Heat Transfer Characteristics and Pressure Drop of a Fluidized Bed Heat Exchanger without Baffle Plate

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ABSTRACT: A new fluidized bed heat exchanger for exhaust gas heat recovery is developed. Compared to the existing ones, the present heat exchanger system is featured by the particle fluidization method which does not depend on conventionally used baffle plate with holes and by the multiple downcomer tubes to extract heat energy from hot particles during the time particles moves down to be fed again to the hot gas line. Particles are introduced to the main hot gas stream alongside the pipe circumference. The heat exchanger performance and pressure drop are evaluated through experiments for the present gas-to-water heat exchanger system.

Nomenclature

C_d : flow factor
ΔT_m : log mean temperature difference [K]
U : overall heat transfer coefficient [W/m$^2\cdot$K]

Greek symbols

β : diameter ratio of orifice, d/D

Subscripts

g : gas side
w : water side

1. Introduction

Furnaces and boilers are the intensive energy-consuming examples in the industry and the wasted energy with the exhaust gas are considered to amount from 30 to 60% of the total energy of the used fuel. In the case of exhaust gases from incinerators, emphasis has been drawn to the environmental post processes rather than to the heat recovery due to the difficulties related with ashes and corrosive exhaust gases. Therefore, it is a nation-wide concern to have a measure for heat recovery from flue gases from industrial furnaces, boilers and incinerators for better use of energy resources. In the flue gas heat recovery process with conventional heat exchangers, the deposition of fly ash on the heat transfer surface results in lower heat transfer performance and corrosion, and accordingly periodical cleaning is necessary for normal operation. Therefore it has been pursued by many to devise effective measure for heat recovery from flue gases avoiding fouling problems.

Since solid particles as glass beads, when they are suspended in the flue gas stream of heat exchangers, are known to help reducing corrosion related problems, to increase heat transfer rates and to have self cleaning characteristics, different types of heat exchangers have

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already been devised and are being used in the abroad, while the development of those kind of heat exchangers is being pursued for practical applications domestically.[1-5] Park[1] measured pressure drop and heat transfer performance in the vertically oriented heat transfer passage with suspended particles and compared the pressure loss with the prediction by using one dimensional pressure loss model, while Park et al.[2] theoretically investigated the heat transfer mechanism of gas-particle direct contact type heat exchangers considering radiation effects. Jung et al.[3] conducted experiments on the horizontal multi-tube type heat exchanger with particulate cross flow and reported the increase in the heat transfer coefficient by 93 percent compared against the case without particles. Lee et al.[4] experimentally investigated by measuring heat transfer coefficients on the surface of vertically oriented baffle type fluidized bed heat exchanger considering the effects of design parameters as pipe diameter, baffle plate hole diameter and the height of mixing chamber, while Jun and Lee[5] conducted three-dimensional flow and heat transfer analyses for the same heat exchanger. In the latter, they investigated on the suspended particles motion in the vertical pipe of finite length.

One of the key problems related with fluidized bed heat exchangers is the excessive pressure drop and flow instability, which results from the particle fluidization method using baffle plates. Therefore, it seems necessary to consider other type fluidization mechanism that minimizes the pressure drop and flow instability while maintaining higher heat transfer performance for longer period. In the present study, the heat transfer performance and pressure drop have been measured for a lab-scale heat exchanger for flue gas heat recovery application uniquely featured by non-baffle plate fluidization method.

2. Fluidized bed heat exchanger test apparatus

Figure 1 shows the schemetic diagram of a fluidized bed heat exchanger test apparatus, which is composed of combustor section and heat exchanger section. The combustor section provides constant flow rate of hot gas to the inlet of the vertically oriented heat exchanger, which is required for thermal analysis. A 35 kW household boiler (30,000 kcal/hr) fueled by kerosene is modified for use as the combustor section of the test apparatus, by eliminating the heat transfer enhancement device and by prohibiting the circulation of feedwater. About 450℃ of hot gas could be supplied to the heat exchanger inlet in this arrangement. The hot

