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Thermal Performance of a Finned-tube Heat Exchanger used in Condensing Gas Boiler

Hie-Chan Kang†, Bok-Bin Lim*, Jung-Man Lee*, Moo-Han Kim**

School of Mechanical & Automotive Eng., Kunsan National University, Gunsan 573-701, Korea

**Department of Mech. Eng., Graduate School of Kunsan National University*

***Department of Mech. Eng., POSTECH, Pohang 790-784, Korea*

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Abstract

In the present study, an experiment was conducted to investigate the heat and mass transfer performance of heat exchangers used in the condensing gas boiler. Two types of spiral circular fin-tube heat exchangers and a plain tube were tested in the flue gas of propane and dry air. Heat and mass transfer coefficients were measured and compared with the previous correlations. The experimental data for the sensible heat transfer of the plain tube reasonably agreed with the previous correlations for dry air and flue gas. However, the mass transfer coefficient of the plain tube was greater than the previous correlations. The pH, NO_x, and SO_x data of condensate were provided.

Key words: Boiler, Condensing, Fin, Heat transfer, Heat exchanger, Mass transfer

Nomenclature

A_b : Tube base area [m^2]
A_f : Fin surface area [m^2]
A_s : Total surface area [m^2]
c_p : Heat capacity [J kg^{-1}K^{-1}]
D : Tube outer diameter [m]
D_{gv} : Mass diffusivity [m^{2}s^{-1}]
D_i : Tube inner diameter [m]
D_o : Fin outer diameter [m]
h : Enthalpy [J kg^{-1}]
h_m : Mass transfer coefficient [kg m^{-2}s^{-1}]
h_s : Sensible heat transfer coefficient [W m^{-2}K^{-1}]
k : Thermal conductivity of fin [W m^{-1}K^{-1}]
L : Tube length [m]
$\dot{m}$: Mass flow rate [kg s^{-1}]
Nu_D : Nusselt number [-]
P_f : Fin pitch [m]
Pr : Prandtl number [-]
Q : Heat transfer rate [W]
Re_D : Reynolds number [-]
Sc : Schmidt number [-]
Sh_D : Sherwood number [-]
t : Fin thickness [m]
T : Temperature [K]
V_{max} : Gas velocity at the minimum cross section [m s^{-1}]
W : Humidity ratio [kg kg^{-1}]

Subscripts

b : Bulk
ex : Exit
g : Flue gas
i : Interface
in : Inlet
l : Latent
LM : Logarithmic mean
v : Vapor
w : Tube wall or water

Greek symbols

μ : Viscosity [kg m^{-1} s^{-1}]
θ : Angle of spiral fin [deg]
ρ : Density [kg m^{-3}]
ω : Ratio of gas mass concentration [-]

†Corresponding author. Tel.: +82 63 469 4722, Fax.: +82 63 469 4727
E-mail address: hckang@kunsan.ac.kr

1. Introduction

The recovery of sensible and latent heat from the exhaust flue gas of a boiler is important to save energy. Recently, clean fuels such as natural gas and propane gas have been widely used in boilers for environmental and biological safety. These gases have high hydrogen content; therefore, the concentration or partial pressure of steam in the exhaust flue gas is relatively high. Commercial gas boilers called as condensing boilers have been developed to increase thermal efficiency by recovering the sensible and latent heat from the flue gas. Therefore, understanding the heat and mass transfer characteristics of flue gas is very important. Further, most gas boilers have adopted fin-tube heat exchangers. The effect of configuration is important because thermal resistance occurs mostly in the flue gas side.

Numerous studies have measured the condensation heat transfer of steam or steam with non-condensable gas on a tube or tube bundle. However, there are not many researches for the latent heat recovery from a boiler. Taniguchi *et al.*[1] performed an experiment and compared its results with the theoretical results. They found that the convective heat and mass transfer were enhanced several tens percents by increasing water vapor concentrations. Kanzaka *et al.*[2] performed a test for the spiral fin-tube heat exchanger by using flue gas. The mass transfer was increased by increasing water vapor contents; however, the heat and mass transfer analogy was not available. Recently, Osakabe *et al*[3] conducted an experimental study for the horizontal tube; it was found that the heat and mass transfer behavior was well predicted in the high wall temperature region. At the high steam concentration, the heat transfer was higher than that predicted by the simple correlation. He suggested a model for the high humid mixture.

