	20~25	up to 60

radiation, convection, and evaporation. In the case of simultaneous heat and mass transfer, the heat transfer by evaporation is much higher than those by both of radiation and convection. Thus, neglecting those effects, the evaporation heat transfer can be obtained as follows ;

$$\begin{aligned} \dot{q}'' &= \dot{m}''_{evap} h_{fg}(T_i) \\ &= h_{evap}(T_i - T_g) \end{aligned} \tag{1}$$

From Eq. (1), the evaporation heat transfer coefficient is defined as follow;

$$h_{evap} = \frac{\dot{m}''_{evap} h_{fg}(T_i)}{T_i - T_g} \tag{2}$$

In Eqs. (1) and (2), the liquid interface temperature can be hardly measured directly. Thus, appropriate assumptions should be required. In this experiment, as the water temperature was controlled to remain constant through the test section, the liquid interface temperature was assumed to be same as the water bulk temperature in this study. The evaporation mass flux in Eq. (2) can be determined with local temperatures, relative humidity and flow velocities. The air-steam mixture is assumed to be homogeneous. Because the density ratio of steam to air is less than 0.02 under the experimental conditions, the mixture velocity is dominated by air flow. Thus, the steam velocity may be assumed to be equal to the air velocity. Because the inlet air mass flow rate is conserved through the duct, the local velocity of air-steam mixture can be obtained as follows ;

$$\dot{m}_{air} = \rho_{air,in}\, u_{air,in}\, A_{x,in} \tag{3}$$

$$u_v = u_{air} = \frac{\dot{m}_{air}}{\rho_{air} A_x} \tag{4}$$

where, the area ratio of $A_{x,in}/A_x$ is 2.5 and the air is assumed to be the perfect gas and then the local air density can be determined from the local temperatures. After the local velocities are determined from Eqs. (3) and (4), the steam mass flow rate is determined with the vapor density at air-steam mixture temperature and the relative humidity as follows ;

$$\dot{m}_{steam} = \rho_v u_v A_x \theta \tag{5}$$

Then, the evaporation mass flux can be obtained with the local steam mass flows as,

$$\dot{m}''_{evap} = \frac{\Delta \dot{m}_{steam}}{A_{heat\ area}} \tag{6}$$

From Eqs. (2) and (6), the local evaporation heat transfer coefficient can be finally calculated with various experimental conditions of water temperature, water flow rate and air flow rate.

4. Simple Evaporation Model

In this study, a simple evaporation model has been also developed to evaluate the experimental results, where heat and momentum transfer through the liquid film are not considered. Homogeneous and saturated gas mixture conditions are assumed. For the turbulent flow in a vertical plate, the mass transfer rate can be determined by following correlation[10, 11]

$$\frac{K_g}{u_g} = \frac{f}{2} Sc_t^{-1} \tag{7}$$

where $f/2 = 0.0296 Re_g^{-0.2}$, and $Sc_t = Sc^{-3/2}$. In Eq. (7), the friction coefficient, f/2, does not include the interfacial shear effect by both the countercurrent flow and the wavy effect on liquid film surface. In the condensation process, the interfacial shear increment enhances the heat transfer rate up to 40%[12]. To account this factor McAdams[13] and Zauzuli[5] suggested the wave multiplication factors of 1.28 and 687 $Re_l^{0.11}$, respectively. At the vertical plate, H.C. Kang[14] measured the interfacial shear by wave motion on the liquid film interface for the air flow velocity and showed that the interfacial shear by wave motion is linearly proportional to air velocity and water flow rate. In this study, the effect of wave motion is simply introduced by the linear function of air velocity using the data of H.C.Kang. The total shear stress is sum of rigid body shear and wave motion shear.