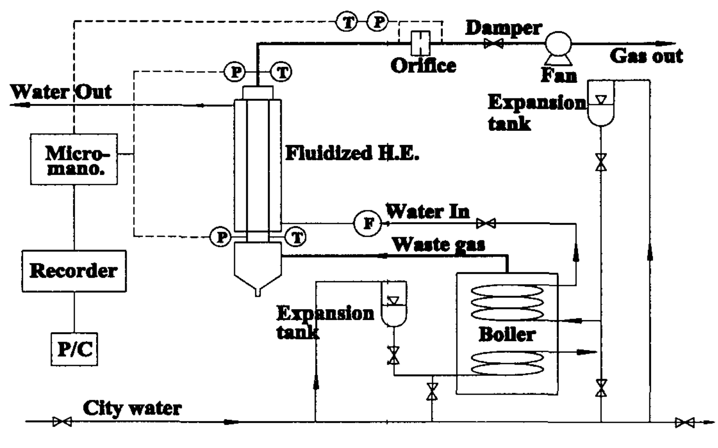

Fig. 1 Schematic diagram of fluidized bed heat exchanger test apparatus.

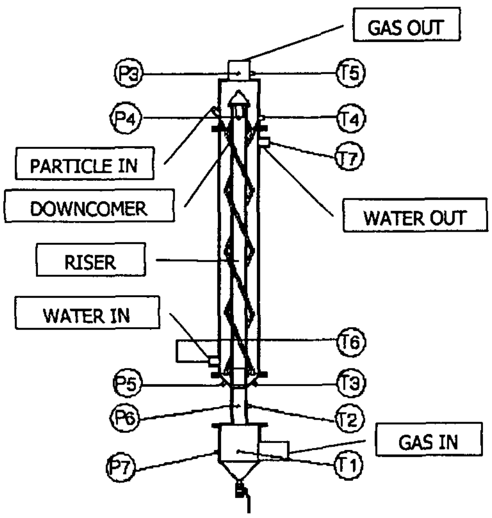

Fig. 2 Tested heat exchanger.

gas supplied from the combustor (boiler) enters the lower part of the heat exchanger to transfer heat energy to cooler part through a vertical tube (riser) and four particle downcomer tubes wound around the riser. The gas flow rate is measured by the orifice installed at the downstream of the heat exchanger and is controlled by a butterfly-type damper. Measured pressure drop through the orifice is converted to the equivalent gas flow rate with temperature correction. The gas flow is driven by a radial fan located downstream of the damper.

The feedwater for the heat exchanger is supplied from the makeup water line of the boiler. The city water is pre-heated to about 60℃ passing through the combustor (boiler) along the wound tube then enters the coolant inlet of the heat exchanger. The water flow rate is measured using a digital flow meter (OVAL Flowpet) right upstream of the heat exchanger inlet. The feedwater is then further heated in the second heat exchanger before being sent for possible usage. The temperature of the feedwater is measured at five locations (from T1 to T5) including the inlet and the outlet of the heat exchanger by using K-type thermocouples (See Fig. 2 and Table 1).

Particles for circulation are dried and weighed

Table 1 Temperature and pressure measurement locations

Pressure tap		T/C	
P1	Orifice upstream	T1	H.E. inlet
P2	Orifice downstream	T2	Riser inlet
P3	H.E. exit	T3	Downcomer exit
P4	Riser exit	T4	Riser exit
P5	Downcomer exit	T5	H.E. exit
P6	Riser inlet	T6	Water inlet
P7	H.E. inlet	T7	Water exit

ahead of feeding and are fed through the ball valve on the upper part of the heat exchanger. These particles first fall in the upper region of the heat exchanger unit where the gas come out from the main vertical pipe (riser of I.D. 54 mm) and then fall again through the four wound tubes (downcomer of I.D. 11 mm), inside of which direct heat transfer between particles and tube surface may occur. Particles are then collected in a small region under the downcomer tubes to be introduced to the hot gas flow at the lower part of the riser through an annular passage. Once the particles are entrained in the main gas stream they move upward along the riser in a suspended condition and inertially separated in the upper region of the unit from the flue gas. In this manner, particles circulate inside the unit, while the flue gas pass through the riser and is exhausted.

