한국에너지공학회지(1998), 제7권 제1호
Energy Engg. J(1998), Vol. 7, No. 1, pp. 122~130

Derivation of a Simplified Heat Transfer Correlation for AP 600 Passive Containment Cooling System

Bum Jin Chung
Nuclear Research & Development Division Ministry of Science & Technology

Abstract — A simplified heat transfer model for the cooling capability of the AP 600 PCCS is proposed in this paper. As the PCCS domain is covered with very thin and long water film, it is phenomenologically divided into 3 regions; water entrance effect region, asymptotic region, and air entrance effect region. As the length of the asymptotic region is estimated to be over 90% of the whole domain, the phenomena in the asymptotic region is focused. Using the analogy between heat and mass transfer phenomena in a turbulent situation, a new dependent variable combining temperature and vapor mass fraction was defined. The similarity between the PCCS phenomena in the asymptotic region and the buoyant air flow phenomena on a vertical heated plate is derived. Using the similarity, the simplified heat transfer correlations for the interfacial heat fluxes and the ratios of latent heat transfer to sensible heat transfer were established. To verify the accuracy of the correlation, the results of this study were compared with those of other numerical analyses performed for the same configuration and they are well within the range of 15% difference.

1. Introduction

The advanced reactor containment design concept adopts a passive containment cooling capability which provides an ultimate heat sink in an accident situation. Most of the proposed passive containment cooling system (PCCS) designs incorporate natural convection cooling. AP 600 PCCS design[1] enhanced its cooling capability by the introduction of water film. But due to the introduction of this additional cooling mechanism, the estimation of the cooling capability by either numerical or experimental method is difficult in spite of the simple geometric configuration.

Much experimental and numerical effort has been made to estimate the cooling capability of the design but the inherent complexity caused by the combined heat and mass transfer at the air-water interface, counter current flow situation of the falling water and the buoyant mixed-air flow, and the highly turbulent natural convection air flow, are not yet clearly identified.

2. Problem Description and Derivation of Governing Equations

Figure 1 shows the AP 600 PCCS configurations[1]. Water film, running down along the steel containment vessel, is heated by the containment wall and cooled by buoyant air flow, which exhibits a counter current situation.

Since the water film is cooled not only by sensible heat transfer but also by latent heat transfer, combined heat and mass transfer phenomena are expected. Nucleate boiling, entrainment of water-va-

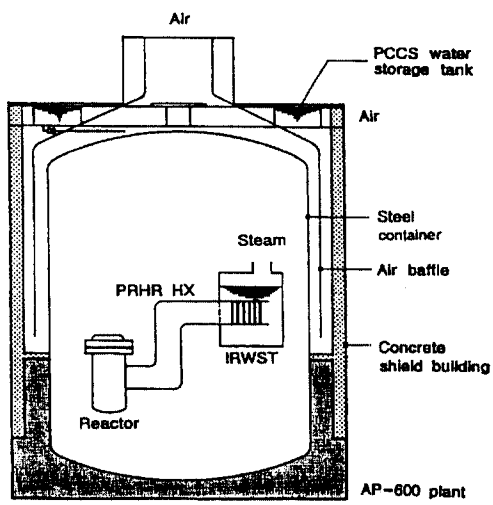

Fig. 1. AP 600 passive containment cooling system.

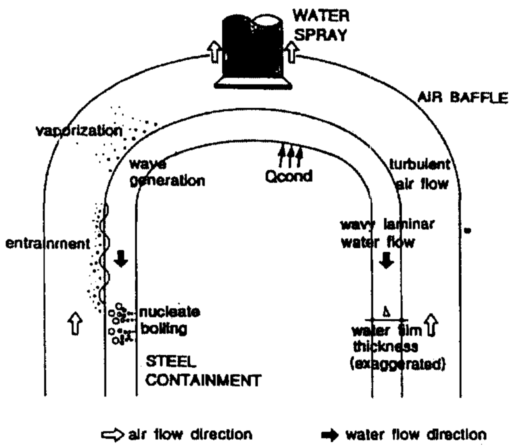

Fig. 2. Schematization of PCCS phenomena.

por, and wave generation at the interface of water and air flow are also possible phenomena but are not expected in view of the PCCS operating conditions. Figure 2 presents the schematization of the possible PCCS phenomena.