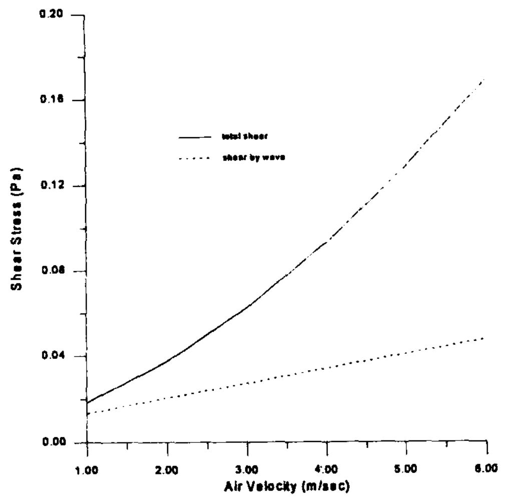

Fig. 2. Calculated Shear Stress vs. Air Velocity

$$\tau = \frac{f}{2}\rho u^2 + \tau_{i,w} = \frac{f'}{2}\rho u^2 \qquad (8)$$

where $\tau_{i,w}$ is shear by wave motion and $f'/2$ is modified friction coefficient including wave motion. Thus, in Eq. (7), the friction coefficient f/2 can be replaced to $f'/2$ to account the wave motion. The calculated shear is shown at Fig. 2.

Eq. (7) is the analogy between momentum and mass transfer, and shows the mass transfer coefficient is proportional to flow velocity only. Thus, as the flow velocity is the control variable, the mass transfer coefficient can be easily obtained. With the mass transfer coefficient, the evaporation mass flux is calculated with vapor partial pressure difference between the water film surface and the vapor bulk as follow;.

$$\dot{m}''_{evap} = K_g(\rho_{v,i} - \rho_{v,b}) \qquad (9)$$

Then the evaporation heat transfer coefficients are determined from Eq. (2).

5. Results

In this study, the experimental and analytical investigations on the evaporation heat transfer are performed. In experiments, the water flow rate, water temperature and air inlet velocity are varied. The measured heat transfer coefficients are compared with analytical models. The two dimensional calculations include the continuity, momentum, energy, concentration and turbulent equations. The two dimensional calculation is not performed in our study but referred in Ph. D. thesis of H.O.Kang[15].

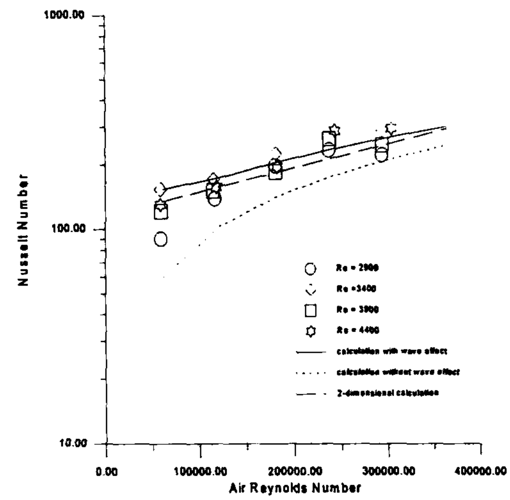

Fig. 3. Nusselt Number vs. Air Reynolds Number (Pr = 4.3)

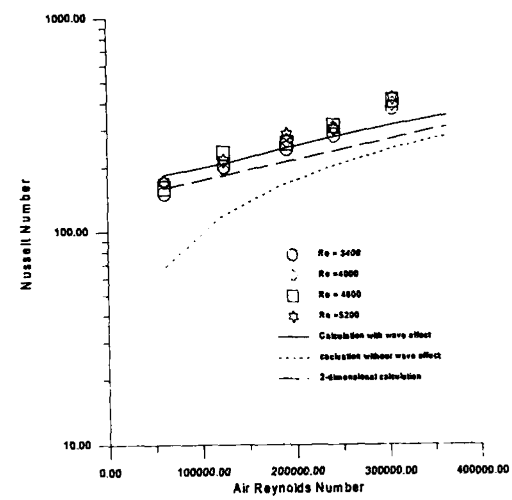

Fig. 4 Nusselt Number vs. Air Reynolds Number (Pr = 3.7)

