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A Proposed Model to Estimate Condensing Heat Transfer Coefficient in Steam-Air Mixture of Containment Atmosphere

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비응축성 가스(공기)가 존재하는 격납용기내에서 증기의 응축 열전달 계수평가에 관한 모델

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초 록

격납 용기 내에 비응축성 가스(공기)가 존재하는 경우에 증기의 응축 열전달 계수를 평가하는 방법을 연구하였다.

유일한 대규모 격납 용기 실험인 CVTR 자료를 이용하여 응축 열전달 계수를 계산하여, 현재 원자력 발전소의 냉각재 상실 사고(LOCA) 및 주 증기 배관 파열사고(MSLB) 시에 격납 용기의 안전 해석에서 공식적으로 사용되고 있는 Tagami와 Uchida 열전달 계수 관계식과 비교해 본 결과 좋은 일치를 보여 주었다.

Nomenclature

A : Heat sink surface area

A_f : Cross-sectional area for condensate flow

C : Constant used in equation(10)

C_p : Specific heat capacity at constant pressure

D_e : Hydraulic diameter

g : Acceleration of gravity

h : Heat transfer coefficient

$\bar{h}$: Average heat transfer coefficient

h_{fg} : Latent heat of evaporation

i : Specific enthalpy

k : Thermal conductivity

L : Characteristic length(height for vertical wall)

M : Mass

$\dot{m}$: Steam condensation rate

Nu : Nusselt number

P : Perimeter

Q : Total energy released

Re : Reynolds number

T : Temperature

ΔT : $T_\infty - T_w$, Temperature difference

t : Time

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U_m : Average velocity of condensate film
V : Containment volume
W : Mass fraction of noncondensable gas (air)
x : Coordinate
y : Mole percent of noncondensable gas (air)

Greek Symbols

ρ : Density
μ : Absolute viscosity
ν : Kinematic viscosity, μ/ρ

Subscripts

a : Air
cond : Condensation
f : Liquid film
g : Vapor
i : Interface
max : Maximum
Nu : Nusselt condensing (pure steam)
p : Peak
sat : Saturation
w : Wall
∞ : Bulk atmosphere

1. Introduction

The containment systems of large nuclear power plants are designed to withstand the pressure and temperature that could result from a loss of coolant accident(LOCA) or the major steam line break (MSLB)[1]. The pressure reducing effects of steam condensation on the pressure temperature response in a containment to a LOCA and MSLB is a subject of safety concern from the view of containment integrity. The containment atmosphere, in the case of a LOCA, is pressurized by the release of high energy coolant into the containment atmosphere, and this pressure rise imposes a potential threat to the structural integrity of the containment building which is the final barrier against the release of radioactive material to the environments.

Various computer codes within the nuclear industry are used to calculate the pressure and temperature transients in containment following a design basis accident (DBA)[2,3]. CONTEMPT-LT computer program is the only publicly available containment analysis code that accounts for both heat and mass removal due to condensation on a heat sink surface. The current approach of both industry and the U. S. Nuclear Regulatory Commission(NRC) has been to use the Tagami or Uchida condensing heat transfer coefficient correlations to calculate the heat removal rate due to heat sink condensation in containment building[1,6].

The peak containment pressure is highly dependent on the heat transfer coefficient between the containment atmosphere and heat absorbing structures. If the heat transfer coefficient used is too large, the temperature and pressure attained will be low. Therefore, from the safety point of view, the containment may be under-designed. On the other hand, if the heat transfer coefficient used in the containment transient calculation is conservatively low, the resulting pressure and temperature will be over-conservative (i.e., too high), thus, the containment building will be over-designed with unnecessary costs involved[4,5].

Heat is transferred from the hot atmosphere to the containment building structures and heat sinks by two important mechanisms: single phase convection and condensation. Radiation is not important because the existing temperature differences in the containment are relatively low. The contribution due to condensation is usually larger than the one due to single phase convection because condensation involves a phase change from vapor to liquid with high enthalpy change.

