Numerical modeling of heat transfer due to particle impact on a wall

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벽면에서의 입자 衝突에 의한 열전달 수치 모델

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Abstract

보일러 및 연소로 등에서의 浮流體 환경에서 입자와 벽면 사이의 열전달 현상을 糾明하기 위한 수치적인 모델링을 행하였다. 본 연구에서는 벽면에 수직 衝突하는 입자에 의한 열전달 현상을 알아보기 위해 2차원 모델을 사용하였다. 입자표면에서 단열된 경계조건과 등온의 경계조건을 사용한 결과를 비교함으로써, 입자가 벽면에 충돌할 때 유체를 매개로한 전도와 입자에 의해 惹起된 대류 현상을 비교할 수 있었다. 계산 결과, 입자가 벽면에 충돌하기 직전에는 입자 크기의 반정도의 거리에 도달할 때까지는 입자의 영향이 별로 없고, 충돌하고 난 후에 영향이 많았다. 또한, Pe 수가 작을 때는 유체를 매개로 한 전도가 지배적이며, Pe 수가 증가할수록 야기된 대류의 효과가 점차 증가하였다.

Introduction

A great many industrial processes involve contact and interaction between solids and fluids . In gas - solid suspension and fluidization flows, the solid particles play important roles on the heat transfer from the wall or immersed surfaces. Heat transfer mechanisms due to impacts are gas - mediated heat conduction, induced gas convection caused by the movement of the particles, and direct conduction through the contact areas connecting the surface and the particles during impact. The contribution of the third process is usually believed to be small due to the short impact duration and small contact area(Botterill, 1975). However, estimations of relative importance of these processes still vary, presumably due to lack of definitive solutions and experimental data. Recently, Kaviany(1988) carried out two - dimensional numerical calculations to show the effect of a moving particle on wall heat transfer in a channel flow. In his study, a single square particle moves with constant speed in a channel with given parabolic inlet velocity. It is very difficult to separate the effect of gas - mediated conduction from induced gas convection. The mechanism of the surface heat transfer can not be fully understood until the individual processes are clarified. No attempts to separate the two effects have been reported yet. It is the main objective of this research to clarify the nature of gas-mediated heat transfer and induced gas convection by carrying out numerical calculations for the proposed model problems. The focus of this research is to clarify the mechanism involved in suspension flows by numerically modeling the collision of particles with the wall. In order to achieve these goals, two-dimensional numerical models were developed. The present work is limited to laminar flow and regular arrays of particles. Although the assumption of the regular array is not realistic, it still can give us some insight about the mechanism of heat transfer between the particle-fluid-wall.

Problem statement

For most practical applications, such as fluidized beds or other suspension flows, particle impacts with the wall usually take place in the presence of many other particles, and more or less concurrently with many other particle-wall collisions, in a stochastic manner. At the present state of understanding and computational capability, modeling such a many-bodied, stochastic process is not feasible and perhaps not justified even if feasible. A more tractable problem is that of a regular array of particles impacting the wall in an otherwise well defined convective environment, such as a laminar or turbulent boundary layer. The present study is undertaken under the assumption that an analysis of such a problem might provide enough qualitative and semi-quantitative insight to make the effort worthwhile, or perhaps even to facilitate a more realistic and more accurate analysis in the future.

In the context of particles impacting a surface in the presence of a pre-existing convection environment, henceforth to be referred to as "background convection", it is reasonable to expect the strength of the pre-existing convection, relative to the disturbance caused by the particle, to be an important parameter. One might expect that when the relative strength of the background convection, measured in terms of the relative velocity or another suitable dimensionless parameter, is sufficiently small, the incremental convection is primarily dependent on the parameters of the particle - its size, velocity, and only weakly dependent on the parameters of the background convective environment. It would then seem reasonable to hope that in this limit one might be able to dispense with the background convection, and substitute in its place a background conductive environment. One such conductive environment which possesses a familiar feature found in convective systems is the time-dependent conduction boundary layer due to a sudden change in the boundary temperature. The conduction boundary layer thickness at the time of particle impact might play a role which is qualitatively and semi-quantitatively similar to that of the thermal boundary layer thickness in a background convection environment.

