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PERFORMANCE OF A NO-DISTRIBUTOR-FLUIDIZED-BED HEAT EXCHANGER

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ABSTRACT
To overcome the fouling of fly ash on the heat transfer surface and erosion and periodical cleaning which are the major drawbacks in conventional heat exchangers for flue gas heat recovery, a no-distributor-fluidized-bed (NDFB) heat exchanger with automatic particle controlling is devised. One of the main advantages of this model is the reduced pressure drop through the entire heat exchanger system, while increasing heat transfer performance. In the present study, the particulate flow behavior is observed at varying flow rate along with the change in particle size. Pressure loss with and without particle ingestion is also measured and the system feature is discussed. Heat transfer performance for the actual setup is also evaluated.

Keywords: Fluidized Bed (FB), Heat Exchanger, Heat Transfer, Pressure Drop, Fouling.

1. INTRODUCTION
Heat recovery from flue gas out of industrial furnaces, boilers and incinerators for better use of energy resources is a nationwide concern in most industrialized countries. Heat Exchangers and heat exchanger networks are extensively used for the purpose of recovering energy. The performance of these heat exchangers decreases continuously with time due to fouling depending on surface temperature, surface condition, construction material, fluid velocity, flow geometry and fluid composition.

Fluidized Beds have been widely used in heat recovery process because of their unique ability of rapid heat transfer and uniform temperature. Energy recovery from the waste heat not only reduces the consumption and expenses related with the purchased energy but also is considered as one of the most direct measures for rational use of energy resources and reduction of the energy usage, which contributes to the CO_2 reduction eventually.

To overcome the fouling of fly ash on the heat transfer surface and erosion and required periodic cleaning, which are the major drawbacks in conventional heat exchangers for flue gas heat recovery, fluidized bed heat exchangers are seriously studied by many investigators. It is generally well accepted that fluidized bed heat exchangers, in which particles are being circulated inside the system with some fluidization mechanism using exhaust gas, have good potential to achieve enhanced heat transfer performance with self-cleaning nature under even corrosive environments. However, due to the nature of two-phase (gas-solid) flow and the possible complexity of phase change such as the condensation in the case of low temperature applications, applications are challenged typically by operational instability and handling problems and different types of heat exchangers are being studied depending on specific applications. Recently, a FB heat exchanger model without a baffle-plate is devised by our research group [1-4]. One of the main advantages of this model is the reduced pressure drop through the entire heat exchanger system, while increasing heat transfer performance. This general trend is very well known, however, serious effort to develop heat exchangers that does not degrade in their performance has not been many. Fouling reduction or maintaining ‘no fouling’ conditions in heat exchanger design and operation has double folded significance. In pursuing heat recovery from waste gases single most challenging issue is a fouling related problem. Due to the fouling and scaling of conventional shell-tube heat exchangers those systems suffer from significant degradation in heat transfer performance, which results in the over-sizing of the heat exchangers and the increased maintenance and downtime costs. In most industrialized countries the capital loss due to the heat exchanger fouling is estimated to amount to 0.25 to 0.30 percent of the gross national product (GNP) [5]. More importantly in an environmental point of view, maintaining initial design performance of the economizer such as in an municipal solid waste (MSW) incinerator applications is critical to the successful operation of the post treatment equipments of flue gas, in which the key role of the economizer is to temper down the gas temperature to a certain value to avoid Dioxin generation rather than to recover waste heat itself. In our previous studies[6,7], a series of no-distributor fluidized bed (NDFB) heat exchangers have been devised for the possible applications to the heat recovery from waste gases such as from municipal solid waste (MSW) incinerators. These heat exchanger models featured particles circulation without having baffle plate for fluidization of particles. Rather fluidization through annular slit is employed with separate down comers, which resulted in significantly reduced pressure drop through the heat exchanger unit. However, due to their sophisticated handling requirements with particles and vulnerable characteristics to congestion of particle passages due to foreign objects as irregular burn-out materials, the practical applicability was considered limited.

