Journal of the Korean Nuclear Society
Volume 28, Number 6, pp. 532~541, December 1996

Development of Heat Transfer and Evaporation Correlations for the Turbulent Natural Convection in the Vertical Channel by Using Numerical Analysis

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(Received March 14, 1996)

Abstract

Theoretical and numerical study on heat transfer and evaporation in the vertical channel has been carried out and basic correlations have been derived for the heat transfer evaluation of PCCS. Analysis program was developed with low-Reynolds-number k-ε model and surface transfer rates were calculated for the turbulent natural convection in the vertical channel. In relation to dry cooling by buoyancy-driven air, first, the system parameters which govern overall heat transfer rate are determined through the adequate nondimensionalization procedure. After comparison with existing experimental data, numerical results are used to derive heat transfer correlation by sensitivity calculations. In relation to wet cooling by falling water film, numerical analysis are carried out for evaporation process with real film surface conditions and evaporation correlation is derived through analogy concept and correction factors.

1. Introduction

The containment systems of existing plants have been constructed to maintain its integrity against any possible thermal and pressure load based on the DBA(Design Based Accidents). But after TMI/2 and Chernobyl accidents, new researches were performed for the severe accidents and the significant improvements of safety systems were demanded for future plants. In the recent years, particular attention was devoted to the intrinsic and passive characteristics of nuclear power plants which was credited to increase reliability and public safety. Natural convection cooling in the double containment is one of the possible concepts for the future containment systems.

Two practical candidates for PCCS(Passive Containment Cooling System) of PWR are the containment cooling (1)by buoyancy-driven air and (2)by evaporative water film. These concepts are adopted in Westinghouse's AP600, Ebasco's HWRF, B&W's ASPWR, and KfK's composite containment[1,2].

Westinghouse carried out several experiments as a confirming procedure of their PCCS design. Some experiments were performed for specific issues such as heat transfer and evaporation coefficients, flow resistance, and wetting fraction. While others were for integral system performance[1]. Ambrosini measured heat transfer and evaporation coefficients in the vertical channel[3]. Westinghouse and Ambrosini used blowers to compensate scale-down effect.

But the heat transfer characteristics of buoyancy-driven flow is different from that of forced flow. Until now, general heat transfer and evaporation correlations for the turbulent natural convection in the vertical channel were not developed. So Ebasco used Sparrow's laminar correlation in their PCCS model with so large conservatism[2,4]. Evaporation correlations were derived from the heat transfer correlations by analogy concept between heat and mass transfer in the above studies.

Experiments for the turbulent natural convection heat transfer in the vertical channel were carried out by Siegel, Hugot, and Miyamoto[5,6,7]. They measured heat transfer coefficients with varying channel width and fixed height. Siegel and Hugot suggested overall heat transfer correlations using only one length scale. Miyamoto obtained local Nu number correlation which was well fitted to data near channel outlet. Two length scales were known to dominate natural convection heat transfer from two vertical plates together[4,5]. Application of the suggested correlations to other aspect ratio(S/H) can induce a few errors. The published data points are insufficient for general heat transfer correlations with two length scales yet. Chang studied numerically the laminar naturally-driven flow in the channel with fully elliptic equations and large reservoir[8]. Cheung did the turbulent flow with boundary layer equations and low-Reynolds-number k-ε model[9].

The purpose of this study is to develop theoretical model for turbulent natural convection in the vertical channel and to derive heat transfer correlations based on numerical results and existing experimental data. Also numerical analysis is carried out for evaporation process with real film surface conditions and evaporation correlation is derived through analogy concept and correction factors. More detailed existing studies on turbulent natural convection heat transfer and evaporation in the vertical channel are described in the reference [10].

2. Theoretical Modeling

In the real annular structure of PCCS, the gap size between steel vessel and concrete wall is very small compared with the diameter of containment vessel. So the effect of curvature can be ignored and the configuration of calculation domain be approximated by a vertical channel, as shown in Fig. 1. Uniform temperature boundary condition is imposed on the left side and adiabatic condition on the right wall in most cases. The left side corresponds to steel wall for heat transfer problem and to film surface for evaporation problem. Heat transfer or evaporation from left side makes nearby gas density low. Light gas flows up by buoyancy.