We consider an infinite fluid whose boundary temperature is suddenly changed at time $t=-t_0$. Separately, a particle of diameter D at a large distance away approaches the wall at a velocity v_P, ultimately colliding with the wall at $t=0$. The fluid is stationary except for the motion induced by the particle. Both the fluid and the particles are isothermal at temperature $T=T_0$ until $t=-t_0$, when the lower boundary temperature is suddenly changed to $T=T_w$. The incremental total heat transfer rate, that above what would have resulted from conduction alone, is integrated with time to obtain the total incremental energy exchange E, which is attributed to particle collision. To investigate the influence of fluid flow to the heat transfer, the following two boundary temperature conditions are used.

A) Adiabatic case : Particles will have zero normal temperature gradient conditions to see the pure effect of induced convection due to particle - disturbed flow field

B) Isothermal case : Assume constant particle temperature to see the combined effect of fluid - mediated conduction and induced convection between particle and wall. This condition means a particle has a very large heat capacity compared to the surrounding fluid or gas and may be close to the real situation.

The difference between these two cases is considered to represent the pure gas - mediated conduction effect of the particle, because the combined effects of gas - mediated conduction and induced convection are obtained from the isothermal particle simulations.

In this paper, two - dimensional computations will be considered, since most of the important phenomena which will be revealed by this model may be similar. The computational domain has height H and width 2W as shown in Fig. 1. Since symmetric conditions exist between particles, the calculation domain can be confined to contain only one particle as illustrated in Fig. 1. The temperature of the top wall is initially T_0 and the corresponding temperature of the semi - infinite conduction problem is imposed later on. And the bottom wall has constant temperature T_W for the temperature boundary conditions. An upper surface is imposed to simplify the boundary condition on the top of the domain. The effect of this boundary conditions will be limited because the conduction thermal boundary layer thickness, δ plays the primary role and should be considered as the relevant parameter instead of H or t_0. The time parameter t_0 is chosen such that the thermal boundary layer thickness, δ, becomes given values when the particle hit the bottom wall. At time $-t_i$, the computation starts and the particle begins to move toward the wall, with normal velocity component v_P. If $-t_0$ is earlier than $-t_i$, then the initial temperature distribution can be computed from the analytical solution of the semi - infinite conduction problem, and then entered as the initial condition. If $-t_0$ is later than $-t_i$, then the initial condition is isothermal, with the wall temperature changing at $-t_0$. Initially the particle is a half diameter away from the top wall. After the particle hits the wall, it is assumed to move away from the wall with the same speed. When the particle reaches a half diameter away from the top wall, a new top wall is applied two diameters below the original location of the top wall. The resulting computation makes it possible to see the remaining effect of the disturbed fluid motion until the incremental heat transfer becomes very small.

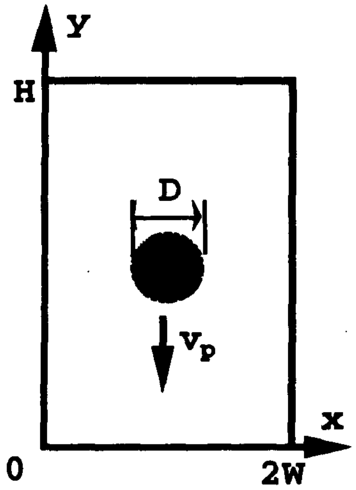


Fig. 1. Domain of the computation.

The given parameters of the problem are the temperature difference $\Delta T = T_w - T_0$, the particle diameter D, the particle velocity v_P, and the thermo-physical properties such as ρ, and α. Dimensionless independent variables can be introduced as follows :

$$x^* = \frac{x}{D}; y^* = \frac{y}{D}; t^* = \frac{t}{D/v_p} \quad (1)$$

$$u^* = \frac{u}{v_p}; v^* = \frac{v}{v_p}; T^* = \frac{T - T_0}{\Delta T}; p^* = \frac{p}{\rho v_p^{\ 2}} \quad (2)$$

Using these variables, the governing equations become :

