Numerical Solution for Temperature and Flow Fields in Crossflow Moving Bed Heat Exchanger

Sang-il Park*

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The steady state solution of flow and temperature distributions in the crossflow moving bed heat exchanger using granular materials was obtained numerically. The effectiveness of crossflow moving bed was compared with that of crossflow heat exchanger with both fluids unmixed. The effect of bed geometry and solid particle size on the performance of moving bed was investigated. Also, the effect of thermal capacities of gas and solid particle and the gas flowrate was studied. The design data to estimate the heat exchanger performance of moving bed are given from the results of calculation in this study.

Key Words : Crossflow, Granular Material, Heat Exchanger, Moving Bed, Numerical Simulation, Temperature and Flow Fields

Nomenclature

Cp : specific heat
c^* : capacity rate ratio
d : diameter
g_c : gravitational constant
h : heat transfer coefficient
m_g : mass flowrate of gas
NTU : number of transfer units
p : pressure
T : temperature
u : gas velocity in x direction
v : gas velocity in y direction
x : coordinate
y : coordinate
y/x : ratio of bed height to length
α : void fraction
μ : viscosity

Subcript

g : gas
i : inlet
L : length of moving bed
p : particle
s : solid particle in bed

* Korea Institute of Energy Research 71-2 Jang-Dong, YouSeong-Gu, Daejeon, Korea

1. Introduction

Problems regarding the performance of a single pass cross-flow heat exchanger have been of continuous interest. Several papers were published to analyze the effect of various parameters. Roetzel and Neubert (1979) calculated the mean temperature difference to obtain an approximate explicit equation together with empirical coefficients. Yamashita et al. (1977) investigated the effect of variable heat transfer coefficient on the heat exchanger performance. In addition, Zhidov et al. (1985) calculated the temperature fields in the crossflow heat exchanger when the hydraulic and temperature condition was not uniform at the inlet.

The crossflow moving bed heat exchanger studied in this paper is a special type of crossflow heat exchanger. A moving bed is generally characterized by the movement of both solid particles and gas during thermal interaction. The number of papers related to the moving bed in crossflow is quite limited. Such crossflow type bed can be operated with small pressure drop during continuous removal of solid particles, e.g. deactivated catalyst pellets under steady state condition (Marb and Vortmeyer, 1988). Most of the crossflow moving beds reported in the literature are found in the petroleum industry (Sittig, 1952; Cha et al., 1973) and flue gas desulpherization plants (Knoblauch et al., 1985).

The performance estimation of the crossflow moving bed needs the calculation of flow and temperature fields within the bed. A theorectical and experimental investigation of crossflow moving bed reactor was performed by Marb and Vortmeyer (1988). But in their analysis of chemical reaction, the detailed information of the velocity variation in the bed was neglected.

The aim of this work is to estimate the effectiveness of heat exchanger by calculating both flow and temperature fields in crossflow moving bed under steady state. Also, the optimum geometry and solid particle size were investigated to obtain high thermal efficiency with relatively small pressure loss.

2. Model Equations and Boundary Conditions

The simplifying assumptions are made in the derivation of the governing differential equations as follows:

(a) Thermal properties of the solid are constant.

(b) Conduction through the solid and gas phases in the bed is negligible.

(c) Internal thermal resistance within the solid particle is negligible.

(d) constant inlet temperatures of the gas and solid phases are assumed.

On the basis of these assumptions, the following 2-dimensional steady state model equations are derived for the geometric configuration as shown in Fig. 1.