2.1. Governing Equations and Boundary Conditions

2.1.1 Heat Transfer Problem

Governing equations are 2-D elliptic type which do not have any assumption of flow profiles. To formulate the problem, following assumptions are employed.

- All physical processes are two-dimensional and taking place at steady state.
- The induced flow is turbulent at inlet. In most cases, the Re number of air is above transition point.

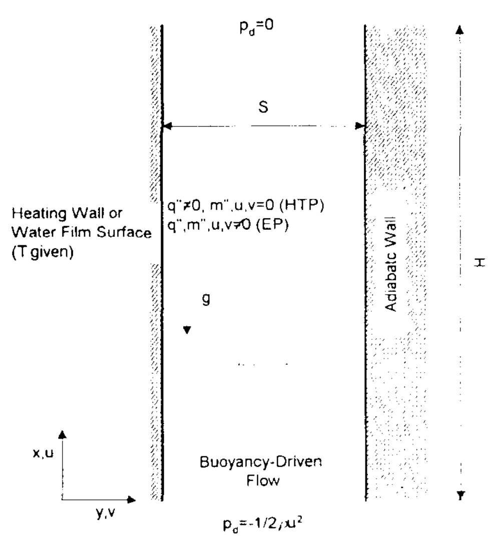

Fig. 1. Calculation Domain and Boundary Conditions

- Boussinesq assumption are not used and all air properties are calculated as the function of temperature.
- pressure work and energy dissipation are negligible.

With the above assumptions, equations governing the conservation of mass, momentum, and energy of air in the channel are given by

$$\frac{\partial}{\partial x_j}(\rho u_j) = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}(\rho u_j u_i) = -\frac{\partial p_d}{\partial x_i} + (\rho - \rho_\infty) g_i + \frac{\partial}{\partial x_j}\left\{(\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\rho k \delta_{ij}\right\} \tag{2}$$

$$\frac{\partial}{\partial x_j}(\rho c_p u_j T) = \frac{\partial}{\partial x_j}\left\{\left(\lambda + \frac{c_p \mu_t}{Pr_t}\right)\frac{\partial T}{\partial x_j}\right\} \tag{3}$$

where p_d is the difference between the local pressure in the channel and the ambient pressure at the same elevation, that is, $p_d = p(x) - p_\infty(x)$. Pr_t is turbulent Prandtl number. A constant value of 0.9 was employed for it in most existing natural convection studies[11]. Same value is for it in the present study. The other symbols are described in the nomenclature.

perature at the inlet, outlet, and both walls are

$$x = 0 : u = u_{in}, T = T_{in}, p_d = -\frac{1}{2}\rho u_{in}^2 \tag{4a}$$

$$y = 0 : u = v = 0, T = T_w \tag{4b}$$

$$y = S : u = v = 0, \frac{\partial T}{\partial y} = 0 \tag{4c}$$

$$x = H : \frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial T}{\partial x} = 0, p_d = 0 \tag{4d}$$

where uniform wall temperature conditions are employed at the heated wall except the calculations for uniform wall heat flux experiments. Adiabatic condition is used at the right wall. Inlet pressure $-1/2\rho u_{in}^2$ is the value needed for stationary ambient air to accelerate to inlet velocity u_{in}, which was suggested by Aihara[12].

(b) Evaporation Problem

In this study, only gas region is solved numerically with given film surface conditions. The governing equations and most boundary conditions for the heat transfer are applied to evaporation with left side regarded as film surface. The properties of air/vapor mixed gas are the mass-averaged values for each nodes. Following equation for vapor mass fraction W is solved with Eq. 1, 2, 3.

$$\frac{\partial}{\partial x_j}(\rho u_j W) = \frac{\partial}{\partial x_j}\left\{\left(\rho D + \frac{\mu_t}{Sc_t}\right)\frac{\partial W}{\partial x_j}\right\} \tag{5}$$

Sc_t is turbulent Schmidt number and has the same value with Pr_t. The u velocity of film surface is given by Nusselt theory from film mass flow rate. Vapor mass fraction and blowing velocity at the film surface are given by

$$W_{int} = \left.\frac{M_v p_v}{M_v p_v + M_a(p_{tot} - p_v)}\right|_{y=0} \tag{6}$$

$$v_{int} = -\left.\frac{D}{(1-W)}\frac{\partial W}{\partial y}\right|_{y=0} \tag{7}$$

