Improved performance of secondary heat exchanger for latent heat recovery from flue gas using mini-tubes

Junpei YAMASHITA[a], Yoshio UTAKA[b*]

[a] Graduated School of Engineering, Yokohama National University, Tokiwadai 79-5, Hodogaya, Yokohama 240-8501, Japan

[b] Faculty of Engineering, Yokohama National University, Tokiwadai 79-5, Hodogaya, Yokohama 240-8501, Japan

ABSTRACT:

A new type of heat exchanger, in which flue gas flows inside thin tubes and cool water is on the shell-side, was proposed to develop the performance and compactness of shell and tube type heat exchangers for latent heat recovery from flue gas. The experimental heat transfer characteristics of single tubes were systematically investigated to determine the effects of tube diameter (1.0-5.0 mm) and length (7-100 mm). Furthermore, a correlation between the non-dimensional bulk mean temperature and the ratio of effective tube length to the thermal entrance region was proposed and was correlated well with the measurement data. Prediction of the heat exchanger performance using this correlation was possible. As a result, it was elucidated that the using mini-tubes is remarkably effective to reduce the size of heat exchanger due to enhancement of heat transfer coefficient and enlargement of heat transfer surface. The volume with 1 mm inner diameter of tubes was approximately 5 percent of that with 5 mm in diameter.

Key Words: Water Heater, Latent Heat Recovery, Compact Heat Exchanger, Thermal Entrance Region

* Corresponding author. Tel/Fax: +81 45 339 3909.

E-mail: utaka@ynu.ac.jp

Nomenclature

Variables

A heat transfer area (mm^2)

A^* flow passage area (mm^2)

d tube diameter (mm)

h heat transfer coefficient ($W/(m^2 \cdot K)$)

i specific enthalpy (kJ/kg)

K overall heat transfer coefficient ($W/(m^2 \cdot K$)

k thermal conductivity ($W/(m \cdot K)$)

L effective tube length (mm)

L_c cooling length (mm)

L_T thermal entrance region (mm)

N_c number of tube bank column

N_R number of tube bank row

M mass flow rate (kg/s)

Pr Prandtl number

ΔP pressure loss (Pa)

Re Reynolds number

S tube spacing (mm)

T temperature (°C)

t tube wall thickness (mm)

U flow velocity (m/s)

x specific humidify (kg/kg)

X heat exchanger width (mm)

Y heat exchanger height (mm)

Greek symbol

ρ density (kg/m^3)

Subscripts

c coolant

d dew point or drain

g moist air

i inner diameter or in-flow

o outer diameter or out-flow

t tube
v steam
w wall

1. Introduction

The objective of this study is to improve the performance of the heat exchanger for latent heat recovery and propose new heat exchanger structures for recovery of the condensation latent heat from flue gases. In the conventional model of a household gas water heater, as shown in Figure 1(a), cool water is heated using only the sensible heat of flue gases, after which the gases are discarded as exhaust. This flue gas contains steam at approximately 200 °C, and thus has a relatively large amount of heat. Recently, water heaters with heat exchangers (secondary (2nd) heat exchangers) that collect both the latent heat and surplus sensible heat of exhaust gas have been used to heat water in advance (Figure 1(b)), which has improved the efficiency of heat utilization by approximately 15%. However, installation of a secondary heat exchanger can be problematic because it increases the size of the water heater and the corrosive condensate obstructs the heat exchange process. Therefore, it is necessary to develop a more compact heat exchanger in which the condensate can be easily treated.

