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Influence of Rayleigh number and solid volume fraction in particle-dispersed natural convection

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ABSTRACT

To study the heat transfer in a natural convection under dense particulate condition with finite-size spherical particles of uniform diameter, a direct numerical simulation is conducted. For calculating the momentum-interaction between the particle and fluid, our original immersed solid method is applied. The heat transfer in the particle-dispersed flow is treated in Eulerian way with the interfacial flux decomposition method. The results shows that, with fixing the thermal conductivity ratio (of the solid to the fluid) to be 100, the temporal- and horizontal-average Nusselt number $\langle Nu\rangle_t$ increases monotonically with solid volume fraction (vf) at Rayleigh number $Ra = 10^4$, while $\langle Nu\rangle_t$ at $Ra = 10^5$ exhibits a local maximum at around $vf = 40\%$, although $\langle Nu\rangle_t$ at $Ra = 10^5$ is always larger than that at $Ra = 10^4$. The heat flux in the particulate system is decomposed into the contributions by convection and conduction through the particles, fluid and interface, and the result shows that the conduction through the interface is the dominant factor to the vertical heat flux in the media. Through visualization of the heat flux through the particle surface, the importance of directly resolving the local heat conduction within the individual particle and through the interface is highlighted.

1. Introduction

Dense particulate flow is observed in industrial applications including energy conversions and chemical processes, and the particles often play an important role in the fluid flow and heat transfer in the particulate media. The heat transfer in particle-dispersed media containing a small amount of nanoparticles of substantially higher effective thermal conductivity than the base fluid has been investigated [1,2], and the large effect on the fluid flow and the improvements in the heat transfer performance were reported. Generally, solid particles of high thermal conductivity could improve the effective thermal conductivity of the particulate media by hundreds or even thousands times greater than that of a base fluid such as water and ethylene glycol (while suppressing the gross viscosity of the mixture), which is a significant way to exploit highly-conductive particles.

Through extensive studies over the decades, a number of numerical techniques have been developed to simulate the motion of a large number of solid particles in a fluid [3–5]. In a combined Eulerian-Lagrangian formulation for the particulate flows, the Lagrangian frame is employed for solving the motion of the solid particles and the particle temperature, while the fluid phase is represented in the Eulerian frame, [4,6,7]. This choice of the frames has enabled simulation of a variety of fluid-particle interaction problems and the heat transfer in a solid-dispersed multiphase flow [8–10]. However, for the problems of heat transfer in particulate media, many researchers assigned a unique temperature for each particle instead of considering temperature distribution inside the individual particle.

Takeuchi et al. [11] discussed the effects of temperature gradient within the individual particles of non-negligible size and inter-particle heat flux due to collision on the flow structure and heat transfer in a particle-dispersed natural convection. They studied the contributions of the convective and conductive components of the heat flux to the overall heat transfer for different bulk solid volume fractions and thermal conductivities of the particles. They showed that, with highly-conductive particles, the conductive component of the heat flux enhances the overall amount of heat transfer in the particulate flow as the bulk solid volume fraction is increased. On the other hand, the balance between the convective and conductive components remains approximately the same irrespective of bulk solid volume fractions for neutrally-conductive particles. While the importance of the heat transfer paths (either through the fluid or particle) near the hot wall has been pointed

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out [12,13], the contribution of interfacial heat transfer in the near-wall region has not been investigated.

In this paper, we focus on the diversity of the heat transfer in a dense particulate media by changing the Rayleigh number and solid volume fraction while fixing the thermal conductivity ratio of solid to fluid, because the above parameters are the primary factors to the heat transfer in a natural convection under dense particulate condition with finite-size spherical particles of uniform diameter. To highlight the effect of the interface, dense particle-dispersed flow, especially the solid volume fraction higher than 50%, is studied, and the heat flux in the particle-dispersed natural convection is further decomposed into the conductive components through the fluid, particle and fluid-particle interface for different Rayleigh numbers.

For simulating the fluid flow around the individual particles and the heat transfer in the dense particulate flow, direct numerical simulation is conducted. For calculating the momentum exchange between the particles and fluid, our original immersed solid method [12,14] is employed, as the method is suitable for simulating collective behavior of thousands of particles [15]. The temperature distribution (in both the fluid and particle phases) is solved by considering the anisotropy of the local thermal conductivity in the interfacial cells [12,13].

The contents of the paper are as follows. Section 2 introduces the governing equations, and Section 3 explains the numerical methods including our original immersed solid method for solving the fluid-particle momentum-interaction, the interfacial flux decomposition method and contact resistance model. Heat transfer in the particulate media between two parallel plates is studied in Section 4. In this section, some snapshots of the temperature and velocity distributions in the particle-dispersed flows are presented (Section 4.2), and the spatial variation of the heat flux along the vertical direction is discussed for different the bulk solid volume fraction (Section 4.3). Then, the heat transfer is decomposed into the conductive and convective components through the fluid and particles (Section 4.4). Finally, the effect of the interfacial heat flux between the fluid and particles on the heat transfer is discussed (Section 4.5).