Energy balances for two-phase model:

gas phase;

$$(\rho uCp)_g\frac{\partial T_g}{\partial x}+(\rho vCp)_g\frac{\partial T_g}{\partial y}=h_{gs}(T_s-T_g) \quad (1)$$

soid phase;

$$(\rho vCp)_s\frac{\partial T_s}{\partial y}=h_{gs}(T_g-T_s) \quad (2)$$

Mass conservation for gas phase:

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}=0 \quad (3)$$

Momemtum conservation for gas phase:

$$\frac{dP}{d\,x}g_c=150\frac{(1-\alpha)^2}{\alpha^3}\frac{\mu\vec{v}}{d_p^2}+1.75\frac{(1-\alpha)}{\alpha^3}\frac{\rho\vec{v}^2}{d_p} \quad (4)$$

Equation of state for gas phase:

$$p=\rho RT \quad (5)$$

The boundary conditions used in the analysis are

$$T_g(x=0)\,T_{g,i} \quad (6)$$

$$T_s(y=y_L)=T_{s,i} \quad (7)$$

$$\int(\rho u)_g dy=\dot{m}_g \quad (8)$$

$$v_g(y=0)=v_g(y=y_L)=0 \quad (9)$$

$$p(x=0)-p(x=x_L)=constant. \quad (10)$$

In the momemtum Eq. (4), the relationship between the pressure drop and gas velocity suggested by Ergun (1952) has been employed instead of the general expression in order to adequately describe the velocity field in the moving bed. In this expression, total energy loss in beds was treated as the sum of viscous and kinetic energy loss which can applies to Forchheimer flow region. With this, the above model equations permits the resolution of the two-dimensional gas

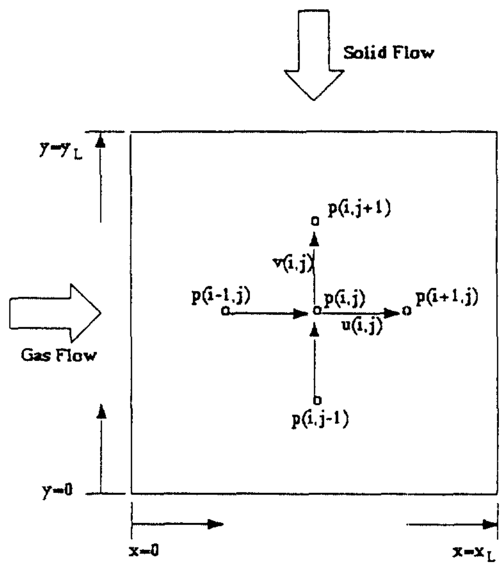

Fig. 1 Model geometry of crossflow moving bed

flow within the bed that may be induced by the pressure difference between the inlet and outlet surfaces and by local temperature gradient. Thus Eq. (4) rather accurately describes the fluid dynamic phenomena in the crossflow moving bed heat exchanger in the usual practical range.

With the constraint by Eq. (8), the total mass flow rates of gas both at inlet and outlet are maintained to be a predetermined value. Eq. (9) describes no gain or loss in gas flow rate. Eq. (10) indicates that the difference between the inlet and outlet gas pressures are maintained to be constant along the y-axis of the interfaces and, as a results of this constraint, the inlet and outlet gas flow rates become nonuniform.

3. Numerical Solution

The temperatures of gas and solid were solved numerically using the finite difference equations derived from Eqs. (1) and (2) with assumed values of other variables. The model geometry for numerical solution is also shown in Fig. 1. The gas density is calculated from Eq. (5) with known gas temperature and assumed pressure. Then, Eq. (3) was discretized to derive the corresponding finite difference equation and the local velocitiy components derived from Eq. (4) were substituted to this equation. As a result, a series of nonlinear equations were all coupled via the only one unknown at this stage, i.e. the local pressure. Subject to the assumed values of inlet and outlet pressures, these equations were solved by iteration to obtain the local pressures and, in turn, local velocities through Eq. (4). Subsequently, a cross-check was made to determine whether or not the earlier mentioned constraint, Eq. (8) was satisfied. If not satisfactory, the inlet and outlet gas pressures are modified using the error in the gas mass flowrates calculated in each iteration. The above procedures were repeated until the ratios of the error of calculated inlet and outlet mass flowrates of gas to the prescribed value were converged to within a preselected tolerance of about 0.001. Number of nodal points used in this study ranged from 30×30 to 60×60. The effect of the number of nodal points on the accuracy of solution was also investigated.