In a typical secondary heat exchanger, water flows inside tubes and the gases flow inside a shell (Fig. 2(a)). Most of these types of heat exchangers employ smooth tubes made of stainless steel as the heat transfer tube. This type of heat exchanger has been investigated to improve performance and be downsized. Osakabe *et al.* [1,2] conducted experimental and numerical investigations on the heat exchange performance of banks of tubes with aligned and staggered arrangements using combustion gas. Effectiveness of narrow tubes was shown for compact heat exchanger. Jeong *et al.* [3] investigated the influence of the inlet combustion gas velocity and temperature to simulate a heat exchanger in which the gas flows downward among horizontally arranged tubes. Their experimental results and prediction are in agreement within an error range of 10 % and the increase in row number of tube bank is effective in enhancing the heat transfer rate. Fujiwara *et al.* [4] conducted a numerical investigation on a heat exchanger for latent heat recovery where the gas flows horizontally among a vertically arranged tube bank. They considered that the heat exchanger size could be compacted by using thinner tubes. Jeong *et al.* [5] proposed an analytical model for heat and mass transfer characteristics in fuel gas to predict the heat transmission and condensation rate from a fuel gas. This model corresponded well with experimental results obtained from their test heat exchanger. Parry *et al.* [6] reported on the development of efficient numerical simulation for a shell and tube type heat exchanger in which heat transfer occurred by fixing the mass of phase change material. They showed that the simulation was in good agreement with measurement temperature and the difference between one dimensional and two dimensional calculations was within 8.5%. Although, the shell and tube type has high efficiency but the proposed type has limitations of downsizing from those researches. Expansion of the heat transfer area has been considered with investigations on fin-tube type heat exchangers. Shi *et al.* [7] showed that the performance of a fin-tube heat exchanger could be predicted within an error of ±7.3% by comparison of numerical and experimental results. Kang *et al.* [8] studied a type of fin-tube with grooves on the fin plate, while Joardar *et al.* [9] realized enhancement of the heat transfer performance by attachment of winglet-fins. Kanzaka *et al.* [10] experimentally investigated the heat and mass transfer characteristics of a heat exchanger with finned tubes and showed that heat transfer tubes with fins have a significant influence on mass transfer. Hwang *et al.* [11] proposed a heat exchanger with titanium finned tubes that has higher corrosion resistance than that with stainless steels tubes. Compact plate-fin type heat exchangers have also been studied to augment the heat transfer area. Kolev *et al.* [12] proposed a new type lamellar heat exchanger that was more compact and exhibited higher performance according to simulation results. Kawaguchi *et al.* [13] developed a new plate-fin type heat exchanger for latent heat recovery and experimentally investigated the heat transfer performance. Cheng *et al.* [14] investigated the polymer compact heat exchanger, which showed high heat transfer coefficient (up to 24000 W/m^2K). Park *et al.* [15] proposed a new method with condensation on the non-wettable surface and it was shown that the method can be extended to various heat exchanger geometries. Although those researchers developed the performances of the compactness of heat exchangers, the structures were complicated, for example, the channel of the heat exchanger with addition of finely arranged thin plate was utilized. Therefore, it is probable that their heat exchanger has low productivity.

Thus, many researches on heat recovery from moist gas has been performed. However, the configuration of heat exchanger proposed in this study is not studied up to now. In this study, a simple and versatile heat exchanger, the shell and tube type, is investigated with a change in the fundamental system. When making compact heat exchangers, thinner tubes are considered to increase the heat transfer area. However, with water flow in these thin tubes, problems such as hydraulic pressure, condensate drainage, and freezing may occur, which places a limit on the compactness that can be achieved with a typical heat exchanger. Therefore, a heat exchanger that employs gas flow within the tubes (Figure 2(b)) is examined with the aim of developing a compact heat exchanger that has a large heat transfer area by implementing thin tubes. Drainage of cooling water on the shell side can be easily achieved and the efficient exclusion of condensate is expected using gas tube flow. It is fundamental to investigate the heat transfer characteristics of a single tube to simulate the characteristics of the heat exchanger. The performance of a heat exchanger can be predicted according to the characteristics of the single tube because the wall temperatures of all tubes are similar and near the coolant temperature for a large flow rate of coolant under the condition of gas water heaters. It is considered that the variations of the inlet gas flow rate among tubes in tube bank are small because the inlet gas is a single phase due to the higher inlet temperature. The differences in the density and composition ratio between flue gas and moist air are small [16]; therefore, the use of moist air as a test gas instead of flue gas is adequate.

Heat transfer with condensation for in-tube air flow including moisture has been widely studied. Hasanein *et al.* [17] conducted an experimental study on condensation heat transfer in vertical tubes with an inner diameter d_i of 46 mm using helium and a mixture of helium and air as non-condensing gases. They proposed a correlation of heat transfer under Reynolds numbers in the range of 850–25000. Jia *et al.* [18] performed experimental and numerical investigations on the heat transfer characteristics in vertical tubes (d_i = 6 and 8 mm) with a combustion gas, and reported good agreement between the experimental results and the analysis model with Re = 2300–5000. Terekhov *et al.* [19] proposed a correlation of the heat transfer in moist air obtained by numerical simulation assuming film condensation, which showed good agreement with the experimental results for d_i = 8 mm and Re = 100000–200000. A numerical study on the influence of parallel plate inclination on the heat transfer of flow with condensation under Re = 1000–2000 was conducted by Siow *et al.* [20] using a two-phase model. Ma *et al.* [21] investigated the heat transfer performance of a steam-air mixture with condensation by using a heat pipe heat exchanger. They showed that the heat transfer was improved 1.6 times under the condition of 20 % of steam and Re = 1500.

