HEFAT2014
10th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
14 – 26 July 2014
Orlando, Florida

NUMERICAL STUDY OF HEAT TRANSFER MODIFICATION IN A PARTICLE-LADEN CHANNEL FLOW CONSIDERING PARTICLE–TURBULENCE INTERACTION

Caixi Liu, Zheng Tang, Yu-Hong Dong*.
*Author for correspondence
Shanghai Institute of Applied Mathematics and Mechanics,
and Shanghai Key Laboratory of Mechanics in Energy and Environment Engineering,
Shanghai University, Shanghai 200072, China.,
E-mail: dongyh@staff.shu.edu.cn

ABSTRACT

Small particles could be as active agents interact with not only velocity field but also temperature field. Understanding and prediction of heat transfer between particle-laden turbulent flow and adjacent wall are of great interest in both applications and fundamentals. In this work, we employ direct numerical simulation and simultaneous Lagrangian tracking to study the influence of inertial particle with different particle thermal response time on heat transfer in particle-laden turbulent channel considering particle–turbulence momentum and energy interaction. As a result, the thermal feedback of the particle has always positive contribution to turbulent heat transfer and the thermal feedback effect gradually strength as the particles specific heat capacity increases..

INTRODUCTION

Understanding and prediction of heat transfer between particle-laden turbulent flow and adjacent solid wall are of great interest in both applications and fundamentals[1-2]. There are many industrial examples illustrating the importance of the thermal transport between the two phases [3]. It is known that heat transfer in single-phase flows have been widely concerned by many researchers, and the mechanisms of heat transfer in dispersed turbulence are poorly understood since the complexity of momentum and energy transport increases when the interactions among small particles and flow field and thermal field exist together.

To date, Eulerian-Lagrangian direct numerical simulations (DNS) and large eddy simulations (LES) have been widely used to examine inertial particle-laden turbulent flow with two-way coupling as well as heat or mass transfer. Zonta & Marchioli[4] studied the influence of dispersed micrometer size particles on turbulent heat transfer in wall-bounded flows and showed that, with respect to single-phase flow, heat transfer fluxes at the walls increase by 2% roughly when the flow is laden with the smaller particles and a opposite trend for the flow with larger particles. A study of Kuerten et al. [5] found the ratio of particles and fluid specific heat play a significant impact on heat transfer. The results of some authors [6-8] present that the importance of the effect of particle-turbulence interaction on the characteristics of flow field and thermal field, and the interaction depends on many factors including physical properties and thermal properties of the two phases. Besides two-way coupling between particles and turbulence, thermal coupling is also important for the modulation of the fluid and particle temperature fields. However, a review of the literature shows that the effect of turbulence–particle interactions on heat transfer in two-phase flows is not well understood.

In this study, the modulation of turbulent flow and heat transfer in particle-laden channel flow is studied by direct numerical simulations. A coupled Eulerian-Lagrangian model [9] in which the momentum and energy transfer between the discrete particles and the continuous fluid are fully taken into account. In the analysis, we employ two-way simulation models not only to solve the momentum equations but also the energy equation，to study the influence of inertial particle with different specific heat capacity on flow and heat transfer..

NOMENCLATURE

Re_τ	Reynolds number of flow based on the friction velocity
Re_p	particle Reynolds number
T	Fluid temperature
d_p	particle diameter
r_f	fluid mass density
r_p	particle mass density
d	half-width of the channel
k_f	thermal conductivity of fluid
f_p^i	momentum feedback terms by the paritcles
q_p	thermal feedback terms by the paritcles

$C_{p,f}$ fluid specific heat capacity at constant pressure
$C_{p,p}$ particle specific heat capacity
T_p particle temperature
$\bar{u}_i$ fluid velocity at particle position
T_f fluid temperature at particle position
Pr Prandtl number
q normal-wall total heat flux
q_{feed} normal-wall feedback heat flux
Nu total Nusselt number
Nu_T turbulent Nusselt number
Nu_{feed} thermal feedback Nusselt number
τ_p particle velocity relaxation time
τ_T particle temperature relaxation time
τ_f fluid viscous timescale
St_T thermal stokes number
τ_t turbulent stress tensor
τ_{feed} particle feedback stress tensor