Two separate time periods are generally considered following a LOCA and MSLB. The first period is called the blowdown period, characterized by high turbulence in the atmosphere as the primary coolant system decompresses[6,7]. Forced convection, together with condensation of steam on the cold walls, are the major mechanisms for heat transfer during this period. The end point of this period is usually not defined explicitly but is the end of the pressurization of the containment resulting from the initial injection of primary coolant or secondary steam into the containment. The second period is referred to as the post-blowdown period, where natural convection and condensation are the main heat transfer mechanisms.

The proposed method in this study uses the technique presented by Carbajo and the recommendation of Rohsenow[4,8]. The numerical calculation using the proposed model is accomplished employing the data measured in Carolinas Virginia Tube Reactor (CVTR) containment test 3 [9].

2. Model Description

2.1. Film Condensation of Pure Vapors

Theoretical analysis of filmwise condensation was originally presented in 1916 by Nusselt. Laminar film condensation on a vertical plate has been actively studied since the pioneering analysis of Nusselt. The physical situation studied by Nusselt and in various succeeding investigations is the simplist of all problems in laminar film condensation: namely, a pure, quiescent, saturated vapor condensing on an isothermal vertical plate. Nusselt formulated the problem in terms of simple force and heat balances within the condensate film and derived the following expression for the heat transfer coefficient on a vertical plate.

$$Nu_x = \frac{h_x x}{k_f} = \left[\frac{g\rho_f(\rho_f-\rho_g)h_{fg}x^3}{4\mu_f k_f(T_g-T_w)} \right]^{1/4} \quad (1)$$

Sparrow and Gregg[10] have reformulated the problem using the mathematical techniques of boundary-layer theory, considering energy-convection and fluid-acceleration terms. The results of these studies showed that the prandtl number of the condensing vapor is a factor that influences the condensation heat transfer coefficient. The prandtl numbers for steam and other common engineering fluids lie between 1 and 10. In this range, the effect of inertia forces on heat transfer for practical purposes was quite small (negligible). But for condensation of liquid metals, the inertia force played an important role in decreasing the heat transfer coefficient below the Nusselt prediction as the parameter $C_p(T_g-T_w)/h_{fg}$ increases.

A comparison with the experiments has shown that the measured heat transfer coefficient is higher than those calculated from the Nusselt model of Eq. (1). This is believed to be mainly due to the behavior of the film. So to speak, pure film condensation is a rarity and some dropwise condensation may take place. Also, it is often found that the films contained ripples which helped liquid mixing in the film. These results in higher values of the heat transfer coefficient. Therefore, McAdams[11] suggested that Nusselt's theory for the average heat transfer coefficient for condensation on a vertical surface should be multiplied by 1.2. In this model, recommendation by Rohsenow[8] is used. He suggested to replace h_{fg} in Eq. (1) by $h_{fg}+0.68C_{pf}(T_g-T_w)$ Then, the local heat transfer coefficient for laminar film condensation over a vertical surface is

$$h_x = \left(\frac{g\rho_f(\rho_f-\rho_g)k_f^3\{h_{fg}+}{4\mu_f x(T_g-} \right.$$

$$\left.\frac{0.68C_{pf}(T_g-T_w)\}}{T_w)}\right)^{1/4} \tag{2}$$

and the average heat transfer coefficient $\bar{h}$ for a vertical surface of height L is given by

$$\bar{h}=\frac{4}{3}\left[\frac{g\rho_f(\rho_f-\rho_g)k_f^3\{h_{fg}+}{4\mu_f L(T_g-}\right.$$
$$\left.\frac{068C_{pf}(T_g-T_w)\}}{T_w)}\right]^{1/4} \tag{3}$$

where liquid properties are evaluated at the arithmetic average of the vapor and wall temperatures.

2.2. Interrelation between the Reynolds Number and the Heat Transfer Coefficient

To establish a criterion for transition from laminar to turbulent flow, a Reynolds number for condensate flow is defined as

$$Re=\frac{\rho_f U_m D_e}{\mu_f} \tag{4}$$

The hydraulic diameter is $D_e\equiv 4A_f/p$ then, the Reynolds number becomes

$$Re=\frac{4\rho_f U_m A_f}{\mu_f p}=\frac{4\dot{m}}{\mu_f p} \tag{5}$$

where the wetted perimeter is equal to the width of the surface perpendicular to flow direction. A mass balance for the condensation of saturated vapor between $x=0$ and $x=L$, gives another correlation:

$$\bar{h}A(T_g-T_w)=\dot{m}h_{fg} \tag{6}$$

where $\dot{m}$ is the total mass flow rate of the condensate at $x=L$.