As the air velocity increases, the friction coefficients which affect the mass transfer increases. That means

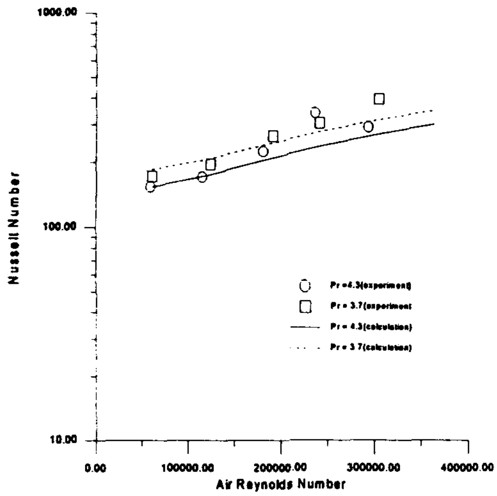

Fig. 5. Nusselt Number vs. Air Reynolds Number ($\Gamma = 0.67kg/s \cdot m$)

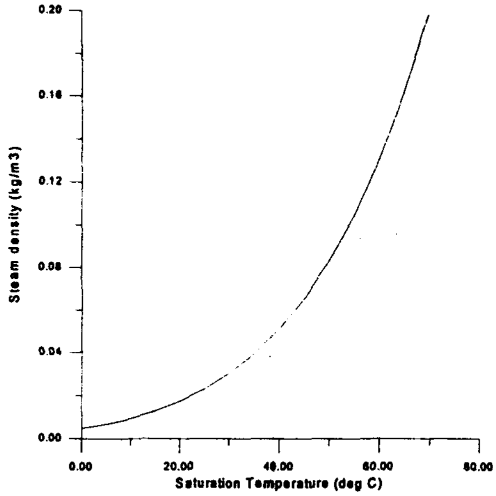

Fig. 6. Steam Density vs. Saturation Temperature

the steam transfer from liquid film surface to air-steam gas mixture is enhanced. Then the steam in the channel can be well removed and, as the results, the steam partial pressure gradient should be large. Thus, the evaporative heat transfer coefficient increases as the air velocity increases, as shown in Figs. 3 and 4.

As the film Reynolds number increases, the wavy motion on the film interface enhance the heat transfer rate. For the laminar film heat transfer coefficient, Chun and Seban[5] suggested $0.8Re^{0.1}{}_{\Gamma}$ as the wavy correction factor. This means that the wavy effect has small dependency on film Reynolds number. Thus, in the simple evaporation model, the wavy effect did not include the effect of film Reynolds number. Also in the experimental results shown in Figs. 3 and 4, the effect of film Reynolds number is not so large.

In Fig. 5, the effect of water inlet temperature is investigated. In this study, the water inlet temperature is represented with water Prandtl number, Pr_l, because Rr_l depends on its temperature. In Eq. (2), the evaporative heat transfer coefficient is depend on the evaporation mass flux and temperature differences. As the water temperature increases, both the temperature difference and the evaporation mass flux become large. Thus, the heat transfer coefficient defined in Eq. (2) has nonlinear behavior with the negative effect of temperature difference but the positive effect of steam density difference. However, the steam density versus temperature has the shape of exponential function as shown in Fig. 6. Additionally, as the result of evaporation mass flux increment, the steam mass fraction in the mixture bulk increases and the noncondensable gas effect reduces, which is the main resistance for vapor diffusion from the film surface to the channel bulk. Thus, the positive effect is larger and the heat transfer coefficient increases as the water temperature increases.

In Figs. 3 and 4, the calculated heat transfer coefficients are well agreed with experimental data, however, the calculated heat transfer coefficients without wave effect underpredict the experimental data and shows large discrepancies at low air velocity. At low air velocity, the portions of wave effect on the total shear is relatively large shown in Fig. 2.

