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CONDENSATION HEAT TRANSFER OF ACTUAL FLUE GAS ON HORIZONTAL TUBES

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ABSTRACT

In order to improve the boiler efficiency, latent heat recovery from an exhaust flue gas is a very important concept. Condensation heat transfer on horizontal stainless steel tubes was investigated experimentally using an actual flue gas from a natural gas boiler. The experiment was conducted at different air ratios and steam mass concentrations of the flue gas, and in a wide range of tube wall temperature. The condensation pattern was similar to the dropwise condensation near the dew point. As the wall temperature was decreased, the wall region covered with a thin liquid film increased. The heat and mass transfer behavior were well predicted with the simple analogy correlation in the high wall temperature region. But in the low wall temperature region, the total heat transfer rate was higher than that predicted by the simple analogy correlation. At a high steam mass concentration artificially generated with steam injection, the total heat transfer rate was higher than that predicted by the simple analogy correlation. The analogy correlation using the modified Sherwood number taking account of the mass absorption effect was proposed. The modified correlation gave a good prediction of the heat flux at the high steam mass concentration.

EXPERIMENTAL APPARATUS AND MAJOR RESULTS

Water cooled test tubes of SUS316L were installed at a transparent polycarbonate test section of which cross-section was 160mm×101mm. Three tubes of 21.7mm diameter were arranged horizontally at pitch of 33.7mm. Sheathed K-type thermocouples of 0.5 mm diameter were imbedded at 15 circumferential locations of tubes to obtain the average wall temperature. The steam and SO_2 concentrations in the flue gas can be controlled with steam and SO_2 injections upstream of the test section. The experiments were conducted at the different wall temperature, flue gas flow rate and properties.

Shown in Fig.A-1 is the relation between the heat flux and the average wall temperature in a flue gas of air ratio 1.29 where the steam mass concentration was approximately 0.1. The solid and dashed lines are predictions by the simple analogy correlation. The condensation heat flux was obtained by the amount of condensed water collected and measured with multiple suction pipes. The total heat flux and the condensation heat flux were well predicted with the simple analogy correlation at the high wall temperature and the low steam mass concentration.

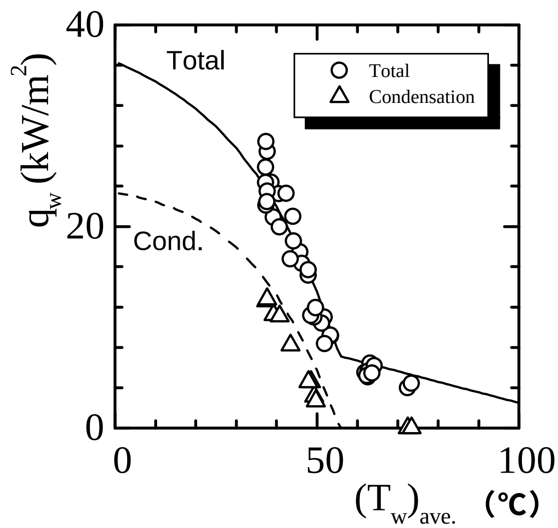

Fig. A-1 Relation between heat flux and average wall temperature

NOMENCLATURE

C: mass concentration per fluid of a unit volume [kg/m^3]
C_p: specific heat [kJ/kg]
d: outer diameter of tube [m]
D: mass diffusivity(m^2/s)
h_V: heat transfer coefficient [kW/(m^{2}K)]
h_C: mass transfer coefficient [m/s]
K_f: modified Nusselt number
L_W: latent heat [kJ/kg]
m_S: injected steam flow rate [kg/s]
Nu: Nusselt number($= h_V d / \lambda$)
P: pressure [Pa]
q: heat flux [kW/m^2]
Pr: Prandtl number($= \nu / \kappa$)
Re: Reynolds number($= ud / \nu$)
R: relative humidity of air
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]
x: flow-directional distance [m]
y: distance from wall [m]
κ: thermal diffusivity ($= \lambda / (\rho C_P)$)
λ: heat conductivity [W/(mK)]
μ: air ratio
ν: kinematic viscosity [m^2/s]
ρ: density [kg/m^3]

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

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 accompanying with the latent heat. So the latent heat recovery from the flue gas is very important to improve the boiler efficiency.