In the present study, an experiment was conducted to provide quantitative data for the condensing gas boiler application. Two types of fin-tube heat exchangers and a plain tube were tested for the flue gas of propane and dry air. Heat and mass transfer coefficients were measured and compared with the previous correlation. The pH and the concentration of NO_x and SO_x of the condensate were measured and discussed. Further, the interfacial condition on the heat exchanger was observed.

2. Experiment

2.1 Experimental apparatus

Fig. 1 is a schematic diagram of the experimental apparatus used in the flue gas test. The apparatus consisted of a gas boiler, a secondary heat exchanger, a test section, a constant water bath, and measuring devices. The gas boiler (18.6 kW) burned propane gas (C_3H_8 98.9, C_3H_4 0.79, C_4H_{10} 0.31 vol%) and produced flue gas. The flue gas was cooled to approximately 170℃ by the water-cooled heat exchanger. The humidity ratio W was approximately 0.09-0.1kg_v/kg_g. The flow rate of the flue gas at the main test section was controlled by a bypass valve and a damper. Two screens were located at the inlet of the test section to enhance the flow quality. The hexahedral main test section was vertically installed and the flue gas flowed from top to bottom. The height of the main test section was 200mm and the cross section was 80mm(L) by 80mm, and the gas velocity was changed from 0.8m/s to 3.2m/s. Three types of heat exchangers, a plain tube, and two fin-tubes were tested in the present study. Two test tubes, the 80mm in length, were horizontally placed in the middle of the main test section. The inter-tube distance was 40mm. The heat exchanger was cooled by supplied water from the constant water bath. The coolant temperature was changed from 20℃ to 65℃. The due point temperature was around 55℃. The drain pot was located at the bottom of

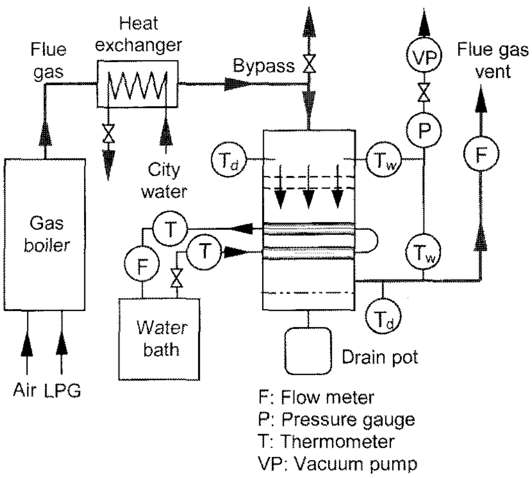

Fig. 1. Schematic diagram of the experimental apparatus used in the flue gas test.

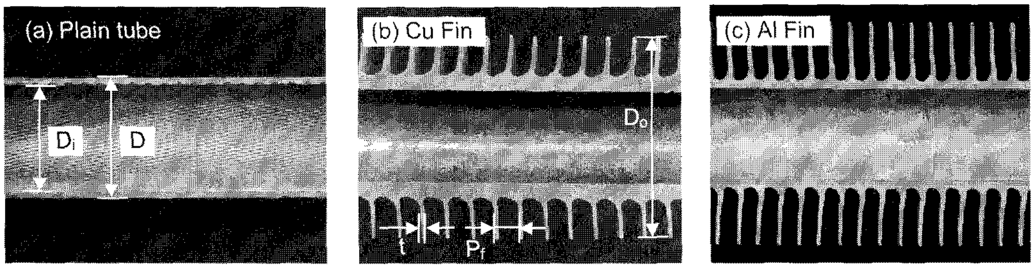

Fig. 2. Heat exchangers tested in the present study.

