Proceedings of the ASME 2011 Power Conference (POWER 2011)
and the International Conference on Power Engineering 2011 (ICOPE-11)
July 12-14, 2011, Denver, Colorado, USA

POWER2011-55359

EFFECT OF HEADER ON LATENT HEAT RECOVERY HEAT EXCHANGER

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ABSTRACT

The most part of energy losses in heat & power system is due to the heat released by the exhaust gas to atmosphere. The exhaust gas consists of non-condensable gas and steam with sensible and latent heat. As a lot of latent heat is included in the exhaust gas, its recovery is very important to improve the system efficiency. Based on the previous basic studies, a thermal hydraulic prediction method for latent heat recovery exchangers was proposed. Two kinds of compact heat exchanger with staggered banks of large and small diameter tubes were designed and fabricated based on the prediction method. In the calculations varying the various parameters, approximately the same heat recovery rate was obtained with both the heat exchangers. The more compactness was obtained with the small tubes at a desired heat recovery rate. The pressure loss in gas side was slightly smaller and that in water side was significantly larger incase of the small tube. By adapting the single header instead of conventional multi header, the pressure loss in the water side could be significantly reduced but the reduction rate of heat recovery was only between 40 to 10%.

INTRODUCTION

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

Based on the previous basic studies [1-3], a prediction method was proposed for the design of heat exchanger to recover the latent heat in the exhaust flue gas. The modified Sherwood number taking account of the mass absorption effect on the heat transfer tubes is used for the condensation of steam in the presence of non-condensable gas. Laminar film of condensate on the tubes is assumed to evaluate the heat resistance due to the inundation. In the calculation procedure, it is possible that the gas temperature coincides with the dew point which is the saturation temperature corresponding to the partial pressure of steam in the exhaust flue gas. When the gas temperature decreases below the dew point, the condensation of steam in the gas takes place and the latent heat increases the gas temperature until it coincides with the dew point.

For condensation from a steam-gas mixture flowing normal to horizontal rows of tubes, an approximate analogy relation between heat and mass transfer was obtained with semi-theoretical consideration taking account of the mass absorption effect on the wall in the previous study [2].

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

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

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

Equations (1) and (2) are heat transfer and mass transfer correlations, respectively. The mass transfer equation can be derived if the heat transfer function of Nu is known. These correlations gave good predictions when the steam mass concentration was less than 25% in single and multiple stages of heat transfer tubes using actual flue gas. Also at the steam mass concentration more than 25%, the good predictions were obtained in the experiment of single and multiple stages using air-steam mixture.

NOMENCLATURE

C_p: specific heat [J/kg]
d_o: outer diameter of tube [m]
d_i: inner diameter of tube [m]
D: mass diffusivity [m^2/s]
h_V: heat transfer coefficient [W/(m^{2}K)]

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

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

CONSTITUTIVE EQUATIONS FOR PREDICTION

Heat resistance of condensate

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

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

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

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

The heat conductivity of film is

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

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

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

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

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

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

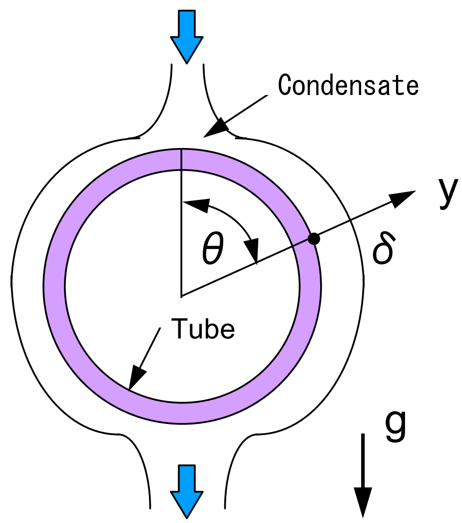

Fig. 1 Heat resistance of condensate film

Heat and mass transfer in gas side

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

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

The convection heat flux is expressed as

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

The condensation heat flux q_C can be expressed as,

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

where W_i is the mass concentration of saturated steam at the wall temperature T_i. Based on the previous studies[4], the Nusselt number Nu_f for the average convective heat transfer coefficient is