Most studies on heat transfer from moist air have been intended for industrial boilers with tube diameters in the range of 10–40 mm and few experiments have been conducted with d_i < 5 mm. However, this study is intended for the 2nd heat exchanger of a household gas water heater. To determine the heat exchanger performance, it is necessary to investigate the heat transfer characteristics of moist air, which imitates flue gas within a single tube with an internal diameter of 5 mm or less.

In our previous study [22], the heat transfer characteristics of moist air rather than flue gas within a single tube were experimentally investigated for tubes with d_i = 1.0–5.0 mm and effective heat transfer lengths of L= 7–100 mm. We reported that the heat transfer in a relatively short length from the inlet was dominant and the change in heat flux decreased steeply with tube length. In addition, a heat exchanger with thinner tubes was effective for compactness. In this study, the heat transfer in a shell and tube type heat exchanger with moist air flow inside the tubes and cooling water in the shell-side is investigated for further improvement of heat exchanger performance. The heat transfer characteristics of single tubes are clarified using improved test sections. Furthermore, non-dimensional correlations of the heat transfer characteristics of moist air are proposed on the basis of experimental results. The performance of heat exchangers with thin tubes is examined using the proposed correlations and the advantages in the compactness of a heat exchanger with thin tubes is discussed.

2. Experimental method

2.1 Experimental system and test apparatus

Figure 3 shows the photograph and schematic of experimental system, which as previously reported [22], consists of a flow meter, a humidifier, a heating apparatus, and a test section. The air used in this experiment is supplied from a blower. The pressure loss in a bent pipe is measured to calculate the gas flow rate. This flow rate measurement is compared with the value presented by using Ito's equation that obtained the pipe friction factor in the bent pipe [23]. The measured relation between pressure loss and flow rate was in good agreement with Ito's equation. As comparison results, this measurement method was confirmed to have sufficient precision. In the humidifier, air flows to a vapor generator and is mixed with steam. This moist air then flows and exchanges heat in the constant temperature bath, where excess humidity is then condensed and discharged. The specific humidity can be determined exactly by measuring the dew point temperature. The moist air flowing in a copper pipe is heated from the outside with a high-temperature blower to obtain the target gas temperature. Figure 4 shows the details of the test section, which consists of gas inlet and heat exchanger parts, and the measurement apparatus. A thermal insulator is inserted between both gas inlet and heat exchanger parts to prevent heat transfer. The temperatures of the gas and water sides of the test section are measured using sheathed thermocouples (0.5 mm diameter) at the inlet and outlet of both fluids, as shown by the red points in Figure 4. Both ends of the test section, which are non-cooling regions for fixing of the tube, are designed to be as short as possible to minimize the measurement error factor of the effective tube length. The length without cooling was shortened from 12 to 5 mm in this experiment. In measurement apparatus, the moist air after exiting the tube is mixed by bifurcation and merge, and the measurement position is covered by the passage of gas flow to achieve an adiabatic condition. Figure 5 shows the details of the apparatus used for measurement of the bulk temperature at the outlet of the test tube. The red allow lines and closed circle denote the gas flow directions and location of temperature measurement in Figs. 5 (a) and (b). Two sizes of measurement apparatus for d_{i} = 1.0 and 2.0 mm, and for d_{i} = 3.0 and 5.0 mm were used because the flow rate of moist air changes with tube diameter. Moist air that has just passed the outlet of test tube is bifurcated at cross-section B and then the air is converged again at cross-section C after passing the right and left channels. The gas is spread peripherally to the entire cross-section D and is discharged at cross-section A. Thus, the thermal effect from the surroundings is minimized and the thermal insulation of the measurement position is improved. The gas temperature is measured with T-type thermocouples (0.1 mm diameter) at the position indicated by the red points in Figure 5. The measurement point at the center of cross-section C is adopted as the mean bulk temperature, T_{B}. The temperature of the tube center is also measured for reference, although the difference was small. Holes of 0.5 mm in diameter are cut at the gas-side of the inlet and outlet to measure the pressure loss using a micro-differential pressure sensor. A condensate outlet is prepared in cross-section A, B, and D, and the condensation rate is determined using the mass measurement of discharged condensate. The temperature of the cooling water flowing in the test section is kept constant using a constant temperature bath. The average velocity of the coolant was the same for all tubes of different diameter.