Subscripts
rms root mean square

. METHODOLOGY

We present the mathematical model for turbulent non-isothermal channel flow with particles as represented in Figure 1. Assuming that the fluid is incompressible and Newtonian, the governing balance equations and a convection-diffusion equation for the fluid (in dimensionless form) read

$$\frac{\partial u_i}{\partial x_i} = 0, \tag{1}$$

$$\frac{\partial u_i}{\partial t} = -u_j \frac{\partial u_i}{\partial x_j} + \frac{1}{\mathrm{Re}_\tau} \frac{\partial^2 u_i}{\partial {x_j}^2} - \frac{\partial p}{\partial x_i} + \delta_{1,i} + f_p^i, \tag{2}$$

$$\frac{\partial T}{\partial t} + u_j \frac{\partial T}{\partial x_j} = \frac{1}{\mathrm{Re}_\tau \mathrm{Pr}} \frac{\partial^2 T}{\partial {x_j}^2} + q_p, \tag{3}$$

where, f_p^i and q_p are the momentum-coupling and thermal-coupling terms, respectively. The terms as follows [14]:

$$f_p^i = -\frac{1}{V} \sum_{j=1}^{n} \frac{3V_p^j C_D^j}{4d_p^j} \left|\mathbf{v}^j - \bar{\mathbf{u}}^j\right| (v_i^j - \bar{u}_i^j), \tag{4}$$

$$q_p = \frac{1}{\mathrm{Re}_\tau \mathrm{Pr}} \frac{1}{V} \sum_{j=1}^{n} Nu_p^j \cdot \pi d_p^j (T_p^j - T_f^j), \tag{5}$$

where V is the volume of a cell, n is the number of particles in each cell, V_p^j is the volume of the j particle.

The particle equations are obtained as follows,

$$\frac{dx_i}{dt} = v_i, \tag{6}$$

$$\frac{dv_i}{dt} = -\frac{3}{4} \frac{C_D}{d_p} \left(\frac{\rho_f}{\rho_p}\right) \left|\mathbf{v} - \bar{\mathbf{u}}\right| (v_i - \bar{u}_i), \tag{7}$$

$$\frac{dT_p}{dt} = \frac{Nu_p}{2} \frac{(T_f - T_p)}{\tau_T}, \tag{8}$$

where, $\bar{u}_i$ and T_f is the fluid velocity and temperature at particle position. Nu_p is the particle Nusselt number and τ_T is the particle thermal response time. The first term on the right-hand side of Eq. (7) represents the Stokes drag force. The non-linear drag coefficient C_D follows a non-linear approximation can be written as[11]:

$$C_D = \frac{24}{\mathrm{Re}_p} (1 + 0.15 \mathrm{Re}_p^{0.687}), \tag{9}$$

where $\mathrm{Re}_p = d_p \left|\mathbf{v} - \bar{\mathbf{u}}\right| / \nu$ represents the particle Reynolds number. The correction for C_D is necessary when Re_p is larger than 1.

The particle Nusselt number is given by the well-known empirical correlation[12] given as:

$$Nu = 2 + 0.6 \cdot \mathrm{Re}_p^{1/2} \cdot \mathrm{Pr}^{1/3} \cdot \tag{10}$$

The particle thermal response time can be defined as

$$\tau_T = C_{p,p} \rho_p d_p^2 / (12 k_f), \tag{11}$$

where $C_{p,p}$ is the specific heat of particle. In this study, the temperature of a particle is uniform, which means the interior temperature of particle being equal its surface temperature.

If particle volume fraction is high enough, the particles exchange both momentum and energy with the carrier fluids, and heat transfer is modulated by the wall-bounded flow laden with particles. It is a key issue to the feedback of particles on turbulent flow in the study of two-phase interaction.

