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HEAT TRANSFER IN COUNTERCURRENT GAS-SOLID FLOW INSIDE THE VERTICAL PIPES

Sang-il Park*

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Abstract-Heat transfer characteristics of the countercurrent gas-solid flow inside verical pipes has been investigated with the shell and tube type heat exchanger. Sand particles having the average particle diameter of 1.0 and 1.7mm were used. The effect of gas velocity and sand paticle flow rate on the heat transfer rate and the pressure drop were examined. At room temperatures, the predicted pressure drop agrees well with the experimental results when the larger sand particles are used. The results shows that there exists an optimum sand particle flow rate at which the heat transfer rate become maximum. The increase in the heat transfer coefficient due to sand particles was obtained up to 62%.

Key Words: Gas-solid Flow, Two-Phase Countercurrent Flow, Fluidization, Heat Transfer Ehnancement

NOMENCLATURE

C_{DM} : Drag coefficient of gas-particle maixture
C_{DS} : Drag coefficient of single particle
D : Inner diameter of pipe
F : Force
f : Friction coefficient
g : Gravitational constant
L : Length of pipe
m : Mass flowrate
Δm_p : Effective mass of sand particles
Nu : Nusselt number (hD/k)
Pr : Prandtl number
Re : Reynold number $(\rho U d_p/\mu)$
t : Time
U : Velocity
z : Axis along the pipe
ε : Porosity
μ : Viscosity
ρ : Density
τ : Shear force

Superscripts

a : Accelerational
b : Bulk temperature
f : Fluid, frictional
g : Gas, gravitational
p : Particle
s : Surface, solid particle
w : Wall

1. INTRODUCTION

Numerous papers on the solid particle laden two phase flow has been published (Depew and Kramer, 1973). In the case of gases as heat transfer fluid, their thermal conductivities and heat capacities are relatively small. Farbar and Morley (1957) tried to overcome this limitation of gas by using the fluid maixture of gas and solid. This two-phase fluid can be applied when the gas temperature is high and the high heat transfer rate is required. In these studies, the loading density is relatively high and thus the pressure drop is generally high. However, studies on the countercurrent gas-solid flow are very limited. Arastoopour and Gidaspow (1979) investigated the countercurrent gas-solid flow in the vertical pipe and the pressure drop and the solid particle velocity profile are calculated with the assumption of the one-dimensional steady state and uniform termperature.

For the heat recovery in a raining bed exchanger, Boumehdi, et al. (1985) studied the heat transfer in the layers of horizontal tubes through which solids and gas flow countercurrently. In this case, the increase in the heat transfer coefficient at tube surface is relatively small and the resulting pressure drop is increased rapidly with the gas flow rate. The purpose of present work is to investigate the heat transfer phenomena of gas-solid two-phase flow inside the pipes to increase heat transfer rate with small pressure drop.

2. EXPERIMENTAL SET-UP

Experimental apparatus used in this study to measure the heat transfer coefficient and the pressure drop of the gas-solid two-phase flow is shown in Fig. 1.

A shell and tube type heat exchanger was installed and the sand particles were falling down through the vertical pipes. Above and below the heat exchanger, the perporated plates were inserted to get more uniform flow distribution of sand particles and gas. The diameter and length of the column was 300mm and 1.2m. The number of pipes of 45.5mm I.D. was 19. The hot sand particles were cooled in the sand cooler and transported to upper storage tank by bucket elevator. The pressure drop and the temperature difference across the heat exchanger were measured. At room temperature experiments, both 1.0mm and 1.7mm diameter sand particles were