Eq. 6 is derived from Dalton's partial pressure law and the state equation of ideal gases. Partial pressure of vapor, p_v is obtained from saturation curve with input film surface temperature. The solubility of air for water is neglected in Eq. 7. The heat flux from the film surface is composed of sensible heat transfer and evaporation.

$$\ddot{q}_{int} = \ddot{q}_c\big|_{y=0} + \ddot{q}_e\big|_{y=0} = -\lambda\left.\frac{\partial T}{\partial y}\right|_{y=0} + \rho v h_{fg}\big|_{y=0} \tag{8}$$

Evaporation occupies most part of overall surface heat flux and the ratio increases with temperature. Whereas sensible heat transfer and evaporation have effect on the driving force of flowing gas together and any mechanism can be neglected.

2.2. Turbulence Model

Eddy viscosity μ_t is introduced to reflect turbulent effect in the above equations. Additional equations are needed to compute local variation of eddy viscosity μ_t and other variables. Standard k-ε turbulence model which has been generally used employs logarithmic wall function. Henkes compared the several k-ε turbulence models for the natural convection boundary layer and reported that erroneous heat flux were predicted with the standard model[9]. So low-Reynolds-number k-ε model which does not need wall function is employed in this study. After reference search for the existing k-ε turbulence models, Jones/Launder's model was adopted, which showed excellent prediction especially for natural convection heat transfer[13]. This model includes six experimental constants and original values are used in this study for both heat transfer and evaporation problems without any change.

Eddy viscosity μ_t is expressed as following

$$\mu_t = C_\mu F_\mu \rho \frac{k^2}{\varepsilon} \tag{9}$$

Turbulent kinetic energy k and turbulent energy dissipation ε is given by

$$\frac{\partial}{\partial x_j}(\rho u_j k) = \frac{\partial}{\partial x_j}\left\{\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right\} + P + G - \rho\varepsilon - 2\mu\left(\frac{\partial k^{1/2}}{\partial x_j}\right)^2 \tag{10}$$

$$\frac{\partial}{\partial x_j}(\rho u_j \varepsilon) = \frac{\partial}{\partial x_j}\left\{\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right\} + \frac{\varepsilon}{k}(C_{\varepsilon 1}F_1 P - C_{\varepsilon 2}F_2\rho\varepsilon) + C_{\varepsilon 3}\frac{\varepsilon}{k}G + 2\mu\mu_t\left(\frac{\partial^2 u}{\partial x_i^2}\right)^2 \tag{11}$$

Inlet k value is expressed as turbulence intensity and inlet ε value is obtained from the k value assuming equilibrium state between turbulence generation and dissipation.

2.3. Nondimensionalization of Governing Equations and Form of Correlations

2.3.1. Heat Transfer Problem

The governing equations are nondimensionalized by using following nondimensional numbers which are referenced to Chang's and Ozoe's studies[14,15].

$$X = \frac{x}{H},\quad Y = \frac{y}{S},\quad A = \frac{H}{S},\quad U = \frac{u}{\alpha/S}$$

$$V = \frac{v}{\alpha/S},\quad P = \frac{p_d}{\rho\alpha^2/S^2},\quad \Theta = \frac{T - T_{in}}{T_w - T_{in}},$$

$$Ra_S = \frac{g\beta(T_w - T_{in})S^3}{\nu\alpha},\quad K = \frac{k}{(\alpha/S)^2},$$

$$E = \frac{\varepsilon}{\alpha^3/S^4},\quad \mu_t^* = \frac{\mu_t}{\mu} \tag{12}$$

In the nondimensional governing equations, parameters which represent the system are Ra_S and aspect ratio(H/S). If these parameters are given, the profiles of all other variables, Nu number, and friction factor are determined uniquely.