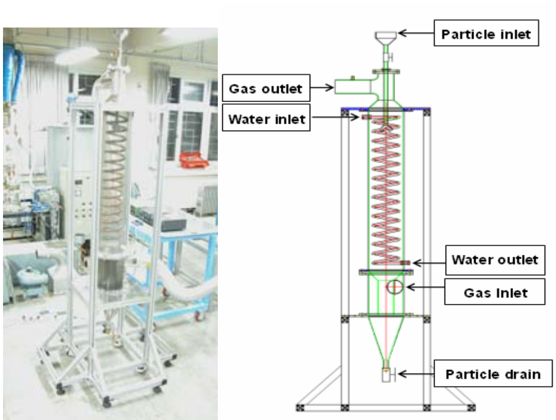
The third model is now under investigation. The present cold flow visualization model which resembles the actual heat exchanger is prepared to investigate the nature of the flow and the pressure drop in simulated operating conditions. In operation and design of fluidized bed reactors, it is important to know the total solids circulation rate, at the moment there are no reliable in situ techniques for measurement of solids circulation rates. Crude estimates are made based on measured pressure drops measured at the gas side inlet and outlet of the model. The actual unit is now under operation and heat transfer data is also presented in this paper.

2. THE COLD FLOW MODEL

Fig. 1 shows a view of cold flow visualization model setup (a) and the graphical description of the test model. The cold flow model test setup is built to demonstrate the characteristics of the particulate flow behavior, from which relevant operating conditions are to be derived, such as the pressure drop through the system, fluidization of particles, separation of particles, particle-surface interaction, etc. For visualization of the flow, the main test section is made visible by using an acrylic tube. The model has the same structure as the actual heat exchanger except that in an actual heat exchanger water jacket is arranged outside the present model. It is composed of a cyclone (lower section), sprouting fluidized bed (middle section) with internal heat transfer module and outer heat transfer surface. Air comes through the inlet of the cyclone and moves through the vertical pipe (the exhaust of the cyclone). Particles which are introduced into the system from the particle inlet are supposed to form a kind of sprouting fluidized bed inside the main tube, while interacting with the internal heat transfer tube module. With the present structure more heat transfer surface area could be secured while the particle fluidization and related problems such as clogging, separation and possible condensation during the transient states could be carefully considered. Visual observation of the particles behavior inside the system is made possible by adopting acrylic material for the outer pipe wall.

2.1 Experimental Setup

Fig. 2 shows a simple test apparatus for the present pressure measurement. The purpose of this cold flow model is to make sure the particles circulation while measuring data for pressure drop at different operating



(a). A view of the model (b). Model description

Fig 1. Cold flow visualization model

conditions such as changing air flow rates and particle sizes. Fig. 2 shows the diagram of the test apparatus to measure the pressure drop. A fan (0.6 kW, Max. Pressure of 352 mmAq, Max. flow rate of 600 m^3/h) with a damper controls the air flow rate, which is measured by an orifice flow meter located at the downstream of the fan. To measure the pressure drop through the heat exchanger module the differential pressures between the gas inlet (P1) and the gas outlet (P2) are measured as well as the differential pressure from the orifice. Measured pressure drop through the orifice is converted to the equivalent gas flow rate. The gas flow is driven by a radial fan located downstream of the damper. Particles for circulation are weighed ahead of feeding and are fed through the ball valve on the upper part of the cold flow system. These particles fall into the middle part of the system where fluidized bed develops with the rising air flow from the internal pipe from the lower cyclone. The rising air flow with particles not only interacts with the outer surface but also with the internal heat transfer module (wound tube lines).

