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Compact Heat Exchanger for Latent Heat Recovery of Exhaust Flue Gas

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Latent heat recovery from exhaust flue gas is a key technology to improve a thermal efficiency of power plants, fuel cells and boilers. A prediction code for the heat exchanger to recover the latent heat in the flue gas has been proposed in the previous basic studies. The code was used on the parametric study of the heat exchanger design for the latent heat recovery. The thermal-hydraulic behavior was calculated for several kinds of heat exchangers using finned tubes or bare tubes. The calculation result indicated that the most compact heat exchanger was that using the bare tube of small diameter. So the compact countercurrent cross-flow heat exchanger using bare tubes of SUS304 was designed and constructed to prove its high ability. The outer and inner diameters of the bare tube were 10.5 and 8.1mm, respectively. The experimental study varying the air ratio of flue gas, feed water temperature and flow rate was conducted. The experimental results for the temperature distribution of water and flue gas in the heat exchanger with bare tubes of small diameter agreed well with the prediction. The proposed compact heat exchanger using small tubes was considered to be preferable for the latent heat recovery from the flue gas and the prediction code was useful for the design of the compact heat exchanger.

Key words: Actual flue gas, Bare tube, Compact, Condensation, Heat and mass transfer, Latent heat

1. INTRODUCTION

The most part of energy losses in a boiler is due to the heat released by the exhaust flue gas to atmosphere. The released heat consists of sensible and latent one. Recently, for a biological and environmental safety, a clean fuel such as a natural gas is widely used in the boiler. As the clean fuel includes a lot of hydrogen instead of carbon, the exhaust flue gas includes a lot of steam accompanied with the latent heat. So the latent heat recovery from the flue gas is very important to improve the boiler efficiency.

In the previous studies [Osakabe, 1998a, 1999a, 1999b], condensation heat transfer on horizontal stainless steel tubes has been investigated experimentally by using the actual flue gas from a natural gas boiler. The experiments were conducted using single and 2 stages of tubes at different air ratios and steam mass concentrations of the flue gas in a wide range of tube wall temperature. The condensation heat transfer was well predicted with a simple analogy correlation in the high wall temperature region and the low steam mass concentration typical in air combustion. In the low wall temperature region less than 30°C or the high steam mass concentration typical in oxygen combustion, the total heat transfer was higher than that predicted by the simple analogy correlation. The condensation heat transfer on spirally finned tubes of single stage was also investigated experimentally and theoretically [Osakabe, 1998b]. The fin efficiency at the condensing region was significantly lower than that at the dry region. The analogous heat and mass transfer model for the finned tube was proposed.

Also in the previous studies[Osakabe, 1999c, 2000], a prediction code for the heat exchanger to recover the latent heat in the flue gas was proposed. In the prediction, the flue gas was treated as a mixture of CO_2, CO, O_2, N_2 and H_2O, and the one-dimensional heat and mass balance calculation along the flow direction of flue gas was adapted. The heat and mass transfer on tubes was evaluated with a simple analogy correlation. For the finned tubes, the fin efficiency at the condensing region was calculated with a semi-empirical correlation obtained in the previous basic study. The effect of condensate film on the tubes was considered to be negligibly small for the heat transfer and pressure loss calculation in the latent heat recovery.

In the present study, the prediction code was used on the parametric study of the heat exchanger design for the latent heat recovery. The thermal-hydraulic behavior was calculated for several kinds of heat exchangers using finned tubes or bare tubes. The calculation result indicated that the most compact heat exchanger was that using the bare tube of small diameter. So the compact countercurrent cross-flow heat exchanger using bare tubes of SUS304 was designed and constructed to prove its high ability. The outer and inner diameters of the bare tube were 10.5 and 8.1mm, respectively. The bare tubes were arranged in a staggered bank of 10-9 rows and 40 stages.

The experimental study varying the air ratio of flue gas, feed water temperature and flow rate was conducted. The temperature distributions of water and flue gas in the heat exchanger were measured with sheathed T-type thermocouples of 0.5 mm in diameter. The pressure loss and the total amount of condensate generated in the heat exchanger were also measured.

