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Effect of particle loading on heat transfer enhancement in a gas-solid suspension cross flow*

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Abstract: Heat transfer between gas-solid multiphase flow and tubes occurs in many industry processes, such as circulating fluidized bed process, pneumatic conveying process, chemical process, drying process, etc. This paper focuses on the influence of the presence of particles on the heat transfer between a tube and gas-solid suspension. The presence of particles causes positive enhancement of heat transfer in the case of high solid loading ratio, but heat transfer reduction has been found for in the case of very low solid loading ratio (M_s of less than 0.05 kg/kg). A useful correlation incorporating solid loading ratio, particle size and flow Reynolds number was derived from experimental data. In addition, the k-ε two-equation model and the Fluctuation-Spectrum-Random-Trajectory Model (FSRT Model) are used to simulate the flow field and heat transfer of the gas-phase and the solid-phase, respectively. Through coupling of the two phases the model can predict the local and total heat transfer characteristics of tube in gas-solid cross flow. For the total heat transfer enhancement due to particles loading the model predictions agreed well with experimental data.

Key words: Multiphase flow, Heat transfer, Particle loading
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INTRODUCTION

The heat transfer characteristics of gas-solid two-phase flow are of great interest in many industrial processes, such as circulating fluidized bed process, pneumatic conveying process, chemical process, drying process, etc. Interest in the influence of particles in solid suspension on convective heat transfer has increased since the 1950s. The heat transfer problem in the pipelines of pneumatic conveying systems was investigated in numerous experimental studies (Farbor et al, 1957; Wilkinson et al., 1967) and theoretical studies (Michaelides, 1986; Kuo et al., 1988; Han et al., 1991). The mechanism of heat transfer enhancement due to particles had been studied (Kurosaki et al., 1990; Yoshida et al., 1990) for the case of impinging jet flows. The influence of particles on heat transfer for the case of a tube or tube bank in cross flow had been investigated primarily in the dilute region of fluidized bed or circulating fluidized bed (George et al., 1982; Wood et al., 1980). Most of these studies showed that the solid loading ratio could alter the heat transfer between tubes and solid suspensions in the splash zone and dilute region.

However, it is difficult to research experimentally this heat transfer mechanism because of imprecise measurement of the solid loading ratio in circulating fluidized bed. So several researchers studied the effect of solid particles on heat transfer of tube in gas-solid cross flow by controlling more precisely the solid loading ratio. The heat transfer characteristics in suspension cross flow was studied experimentally by Woodcock et al. (1966) for an in-line tube bank where the solid loading ratio was in the range of 2 to 8 (kg/kg), and by Murray et al. (1991) for a staggered array of tubes. Murray (1994a, b) investigated the mechanisms associated with the enhancement of convective heat transfer over the front of a tube in a particulate cross flow. Unfortunately, there is little research work for numeri-

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cal simulation especially dealing with the total heat transfer characteristics of a tube in a particulate two-phase flow, especially the numerical simulation.

In this work, the heat transfer enhancement for a gas-particle cross flow around a tube was studied experimentally and theoretically. A model was developed to simulate this enhancement with the coupling of the gas phase and solid phase and particle-to-wall conduction. In addition, the numerical simulation results were also compared with the experimental data.

NUMERICAL PROCEDURE

This section summarizes the main features of the numerical procedure developed to calculate the flow and heat transfer of the particulate two-phase cross flow. The two-equation k-ε Model and the Fluctuation-Spectrum-Random-Trajectory Model (FSRT Model) were used to simulate the flow and heat transfer of gas-phase and solid-phase respectively.

For steady, two-dimensional, incompressible, constant property, turbulent flow the basic equations can be written in the following general form:

$$\frac{\partial}{\partial x}(\rho U\phi)+\frac{\partial}{\partial y}(\rho V\phi)=\frac{\partial}{\partial x}\left(\Gamma_\phi\frac{\partial\phi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\Gamma_\phi\frac{\partial\phi}{\partial y}\right)+S_\phi+S_{\mathrm{p}}^{\phi},\tag{1}$$

where ϕ is a universal parameter (standing for u, v, k, ε, etc.), Γ_ϕ is the relevant coefficient of turbulent diffusion, S_ϕ is the source term and S_{p}^{ϕ} is the particulate source term which represents the net flux of u, v, k, ε, etc. into the fluid phase due to the particle-fluid interaction.