4. Results of Numerical Calculations

The effectiveness, as a function of NTU, for the crossflow heat exchanger with both fluids unmixed was calculated for comparison. The surface area of the solid particles in the moving bed was used in the calculation of NTU. The capacity rate ratio, c^*, for these calculations were selected to be 0.0, 0.5 and 1.0. The values of NTU were ranged from 0 to 10. By assuming the constant heat transfer coefficients, the effectiveness of moving bed heat exchanger was also calculated numerically. In Fig. 2, the effectiveness of the moving bed calculated from this study is compared with that of crossflow heat exchanger reported in Baclic (1978).

The difference in the effectivenesses from two studies is very small when NTU is less than 2 or c^* is equal to 0. The difference increases as NTU and c^* increase, which may be due to the increased temperature gradient of gas with increasing NTU and c^*. The increase in the gas temperature gradient causes the increase in Vg and thus the effectiveness of moving bed is reduced. The maximum difference in effectiveness of the two is about 3% in the calculated range shown in Fig. 2.

In Fig. 3, the inlet and outlet gas velocity distributions were compared for two cases ; First case is $Vg=0$ everywhere and second case is $Vg>0$. This is the main difference between the crossflow heat exchanger and the moving bed.

This figure shows that, in the case of $Vg=0$,

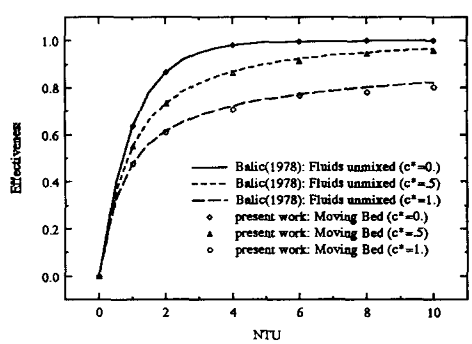

Fig. 2 Comparison of effectiveness of crossflow heat exchanger with that of moving bed

the variations of the inlet and outlet gas velocities are greater. Thus, the increase in Vg results in the reduction of the difference in Ug along the bed height. In the case shown in this figure, the gas temperature increases with bed height. In the higher temperature region, the U velocity may be increased due to reduced gas density and thus the greater pressure drop may be required. But, the pressure drop and the gas flowrate across the bed along the entire bed height were given. Thus, a fraction of gas in the region of higher temperature moves to the colder region. Also, the gas flowrate per unit area at the region of higher temperature is reduced.

In both cases shown in Fig. 3, the outlet gas velocities decrease as the bed height increases. This is mainly due to the differences in the local gas temperatures. As the gas temperature increases under the condition of the same pressure drop across bed, the gas density decreases and thus the gas velocity increases.

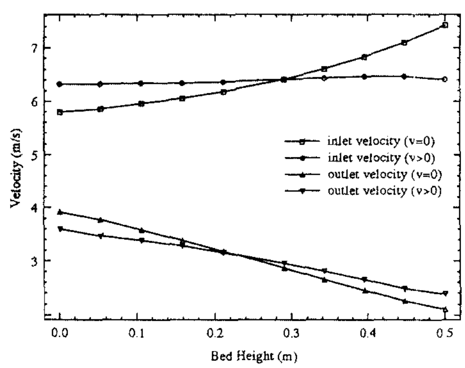

Fig. 3 Comparison of the inlet and outlet gas velocity distributions

Figure 4 shows the typical variation of Vg at 5 different values of y/h in the moving bed.

This figure indicates that V_g is zero at all of the boundaries and greater than zero at the interior region. This gas velocity is affected mainly by the gas temperature distribution and causes the decrease in the effectiveness of moving bed heat exchanger. When compared with the Ug, the ratio of the maximum values of Ug to Vg is about 0.12 in this typical case.