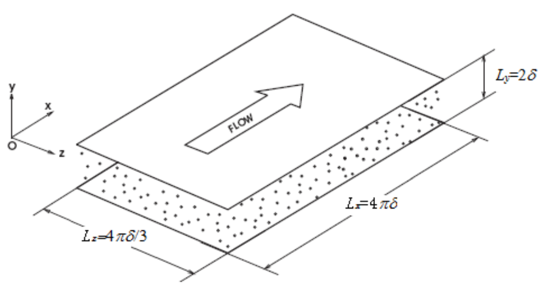


Fig. 1 Sketch of particle-laden channel flow

NUMERICAL METHOD

In this study the governing equation of fluid are solved using the fractional step method on a staggered grid. Periodic boundary conditions are applied in the streamwise and spanwise direction, and no-lip and no-penetration velocity conditions are imposed on the walls. Two different constant temperature are performed on the bottom and top walls.

Lagrangian particle tracking is used to obtain the position, velocity and temperature of particles. The motion equations of particle are advanced in time by a second-order Adams-Bashforth method. Fourth-order lagrangian polynomial was applied to interpolate the fluid velocity and temperature at particle position. Periodic boundary conditions are used to reintroduce particle into the computational domain when particle motion outside the channel in homogeneous directions, whereas perfectly-elastic collision at the smooth wall was assumed when the distance between particle center and wall surface is less than particle radius [9]. At the beginning of simulation, particle velocity and temperature are set equal to those of the fluid at the particle initial position.

To ensure that the computed results are independent of the time steps and the grid sizes, results calculated by different grid numbers and time steps were carefully examined in our present study. Moreover, the present computational method and the relevant code have also been validated and verified in our previous works [9,10]. In which, extensive validations and verifications have been performed in thermal turbulent channel flow and agree well each other (cf. Fig.1 and Fig.2 in [10]). It showed the distributions of turbulent intensities and the temperature fluctuation (not shown here) were in good agreement with the typical DNS and experiments data.

RESULTS AND DISCUSSION

Fluid physical parameters and the material characteristics of particle for each test case are shown in Table 1 and Table 2, respectively. Here, each case consisting of 200,000 particles with different specific heat is tracked in the turbulent channel flow.

Table 1. Fluid physical parameters.

ρ_f	$C_{p,f}$	k_f	T_w^h / T_w^c	Pr	Re_τ
1.25kg/m^3	1.005$kJ/(kg.K)$	0.059$W(m^{-1}K^{-1})$	$20°C / 40°C$	0.71	180

Table 2. Definition of the test cases.

Case	d_p(μm)	d_p / δ	$C_{p,p} / C_{p,f}$	St	St_T
A1.1	140	6×10^{-3}	0.5	125	66
A1.2	140	6×10^{-3}	8	125	1062
A1.3	140	6×10^{-3}	24	125	3185
A1.4	140	6×10^{-3}	50	125	6635
C1.1	—	—	—	—	—

Fig. 1 shows the profile of the rms temperature fluctuation and mean temperature of fluid in the channel flow laden particles for different ratios of particle-to-fluids specific heat capacity $C_{p,p}/C_{p,f}$ from 0.5 to 24. The behavior of the fluid temperature averaged over the homogeneous directions is shown in Fig 1(a). The reduction of the temperature fluctuations becomes more obvious with the increase of the particle specific heat. It is seen that the slope of the mean temperature profile exhibits steeper and steeper in the near wall region with specific heat increasing.

At a relative high loading, the ratio of particle-to-fluids specific heat capacity is dominant factor affecting the heat transfer process. Fig. 1(b) shows that in A1.1 case ($C_{p,p}/C_{p,f}$ =0.5) the mean temperature becomes more uniformly distributed in channel and yet the opposite trend in A1.2($C_{p,p}/C_{p,f}$ =8) and A1.3($C_{p,p}/C_{p,f}$ =24) cases. That indicates the normal-wall gradient of fluid mean temperature increases in the near wall region with the particle specific heat larger more than the fluid, and it decrease with the particles with particle specific heat becoming smaller. Regardless of the particles specific heat, the particles always decreases the intensity of the temperature fluctuations. And the modulation of the temperature fluctuations becomes more obvious with the increase of the particles specific heat.