The Lagrangian treatment of the particulate phase combined with the Eulerian approach for the fluid has been used by many researchers, especially by Schuh et al. (1989) for the numerical calculation of particle-laden gas flows past tubes. In this paper, the Fluctuating -Spectrum-Random-Trajectory model (FSRT model) proposed by Cen et al. (1992) is used to consider the effect of turbulence on particle dispersion.

For the particle trajectory calculation, particularly for the analysis of dilute particle-laden flows, it was assumed that effects of virtual mass, Bassett and Magnus forces and particle collisions can be neglected. With these assumptions, the particle motion equation is:

$$M_{\mathrm{p}}\frac{\mathrm{d}\boldsymbol{V}_{\mathrm{p}}}{\mathrm{d}t}=C_{\mathrm{D}}\rho_{\mathrm{g}}A_{\mathrm{p}}\frac{|\boldsymbol{V}_{\mathrm{g}}-\boldsymbol{V}_{\mathrm{p}}|}{2}(\boldsymbol{V}_{\mathrm{g}}-\boldsymbol{V}_{\mathrm{p}})+M_{\mathrm{p}}\boldsymbol{g},\tag{2}$$

where $\boldsymbol{V}_{\mathrm{p}}$ is the instantaneous velocity of particle-phase, $\boldsymbol{V}_{\mathrm{g}}$ is that of gas-phase, which can be separated into mean velocity $\bar{\boldsymbol{V}}_{\mathrm{g}}$ and fluctuating velocity $\boldsymbol{V}'_{\mathrm{g}}$, that is:

$$\boldsymbol{V}_{\mathrm{g}}=\bar{\boldsymbol{V}}_{\mathrm{g}}+\boldsymbol{V}'_{\mathrm{g}}.\tag{3}$$

The fluctuating velocity of the gas phase can be simulated by a random Fourier series:

$$\begin{aligned}u'_{\mathrm{g}}&=\sum_n R_1U_{\mathrm{m}i}\cos(i\omega_i t-R_2\alpha_i^u),\\ v'_{\mathrm{g}}&=\sum_n R_3V_{\mathrm{m}i}\cos(i\omega_i t-R_4\alpha_i^v),\end{aligned}\tag{4}$$

where R_1 and R_2 are random values with Gaussian distribution, ω_i is the turbulent fluctuating frequency, α_i^u and α_i^v are the fluctuating initial phase angles, and $U_{\mathrm{m}i}$ and $V_{\mathrm{m}i}$ are the amplitudes determined by the turbulent fluctuating spectrum.

The drag coefficient C_D in a turbulent flow is obtained from the following expression (Fan et al., 1987):

$$C_{\mathrm{D}}=\frac{24}{Re_{\mathrm{p}}}(1+0.19Re_{\mathrm{p}}^{0.62})\left[1+0.095\left(\frac{f_{\mathrm{b}}Re_{\mathrm{p}}}{\sigma}\right)^{0.0287}\right],\tag{5}$$

Moreover, the particle trajectory can be obtained by solving the following equations:

$$\begin{aligned}x_{\mathrm{p}}&=x_{\mathrm{po}}+(u_{\mathrm{p}}+u_{\mathrm{po}})\Delta t/2,\\ y_{\mathrm{p}}&=y_{\mathrm{po}}+(v_{\mathrm{p}}+v_{\mathrm{po}})\Delta t/2.\end{aligned}\tag{6}$$

Fig. 1 is a sketch of a single particle impinging on and rebounding from the surface of a tube. For determining the position of impact, a small region around the tube of thickness δ_{b} is proposed to be set to $0.005D$ (Schuh et al., 1989), where D is the diameter of the tube. When the particle first approached the surface, the solution algorithm is unaware of its presence and forces the particle to cross the surface. When this happens, the particle is returned to its previous position, the step size is decreased by a factor of 2, and calculating is retried. This procedure is repeated until the particle falls into the δ_{b} region. Assuming that the velocity of the particle remains constant within δ_{b}, the velocity and direction of the rebounding particle can be obtained by the following formulas

$$\frac{|\bar{V}_{p2}|}{|\bar{V}_{p1}|}=(1.0-0.02108\alpha_1+0.0001417\alpha_1^2)\cdot\sqrt{\frac{1+\operatorname{ctg}^2\alpha_2}{1+\operatorname{ctg}^2\alpha_1}},$$

$$\alpha_2=\operatorname{ctg}^{-1}\left[\left(\frac{0.95+0.00055\alpha_1}{1.0-0.02108\alpha_1+0.0001417\alpha_1^2}\right)\operatorname{ctg}\alpha_2\right], \quad (7)$$

where $\bar{V}_{p1}$ and $\bar{V}_{p2}$ are the velocities of the particle before and after collision with the tube, α_1 and α_2 are angles of collision and rebounding, respectively.