Since the diameter of the containment is large, the effects of curvature are neglected and the calculation domain is safely approximated to 2-dimensional rectangular domain[2], which is presented in Fig. 3.

Due to the high Rayleigh number ($>10^{13}$) in this system, strong turbulent intensities are expected in the natural convection mixed air flow.

According to the conceptual design of the AP 600 PCCS, the containment height is 50~60 m in scale

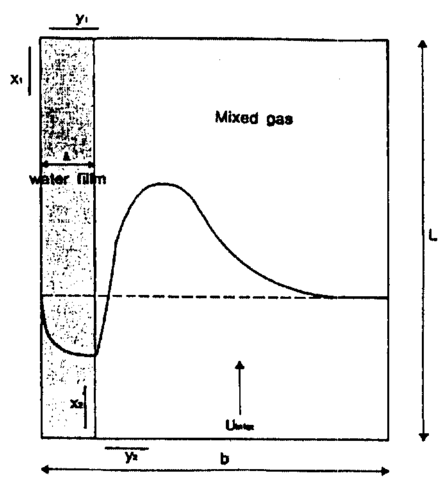

Fig. 3. Simplified calculation domain.

and the water film thickness is less than 1 mm in scale. This means that the height to thickness ratio of the water film is extremely large. Water film flow characteristics depend on film Reynolds number which is a function of both film thickness and viscosity. For the thin water film in this system, a laminar water film flow assumption is relevant. The data on AP 600 PCCS were based upon the Westinghouse reports[1]. Thus whether the thin film can reach the bottom and how the non-wetting parts can be prevented on the containment surface are not considered in this study. But the film thickness variations along the flow under containment heat flux conditions are reported to be less than 15% and most of thin film condensation problems neglect the variation of film thickness. Thus it is reasonable to make a constant film thickness assumption[3].

Nucleate boiling and entrainment of water vapor were not considered in the analysis due to the thin water film. The waves generated at the interface of water and air flow are reported to have some effects on the interface heat flux but they do not change the major scales. Therefore wave effects are not considered in the analysis. Table 1 summarizes the critical parameter values used in phenomena modeling[4,5,6].

The 2-dimensional, steady state, laminar flow in the water film is described by the following equations:

Momentum

$$\frac{\partial}{\partial y_1}\left(v_1 \frac{\partial u_1}{\partial y_1}\right)+g=0 \tag{1}$$

Energy

$$u_1 \frac{\partial T_1}{\partial x_1} = \frac{\partial}{\partial y_1}\left(\alpha_1 \frac{\partial T_1}{\partial y_1}\right) \tag{2}$$

Entrance Condition

$$x_1=0\,;\ u_1=0\,,\ T_1=T_\infty \tag{3}$$

Table 1. Critical parameter values for gravity film and buoyancy-induced gas flow.

Assumptions	Critical parameter values
Laminar liquid film	$Re_f \leq 1500$
No film break-up	$\delta \geq 0.01$ mm
No nucleate boiling	$q_w \leq 8$ kW/m^2
Laminar gas flow	$Ra_H \leq 10^9$
No entrainment	$\delta < 5$ mm

Heated Wall Condition

$$y_1 = 0\ ;\ u_1 = 0\ ,\ T_1 = T_w = \text{constant.} \quad (4)$$

The 2-dimensional, steady, turbulent, boundary layer flow in the mixed gas side is governed by the following conservation equations along with Boussinesq assumption, which assumes the flow is incompressible except for the buoyancy terms in the momentum equation:

Continuity

$$\frac{\partial u_2}{\partial x_2} + \frac{\partial v_2}{\partial y_2} = 0 \quad (5)$$

Momentum

$$u_2 \frac{\partial u_2}{\partial x_2} + v_2 \frac{\partial u_2}{\partial y_2} = \frac{\partial}{\partial y_2}(\nu_{2,e} \frac{\partial u_2}{\partial y_2}) + g\,[\beta(T_2 - T_\infty) + \beta^* (W_2 - W_\infty)] \quad (6)$$