From equations (5) and (6), a form of the Reynolds number at $x=L$ is given by

$$Re=\frac{4L\bar{h}(T_g-T_w)}{\mu_f h_{fg}} \tag{7}$$

Therefore, Eq. (3) can be rewritten in terms of this Reynolds number as

$$\bar{h}=\frac{4}{3}\left[\frac{g\rho_f(\rho_f-\rho_g)k_f^3}{\mu^2_f Re}\bar{h}+\right.$$
$$\left.\frac{0.68gC_{pf}\rho_f(\rho_f-\rho_g)k_f^3}{4\mu_f L}\right]^{1/4} \tag{8}$$

With film condensation of vapor on tall vertical surface, condensation rates may easily be sufficiently large to cause turbulent flow in the film. The critical value at which transition from laminar to turbulent flow occurs is found to be about 1800 for vertical surface when the Reynolds number is defined as Eq. (5). After the start of turbulence the average heat transfer coefficient $\bar{h}$ is obtained by the following empirical relation proposed by Kirkbride[11].

$$\bar{h}=0.0077\left[\frac{g\rho_f(\rho_f-\rho_g)k_f^3}{\mu_f^2}\right]^{1/3}Re^{0.4} \tag{9}$$

2.3. The Effects of Noncondensable Gas

A class of condensation problems of much greater complexity is encountered when consideration is given to vapors which contain noncondensable gas. In such situations, concentration and temperature gradients are set up in the vapor-gas mixture. The presence of a noncondensable gas, such as air, considerably reduces the heat transfer coefficient relative to the value obtained with pure condensable steam. This important effects has heen extensively studied analytically by Sparrow and Lin, Mincowycz and Sparrow, and many others[12,13,14].

Condensation on a cold wall in the presence of a noncondensable gas is composed of three heat transfer resistances. (Fig. 1): (a) condu-

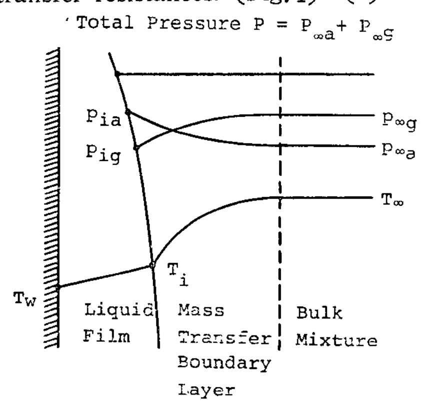

Fig. 1 Physical picture of condensing surface in the presence of noncondensable air

ction into the cold wall, (b) conduction/convection through the liquid film on the wall, and (c) mass transfer of steam and convective heat transfer through the gas/vapor boundary layer.

Direct enginering approach for the calculation of heat transfer coefficient in the presence of noncondensable gas (air) which are widely used in containment analysis following a LOCA and MSLB in nuclear power plants are the Uchida's experimental values and the Tagami's correlation[6,7].

Uchida's condensing heat transfer coefficient is based on a series of experiments where various mixtures of steam and noncondensable gases are cooled on a vertical surface 14cm× 30cm. The Uchida experiments have values of the heat transfer coefficients as a function of the air/steam weight ratio (Fig. 2). Uchida condensing heat transfer coefficient was found to be decreasing with the increase of noncondensables in the mixtures, and the coefficient is independent of the kind of noncondensables.