2.2 Experimental conditions

Table 1 shows the experimental conditions used for the proposed 2nd heat exchanger and that for a typical product. The experimental conditions used in this study are based on the typical conditions of use for a secondary heat exchanger installed in a typical household gas water heater. The typical 2nd heat exchanger and desired performance are referred to as PH here and are considered as standard values for heat exchanger performance. The inlet temperature and specific humidity of moist air for the present study are configured to be the same as those for PH; an inlet temperature of 180 °C and specific humidity of 0.11 kg/kg (dew point = 56.5 °C). Four heat transfer tubes with four different inner diameters (1, 2, 3, and 5 mm) and made from SUS304 were used. The flow velocity of the flow gas was the same for all four tube diameters at 2.5 m/s. The mass flow rate of moist air corresponding to each tube inner diameter was 1.5-37.5×10^{-6} kg/s. The coolant temperature was kept at 20 °C and at a velocity of 1.2 m/s on the basis of the maximum water flow rate of PH. The cooling length L_c in the present experiment is the length from the inlet to the outlet of coolant flow and five different cooling lengths in the test section of 7, 20, 30, 50, and 100 mm were investigated. The density and specific heat of moist air were obtained from the vapor pressure and the molar fraction of vapor. The thermal conductivity and viscosity were calculated using the Lindsay-Bromley [24] and Wilke equations [25], respectively.

2.3 Measurement of mean bulk temperature and determination of the effective tube length

Measurement of mean bulk temperature T_B is important to estimate the heat transfer characteristics of moist air when there are temperature and velocity distributions in the tube flow. The measurement method of T_B is investigated to compare the measurement results with theoretical values using dry air condition [22]. That is, since there are the short non-cooling regions of test tubes near the inlet and outlet to fix in the apparatus, the tube length with heat transfer effectively is different from actual length. Therefore, it was necessary to determine the effective tube length, which was decided by comparing the measurement and the theoretical value obtained from numerical calculation using dry air. The mean bulk temperature at the tube outlet was determined to be the same as that measured at the convergence point (cross-section C in Figure 5); thus, the effective tube lengths for heat exchange in this experimental system were determined. The non-cooling region of test section near the inlet and outlet was shortened in this study; therefore, it is necessary to again determine the effective tube length. The effective tube length was determined by comparing the measurement value and the theoretical value obtained from numerical calculation using dry air, as for the previous study [18]. Figure 6 shows the calculation model with a two-dimensional cylindrical coordinate system for derivation of the theoretical value. The air (inlet conditions: U_{gi} = 2.5 m/s, T_{gi} = 180 °C）flows along the z-axis and the outside wall is cooled by cool water at constant temperature (T_{w} = 20 °C). The temperature distribution of dry air is calculated for laminar forced convection because R*e* for the dry inlet air in this experiment was approximately 72–360 for d_{i} =1.0-5.0 mm, respectively. The temperature distribution of r-direction in the range from 0 to 110 mm was obtained by simulation and the theoretical mean bulk temperature was calculated using Eqs. (1) and (2):

$$T_B(z) = \frac{2}{UR^2}\int_{-R}^{R} T(r)u(r)rdr \qquad (-1/2d_i \le r \le 1/2d_i) \tag{1}$$

$$u(r) = 2U\left(1 - \frac{r^2}{R^2}\right) \tag{2}$$

Figure 7 shows a comparison between the experimental and calculated results for the mean bulk temperature of dry air at the outlet. The range bars indicate the ranges of calculated T_{B} between the tube length cooled under total tube length and the length that was directly cooled by coolant. The range bars are omitted where there was no significant change in temperature against the tube length of $L \geq 23$ mm, d_{i} = 1.0 mm and $L \geq 53$ mm, d_{i} = 2.0 mm. The differences between the measured and theoretical T_{B} for these tube lengths are comparatively small; 90% of all measured values are within ±3 °C and all of the measured values are within ±5 °C. From these results, the mean bulk temperature at the tube outlet is approximated by the temperature measured at the convergence point and the effective tube lengths for heat exchange in this experimental system were determined to be L=10, 23, 33, 53, 103 mm for direct cooling lengths of 7, 20, 30, 50, 100 mm, respectively.

3. Results and discussion

3.1 Effects of effective tube length on mean bulk temperature and condensation rate

Figure 8 shows the relation between the effective tube length L and the mean bulk temperature T_B for each inner diameter d_i (1.0–5.0 mm) when the moist air velocity is 2.5 m/s. The data points and dashed horizontal line show T_B and the coolant temperature, respectively. Although the inlet gas velocity was the same for all d_i, the moist air temperature at outlet T_{go} varied significantly for different d_i. T_B varied significantly for smaller d_i and shorter L, which confirms that heat transfer is improved when using thinner tubes.