2. CONSTITUTIVE EQUATIONS FOR PREDICTION

2.1 Fuel Combustion

The composition of natural gas fuel used in the test boiler is shown in Table 1. Volumetric concentrations of N_2, CO_2, O_2 and CO in the dry gas were measured by a gas analyzer. By using the measured concentration, the air ratio μ is calculate as

$$\mu = \frac{N_2}{N_2 - \frac{0.79}{0.21}(O_2 - 0.5 \cdot CO)} \quad (1)$$

The molar fraction of the carbon in the fuel of 1 mol can be calculated as,

$$CCR = 1 \times 0.88 + 2 \times 0.058 + 3 \times 0.045 + 4 \times 0.017 \quad \text{mol}$$

By using the volumetric flow rate of the fuel V_F, the volumetric flow rate of the carbon dioxide and monoxide, V_{COX} , is

$$V_{COX} = V_F \cdot CCR \quad (2)$$

The volumetric flow rate of dry gas V_d is,

$$V_d = \frac{V_{COX}}{CO_2 + CO} \quad (3)$$

The molar fraction of the hydrogen corresponding to CO_X of 1 mol can be calculated as,

$$CHR = (4 \times 0.88 + 6 \times 0.058 + 8 \times 0.045 + 10 \times 0.017) / CCR \quad \text{mol}$$

Considering that the air flow rate necessary for the combustion of the fuel is $V_d N_2/0.79$, the volumetric fraction of H_2O in the flue gas can be estimated as

$$H_2O = \frac{(CO_2 + CO) \cdot CHR/2 + N_2 \cdot R \cdot P_{sat}/(P_a \cdot 0.79)}{1 + (CO_2 + CO) \cdot CHR/2 + N_2 \cdot R \cdot P_{sat}/(P_a \cdot 0.79)} \quad (4)$$

where P_{sat} is the saturation pressure of steam in combustion air.

The volumetric flow rate of wet gas, V_{wt}, is

$$V_{wt} = \frac{V_d}{1 - H_2O} \quad (5)$$

When the flue gas temperature is T_f°C, the steam mass concentration C_f per unit volume of flue gas is,

$$C_f = \frac{H_2O \cdot 18}{22.4} \cdot \frac{273.15}{273.15 + T_f} \tag{6}$$

The steam mass concentration W_f per unit mass of flue gas is

$$w_f = \frac{H_2O \cdot 18}{H_2O \cdot 18 + (1 - H_2O)(CO_2 \cdot 44 + CO \cdot 28 + N_2 \cdot 28 + O_2 \cdot 32)} \tag{7}$$

CH_4	88.0 %
C_2H_6	5.8
C_3H_8	4.5
C_4H_{10}	1.7

TABLE 1. Composition of natural gas fuel(13A).

2.2 Heat and Mass Transfer in Gas Side

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

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

The convection heat flux is expressed as

$$q_V = h_V(T_f - T_W) \tag{9}$$

The condensation heat flux can be expressed as,

$$q_C = h_C L_W (C_f - C_W) \tag{10}$$

where C_w is the mass concentration of saturated steam at the wall temperature T_w. Based on the previous studies, the Nusselt number Nu_f for the average convective heat transfer coefficient in the range of $2\times10^3 < Re_f \leqq 5\times10^5$ is

$$Nu_f = c\, Re_f^{0.6}\, Pr_f^{m} (Pr_f / Pr_W)^{0.25} \tag{11}$$

Zukauskas(1972) proposed m=0.36 and

$$\text{For } S_1/S_2 < 2 \qquad c = 0.35\,(S_1/S_2)^{0.2} \tag{12}$$

$$\text{For } S_1/S_2 \geqq 2 \qquad c = 0.40 \tag{13}$$

for a staggered bank. For an analogous mass transfer process, the Nusselt number and Prandtl number in the heat transfer relation Eq.(11) are simply replaced by the Sherwood number and the Schmidt number, respectively. This procedure gives

$$Sh_f = c\, Re_f^{0.6}\, Sc_f^{m} (Sc_f / Sc_W)^{0.25} \tag{14}$$

The rule gives a correct relation for the limiting situation in which the differences in temperature and concentration are vanishingly small, and it is valid for independent analogous heat and mass transfer situations as well as for a combined heat and mass transfer process.

