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PREDICTION AND PERFORMANCE OF COMPACT LATENT HEAT RECOVERY HEAT EXCHANGER WITH SMALL DIAMETER TUBES

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ABSTRACT

The most part of energy losses in power system such as fuel cells is due to the heat released by the exhaust gas to atmosphere. The exhaust gas consists of non-condensable gas and steam with sensible and latent heat. As a lot of latent heat is included in the exhaust gas, its recovery is very important to improve the power system efficiency. Based on the previous basic studies, a thermal hydraulic prediction method for latent heat recovery exchangers was proposed. For the condensation of steam on heat transfer tubes, the modified Sherwood number taking account of the mass absorption effect on the wall was used. Two kinds of compact heat exchanger with staggered banks of bare tubes of 10.5 or 4mm in outer diameter was designed with the prediction method. The more compactness was obtained with the smaller tubes at a designed heat recovery. The thermal hydraulic behavior in the compact heat exchangers was experimentally studied with air-steam mixture gas. In the parametric experiments varying the steam mass concentration, the temperature distributions of cooling water and mixture gas were measured. The experimental results agreed well with the prediction proposed in this study and the more compactness with the smaller tubes was proved.

INTRODUCTION

Recently, for a biological and environmental safety, clean fuels such as natural gas or hydrogen are recommended to use in the power system. As the clean fuel includes a lot of hydrogen instead of carbon, the exhaust flue gas includes a lot of steam accompanying with the latent heat. As a next power system, the fuel cells using only the hydrogen is planned and developed in worldwide. The most part of energy losses in fuel cells is due to the heat released by the exhaust gas to atmosphere. The exhaust gas consists of air and steam with sensible and latent heat. As a lot of latent heat is included in the mixture gas, its recovery is very important to improve the power system efficiency using the fuel cell.

Based on the previous basic studies, a prediction method was proposed for the design of heat exchanger to recover the latent heat in the air-steam mixture gas. The modified Sherwood number taking account of the mass absorption effect on the heat transfer tubes is used for the condensation of steam in the presence of non-condensing gas. Laminar film of condensate on the tubes is assumed to evaluate the heat resistance due to the inundation. In the calculation procedure, it is possible that the gas temperature coincides with the dew point which is the saturation temperature corresponding to the partial pressure of steam in the mixture gas. When the gas temperature decreases below the dew point, the condensation of steam in the mixture gas takes place and the latent heat increases the gas temperature until it coincides with the dew point.

For condensation from a steam-gas mixture flowing normal to horizontal rows of tubes, an approximate analogy relation between heat and mass transfer was obtained with semi-theoretical consideration taking account of the mass absorption effect on the wall in the previous study.

$$Nu = f(Re,Pr) \tag{1}$$

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

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

Equations (1) and (2) are heat transfer and mass transfer correlations, respectively. The mass transfer equation can be derived if the heat transfer function of Nu is known. These correlations gave good predictions when the steam mass concentration was less than 25% [1-10] in single and multiple stages of heat transfer tubes using actual flue gas. Also at the steam mass concentration more than 25%, the good predictions were obtained in the experiment of single stage using air-steam mixture [11]. It is very important to verify the availability of the analogy relation in the heat exchanger configuration at the steam mass concentration more than 25%.

The compactness can be obtained in the heat exchanger for the latent heat recovery when bare tubes of small diameter are used instead of conventional finned tubes [10]. The compact countercurrent cross-flow heat exchanger using small bare tubes of SUS304 was designed and constructed to prove its high ability. Two kinds of compact heat exchanger with staggered banks of bare tubes of 10.5 or 4mm in outer diameter was designed with the prediction method. The experimental study varying the steam concentration of mixture gas, feed water temperature and flow rate was conducted.

