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New analogy relation of heat and mass transfer in wide range of non-condensing gas concentrations

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For condensation from a steam-gas mixture, analogy correlations of heat and mass transfer taking account of the mass absorption effect on the wall were proposed in the previous study. These correlations gave good predictions on heat exchanger calculations when the concentration of non-condensing gas was more than 75%. The analogy method is preferable for the design of heat exchangers where the heat transfer correlations have been empirically established. However, at the concentration less than 75%, some modification was necessary on the mass transfer correlation because it was originated and estimated from the heat transfer correlation without condensation. Condensation heat transfer on a row of horizontal stainless steel tubes was investigated experimentally in the non-condensing gas concentration of 0~78% and new analogy relation covering the wide range was proposed.

Keywords: Analogy correlations, Heat and mass transfer, Mass absorption effect, Non-condensing gas, Horizontal tubes

1. INTRODUCTION

The condensation with a certain amount of non-condensing gas is frequently observed in various industrial applications. The condensation heat transfer at a small amount of non-condensing gas has been studied for the air leakage problem in the condenser of steam turbine system. For a large amount of non-condensing gas, experimental and theoretical studies have been conducted for the latent heat recovery from the exhausted flue gas and the cooling of nuclear containment vessel. As the previous studies of condensation heat transfer focused on a limited range of non-condensing gas concentration, the proper estimation method covering the wide range of non-condensing gas concentration is needed for the various applications.

When the flow induced with the condensation is negligibly small in the two-dimensional flow field, the heat transportation equation is

$$\frac{\partial \tilde{T}}{\partial \tilde{t}}+\tilde{u}\frac{\partial \tilde{T}}{\partial \tilde{x}}+\tilde{v}\frac{\partial \tilde{T}}{\partial \tilde{y}}=\frac{1}{\mathrm{Re}_f\,\mathrm{Pr}_f}\left(\frac{\partial^2 \tilde{T}}{\partial \tilde{x}^2}+\frac{\partial^2 \tilde{T}}{\partial \tilde{y}^2}\right) \quad (1)$$

where non-dimensional temperature, time, velocities and coordinates are defined as,

$$\tilde{T}=\frac{T-T_i}{T_f-T_i},\ \tilde{t}=\frac{tu_m}{D},\ \tilde{u}=\frac{u}{u_m},\ \tilde{v}=\frac{v}{u_m},\ \tilde{x}=\frac{x}{D}\ \text{and}\ \tilde{y}=\frac{y}{D}$$

The mass transportation equation can also be described as,

$$\frac{\partial \tilde{w}}{\partial \tilde{t}}+\tilde{u}\frac{\partial \tilde{w}}{\partial \tilde{x}}+\tilde{v}\frac{\partial \tilde{w}}{\partial \tilde{y}}=\frac{1}{\mathrm{Re}_f\,Sc_f}\left(\frac{\partial^2 \tilde{w}}{\partial x^2}+\frac{\partial^2 \tilde{w}}{\partial y^2}\right) \quad (2)$$

where non-dimensional mass concentration is defined as,

$$\tilde{w}=\frac{w-w_i}{w_f-w_i}$$

Equations (1) and (2) indicate that the distribution of mass concentration can be estimated with that of temperature by replacing Pr with Sc. So when the heat transfer can be described by,

$$Nu_f=\left[\frac{\partial \tilde{T}}{\partial \tilde{y}}\right]_i=f(\mathrm{Re}_f,\mathrm{Pr}_f), \quad (3)$$

The mass transfer can be described by replacing Pr with Sc as,

$$Sh_f=\left[\frac{\partial \tilde{w}}{\partial \tilde{y}}\right]_i=f(\mathrm{Re}_f,Sc_f), \quad (4)$$

It should be noted that Nu and Sh are the interfacial gradients of non-dimensional temperature and concentration distributions, respectively, when the flow induced with the condensation is negligibly small. This useful and simple relation is called as the simple analogy relation and the condensation heat transfer at large amount of non-condensing gas can be well described with the relation. The analogy method is preferable for the design of heat exchangers where the heat transfer correlations have been empirically established.

For small amount of non-condensing gas, the vigorous condensation can be expected and the effect of flow towards the condensing surface (mass absorption effect) has to be considered.