Table 1. Dimensions of test finned-tube heat exchangers.

Symbol	Unit	Plain	Cu Fin	Al fin
D_i	mm	14.0	12.7	12.3
D	mm	15.9	16.7	14.8
D_o	mm	15.9	28.3	30.0
t	mm	-	0.50	0.57
P_f	fin/m	-	285	357
θ_{min}	deg	-	2.2	1.7
A_b / L	m^2/m	-	0.045	0.037
A_f / L	m^2/m	-	0.220	0.388
A_s / L	m^2/m	0.050	0.265	0.425
F_{fin}	A_s / A_b	1.0	5.1	9.1
Material		Copper	Copper	Aluminum

the test section to collect the condensate from the test tube. A baffle was placed between the drain pot and test tubes to avoid the re-evaporation of condensate. The dry and wet bulb temperatures were measured at the inlet and outlet, respectively. A window was made on a sidewall of the main test section so as to observe the condensate that formed on the heat exchanger. The flow rate of the flue gas was measured by a nozzle flow meter at the outlet of the test section.

2.2 Test heat exchangers

Fig. 2 shows a cross-sectional view of the three heat exchangers used in the present study. The plain tube (Fig. 2(a)) is made of copper, without fins, and has a diameter of 15.9mm. The other heat exchangers are spiral circular-fin tube heat exchangers. The heat exchanger shown in Fig. 2(b) is made of copper and that shown in Fig. 2(c) is made of aluminum. The aluminum fin has more hydrophobic characteristics than the copper fin. Further, the fin pitch of the copper tube (285fin/m) was greater than that of the aluminum tube (357fin/m). The details of dimensions are listed in Table 1.

2.3 Experimental method

The heat transfer rates were measured in the flue gas and coolant water-sides respectively. Two K-type thermocouples were used to measure flue gas temperature at the inlet and outlet of the test heat exchanger. An aspirating device was used to measure the wet bulb temperature. Wet cotton was filled in an insulated tube and a K-type thermocouple was located at the center of the wet cotton. A vacuum pump aspirated the flue gas from the test section and the flue gas passed through the wet cotton. In this process, the measuring device was insulated and the vacuum pressure was controlled at not more than 700Pa from atmospheric pressure. The present method gave less than 2.0% error in the humidity ratio measurement under the ambient condition when compared to the ASHRAE recommended method when the operation or aspirating time was more than 5 min. The wet cotton was replaced after each measurement. A flow nozzle (recommended ISA) having a diameter of 45.0mm was used to monitor the flow rate of the flue gas. The pressure difference between the upstream and downstream of the flow nozzle was measured by a pressure transducer (Furness, FC012). Two RTD thermometers and a magnetic flow meter were used to measure the heat transfer rate in the coolant side. The outputs of temperature, pressure, and flow sensors were treated by a data acquisition system (AOIP, SA32). All thermometers were calibrated to within 0.1℃, especially 0.01℃ for RTDs by using a NIST traceable bulb thermometer.

It took approximately 60min to reach the steady state after the gas boiler was turned on. All data were sampled for 2min and averaged when the steady state was reached. Uncertainties of heat loss through the test section, around 15% to the total heat transfer, could not be avoided because the size of the test heat exchanger used in the flue gas test was small. Further, the re-evaporation of condensate downstream of the heat exchanger could not be completely avoided in the present study. This gave some error in the measurement of sensible and latent heat transfer. The dry air test was conducted to measure the sensible heat transfer only in the other wind tunnel. The properties of the collected condensate were tested by the probe-type pH meter and ion chromatography (Bonex Co.) to measure the NO_x and SO_x values.