For the latent heat recovery from a boiler, condensation heat transfer experiments have been conducted by using steam-air mixtures(Taniguchi et. al., 1987; Kanzaka et. al., 1992 and Kawamoto et. al., 1995). These experiments yield different opinions on the analogy for heat and mass transfer. Furthermore, condensation in actual flue gas is more complex and least understood. So it was difficult to predict the accurate thermo-fluid behavior of an actual condensing heat exchanger using an actual flue gas. Condensation heat transfer on horizontal stainless steel tubes was investigated experimentally by using the actual flue gas from a natural gas boiler. The experiment was conducted at different air ratios and steam mass concentrations of the flue gas in a wide range of tube wall temperature.

EXPERIMENTAL APPARATUS AND METHOD

Shown in Fig.1 is a schematic of experimental apparatus. The flue gas from a natural gas test boiler was cooled from 270°C to 120~200°C by a heat exchanger without the condensation . The steam and SO_2 concentrations in the flue gas can be controlled with steam and SO_2 injections systems upstream of the test section.

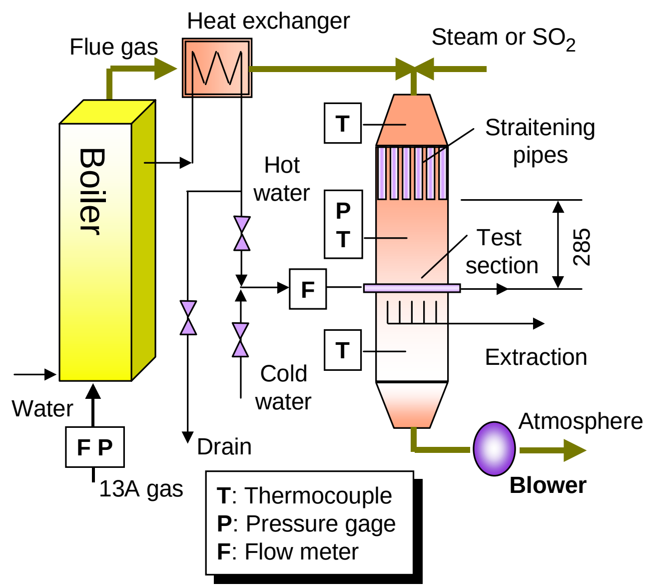

Fig. 1 Schematic of experimental apparatus

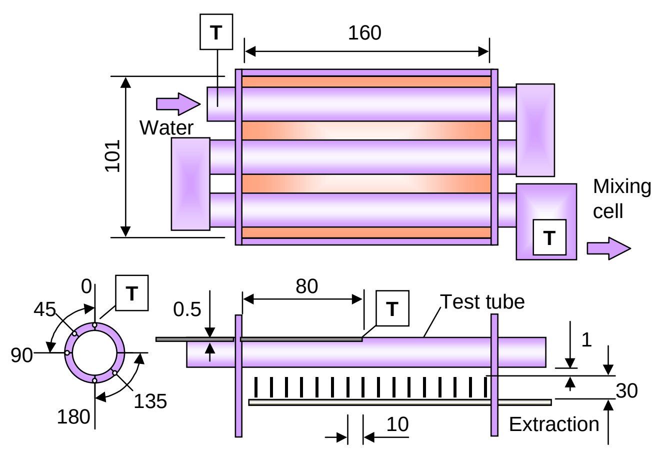

Fig. 2 Test tubes and extraction system

The steam was supplied from another steam boiler and measured with a vortex-shedding flow meter. The error was within ±2% of the measured value. The flow rate and pressure of the natural gas fuel supplied to the test boiler were measured. The error of the flow rate was within ±0.5%. The flue gas temperatures just above and below the test tubes were measured with sheathed K-type thermocouples of 0.5 mm in diameter.