The non-condensing gas flows towards the surface with the condensing gas at the normal velocity v_i and is removed with the diffusion as shown in Fig.1. The continuous equation of the non-condensing gas gives

$$v_i=-\frac{a_f}{1-w_i}\left[\frac{\partial w}{\partial y}\right]_i \quad (5)$$

As a first approximation, the mass absorption effect is considered only in the mass transfer equation, not in the heat transfer equation. The integral equations for the mass and energy conservation are

$$\frac{d}{dx}\int \rho_f u(w_f-w)dy-\rho_f v_i(w_f-w_i)=\rho_f \alpha_f\left[\frac{\partial w}{\partial y}\right]_i \quad (6)$$

$$\frac{d}{dx}\int \rho_f u(T_f-T)dy=\rho_f \kappa_f\left[\frac{\partial T}{\partial y}\right]_i \quad (7)$$

By using the non-dimensional temperature and mass concentration, Eqs (6) and (7) become

$$\frac{d}{dx}\int \rho_f u(1-\tilde{w})dy=\frac{1-w_f}{1-w_i}\rho_f \alpha_f\left[\frac{\partial \tilde{w}}{\partial y}\right]_i \quad (8)$$

$$\frac{d}{dx}\int \rho_f u(1-\tilde{T})dy=\rho_f \kappa_f\left[\frac{\partial \tilde{T}}{\partial y}\right]_i \quad (9)$$

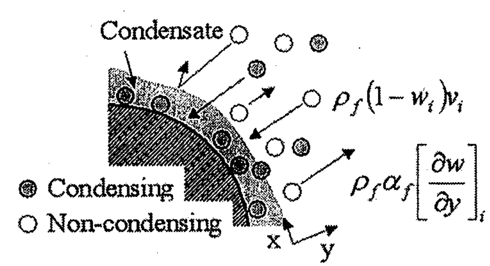

FIGURE 1 Behavior of non-condensing gas

when

$$\kappa_f = \frac{1-w_f}{1-w_i}\alpha_f \tag{10}$$

is presumed, the distribution of non-dimensional mass concentration coincides with that of non-dimensional temperature. Equation (10) can be modified as

$$\Pr_f = \frac{1-w_i}{1-w_f}Sc_f \tag{11}$$

When Nu is described with function f

$$\left[\frac{\partial \tilde{T}}{\partial y}\right]_i = \frac{1}{D}Nu_f = \frac{1}{D}f(\mathrm{Re}_f, \Pr_f) \tag{12}$$

Equations (11) and (12) give the gradient of non-dimensional mass concentration as

$$\left[\frac{\partial \tilde{w}}{\partial y}\right]_i = \frac{1}{D}f(\mathrm{Re}_f, \frac{1-w_i}{1-w_f}Sc_f) \tag{13}$$

The condensation rate m_c can be described with considering the normal velocity to the surface and the diffusion as

$$m_c = -\rho_f v_i w_i + \rho_f \alpha_f \left[\frac{\partial w}{\partial y}\right]_i \tag{14}$$

The normal velocity v_i given by Eq(5) and Eq(14) give

$$m_c = -\rho_f v_i w_i - \rho_f v_i (1-w_i) = -\rho_f v_i \tag{15}$$

The condensation rate can be described with the simple multiplication of mixture density and the normal velocity. When the normal velocity vi given by Eq(5) is again used in Eq(15),

$$m_c = -\rho v_i = \frac{\rho\alpha_f}{1-w_i}\left[\frac{\partial w}{\partial y}\right]_i = \frac{w_f - w_i}{1-w_i}\rho\alpha_f\left[\frac{\partial \tilde{w}}{\partial y}\right]_i \tag{16}$$

Equations (13) and (16) give Sh for small amount of non-condensing gas as,

$$Sh_f = \frac{m_c D}{\rho_f \alpha_f (w_f - w_i)} = \frac{1}{1-w_i}f(\mathrm{Re}_f, \frac{1}{\omega}Sc_f) \tag{17}$$

where $$\omega = \frac{1-w_f}{1-w_i}$$

So the mass transfer equation can easily be derived if the heat transfer function of Nu is known. These correlations gave good predictions when the non-condensing gas concentration was more than 75%[1-5] in single and multiple stages of heat transfer tubes using actual flue gas. But in the analogy relation, the mass transfer correlation was originated and estimated with the heat transfer correlation without the condensation; i.e. no mass absorption effect. So the estimation error can be expected as the heat transfer equation is affected with the mass absorption effect when the significant condensation occurs at the lower concentration of non-condensing gas.