NOMENCLATURE

C_p: specific heat [J/kg]
d: outer diameter of tube [m]
d_i: inner diameter of tube [m]
D: mass diffusivity [m^2/s]
h_V: heat transfer coefficient [W/(m^2K)]
h_C: mass transfer coefficient [m/s]
L_W: latent heat [J/kg]
Nu: Nusselt number [$= h_V d / \lambda$]
P: pressure [Pa]
q: heat flux [kW/m^2]
Pr: Prandtl number [$= \nu / \kappa$]
Re: Reynolds number [$= ud / \nu$]
S_1: spanwise pitch [m]
S_2: flow-directional pitch [m]
Sh: Sherwood number [$= h_C d / D$]
Sc: Schmidt number [$= \nu / D$]
T: temperature [°C]
u: velocity at minimum flow area [m/s]
V: volumetric flow rate [m_N^3/s]
w: mass concentration per fluid of an unit mass [kg/kg]
κ: thermal diffusivity [$= \lambda / (\rho C_P)$]
λ: heat conductivity [W/(mK)]
ν: kinematic viscosity [m^2/s]
ρ: density [kg/m^3]

subscript

C: condensation
f: mixture gas
i: interface(condensation surface)
V: convection
W: total or wall of tube
N: standard condition at 0°C and atmospheric pressure
sat: saturated condition of steam

CONSTITUTIVE EQUATIONS FOR PREDICTION

Heat resistance of condensate

Though a part of condensate falls down between the tubes and on the duct wall, it is assumed that all the condensate generated at the upper stage flows on the tubes as a laminar film. The momentum balance dominated by viscous and gravity force gives the velocity distribution at θ° from the tube top in Fig.1. :

$$u = \frac{(\rho_L - \rho_G) g \sin\theta}{\mu_L}\left(y\delta - \frac{y^2}{2}\right) \tag{3}$$

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

$$\delta = \left[\frac{1.5\mu_L m}{\rho_L(\rho_L - \rho_G) g \sin\theta}\right]^{1/3} \tag{4}$$

The heat conductivity of film is

$$K = \frac{\lambda_L}{\delta} = \left[\frac{\lambda_L{}^3 \rho_L(\rho_L - \rho_G) g \sin\theta}{1.5\mu_L m}\right]^{1/3} \tag{5}$$

Equation (5) gives the heat flux through the film when the temperature difference between the film is multiplied. The average conductivity from θ= 0° to θ=π is

$$\overline{K} = \frac{1}{\pi}\int_0^{\pi} K d\theta = 0.72\left[\frac{\lambda_L{}^3 \rho_L(\rho_L - \rho_G) g}{\mu_L m}\right]^{1/3} \tag{6}$$

The average heat resistance of film is defined as the inverse of the above average conductivity. The average film thickness is

$$\overline{\delta} = \frac{\lambda_L}{\overline{K}} \tag{7}$$

In the calculation, the mass flow rate, m, at a certain stage includes the condensate generated at the stage for the conservative estimation.

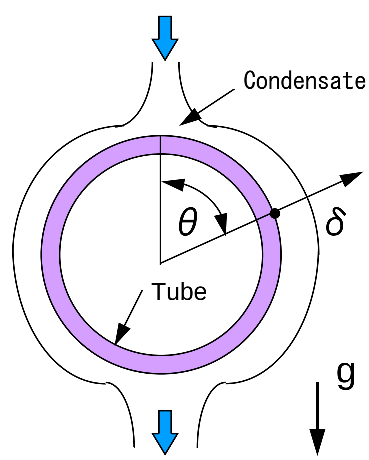

Fig. 1 Heat resistance of condensate film

Heat and mass transfer in gas side

The total heat flux q_W consists of the convection heat flux q_V and the condensation heat flux q_C as

$$q_w = q_V + q_C \tag{8}$$

The convection heat flux is expressed as

$$q_V = h(T_f - T_i) \tag{9}$$

The condensation heat flux q_C can be expressed as,

$$q_C = h_C L_W \rho_f (w_f - w_i) \tag{10}$$

where W_i is the mass concentration of saturated steam at the wall temperature T_i. Based on the previous studies[12], the Nusselt number Nu_f for the average convective heat transfer coefficient in the range of $10^3 < Re_f \leqq 2\times 10^5$ is

$$Nu_f = c\, Re_f^a\, Pr_f^b (Pr_f / Pr_w)^{0.25} \tag{11}$$

Zukauskas[12] proposed a=0.6, b=0.36 and

For $S_1/S_2 < 2$ $\quad c = 0.35(S_1/S_2)^{0.2}$ (12)

For $S_1/S_2 \geqq 2$ $\quad c = 0.40$ (13)

for a staggered bank. For the condensation of steam on heat transfer tubes, the modified analogy relation of Eqs.(1) and (2) gives

$$Sh_f = M_f c\, Re_f^a\, Sc_f^b (Sc_f / Sc_w)^{0.25} \tag{14}$$

where $M_f = \dfrac{Max(1,\ 2-1.2\omega)}{1-w_i}\left(\dfrac{1}{\omega}\right)^b$

The Sh number increases sharply at the steam mass concentration of 1 in Eq.(14). This indicates the mass transfer at the pure steam condition is enough high to neglect the interfacial resistance of mass transfer. In the calculation for pure steam without air, the modification factor M_f of 100 was used to avoid the calculation error divided by zero.