Water cooled test tubes of SUS316L were installed in a transparent polycarbonate duct with the cross-section of 160mm×101mm as shown in Fig.2. The water flow rate was measured with a turbine flow meter within ±2%. The connecting pipes outside the test section were thermally insulated with a glass wool. Three tubes of 21.7mm in diameter were arranged horizontally at a pitch of 33.7mm. The pitch to diameter ratio p/d was 1.55. Sheathed K-type thermocouples of 0.5 mm in diameter were imbedded at circumferential locations of 0, 45, 90, 135 and 180° from the top of 3 tubes to obtain the average wall temperature. The inlet and outlet water temperature of the test tubes were measured also with the sheathed K-type thermocouples of 0.5 mm in diameter. The outlet temperature was measured at a mixing chamber to obtain a well-mixed bulk temperature. The temperature difference of the cooling water through the tubes was kept approximately at 1~4 K. The heat flux of the test tubes was calculated with the flow rate and the temperature difference of cooling water through the tubes. The measurement error of the heat flux was estimated to be 20% as the maximum in the non-condensation region and 5% as the minimum in the condensation region. 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 condensate generated on the test tubes was collected with an extraction system as shown in Fig.2. The multiple stainless-steel tubes of 1 mm inner diameter were attached to a header piping of 6 mm in diameter which led to a vacuum pump. The condensate hanging to the tube bottom was immediately extracted to a measuring pot with the multiple tubes and the amount collected during 5 to 10 minutes period was measured. By using the collected amount and the latent heat of steam in the flue gas, the condensation heat flux was estimated. It was confirmed that the extraction did not affect the condensation heat flux and the steam mass involved in the extracted gas was negligibly small. The pH value of the condensate at 25°C was measured with an electric-conductivity pH sensor.

Table1. Composition of natural gas fuel(13A)

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

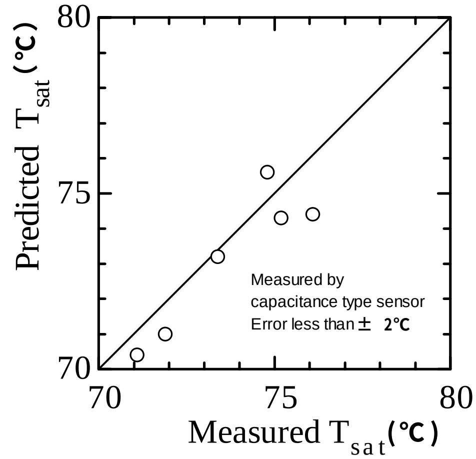

Fig. 3 Predicted and measured dew points

The composition of natural gas fuel used in the test boiler are 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. The measurement error was within ±10 % for CO_2, within ±0.7 % for O_2 and within ±15 % for CO. By using the measured concentration, the air ratio μ is calculated as

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

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

$$V_{COX} = V_F \cdot CCR \tag{2}$$

The volumetric flow rate of dry gas V_d is,

$$V_d = \frac{V_{COX}}{CO_2 + CO} \tag{3}$$

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 $V_dN_2/0.79$, the volumetric fraction of H_2O in the flue gas can be estimated as

$$H_2O = \frac{(CO_2 + CO) \cdot CHR / 2 + N_2 \cdot R \cdot P_{sat} / (P_a \cdot 0.79) + m_S \cdot 22.4 / (18 \cdot V_d)}{1 + (CO_2 + CO) \cdot CHR / 2 + N_2 \cdot R \cdot P_{sat} / (P_a \cdot 0.79) + m_S \cdot 22.4 / (18 \cdot V_d)} \tag{4}$$

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

Shown in Fig.3 is a comparison of the predicted dew points by using Eq.(4) and the measured ones with a capacitance-type humidity sensor. The measurement error of the dew points was

±2°C in the experimental condition in Fig.3. The calculated dew points by Eq.(4) agree well with the measurement.

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

$$V_{wt} = \frac{V_d}{1 - H_2O} \tag{5}$$

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

Table2. Low velocity experimental condition

Experiment No.	L-1	L-2	L-3
Air ratio μ	1.09	1.2	1.33
Flow rate V_{wt} (m_N^3/h)	32.2	33.6	36.3
Gas temp. T_f (°C)	145	167	192
Steam conc. w_f (kg/kg)	0.111	0.108	0.097
Dew point T_{sat}(°C)	57.1	56.4	54.4