*Korea Institute of Energy & Resources, Box 339, Deajoen 305—343, Korea

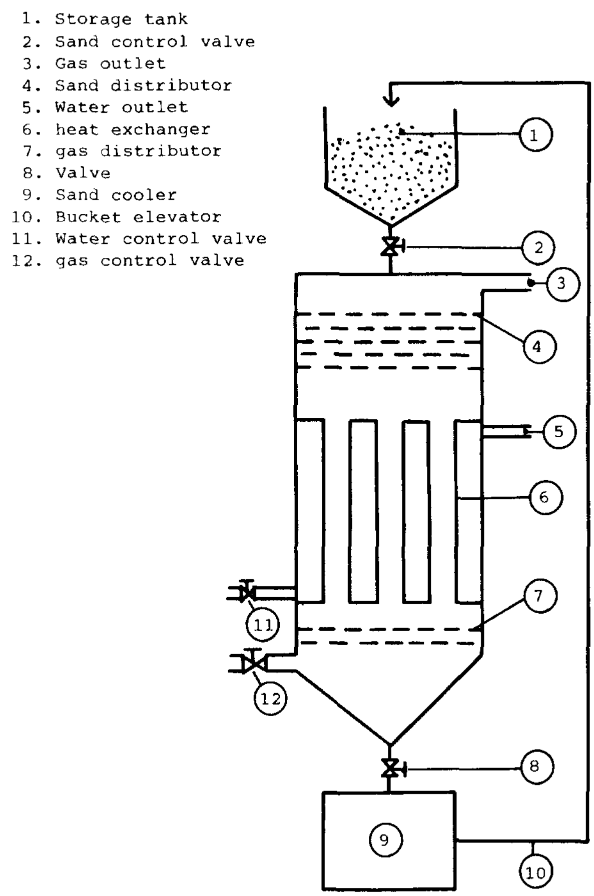

Fig. 1 Schematic diagram of experimental apparatus

used to examine their effect on the pressure drop. In high temperature experiments, only 1.7mm diameter sand particles were used because they maintained the ralateivly uniform distribution even at high gas velocities.

3. THEORETICAL ANALYSIS

3.1 Solid Particle Velocity Profile and Pressure Drop in the Vertical Pipe

When the sand particles fall through the vertical pipes, the velocity variation of the particles along the length of the pipe can be calculated by considering the balance of various forces acting on the sand particles. If the velocity distribution of the sand particles is assumed to be one-dimensional because the volume fraction of sand particles in the gas-solid mixture is small, the following equation can be obtained.

$$\Delta m_p \frac{du_p}{dt} = -F_d + F_g - F_f \tag{1}$$

where Δm_p is the effective weight of solid particles in the cross-sectonal area of the vertical pipe. F_d, F_g and F_f are the drag force, the gravitational force and the viscous force, respectively.

The drag force can be expressed as follows :

$$F_d = 0.75 C_{DS} \frac{\rho_f (U_f + U_p)^2}{(\rho_p - \rho_f) D_p} \Delta m_p \tag{2}$$

where C_{DS} is the drag coefficient on the single solid particle and can be obtained from the following equation.

$$\begin{aligned} &C_{DS} = 24/\mathrm{Re} && \mathrm{Re} < 0.2 \\ &C_{DS} = (24/\mathrm{Re})(1 + 0.15\mathrm{Re}^{0.687}) && 0.2 < \mathrm{Re} < 500 \\ &C_{DS} = 0.44 && 500 < \mathrm{Re} < 200{,}000 \end{aligned} \tag{3}$$

When the solid particles are fluidized, the drag coefficient should be modified as follows (Yang, 1976) :

$$C_{DM} = C_{DS} * \varepsilon^{-4.7} \tag{4}$$

where ε is the porosity which can be defined as follows :

$$\varepsilon = 1 - \frac{4 m_p}{(\rho_p - \rho_f) \pi D^2 U_p} \tag{5}$$

The gravitational force and the friction force can be expressed as follows :

$$F_g = g \Delta m_p \tag{6}$$

$$F_f = (2 f_s U_p^2 / D) \Delta m_p \tag{7}$$

where f_s is the friction coefficient of the solid particles at the wall. According to Konno and Saito(1969), it can be determined from the following :

$$f_s = 0.028 \sqrt{g D_p}\, U_p^{-1} \tag{8}$$

Therefore, the following equation on the velocity variation of the solid particles in the vertical pipe can be obtained.

$$\frac{dU_p}{dt} = -0.75 C_{DM} \rho_f \frac{(U_f + U_p)^2}{(\rho_p - \rho_f) D_p} + g + \frac{2 f_s U_p^2}{D} \tag{9}$$

The pressure drop of the gas-solid two-phase flow in the pipe can be obtained from the momentum equation as suggested by Arastoopour and Gidaspow(1979). However in the present study the properties of gas such as gas velocity are assumed constant because the porosity of the gas-solid mixture is very large. Therefore the resulting momentum equation is as follows ;

$$\frac{dP}{dz} + \frac{4\tau_w}{D} - \rho_p (1-\varepsilon) g + m_p \frac{dU_p}{dz} = 0 \tag{10}$$

Thus the expression for the total pressure drop can be obtained as the following equation by integrating Eq.(10) with respect to z.