3. Test conditions

3.1 Orifice calibration

The flow rate of the flue gas is measured by using D and D/2 taps orifice downstream of the heat exchanger unit. In the present study, a pipe of 70 mm inner diameter (D) and an orifice plate with diameter ratio ($\beta = d/D$) of 0.57 are used. The gas flow rate is expressed as

$$Q = C_d A_t \left[\frac{2(p_1 - p_2)/\rho}{1-\beta^4} \right]^{1/2} \tag{1}$$

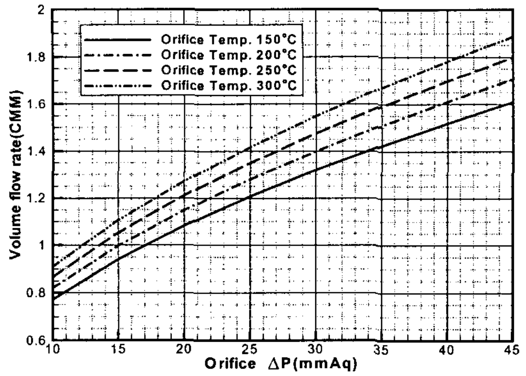

Fig. 3 Flow rate (Q) versus orifice pressure drop (Δp) at different gas temperature.

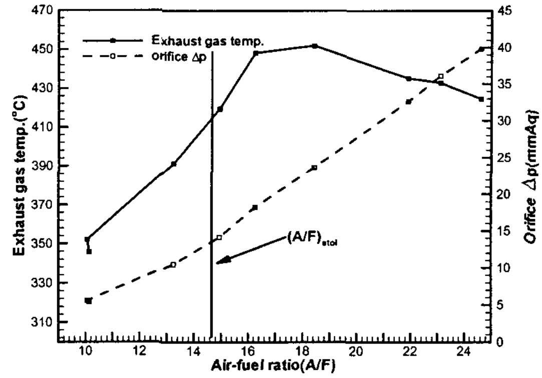

Fig. 4 Air-to-fuel ratio vs. exhaust gas temperature.

where the dimensionless discharge coefficient C_d for D and D/2 taps orifice is as follows[6];

$$C_d = f(\beta) + 91.71\beta^{2.5}\mathrm{Re}_D^{-0.75} + \frac{0.09\beta^4}{1-\beta^4}F_1 - 0.0337\beta^3 F_2 \tag{2}$$

with

$$f(\beta) = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 \tag{3}$$

The values of empirical constants F_1 and F_2 are 0.4333 and 0.47, respectively. The discharge coefficient C_d is a function of β and the Reynolds number. When the value of β is 0.57, C_d remains almost constant at Reynolds number ($\mathrm{Re}_D = UD/\nu$) higher than 10^6 while it decreases from 0.615 to 0.605 as the Re_D increases in the range from 3×10^4 to 10^6. The density change with temperature is considered through Sutherland law. Fig. 3 shows the volume flow rate of gas versus the pressure drop through the orifice at different gas temperatures. From this figure it could be noticed that 15~20% of difference in the volume flow rates may occur if the gas temperature varies by 150℃ (for example from 150 to 300℃) at the same pressure difference reading. It is to be emphasized therefore that care should be paid with temperature especially when one deals with hot gases.

3.2 Determination of air flow rate

For the flue gas is directly supplied from the burner (nominal fuel flow rate of 3,900 cm^3/hr) *it is necessary to monitor the gas temperature* with the change of gas flow rate. Fig. 4 shows the measured temperature at the heat exchanger inlet with respect to the flow rate variation converted from the pressure drop through the orifice. If the air supply decreases from the stoichiometric condition, the gas temperature decreases, while as the excess air increases the gas temperature increases to a certain point and decreases. In the case of Kerosene, the density of 791 kg/m^3, lower heating value of 46,035 kJ/kg and the stoichiometric air-to-fuel ratio of 14.79 are used for calculation. According to the figure, the highest gas temperature is obtained at A/F=18.4 which is 20% higher than the stoichiometric A/F of 14.8 and the value is 452℃ at Δp_{orif} of 231 Pa (23.5 mmAq). It could be concluded from the experiments that in order to get hot gas about 450℃ at the heat exchanger inlet it is necessary to maintain the orifice pressure drop ranging from 176 to 235 Pa (18 to 24 mmAq). If the air flow rate is less than this, soot may appear due to the unburned carbon fraction and lower gas temperature is expected. The temperature drop from the heat exchanger exit to the orifice ranged from 85 to 100℃ in the present experiments.