Energy

$$u_2 \frac{\partial T_2}{\partial x_2} + v_2 \frac{\partial T_2}{\partial y_2} = \frac{\partial}{\partial y_2}(\alpha_{2,e} \frac{\partial T_2}{\partial y_2}) \quad (7)$$

Vapor Mass Concentration

$$u_2 \frac{\partial W_2}{\partial x_2} + v_2 \frac{\partial W_2}{\partial y_2} = \frac{\partial}{\partial y_2}(D_{2,e} \frac{\partial W_2}{\partial y_2}) \quad (8)$$

Entrance Condition ($x_2 = 0$)

$$u_2 = u_{2,\,inlet},\ v_2 = 0,\ T_2 = T_\infty,\ W_2 = W_\infty \quad (9)$$

Adiabatic Wall Condition

$$y_2 = b\ ;\ u_2 = v_2 = 0,\ \frac{\partial T_2}{\partial y_2} = \frac{\partial W_2}{\partial y_2} = 0 \quad (10)$$

The solutions from the liquid and the gas side must satisfy the following interface conditions:

Continuity Conditions

$$\text{velocity} \quad u_1 = u_2 = u_I,$$
$$\text{temperature} \quad T_1 = T_2 = T_I,$$
$$\text{shear stress} \quad \tau_1 = \tau_2 = \tau_I \quad (11)$$

Thermodynamic Equilibrium Conditions

$$q_I = \left[-k_1 \frac{\partial T_1}{\partial y_1} \right] = q_{sensible} + q_{latent}$$

$$= \left[-k_{2,e} \frac{\partial T_2}{\partial y_2} \right] + m_I\, h_{fg}\ , \quad (12)$$

$$m_I = \rho_s v_I\ , \quad (13)$$

$$v_I = -\frac{D_{2,e}}{1 - W_2} \frac{\partial W_2}{\partial y_2}\ ;\ y_2 = 0\ . \quad (14)$$

3. Phenomena Simplification

3-1. New dependent variable

A new dependent variable combining temperature and mass concentration can be formulated in this specific problem. In a turbulent flow situation, the Reynolds analogy between turbulent heat and momentum transfer can be readily extended to turbulent mass transfer. Thus Lewis number unity assumption is more valid[7]. The values of the thermal diffusivity and the mass diffusion coefficient in turbulent flows are the same and both the energy and vapor mass concentration equations are homogeneous. Thus one can conclude that if the boundary conditions of energy and mass concentration equations fall in the same class (e.g. Dirichlet or flux boundary condition), the profiles of the solution regardless of their absolute value should be the same. We will assume that the interface conditions for the energy and mass concentration equations have the same form but they will be verified later in this paper.

$$\theta_2 \equiv \beta(T_2 - T_\infty) + \beta^* (W_2 - W_\infty)$$
$$\equiv \beta'(T_2 - T_\infty) \quad (15)$$

3-2. Interface conditions

Interface conditions are the mathematical descriptions of physics at the air-water interface. In this problem, the physical quantities which appear in the interface conditions are u_I, τ_I, T_I, q_I, m_I, and W_I. Among the six quantities, only two are independent. If u_I is fixed, τ_I is automatically determined and vice versa. And if the interface temperature, T_I, is known, the quantities, q_I, m_I, and W_I are determined thermodynamically. For the gas-liquid flow interface problem, the effect of the interface shear stress is generally assumed to be negligible[4]. Thus in this problem, there remains only one independent quantity, T_I.

3-3. Duct flow

The PCCS geometry is a natural convection flow duct. The bulk temperature of mixed air increases as it moves upward so that the buoyancy forces are different according to the elevations. But in a duct flow situation, the mass flow rate at all elevations should be the same except for the negligibly small addition of water vapor. Thus with the buoyancy effect, the chimney effect, which is very similar to the forced convection phenomena, is expected. Thus duct height can be divided by the prevailing mechanisms; chimney effect dominant region and natural convection effect dominant region. As the chimney effect dominant height can be a scale, the minimum requirement for the experimental facility height can be formulated by comparing the forced and natural convection boundary layer expressions. The boundary layer expressions of natural convection and forced convection is formulated as:

$$(\delta_T)_{NC} \sim L_3 Ra_{L_3}^{-\frac{1}{4}} Pr^{-\frac{1}{4}} , \quad (16)$$

$$(\delta_T)_{FC} \sim L_3 Re_{L_3}^{-\frac{1}{2}} Pr^{-\frac{1}{2}} . \quad (17)$$