The Tagami correlation is based on experiments performed in a small steel cylinder (15cm diameter 45cm height), which was agitated by the introduction of large quantities of steam. This correlation can be expressed as follows:

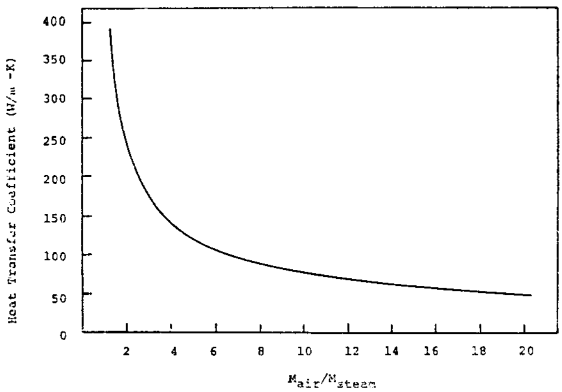

Fig. 2 Uchida heat transfer coefficients

for the maximum heat transfer coefficient

$$h_{max}=C\left(\frac{Q}{Vt_p}\right)^{0.62} \tag{10}$$

where $C=0.603$ for SI units

$C=72.5$ for English units

t_p=the time interval until peak pressure (sec)

and for the heat transfer coefficient as a function of time

$$h=h_{max}\left(\frac{t}{t_p}\right) \tag{11}$$

Tagami correlation represents a linear time interpolation between an initial value of zero and h_{max} which occurs at $t=t_p$. For times beyond t_p the Tagami correlation is not used.

Both experimental and analytical investigations for condensation in the presence of large concentration of noncondensables are lacking. For laminar film condensation, in this model, the heat transfer coefficient is corrected for the amount of air using the Henderson-Marchello correlation[15],

$$\frac{h_{cond}}{h_{Nu}}=\frac{1}{1+0.51y} \tag{12}$$

For turbulent film condensation, the results obtained by Minkowycz and Sparrow[13,14] is extrapolated because the mass fraction of air in their analysis was restricted to a value of 0.1. In turbulent conditions, the reduction of heat transfer coefficient is less sensitive to the presence of noncondensable air than the one in laminar, however, slightly increases as the temperature difference ($\Delta T=T_\infty-T_W$) is increasing and as bulk temperature is decreasing. A simplified correlation including the influence of noncondensable air, temperature difference and bulk temperature, in turbulent film condensation, is obtained as follows (based on the analytical results of Ref. 14)

$$\frac{h_{cond}}{h_{Nu}}=0.813\times 10^{-(0.1\Delta T/T_\infty+W_\infty)} \tag{13}$$

3. Computation Procedure

The calculation procedure for condensing heat transfer coefficient is as follows:

(a) An initial value of the heat transfer coefficient, h_{cond} is assumed.

(b) The condensation rate, $\dot{m}$, is determind by

$$\dot{m}=\frac{h_{cond}A(T_{\infty}-T_W)}{i_g(T_{\infty})-i_f(T_W)} \tag{14}$$

(c) Reynolds number is calculated from equation (5). If Re is greater than 1800, go to step (h), otherwise go to step (d).

(d) Nusselt heat transfer coefficient, h_{Nu}, is calculated from equation (8) for laminar film condensation.

(e) Mass fraction of air, is determined by performing a mass balance of steam, condensate and air.

(f) The mole percent of air is calculated using the following equation

$$y=\frac{1800W_{\infty}}{29-11W_{\infty}} \tag{15}$$

(g) Correction for the influence of noncondensable air is carried out by equation (12) for laminar film condensation.

(h) Nusselt heat transfer coefficient, h_{Nu}, is calculated from equation (9) for turbulent film condeosation.

(i) Correction for the influence of noncondensable air is carried out by equation (13) for turbulent film condensation.

(j) Comparison of the new value of h_{cond} obtained from step (g) or (i) with the initial value is accomplished. The iteration is continued until both values converge.

(k) This process is repeated for each time step.

4. Results and Discussion

Only limited experimental data exist for the analysis of the containment response to postulated accident conditions. Numerical results are obtained using the experimental data of the CVTR containment test 3[9]. The CV-TR simulated DBA tests provide the only large-scale containment (Fig. 3) response data available for evaluating computational techniques used in safety analysis of power reactors.