3.3 Velocity prediction in the riser

Although it is necessary to know the gas velocity passing through the riser to determine the condition for particle circulation, it is difficult to measure directly by penetrating the probe inside the tube. For this reason a computer simulation tool[7] is adopted. Figure 5 shows three-dimensional solution domain used for numerical simulation and a part of the results representing velocity distribution at a mid-cross section with incoming gas flow rate of 1.31 m^3/min (Inlet velocity of 5.65 m/s). There exist a main vertical passage (riser) surrounded by four tubes through which particles may fall down. According to the numerical simulation results, the average velocity ratio of the center section to the surrounding tube sections are known to be about 0.27 at the two different inlet conditions, i.e., inlet velocity of 5.2 m/s and 5.65 m/s, respectively. Based on this information, it is possible to estimate the rising gas velocity at the center section as

$$V_p = \frac{Q}{A_p + 1.08A_{tube}} \tag{4}$$

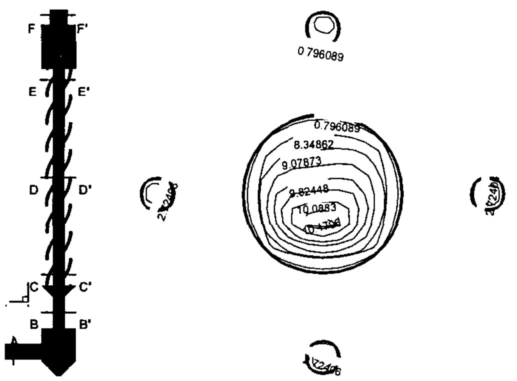

Fig. 5 Computational domain and velocity contour at mid-section D-D′(Q=1.31 m^3/min; velocity in m/s).

where the cross-sectional area of the riser pipe $A_p = \frac{\pi}{4}\left(\frac{54.0}{10}\right)^2 = 22.90\ cm^2$ and the area of each surrounding tube $A_{tube} = \frac{\pi}{4}\left(\frac{11.0}{10}\right)^2 = 0.95\ cm^2$. It is estimated that the flow rate passing through the surrounding tubes is approximately 4.3% of the total gas flow rate. When the orifice pressure drop is set to be 195.8 Pa (20 mmAq) for experiment, the estimated flow rate is Q=1.08 m^3/min and the gas velocity through the riser is 7.75 m/s.

3.4 Particles characteristics and circulation

To enhance the heat transfer performance and to reduce deposition of ash (fouling) by circulating particles, appropriate particles need to be selected considering the material property (related with the density, shape, and thermal conductivity) and size (diameter). For the present study glass beads commercially designated as JB-700 (diameter range 425~850 μm, S.G. 2.62) is selected and used. Visualization test (cold flow test) to make sure the particle circulation is conducted ahead of the heat exchanger experiments.

3.5 Water supply

In order to avoid condensation it was necessary to maintain the water temperature at the heat exchanger inlet to be above 60℃ and the resulting water flow rate is set to be 7,900 cm^3/min.

After all the above provisions are met, the tests for the fluidized bed heat exchanger is performed. Tests are arranged in three steps, i. e., (1) without particle, (2) with 300 cm^3 of particles, (3) with 600 cm^3 of particles. For measurement of temperature and pressure a digital recorder DA-100 (Yokogawa, 30 Ch.) is used with micromanometers.