They represent the thermal distances between the heat-exchanging entities. The type of convection mechanism is decided by the smaller of the two distances, since the wall will leak heat to the nearest heat sink. Thus the natural convection dominant criterion is

$$(\delta_T)_{NC} < (\delta_T)_{FC} \cdot \quad (18)$$

From this criterion a length scale is formulated;

$$L_3 \sim \frac{u_{inlet}^2}{g\Delta\theta} \quad (19)$$

where $\Delta\theta \equiv \beta\Delta T + \beta^* \Delta W$. Using the reference data, the length of the air entrance region is estimated to be 1 m in order.

This means that the prevailing heat transfer mechanism in the air inlet and in the rest of the duct are different. Thus the test facility height should be high enough to prevent the entrance effects.

Table 2. Reference and input data[3].

α=0.16 cm^2/s	L=2- m
ν=0.48 cm^2/s	Δ=0.3 mm
$\beta \sim 1/T_\infty$=0.003	$\Delta T \sim 75°C$
$\beta^* \sim M_{air}/M_{vap}$-1	$\Delta W \sim 0.5353$
=0.6081	$u_{inlet} \sim 2$ m/s
g=9.8 m/s^2	T_∞=15°C
D~3×10^{-5} m^2/s	T_w:60~90°C

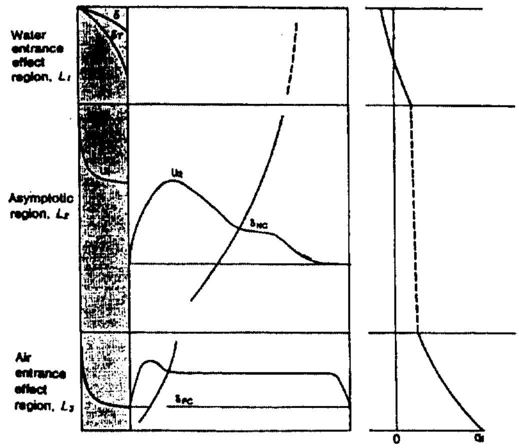

Fig. 4. Phenomenological separation of the calculation domain.

3-4. Asymptotic region

The film thickness is very thin compared to its height. This means that the velocity and temperature profiles of the water film will be quickly fully developed. Also due to the duct flow situation of the air flow, there must be two length scales in the water flow direction. This situation is presented in Fig. 4. The length scale of the water entrance effect region can be calculated by the fully developing distance i.e. the distance that the evolving boundary layer of the water flow approaches the film thickness.

$$L_1 \sim \frac{u_{max}\Delta^2}{\alpha_1} \sim \frac{g\Delta^4}{2\alpha_1\nu_1} \quad (20)$$

This length scale is estimated to be less than 1 m. Hence from the two length scales, one can conclude that the length of the asymptotic region is estimated to be over 90% of the whole domain.

3-5. Vapor momentum

It is well known that the momentum contribution in the mixed air flow due to water vaporization can be neglected in view of the mass transfer analogy. Thus the blowing velocity at the interface was not considered in the analysis.

4. Flow Evolution Characteristics

In the derivation of the similarity, interface temperature condition is a key parameter but it is also an unknown in this problem. Nevertheless one can expect that the interface temperatures are determined by the physical characteristics of each flow and also have a trend along the flow direction. Hence in this analysis, we assumed the interface temperature trend along the flow direction and then derived relevant interface temperature condition physically based upon each flow characteristics.