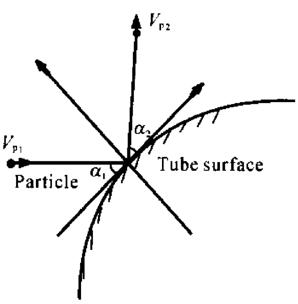

Fig. 1 A particle impinging on and rebounding from the tube

The heat conduction between impacting particles and a tube wall was studied by Sun et al. (1988) who found that the total energy exchange on impinging could be given by

$$q_c=\frac{A_{cmax}(T_w-T_{p1})\pi t_c)^{-1/2}}{(\rho_p C_{pp} K_p)^{-1/2}+(\rho_w C_{pw} K_w)^{-1/2}}, \quad (8)$$

where t_c is the contact time, (T_w-T_{p1}) is the initial temperature difference between the particle and the wall, C is the thermal capacity and K is the thermal conductivity, A_{cmax} is the maximum contact area, $A_{cmax}=0.87\pi r_c^2$, r_c is the contact radius. Subscripts p and w refer to the particle and tube wall.

EXPERIMENTAL FACILITIES AND PROCEDURES

The heat transfer measurements were carried out in a closed-loop circulating wind tunnel with a particle feed and separation system as shown in Fig. 2.

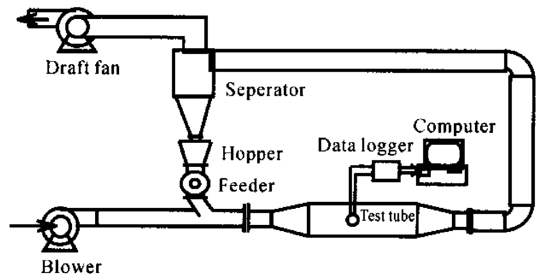

Fig. 2 Schematic of the experimental facility

Air was drawn into a 200×200 mm pipeline where the flow rate was monitored by a swing type air meter. The solid particles used in this investigation were ash with mean diameter of 65 micron and sands with mean diameter of 120 micron and 328 micron respectively. The particles were introduced into the main flow using a rotary valve rotation rate controlled by an electric motor. The solid loading ratio could be up to 4.5 kg per kg air. A cyclone was used to separate the solid particles from the air and return them to a storage hopper connected to the feeder. The total heat transfer rate was measured using an electrically heated probe.

RESULTS AND DISCUSSION

It is well known that the solid loading ratio is one of the main factors that have great influence on the heat transfer between surface and suspension. Fig. 3 shows the effect of the solid loading ratio (where $M_s<0.05$ kg/kg) for particle size of 63 μm, 150 μm and 385 μm. Fig. 4 shows the effect of the solid loading ratio (where $0.05<M_s<2.5$ kg/kg) for particle size of 63 μm and 385 μm. Heat transfer was evidently enhanced with increasing solid loading ratio, but the presence of solid particles reduced instead of enhanced the heat transfer in the range of very low solid loading ratio (about $M_s<0.015$ kg/kg). This case is similar to that of suspension heat transfer in pipelines (Farber et al., 1957). It is possible that the presence of particles in a gas flow reduces the level of turbulence as a result of eddy-particle interactions (Hetsroni, 1989). Analysis of this negative effect (Gao et al., 1996) using particle critical value θ showed that this negative effect was mainly due to turbulence suppression by particles, with consequent lowering of the heat transfer rate.

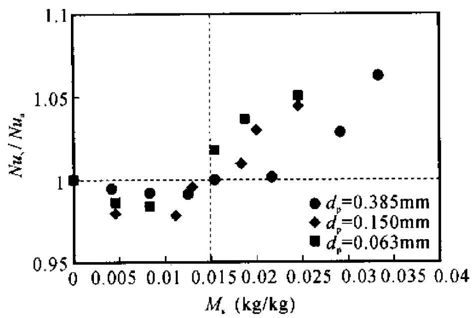

Fig. 3 Variation in Nusselt number with the solid loading ratio (where $M_s < 0.05$ kg/kg)

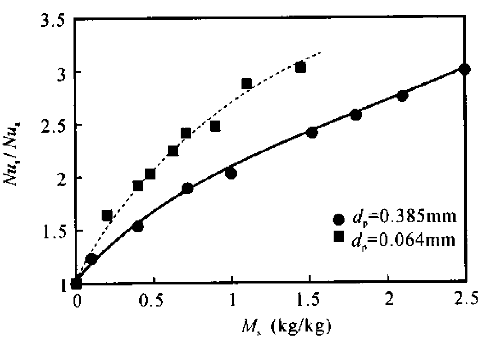

Fig. 4 Variation in Nusselt number with the solid loading ratio (where $0.05 < M_s < 2.5$ kg/kg)