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

$$\frac{\partial u^*}{\partial t^*} + u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} = -\frac{\partial p^*}{\partial x^*} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}}\right) \quad (4)$$

$$\frac{\partial v^*}{\partial t^*} + u^* \frac{\partial v^*}{\partial x^*} + v^* \frac{\partial v^*}{\partial y^*} = -\frac{\partial p^*}{\partial y^*} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 v^*}{\partial x^{*2}} + \frac{\partial^2 v^*}{\partial y^{*2}}\right) \quad (5)$$

$$\frac{\partial T^*}{\partial t^*} + u^* \frac{\partial T^*}{\partial x^*} + v^* \frac{\partial T^*}{\partial y^*} = \frac{1}{Pe}\left(\frac{\partial^2 T^*}{\partial x^{*2}} + \frac{\partial^2 T^*}{\partial y^{*2}}\right) \quad (6)$$

$$T^* = 1 \, at \; y^* = 0, t^* \geq -t_0^{\ *} \quad (7)$$

$$T^* = 0 \; at \; t^* < -t_0^{\ *} \quad (8)$$

$$\frac{\partial u^*}{\partial y^*} = 0, v^* = 0, T^* = T^*_{top}(t) \; at \; y^* = H/D \quad (9)$$

$$u^* = 0, \frac{\partial v^*}{\partial x^*} = 0, \frac{\partial T^*}{\partial x^*} = 0, at \; x^* = 0 \; and \; x^* = 2W/D \quad (10)$$

$$T^* = 1 \, at \; y^* = 0, t^* \geq -t_0^{\ *} \quad (11)$$

$$T^* = 1 \, at \; t^* < -t_0^{\ *} \quad (12)$$

where

$$t^*_{top}(t^*) = 0, t^* < -t^*_0 \quad (13)$$

$$t^*_{top}(t^*) = T^*_a(t^* + t^*_0, H/D), t^* \geqq -t^*_0 \quad (14)$$

At the particle surface

$u^* = 0$, $v^* = 1$, (15)

$T^* = 0$ for isothermal,

$$\frac{\partial T^*}{\partial n} = 0 \text{ for adiabatic case} \quad (16)$$

$$T^*(x^*, y^*) = 0 \; at \; t^* = -t^*_0 \; if \; -t^*_i < -t^*_0 \quad (17)$$

$$T^*(x^*, y^*) = T^*_a(t^*_0, -t^*_i, y^*) \; at \; t^* = -t^*_0 \; if \; -t^*_i > -t^*_0 \quad (18)$$

where

$$T^*_a(t^*, y^*) = 1 - erf\left(\frac{y^*}{2}\sqrt{\frac{Pe}{t^*}}\right) \quad (19)$$

Heat transfer rates per unit depth, q and the energy exchange per unit depth, E which was transferred from the bottom wall to the fluid during the calculation are defined as follows :

$$q(t) = \int_0^{2w} q''(x,t)\,dx \quad (20)$$

$$E(t) = \int_{-t_0}^{t} q \, dt \quad (21)$$

where q″(x, t) is heat flux at the bottom wall. The net incremental heat transfer and energy exchange are defined in the following equations.

$$\Delta q(t) = q(t) - q_0(t) \quad (22)$$

$$\Delta E(t) = E(t) - E_0(t) \quad (23)$$

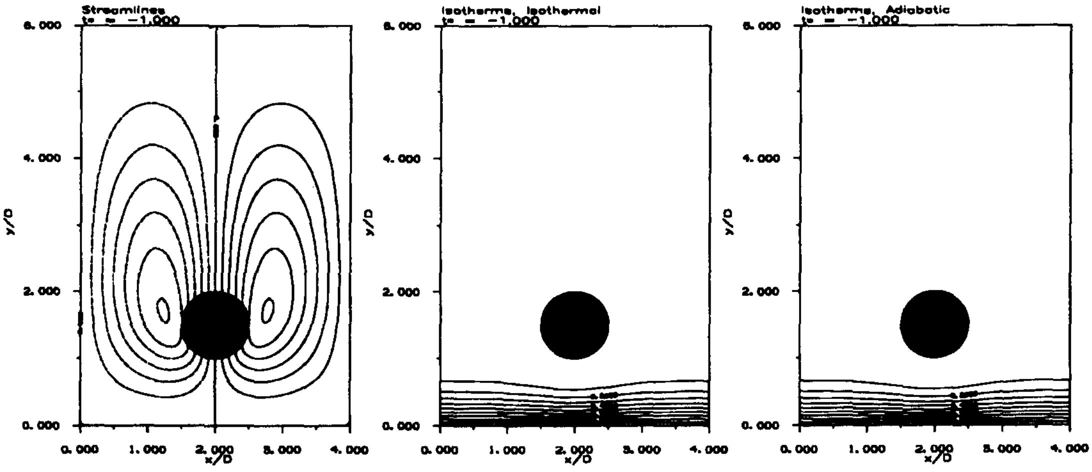
Where, q_0 is the heat transfer without the particle and E_0 is the energy exchange without the particle.