To make clear the evaluation error at the lower concentration of non-condensing gas, steam condensation heat transfer on horizontal stainless steel tubes was experimentally investigated using a row of three tubes in a transparent duct at non-condensing gas concentration of 0~78 %. The tube outer diameter was 21.7mm and air was used as the non-condensing gas. The temperatures of air/steam mixture gas, cooling water in tubes and tube surfaces as well as the steam mass concentration and pressure were measured for the modification of analogy relation.

2. EXPERIMENTAL APPARATUS AND METHOD

Shown in Fig.2 is a schematic of experimental apparatus. The steam of approximately 120°C mixed with air as non-condensing gas was supplied to the test section of atmospheric pressure. Steam and air was well mixed with the special nozzles where air and steam jets had different rotational directions. The steam was supplied from a couple of steam boilers and the mass flow rate was measured with a vortex-shedding flow meter or V-cone mass flow meter. The error was within ±2% of the measured mass flow rate. The airflow rate was measured with several rotor flow meters. The mixture gas temperatures just above and below the test tubes were measured with sheathed T-type thermocouples of 0.5 mm in diameter.

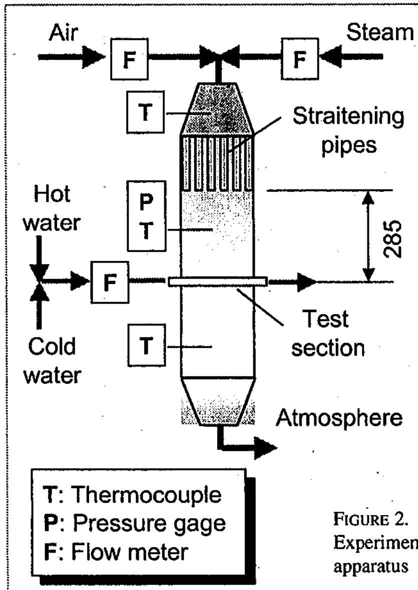

FIGURE 2. Experimental apparatus

Water cooled test tubes of SUS316L were installed in a transparent polycarbonate duct with a cross-section of 160mm×101mm as shown in

Fig.3. The water flow rate was measured with a rotor flow meter. The connecting pipes outside the test section were thermally insulated with a glass wool. Three tubes of 21.7mm in diameter were arranged horizontally at a pitch of 33.7mm. The pitch to diameter ratio p/d was 1.55 that was typical for the conventional heat exchanger. Sheathed T-type thermocouples of 0.5 mm in diameter were imbedded at circumferential locations of 0, 45, 90, 135 and 180° from the top of 3 tubes to obtain the average wall temperature. The inlet and outlet water temperatures of the test tubes were measured also with the sheathed K-type thermocouples of 0.5 mm in diameter. The outlet temperature was measured at a mixing chamber to obtain a well-mixed bulk temperature.

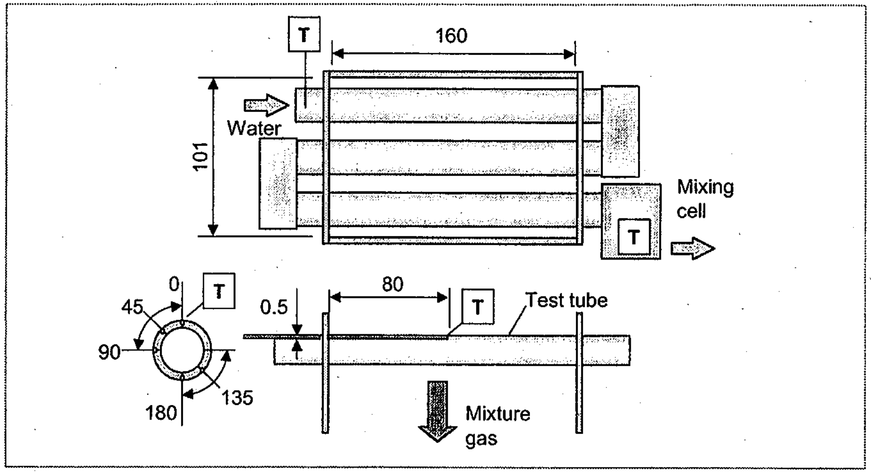

FIGURE 3. Test section

The temperature difference of the cooling water through the tubes was kept approximately at 1.7~15.7K. The heat flux of the test tubes was calculated with the flow rate and the temperature difference of cooling water through the tubes. The measurement error of the heat flux was estimated to be ±12% as the measurement error of the temperature by thermocouples was within ±0.1 K. The thermocouple signals were transferred to a personal computer by a data logger and analyzed.