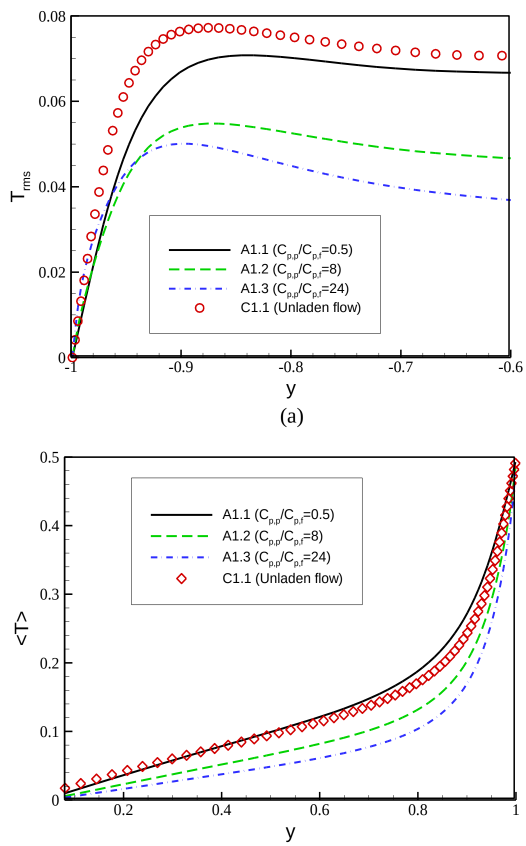
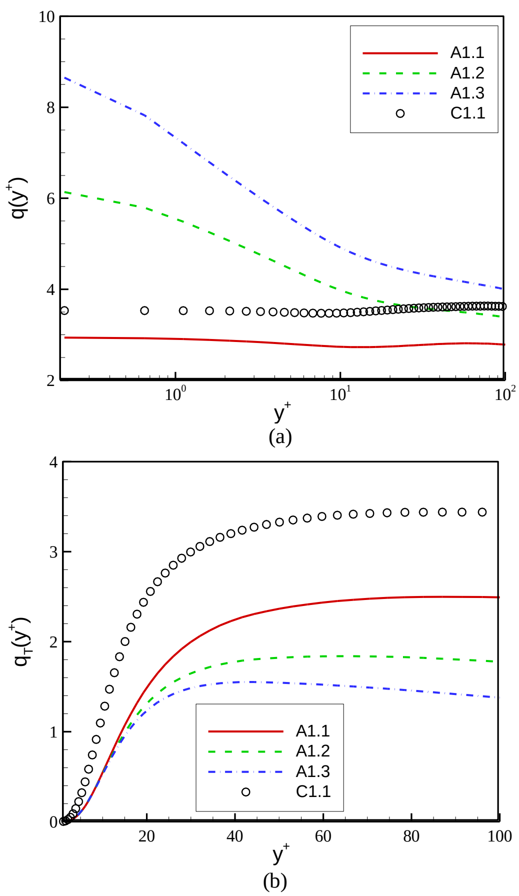


Figure 1. Primary fluid temperature statistics both unladen flow and particle-laden flows: (a) Rms of fluid temperature fluctuations; (b)Mean fluid temperature.

The modification induced by the particles on the wall-normal heat flux is investigated upon comparison against an unladen flow (C1.1case). The total wall-normal is defined as

$$q(y)=\underbrace{\frac{\partial <T>}{\partial y}}_{q_v(y)}\underbrace{-\mathrm{Re}_\tau \Pr <T'u_2'>}_{q_T(y)}+\underbrace{\mathrm{Re}_\tau \Pr\int_{-1}^{y}<q_p>dy}_{q_{feed}(y)}, \tag{11}$$

The total heat flux in Eq.(8) is split into three parts i.e., the turbulent flux, the viscous flux and the particle feedback component. Fig.2 shows the heat flux along wall-normal direction. In comparison with unladen flow, the difference of the total heat flux shows that the thermal coupling is quite important and particles has notably influence on the total heat flux. In A1.1 cases ($C_{p,p}/C_{p,f}$ =0.5), the total heat flux is reduced by the smaller specific heat particles. When the particle specific heat becomes larger, the total heat flux increases gradually in Fig 2(a). In the viscous layer, the viscous heat flux plays a dominant role in the total heat flux. The viscous heat flux is proportional to the mean fluid temperature gradient, and the clutter of particles in the near wall region increases the mean fluid temperature gradient with the increase of the particle specific heat. That leads to the total heat flux becomes increasingly larger in the near-wall region with the particle specific heat increases. For the turbulent heat flux, it becomes more important to the total heat flux outside the buffer layer. However, the turbulent heat flux is significantly reduced by the particles for all cases in Fig. 2(a). The reason for this is that wall-normal and temperature turbulent intensity are restrained.