Table3. High velocity experimental condition

Exp. No	H-1	H-2	H-3	H-4	H-5	H-6	H-7
Air ratio μ	1.29	1.2	1.22	1.19	1.19	1.22	1.18
V_{wt} (m_N^3/h)	212	234	234	246	249	268	255
T_f (°C)	124	118	118	118	118	116	120
m_S (kg/h)	0	31	34	43	46	51	53
w_f (kg/kg)	0.103	0.216	0.222	0.249	0.262	0.263	0.279
T_{sat} (°C)	55.5	70.4	71.0	73.2	74.3	74.4	75.6

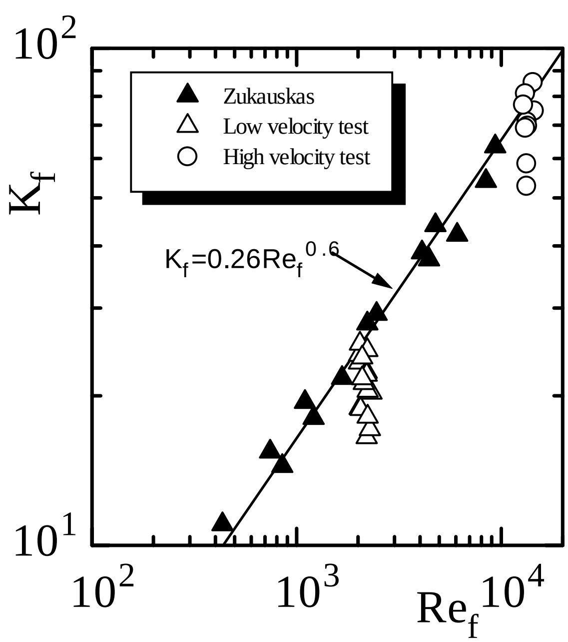

Fig. 4 Modified Nusselt number under non-condensing condition

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

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

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

Two series of experiments with low and high flue gas velocities were conducted. The flue gas velocity at the minimum flow area of the test section was approximately 3.5m/s in the low velocity experiment and approximately 15m/s in the high velocity experiment, respectively. In the low velocity experiment, the boiler was operated at three kinds of air ratios by controlling the rotational speed of the blower with an inverter. In the high velocity experiments, steam concentration of the flue gas was controlled by injecting a steam into the flue gas. The experimental conditions for the low and high velocity experiments are shown in Tables.2 and 3, respectively.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Low velocity experiment

The maximum inner wall temperature of the polycarbonate test section was 100°C. The maximum contribution ratio of the radiation heat flux from the inner wall to the test tubes was within 9% of the total heat flux q_T when the emissivity of the polycarbonate wall and the tubes covered with the condensate were assumed as 1. As an accurate estimation of the radiation heat transfer is difficult, it was neglected in the following analysis. The total heat flux q_T consists of the convection heat flux q_V and the condensation heat flux q_C as

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

In the experiments, q_T was obtained from the temperature difference and flow rate of the cooling water. The convection heat flux is expressed as

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

The condensation heat flux q_C 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 is

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

The modified Nusselt number K_f is

$$K_f = Nu_f \cdot Pr_f^{-m} (Pr_f / Pr_W)^{-0.25} = c\, Re_f^{0.6} \tag{12}$$

Zukauskas(1972) proposed c=0.26, m=0.37 for the first row tubes of an in-line bank with p/d=1.6, which is approximately the same as that in the present study. Shown in Fig. 4 is the experimental data of the modified Nusselt number under non-condensation condition in the low and high velocity experiments along with previous data by Zukauskas(1972). The non-condensation heat transfer data agrees fairly well with the previous data and correlation (12). These results show that the heat flux and the flue gas velocity were measured with efficient accuracy in the present study. 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

$$Sh_f = c\,Re_f^{0.6}\,Sc_f^{m}\,(Sc_f / Sc_w)^{0.25} \tag{13}$$

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 an first attempt, the mass diffusivity of steam in flue gas was estimated with the well-known mass diffusivity of steam in air as