TABLE 1. Experimental condition

Re_f	1350 ~ 11300
Air mass ratio 1-W_f	0 ~ 0.78
Mixture gas temp. T_f (°C)	73.0 ~ 129.1
Average wall temp. T_W (°C)	28.8 ~ 91.0
Inlet cooling water temp. (°C)	8.0 ~ 74.0

The experiments were conducted at atmospheric pressure. The mixture Re number, the non-condensing gas concentration and average surface temperature of tubes were varied for the parametric study. The major test conditions are shown in Table 1.

3. CONSTITUTIVE EQUATIONS FOR PREDICTION

3.1 Heat resistance of condensate

The momentum balance dominated by viscous and gravity force gives the velocity distribution at θ° from the tube top in Fig.4. :

$$u=\frac{(\rho_L-\rho_G)g\sin\theta}{\mu_L}\left(y\delta-\frac{y^2}{2}\right) \tag{18}$$

Integrating the above velocity profile and using the condensate mass flow rate per unit of tube length, *m*, yields the film thickness as,

$$\delta=\left[\frac{1.5\mu_L m}{\rho_L(\rho_L-\rho_G)g\sin\theta}\right]^{1/3} \tag{19}$$

The heat conductivity of film is

$$K=\frac{\lambda_L}{\delta}=\left[\frac{\lambda_L^3\rho_L(\rho_L-\rho_G)g\sin\theta}{1.5\mu_L m}\right]^{1/3} \tag{20}$$

Equation (20) gives the heat flux through the film when the temperature difference between the film is given. The average conductivity from θ = 0° to $\theta=\pi$ is

$$\bar{K}=\frac{1}{\pi}\int_0^\pi K d\theta=0.72\left[\frac{\lambda_L^3\rho_L(\rho_L-\rho_G)g}{\mu_L m}\right]^{1/3} \tag{21}$$

The average interfacial temperature T_i can be described with

$$T_i = \frac{q_w}{K} + T_w \tag{22}$$

the average wall temperature T_w and heat flux q_w as,

The heat flux qw can be calculated with Eq.(22) and the condensation heat flux equation mentioned below in the mixture gas side which strongly depends on the interfacial temperature T_i. In the present study of single stage, the average flow rate of condensate on unit of tube length is used as the condensate mass flow rate, m. These equations which give the average conductivity based on the condensate mass flow rate are very important to calculate the inundation effects on the tubes in the multi-stages tube bank.

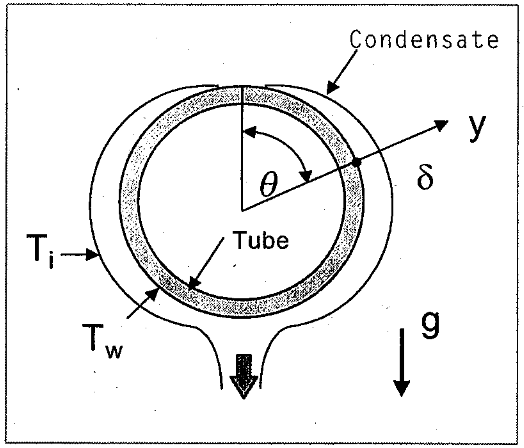

FIGURE 4. Condensate on tube

3.2 Heat and mass transfer in gas side

The total heat flux q_W consists of the convective heat flux q_v and the condensation heat flux q_c as;

$$q_w = q_V + q_C \tag{23}$$

The convection heat flux q_v is expressed as,

$$q_V = h(T_f - T_i) \tag{24}$$

The condensation heat flux q_c can be expressed as,

$$q_C = h_C L_W \rho_f (w_f - w_i) \tag{25}$$

where the steam mass concentration w_i [kg/kg] is the saturated concentration corresponding to the interfacial temperature T_i.