In the case of natural convection near vertical plate, there is only one independent length scale, height H. Nu number depends on only Rayleigh number Ra_H in the most heat transfer correlations and the form is $Nu_H = C\ Ra_H^m$. Whereas there are two independent length scale, that is, height H and width S in case of the natural convection from vertical channel. Nu number is expressed in Ra_S and aspect ratio H/S in most existing laminar heat transfer correlations as following[3] :

$$Nu_S = C\ Ra_S^{m_1}\left(\frac{S}{H}\right)^{m_2} \tag{13a}$$

$$\text{where}\ \ h = q''/(T_w - T_{in}) \tag{13b}$$

Laminar heat transfer correlation derived by Elenbaas has unit exponents, m1 and m2 values for the thin channel[15]. The exponents have same values in the correlations derived by Sparrow and Chang[3,8]. Though Sigel expressed his turbulent data by using one length scale, Kreith recommended that the experimental data were more adequately fitted to Eq. 13 type of correlation[4].

The turbulent natural convection heat transfer correlation which is the subject of this study can be expressed in the form of Eq. 13 but the same value of m1 and m_2 is not expected. The coefficients and each exponents will be determined based on numerical results and experimental data. Heat transfer coefficient is defined with air average temperature T_m at the starting elevation of developed region in the correlation.

2.3.2. Evaporation Problem

Instead of α, diffusivity of vapor D is used for nondimensional numbers of the evaporation problem. Ra' number and aspect ratio determine Sherwood number and the relation can be expressed in the following equation.

$$Sh_S = C\ Ra_S'^{m_1} \left(\frac{S}{H}\right)^{m_2} \quad (14)$$

where the density difference of Ra number is composed of temperature and vapor mass fraction difference. Sh number is defined as

$$Sh_S = \frac{m''S}{\rho D} \frac{(1 - W_{int})}{(W_m - W_{int})} \quad (15)$$

The coefficients and each exponents are determined in the same manner with the case of heat transfer.

2.4. Numerical Solution Method

One major requirement in the numerical prediction of the turbulent flow is that fine grids are needed near the wall. This is especially true when using the low-Reynolds-number turbulence models. To save computation time and to improve the accuracy in predicting the near-wall turbulence behavior, fine grid spacing is used in the near-wall regions, whereas the grids in the central region of the channel are arranged in such a way that the grid spacing increases toward the centerline.

Control volume scheme and staggered grids system are used for the discretization of the governing equations and boundary conditions[16]. The control volume scheme has advantage for the conservation of mass and momentum, compared to Taylor series expansion. The staggered grids system is used to avoid pressure fluctuation during converging sequence.

The governing equations are coupled nonlinearly with each other and many iterations are needed to obtain final converging solution. SIMPLER(Semi-Implicit Methods for Pressure Linked Equation Revised) suggested by Patankar is employed to couple pressure and velocities equations[16]. CGM(Conjugate Gradient Method) is used to solve the matrix equation for pressure and MSI(Modified Strongly Implicit) for the other equations[17]. Numerical solutions have the possibility to diverge during the iterations. Under-relaxation factors have values between 0.3 and 0.8. If a large number of grids are used, numerical solution often fluctuates periodically. Multi-grids method is employed to prevent the fluctuation.

3. Results and Discussion

3.1. Heat Transfer Problem

The thickness of viscous sublayer is about y/η_T = 1.7 as described by George/Capp in the turbulent natural convection boundary layer[18]. Its value is about same to $y^+ = 4$.

All solutions are obtained with first grid located wit-

Table 1. The Effect of Distance from Wall to First Node on Nu Number for Heat Transfer Problem (H=6m, S=0. 1m, T=40~160℃)

Ra	Nu/y^+			
	1.83E-4*	6.14E-5	3.32E-5	1.57E-5
4.25E6	10.60/0.86	10.37/0.15	10.35/0.07	–
8.51E6	14.02/1.09	13.22/0.36	13.15/0.09	–
1.28E7	–	15.57/0.40	15.38/0.21	15.27/0.10
1.70E7	–	22.85/0.47	18.61/0.24	17.54/0.11

* Distance of first node from wall

hin $y^+ = 0.5$ in this study. The calculated Nu numbers with different first grid locations are shown in Tab. 1. The table shows converging trends as first grid approaches the wall.

There exist a few experimental heat transfer correlations for laminar natural convection in the channel. To verify the developed program except turbulence part, the numerically obtained Nu numbers are compared with the value from Sparrow's and Bar-Cohen's laminar correlations, which shown in Fig. 2[19,20]. The numerically obtained Nu numbers agree to correlations very well in the figure. Hugot carried out turbulent natural convection experiment where both walls were heated in the condition of uniform temperature. This experiment is simulated numerically. Fig. 3 shows heat transfer coefficient trends at the different heights. The numerically obtained Nu numbers have same trends with experimental data but are located somewhat at the lower positions in the figure.