4-1. Water film flow evolution characteristics

The momentum equation for the water film is an ordinary differential equation and the boundary condition on heated wall is known and the interface condition can be assumed to be a free shear stress condition due to the previous reasoning on liquid-gas interface. Thus the solution is

$$u_1(y_1) = -\frac{g}{2\nu_1} y_1{}^2 + \frac{g\Delta}{\nu_1} y_1 . \tag{21}$$

Since the maximum-minimum principle is valid in the heat equation[8], the analytic solution of the heat equation using the maximum velocity is possible with the unknown interface condition. The assumed interface temperature condition can be written as;

$$T_t(x_2) \sim x_2^n . \tag{22}$$

Thus the analytic solution for the heat equation is

$$\overline{T_1} = 1 - \overline{y_1} + P\overline{y_1}(L_2 - x_1)^n + P\sum_{j=1}^{\infty}\int_0^{x_1} \exp(x\pi^2 j^2 (x_1 - \lambda)) \frac{2n}{j\pi} \cos j\pi (L_2 - x_1)^{n-1} d\lambda \cdot \sin j\pi \overline{y_1} , \tag{23}$$

where

$$\overline{T_1} \equiv \frac{T_1 - T_\infty}{T_w - T_\infty} , \quad \overline{y_1} \equiv \frac{y_1}{\Delta} . \tag{24}$$

4-2. Buoyant air flow evolution characteristics

Using the new dependent variable, the governing equations for the buoyant air flow are transformed as follows;

$$\frac{\partial u_2}{\partial x_2} + \frac{\partial v_2}{\partial y_2} = 0 , \tag{25}$$

$$u_2 \frac{\partial u_2}{\partial x_2} + v_2 \frac{\partial u_2}{\partial y_2} = \nu_{2,e} \frac{\partial^2 u_2}{\partial y_2^2} + g\theta_2 , \tag{26}$$

$$u_2 \frac{\partial \theta_2}{\partial x_2} + v_2 \frac{\partial \theta_2}{\partial y_2} = p_{2,e} \frac{\partial^2 \theta_2}{\partial y_2^2} , \tag{27}$$

where $p_{2,e} = \alpha_{2,e} = D_{2,e}$.

The self-preservation analyses are possible. This means that if we change the coordinate systems from x_2-y_2 to x_2-y_2/δ, where δ is the boundary layer thickness, the solutions can be divided by the amplitude function which depends only on x_2 and the profile function which depends only on y_2/δ. Substituting the following equations for the transformed equations, the balance between each quantity reveals.

$$u_2(x_2, y_2) = A(x_2)\, f(\eta) , \tag{28}$$

$$v_2(x_2, y_2) = B(x_2)\, h(\eta) , \tag{29}$$

$$\theta_2(x_2, y_2) = T^*(x_2)\, k(\eta) , \tag{30}$$

$$\text{where } \eta = \frac{y_2}{\delta(x_2)} . \tag{31}$$

The results are

$$\delta \sim x_2^m , \quad B \sim x_2^{-m} , \quad A \sim x_2^{1-2m} , \quad T^* \sim x_2^{1-4m} . \tag{32}$$

The above results reveal the relationship between all the physical quantities and the assumed interface condition (22).

3-3. Interface temperature condition

As denoted in Fig. 5, the analytic solution of the energy equation of the water film shows the interface heat flux behavior with respect to x_1 and n;

$$q_{I,1} = -\frac{1}{P} + (L_2 - x_1)^n + \sum_{j=1}^{\infty}\left(-\int_0^{x_1} \exp[-x\pi^2 j^2 (x_1 - \lambda)]\, d\lambda\right) \cdot 2n (L_2 - x_1)^{n-1} \tag{33}$$

Meanwhile from the self-preservation analysis of the buoyant air flow, the interface heat flux should be

$$q_{I,2} \sim (L_2 - x_1)^{\frac{5}{4}n - \frac{1}{4}} = Q (L_2 - x_1)^{\frac{5}{4}n - \frac{1}{4}} . \tag{34}$$

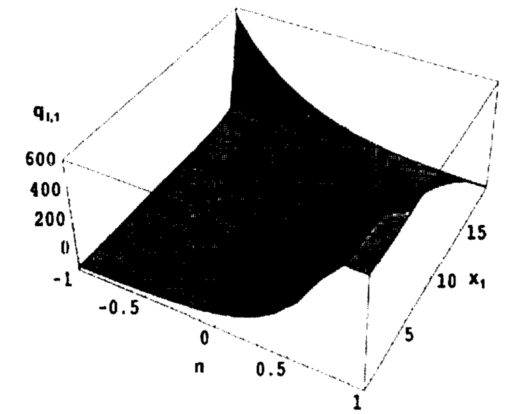

Fig. 5(a). Behavior of interface heat flux w.r.t x_1 and n from the analytic solution of water film.