$$D = D_{air}\left(\frac{\kappa}{\kappa_{air}}\right) \tag{14}$$

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),

$$D_{air} = 7.65\times10^{-5}\,\frac{(T+273.15)^{11/6}}{P} \tag{15}$$

Shown in Fig.5 is the relation between the heat flux and average wall temperature at the air ratio of 1.33. The key (O) is the total heat flux and the key (Δ) is the condensation heat flux obtained from the amount of condensate. The solid line shows the total heat flux predicted by the simple analogy correlations Eqs.(11) and (13). The dashed line is the prediction of the condensation heat flux by the Eq.(13). The predicted heat flux begins to increase at the dew point of 54.4°C due to the initiation of condensation. The tube surface was covered with small droplets near the dew point. Condensation initiated at the dew point corresponding to the partial pressure of steam in the flue gas. Decreasing the wall temperature furthermore, the surface area covered with a thin film increased but a partially dry region covered with droplets could be recognized. The experimental data for the condensation heat flux agree well with the predictions. The total heat flux also can be predicted well with the correlation except for the low temperature region where the heat flux is slightly higher than the prediction. The same heat transfer behaviors were observed in the experiments of the different air ratio.

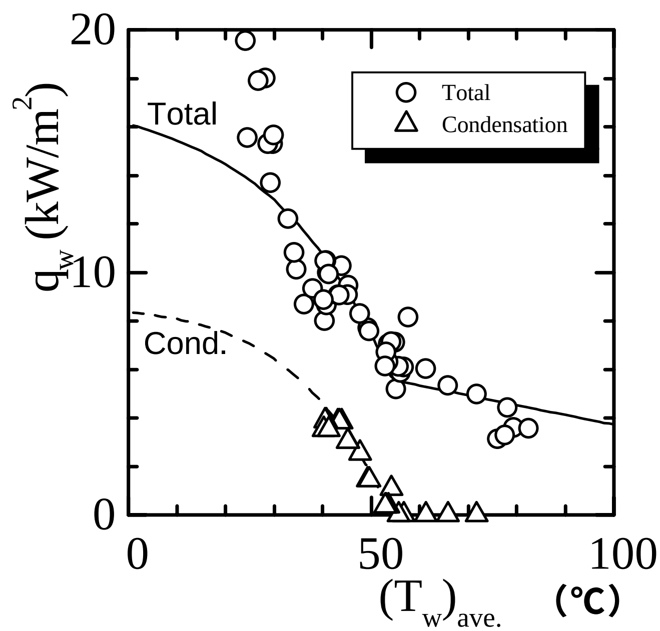

Fig. 5 Relation between heat flux and average wall temperature at air ratio of 1.33

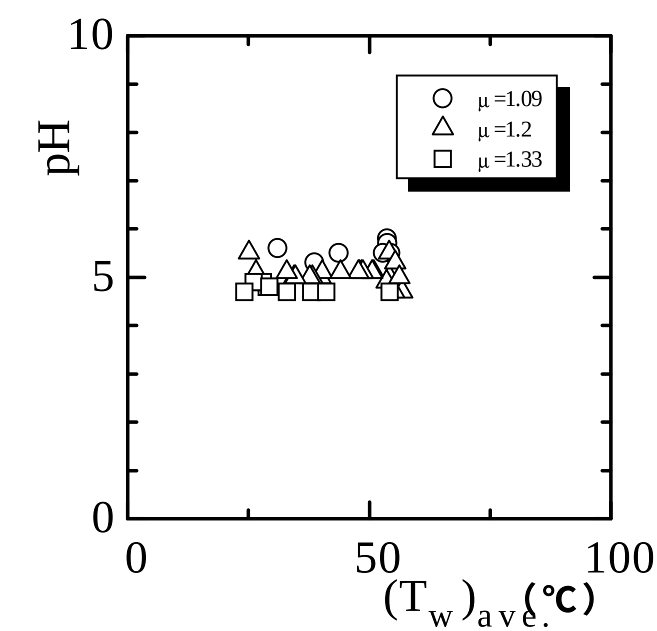

Fig. 6 pH value of condensate

Shown in Fig.6 is relation between the pH value of the condensate and the average wall temperature in the experiments of the different air ratio. The pH value does not depend on the wall temperature and it decreases slightly with the air ratio. It is possible that the concentration of NO_X in the flue gas increases with an increase in the air ratio. The increase of NO_X is expected to decrease the pH value of condensate.

The simple analogy correlation underpredicted the convective and condensation heat transfer in the experiment by Taniguchi et. al.(1987) using a wet air. In their experiment, the wall temperature was between 18.8°C and 30.4°C. The underprediction of the total heat flux is also observed at a low average wall temperature less than approximately 30°C in the present experiment.