Mixture gas was treated as a 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 [14] and Wilke[15], respectively. It is considered that a strong correlation exists between the thermal and mass diffusivities. As a first attempt, the mass diffusivity of steam in mixture gas was estimated with the well-known mass diffusivity of steam in air as

$$D = D_{air}\left(\frac{\kappa}{\kappa_{air}}\right) \tag{15}$$

where κ and κ_{air} are the thermal diffusivities of flue gas and dry air, respectively. The diffusivity of steam in air can be expressed as[16],

$$D_{air} = 7.65\times 10^{-5}\,\frac{(T+273.15)^{11/6}}{P} \tag{16}$$

The one-dimensional heat and mass balance calculation along the flow direction of flue gas was conducted. The steam mass concentration and the flue gas temperature at N+1th stage can be calculated from those at Nth stage as shown in Fig.2.

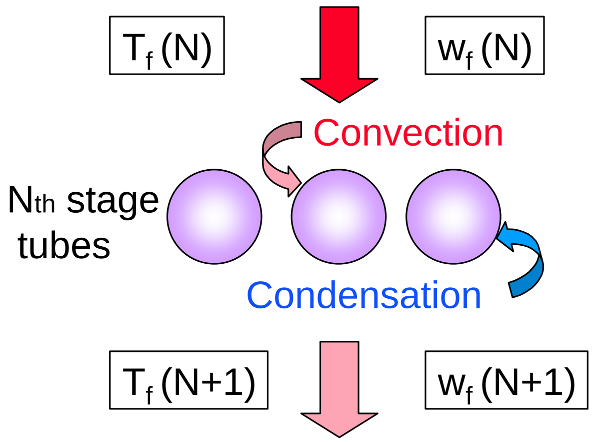

Fig. 2 One-dimensional calculation

The heat and mass balance equations are**;**

$$w_f(N+1) = \frac{m_f w_f(N) - q_C A_W / L_W}{m_f - q_C A_W / L_W} \tag{17}$$

$$T_f(N+1) = T_f(N) - \frac{q_V A_W}{C_{Pf}\, m_f} \tag{18}$$

where A_w is the heat transfer area per a stage.

It is possible that the gas temperature merges with the dew point which is the saturation temperature corresponding to the partial pressure of steam in the flue gas. When the gas temperature decreases below the dew point, the condensation of steam in the flue gas takes place and the latent heat increases the gas temperature until the gas temperature coincides with the dew point. In this case, the energy balance gives the relation between the increase of the gas temperature, ΔT_f, and the decrease of steam concentration, Δw_f, as;

$$\Delta T_f = \frac{L_W}{C_{Pf}} \Delta w_f \tag{19}$$

Heat conduction in tube

The heat conductivity for the inconel or austenite stainless steel is given with the following approximate correlation [17].

$$\lambda_t = 13.2 + 0.013 T_t \qquad \mathrm{W/(m\ K)} \tag{20}$$

where T_t is the average temperature of tube as,

$$T_t = \frac{T_W + T_{Wi}}{2} \tag{21}$$

where T_w and T_{wi} are the outer and inner wall temperatures, respectively. The heat flux at the outer wall is,

$$q_W = \frac{2\lambda_t (T_W - T_{Wi})}{d_O \ln(d_O / d_i)} \tag{22}$$

Heat transfer in water side

Heat transfer correlation by Dittus-Boelter taking account of the pipe inlet region is used. The coefficient by McAdams[18] was used for the modification.

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.4} \left(1 + \left(\frac{d_i}{L} \right)^{0.7} \right) \tag{23}$$

where L is the heating length of tube.

EXPERIMENTAL APPARATUS AND METHOD

Shown in Fig.3 is a schematic of experimental apparatus. Steam and air were supplied from a boiler and a compressor, respectively, and well mixed before entering the test heat exchanger. The mixture gas was led to the upper plenum and flowed downward in the heat exchanger. The mixture gas was finally released to atmosphere from the outlet plenum. The supplied steam temperature was approximately 120°C and mixed with air of room temperature.