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

Zukauskas[4] proposed a=0.6, b=0.36 and

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

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

for a staggered bank in the range of $10^3 < Re_f \leqq 2 \times 10^5$. For the condensation of steam on heat transfer tubes, the modified analogy relation of Eqs.(1) and (2) gives

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

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

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

Mixture gas was treated as a mixture of N_2, O_2 and H_2O and its property was estimated with special combinations of each gas property proposed by the previous studies. For example, the heat conductivity and the viscosity were estimated with the methods by Lindsay&Bromley [5] and Wilke[6], respectively. It is considered that a strong correlation exists between the thermal and mass diffusivities. As a first attempt, the mass diffusivity of steam in mixture gas was estimated with the well-known mass diffusivity of steam in air as

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

where κ and κ_{air} are the thermal diffusivities of flue gas and dry air, respectively. The diffusivity of steam in air can be expressed as[7],

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

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

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

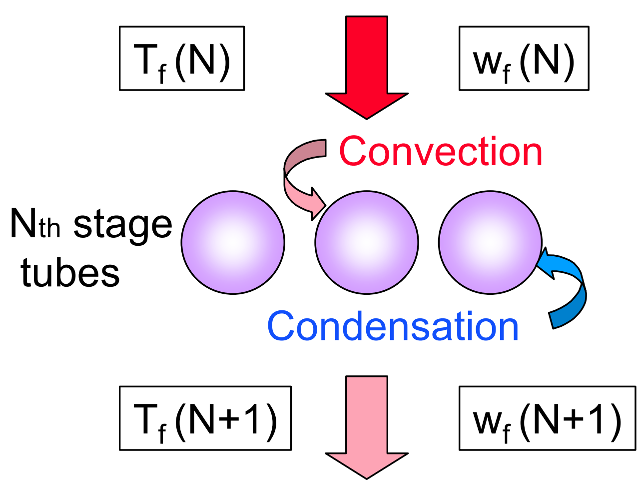

Fig. 2 One-dimensional calculation

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

where A_w is the heat transfer area per a stage.

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

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

Heat conduction in tube

The heat conductivity for the inconel or austenite stainless steel is given with the following approximate correlation [8].

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

where T_t is the average temperature of tube as,

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

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

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

Heat transfer in water side

Heat transfer correlation by Dittus-Boelter taking account of the pipe inlet region is used. The coefficient by McAdams[9] was used for the modification.

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

where L is the heating length of tube.

Pressure loss calculation

The pressure loss per a stage of tube in gas side is,

$$\Delta P = 2 f \rho_f u^2 \tag{24}$$

For the staggered bank of bare tube, Jacob[10] proposed the following coefficient f,

$$f = \left[0.25 + \frac{0.118}{\left\{\left(\dfrac{S_1}{d_o}\right) - 1\right\}^{1.08}}\right] \mathrm{Re}_f^{\,-0.16} \tag{25}$$

This correlation can be used also in the range of $10^3 < Re_f \leqq 2\times 10^5$. The pressure loss per a tube in water side is expressed as followings assuming the inlet and outlet pressure loss coefficient of 1.5,

$$\Delta p = \left(1.5 + 4f\frac{L}{d_i}\right)\frac{\rho u^2}{2} \tag{26}$$

where the friction coefficient f is,

$f = 16/Re$ for laminar flow,

$f = 0.079\,Re^{-0.25}$ for turbulent flow.

COMPACT HEAT EXCHANGER

Shown in Fig.3 is a schematic of heat exchanger. Heat transfer tubes were installed in a rectangular duct of 205x205mm. The tubes at each stage were connected with a header to maintain the same flow rate of feed water. The feed water was supplied at the downstream of gas flow and flows counter-currently to the upstream. In the present study, two kinds of heat transfer tubes with the different diameter were used. The height L of the heat exchanger necessary to recover a desired heat strongly depends on the diameter of heat transfer tubes.