The total heat transfer rate can be separated into the sensible and latent heat transfer as given below.

$$\begin{aligned} Q &= Q_s + Q_l \\ &= \dot{m}_g C_{p,g}(T_{in} - T_{ex}) + \dot{m}_g h_v (W_{in} - W_{ex}) \end{aligned} \tag{1}$$

The exit temperature was corrected considering the heat loss at the test wall. The averaged heat capacity and enthalpy of the flue gas were used for each test condition. The ratio of heat capacity ($\dot{m}C_p$) of the flue gas to that of the coolant was around 4%, so the present cross flow can be assumed as the counter flow in the heat transfer coefficient evaluation. The heat and mass transfer coefficients were defined as given below:

$$h = \frac{Q_s}{A_s \Delta T_{LM}} \tag{2}$$

$$h_m = \frac{Q_l}{A_s h_v \rho_g \Delta W_{LM}} \tag{3}$$

$$\Delta T_{LM} = \frac{(T_{in} - T_w) - (T_{ex} - T_w)}{\ln[(T_{in} - T_w)/(T_{ex} - T_w)]} \tag{4}$$

$$\Delta W_{LM} = \frac{(W_{in} - W_w) - (W_{ex} - W_w)}{\ln[(W_{in} - W_w) - (W_{ex} - W_w)]} \tag{5}$$

The wall temperature was calculated by using the relation among the coolant temperature, heat transfer rate, and the thermal resistances of the coolant and the tube wall. It is difficult to find reliable correlations for the temperature of interface, especially the fin-tube geometry. Therefore, the temperature of the condensate interface was assumed to be that of the wall in the present study. The fin efficiency assumed as 100% because the parameter $(D_o - D)\sqrt{h/2kt}$ in the present experiment was near 0.1. The humidity ratio W was estimated by the dry and wet bulb temperatures according to the ASHRAE handbook[4]. However we replaced the heat capacities of flue gas and vapor at high temperature instead of the standard condition of air. The Nusselt, Sherwood, and Reynolds numbers are as follows:

$$\mathrm{Nu}_D = \frac{hD}{k_g} \tag{6}$$

$$\mathrm{Sh}_D = \frac{h_m D}{D_{gv}} \tag{7}$$

$$\mathrm{Re}_D = \frac{\rho_g V_{max} D}{\mu} \tag{8}$$

where the flue gas properties such as density, viscosity, thermal conductivity, and mass diffusivity were obtained considering the gas composition by Reid *et al.*[5] The reference temperature for the flue gas was the film temperature, i.e., the average of inlet and outlet gas temperatures. The maximum gas velocity V_{max} was defined as the representative gas velocity at the minimum cross section.

3. Results and discussions

3.1 Heat transfer

Fig. 3 shows the Nusselt number versus Reynolds number for dry air and flue gas. The heat transfer coefficient is defined for the total surface area and the Nusselt number is defined for tube diameter D. In the case of the dry air test, the void symbol in Fig. 3, the Nusselt number of the

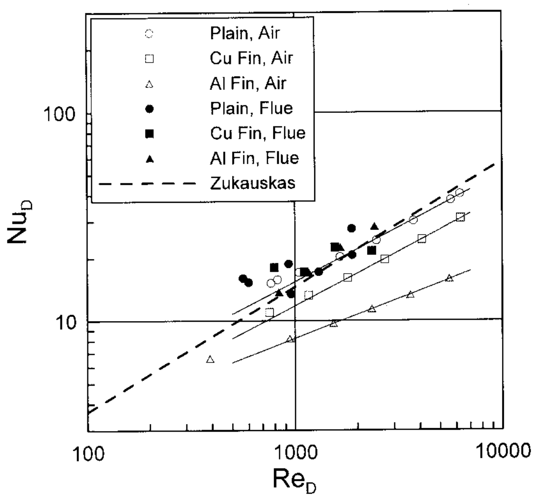

Fig. 3. Nusselt number versus Reynolds number for dry air and flue gas flow.

plain tube is high and that of the aluminum fin is small. This is because of the fin and the close gap between the fins, i.e., the fin pitch of the aluminum fin is the largest. The present data for the dry air test agreed well with the simple correlation of Zukauskas[6] for the sensible heat transfer in the plain tube. The correlation is given below:

$$\mathrm{Nu}_D = 0.26\,\mathrm{Re}_D^{0.6}\,\mathrm{Pr}_b^{0.37}(\mathrm{Pr}_b/\mathrm{Pr}_w) \tag{9}$$

The sensible heat transfer coefficient of flue gas is similar to that of dry air. However, the mass transfer is larger than that in the dry test for the fin-tube heat exchangers. It is believed that the absorption effect due to the condensation is coupled in the heat transfer process, as discussed by Osakabe *et al*[3].