Flue gas was treated as a mixture of N_2, CO_2, O_2, CO and H_2O and its property was estimated with special combinations of each gas property proposed by the previous studies (JSME, 1983). For example, the heat conductivity and the viscosity were estimated with the methods by Lindsay&Bromley(1950), and Wilke(1950), 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 flue 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 air, respectively. The diffusivity of steam in air can be expressed as (Fujii *et. al.*, 1977),

$$D_{air} = 7.65\times10^{-5}\,\frac{(T + 273.15)^{11/6}}{P} \tag{16}$$

The previous experimental study for single stage of finned tube [Osakabe, 1998b] showed the following empirical correlation was available in the range of $2\times10^3 < Re_f \leqq 5\times10^5$.

$$Nu_f = j\, Re_f\, Pr_f^{0.33} \tag{17}$$

where

$$j = C_1 C_3 C_5 \left(\frac{d + L_F}{d}\right)^{0.5} \tag{18}$$

$$C_1 = 0.25\, Re_f^{-0.35} \tag{19}$$

$$C_3 = 0.35 + 0.65 e^{-0.25 L_F / S_F} \tag{20}$$

$$C_5 = 0.7 \tag{21}$$

The fin efficiency can be calculated by,

$$\eta = Y_F\left[0.45\ln\left(\frac{d+L_F}{d}\right)(Y_F-1)+1\right] \tag{22}$$

where

$$Y_F = X_F(0.7+0.3X_F) \tag{23}$$

$$X_F = \frac{\tanh(mb)}{mb} \tag{24}$$

$$m = \left[\frac{2h}{\lambda_F t_F}\right]^{0.5} \tag{25}$$

$$b = L_F + t_F/2 \tag{26}$$

The fin efficiency strongly depends on the heat transfer coefficient, h, in Eq.(25). These correlations were obtained from the single phase experiment where the heat flux is the multiple of the constant heat transfer coefficient and the temperature difference between the wall and the fluid. Although the condensation heat flux can not be expressed with the above simple relation, it is considered that the Eq.(25) with the larger heat transfer coefficient taking account of the condensation heat transfer would give an approximation. The previous study showed the fin efficiency could be evaluated with following equivalent heat transfer coefficient in the condensation region.

$$h = h_V + \beta\frac{h_C L_W(C_f - C_W)}{T_f - T_W} \tag{27}$$

where β=1 for the usual calculation.

For an analogous mass transfer, the simple analogy between heat and mass transfer gives,

$$Sh_f = j\,Re_f\,Sc_f^{0.33} \tag{28}$$

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;

$$H_2O(N+1) = \frac{H_2O(N)\cdot V_{wt} - \dfrac{q_C A_W}{L_W}\cdot\dfrac{22.4}{18}}{V_{wt} - \dfrac{q_C A_W}{L_W}\cdot\dfrac{22.4}{18}} \tag{29}$$

$$T_f(N+1) = T_f(N) - \frac{q_V A_W}{C_{Pf}\rho_f(273.15+T_f)/273.15\cdot V_{wt}} \tag{30}$$

In the above equations, A_w is the effective heat transfer area per a stage defined as,

$$A_W = A_B + \eta A_F \tag{31}$$

where A_B is the base tube surface and A_F is the heat transfer area of fin. In case of bare tube, A_B is the outer surface of tube and A_F is zero.

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 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, ΔH_2O, as;

$$\Delta T_f = \frac{18}{22.4}\cdot\frac{L_W}{C_{Pf}\rho_f(273.15+T_f)/273.15}\cdot\Delta H_2O \tag{32}$$

The average film thickness of condensate was calculated by the method shown in Appendix. As the existence of the film did not affect significantly the prediction for the present experiment, the thermal resistance of the film was neglected.

2.3 Heat Conduction in Tube

The heat conductivity for the inconel or austenite stainless steel is given with the following approximate correlation [Osakabe, 1989].

$$\lambda_t = 13.2 + 0.013T_t \qquad \text{W/(m K)} \tag{33}$$

where T_t is the average temperature of tube as,

$$T_t = \frac{T_W + T_{Wi}}{2} \tag{34}$$

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

$$q_W = \frac{2\lambda_t(T_W - T_{Wi})}{d\ln(d/d_i)} \tag{35}$$

2.4 Heat Transfer in Water Side

Heat transfer correlation by Dittus-Boelter taking account of the pipe inlet region is used. The inlet coefficient by McAdams(1954) is used for the modification.

$$Nu = 0.023\,Re^{0.8}\,Pr^{0.4}\left(1+\left(\frac{d_i}{L}\right)^{0.7}\right) \tag{36}$$

where L is the heating length of tube.