Shown in Fig.4 is a schematic of heat exchanger. Heat transfer tubes were installed in a rectangular duct of 205X205mm. The tubes at each stage were connected with a header to maintain the same flow rate of feed water. The feed water was supplied at the downstream of gas flow and flows counter-currently to the upstream. In the present study, two kinds of heat transfer tubes with the different diameter were used. The height L of the heat exchanger strongly depends on the diameter of heat transfer tubes.

Shown in Fig.5 is the arrangement of heat transfer tubes. The staggered tube bank with the same flow-directional and span-wise pitch was adopted. Two kinds of bare tubes of 10.5 or 4mm in outer diameter were installed in the rectangular duct. The heat exchanger with 10.5 mm tubes was called as Type A and that with 4mm was called as Type B.

Shown in Table 1 are the major dimensions of the two heat exchangers designed with the same heat recovery rate. The height of Type A heat exchanger with the larger tubes was 820mm, on the other hands, that of Type B with the smaller tubes was only 160mm. The total number of heat transfer tubes was 380 in the Type A and 500 in the Type B. The total weight of heat transfer tubes was 21.6kg in the Type A and 7.65kg in the Type B. The tube weight of Type B was approximately 1/3 of Type A. The heat transfer area at the gas side of Type B was approximately the half of Type A. The compactness was achieved with the smaller tubes.

Shown in Fig.6 the comparison of calculated heat recovery rate for Type A and B heat exchangers at the mixture gas temperature of 100°C and the feed water flow rate of 600kg/h. Even when the steam mass concentration was varied between 0.12 and 0.93, the same heat recovery rate was successfully obtained with both heat exchangers.

The temperature distributions of water and flue gas in the heat exchanger were measured with sheathed thermocouples. The thermocouple signals were transferred to a personal computer with a GPIB line and analyzed. The measurement error of the temperature in this study was within ±0.1 K. The parametric study varying the mixture gas flow rate, steam mass concentration, feed water flow rate and temperature was conducted. The major test conditions are shown in Table.2. The steam mass flow rate was varied between 21.6 to 86.9 kg/h and the air mass flow rate was controlled to obtain a desired steam mass concentration between 0.2 and 1. As the result of mixing of steam and air, the mixture gas temperature was between 77 and 113 °C.