3.2 Mass transfer

Fig. 4 shows the comparison of mass transfer coefficients for three heat exchangers along with the Reynolds number. The Sherwood number increases for increasing the Reynolds number, and the plain tube shows a high value when compared to fin tubes. The differences in the mass transfer performance of heat exchangers are increased when compared to the sensible heat transfer, as shown in Fig. 3. The correlations suggested by Zukauskas[6], Osakabe *et al*[3], Mills[7], and Fujii[8] for the mass transfer coefficient of the plain tube are compared and they are given below, respectively:

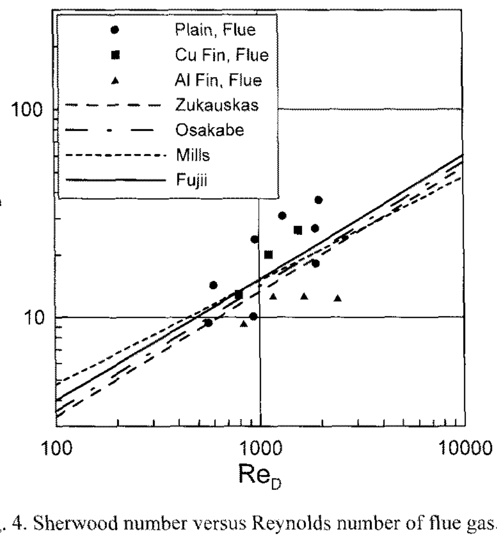

Fig. 4. Sherwood number versus Reynolds number of flue gas.

$$\mathrm{Sh}_D = 0.26\,\mathrm{Re}_D^{0.6}\,\mathrm{Sc}_b^{0.37}(\mathrm{Sc}_b/\mathrm{Sc}_w) \tag{10}$$

$$\mathrm{Sh}_D = 0.26\,\mathrm{Re}_D^{0.6}\,\mathrm{Sc}_b^{0.37}(\mathrm{Sc}_b/\mathrm{Sc}_w)^{0.25}\frac{1}{1-W_i}\left(\frac{1-W_i}{1-W_b}\right)^{0.37} \tag{11}$$

$$\mathrm{Sh}_D = 0.57\,\mathrm{Re}_D^{1/2}\,\mathrm{Sc}_b^{1/3}\left\{\left[\frac{1.57\mathrm{Sc}_b^{0.1}(1-\omega)}{(1+\mathrm{Sc}_b)^{1/2}\omega^{1/2}}\right]^{3/2}+1\right\}^{2/3} \tag{12}$$

$$\mathrm{Sh}_D = \mathrm{Re}^{1/2}\frac{[1+\chi\mathrm{Sc}^{0.4}(\omega^{-1}-1)]^{1/2}-1}{2(1-\omega)} \tag{13}$$

where

$$\chi = 4(0.3+0.1\,\mathrm{Re}^{0.17}) \tag{14}$$

The ω is the ratio of the non-condensable gas concentration between bulk and interface, i. e., W_{gb}/W_{gi}. In Fig. 4, the present data for the plain tube are approximately 40% higher than the correlations. The trend of the present data agrees more with Fujii's correlation, as discussed by Jeong *et al.*[9] The copper fin shows better mass transfer performance than the aluminum fin. The reason is supposed that the blocking rate of the condensate between the fins degrades the mass transfer in the small fin pitch.

Fig. 5 shows the state of condensate formed in the copper fin-tube heat exchanger. Small droplets are formed on the surface; however, the condensate is formed as films on most of the surface above the main tube. Large droplets having fin pitch dimension in their diameter are formed at the bottom of the fins.