2.5 Pressure Loss Calculation

The pressure loss per a stage of tube is,

$$\Delta P = 2f\,\rho_f\,u^2 \tag{37}$$

For the staggered bank of bare tube, Jacob(1938) proposed the following coefficient f,

$$f = \left[0.25 + \frac{0.118}{\left\{\left(\frac{S_1}{d}\right)-1\right\}^{1.08}}\right] Re_f^{-0.16} \tag{38}$$

For the staggered bank of finned tube, the empirical correlation by ESCOA(1979) is,

$$f = C_2 C_4 C_6 \left(\frac{d+L_F}{d}\right)^{0.5} \tag{39}$$

where

$$C_2 = 0.07 + 8\,Re_f^{-0.35} \tag{40}$$

$$C_4 = 0.11\left[0.05\frac{S_1}{d}\right]^{-0.7(L_F/S_F)^{0.20}} \tag{41}$$

$$C_6 = 1.1 + \left[1.8 - 2.1e^{-0.15N_f^2}\right]e^{-2.0S_2/S_1} - \left[0.7 - 0.8e^{-0.15N_f^2}\right]e^{-0.6S_2/S_1} \tag{42}$$

3. PARAMETRIC STUDY

Heat transfer tubes are installed in the rectangular duct of 205×205mm to recover the latent heat in flue gas as shown in Fig.1. The horizontal heat transfer tubes are arranged in multi-stages and the neighboring stages are connected with headers. The water flow rate and temperature to each tube at a stage is maintained at the same values with the header. The flue gas and the generated condensate vertically cross the horizontal tubes. This duct size is approximately the same as the flue gas duct of test boiler used in this experiment. The outlet temperature of flue gas from the boiler is 280°. The flue gas is generated with natural gas 13A at the flow rate of 15 m_N^3/h and the air ratio of 1.2. The flow rate and inlet temperature of feed water are 600kg/h and 20°, respectively. The temperature of the feed water is increased from 20° to 60° with the heat recovery.

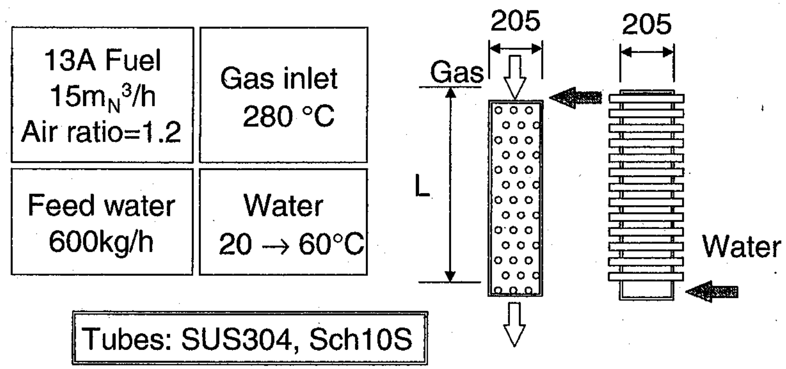

FIGURE 1. Boundary conditions for parametric study.

The various kinds of heat transfer tubes are installed in the duct as the parametric design study as shown in Table 2. Bare tubes of 21.7 and 10.5 in outer diameter, spirally finned tubes are selected in the study. Figure 2 shows the schematic of spirally finned tube considered in this study. The plate fin of 1mm in thickness is welded on the base tubes of 21.7 mm in diameter and the fin heights are 12, 8 and 3mm. The vertical and horizontal pitch of tube arrangement is the same and larger than the tube outer diameter by 10mm considering the fabrication ability of holes at the tube sheets. The maximum number of tubes was 333 using the tube of 10.5 mm in outer diameter. The minimum number was 102 using the finned tube of 12mm in height, which is often used in conventional economizer for the sensible heat recovery. The most compact economizer was that using small bare tubes of 10.5mm in outer diameter. The height and the total tube weight of the economizer using small bare tubes were 718mm and 18.9kg, respectively. The pressure loss in the waterside was slightly higher than the others but could be allowable. On the other hand, the total weight of that using the finned tubes of 12mm in height was 62.6kg. The bare tube of small diameter is preferable for the compact design of economizer for the latent heat recovery. As the fin efficiency is relatively low due to the high heat transfer coefficient of condensation, the bare tube is preferable in the latent heat recover region.