2.2 Experimental Procedures and Particles Ingestion

The experiments are to be performed to measure the pressure loss at different air flow rates with and without

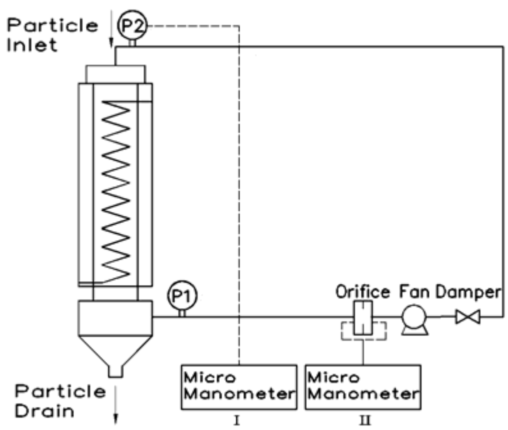


Fig 2. Test apparatus

particles. From this experiment, the system pressure loss data is to be obtained by measuring pressures at the gas inlet and gas outlet, respectively. For the pressure measurements, a micro-manometer (FCO-12, 0-199mmAq) is used. Particles used are glass beads (S.G. 2.62) which ranges from 150 microns to 800 microns in diameter, and is grouped into four size groups, that is, 150-250, 250-425, 425-600, 600-800 microns, respectively. Visual observation is also important in this process to check whether the fluidization and the trajectories of individual particles are appropriate or not. Through this study, the minimum and the maximum air flow rates are to be identified for individual particle size group. The experiments are to be performed to measure the pressure loss at different air flow rates with and without particles.

2.3 Calculation of the Air Flow Rate

The flow rate of the flue gas which is measured by using D and D/2 tap orifice downstream of the heat exchanger unit can be obtained by

$$Q = C_d A_t \left[\frac{2(P_1 - P_2)/\rho}{1-\beta^4}\right]^{1/2} \tag{1}$$

with the measured pressure difference between the upstream and downstream of the orifice. In the present case, a pipe of 103 mm inner diameter (D) and an orifice plate with diameter ratio (β=d/D) of 0.50 are used. The density at high temperature is obtained from the ideal gas law. Here the dimensionless discharge coefficient C_d for D and D/2 taps orifice [8] is known to be

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

with

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

where the values of empirical constants F_1 and F_2 are 0.4333 and 0.47, respectively.

3. PRESSURE DATA

3.1 Simulated Flow Pattern and Pressure Distribution

Fig. 3 shows the flow pattern of the air flow and the pressure distribution obtained from our preliminary simulation results. For the present analyses, a general purpose thermal-fluid analyzer, STAR-CD [9], is used. This software is based on the finite volume method and equipped with several turbulence models including k-ε high Reynolds number model, which is used for the present simulation. As it can be seen from the figure, at the lower part of the heat exchanger model, which is virtually a cyclone, the flow is basically vertical in nature and the resulting uprising flow through the internal pipe shows the maximum flow velocity. When the system is

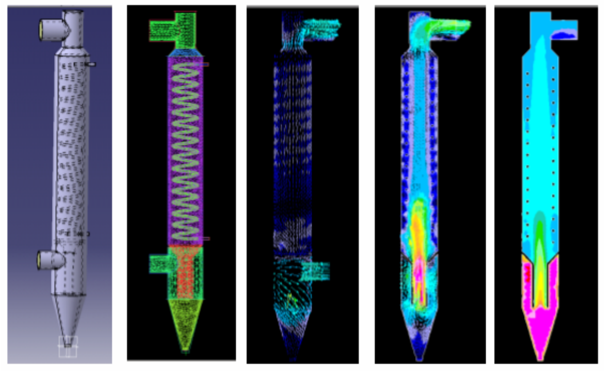


Fig 3. Numerical simulation results

to be designed, this uprising velocity should be high enough to move particles upward in order not to have falling particles through this region. Once fluidized, particles are supposed to move up, but not supposed to escape the system, which sets the flow condition requirement in the flow system design. Uprising flow after the maximum velocity region are exposed to a diffuser like region, at which the vertical component of the flow velocity become more dominant due to the abrupt expansion of the passage causing the reduction of the axial velocity component. Also, the pressure field shows that appreciable pressure will be built in the lower part, resulting in significant pressure drop through the system. This three-dimensional numerical simulation results suggest the possible vertical motion of both the flow and the particles inside the main heat exchanger, which could be identified through the visual observation in most of the tested condition.