$$\begin{aligned} \Delta P_T = & \int_0^L \rho_p (1-\varepsilon) g\, dz + \frac{2 f_g \rho_g U_g^2 L}{D} \\ & - \int_0^L 2 f_s \rho_p (1-\varepsilon) U_p^2 / D\, dz - \rho_p (1-\varepsilon) U_p(L)^2 \end{aligned} \tag{11}$$

3.2 Heat Transfer in Pipe

The empirical equations for the gas-solid two-phase flow in pipe have been reported only for the particle-laden flow and are applicable when the loading density is greater than 1.0. The heat transfer rate for the countercurrent gas-solid flow is not yet reported. Therefore in this study the well-known equations for the heat transfer coefficient applicable when only gas flows in pipe are selected to be compared with the experimental values. For the long pipe in which the turbulent flow is fully developed, the following two equations are frequently empoyed (Colburn, 1933 and Sider and Tate, 1936).

$$\text{Colburn's Eq.}: Nu = 0.023\ Re^{0.8}Pr^{0.33} \quad (12)$$
$$\text{Sider-Tate Eq.}: Nu = 0.027Re^{0.8}Pr^{0.33}(\mu_b/\mu_s)^{0.14} \quad (13)$$

When the length of the pipe is not long enough and the entrance effect is expected, the following equation can be used with the modification for the pipe length and diameter (Nusselt, 1931).

$$Nu = 0.036Re^{0.8}Pr^{0.33}(D/L)^{0.055} \text{ for } 10 < L/D < 400 \quad (14)$$

4. EXPERIMENTAL RESULTS AND DISCUSSION

In the room temperature experiments using the sand particles of diameter 1.0mm and 1.7mm, the pressure drop across the pipe was measured when the gas velocity increased up to 5m/s. The measured pressure drop is compared with the calculated values from Eq.(11) in case of 1.0mm diameter sands in Fig. 2.

In this Figure, the pressure drop increases almost linearly with the gas velocity up to about 3.2m/s and after that its rate of increase is reduced for both sand particle flowrates tested. This may indicate that above this gas velocity the distribution of sand particles is not even and thus the flowrates of sand particles through pipes become different each other. The predicted pressure drop continues to increase and thus above the deflection points the deviation between the calculated and the measured become large.

In Fig.3, the measured pressure drop is compared with the predicted values when 1.7mm diameter sand particles are used. This figure shows that the distribution of sand particles is maintained to be uniform and the pressure drop continues to increase even at high gas velocity. The predicted values agree well with the measured values with the maximum error about 0.4mmAq. More uniform distribution of larger sand particles may be due to their insensibility to the uneven gas flow distribution.

Therefore, in the heat exchanger performance test, only 1.7mm diameter sand particles were used. First, the flowrates and temperatures of gas mand water were measured in the experiment without sand particles. The the heat transfer coefficient at the gas-wall heat transfer surface was determined from these measured mvalues. The measured heat transfer coefficients are compared with the calculated values from Eqs.(12), (13) and (14) in Fig. 4.

This figure shows that the experimental results rather agree will with the predicted values from Eq.(14) and there exists the entrance effect because the lengthe of the pipes is not long enough. Thus, the effects of sand particles on the heat transfer rate were examined by comparing the experimental results with the predictions obtained from Eq.(14).

In the heat transfer experiments, the effect of sand particle flowrate was investigated when the gas velocity is maintained to be 7.8m/s. The sand particle flowrate was increased up to 132g/sec at which the loading density is 0.88. The inlet gas temperatures are maintained to be about 370℃. The measured pressure drop against the flowrate of sand particles is compared with the calculated values in Fig. 5.

This figure indicates that the measured pressure drop is much higher than the calculated value and this is quite different from the room temperature experimental results. The inlet gas velcoity is higher than the outlet gas velocity due to its higher temperature. The relatively samll sand

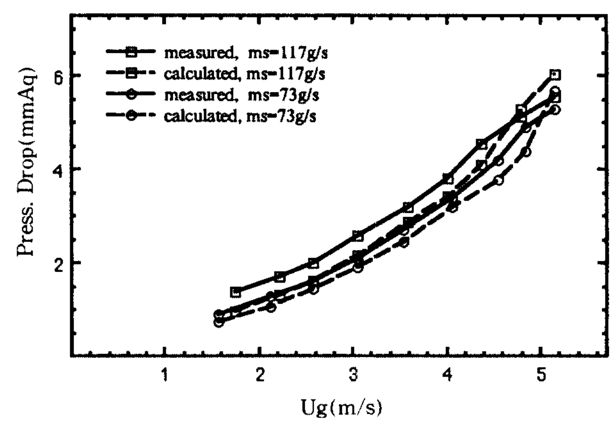

Fig. 3 Variation of pressure drop against gas velocity at room temperature ($D_p = 1.7$mm)