Discretization

The governing equations were discretized into a set of difference equations to solve them. Instead of using the commonly used stair-case boundary (or block-off boundary) at the particle boundary in the Cartesian coordinate, velocity and temperature boundary conditions are applied at the exact locations of its boundaries (Li, 1992) to apply more accurate boundary conditions. Momentum and energy equations are discretized by Taylor - series approximation and the continuity equations are discretized by control volume method to ensure the mass conservation at the particle boundary. Because of the non - linearity and coupling between the velocities and the pressure, iterative algorithm was used to solve the equations. SIMPLER algorithm was used for the simulation models. Numerical results were obtained for selected Prandtl numbers and Peclet numbers and background thermal boundary layer thickness, δ/D. The pressure equation which is derived from the continuity and momentum equations was solved by iterations. Uniform 32 ×96 meshes were used to discretize the governing equations. Numerical calculations were conducted mostly on the Axill - 311 workstation. The solutions were judged to be convergent when maximum value of residuals falls within the convergence criterion. The convergence criterion $\varepsilon = 1 \times 10^{-6}$ was used for these calculations.

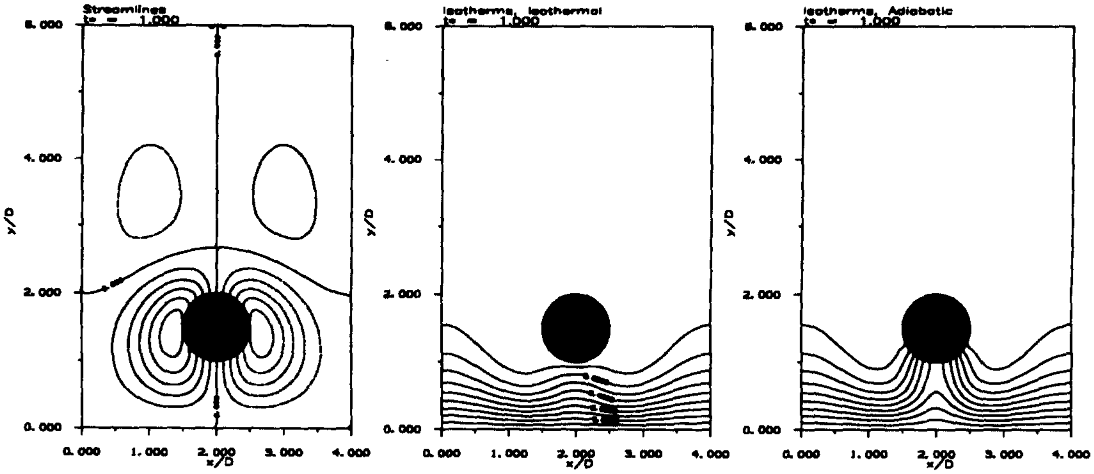
Results and discussion

Figuress 2 - 3 are representative plots of the streamlines and the temperature isotherms for both adiabatic and isothermal cases at various computation times for Re=20 and Pr=1. Negative t* means that particle is moving closer to the wall and positive t* means that it is moving away from the wall after hitting the bottom wall. It is seen that the fluid around the particle is pulled and pushed by the particle and recirculated at the boundaries. In Fig. 2, it is shown that at t*= - 1, the temperature profile is slightly disturbed by the particle near the point of impact due to the flow movement, hence increasing the heat transfer near the point of impact. Up to this time, both the isothermal and adiabatic cases exhibit almost identical temperature profiles which suggests that the conduction plays a minimal role in heat transfer at the wall, while the convection effect is commanding until the particle reaches one particle diameter away from the wall. It is also seen from Fig. 3 that the temperature



(a) (b) (c)

Fig. 2. (a) Streamlines, (b) temperature isotherms for the isothermal case, and (c) temperature isotherms for the adiabatic case at t* = - 1.