3.2 Baseline Pressure Loss versus Flow Rate (without particles)

Fig.4 shows a calibration curve for the air flow rate with respect to the pressure difference from the orifice, ΔP_{orif}. Under the present set up, the air flow rate could be changed from 0.0245 m^3/s to 0.0641 m^3/s with ΔP_{orif} from 18.9 mmAq to 131.7 mmAq. (3)

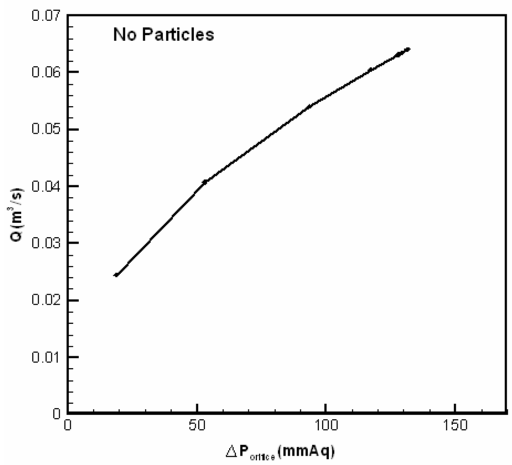


Fig 4. Air flow rate versus ΔP_{orif} (no particle)

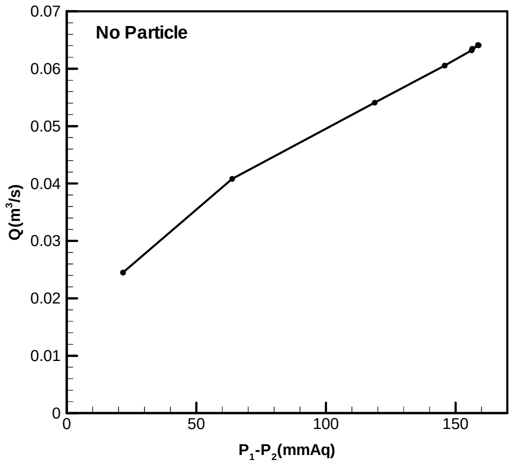


Fig 5. Pressure drop (P_1-P_2) versus Q (no particle)

Fig. 5 shows the pressure loss, P1-P2, through the cold flow model of the heat exchanger, from which one can notice that the maximum pressure loss was found to be 158.8 mmAq at the flow rate of 0.0641 m^3/s. According to the measured data, the pressure increases with flow rate increase.

3.3 The Effect of Particle Ingestion on the System Pressure Loss (with particles)

Once the base line system behavior is understood, the effect of ingested particles to the pressure loss is evaluated. For this test, a nominal air flow rate of 0.0641 m^3/s is selected and fixed corresponding a specific damper setting. Four different particle size groups (glass beads) are considered, that is 150~250 µ m, 250~400 µ m, 400~600 µ m and 600~800 µ m, respectively. Fig. 6 shows the measured pressure loss for different particle size groups with the amount of ingested particles as a parameter. As the particle mass increases step by step (by 200grams), the pressure drop in general increase with ingested particle mass, but the amount of increase were not appreciable except for the particle group III.

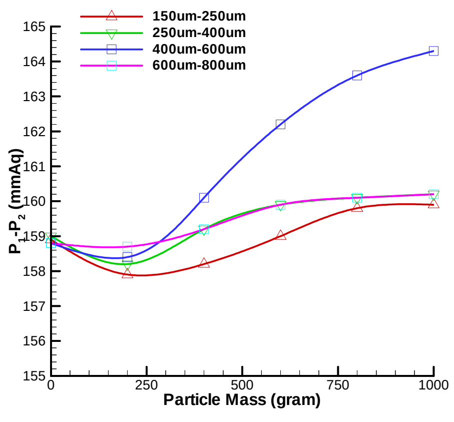


Fig 6. Pressure drop versus particle mass
(Q = 0.0641 m^3/s)

Through the visual inspection, particles rotating motion could be found for all the cases, but with smaller particle groups, I and II, particles tend to escape the system, while larger particles, IV, circulates only at the lower part of the heat exchanger surface. Particle group III is unique in a sense that particles are massively circulating with good distribution pattern. It seems that the pressure loss increase is responsible for the massive fluidization through the internal vertical pipe. One unusual phenomenon could be observed during the pressure measurement, which is associated with the decrease in the pressure loss with the increased ingested particle mass. The reason for this decrease unfortunately can not be explained clearly at this point.