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

Shown in Fig.5 are photograph of the two heat exchangers designed with the same heat recovery rate. The height of “Large” heat exchanger with the larger tubes was 820mm, on the other hands, that of “Small” with the smaller tubes was only 160mm.

Generally the heat transfer is described as,

$$Nu \approx \mathrm{Re}^m$$

where m is between 0 and 0.8. So the heat transfer coefficient *h* can be expressed as,

$$h \approx \frac{1}{d^{1-m}}$$

The smaller diameter *d* of tube results as the higher heat transfer coefficient and the analogy relation gives the higher mass transfer coefficient. The more compactness of heat exchanger can be obtained with the smaller heat transfer tubes.

Table 1 shows the major dimensions. The total number of heat transfer tubes was 380 in the “Large” and 500 in the “Small”. The total weight of heat transfer tubes was 21.6kg in the “Large” and 7.65kg in the “Small”. The tube weight of “Small” was approximately 1/3 of “Large”. The heat transfer area at the gas side of “Small” was approximately the half of “Large”. The compactness was achieved with the smaller tubes.

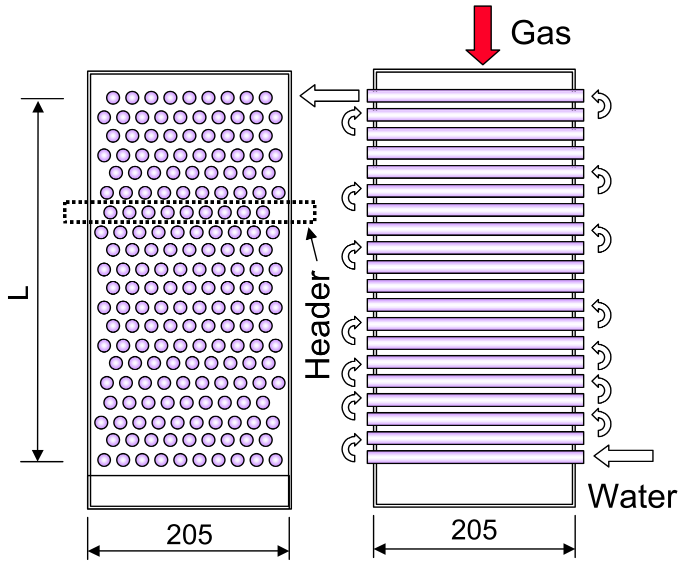

Fig. 3 Schematic of tube bank

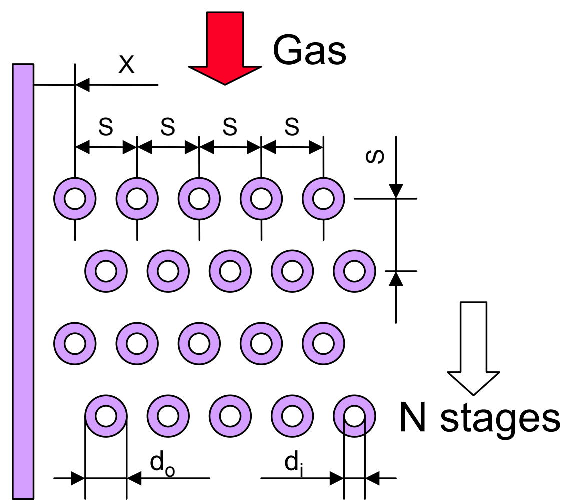

Fig. 4 Schematic of tube array

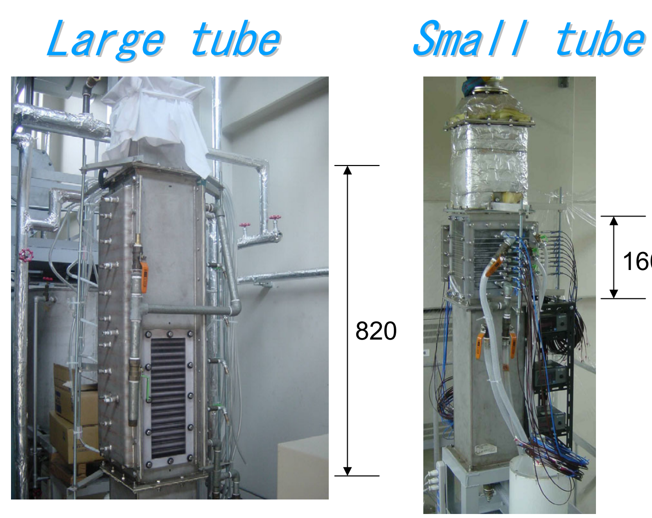

Fig. 5 Photograph of two HXs

Table1 Major dimensions

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

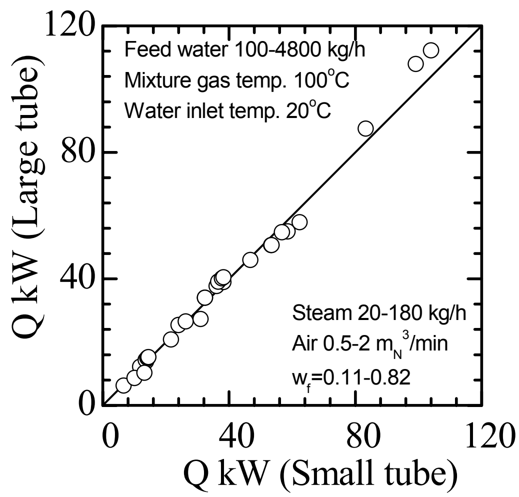

Fig. 6 Comparison of calculated heat recovery for "Large" and "Small" heat exchangers