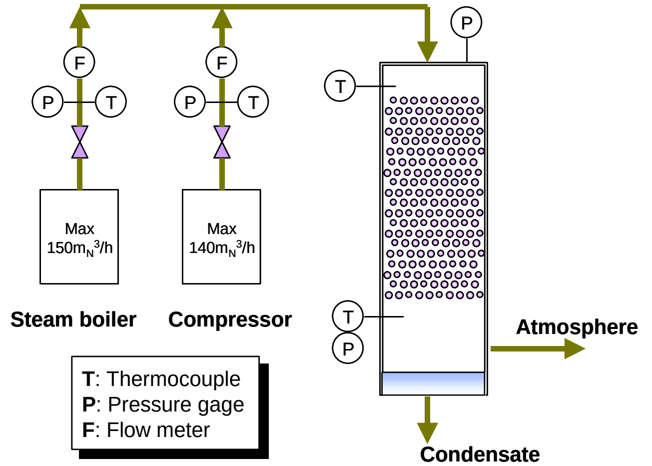

Fig. 3 Schematic of experimental apparatus

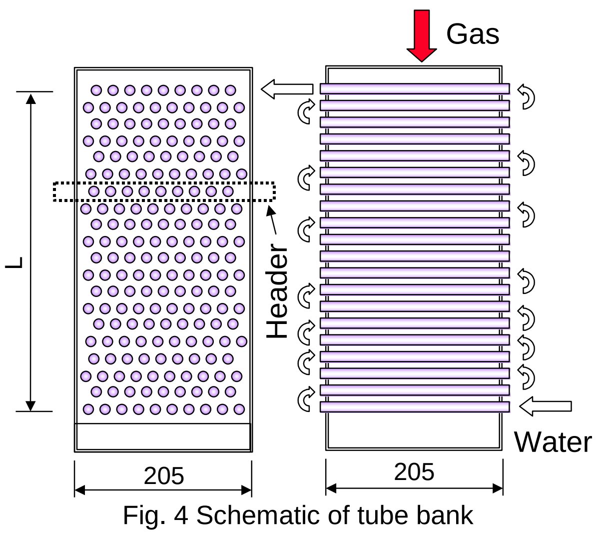

Fig. 4 Schematic of tube bank

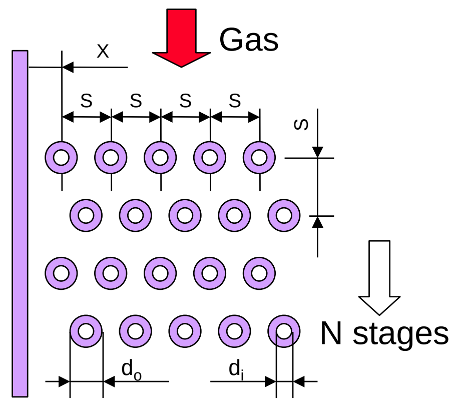

Fig. 5 Schematic of tube array

Table1 Major test conditions

	Type A	Type B
L (mm)	820	160
S (mm)	20.5	8
d_o (mm)	10.5	4
d_i (mm)	8.1	2
Stages	40	20
Number of tubes	380	500
X(mm)	10.3	6.5
Gas-side Heat transfer area (m^2)	2.57	1.29
Weight of tubes (kg)	21.6	7.65

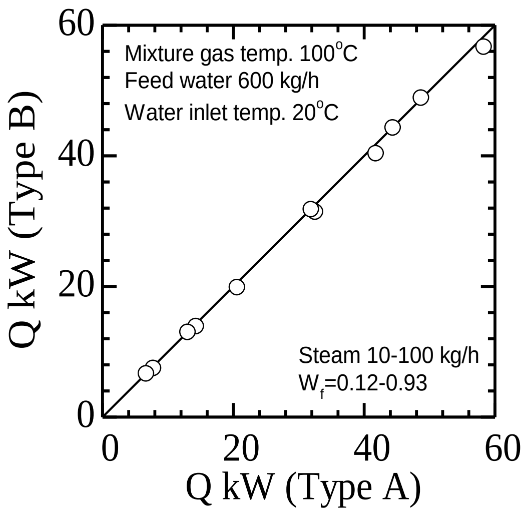

Fig. 6 Comparison of recovered heat

Table2 Major test conditions

Steam mass flow rate kg/h	21.6 ~ 86.9
Steam mass concentration w_f	0.2~ 1
Mixture gas temp. T_f (°C)	77.0 ~ 113.2
Cooling water flow rate kg/h	360~ 614
Inlet temp. of cooling water (°C)	8.7 ~ 26.8

COMPARISON OF EXPERIMENTAL RESULT AND PREDICTION

Shown in Fig.7 is the comparison of the experimental result and prediction at the steam mass concentration w_f=0.35 in Type A heat exchanger. Feed water and steam mass flow rates are 600 and 23.5 kg/h, respectively. The key ○ and △ are the measured temperatures of gas and water, respectively. The lines in the figure are the predictions. The solid lines are the temperatures of gas and water in the tube bank. The a-dot-dashed line and the two-dots-dashed line are the interfacial temperature of condensate and the inner wall temperature of tubes, respectively. The dashed line which is the saturation temperature (dew point) corresponding to the partial pressure of steam in the mixture gas agrees well with the gas temperature. As the outer wall temperature is smaller than the dew point, the condensation on the wall takes place throughout the heat exchanger. The dew point and gas temperature decrease with increasing stages indicating the condensation of steam in the mixture gas.

In the present experiments, the entrained and dispersed condensate in the mixture gas tends to wet the thermocouples among the tube bank in spite of the small umbrella installed above the thermocouple. The wet thermocouples indicate the lower value than the actual gas temperature. Though the prediction for the gas temperature is slightly higher than the experimental result, the prediction for the water temperature agrees well with the experimental result indicating the proper prediction.