4. PERFORMANCE TEST APPARATUS

Fig. 7 shows the third generation model of the heat exchanger and Fig.8 shows the schematic diagram of performance test apparatus for a single riser NDFB heat exchanger. A household boiler is used to supply a constant flow rate of hot gas to the inlet of the vertically oriented heat exchanger. About 800^0C of hot gas could be supplied to the heat exchanger inlet by eliminating the heat transfer enhancement device. The hot gas supplied from the combustor (boiler) enters the lower part of the heat exchanger to transfer heat energy to cooler part (water) through the heat transfer surfaces. The gas flow rate is measured by the orifice installed at the downstream of the heat exchanger and is controlled by a 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.

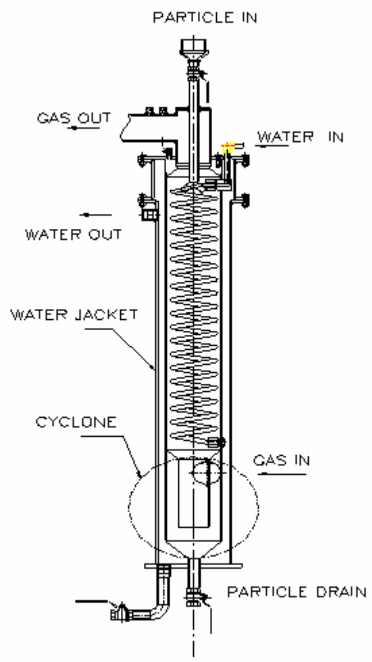


Fig 7. Schematic of the third generation model of the heat exchanger

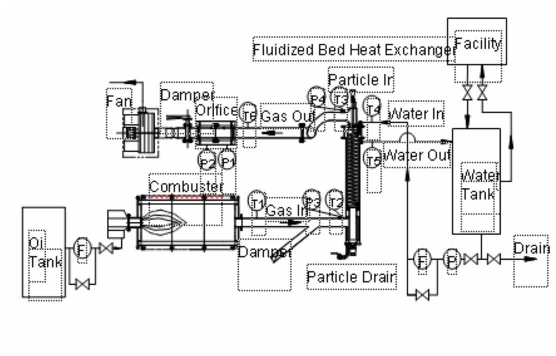


Fig 8. Schematic diagram of fluidized bed heat exchanger test apparatus.

Feed water for the heat exchanger is supplied from the makeup water line of the boiler. The city water is pre-heated passing through the combustor (boiler) along the wound tube then enters the inlet of the NDFB heat exchanger. The water flow rate is measured using a digital flow meter (OVAL) right upstream of the heat exchanger inlet. The temperature is measured at six locations (from T1 to T6) including the inlet and the outlet of the heat exchanger by using K-type thermocouples. For the measurement of temperature and pressure a 30 channel digital recorder, DA-100 (Yokogawa), is used.