Tube type	Bare1	Bare2	Fin1	Fin2	Fin3
Pipe diameter (mm)	10.5	21.7	21.7	21.7	21.7
Pitch (mm)	20.5	34.2	51.0	41.0	34.2
Stage	35	40	29	25	27
Pipe no.	333	220	102	113	149
Fin pitch	-	-	5	5	5
Fin height	-	-	12	8	3
Length L (mm)	718	1370	1480	1030	923
Heat transfer area (m^2)	2.25	3.07	12.4	8.73	5.04
Tube weight (kg)	18.9	46.1	62.6	51.1	42.5
Gas pressure loss (mmAq)	11.5	21.6	19.8	22.1	22.4
Water pressure loss (mmAq)	515.0	65.3	117.0	60.5	43.9

TABLE 2. Comparison of bare and finned tubes heat exchanger.

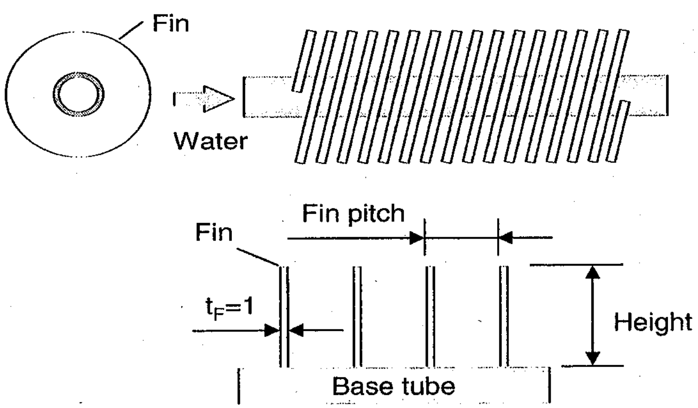

FIGURE 2. Schematic of spirally finned tube.

4. COMPACT HEAT EXCANGER EXPERIMENT

The calculation result indicated that the most compact heat exchanger was that using the bare tube of small diameter. So the compact countercurrent cross-flow heat exchanger using bare tubes of SUS304 was designed and constructed to prove its high ability. Shown in Fig.3 is a schematic of experimental apparatus. The experiment was conducted by using flue gas generated with the combustion of air and natural gas fuel. The flue gas from a natural gas boiler was led to the inlet plenum of the test heat exchanger. The flue gas was released to atmosphere from the outlet plenum. The countercurrent cross-flow heat exchanger consisted of bare tubes of 10.5 and 8.1mm in outer and inner diameter, respectively. The horizontal bare tubes of small diameter were arranged as a bank of 10-9 rows and 40 stages and the neighboring stages were connected with the rectangular header. The effective heating length of the tubes was 200mm. The temperature distributions of water and flue gas in the heat exchanger were measured with sheath T-type thermocouples of 0.5mm in diameter. The thermocouple signals were transferred to a data logger and analyzed.

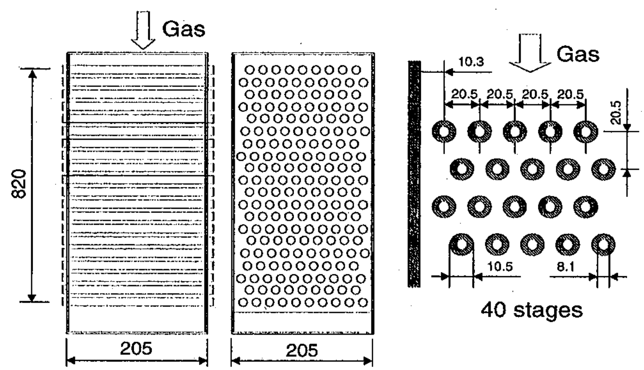

FIGURE 3. Experimental apparatus.