Fig. 5 shows the effect of particle size on Nusselt number for a Reynolds number of 12 000 and a solid loading ratio of 1.0 kg/kg. It can be seen that the enhancement increases with reducing particle size. The smaller the particles are, the larger the number of particles in a unit volume are for the same solid loading ratio. Thus the heat transfer enhancement from the reduction in the boundary layer thickness becomes large as the number of particles passing through the boundary layer increases. As mentioned in Murry (1994a), the thermal response time of small particles becomes shorter than that of large particles. Thus the suspension will have higher thermal capacity for small particles and more stored thermal energy is transported out of the heated fluid zone by rebounding particles. For very small particles, where $\tau_t \ll \tau_e$, Owen (1969) suggested that the rate of turbulent energy dissipation is increased, compared to single-phase flow, by a ratio of $(1 + M_s/\rho_g)^{1/2}$. (The relaxation time, τ_t, can be written as $d_p^2\rho_p/18\upsilon\rho_g$; the characteristic time scale for the eddies, τ_e, is their characteristic size divided by characteristic velocity, l/u_g'.) For larger particles with $\tau_t > \tau_e$, Owen (1969) suggested that the turbulent fluctuations in the presence of the particles should decrease as $(1 + M_s/\rho_g \cdot \tau_t/\tau_e)^{-1/2}$. Therefore, heat transfer reduction due to turbulence suppression increases for larger particles. All of the above factors induce the trend of increasing enhancement of heat transfer with decrease in particle size.

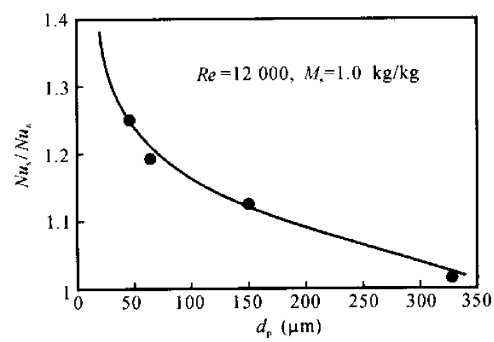

Fig. 5 Variation in Nusselt number with particle size

Reynolds number is also one of the factors which influence the heat transfer enhancement. In Fig. 6, the effect of varying Reynolds number

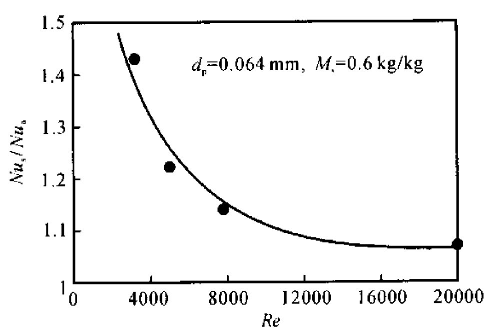

Fig. 6 Variation in Nusselt Number with flow Reynolds number

is examined for the particle size of 64 μm and solid loading ratio of 0.6 kg/kg. In this case, a trend of increasing enhancement with decreasing Reynolds number can be found. According to the changes in the boundary layer characteristics in the presence of particles, the boundary layer around the tube is thinner for higher Reynolds number and the residence time of particles passing through the boundary layer is smaller than that for lower flow Reynolds number. Hence, heat transfer enhancement reduces for high Reynolds number with reducing disturbance of particles.

The effect of main factors, such as solid loading ratio, particle size and flow Reynolds number, have been investigated above. Experimental data yielded the following useful correlation for the heat transfer of a tube in suspension cross flow:

$$\frac{Nu_s}{Nu_a} = 1 + 4.1 Re^{-0.31} d_p^{-0.233} M_s^{0.72}, \quad (9)$$

where $Re = 3000 - 20{,}000$, $d_p = 64 - 328$ μm, $M_s = 0.05 - 2.5$ kg/kg.

After flow field and temperature field have been calculated by SIMPLE scheme, particle trajectories and temperature can be calculated with the flow field and temperature field and then particle source terms can be obtained. Numerical simulation results are outputted after the coupling of the flow phase and solid phase.