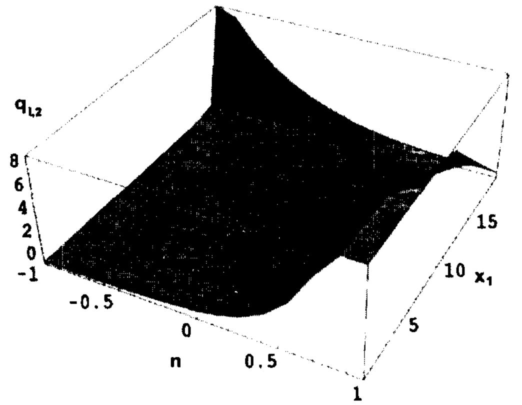

Fig. 5(b). Behavior of interface heat flux w.r.t. x_1 and n from the self-preservation of buoyant air flow.

where P, Q are unknown constants, defined at the arbitrary x_1, where the water film temperature is nearly linear.

As in Fig. 5, the two interface heat flux behaviors are by chance very similar.

The above two equations should be simultaneously satisfied at the interface. Equating the two equations at the interface, the interface temperature condition can be estimated.

$$\frac{T_w-T_\infty}{T_w-T_I}-\frac{T_I-T_\infty}{T_w-T_I}\left\{(L_2-x_1)^n+\sum_{j=1}^{\infty}(\ldots)\right\}$$

$$=(L_2-x_1)^{\frac{5}{4}n-\frac{1}{4}} \tag{35}$$

where $(\ldots)=\int_0^{x_1}\exp(x\pi^2 j^2(x_1-\lambda))\,2j\,(L_2-x_1)^{n-1}\,d\lambda$

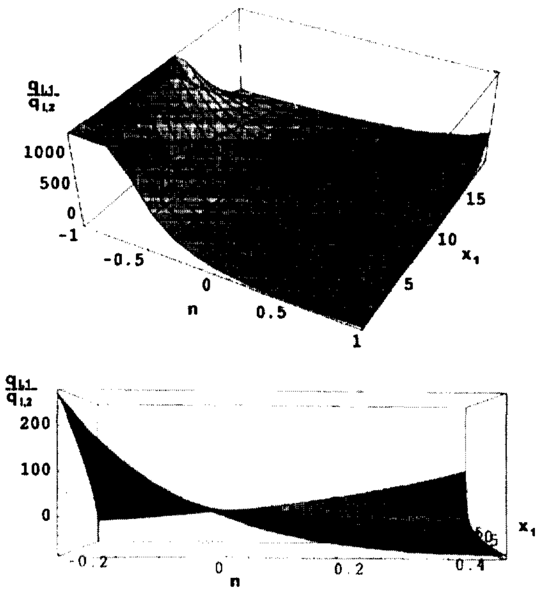

Fig. 6. Behavior of interface heat flux ratio w.r.t. x_1 and n.

Figure 6 shows the interface heat flux ratio with respect to x_1 and *n*. Regardless of x_1, the interface heat flux ratio should be unity in order to simultaneously satisfy both water and air flow characteristics. From Fig. 6, one can conclude n should be 0 i.e. Interface temperature should be constant. Since this condition is satisfied, the above assumption on the new dependent variable, θ, is proved to be reasonable.