Based on the previous studies, the Nusselt number Nu_f for the average convective heat transfer coefficient is

$$Nu_f = c\,\mathrm{Re}_f^a\,\mathrm{Pr}_f^b(\mathrm{Pr}_f/\mathrm{Pr}_w)^{0.25} \tag{26}$$

where subscript f and w indicate the physical values defined at mixture gas and wall, respectively. Zukauskas[6] proposed a = 0.6, b = 0.37, c = 0.26 for the first row tubes of an in-line bank with p/d=1.6, which is approximately the same as that in the present study, in the range of $10^3 < \mathrm{Re}_f < 2\times10^5$.

The analogy relation of Eqs.(12) and (17) give the Sherwood number describing the average mass transfer coefficient as,

$$Sh_f = \frac{1}{1-w_i}\left(\frac{1-w_i}{1-w_f}\right)^b c\,\mathrm{Re}_f^a\,Sc_f^b(Sc_f/Sc_w)^{0.25} \tag{27}$$

The mixture gas was treated as the mixture of N_2, O_2 and H_2O and its property was estimated with special combinations of each gas property proposed by the previous studies. For example, the heat conductivity and the viscosity were estimated with the methods by Lindsay&Bromley equation[7] and Wilke equation[8], respectively. The steam mass diffusivity of mixture gas was obtained with the modification[1-3] of Fujii's correlation[9] for steam mass diffusion in air.

4. EXPERIMENTAL RESULTS AND CONSIDERATION

4.1 Pure steam experiment

The heat resistance of condensate dominates the heat transfer in the pure steam condition as the convective and diffusive resistance in the gas side is negligible. So the heat resistance of condensate was studied experimentally using pure steam before the mixture gas experiments. When the laminar film of condensate without the interfacial shear is assumed on the heat transfer tube, the average heat transfer coefficient can be described with the following Nusselt correlation.

$$Nu = \frac{hd}{\lambda_L} = 0.728\left(\frac{Ra}{Ja}\right)^{1/4} \tag{28}$$

where $$Ra = \frac{g(\rho_L - \rho_f)\mathrm{Pr}_L d^3}{\rho_L \nu_L^2}, \quad Ja = \frac{C_{pL}(T_{sat} - T_W)}{L_W}.$$

Though Eq.(21) was obtained with the constant flow rate of condensate, the actual flow rate of condensate increases as flowing down from the top to the bottom of tube. When 0.424 times of the total generated condensate is used as m in Eq. (21), Eq.(28) coincides with Eq.(21). The average flow rate of condensate for the evaluation of heat transfer is slightly smaller than the half of total generated condensate on the tube.

On the other hand, Fujii[10] proposed the following empirical correlation for the condensation heat transfer on a single tube placed in a wide space taking account of the effect of cross flow velocity.

$$Nu = 0.96X^{0.2}\sqrt{\mathrm{Re}_L} \tag{29}$$

where $X = \frac{\mathrm{Pr}_L}{FrJa}$, $Fr = \frac{u_\infty^2}{gd}$, $\mathrm{Re}_L = \frac{u_\infty d}{\nu_L}$,

in the applicable range of $0.03 < X < 600$. In the above correlation, $u\infty$ is the main flow velocity approaching to the single tube.

Shown in Fig.5 is the relation of steam Re and non-dimensional heat flux divided by Nusselt and Fujii's predictions. The key Δ is the non-dimensional heat flux divided by the heat flux obtained with the average conductivity of laminar film, Eq.(21), where 0.424 times of the total generated condensate is used as the average condensate mass flow rate m. The total mass of generated condensate was estimated with the measured heat flux. In the present experimental range, the experimental data agree well with the prediction assuming the laminar film of zero interfacial shear. As the Fujii's correlation was obtained with horizontal single tube experiments crossed by the horizontal flow of low-pressure steam, the lower heat transfer in the present experiment is considered to be due to the different experimental geometry.

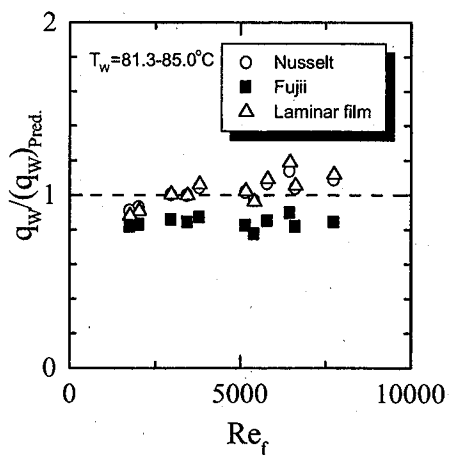

FIGURE 5. Condensation without non-condensing gas.