The thermal hydraulic behavior in the compact heat exchangers was experimentally studied with air-steam mixture gas in previous study [3]. In the parametric experiments varying the steam mass concentration, the temperature distributions of cooling water and mixture gas were measured. It is reported that the experimental results agreed well with the present prediction method.

Shown in Fig.6 is the comparison of calculated heat recovery for "Large" and "Small" heat exchangers when the mixture gas temperature is fixed at 100°C and the feed water temperature is fixed at 20°C. The nominal feed water flow rate is 600 kg/h but in the calculation the flow rate is varied between 100 and 4800kg/h. Even when the steam mass concentration was varied between 0.11 and 0.82, the same heat recovery rate is successfully obtained with both heat exchangers.

PRESSURE LOSS IN HEAT EXCHANGER

The gas-side pressure loss ratio of "Small" to "Large" HX was calculated at the different steam concentration as shown in Fig.7. The flow rate of mixture gas and feed water is same in "Small" and "Large" HX. The calculated conditions are fixed at the mixture gas temperature of 100°C, the feed water temperature of 20°C and the feed water flow rate of 600kg/h. The pressure loss of "Small" HX is approximately 40% smaller than that of "Large" HX.

Shown in Fig.8 is the water-side pressure loss ratio of "Small" to "Large" HX at the different steam concentration. The flow rate of mixture gas and feed water is same in "Small" and "Large" HX. The calculated conditions are also fixed at the mixture gas temperature of 100°C, the feed water temperature of 20°C and the feed water flow rate of 600kg/h. The pressure loss of "Small" HX is approximately 40 times larger than that of "Large" HX.