4. Similarity

Based on the above analyses, since the interface temperature is constant in the asymptotic region, the buoyant mixed air flow in the PCCS and the natural convection air flow are governed by the same class of equations and boundary conditions. In other words, the equation (25)~(27) have the same form as the governing equations and boundary conditions for a buoyant air flow system except for using B' and k' instead of β and k. Therefore the two systems are similar. In the combined heat and mass transfer phenomena interface heat flux are expressed by

$$q_I=-k'\left(\frac{\partial T_2}{\partial y_2}\right)_I \tag{36}$$

where

$$k' \equiv -\left(k_{2,e}+\rho_s D_{2,e} h_{fg}\frac{1}{1-W_{2,I}}\frac{W_{2,I}-W_{2,\infty}}{T_{2,I}-T_{2,\infty}}\right)$$

And buoyance force term expression is

$$g\,\theta_2 = g\,\beta'(T_2 - T_\infty) \tag{37}$$

where $\beta' \equiv \beta+\beta^{*}\left(\frac{W_2-W_\infty}{T_2-T_\infty}\right)_I$

If the parameters are properly absorbed in the transformation procedure, both systems are governed by one set of equations. This means that the solution for each system is identical in form. Thus one can conclude that if β' and k' are used in place of β and k in buoyant air flow solution and/or correlation, the interface heat flux for the mixed buoyant air should be estimated.

5. Verification

Using these newly defined variable and coefficients, the interfacial heat fluxes and the ratios of latent heat transfer to sensible heat transfer are computed using McAdams correlation, which is one of the adopted correlations in WH-GOTHIC[9]. It is well known that the heat transfer from the water film is governed by the gradient of the air flow temperature profile and the adiabatic wall mainly affects the tail part of the temperature profile. In this sense, it is a proper natural convection heat transfer correlation for this Ra number range.

Substituting the equivalent conductivity and equivalent buoyant coefficient to McAdams correlation,

$$Nu = 0.13\,(Gr\cdot Pr)^{1/3} \tag{38}$$

or

$$h = k'\times 0.13\left(\frac{g\beta'\Delta T}{\alpha\nu}\right)^{1/3}. \tag{39}$$

The interface heat flux expressions from water and air are:

$$q_1 = -k_1\frac{T_W-T_I}{\Delta},$$

$$q_2 = h(T_I)(T_I - T_\infty)\,. \tag{40}$$

Equating the above equations, the interface temperature and heat flux are calculated. Also the ratio of latent and sensible heat transfer can be calculated by

$$\frac{q_{latent}}{q_{sensible}} = \frac{\left(\frac{\rho_s D_{2,e} h_{fg}}{1-W_{2,I}}\frac{W_{2,I}-W_{2,\infty}}{T_{2,I}-T_{2,\infty}}\right)}{k_{2,e}} \tag{41}$$

To verify the accuracy of the modified correlation, the results of this study are to be compared with other studies. But in view of the Rayleigh number expression, the containment height is the only parameter that affects Rayleigh number in the third power. The Rayleigh number in this system is larger than 10^{13}. To achieve the similar turbulence intensity and to overcome the entrance effect of water film and air, the experimental facility height scale should be the same scale, in order 10 m. No other method, say increasing ΔT, will be sufficient in achieving similar Rayleigh number. Also the questions on whether the thin water film covers the whole containment outer vessel and how to prevent the non-wetting area, are not clearly solved yet.

Due to the present lack of relevant experimental data, the results of this study should be compared with the results of numerical analyses. The results of the numerical study in reference 3 and the data provided by personal contact with author, Dr. Kang were used in comparison. The study in reference 3 adopted the same governing equations and boun-

Table 3. Comparisons of the results from scale analysis numerical method.

(a) Interface heat flux (W/m^2)

T_w (°C)	Scale analysis	Numerical method	Relative error (%)
70	3781	4442	-14.9
80	6902	6573	5.0
90	12730	11753	8.3

(b) Ratio of latent to sensible heat transfer

T_w (°C)	Scale analysis	Numerical method	Relative error (%)
60	6.8	7.2	5.4
70	9.3	9.7	4.2
80	12.9	12.6	2.2
90	18.5	18.1	2.2

dary conditions and were performed for the same configuration. But in this verification only constant wall temperature condition and 0.06 kg/m.s film flow rate condition were considered. As denoted in Table 3, the results are well within the range of 15% difference.

6. Conclusion

The similarity between the PCCS phenomena, which contains the sensible and latent heat transfer, and the buoyant air flow on a vertical heated plate was derived. This means that if we know the solution (or correlation) for the buoyant air flow system, we can use the solution (or correlation) also for the AP 600 PCCS in the asymptotic region. In this sense, this interpretation model is very simple to use and to apply in a similar situation such as water drop covers the whole vessel without running down.