(a) (b) (c)

Fig. 3. (a) Streamlines, (b) temperature isotherms for the isothermal case, and (c) temperature isotherms for the adiabatic case at t*= − 1.

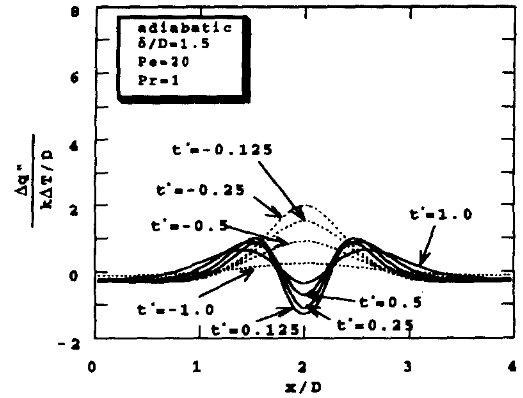


Fig. 4. Incremental heat flux at the obttom wall for adiabatic case.

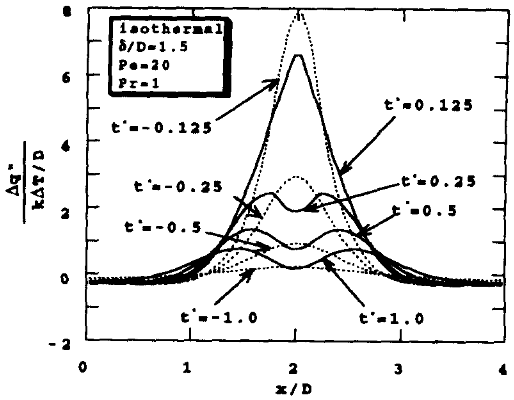


Fig. 5. Incremental heat flux at the bottom wall for isothermal case.

fields are disturbed to a greater extent by the movement of the particle especially near the impact point when it moves away from the wall after hitting the wall. Plots of the incremental heat flux at the wall for the adiabatic particle are illustrated in Fig. 4 for Pr=1, Pe=20, and δ/D=1.5 at different dimensionless times. Fig. 5 is a corresponding plot for the isothermal particle, where δ/D is the ratio of background thermal boundary layer at the time of impact to the diameter of the particle. When the particle approaches the wall, the results are represented by solid lines and when it moves away from the wall by dot lines. It is seen in Fig. 4 that heat flux for the adiabatic case is large near the impact point(x/D=2) due to the downward flow of the fluid induced by the particle when the particle is one diameter and one half diameter away from the bottom wall respectively(i.e., at t*= − 1.0 and t*= − 0.5). However, the heat flux near the boundaries(x/D=0, x/D=4) is smaller due to the upward flow movement shown in Fig. 4. As the particle moves closer to the wall, for example at $t^* = -0.5$ and $t^* = -0.25$, the heat flux increases at the impact point. After hitting the wall, the heat flux is smaller near the impact point even when the particle is more than one diameter away from the wall($t^* = 1$), because of the reversed flow field. On the contrary, for the isothermal case, Fig. 5 shows that the heat flux becomes large near the impact point when the isothermal particle approaches or moves away from the wall. The heat flux near the impact point becomes very large when the particle is very close to the wall. These indicate that the gas - mediated conduction effect is dominant over the induced gas convection caused by the movement of the particle when the particle is very close to the wall. It is easily inferred from Fig. 5 that there has been a larger temperature fluctuation near the impact point than at the boundary.

The net incremental energy exchange due to particle impact is equal to the net increase of the total heat transfer integrated over the entire wall and integrated with time. It is considered the total heat transfer above that of the background state without particle as defined as equation(23). It is an ultimate goal to get the net incremental energy exchange for the several different parameters. One representative plot of net incremental energy exchange as a function of time, is illustrated in Fig. 6. It also shows the effect of the height of the computational domain. The calculations were done for $H/R = 12$ and $H/R = 16$, and the effect of the height on the results is seen to be almost unnoticeable. Hence all other calculations were done for $H/D = 12$. This also justifies the selection of the boundary condition at the top computation domain. The net incremental energy exchange for isothermal particle is much larger than that of the adiabatic particle, which shows the effect of conduction is still very large in spite of their relatively short duration near the wall. Notice the difference between the adiabatic and the isothermal cases is very small until just before particle hits the wall, i.e., $t^* = 0$, but becomes large thereafter.