High velocity experiment

The maximum inner wall temperature of the polycarbonate test section was 85°C. The maximum contribution ratio of the radiation heat flux from the inner wall to the test tubes was within 5% of the total heat flux q_T when the emissivity of the polycarbonate wall and the tubes covered with the condensate were assumed as 1. Shown in Fig.7 is the relation between the average tube wall temperature and the heat flux at $Re_f \cong 13500$, $\mu \cong 1.29$ and $T_f \cong 124$°C. The experimental results for the total and condensation heat flux agree well with the prediction by the simple analogy correlations Eqs.(11) and (13).

The pH value of the condensate in the high velocity experiment was approximately 4 which was slightly lower than that in the low velocity experiment. It is considered that the concentration of NO_X increases with the increase of flame temperature due to the increase of fuel flow rate in the burner. The fuel consumption rate in the high velocity experiment was approximately 5 times that in the low velocity experiment. The increase of NO_X is expected to decrease the pH value of the condensate in the high velocity experiment. Shown in Table.4 is the ion concentration of the condensate in the high velocity experiments. The condensate was sampled at different subcoolings of the tube from the dew point. The NO_X and SO_X ions are considered to reduce the pH of condensate and promote the corrosion. The high concentration of these ions can be recognized near the dew point.

To study the effect of steam concentration in the flue gas, experiments were conducted with steam injection into the flue gas upstream of the test tubes. The injected steam flow rate m_S was between 31 and 53 kg/h. The steam concentration w_f in the flue gas varied between 0.216 and 0.279kg/kg and the corresponding dew point varied between 70.4 and 75.6°C. Shown in Fig.8 is the enlarged photograph of tube wall at 90° from the tube top . The dew point was 71°C and the tube wall subcooling was 12.3 K. The tube surface was almost completely covered with a thin condensate film partially accompanied by droplets. The partial dry region could be recognized near the droplets even in the steam injection experiment.

The above simple analogy correlations Eqs.(11) and (13) predict well the experimental heat flux at a relatively small concentration of steam in the flue gas. When the steam concentration is large, the effect of mass absorption at the wall should be considered(Rose, 1980 and Shiozaki et. al., 1998).

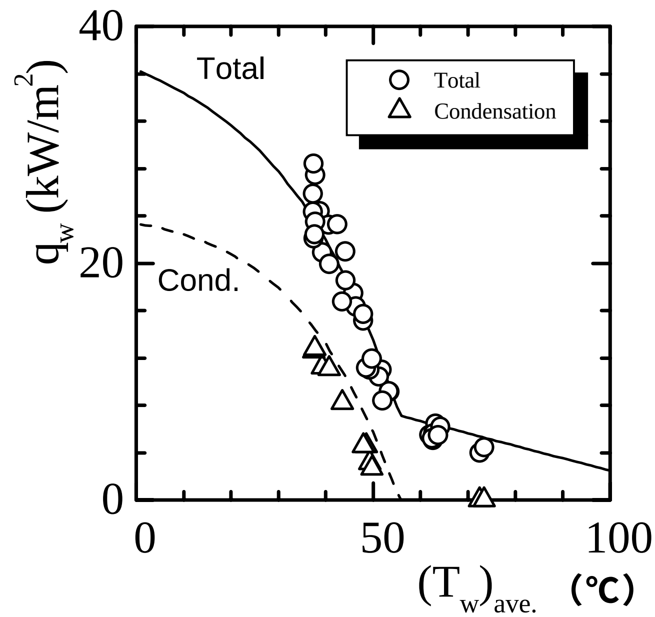

Fig. 7 Relation between heat flux and average wall temperature in high velocity experiment

Table4. Ion concentration in condensate

ΔT_{sub} (K)	~1	~10	~20
Cl^- (mg/l)	1.0	< 0.5	< 0.5
SO_4^{2-} (mg/l)	8.6	3.2	2.3
NO_3^{2-} (mg/l)	24	8.8	7.1
NO_2^{2-} (mg/l)	17	11	11
CO_3^{2-} (mg/l)	< 1	< 1	< 1