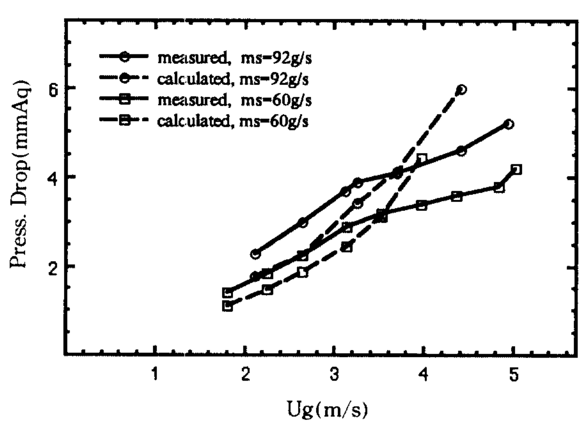

Fig. 2 Variation of pressure drop against gas velocity at room temperature ($D_p = 1.0$mm)

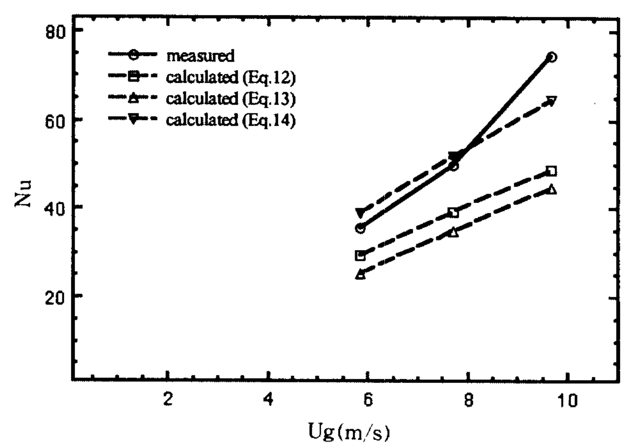

Fig. 4 Comparison of measured heat transfer rate with calculated values without sand

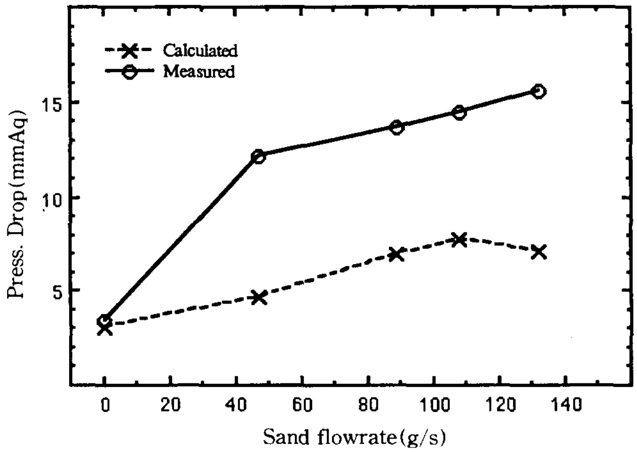

Fig. 5 Variation of pressure drop against sand flowrate

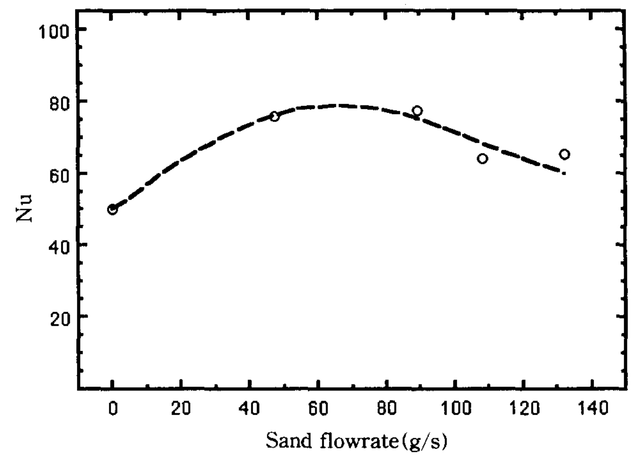

Fig. 6 Effect of sand flowrate on heat transfer rate

particles that their transport velocities are higher than the outlet gas velcoity and lower than the inlet gas velcoity will be remained in the pipes. Thus, this high pressure drop in the high temperature experiments is because these relatively small sand particles are fluidized in the pipes.