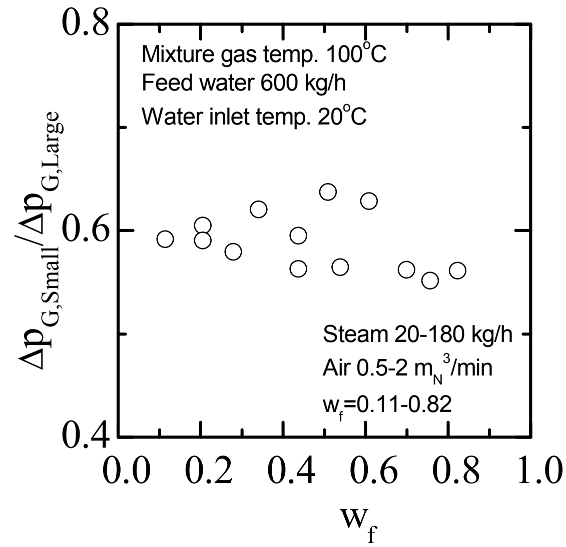

Fig. 7 Ratio of pressure loss in gas side at different steam concentration

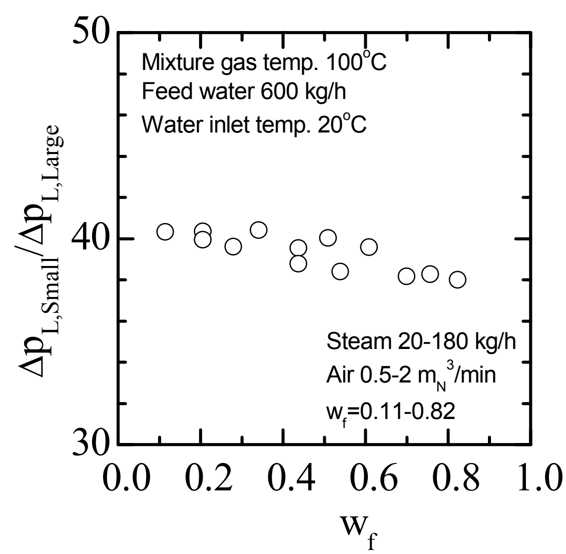

Fig. 8 Ratio of pressure loss in water side at different steam concentration

EFFECT OF HEADER

By using the smaller tubes, the more compactness of heat exchanger for the latent heat recovery was successively achieved. But the pressure loss in water side increased significantly compared to the conventional heat exchanger using the larger tubes. To reduce the pressure loss in water side, the single header was proposed instead of the conventional multi header.

Shown in Fig.9 is the comparison of multi and single header. In the multi header, the tubes at each stage were connected with a header to maintain the same flow rate of feed water in each tube at a stage. The feed water was supplied at the downstream of gas flow and flows counter-currently to the upstream. The temperature of feed water increases stage by stage heated with the mixture gas. In the single header, the tubes of right and left-side were connected with the single header. So the feed water flows simultaneously into all the tubes from the right side header to the left side header.

The water-side pressure loss ratio of single to multi header was calculated at the different feed water flow rate Q_L and steam concentration in "Small" heat exchanger as shown in Fig.10. At the nominal feed water flow rate of 600 kg/h, the ratio is less than 10^{-3} indicating the significant reduction of water-side pressure loss by adapting the single header. The ratio further decreases with increase of feed water flow rate.

The single header provides the smaller waterside pressure loss and has a possibility to reduce the heat recovery rate. The heat recovery ratio of single to multi header was calculated at the different feed water flow rate Q_L and steam concentration w_f in "Small" heat exchanger as shown in Fig.11. At the nominal feed water flow rate of 600 kg/h, the ratio is approximately 0.7 in spite of the significant reduction of water-side pressure loss by adapting the single header. The ratio further increases with increase of feed water flow rate. At the high feed water flow rate, the increase of water temperature is suppressed and the difference of header can be negligible. It is interesting that the ratio is smaller at the higher steam concentration.