3.3 Heat and mass transfer analogy

Fig. 6 shows the comparison of heat and mass transfer coefficients of three heat exchangers for the flue gas. The ratio of mass transfer ($\mathrm{Sh}_D\mathrm{Re}_D^{-0.5}$) to heat transfer ($\mathrm{Nu}_D\mathrm{Re}_D^{-0.5}$) varies from 0.4 to 1.8. Even though there are some experimental errors, we can find that the mass transfer of the heat exchanger affects the fin geometry, especially the fin pitch. Aluminum has a hydrophobic surface and so more condensate will be formed when compared to copper. Fig. 7 shows the mass transfer ($\mathrm{Sh}_D\mathrm{Re}_D^{-0.5}$) along $\omega\,(=W_{gb}/W_{gi})$. A similar trend for the effect of heat exchanger configuration can be found in the results.

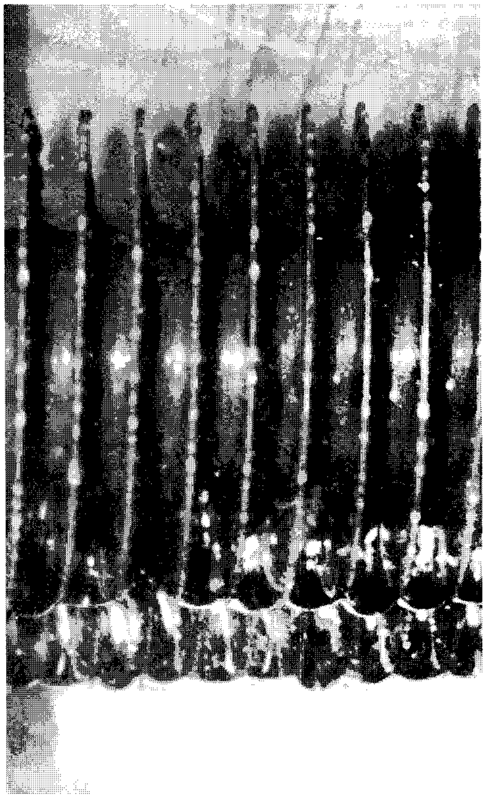

Fig. 5. Photograph of wet surface, copper finned-tube heat exchanger, and flue gas, V_{max} =1.38 m/s, $T_{g,in}$ =170 ℃, $T_{g,ex}$ = 123 ℃, T_w = 42 ℃, W_{in} = 0.095, W_{ex} = 0.077

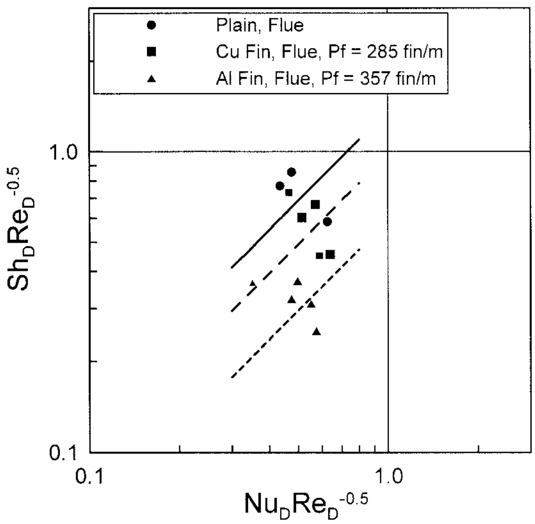

Fig. 6. Comparison of heat and mass transfer coefficients for flue gas.