In Fig.6, the variation of the measured heat transfer rates against the sand particle flowrates is shown. This figure shows that the heat transfer rate increases rapidly even at low sand particle flowrate. This is because the small sand particles which remain in the pipes due to the velocity difference between the inlet gas and the outlet gas are fluidized near the wall of the pipes. The small fluidizing particles and the large falling particles will disturb the boundary layer. Thus, the heat transfer through this boundary layer will be enhanced. Also, it can be seen that there exists an optimum sand particle flowrate at which the heat transfer rate becomes maximum. The maximum incrase in the heat transfer coefficient due to sand particles is about 62%. At this optimum point, the loading density is about 0.5. This is quite different from the results of the particle-laden flow.

In the case of particle-laden flow, the temperature of solid particles can be assumed to be same as that of the surrounding gas. But, in the countercurrent gas-solid flow of present study, the falling particles are heated by the high temperature gas in the pipe. Also, the temperature of the particle may depend on the falling velocity of the particle and thus the smaller particles may be closer to the temperature of the surrounding gas because of their residence time is longer.

When the particle flow rate is greater than the optimum value, the particles absorb more heat from the gas and more solid particles of lower temperatures are fluidized near wall of the pipe. The gas temperature near the pipe wall is cooled by these lower temperature solid particles. Therefore, the actual temperature potential at the heat transfer surface is significantly reduced by the falling particles. This effect becomes more pronounced than the heat transfer enhancemence effect due to the particle fluidization at high sand flowrates. Also, the optimum particle flow rate may be changed by the particle inlet temperature. When the particle inlet temperature is higher than gas inlet temperature, the optimum particle flowrate may not exist.

Also, the effect of gas velocity on the heat transfer rates was examined and the experimental results are compared with the values calculated from Eq.(14) with the assumption

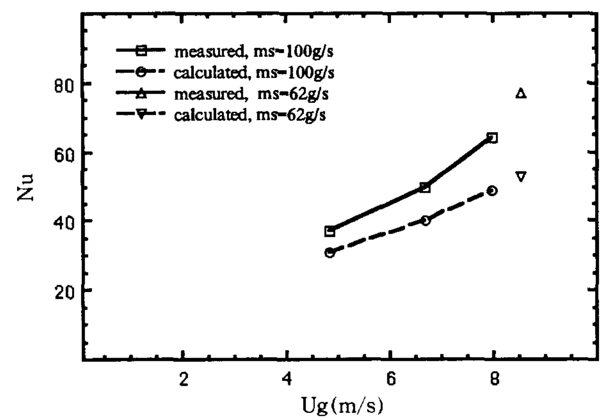

Fig. 7 Effect of gas velocity on heat transfer rate

of no sand particles in pipes in Fig. 7.

The measured heat transfer rates are increased with gas velcity and the trend of the calculated values is quite similar to that of the measured values. Also, the difference between the measured and the calculated increases with the gas velocity and this indicates that the effect of sand particles become more pronounced.

When the gas velocity and the sand particle flowrate are 7.7m/s and 89 g/s, respectively, Nusselt number is 77.3 and the pressure drop is 13.7mmAq. To get same Nusselt number without sand particles, the gas velocity should be increase to 14.2m/s. In this case, the pressure drop is calculated to be 9.2mmAq. However, when the fouling effect is consdiered and if the fouling factor is assumed to be 0.0035m^{2}K/W [Weierman, 1982], the gas velocity should be increased to 19.0m/s to obtaine the Nusselt number of 77.3. In this case the pressure drop will be about 16.4mmAq which is about 20% higher than 13.7mmAq. Thus, the enhancement method of the heat transfer rate with the countercurrent gas-solid flow in the verticle pipes can be employed especially when the severe fouling problem is expected.

5. CONCLUSION

From the room temperature experimmtal and theoretical analysis in this study, the pressure drop of the gas-solid flow in the vertical pipes is proved to be predicted when the solid flow is uniform. Also, from heat transfer experiments, it can be said that there exists an optimum sand particle flowrate to maximize the heat transfer rate in the vertical pipes. The heat transfer is enhanced by the fluidization of small particles near the pipe wall. The pressure drop of the gas-solid flow through the pipe is relatively small. The maximum increase in heat transfer rate obtained in this study is about 62% and the loading density at this point is about 0.5. The countercurrent gas-solid two-phase flow in the vertical pipes can be employed when the severe fouling problem is encountered.

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