4.1 Heat Exchanger Performance Analysis

Total heat transferred can be expressed as

$$q = \rho_g Q_g c_{pg} p_g \Delta T_g \quad \text{(Gas side)} \tag{4}$$

$$q = \rho_W Q_W c_{p_W} \Delta T_W \quad \text{(Water side)} \tag{5}$$

The overall heat transfer coefficient U of the heat exchanger is related as

$$q = UA\Delta T_m \tag{6}$$

where ΔT_m is the log mean temperature difference defined as

$$\Delta T_m = \frac{(T_{g,e} - T_{w,e}) - (T_{g,i} - T_{w,i})}{\ln\left[(T_{g,e} - T_{w,e})/(T_{g,i} - T_{w,i})\right]} . \tag{7}$$

Table 1: Temperature and pressure measurement locations

Channel	Measured Location
T1	Gas-burner outlet
T2	Gas-heat exchanger inlet
T3	Gas-heat exchanger outlet
T4	Water-heat exchanger inlet
T5	Water-heat exchanger outlet
T6	Gas-orifice
$\Delta P_{orifice}$	Orifice (P1-P2)
$\Delta P_{H.E.}$	Heat Exchanger (P3-P4)

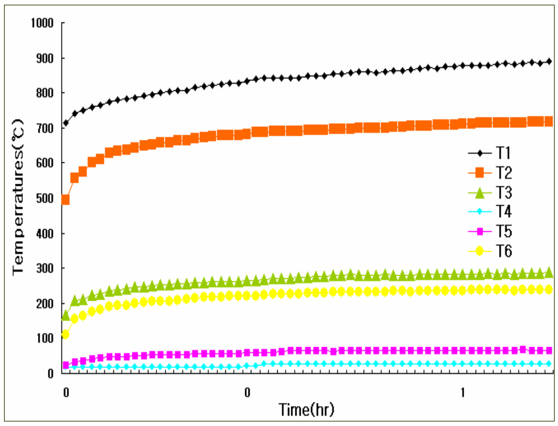


Fig 9. Temperature variations with respect to time

For the present case, $T_{g,e}$ =*T3*, $T_{g,i}$ =*T2*, $T_{w,e}$ =*T5*, $T_{w,i}$ = *T4* (See Fig. 8).

4.2 Heat Transfer Results

Fig. 9 shows typical temperature variations with time of the no-distributor-fluidized-bed heat exchanger. For a water flow rate of 0.135 kg/s and gas flow rate of 0.0587 kg/s, the heat exchanger was operated about 1hour and 20 minutes. The temperature difference in heat exchanger was about 428.5 ℃ and water temperature rise by 39.5 . The overall performance of the heat exchanger is shown in table 2.

5. CONCLUSIONS

In this paper, a no-distributor-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 applied to the heat recovery from waste gases from furnaces, boilers and incinerators with higher efficiency and reducing fouling related problems. Major feature of the proposed heat exchanger may be summarized as follows:

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

Table 2: Performance test analysis results

Item (Unit)	Values
m_w (kg/s)	0.135
m_g (kg/s)	0.0587
Gas Inlet (℃)	715.6
Gas Outlet (℃)	287.1
Water Inlet (℃)	26.4
Water Outlet (℃)	65.9
$\Delta T_{2\text{-}3}$ (℃)	428.5
$\Delta T_{4\text{-}5}$ (℃)	39.5
q_g (kcal/hr)	27.51
U(W/m^{2}K)	58.73
Efficiency(%)	81.0

(2) By using a cold flow model for the fluidized heat exchanger, particulate flow behavior could be identified with some quantitative data such as the pressure loss as well as visual observation.
(3) Optimum operating condition and the particle size group could be identified by performing the present experimental study. Present model showed unique behavior that the pressure loss decrease consistently when particles are ingested at specific amount, to say 200 grams 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.

6. ACKNOWLEDGEMENT

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8. NOMENCLATURE

Symbol	Meaning	Unit
A	heat transfer area	(m^2)
C_d	discharge coefficient	
d	orifice inner diameter	(m)
D	orifice outer diameter	(m)
m	mass flow rate	(kg/s)
P	pressure	(Pa)
ΔP	pressure drop along heat exchanger	(Pa)
q	rate of heat transfer	(W)
Q	flow rate	(m^3/s)
T	temperature	(K or ^{0}C)
ΔT_m	Log mean temperature difference	(K)
U	overall heat transfer coefficient	(W/m^2.K)
β	diameter ratio(=d/D)	(m/m)