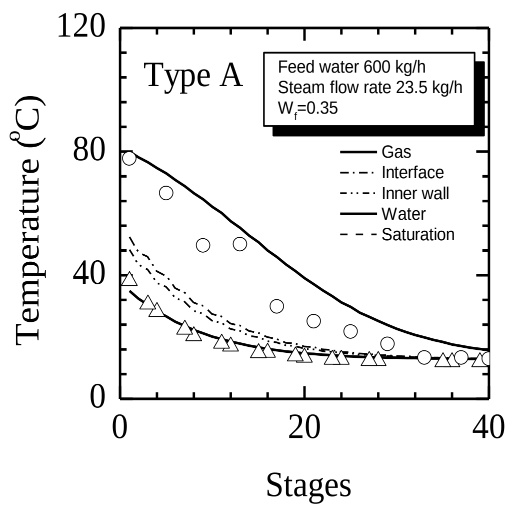

Fig. 7 Temperature distribution at w_f=0.35 in Type A

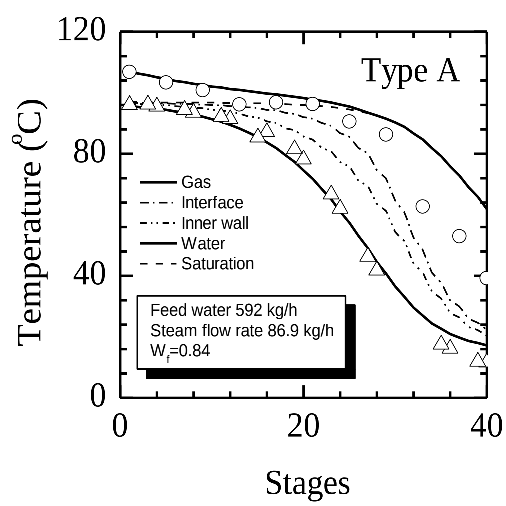

Fig. 8 Temperature distribution at w_f=0.84 in Type A

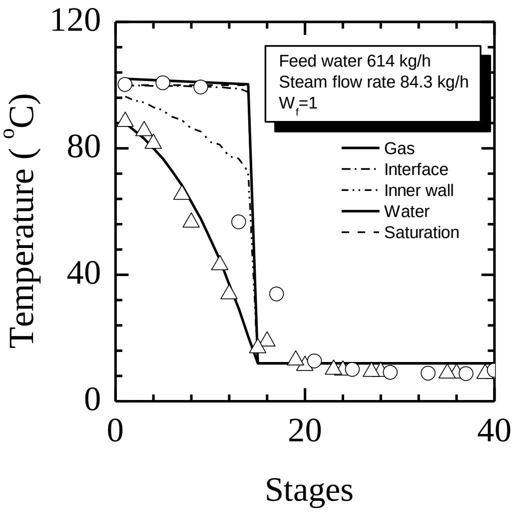

Fig. 9 Temperature distribution at w_f=1 in Type A

Shown in Fig.8 is the comparison of the experimental result and prediction at the steam mass concentration w_f=0.84 in Type A heat exchanger. The temperature increase of cooling water is strongly depressed as the saturation temperature and the cooling water temperature is nearly equal at the upper heat exchanger. When the cooling water temperature becomes enough below the saturation temperature approximately at 12th stage, the condensation on the tubes initiates and the temperature rise of cooling water significantly increases.

The saturation temperature merges with the gas temperature approximately at 25th stage and the condensation of steam occurs not only on the tubes but also in the mixture gas below this stage. The wet thermocouples also indicate the lower value than the actual gas temperature. Though the prediction for the gas temperature is slightly higher than the experimental result, the prediction for the water temperature agrees well with the experimental result indicating the proper prediction.

In the case of pure steam, w_f=1, in Type A heat exchanger as shown in Fig.9, the inlet gas temperature was equal to the saturation temperature of 100°C as the outlet of heat exchanger is open to the atmosphere. Heat resistance of condensate film governs the heat transfer process in this case. When the gas temperature sharply dropped approximately at 14th stage, all the steam has condensed above this height. Below the height, air can invade from the outlet of heat exchanger and the heat transfer was strongly depressed. So the temperature field is kept nearly at the inlet temperature of cooling water. Even in this case, the prediction using the modification factor M_f of 100 agrees well with the experiment.

Shown in Fig.10 is the comparison of the experimental result and prediction at the steam mass concentration w_f=0.37 in Type B heat exchanger using the smaller tubes when the steam flow rate is 24.1 kg/h. The tube outer diameter of Type B is 4 mm and nearly equal to the droplets size falling from the upper condensing tubes. So the comparison of the experimental result and prediction assuming the uniform thin film of condensate is interesting and important for the further improvement of prediction method. Though the prediction for the gas temperature is slightly higher than the experimental result due to the wet thermocouples, the prediction for the water temperature agrees well with the experimental result indicating the proper prediction for the heat exchanger of smaller tubes.