The influence of particle spacing or width of the computational domain on the numerical results, is illustrated in Fig. 7. It can be seen that the effect of computation width or particle

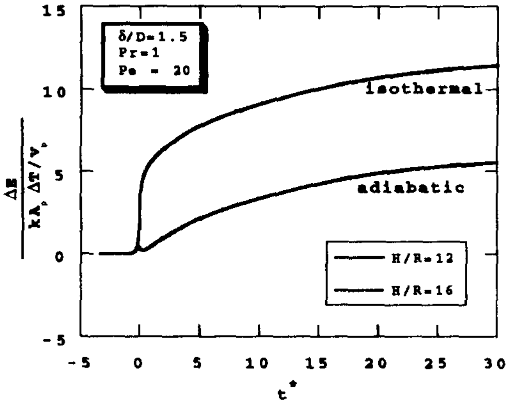


Fig. 6. Incremental energy exchange at the bottom wall for both adiabatic and isothermal cases with different domain heights.

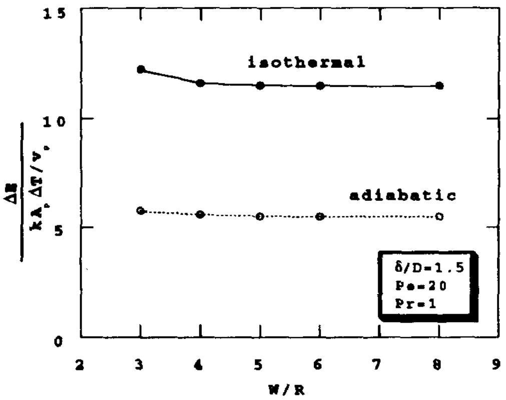


Fig. 7. Incremental energy exchange at the bottom wall versus computational domain width for both adiabatic and isothermal cases.

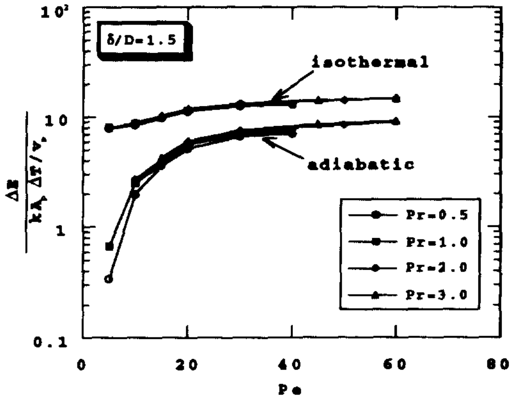


Fig. 8. Incremental energy exchang nondimensionalized with energy scale $kA_p\Delta T/v_p$ at various Pr and Pe numbers.

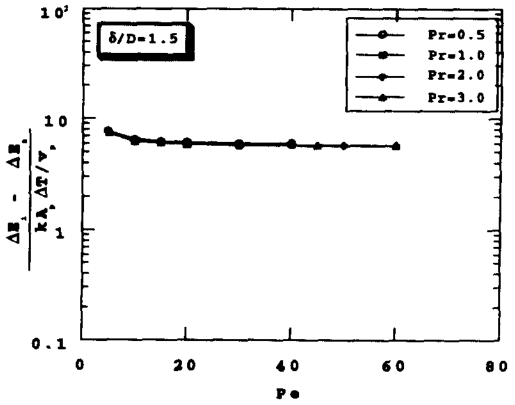


Fig. 9. Incremental energy exchange to represent the gas mediated conduction effect at various Pr and Pe numbers.

spacing becomes small as W/D increases. This allows us to carry out some calculations with W/D=4 to save computation time. Although the results may be quite different when the width is very small, it is not the prior objective to investigate the effect of width on the heat transfer mechanism in this particle-fluid-wall interaction model.