4.2 Mixture gas experiment

The observation showed that the heat transfer surface was covered always with the thin film of condensate at the lower mass concentration of air but the several dry spots appeared at the higher concentration of air. In the present prediction, the laminar film of condensate was assumed as a first approximation. The heat flux was calculated with Eq.(21) to obtain the interfacial temperature and the heat & mass transfer correlations in the gas flow side. As mentioned above, 0.424 times of the total generated condensate was used as the average condensate mass flow rate m. The total generated condensate was calculated with the mass transfer determined by the predicted Sh number in the mixture gas side. As the film of condensate was not continuous and several dry spots could be observed at the higher mass concentration of air, the assumption of continuous laminar film in the present prediction is not appropriate in the strict sense. However, the heat transfer in this region was dominated with the convection and diffusion of the mixture gas side and the film thickness of condensate was negligibly small. So it was considered that the effect of non-continuous film appeared as the dry spots could be neglected.

Figure 6 shows the relation of the non-dimensional heat flux and the mixture gas Re. The non-dimensional heat flux is the experimental heat flux divided by the prediction assuming the laminar film and the conventional analogy relation described with the equations (26) and (27). The systematic trends or characteristics depending on the mixture gas Re are not clear.

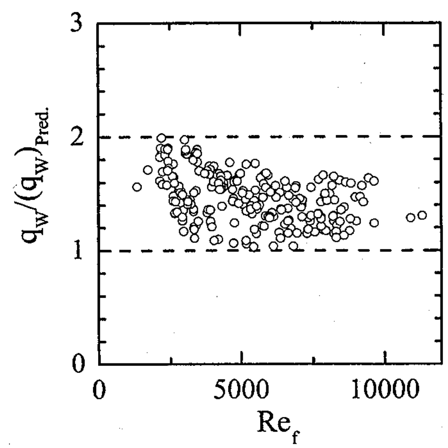

FIGURE 6. Relation of heat flux divided by conventional predictions and Re of mixture gas.

Shown in Fig.7 is the relation of non-dimensional heat flux and air mass concentration ratio between main flow and interface, $\omega = (1-W_f)/(1-W_i)$. The parameter ω is the important in Rose and Fujii's correlations and approximately proportional to $1-W_f$ in the present experiment. The experimental heat flux approximately agrees with the prediction at the lower ω because the heat transfer in this region is dominated with the heat conductivity of film, and increases with an increase of ω. The non-dimensional heat flux takes a maximum and gradually decreases to 1 approximately at = 0.8 indicating the proper prediction.

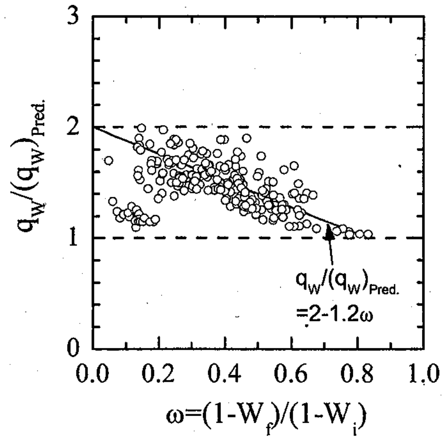

FIGURE 7. Relation of heat flux divided by conventional predictions and $\omega = (1-W_f)/(1-W_i)$

When ω is larger than 0.8, the modification of correlation is not necessary as the flow induced with the condensation becomes negligibly small. The previous experimental data at the non-condensing gas concentration larger than 75%($\omega > 0.75$) agreed well without the modification [1-5]. The non-dimensional heat flux can be described with the following correlation except the region of low non-condensing gas concentration.