For the application of heat transfer in PCCS, a new evaporative heat transfer coefficient correlation is required. Thus, in this study, the evaporation heat transfer coefficient correlation is developed using experimental data. In the evaporation phenomena, important dimensionless variables are Nu, Re, Pr numbers for heat transfer and Sh, Re, Sc numbers for mass transfer. In the application of PCCS safety analysis, the major interest is only the heat removal rate by evaporation. Thus, in this study, the correlation is obtained for heat transfer only expressed with functions of air Reynolds number, water Prandtl number and Film Reynolds number as,

$$Nu = 0.021 Re_g^{0.8} Pr_l^{-3/4} Re_l^{0.1} \quad (10)$$

In Eq. (10), the air Reynolds number dependency is $Re^{0.8}{}_g$, and this corresponds to the characteristic behavior of turbulent flows shown in the friction coefficients used in Eq. (7). The Film Reynolds number dependency is $Re^{0.1}{}_l$ and this is much smaller than the other parameters. And the Prandtl number dependency is $Pr_l^{-3/4}$ in this study. The Fig. 7 shows the correlated heat transfer coefficients vs. measured heat transfer coefficient data. Most data are within 20% error band. Thus, the correlation is well agreed with experimental data.

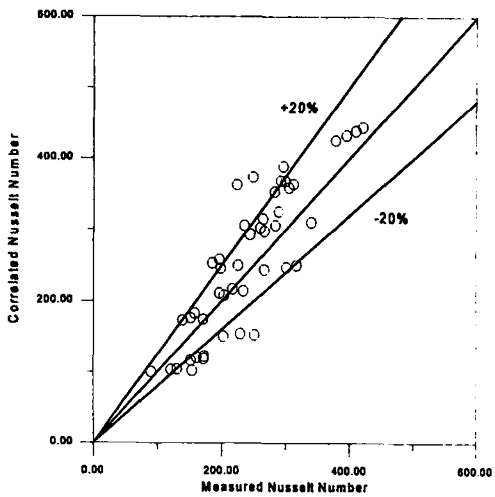

Fig 7. Correlated Nusselt Number vs. Measured Nusselt Number

6. Conclusions

In this study, experimental and analytical studies on the evaporative cooling in PCCS were performed. As the water inlet temperature, Film Reynolds number, and air inlet velocity are increased, the evaporative heat transfer coefficients are increased. Among them, the film Reynolds number has the smallest effects than the other parameters on the evaporative heat transfer coefficients. The analytical evaporation models are compared with experimental results and quite well agreed. However, if the wave effect is not included, the analytical results underestimate experimental results.

Using experimental data, a new heat transfer coefficient correlation is developed as functions of air Reynolds number, water Prandtl number and Film Reynolds number. The resultant correlation of the evaporation heat transfer in a closed channel such as PCCS is suggested as,

$$Nu = 0.021 Re_g^{0.8} Pr_l^{-3/4} Re_l^{0.1}$$

In the proposed correlation, the air Reynolds number dependency is $Re^{0.8}{}_g$, and this corresponds to the characteristic behavior of turbulent flows suggested from other studies. Most data of correlation are within 20% error band. Thus, the correlation is well agreed with experimental data.

Nomenclature

$A_{heatarea}$ = Heat transfer area of vertical plate between two measuring points

A_x = Cross section area of gas mixture flow(gap thickness × width)

L = The length of vertical plate

Nu = Nusselt number($\frac{hL}{k_l}$)

Re_g = gas reynolds number($\frac{uL}{\nu_g}$)

Pr_l = Water Prandtl number($\frac{\mu c}{k_l}$

Sc_t = turbulent Schmidt number

T = temperature
u = velocity
Γ = Water flow rate per unit depth
ρ = density
θ = relative humidity

Subscripts

b = air-steam mixture bulk
evap = *evaporation*
i = liquid film interface
l = liquid water
g = air-steam mixture
v = steam
x = local positions according to the vertical dirction

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