The experimental study varying the air ratio of flue gas, feed water temperature and flow rate was conducted. The major experimental conditions are listed in Table 3. The typical comparison of the experimental result and prediction is shown in Fig.4. In the figure the solid lines are the temperatures of gas and water in the bank. The a-dot-dashed line and the two-dots-dashed line are the inner and outer wall temperature of tubes, respectively. The dashed line is the saturation temperature (dew point) corresponding to the par-

Air ratio	1.36-1.57
Feed water (kg/h)	479-616
13A Fuel (m_N^3/h)	15.8-16.8
Gas inlet temp. (°C)	286-294
Water inlet temp. (°C)	14.3-21.7

TABLE 3. Experimental conditions.

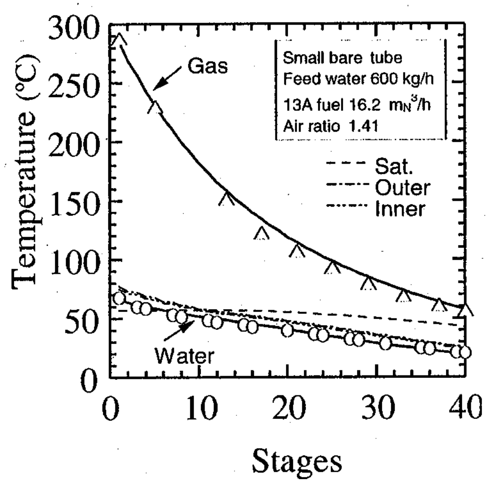

FIGURE 4. Temperature distribution in compact heat exchanger.

tial pressure of steam in the flue gas. When the outer wall temperature is smaller than the dew point, condensation on the wall takes place. The dew point decreases along the stages at the condensation region as the steam concentration decreases with the condensation. As the gas and water velocity changes stage-by-stage corresponding to the change of flow area in the present experimental apparatus, the wall temperatures shows zigzag polygonal line. The key O and Δ are the measured temperatures of gas and water, respectively. The experimental results for the temperature distributions of water and flue gas in the heat exchanger agree well with the prediction. The proposed compact heat exchanger using small bare tubes was considered to be preferable for the latent heat recovery from the flue gas and the prediction code was useful for the design of the compact heat exchanger.

Shown in Fig.5 is the comparison of experimental result and prediction for the pressure loss throughout the heat exchanger. As the empirical correlation obtained in the non-condensing region was used in the one-dimensional calculation, the experimental results for the condensing banks of small bare tubes were slightly larger than the prediction. The effect of condensate film on the tubes should be considered for the precise calculation of pressure loss.

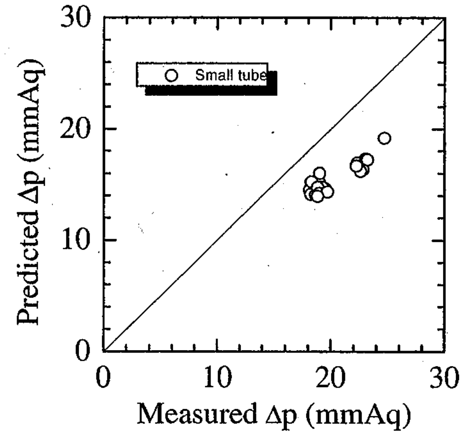

FIGURE 5. Predicted and measured pressure difference.

5. CONCLUSION

1. The prediction code was used on the parametric study of the heat exchanger design for the latent heat recovery. The thermal-hydraulic behavior was calculated for several kinds of heat exchangers using finned tubes or bare tubes. The calculation result indicated that the most compact heat exchanger was that using the bare tube of small diameter.
2. The compact countercurrent cross-flow heat exchanger using bare tubes of SUS304 was designed and constructed to prove its high ability. The outer and inner diameters of the bare tube were 10.5 and 8.1mm, respectively. The bare tubes were arranged in a staggered bank of 10-9 rows and 40 stages. The experimental study varying the air ratio of flue gas, feed water temperature and flow rate was conducted. The experimental results for the temperature distributions of water and flue gas in the heat exchanger with bare tubes of small diameter agreed well with the prediction. The experimental results of

the pressure loss through the condensing banks of small bare tubes were slightly larger than the prediction as the empirical correlation obtained in the non-condensing region was used.

3. The proposed compact heat exchanger using small bare tubes was considered to be preferable for the latent heat recovery from the flue gas as the fin efficiency was relatively low due to the high heat transfer coefficient of condensation.