In the turbulent forced convection, temperature profiles near wall can be expressed in the logarithmic functions if they are nondimensionalized with friction velocities. Most heat transfer correlations for the forced convection were derived based on the profile assumption. Also the temperature values of the nearest grid from the wall are calculated by using the logarithmic function in the standard k- model. Fig. 4 shows two groups of profiles, that is, the calculated temperature profiles near the wall which are nondimensionalized with friction velocities and the logarithmic functions for the forced convection. In the viscous sublayer, the calculated profiles agree to the linear function very well as described by Siebers [21]. Whereas the calculated profiles deviate from the logarithmic function in the turbulent layer, which reveals the heat transfer characteristics of buoyancy-driven flow is different from that of forced flow.

The coefficient and exponents of Eq. 13 are determined through following procedure. First, Nu numbers are calculated for different Ra numbers with aspect ratio fixed. Exponent, m1 is obtained from these calculations. Above calculations are performed with

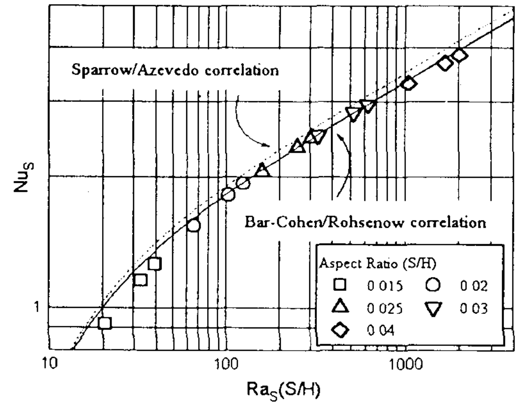

Fig. 2. The Calculated Nu numbers for Laminar Convection.

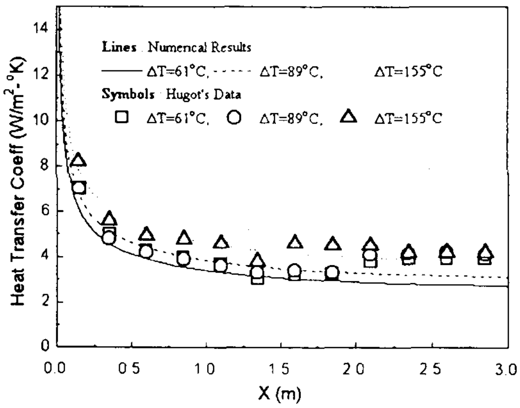

Fig. 3. Comparison of the Calculated Heat Transfer Coefficients and Experimental Data

other aspect ratio and then the value of m1 is confirmed to be same for other aspect ratio. Finally, the coefficient and exponent m_2 are fitted from the values of Nu/Ra^{m1}. Nu number is defined as the averaged value of the developed region. Calculation conditions are shown in Tab. 2. The internal condition of channel determines Re number of induced air flow in the natural convection. Fig. 5 shows the Re number as a function of Ra number and aspect ratio. Fitting function for the calculated data is

$$Re_{S'} = 0.471\ Ra_{S'}^{0.47} \left(\frac{H}{S'}\right)^{0.565} \tag{16}$$

where S'(2S) is used for convenience, instead of S and all properties are calculated at the ambient temperature. Fig. 5 also shows the experimental data of Miyamoto[7]. The experimental data are shown to deviate from the fitting function at the wide channel (H/S' = 12.5). Miyamoto described that the characteristics of flow in this width was close to that of natural convection near single plate.