In real case, the heat transfer coefficient between the gas and the bed material varies with the gas velocity, as reported by Wakao et al. (1979). The thermal effeciency and the pressure drop of moving bed were calculated by changing the particle diameter and bed configuration(y/x). In Fig. 5, variations of the effectiveness with y/x are shown.

For each particle diameter, the efficiency decreases with increasing y/x. But, when y/x is less than 2 and the particle diameter is less than 0. 02 m, the rate of decrease is very small. This indicates that the Vg inside the bed is significantly increased and thus the efficiency is reduced.

The variations of pressure drop against y/x are calculated and the results are shown in Fig. 6. This figure shows that the pressure drop is decreased with increasing y/x and the rate of decrease in pressure drop is great when y/x is less than about 5. This indicates that it is economically practical to keep y/x to be greater than about 5 to avoid the excessive pressure drop while maintaining reasonable efficiency.

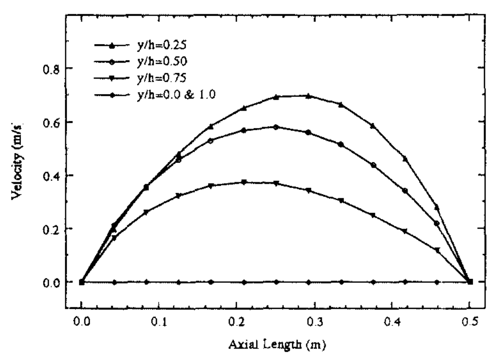

Fig. 4 Variations of Vg in the moving bed

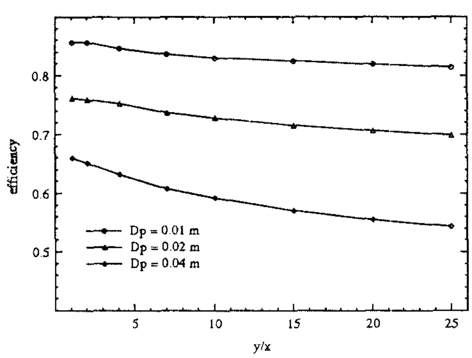

Fig. 5 Efficiency as a function of y/x

Figure 7 shows the variation of the efficiency with the pressure drop that was calculated for various particle diameters. It is readily observable that the efficiency increases as the pressure drop increases. But the rate of increase in efficiency is reduced with increasing pressure drop. In the tested range, the efficiencies of bed having fine particle size are greater than those having large particle size. Also, for given pressure drop across the bed, the fine particle size results in higher efficiency than the large particle size. This indicates that, to maintain high thermal efficiency, it is better to use small particle size and the bed configuration of large y/x.

Next, the effect of thermal capacitance ratio of the bed material to the gas on the bed thermal efficiency was investigated. The thermal capacitance ratio varies with the thermal capacities and the mass flowrates of bed material and gas. The effect of flowrate of bed material on the moving bed performance is as much as that of the thermal capacity of bed material.

The effect of the thermal capacity of bed material on the thermal efficiency was investigated when the particle diameter equals to 0.02 m. The values of y/x considered are 4 and 15 which can be typical values in practical use. It is shown in Fig. 8 that the efficiency increases almost linearly and its increasing rate is great when the capacitance ratio is less than 1.0. Also, in this range, the effect of y/x is relatively small. For the capacitance ratio over 1.0, the rates of increase are reduced, whereas the effect of y/x becomes significant.

Figure 9 shows the calculated results to investigate the effect of thermal capacity of gas. This figure indicates that the general trend of increasing efficiency is similar to that for the variation of thermal capacity of bed material. But the effect of y/x is significant only when the capacitance ratio is in the range from 1.0 to 2.5. Also, at high values of capacitance ratio, the efficiency approaches 1.0.