Figure 9 shows the measured condensation rates M_d for various effective tube lengths and inner diameters of the tubes. Figure 10 shows the ratio of condensation rate to the theoretical maximum condensation rate in which moist air (x = 0.11 kg/kg) is cooled at 20 °C to the measured condensation rate. The variation of M_d against L decreases with the tube diameter in the measured range of tube length. The variation of M_d at larger L is small for small d_i; therefore, both the major part of the sensible heat of the gas and the latent heat of condensation is transferred for short length tubes. Condensation can be confirmed even if T_B is higher than the dew point (56.5 °C), even for longer tubes of $10 \leq L \leq 33$ with d_i = 3.0 mm and for all L with d_i = 5.0 mm. The temperature of the tube wall can approximate the coolant temperature because the heat transfer coefficient of the coolant is larger than that of inside tube; therefore, condensation occurs inside the tube in the vicinity of the wall even under higher T_B. It is difficult to determine exactly the humidity of the outlet air from T_B, because the measured values of T_B are effective only for the estimation of the sensible heat of moist air in the temperature range between that of the inlet and outlet (T_B). Thus, it is necessary to use the measured values of collected condensate for estimation of the latent heat of condensation.

3.2 Heat transfer characteristics of moist air inside tubes

The heat transfer characteristics are investigated here on the basis of the measured results presented in Sec. 3.1. The heat transfer rate of the gas is calculated from the enthalpy difference between the inlet and outlet with moist air. Equation (3) is used to estimate the specific enthalpy of moist air. The specific heats of air and water and latent heat of vaporization are given here. The temperatures measured at the inlet and outlet are used for T_g in Eq. (3). The specific humidity x (kg/kg) is the mass ratio of steam and dry air, and specific humidity at the inlet x_i is 0.11 kg/kg for all tube diameters and effective tube lengths. The specific humidity at the outlet x_o is calculated from Eq. (4) using the measured condensation rate.

$$i = 1.005T_g + (2501 + 1.846T_g)x \tag{3}$$

$$x_o = \frac{M_{vi} - M_d}{M_{gi} - M_{vi}} \qquad M_{vi} = 0.11M_{gi} \tag{4}$$

Although Eq. (3) was originally used in the range of 0 to 99 °C, Niitsu et al. [26] investigated the precision of Eq. (3) in the range of 0 to 200 °C at atmospheric pressure and determined the error was within approximately 1% on average. Therefore, it is possible that Eq. (3) can be adopted in this study. Figure 11 shows the relation between L and specific enthalpy difference Δi. The enthalpies of moist air transported under these conditions are almost completed when d_i is 1.0, 2.0 and 3.0 mm. The variations of Δi for d_i =1.0 and 2.0 mm are small against L in the range of measured tube length, as with those in Figures 8 and 10. In particular, Δi for d_i =1.0 mm against the entire measured range of L is close to the maximum, i.e., 93% of the maximum is attained, even with L=10 mm. When d_i is 2.0 mm, approximately 93 and 72% of the heat is obtained in the case of L = 33 and 10 mm, respectively. Although the heat transfer rate decreases correspond to the decrease in L with larger d_i, the decreasing rate of heat transfer is smaller than that of L.

3.3 Heat transfer correlation of moist air flowing inside tubes

In Sec. 3.2, the heat transfer rate was shown to decrease steeply in the smaller region of L and then gradually decrease with increasing L because the heat transfer coefficient is large in the thermal entrance region where a thermal boundary layer is developed by the in-tube flow. Therefore, it is worthwhile to examine the effect of the thermal entrance region length on the heat transfer system for moist air. Since Re is 78–389 for d_i = 1.0–5.0 mm according to the inlet gas conditions, the flow regime could be approximated as laminar flow even with condensation. The relationship between the bulk mean temperature and the thermal entrance region length L_T for laminar flow is expressed by Eq. (5):