Fig. 8 Photograph of tube wall at 90° from tube top, ΔT_{sub}=12.3 K and m_S= 34 kg/h

As an first approximation, the effect is considered only in the mass transfer equation, not in the heat transfer equation similar to the Hijikata's method(1989a, b) for a binary mixture. The integral equations for the mass and energy conservation are

$$\frac{d}{dx}\int \rho u(w_f - w)dy - \rho_i v_i (w_f - w_i) = \rho_i D \left[\frac{\partial w}{\partial y}\right]_i \tag{16}$$

$$\frac{d}{dx}\int \rho u(T_f - T)dy = \rho_i \kappa \left[\frac{\partial T}{\partial y}\right]_i \tag{17}$$

where v_i is the normal velocity to the wall generated by condensation. From the equation of continuity for the non-condensing components,

$$v_i = -\frac{D}{1-w_i}\left[\frac{\partial w}{\partial y}\right]_i \tag{18}$$

Non-dimensional concentration and temperature are defined as

$$w^* = \frac{w - w_i}{w_f - w_i} \tag{19}$$

$$T^* = \frac{T - T_i}{T_f - T_i} \tag{20}$$

By using Eqs.(18), (19) and (20), Eqs. (16) and (17) can be transformed to the following:

$$\frac{d}{dx}\int \rho u(1 - w^*)dy = \frac{1-w_f}{1-w_i}\rho_i D \left[\frac{\partial w^*}{\partial y}\right]_i \tag{21}$$

$$\frac{d}{dx}\int \rho u(1 - T^*)dy = \rho_i \kappa \left[\frac{\partial T^*}{\partial y}\right]_i \tag{22}$$

The boundary conditions for the non-dimensional concentration and temperature are same and are $w^*=T^*=0$ at $y=0$ and $w^*=T^*=1$ at $y=\infty$. So the relation of

$$\kappa = \frac{1-w_f}{1-w_i}D \tag{23}$$

is presumed in Eqs.(21) and (22), non-dimensional concentration profile coincides with that of the temperature. In the other words, the non-dimensional concentration profile can be estimated from that of the temperature by using Eq.(23). The empirical correlation for the heat transfer gives

$$\left[\frac{\partial T^*}{\partial y}\right]_i = \frac{Nu_f}{d} = \frac{1}{d} c\, Re_f^{0.6}\, Pr_f^m (Pr_f / Pr_w)^{0.25} \tag{24}$$

The gradient of concentration can be estimated by using Eq.(23) in Eq.(24) as

$$\left[\frac{\partial w^*}{\partial y}\right]_i = \frac{1}{d} c\, Re_f^{0.6}\, Sc_f^m (Sc_f / Sc_w)^{0.25}\left(\frac{1-w_i}{1-w_f}\right)^m \tag{25}$$

By using Eq.(18), the condensation rate m_C is expressed as,

$$m_C = -\rho_i v_i = \frac{\rho_i D}{1-w_i}\left[\frac{\partial w}{\partial y}\right]_i = \frac{w_f - w_i}{1-w_i}\rho_i D \left[\frac{\partial w^*}{\partial y}\right]_i \tag{26}$$

Equations (25) and (26) give the modified Sherwood number taking account of the mass absorption effect at the wall. The modified Sherwood number becomes

$$Sh_f = \frac{m_C d}{\rho_i D(w_f - w_i)} = \frac{1}{1-w_i}\left(\frac{1-w_i}{1-w_f}\right)^m c\, Re_f^{0.6}\, Sc_f^m (Sc_f / Sc_w)^{0.25} \tag{27}$$

Shown in Figs.9 and 10 is the relation between the total heat flux and the average tube wall temperature with and without steam injection. The steam mass concentration w_f was 0.103 without steam injection and increased to 0.216~0.279 with injection. The corresponding dew point increased from 55.5°C to 70.4~75.6°C. The dashed line predicted by the simple analogy correlation agrees well with the experimental heat flux without steam injection but is smaller than that with steam injection. The solid line calculated with the modified Sherwood number agrees well with the measured results with steam injection. The dot-dashed line is the prediction with the semi-empirical correlation for a single tube by Rose(1980). The Rose's prediction is slightly lower than the present data and the present modified correlation at the high wall temperature region near the dew point.