The thermal capacitance ratio can also be changed by the gas flowrate. The effect of gas flowrate was studied and the variations of ef-

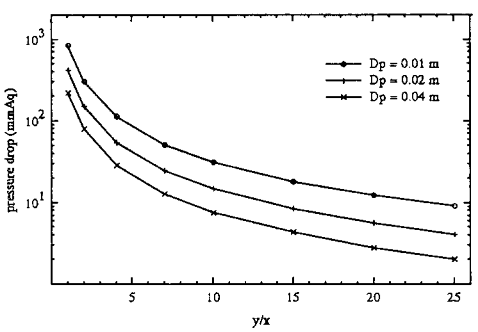

Fig. 6 Pressure drop as a function of y/x

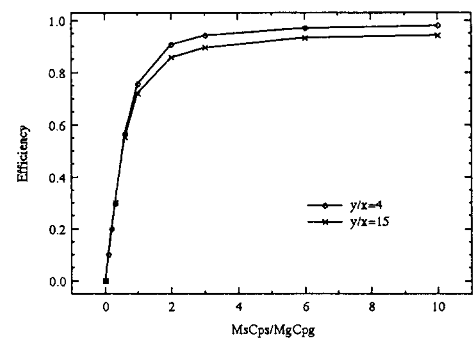

Fig. 8 Effect of thermal capacity of bed material

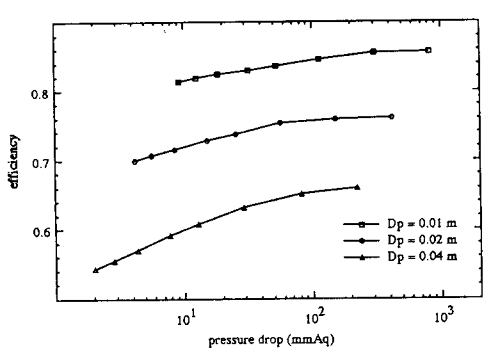

Fig. 7 Efficiency as a function of pressure drop

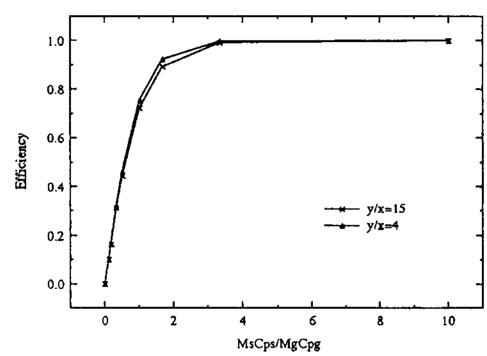

Fig. 9 Effect of thermal capacity of gas

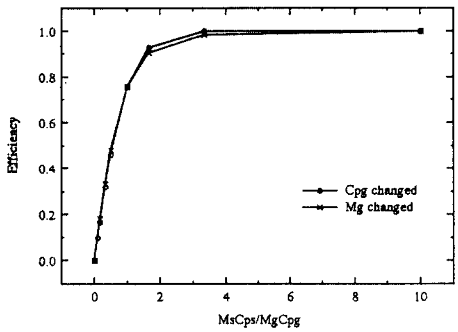

Fig. 10 Effect of gas flowrate

ficiencies are compared in Fig. 10. As seen in Fig. 10, as the gas flowrate is increased, the gas velocity through the bed also increases and thus the heat transfer coefficient is increased. The difference between the two cases, shown in Fig. 10, represents the effect of resulting changes in the heat transfer coefficient. Thus, the efficiencies in the case that the gas flowrates vary are higher when the thermal capacitance ratio is less than 1.0 and lower when it is greater than 1.0.

5. Conclusions

The performance of moving bed heat exchanger was calculated numerically and the following conclusions were drawn :

(1) The efficiency of moving bed is less than that of the crossflow heat exchanger with both fluids unmixed and the difference of the two becomes significant as NTU and c^* increase.

(2) To obtain high heat exchanger performance, it is better to use finer particle size and the bed configuration having greater value of y/x.

(3) As the thermal capacitance ratio increases, the efficiency increases rapidly for the lower values of thermal capacitance ratio and the rate of increase is greatly reduced thereafter. Thus it is desirable to operate the moving bed heat exchanger at the thermal capacitance ratio that is greater than 1.0.

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