$$L_T / d = 0.05 RePr \qquad Re < 2300 \tag{5}$$

Figure 12 shows the relationship between the non-dimensional length L/L_T, which is the ratio of effective tube length L and thermal entrance region L_T, and the non-dimensional temperature difference $(T_B\text{-}T_C)/(T_{gi}\text{-}T_C)$, which is the ratio of the differences between the bulk temperature of moist air and the coolant temperature at the inlet to that at the outlet. Figure 12(b) shows an enlarged view of Figure 12(a) to distinguish the major region of $0 < L/L_T < 1$. Figure 13 shows the relationship between L/L_T and the non-dimensional specific enthalpy difference $\Delta i/\Delta i_{max}$, which is the ratio of the local specific enthalpy difference Δi, to the maximum specific enthalpy difference between the inlet and outlet, Δi_{max}. All measured data reveal very close trends in the variations of the non-dimensional temperature difference and specific enthalpy against the non-dimensional length that are both qualitatively and quantitatively independent of tube diameter. Approximately 63% of the non-dimensional temperature dropped and 73% of the enthalpy transferred within the entrance region at L/L_T <1. Thus, it was confirmed that the major energy transfer of moist air occurs in the tube length of the entrance region, which confirms that it is effective to use thin tubes such as d_i = 1.0 and 2.0 mm with shorter thermal entrance regions to achieve a compact heat exchanger. It is possible to express these trends among the non-dimensional expressions using single curves; the solid lines in Figures 12 and 13 show that Eqs. (6) and (7) correlate well with the experimental data obtained in this study.

$$\frac{T_B - T_c}{T_{gi} - T_c} = a\left(\frac{L}{L_T} + a^{\frac{1}{n}}\right)^{-n} \qquad a = 1.5, n = 1.7 \quad \left(\frac{L}{L_T} \geq 0, \quad 0 < \frac{T_B - T_c}{T_{gi} - T_c} \leq 1\right) \tag{6}$$

$$\frac{\Delta i}{\Delta i_{\max}} = 1 - b\left(\frac{L}{L_T} + b^{\frac{1}{m}}\right)^{-m} \qquad b = 0.7, m = 1.6 \quad \left(\frac{L}{L_T} \geq 0, \quad 0 < \frac{\Delta i}{\Delta i_{\max}} \leq 1\right) \tag{7}$$

Figure 14 shows a comparison between the measured data shown in Fig. 8 and that obtained using Eq. (6), while Figure 15 shows that between the data shown in Figure 11 and that obtained from Eq. (7). Equations (6) and (7) are confirmed to correlate well with the experimental data of temperature variations and enthalpy changes against the effective tube length for all tube diameters of d_i = 1-5 mm, as shown in Figure 14 and 15, respectively.

Although the relations between $(T_B\text{-}T_C)/(T_{gi}\text{-}T_C)$ and L/L_T and between and $\Delta i/\Delta i_{max}$ and L/L_T could be expressed by single relations in the Graetz solution for single phase heat transfer in tube flow, there is no theoretical background to have a similar trend to the Graetz solution for this moist air system with condensation. However, it is probable that the heat transfer rates, both from sensible heat and latent heat of condensation from moist air, have similar trends against tube length normalized according to the thermal entrance region, even with different tube diameters.

3.5 Discussion of the proposed heat exchanger performance

Figure 16 shows a dimensions of heat exchanger that is similar to a standard structure with tube bank arrangement that was used for comparison of the heat exchanger volume among cases under different tube diameters. The homothetic ratios between tube diameter and tube gap for d_i = 1-5 mm were equal to that of the PH standard. The effective tube length and the difference of specific enthalpy were calculated from Eqs. (6) and (7). The heat transfer rate is calculated by the difference in specific enthalpies at the inlet temperature T_{gi} = 180 °C and outlet temperature T_B=65 °C, which is equal to that of PH, as shown by dashed line in Figure 14. The flow rates of moist air for each tube were equivalent. Thus, the volume of heat exchangers with the same total heat transfer rate as PH were investigated for each diameter tube, with the assumption of following items:

1. Total heat transfer rate of the proposed heat exchanger is the same as that of PH.
2. The ratio of gas and water flow rates per area are the same as PH.
3. The inlet gas velocity is constant at 2.5 m/s.
4. The mean bulk temperature of the outlet gas is the same as that of PH (65 °C).
5. The cross sections for different tube diameters are similar to that of PH.

The volume ratios for heat exchange between the heat exchangers in this study and the conventional type (PH) are given in Table 2 using Eqs. (6) and (7). The volume ratio for d_i = 1.0 and 2.0 mm are calculated as 4.3 and 17%, respectively, which indicates the possibility to realize a compact heat exchanger using thin tubes.