$$\frac{q_w}{(q_w)_{Pred.}} = 2 - 1.2\omega \qquad (30)$$

The mass transfer correlation was estimated from the convective heat transfer correlation without condensation in the analogy relation described with Eqs.(26) and (27). So the mass transfer correlation underestimates the experimental result when the convective heat transfer is enhanced due to the significant condensation at the low mass concentration of non-condensing gas. However, the convective heat flux is negligibly small compared to the condensation heat flux when the enhancement due to the condensation appears. So the modification of convective heat transfer correlation due to the mass absorption effect is not important compared to that of the mass transfer correlation. On the other hand, when the enhancement due to the mass absorption effect can be neglected at the high concentration of non-condensing gas, the modification of heat transfer correlation is considered to be also unnecessary. As a first approximation, the heat transfer correlation without the mass absorption effect was used and only the mass transfer correlation was modified with Eq.(30) as,

$$Sh_f = \frac{Max(1\,,\,2-1.2\omega)}{1-w_i}\left(\frac{1}{\omega}\right)^b c\,\mathrm{Re}_f^a\, Sc_f^b (Sc_f / Sc_w)^{0.25} \qquad (31)$$

Considering the previous study of the non-condensing gas concentration more than 75%, the larger value of 1 or 2-1.2ω is given by the Max function in Eq.(31). The estimated uncertainty for the Sh number is considered to be within ±20%.

Shown in Figure 8 are the comparison of experimental and predicted heat flux with the previous and new analogy relation. The underestimation with the previous analogy relation is successfully improved in the new analogy relation. The slight underestimation at the higher heat flux region is considered to be due to the conservative assumption of laminar film with zero interfacial shears.

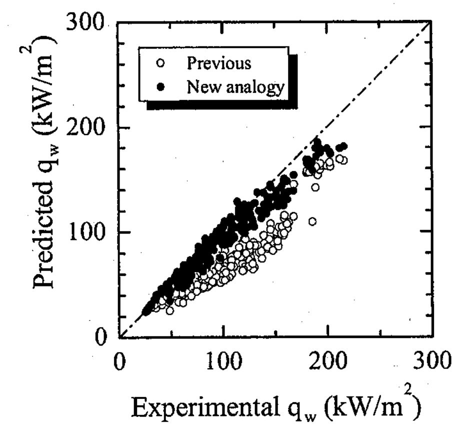

FIGURE 8. Comparison of predicted and experimental heat flux.

New analogy relation gives the condensation flow rate and the normal velocity to the surface can be calculated with Eq(16). Shown in Figure 9 is the relation of normal velocity and ω. The higher normal velocity due to the vigorous condensation is obtained at the lower ω. The normal velocity decreases with an increase of ω and approaches to zero at ω= 0.8 indicating the weak condensation.

To verify the new analogy relation in the different geometry is important, the relation was applied to a single tube placed in a wide space. Rose[11] proposed the following correlation for a single tube placed in a wide space and crossed by steam with non-condensing gas.

$$Sh_f = \frac{\left[1+2.28Sc_f^{1/3}\left(\frac{1}{\omega}-1\right)\right]^{1/2}-1}{2(1-\omega)}\sqrt{\mathrm{Re}_{f\infty}}\frac{1}{1-w_i} \quad (32)$$

where $\omega = \frac{1-w_f}{1-w_i}$.

For the convective heat transfer with the condensation is expressed with,

$$Nu_f = \frac{0.57\sqrt{\mathrm{Re}_{f\infty}}\,\mathrm{Pr}_f^{1/3}}{1+\beta\,\mathrm{Pr}_f} + \beta\,\mathrm{Pr}_f\sqrt{\mathrm{Re}_{f\infty}} \quad (33)$$

where $\beta = \frac{Sh_f(1-w_i)(1-\omega)}{\sqrt{\mathrm{Re}_{f\infty}}\,Sc_f}$.

The applicable range of the equation is $10 < Re_{f\infty} < 10^4$.

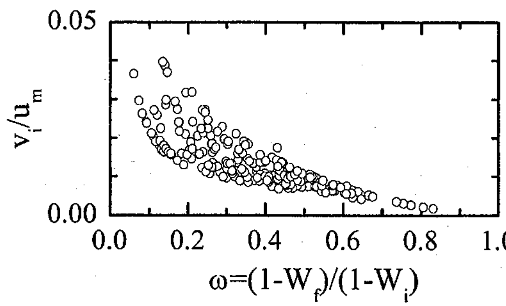

FIGURE 9. Relation of normal velocity to surface and $\omega=(1-W_f)/(1-W_i)$.