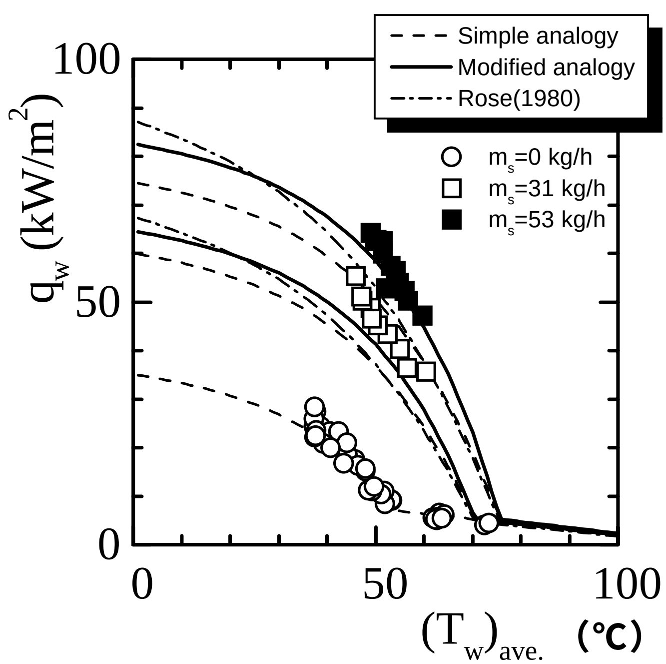

Fig. 9 Relation between heat flux and average wall temperature with and without steam injection

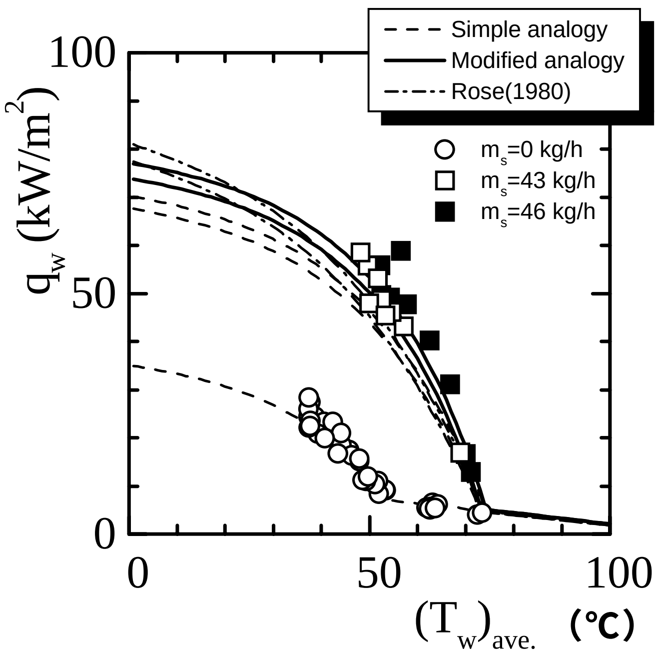

Fig. 10 Relation between heat flux and average wall temperature with and without steam injection

CONCLUSION

Condensation heat transfer of an actual flue gas from a natural gas boiler on horizontal stainless steel tubes was investigated experimentally. The experiments were conducted at different air ratios and steam mass concentrations of the flue gas in a wide range of tube wall temperature.

1. The boundary between the dry and condensation regions was given by the dew point defined as the saturation temperature corresponding to the partial pressure of steam in the flue gas. The pH value of the condensate was approximately 4~5 and slightly decreases with the air ratio and the fuel flow rate.
2. The condensation pattern was similar to dropwise condensation near the dew point. Decreasing the wall temperature, the wall region covered with a thin liquid film increased.
3. The condensation heat transfer was well predicted with the simple analogy correlation in the high wall temperature region. In the low wall temperature region less than 30°C, the total heat transfer was higher than that predicted by the analogy correlation.
4. At a high steam mass concentration artificially generated with steam injection, the total heat transfer was higher than the prediction by the simple analogy correlation. The analogy correlation using the modified Sherwood number taking account of the mass absorption effect was proposed. The modified correlation gave good prediction on the heat flux at the high steam mass concentration.

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