Figure 17 shows the relation between L and the pressure loss ΔP in moist air. The measurement results in pressure loss is coincided with previous study [22]. Although ΔP increases with decreasing d_i, it is possible to use shorter L due to improvement of the heat exchange performance for smaller d_i. In the present gas-water heater system, the maximum acceptable pressure loss in the secondary heat exchanger is approximately 200 Pa. Since L required for heat exchange can be reduced for a thin tube, it is possible to realize the pressure loss within limits. Moreover, the temporal variation of pressure loss by the occurrence and rejection of condensate is observed in small diameter tubes. This variation becomes a maximum at L = 103 mm with d_i=1.0 mm. However, the difference in pressure loss between the temporal variation and the average was approximately ±23% at the maximum. In the cases of d_i=2.0, 3.0 and 5.0 mm, the variations of pressure loss were approximately ±13, ±8, and ±3% at the maximum, respectively. It is possible to realize the pressure loss within 200 Pa for all tube diameters with the lengths shown in Table 2. Furthermore, the flow rate for each tube considering the tube bank in the heat exchanger is not considered to be significantly uneven, because the moist air temperature at the inlet is sufficiently higher than the dew point so that the gas is single phase at the inlet.

4. Conclusions

A new shell and tube type heat exchanger, in which flue gas flows inside tubes and water is on the shell-side, was proposed to develop improved performance and compactness for latent heat recovery from flue gas using thin tubes. The heat transfer characteristics of the flue gas with condensation in single tubes (d_i = 1.0-5.0 mm) were studied with effective tube lengths of 10-103 mm and the results are summarized as follows:

(1) The heat transfer coefficient of moist air was enhanced for thinner tubes, and steep temperature

drops occur over shorter lengths from the inlet.

(2) The majority of energy from moist air is transferred in the tube length of the thermal entrance region for all tube diameters, and all measured data are located very close to each other, which indicates the single relations of non-dimensional temperature difference and specific enthalpy against non-dimensional length independent of the tube diameter. Thus, the equations below correlate well with the experimental data for the temperature variations and enthalpy changes against effective tube length for all tube diameters of d_{i} = 1-5 mm.

$$\frac{T_B - T_c}{T_{gi} - T_c} = a\left(\frac{L}{L_T} + a^{\frac{1}{n}}\right)^{-n} \qquad a = 1.5, n = 1.7 \quad \left(\frac{L}{L_T} \geq 0, \qquad 0 < \frac{T_B - T_c}{T_{gi} - T_c} \leq 1\right)$$

$$\frac{\Delta i}{\Delta i_{\max}} = 1 - b\left(\frac{L}{L_T} + b^{\frac{1}{m}}\right)^{-m} \qquad b = 0.7, m = 1.6 \quad \left(\frac{L}{L_T} \geq 0, \qquad 0 < \frac{\Delta i}{\Delta i_{\max}} \leq 1\right)$$

(3) The volume ratios for heat exchange between the heat exchangers in this study and the conventional type (PH) are shown using Eqs. (6) and (7). The volume ratio for d_{i} = 1.0 and 2.0 mm were calculated as 4.3 and 17%, respectively. Therefore, it was confirmed that a compact heat exchange can be realized using thin tubes.

Acknowledgements

The authors would like to express their heartfelt gratitude to Mr. Masakazu Kobayashi and Yasuhiko Sano for their cooperation in the research, and to Paloma Corporation for partial funding.

References

[1] Osakabe M., Yagi K., Itoh T., Ohmasa K., Condensation heat transfer on tubes in actual flue gas (Parametric study for condensation behavior), Proceedings of the 12th International Heat Transfer Conference 32 (2003) 153-166

[2] Osakabe M., Horiki S., Hanaki Y., Prediction and performance of compact heat exchanger with small diameter tubes for latent heat recovery, Transactions of the JSME (B) 73 (2007) 253–259 (in Japanese)

[3] Jeong D. W., Lee S. Y., Lee H. J., Performance prediction of heat exchangers for waste heat recovery from humid flue gases, Proceedings of 4th JSME-KSME Thermal Engineering