In Fig. 8, semi-log plots of the incremental energy exchanges for δ/D=1.5, are shown for several Pr and Pe number for both adiabatic and isothermal case. Non-dimensionalized values using the energy scale $kA_p\Delta T/v_p$, which is based on the heat conduction mechanism are used in this figure. A_p is the projection area of the particle to the wall. In two-dimensional case, A_p is equal to DL where D is a diameter of the particle and L, which is equal to unit length 1 in this model, is a length of the cylindrical particle. Even though the effect of Pr is small for the range of Pr, 0.5-3, the incremental energy exchange increase as Prandtl number increases for both the isothermal and the adiabatic case. The figure also reveals that the incremental energy exchange increases as the Pe increases, especially for the adiabatic case showing the induced convection effect becomes important for high Pe.

The difference between non-dimensionalized incremental energy exchange for the two particle models(adiabatic and isothermal) demonstrates the sole effect of gas-mediated heat conduction. The results are illustrated in Fig. 9 for δ/D=1.5. We can see the effect of the gas-mediated heat conduction is almost independent of the Peclet number, thus showing the energy scale, $kA_p\Delta T/v_p$, is a appropriate for representing the effect of gas-mediated heat conduction. The gas-mediated conduction effect decreases as Pr increases even though the effect is barely noticeable.

The incremental energy exchange normalized with a different energy scale, $\rho c_f \mathrm{Vol}\Delta T$, based on the heat capacity of fluid of equal volume as the particle, is plotted in Fig. 10. The volume of the particle, Vol is equal to πDL in the two-dimensional model. This figure also reveals that at small Pe or Re, gas-mediated conduction plays a dominant role and at large Pe or Re induced gas convection caused by the flow movement plays an important role. From figures 8-10, we can conclude that the energy

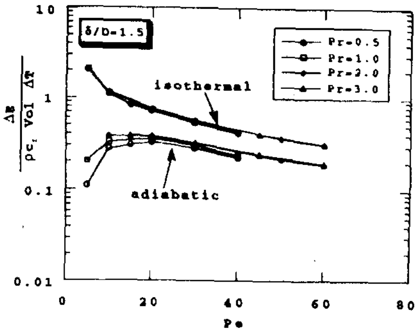


Fig. 10. Incremental energy exchange nondimensionalized with energy scale $\rho c_f Vol\Delta T$ at various Pr and Pe numbers.

scale, $kA_p\Delta T/v_p$, is better for the correlation of gas – mediated – conduction and the energy scale, $\rho c_f Vol\Delta T$, is more appropriate for induced gas convection.

Conclusions

A two – dimensional finite difference model for the particle – wall – fluid interaction environment was developed in order to understand the important phenomena in that environment. The model presented information describing the relative importance of the gas – mediated conduction effect and the induced convection effect. The results of the two – dimensional model show the following.

(1) By comparing the results using (a) adiabatic boundary conditions on the particle and (b) uniform, elevated temperature conditions on the particle, the contributions of fluid – mediated conduction and particle induced convection were successfully separated.

(2) The energy scale, $kA_p\Delta T/v_p$, based on the conduction near the wall, is a fairly good choice for representing the effect of the gas – mediated heat conduction. The energy scale, $\rho c_f Vol\Delta T$, based on the heat capacity of the fluid of equal volume as the particle, is a good choice for representing the induced gas convection effect.

(3) When the particle moves towards the wall, the gas – mediated conduction and the induced convection effect are very minor until the particle reaches a distance of one to one half diameter away from the wall.

(4) The temperature profiles for both the adiabatic and the isothermal cases are more disturbed when the particle is bouncing from the wall than when the particle is approaching the wall.

(5) The gas – mediated conduction effect is dominant over the induced gas convection effect when Pe is small. The induced gas convection effect becomes significant as Pe increases.

Abstract

A numerical study was undertaken to clarify the mechanisms of heat transfer in fluid – particle suspension flows. Such flows, including fluidization, are of considerable industrial importance. The present study uses 2 – D numerical computations of collisions of normal incidence between a particle and a wall. By comparing the results using (a) adiabatic boundary conditions on the particle and (b) uniform, elevated temperature conditions on the particle, the contributions of fluid – mediated conduction and particle induced convection were successfully separated. Computational expedience led to the use of a transient conduction thermal layer as the background thermal field for the analysis. The results shows that the effect of particle movement is very small until the particle reaches a distance of one to one half diameter away from the wall. The gas – mediated conduction effect is dominant over the induced gas convection effect when Pe is small and the induced gas convection effect becomes significant as Pe increases.

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