Fig. 7 and Fig. 8 show the predicted effects of particle size and Reynolds number on the thermal boundary layer, respectively. The modified thermal effectiveness factor, η_t, defined by Murry (1994a) can indicate the influence of a particle on the thermal boundary layer, is also used in our present study, can be written as $(T_{pmax} - T_\infty)/(T_w - T_\infty)$, where T_{pmax} represents the maximum particle temperature reached on passing through the thermal boundary layer. But in Murry's study, the thermal response time was constant while a particle traveled through the thermal boundary layer. It may induce some errors in predicting this influence. In our present investigation, the change in the particles temperature and velocity was directly obtained by integrating the particle motion and energy equation using the fourth-order Runge-Kutta scheme. It was found that the level of the particle's influence reduces dramatically with increasing particle size and flow Reynolds number. This may explain why there is little heat transfer enhancement for large particles size and high Reynolds number.

The numerical model can also predict the local heat transfer characteristics of the tube in suspension cross flow. The result can be seen in Fig. 9 for a Reynolds number of 13 400 and particle size of 64 μm. The curves shown are for solid loading ratio of 0, 1.0 and 2.0 (kg/kg), respectively. As shown in Fig. 9, a trend of local heat transfer enhancement can be identified. The largest enhancement can be found at the front point and the second largest at the rear

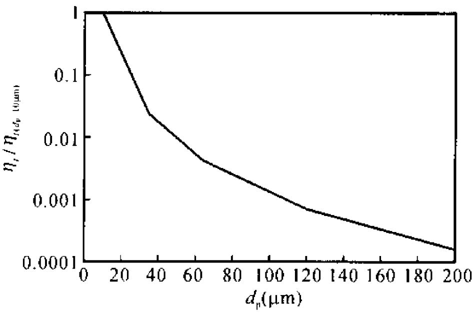

Fig. 7 The predicted effect of particle size on the level of the particle's influence on the thermal boundary layer over the front of the tube

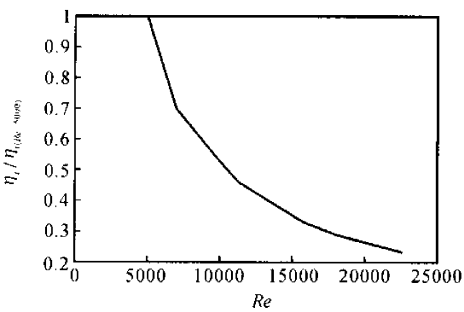

Fig. 8 The predicted effect of flow Reynolds number on the level of the particle's influence on the thermal boundary layer over the front of the tube

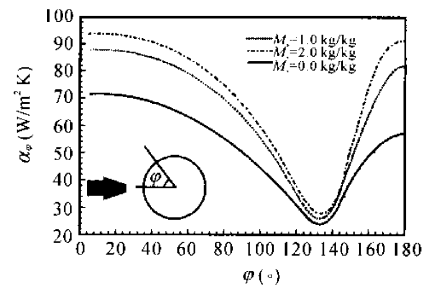

Fig. 9 The predicted local heat transfer factor of a tube in a suspension cross flow (Re = 13,400, d_p = 0.064mm)

point of the tube (where φ is equal to 0 and 180 degrees, respectively). But there was little local enhancement at the point of $\varphi = 125 - 145$ degrees. This result is in good agreement with the experimental result of Murry (1994b).

The total heat transfer enhancement with increasing solid loading ratio was also predicted by the numerical model, as shown in Fig. 10. A trend of increasing enhancement with increasing solid loading ratio in the simulated results could be found, although the predicted increase in Nusselt number was smaller than that in the present experimental data.

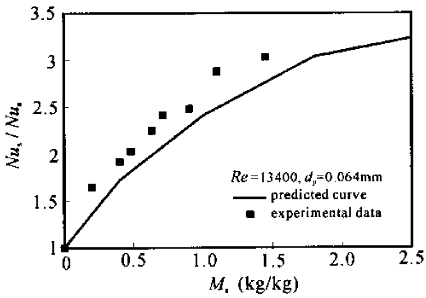

Fig. 10 Predicted increase in Nusselt number compared with that in the experimental data (Re = 13,400, d_p = 0.064 mm)

CONCLUSIONS

Investigation of the effect of solid particles on the heat transfer characteristic of a tube in suspension cross flow yielded a useful correlation incorporating solid loading ratio, particle size and flow Reynolds number. Heat transfer reduction has been found for very low solid loading ratio ($M_s < 0.05$ kg/kg). The local heat transfer characteristics around the tube were successfully predicted by the present numerical model. For total heat transfer enhancement due to particles the model have been shown to give good agreement between predictions and experimental data.

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