In the course of the study, three length scales were derived - L_1, L_3, and 10 m - two length scales reflect the air and water entrance effect and one the turbulent intensity. Using the length scales, one can design an experimental facility for his/her purpose.

This study also has some restriction. If the containment wall temperature is so high that the water film dries out running down, and if the water film thickness is so large that the entrance effect becomes too large to provide asymptotic region the similarity might not be established. And also wave effects are not included in this study.

Mathematically the similarity for the constant wall heat flux condition can be shown. But physically, due to the drying-out possibility, the result is less feasible. Surely in a containment situation, the constant temperature assumption is more likely than the constant wall heat flux assumption.

Based upon the comparison with another calculations, the results of this study are well within the range of 15% difference, which is reasonable for this analysis. For the most probable operating conditions, the error ratios are nearer 5%. This interpretation model is therefore considered to reflect the PCCS phenomena reasonably well.

Table of Nomenclature

b	: gap thickness
D	: diffusion coefficient
F	: nondimensionalized stream funtion
g	: gravitational acceleration
h	: heat transfer coefficient
h_{fg}	: latent heat of evaporation
k	: coductivity
L	: containment height
L_1	: length of film entrance region
L_2	: length of the asymptotic region
L_3	: length of water entrance region
M	: molecular weight
m	: mass transfer rate
P, Q	: unknown coefficient
p	: pressure
q	: heat flux
T	: temperature
$\overline{T}$	: nondimensionalized temperature
u	: x velocity component
v	: y velocity component
x	: longitudinal coordinate
y	: transversal coordinate
W	: mass fraction of vapor
Ra	: Rayleigh number
Re	: Reynolds number
Pr	: Prandtl number
Nu	: Nusselt number
α	: thermal diffusivity
β	: thermal expansion coefficient
β^*	: concentration expansion coefficient
δ	: boundary layer thickness
μ	: viscosity
ν	: kinematic viscosity
η	: similarity variable
Δ	: water film thickness
τ	: shear stress
θ	: modified temperature
ρ	: density
A,B,T*(x)	: amplitude function for u, v, and Θ
f, h, k(η)	: profile function for u, v, θ
ρ	: density
1	: water film
2	: mixture air
a	: air

f : film
I : interface
inlet : inlet
e : effective coefficient
v : vapor
w : wall
x : local value
∞ : ambient
FC : forced convection
NC : natural convection

References

1. H.J. Brushi, "AP-600 Overview", Westinghouse Nuclear Plant Program (1991).
2. Y. Jaluria, "Natural Convection Heat and Mass Transfer", Pergamon, Oxford (1980).
3. H. Kang, J. Song and U. Lee, "Numerical Analysis of Cooling Characteristics By Water Film in Passive Containment Cooling System", Int. Conf. on New Trends in Nuclear System Thermohydraulics, Pisa, Italy (1994).
4. R.B. Bird, W.E. Stewart and E.N. Lightfoot, "Transport Phenomena", John Wiley & Sons Inc. (1960).
5. H. Chiang and C. Kleinstreuer, "Analysis of Passive Cooling in a Vertical Finite Channel Using a Falling Liquid Film and Buoyancy-Induced Gas-Vapor Flow", Int. J. Heat Mass Transfer, Vol. 34, No. 9, 2339-2349 (1991).
6. A. Koestel, R.G. Gido and J.S. Gilbert, "Film Entrainment and Drop Deposition for Two-Phase Flow", NUREG/CR-1634.
7. William M. Kays and Michaele E. Crawford, "Convective Heat and Mass Transfer 2nd edition", McGraw-Hill Book Company (1980).
8. P. Duchateau and D.W. Zachmann, "Theory and Problems of Partial Differential Equations", Schaum's Outline Series in Mathematics, McGraw-Hill Book Company (1986).
9. M.D. Kennedy, J. Woodcock, R.F. Wright, and J.A. Gresham, "Westinghouse-GOTHIC Comparisons with 1/10 Passive Containment cooling Tests", Int. Conf. on New Trends in Nuclear System Thermohydraulics, Pisa, Italy (1994).