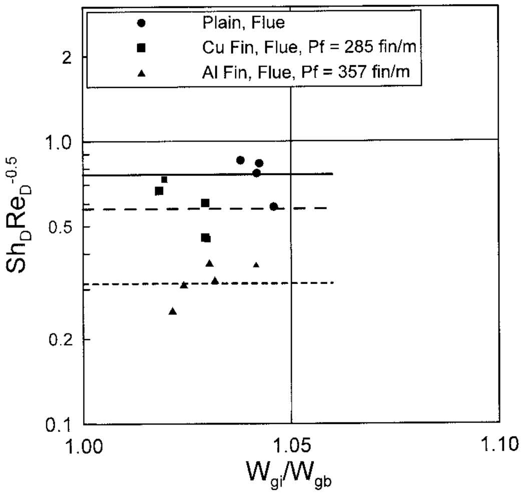

Fig. 7. Relation between $Sh_D Re_D^{-0.5}$ and W_{gb}/W_{gi} for flue gas.

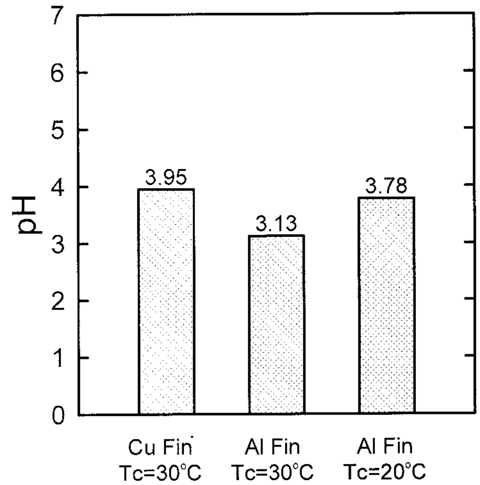

Fig. 8. pH of condensate.

3.4 pH of condensate

Fig. 8 shows the comparison between the pH of condensate for the surface conditions and coolant temperature (or wall temperature). The pH value of the present study ranged from 3 to 4. For the similar wall temperature, the copper fin shows a higher pH value than the aluminum fin. We suppose the reason to be the effect of fin pitch and wetability. For the same heat exchanger, the low wall temperature or the higher concentration potential gives a higher pH value. The condensation rate is increased for increasing the concentration potential of vapor; therefore, the condensate becomes diluted. Fig. 9 compares the concentration of NO_x

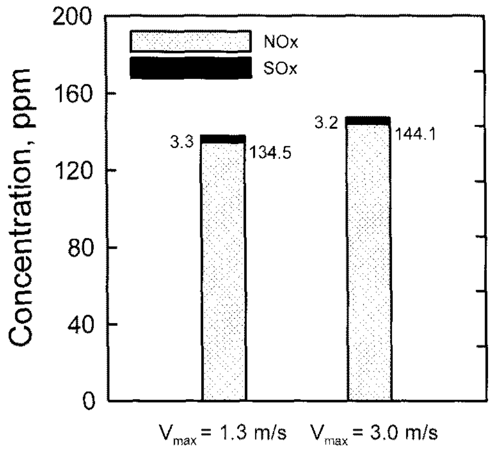

Fig. 9. SO_x and NO_x of condensate.

and SO_x in the condensate for the copper fin heat exchanger for the gas velocity. The NO_x and SO_x concentrations are approximately 140ppm and 3.3ppm, respectively. The concentration of NO_x slightly increases with flue gas velocity.

4. Conclusions

An experiment was conducted to provide quantitative data for the condensing gas boiler application. Two types of fin-tube heat exchangers and a plain tube were tested for the flue gas of propane and dry air. Heat and mass transfer coefficients and the pH, NO_x and SO_x concentration of the condensate were measured. We obtained the following conclusions.

(1) The experimental data for sensible heat transfer of the plain tube reasonably agreed with the previous correlation for the dry air and flue gas.
(2) The mass transfer coefficient of the plain tube showed similar trends; however, the values were large when compared to the previous correlations.
(3) The fin-tube heat exchangers having the extended surface showed lower mass transfer coefficients than the plain tube.
(4) The pH, NO_x, and SO_x data of the condensate was provided. The aluminum fins, having the small fin pitch, showed lower pH than the copper fins, having the large fin pitch.

A more precise experiment to find out the details of heat and mass transfer characteristics is required.

Acknowledgement

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