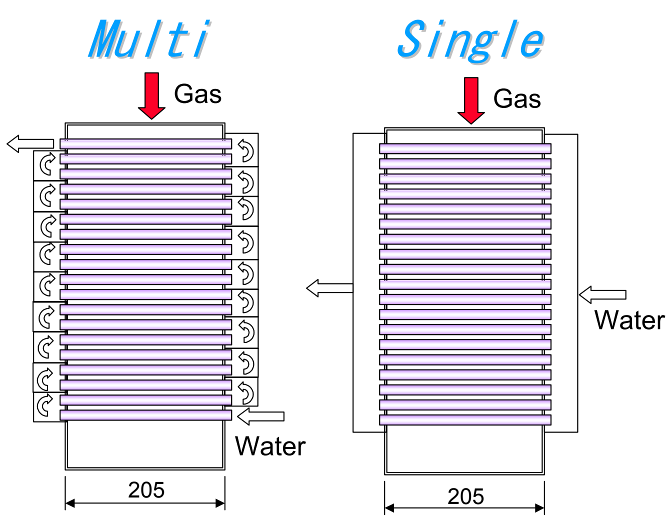

Fig. 9 Comparison of multi and single header

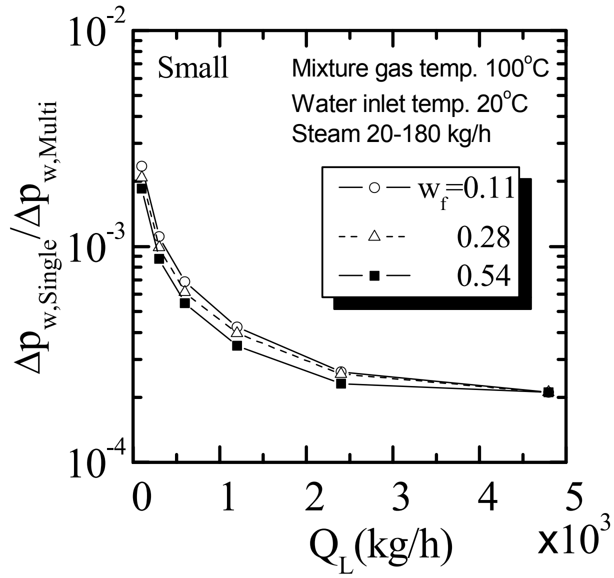

Fig. 10 Ratio of pressure loss in water side at different water flow rate in "Small" HX

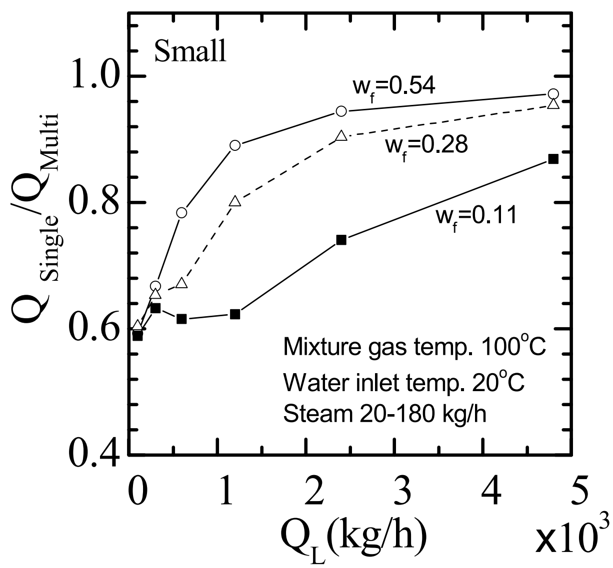

Fig. 11 Ratio of heat recovery at different water flow rate in "Small" HX

When the feed water flow rate is less than 600 kg/h, non-condensing region appears in the upper parts of heat exchanger. The existence of dry region is affected with the header type and the steam concentration. The non-monotonous increase of ratio in Fig.11 is considered to be due to the dry region. It should be noted that the pressure loss in the waterside could be significantly reduced but the reduction rate of heat recovery was only between 40 to 10% by using the single header.

CONCLUSION

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

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

(3) The thermal hydraulic behavior in the compact heat exchangers of bare tubes of 10.5 and 4mm was calculated. In the parametric calculations varying the steam mass concentration, approximately the same heat recovery rate was obtained with both the heat exchangers.

(4) The pressure loss in the gas side was slightly smaller in the smaller tube. However, the pressure loss in the waterside was significantly larger in the smaller tube.

(5) By using the single header, the pressure loss in the waterside could be significantly reduced but the reduction rate of heat recovery was only between 40 to 10%.

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