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NOMENCLATURE

A	heat transfer area
A_B	heat transfer area of base tube
A_F	heat transfer area of fin
C	mass concentration per fluid of a unit volume
C_p	specific heat
d	outer diameter of tube
d_i	inner diameter of tube
D	mass diffusivity
g	acceleration due to gravity
h_V	heat transfer coefficient
h_C	mass transfer coefficient
L_W	latent heat
Nu	Nusselt number [$= h_V d / \lambda$]
Nr	total number of stages
P	pressure
Pr	Prandtl number [$= \nu / \kappa$]
q	heat flux
R	relative humidity of combustion air
Re	Reynolds number [$= ud / \nu$]
S_1	spanwise pitch
S_2	flow-directional pitch
Sh	Sherwood number [$= h_c d / D$]
Sc	Schmidt number [$= \nu / D$]
T	temperature
u	velocity at minimum flow area
V	volumetric flow rate
w	mass concentration per fluid of an unit mass
x	relative humidity of air

Greek

η	fin efficiency
κ	thermal diffusivity [$\lambda / \rho Cp$]
λ	heat conductivity
μ	air ratio
ν	kinematic viscosity
ρ	density

Subscript

a	atmosphere
C	condensation
COX	carbon dioxide and monoxide
d	dry gas
F	fuel
f	flue gas
i	condensation surface
V	convection
W	wall
N	standard condition at 0°C and atmospheric pressure
sat	saturated condition of steam
sub	subcooling
wt	wet gas

REFERENCES

ESCOA FINTUBE CORPORATION, 1979, SOLIDFIN HF.

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

Jakob, M., 1938, “Heat transfer and flow resistance in cross flow of gases over tube banks”, *Trans. ASME*, 60, 384.

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

Lindsay, A.L. and Bromley L.A., 1950, “Thermal conductivity of gas mixtures”, *Indust. Engng. Chem.*, 42, 1508-1510.

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

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

Osakabe, M., Ishida, K., Yagi, K., Itoh, T. and Ohmasa, M., 1998a, "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, 3378-3383.

Osakabe, M., Itoh and T. Ohmasa, 1998b, "Condensation heat transfer on tubes in actual flue gas (Experiment using spirally finned tubes)", (in Japanese), *Proc. of MESJ 61st Meeting*, 70-73.

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

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

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

Osakabe, M., 2000, "Latent Heat Recovery from Oxygen-Combustion Flue Gas", *Proc. of 35th Intersociety Energy Conversion Conference*, Vol.2, 804-812

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

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

APPENDIX

It is assumed that all the condensate generated at the upper stage flows on the tubes as a laminar film as shown in Fig. 6. The momentum balance dominated by viscous and gravity force gives the velocity distribution at θ° from the tube top:

$$u = \frac{(\rho_L - \rho_G) g \sin\theta}{\mu_L}\left(y\delta - \frac{y^2}{2}\right) \tag{42}$$

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

$$\delta = \left[\frac{1.5\mu_L m}{\rho_L(\rho_L - \rho_G) g \sin\theta}\right]^{1/3} \tag{43}$$

The heat conductivity of film is

$$K = \frac{\lambda_L}{\delta} = \left[\frac{\lambda_L{}^3 \rho_L(\rho_L - \rho_G) g \sin\theta}{1.5\mu_L m}\right]^{1/3} \tag{44}$$

The average conductivity from q= 0° to q=p is

$$\overline{K} = \frac{1}{\pi}\int_0^{\pi} K\, d\theta = 0.72\left[\frac{\lambda_L{}^3 \rho_L(\rho_L - \rho_G) g}{\mu_L m}\right]^{1/3} \tag{45}$$

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

$$\overline{\delta} = \frac{\lambda_L}{\overline{K}} \tag{46}$$

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

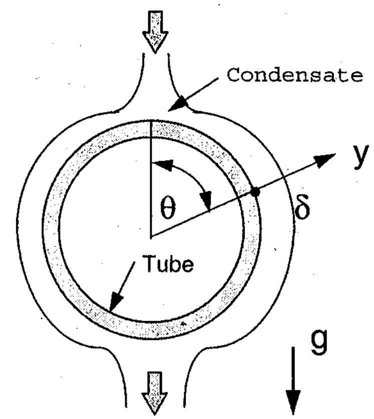

FIGURE 6. Heat conductance of condensate film.