Fig. 6 shows the calculated Nu numbers as a function of Ra number and aspect ratio. Fitting function for the calculated data is

$$Nu_{S'} = 0.187\ Ra_{S'}^{0.312} \left(\frac{S'}{H}\right)^{0.173} \tag{17}$$

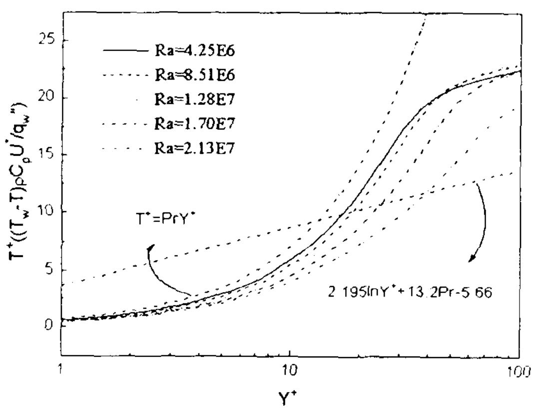

Fig. 4. Nondimensional Temperature Profiles Near Wall

Table 2. Conditions for Sensitivity Calculations

Reference Dimension	
Channel Height	3~15 m
Channel Width	0.05~0.3m
Wall Temperature	40~200℃ (HTP*)
	50~80℃ (EP**)
Ambient Temperature	20℃
Humidity of Ambient Air	50%
Dimensionless Groups	
Aspect Ratio(S'/H)	0.02~0.05
Ra Number	6E5~5E9(HTP)
	5E6~1E9(EP)
Pr number	0.71

* HTP : Heat Transfer Problem

** EP : Evaporation Problem

The experimental data of Hugot, Siegel, and Miyamoto are also shown in the figure. Hugot and Siegel carried out their experiments with uniform wall heat flux condition and used the modified Ra numbers, Ra*. Parameter conversion, $Ra_{S'}^* = Ra_{S'}Nu_{S'}$ is employed for their data. The calculated Nu numbers are distributed a little low compared to the experimental values but have same trends with them in the figure. Both of the calculated and the experimental Nu numbers show linear trend, which proves the adequacy of Eq. 13 for the turbulent natural convection in the channel. The coefficient C of correlation is changed to 0.22 with same exponents, if the experimental data are fitted.

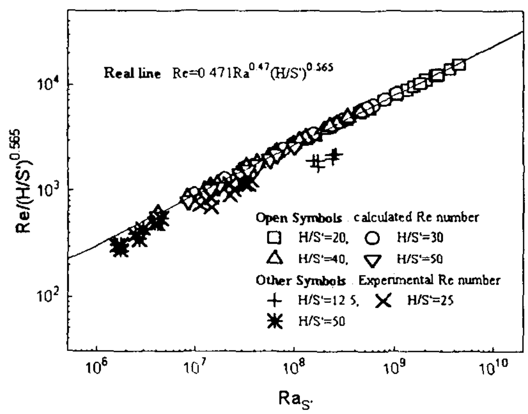

Fig. 5. Re Number of Induced Flow as Function of Ra Number

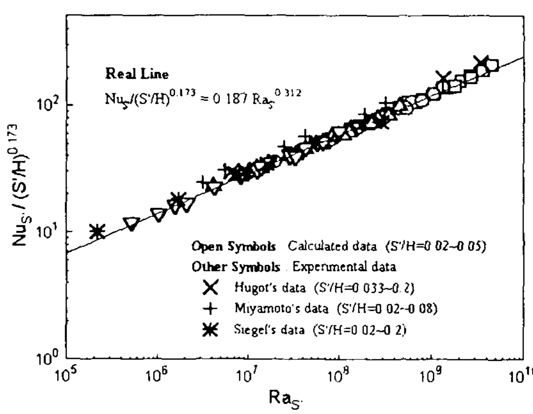

Fig. 6. Nu Number as a Function of Ra Number

3.2. Evaporation Problem

The procedure for heat transfer are performed for evaporation correlation with one additional variable of film mass flow rate. In this case, the left side has non-zero film velocity, which induces the decrease of incoming air mass flow rate. Also very small but non-zero blowing velocity from film surface has effect on evaporation rate. These two effect have to be included in the evaporation correlation. In this study, correction factor method is employed to reflect the effect.