Shown in Fig.11 is the comparison of the experimental result and prediction at the steam mass concentration w_f=0.37 in Type B heat exchanger when the feed water flow rate is reduced to 360 kg/h. The non-uniform flow in the mixture gas side can be suggested with the scattering of cooling water temperature at the upper heat exchanger. This scattering can be seen also in Fig.10.

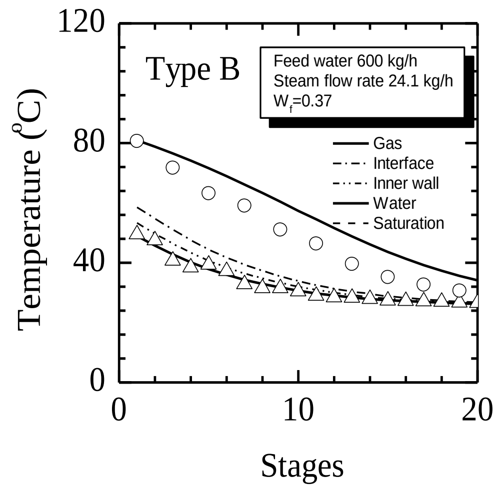

Fig. 10 Temperature distribution at feed water flow rate of 600 kg/h and w_f=0.37 in Type B

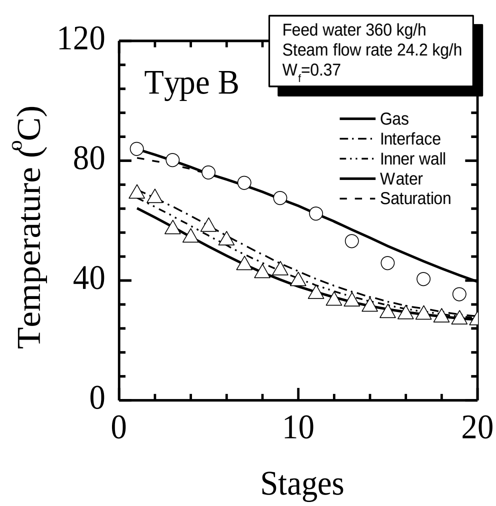

Fig. 11 Temperature distribution at feed water flow rate of 360 kg/h and w_f=0.37 in Type B

However the increase of cooling water temperature due to the reduced flow rate can be properly predicted. The reduced amount of condensate assures the correct measurement of gas temperature in this case. The proper prediction is also indicated in the comparison of the experimental results and prediction.

Shown in Fig.12 is the comparison of the experimental result and prediction at the steam mass concentration w_f=0.49 in Type B heat exchanger when the steam flow rate is increased to 60.2 kg/h. The scattering of cooling water temperature at the upper heat exchanger disappears suggesting the uniform mixture gas flow at the increased flow rate. The increase of cooling water temperature due to the increased steam flow rate can be properly predicted. The proper prediction is also indicated in the comparison of the experimental results and prediction.

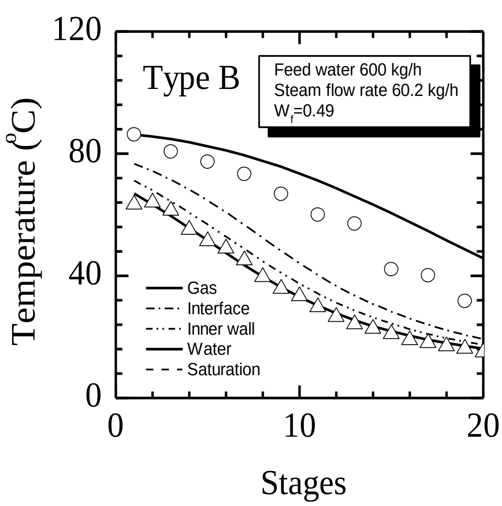

Fig. 12 Temperature distribution at feed water flow rate of 600 kg/h and w_f=0.49 in Type B

CONCLUSION

(1) Based on the previous basic studies, a thermal hydraulic prediction method for latent heat recovery exchangers was proposed. For the condensation of steam on heat transfer tubes, the modified Sherwood number taking account of the mass absorption effect on the wall was used.