Fujii[10] proposed the following correlation instead of Eq.(32) to fit to their experimental results.

$$Sh_f = 0.73(1+0.0028\sqrt{\mathrm{Re}_{f\infty}})\frac{Sc_f^{1/3}}{\sqrt{\omega(1+\omega)}}\frac{\sqrt{\mathrm{Re}_{f\infty}}}{1-w_i} \quad (34)$$

In the correlations by Rose and Fuji, the physical values were defined at the mean temperature of mixture gas and interface; i.e. film temperature.

In Rose and Fujii's correlations, the base correlation of heat transfer to estimate the mass transfer correlation is,

$$Nu_f = 0.57\,\mathrm{Re}_{f\infty}^{1/2}\,\mathrm{Pr}_f^{1/3} \quad (35)$$

The new analogy relation in the present study simply and successfully estimates the mass transfer correlation as,

$$Sh_f = \frac{Max(1\,,\,2-1.2\omega)}{1-w_i}\left(\frac{1}{\omega}\right)^{1/3} 0.57\,\mathrm{Re}_{f\infty}^{1/2}\,Sc_f^{1/3} \quad (36)$$

Shown in Fig.10 is the comparison of this correlation with the Rose and Fujii's correlations. In the Rose and Fujii's correlations, Sc_f of 0.8 and $Re_{f\infty}$ of 5000, respectively, were used as the typical values. The mass transfer correlation given by the new analogy relation agrees well with the Rose and Fujii's correlations. The new analogy relation is useful not only for the tube bank but also for the single tube.

When the single tube correlation has been applied to the heat exchanger geometry, the special velocity taking account of the blockage effect in the restricted duct has been empirically used. These empirical methods have been applied though the flow in the heat exchanger is apparently different from that around the single tube in a wide space. The application of these methods to the different geometries of heat exchangers such as using fins or spacers to enhance the heat transfer of tube is very difficult. Also when the flow slantingly crosses the tubes or the much smaller tubes of diameter than the conventional tubes are used, the application of these methods should be careful. Even when the heat transfer characteristics are significantly different from that of the conventional single tube in wide space, the present new analogy relation can give the mass transfer correlation if the heat transfer correlation is empirically given. So the wide applications of the new analogy relation are expected on the future design of condensing heat exchangers of complex geometries.

5. CONCLUSION

Condensation heat transfer on horizontal stainless steel tubes was investigated experimentally for the modification of mass transfer correlation. The experiment was conducted in a wide range of non-condensing gas concentration of 0~78% and the new analogy relation was proposed as,

$$Nu = f(\mathrm{Re},\mathrm{Pr})$$

$$Sh = \frac{Max(1\,,\,2-1.2\omega)}{1-w_i} f(\mathrm{Re},\frac{1}{\omega}Sc)$$

where function Max yields the larger value of 1 or $2\text{-}1.2\omega$.

NOMENCLATURE

C_p:	specific heat [J/(kgK)]
d:	outer diameter of tube [m]
D:	mass diffusivity [m^2/s]
h:	heat transfer coefficient [W/(m^2K)]
h_C:	mass transfer coefficient [m/s]
L_W:	latent heat [J/kg]
m:	condensate mass flow rate per unit of tube length[kg/(s m)]
Nu_f:	Nusselt number [$= hd / \lambda_f$]
Pr:	Prandtl number [$= \nu / \kappa$]
q:	heat flux [kW/m^2]
Re_f:	Reynolds number [$= u_m d / \nu_f$]
$Re_{f\infty}$:	Reynolds number [$= u_\infty d / \nu_f$]
Sh_f:	Sherwood number [$= h_c d / D_f$]
Sc:	Schmidt number [$= \nu / D$]
T:	temperature [°C]
u_m:	velocity at minimum flow area [m/s]
u_∞:	velocity at maximum flow area [m/s]
v:	velocity[m/s]
w:	mass concentration per fluid of an unit mass [kg/kg]
κ:	thermal diffusivity [$= \lambda / (\rho C_p)$] [m_2/s]
λ:	heat conductivity [W/(mK)]
ν:	kinematic viscosity [m^2/s]
ρ:	density [kg/m^3]

Subscript

f:	mixture gas
g:	air
i:	interface(condensation surface)
L:	condensate
sat:	saturated condition of steam
w:	wall

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