Conference (2000) 673-678
[4] Fujiwara Y., Fujiyama J., Asano H., Kimura K., A study on compactification of latent heat recovery heat exchanger for gas water heater, Thermal Engineering Conference 2007 7 (2007) 205–206 (in Japanese)
[5] Jeong K., Kessen M. J., Bilirgen H., Levy E. K., Analytical modeling of water condensation in condensing heat exchanger, International Journal of Heat and Mass Transfer 53 (2010) 2361-2368.
[6] Parry, J.A., Eames, C.P., Agyenim, B. A., Modeling of Thermal Energy Storage Shell-and-Tube Heat Exchanger, Heat Transfer Engineering, 35(1) (2014) 1-14
[7] Shi X., Che D., Agnew B., Gao J., An investigation of the performance of compact heat exchanger for latent heat recovery from exhaust flue gases, International Journal of Heat and Mass Transfer 54 (2011) 606-615
[8] Kang H. C., Kim M. H., Ha S. C., Experimental method to evaluate the performance of finned-tube heat exchangers under dehumidifying conditions by using the scaled model, Proceedings of 4th JSME-KSME Thermal Engineering Conference (2000) 769-774.
[9] Joardar A., Jacobi A. M., Heat transfer enhancement by winglet-type vortex generator arrays in compact plain-fin-and-tube heat exchangers. International Journal of refrigeration 31 (2008) 87-97
[10] Kanzaka M., Soda M., Yokoo K., Iwabuchi M., Osada I., Heat and mass transfer on the spirally finned tube, Transactions of the JSME (B) 58(545) (1992) 248–253(in Japanese)
[11] Hwang K., Song C. H., Saito K., Kawai S., Experimental study on titanium heat exchanger used in a gas fired water heater for latent heat recovery, Applied Thermal Engineering 30 (2010) 2730-2737
[12] Kolev D., Kolev N., Performance characteristics of a new type of lamellar heat exchanger for the utilization of flue gas heat, Applied Thermal Engineering 22 (2002) 1919-1930
[13] Kawaguchi K., Okui K., Shimoura T., Ohkouchi K., Osakabe H., Ninagawa T., The heat transfer and pressure loss characteristics of the heat exchanger for recovering latent heat (Effects of straight fin length and fin pitch on the heat transfer and pressure loss characteristics). Transactions of the JSME (B), 72(713) (2006) 104–111(in Japanese)
[14] Cheng, L., Geld, C. W. M. V. D., Experimental Study of Heat transfer and Pressure Drop Characteristics of Air/Water and Air-Steam/Water Heat Exchanger in a Polymer Compact Heat Exchanger, Heat Transfer Engineering 26(2) (2005) 18-27
[15] Park, Y.G., Jacobi, M. A., A simple Air-Side Data Analysis Method for Partially Wet Flat-Tube Heat Exchanger, Heat Transfer Engineering 32(2) (2011) 133-140
[16] The Japan Gas Association (http://www.gas.or.jp/books/notice2010050701.)
[17] Hasanein H. A., Kazimi M. S., Golay M. W., Forced convection in-tube steam condensation in

the presence of noncondensable gases, International Journal of Heat and Mass Transfer 39 (1996) 2625-2639

[18] Jia L., Peng X. F., Yan Y., Sun J. D., Li X. P., Effects of vapor condensation on the convection heat transfer of wet flue gas in a vertical tuber International Journal of Heat and Mass Transfer 44 (2001) 4257-4265

[19] Terekhov V. I., Terekhov V. V., Sharov K. A., Heat and mass transfer in condensation of water vapor from moist air. Journal of Engineering of Physics and Thermophysics 71(5) (1998) 771-777

[20] Siow E. C., Ormiston S. J., Soliman H. M., Two-phase modeling of laminar film condensation from vapor-gas mixtures in declining parallel-plate channels. International Journal of Thermal Sciences 46 (2007) 458-466

[21] Ma, X., Lan, Z., Xu, W., Zhang, C., Fang, Z., Bai, T., Convection condensation heat Transfer of Steam-air Mixture with Heat Pipe Heat Exchanger, Heat Transfer engineering 35 (6-8) (2013) 600-609

[22] Yamashita J., Utaka Y., Kobayashi M., Sano Y., High Performance Heat Exchanger for Latent Heat Recovery from Fuel Gas, Transactions of the JSME (B) 79(803) (2013) 1363–1374

[23] Ito, H., Laminar flow in curved pipes, Journal of Applied Mathematics and Mechanics 49 (1969) 653-663

[24] Lindsay A. L., Bromley L. A., Thermal Conductivity of Gas Mixtures, Industrial and Engineering Chemistry 42(8) (1950) 1508-1511

[25] Wilke C. R., A Viscosity Equation for Gas Mixtures. The Journal of Chemical Physics 18 (1950) 517-519

[26] Niitsu Y., Kurahashi M., Misumi M., New Psychrometric Charts for the Use of Drying, Humidification, and Air Conditioning. Part 1: Constitution of Charts, Journal of Textile Machinery Society of Japan 9(1) (1956) 9-19 (in Japanese)

[27] Japan Society of Mechanical Engineers, JSME Mechanical Engineers’ Handbook Fundamentals α4: Fluids Engineering (2006) (in Japanese)