4. Experimental results and discussion

4.1 Heat transfer performance

Figure 6 shows the heat transfer test results

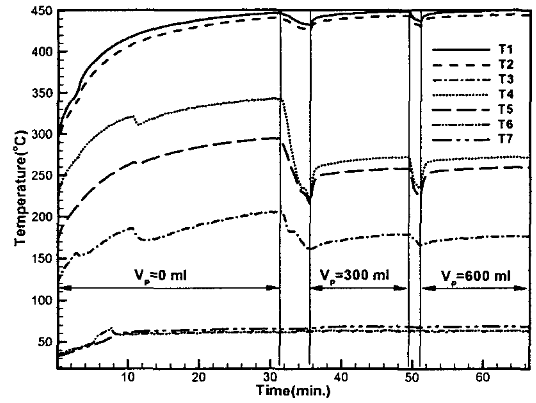

Fig. 6 Temperature vs. time (Q_g=1.08 m^3/min, Q_w=7,900 cm^3/min).

with the gas flow rate Q_g=1.08 m^3/min (equivalent to Δp_{orif}=196 Pa), water flow rate Q_w= 7,900 cm^3/min.

Once the steady state condition is reached after the ignition of the burner, particles are fed into the heat exchanger by 300 cm^3 through the hole on the top of the heat exchanger unit. Another dose of 300 cm^3 particles are added into the system after the steady condition is obtained again. In the figure legend, measured temperature of the gas are represented by T1 to T5, while T6 and T7 represent inlet and outlet water temperature, respectively.

According to the test results, the gas temperature at the heat exchanger inlet (T1) and vertical pipe inlet (T2) steadily approach the equilibrium temperature of approximately 450℃. T2 shows slightly lower values than T1 by 5℃, which seems due to the aerodynamic effect. When first 300 cm^3 of particles are fed, significant temperature difference at the exit is observed, however no significant change was monitored with the second dose. Similar behavior is monitored in the pressure drop through the vertical pipe. This seems due to the limitation of particle transport capacity of carrier gas flow which may be influenced by the fluidization methods, flow condition (velocity and the velocity fluctuation) and particles characteristics. By introducing particles outlet temperatures (T4 and T5) shows significant variation, while inlet temperatures (T1 and T2) remain unaffected.

Table 2 Measured temperature and pressure difference

Item (unit)	Particle vol. (ml) 0 (1)	600 (2)	Ratio (2)/(1)
T1 (℃)	446.4	448.8	0.995
T2 (℃)	440.1	443.2	1.007
T3 (℃)	204.8	178.2	0.870
T4 (℃)	342.2	271.6	0.794
T5 (℃)	294.0	258.1	0.878
T6 (℃)	61.9	63.3	1.023
T7 (℃)	65.9	68.4	1.038
$\Delta p_{\text{orifice}}$ (mmAq)	20.1	19.2	0.955

For quantitative analysis, the test conditions and the temperature measurement results are summarized before and after the particle feeding and is shown in Table 2. According to this table the temperature drop through the vertical pipe (riser) [ΔT_{2-4}] increases from 97.9℃ with no particles to 171.6℃ with 600 cm^3 particles resulting in 75% increase. The temperature drop through the entire heat exchanger [ΔT_{1-5}] changes from 152.4℃ with no particles to 190.7 ℃ resulting in 25% increase.

Total heat transferred can be expressed as

$$q = \rho_g Q_g c_{p_g} \Delta T_g \quad \text{(gas side)} \tag{5}$$

$$q = \rho_w Q_w c_{p_w} \Delta T_w \quad \text{(water side)} \tag{6}$$

and the total heat trasferred from the gas side is increased by 24%.

For the evaluation of heat transfer performance of the heat exchanger overall heat transfer coefficient U[8] is determined by using

$$q = UA\Delta T_m \tag{7}$$

where ΔT_m is the log mean temperture difference defined as

Table 3 Effect of particle loading on the heat transfer performance

Description	ΔT_m (℃)	A (m^2)	U ($W/m^2 \cdot K$)
No particle	299.6	0.39	19.96
V_p=600 mL	276.1	0.39	26.92

$$\Delta T_m = \frac{(T_{g,e} - T_{w,e}) - (T_{g,i} - T_{w,i})}{\ln[(T_{g,e} - T_{w,e})/(T_{g,i} - T_{w,i})]} \quad (8)$$

For the present case, $T_{g,e} = T_5$, $T_{g,i} = T_1$, $T_{w,e} = T_7$, $T_{g,i} = T_6$ are used to determine ΔT_m. Through the analysis it is found that by circulating particles with the flue gas approximately 35% of increase in the overall heat transfer coefficient U could be achieved.