(2) Two kinds of compact heat exchanger with staggered banks of bare tubes of 10.5 or 4mm in outer diameter was designed with the prediction method. The more compactness was obtained with the smaller tubes at a designed heat recovery.

(3) The thermal hydraulic behavior in the compact heat exchangers of bare tubes of 10.5 or 4mm was experimentally studied with air-steam mixture gas. In the parametric experiments varying the steam mass concentration, the temperature distributions of cooling water and mixture gas were measured. The experimental results agreed well with the prediction proposed in this study and the more compactness with the smaller tubes was proved.

REFERENCES

(1) Osakabe, M., Ishida, K., Yagi, K., Itoh, T. and Ohmasa, M., "Condensation heat transfer on tubes in actual flue gas (Experiment using flue gas at different air ratios)", (in Japanese), *Trans. of JSME* , 64-626, B, (1998), pp.3378-3383.

(2) Osakabe, M., Yagi, K., Itoh, T. and Ohmasa, M., "Condensation heat transfer on tubes in actual flue gas (Parametric study for condensation behavior)", (in Japanese), *Trans. of JSME*, 65-632, B, (1999), pp.1409-1416.

(3) Osakabe, M., Itoh, T. and Ohmasa, M., "Condensation heat transfer on spirally finned tubes in actual flue gas", (in Japanese), *J. of MESJ* , 35-4, (2000), pp.260-267.

(4) Osakabe, M., Itoh, T. and Yagi, K., "Condensation heat transfer of actual flue gas on horizontal tubes", *Proc. of 5th ASME/JSME Joint Thermal Eng. Conf.*, AJTE99-6397, (1999).

(5) Osakabe, M., "Thermal-hydraulic behavior and prediction of heat exchanger for latent heat recovery of exhaust flue gas", *Proc. of ASME*, HTD-Vol.364-2, (1999), pp.43-50.

(6) Osakabe, M., Tanaka, O. and Kawakami, A., "Prediction and behavior of heat exchanger for latent heat recovery of exhaust flue gas", (in Japanese), *Trans. of JSME*, 66-649, B, (2000), pp.2471-2477.

(7) Osakabe, M., "Thermal-hydraulic behavior and prediction of heat exchanger for latent heat recovery of exhaust flue gas", *Proc. of ASME*, HTD-Vol.364-2, (1999), pp.43-50.

(8) Osakabe, M., "Latent heat recovery from oxygen-combustion flue gas", *Proc. of 35th Intersociety Energy Conversion Conference*, Vol.2, (2000), pp.804-812.

(9) Osakabe,M., Horiki,S., Itoh,T. and Mouri,K., "Latent Heat Recovery from Oxygen-Combustion Boiler", *Proc. of RAN2001 (Nagoya)*, (2001).

(10) Osakabe,M., Horiki,S., Itoh,T. and Haze,I., "Latent Heat Recovery from Actual Flue Gas", *Proc. of IHTC 12 (Grenoble)*, (2002).

(11) Osakabe,M. and Ikeda,N., "Condensation Heat Transfer in Wide Range of Non-condensing Gas Fraction", (in Japanese), *Trans. of JSME*, 69-685, (2003), pp.2107-2113

(12) Zukauskas, A., *Advances in Heat Transfer*, 8, Academic press, New York, (1972), pp.93-160

(13) JSME, Data Book: *Heat Transfer 3rd Edition*, (in Japanese), 1983.

(14) Lindsay, A.L. and Bromley L.A., "Thermal conductivity of gas mixtures", *Indust. Engng. Chem*., 42, (1950), pp.1508-1510.

(15) Wilke, C.R., "A viscosity equation for gas mixture", *J. Chem. Phys*., 18, (1950), pp.517-519.

(16) Fujii, T., Kato, Y. and Mihara, K., "Expressions of transport and thermodynamic properties of air, steam and water", *Univ. Kyushu Research Institute of Industrial Science Rep.66*, (1977), pp.81-95.

(17) Osakabe, M., "Thermal-hydraulic study of integrated steam generator in PWR", *J. Nucl. Sci. & Technol*., 26(2), (1989), pp.286-294.

(18) McAdams, W.H., *Heat transmission*, McGRAW-HILL, (1954).