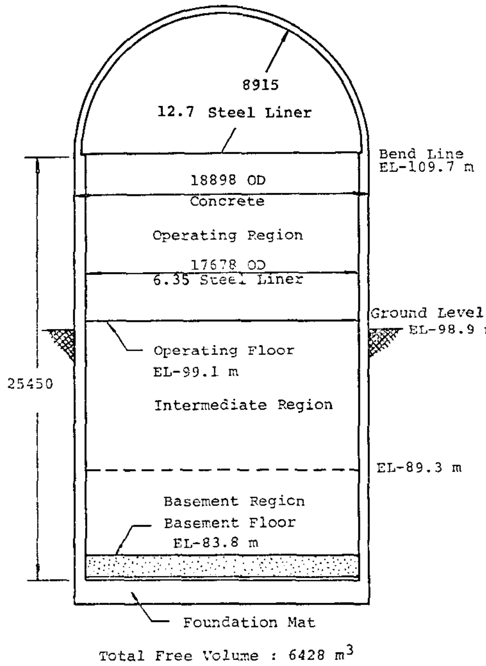

Fig. 3 CVTR containment structure

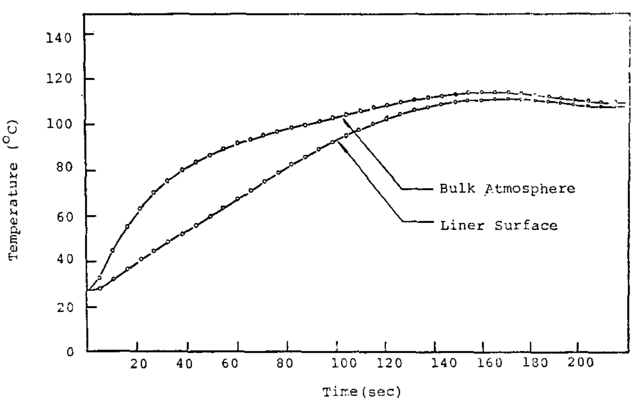

Fig. 4 Temperature transient data of heat plug 2

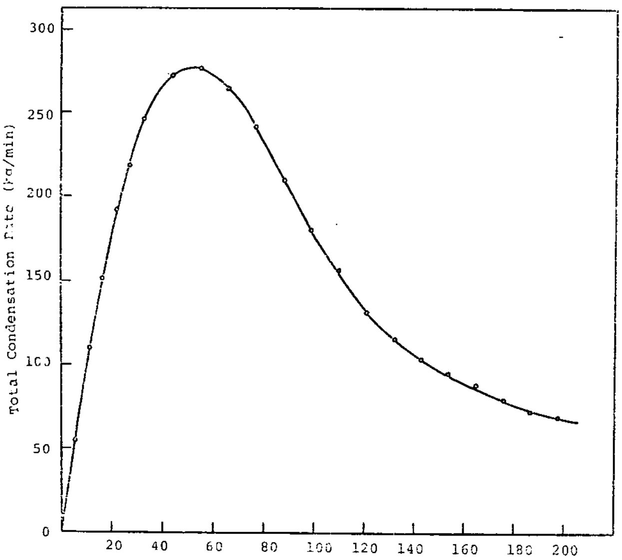

Fig. 5 Calculated condensation rate with time

In CVTR test 3, saturated steam with average enthalpy of 2780kJ/kg was injected to the containment atmosphere during 166.4 sec. The input values for the temperature of the bulk atmosphere and the wall were taken from the data of heat plug 2 of the CVTR report (Fig. 4). The average condensation rate is plotted in Fig. 5. Average condensation rate is proportional to the temperature difference because it is determined from equation (6). Condensation rate increases rapidly in early portion of the steam injection, then, decreases slowly. In the early portion of the blowdown period, as mentioned at introduction, the forced convection and the condensation of steam on the cold walls are the major mechanisms for heat transfer. When forced convection exists on the heat sink surface, convection effects become dominant. The fraction of condensation mode increases rapidly with time on the other hand convection effect decreases.