4.2 Pressure loss

Major pressure loss arise from the riser through which gas-particle mixture is raised. Therefore the measurement for pressure drop between the inlet (P7) and the outlet (P8) of the riser pipe is monitored. The measured pressure drop characteristics is shown in Fig. 7. According to the figure the pressure drop through the riser pipe is 78 Pa (8.0 mmAq) with no particles and increases to 278 Pa (28.4 mmAq) as 300 cm^3 of particles are fed. However, further pressure drop is not appreciable with the second dose of particles, which is due to the limitation of particle transport capacity of the carrier gas flow as mentioned in the previous section.

To demonstrate the reduced pressure drop characteristics of the present approach, the pressure loss is expressed in terms of so called minor loss coefficient ($K = h_l/(V^2/2)$)[9] typically used for loss estimation in the pipe system. Table 4 shows the comparison of the minor loss coefficients from the present experiments and from the previous experiments using baffle plate. The compared data is obtained from the test which has the pipe-to-baffle plate hole diameter (D_p/D_h) of 1.33. According to the compared data, the value of K for the present case (annular injection) is about 13% of the value of K using baffle plate type. This is important because, by using this approach, additional pressure drop arising from particle circulation can be minimized.

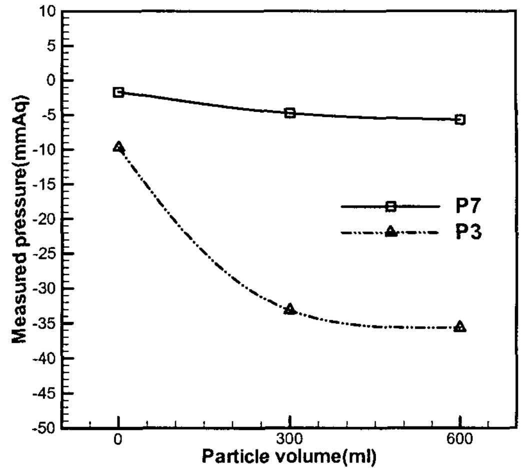

Fig. 7 Measured pressure at inlet (P7) and outlet (P3) of the heat exchanger.

5. Conclusions

In this study, a fluidized bed type heat exchanger model that may be used for heat recovery from flue gases is proposed. This type of heat exchanger can be appied to the heat recovery from waste gases from furnaces, boilers and incinerators with higher efficiency and reducing fouling related problems. Major feature

Table 4 Minor loss coefficients (K) for two different fluidizing methods

Test condition	Q (m^3/min)	D_{in} (mm)	V_{in} (m/s)	h_l (mm)	$K \times 10^3$
W/O baffle plate	1.08	70	4.68	8.0	0.7305
With baffle plate D_p/D_h=1.33	0.46	70	1.99	11.3	5.707

of the proposed heat exchanger may be summarized as follows;

(1) The present heat exchanger does not use (perforated) baffle plate for fluidization of particles rather aerodynamically suspends particles inside the pipe flow.

(2) The pressure drop through the heat exchanger is reduced significantly as demonstrated.

(3) By introducing particles successfully for circulation inside the heat exchanger system, it is demonstrated that the overall heat transfer coefficient U based on logarithmic mean temperature difference (LMTD) is increased by 35% in the present study.

Fluidized bed type heat exchangers have significant potential to recover wasted heat energy contained in the exhaust gases as demonstrated in the present study and need to be investigated further for practical applications. Further researches on the particle fluidization and circulation, design optimization, measurement technique, effects of particle sources, heat transfer mechanism of gas-particulate flow, and fouling problems are required.

Acknowledgements

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