2. Governing equations

For the fluid phase, an incompressible iso-viscous Newtonian fluid is assumed. The governing equations for the fluid are the equations of continuity, momentum and energy given by the following equations:

$$\nabla \cdot \boldsymbol{u}_f = 0, \tag{1}$$

$$\frac{\partial \boldsymbol{u}_f}{\partial t} + (\boldsymbol{u}_f \cdot \nabla)\boldsymbol{u}_f = -\frac{1}{\rho_f}\nabla p + \nu \nabla^2 \boldsymbol{u}_f - \beta(T - T_0)\boldsymbol{g}, \tag{2}$$

$$\frac{\partial T}{\partial t} + \boldsymbol{u}_f \cdot \nabla T = \frac{\lambda_f}{\rho_f c_f}\nabla^2 T, \tag{3}$$

where $\boldsymbol{u}_f$ is the fluid velocity, the subscript "f" denotes the fluid phase, t is the time, ρ_f is the fluid density, p is the pressure, ν is the kinematic viscosity of the fluid, β is the thermal expansion coefficient, T is the temperature, $\boldsymbol{g}$ is the gravitational acceleration, λ_f is the thermal conductivity of the fluid, and c_f is the specific heat of the fluid. The dimensionless forms of the above equations are respectively given as follows:

$$\nabla \cdot \boldsymbol{u}^* = 0, \tag{4}$$

$$\frac{\partial \boldsymbol{u}^*}{\partial t^*} + (\boldsymbol{u}^* \cdot \nabla)\boldsymbol{u}^* = -\nabla p^* + \sqrt{\frac{Pr}{Ra}}\nabla^2 \boldsymbol{u}^* - \beta(T^* - T_0^*)\boldsymbol{j}, \tag{5}$$

$$\frac{\partial T^*}{\partial t^*} + \boldsymbol{u}^* \cdot \nabla T^* = \frac{1}{\sqrt{Pr \cdot Ra}}\nabla^2 T^*, \tag{6}$$

where $\boldsymbol{j} = \boldsymbol{g}/|\boldsymbol{g}|$ is the unit vector in the direction of the gravity. The dimensionless variables are represented with the superscript '$*$' and those are listed in Table 1. Here, the reference velocity is $U = \sqrt{g\beta\Delta T H}$ with the characteristic temperature difference ΔT. The reference length H is explained in Section 4.1. The following two non-dimensional numbers are introduced: Prandtl number $Pr = \nu\rho_f c_f/\lambda_f$ and Rayleigh number $Ra = Pr \cdot (UH/\nu)^2$. Hereafter, the notation '$*$' is omitted as there is no possibility of misunderstandings.

3. Numerical method

3.1. Immersed solid method

Based on our original immersed solid method [15], the procedure of the interaction between the fluid and particle is briefly explained in the following. In the method, one particle diameter is typically resolved more than 10 grid points. Hereafter, the cell partially occupied by the solid object is referred to as interfacial cell. The velocity in the interfacial cells is established by volume-averaging the fluid velocity and the particle velocity:

$$\boldsymbol{u} = (1 - \alpha)\boldsymbol{u}_f + \alpha \boldsymbol{u}_s, \tag{7}$$

where α is the local solid volume fraction and the subscript "s" denotes the solid phase. The velocity field of the interface is assumed to obey Eq. (5) and the intermediate velocity field $\boldsymbol{u}^{F2}$ is obtained as follows:

$$\boldsymbol{u}^{F1} = \boldsymbol{u}^n + \int_{t^n}^{t^{n+1}} \left[-\nabla p - (\boldsymbol{u} \cdot \nabla)\boldsymbol{u} + \sqrt{\frac{Pr}{Ra}}\nabla^2 \boldsymbol{u} - \beta(T - T_0)\boldsymbol{j} \right] dt, \tag{8}$$

$$\nabla^2 \phi = \frac{\nabla \cdot \boldsymbol{u}^{F1}}{\Delta t}, \tag{9}$$

$$\boldsymbol{u}^{F2} = \boldsymbol{u}^{F1} + \nabla\phi \Delta t, \tag{10}$$

where $\Delta t = t^{n+1} - t^n$ is the time increment and $\phi = p^{n+1} - p^n$. In this paper, the convective and viscous terms are integrated in time with the second-order Adams-Bashforth and Crank-Nicolson methods, respectively. The momentum-interaction force between the phases is modeled as follows [12,15]:

$$\boldsymbol{f}_{\mathrm{I}}^n = \frac{\alpha^n(\boldsymbol{u}_s^n - \boldsymbol{u}_f^{F2})}{\Delta t}, \tag{11}$$

and the force is imposed at the interfacial cell to satisfy the no-slip condition at the object surface, and the velocity field is time-updated to the next time level $(n + 1)$ as:

$$\boldsymbol{u}^{n+1} = \boldsymbol{u}^{F2} + \boldsymbol{f}_{\mathrm{I}}^n \cdot \Delta t. \tag{12}$$

The motion of the solid particle is calculated in the Lagrangian frame. The translating and angular velocities of the particle are updated as follows:

$$\boldsymbol{u}_s^{n+1} = \boldsymbol{u}_s^n + m_s^{-1} \int_{t^n}^{t^{n+1}} dt \int_{V_s} \left(-\rho_f \boldsymbol{f}_{\mathrm{I}} \right) dV_s, \tag{13}$$

$$\boldsymbol{\omega}_s^{n+1} = \boldsymbol{\omega}_s^n + \boldsymbol{I}_s^{-1} \cdot \int_{t^n}^{t^{n+1}} dt \int_{V_s} \boldsymbol{R}_s \times (-\rho_f \boldsymbol{f}_{\mathrm{I}}) dV_s, \tag{14}$$

where m_s is the particle mass, V_s is the volume enclosing the particle, $\boldsymbol{\omega}_s$ is the angular velocity vector, $\boldsymbol{I}_s$ is the inertia tensor of the particle and $\boldsymbol{R}_s$ is the radial vector from the particle center to the