Two categories of calculations are carried out. First category is with zero surface velocity and zero blowing velocity. Second category is with real surface condition for the given film mass flow rate. The purpose of first category calculation is the confirming of analogy between heat transfer and mass transfer. In this case, the introduction of correction factors is not needed. Correction factors are determined from the second category calculation. Fig. 7 shows the result, where ◇ symbols are obtained from first calculations and ● from second. The coefficient and exponents of evaporation correlation in the figure are same to those of heat transfer problem. The ◇ symbols are fitted by the correlation very well. This fact shows that the analogy methodology between heat and mass transfer can be used for the turbulent natural convection in the channel. ● symbols are $Sh_{S'}/(S'/H)^{0.173}$ divided by following correction factors.

$$\text{Blowing CF} = \frac{\ln(1+B)}{B},\ B = \frac{m''/\rho_g u_g}{St_{AB}} \quad (18a)$$

$$\text{Surface Velocity CF} = 0.819\left(\frac{Re_{film}}{Ra_{S'}^{0.47}}\right)^{-0.033} \quad (18b)$$

Eq. 18a expresses the decrease of evaporation by blowing effect and Eq. 18b reflects the effect of film surface velocity. The characteristic scale of Re_{film}, $Ra_{S'}^{0.47}$ is obtained from Eq. 16. The ● symbols are also fitted by the correlation very well in the figure.

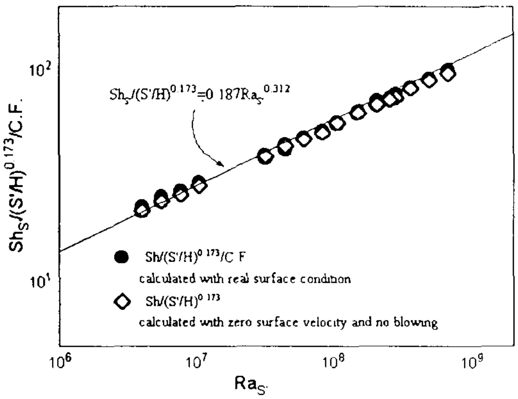

Fig. 7. Sh Number as a Function of Ra Number (film mass flow rate=0.05~0.15kg/m-sec, T_{int}=50~80℃)

4. Conclusions

Theoretical study on turbulent natural convection heat transfer and evaporation in the vertical channel using numerical analysis was performed and basic correlations have been derived for the heat transfer model of PCCS. Comparison between the numerically obtained results and the existing heat transfer correlations or experimental data showed it possible for heat transfer correlation to be determined from the developed program.

The system parameters that dominate transfer rate are derived through the nondimensionalization of governing equations. The coefficient and exponents of correlations are determined by sensitivity calculation for the system parameters. Nu numbers from the obtained correlation are a little lower than experimental data but show the same trend. Sh numbers from evaporation calculation with real film surface conditions very well correspond to the evaporation correlation obtained from analogy concept and correction factors. The developed correlations are for the naturally driven flow into the channel with ideal inlet/outlet boundary conditions and for fully wetted surface. Additional studies on friction increase in the real structure and wetting fraction of water film may be needed to apply the developed correlations to real containment systems.

Nomenclature

B parameter used in blowing correction
C_p specific heat
CF correction factor
D diffusion coefficient
G turbulent production by buoyancy
g gravitational acceleration
H channel height
h heat transfer coefficient
h_m mass transfer coefficient
h_{fg} latent heat of evaporation
k turbulent kinetic energy
M molecular weight
m'' evaporation flux
Nu Nusselt number, hS'/λ
P turbulent production by shear stress
p pressure
p_d dynamic pressure
Pr Prandtl number, ν/α
q'' heat flux
Ra Rayleigh number for heat transfer$(g\beta\Delta TS'^3)/\alpha\nu$
Ra' Rayleigh number for evaporation
$\{g(\beta^*\Delta W+\beta\Delta T)S'^3\}/D\nu$
Ra^* modified Rayleigh number,
$(g\beta q''S'^4)/\lambda\alpha\nu$
Re Reynolds number, uS'/ν
S channel width
S' 2S
Sc Schmidt number, ν/D
Sh Sherwood number, h_mS'/D
St_{AB} Stanton number for mass transfer
h_m/u
T temperature
u,v velocity components of x, y direction
W vapor mass fraction
x longitudinal coordinate
y transversal coordinate

Greek letters

α thermal diffusivity
β thermal expansion coefficient
η_T inner region length scale
$[(\nu/Pr)^2/(g\beta\Delta T)]^{1/3}$
ε turbulent dissipation rate
λ conductivity
μ viscosity
ν kinematic viscosity
ρ density

Subscripts

e evaporation
c convection
i,j coordinate index
in inlet
int interface
m average value
t turbulent
v vapor
w wall
∞ ambient condition

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