The calculated values for the condensing heat transfer coefficient are plotted in Fig. 6 and compared with Tagami and Uchida correlations. Both Tagami and Uchida correlations are not analytical results but direct engineering approach, consequently, accurate comparison of them with heat transfer coefficient calculated in this model is impossible. Nevertheless good agreement with these empirical coreelations is obtained. The weight of air initially in the containment was approximately 7439kg and leakage of air during the test period was assumed to be negligible. It is assumed that initial mass of vapor is 81.2 kg and the containment atmosphere sustains at saturated state according to given temperature input. Laminar film condensation is always reported because the calculated heat transfer coefficients are average values unifo-

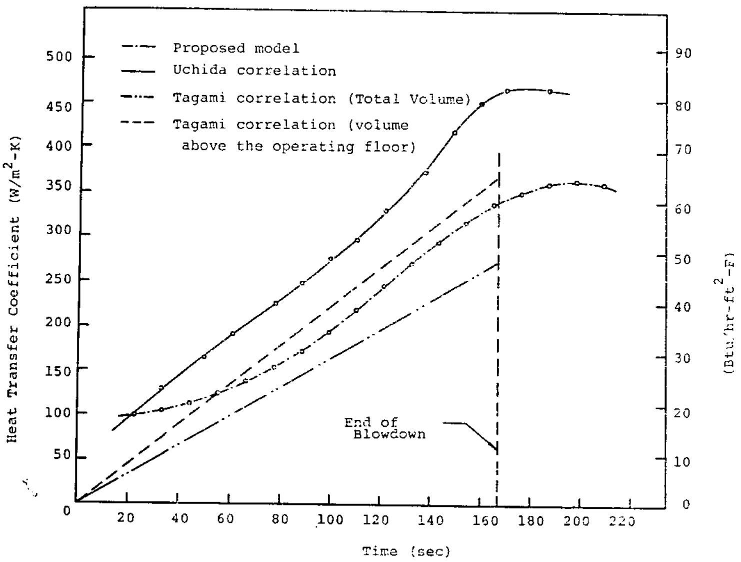

Fig. 6 Condensing Heat Transfer Coefficients for the CVTR test

rmly assigned for all heat sinks inside the containment. But locally turbulent film condensation may occur in real situation. The calculated heat transfer coefficients are slightly lower than the Tagami correlation based on the volume above the operating floor(3993m^3) but higher than the one based on total free volume (6428m^3). In licensing calculation for the evaluation of containment integrity of nuclear power plant, total free volume is used for conservatism. Uchida heat transfer coefficients determined from the air/mass ratio calculated in this model are slightly higher than the Tagami correlation. The reason for this result is believed as those assumptions that all the blowdown steam is introduced into the containment vapor region and the removal of steam is by condensation only. Therefore, the bulk air to steam ratio is more or less underestimated. Uchida correlation considers the bulk air to steam ratio only and the Tagami correlation deals with this ratio indirectly.

The range of heat transfer coefficients experimentally measured in CVTR tests was 28-1590W/m^2-K according to various position and measuring method. The lower values of calculated heat transfer coefficients than the experimental ones in CVTR tests are mainly due to average value uniformly assigned for all heat sinks and consideration of latent heat only on the assumption of saturated state.

5. Conclusions

The heat transfer process in containment building is dependent on many variables such as heat transfer mode(single phase convection, condensation or both), condensation mode (filmwise or dropwise), surface condition of heat sinks, time after accident, geometry and location of heat sinks, steam concentration and distribution, steam velocity, and temperatures of heat sinks and atmosphere. Generally a detailed space-time solution for localized heat transfer coefficient is unobtainable. A practice in containment response calculation for the safety analysis of power reactors is to use area-averaged heat transfer coefficient. The average coefficient permits estimation of total structural heat absorption and, consequently, the average containment pressure-temperature transient. However, the localized response may significantly differ from the average behavior.

Although the dominant condensation mode on the wall of containment building is filmwise condensation, lecal dropwise condensation could occur especially in early portion of blowdown phase. The extraordinary heat flux of dropwise condensation can cause serious thermal loadings on the affected structres. Therefore, the margin of safety is always required.

Calculated condensing heat transfer coeficient are lower than the experimentally measured ones in CVTR tests, but agree well with the Tagami and the Uchida empirical correlations which are widely used in containment analysis because of its apparent trend to conservatism. Therefere, this method can be used in conservative analysis for containment integrity. Further investigations are necessary concerning the effect of the size and geometrical configurations of the containment.

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