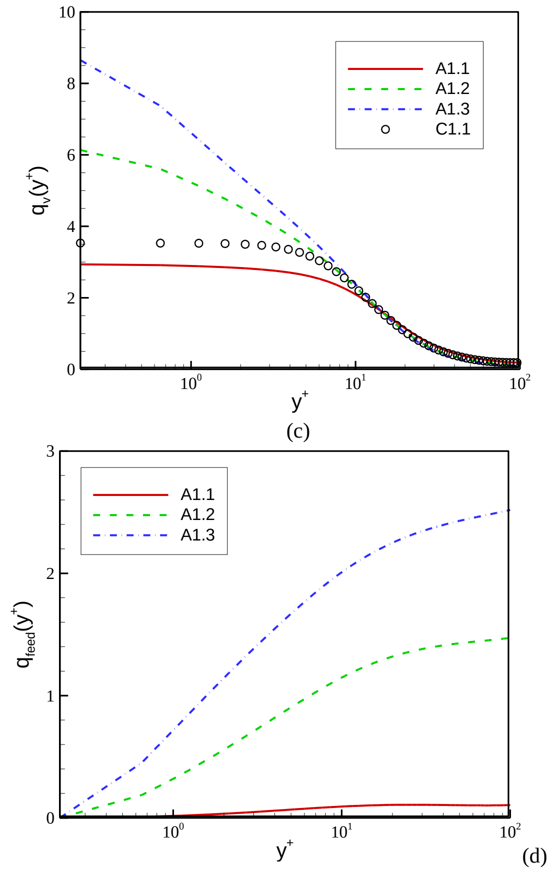


Figure 2. Wall-normal heat flux both unladen flow and particle-laden flows: *(a)* Total heat flux; *(b)* Turbulent heat flux; *(c)* Viscous heat flux; *(d)* Particle feedback heat flux.

For A1.3 case, the turbulent heat flux decreases while the total heat flux increases in the near wall region and outside buffer layer, shown as Fig. 2(a) and 2(b). The reason is that the particle thermal feedback heat flux is larger than the reduction of the turbulent heat flux. The essence of thermal feedback of the particles on turbulence is heat exchange between the particles and surrounding fluid. As the particle specific heat increases, the particle thermal response time becomes gradually larger. So the temperature difference between the particles and surrounding fluid increases in Fig. 3. That leads the particle thermal feedback increases in Fig. 2(d).

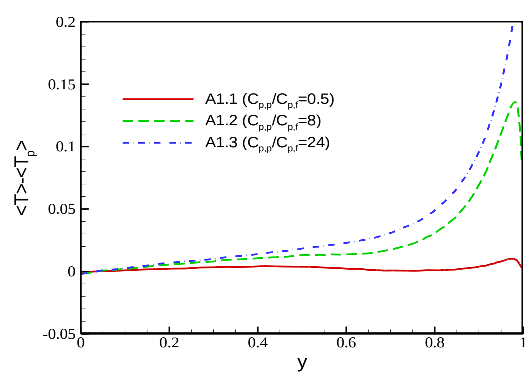


Figure 3. The relative temperature between the particles and surrounding fluid.

The Nusselt number is an important parameter relevant to the heat transfer coefficient. It reflects the ability of the convective heat transfer compared to conductive heat transfer in fluid. The Nusselt number (Nu) can be decomposed into three parts, i.e., a constant viscous contribution, a turbulent contribution, and the particles feedback contribution as shown in Fig. 4, the computed Nusselt numbers, as well as their turbulent contribution and the particles feedback contribution are plotted as a function of the thermal stokes number. the total Nusselt number and the feedback Nusselt number are monotonically increasing with the particle thermal Stokes number and yet the turbulent Nusselt number is opposite. Furthermore, we find the ability of heat transfer for particle-laden flow is enhanced relative to unladen flow when a logarithmic value of the particle thermal Stokes number is larger than 7.05 in Fig. 4(a).

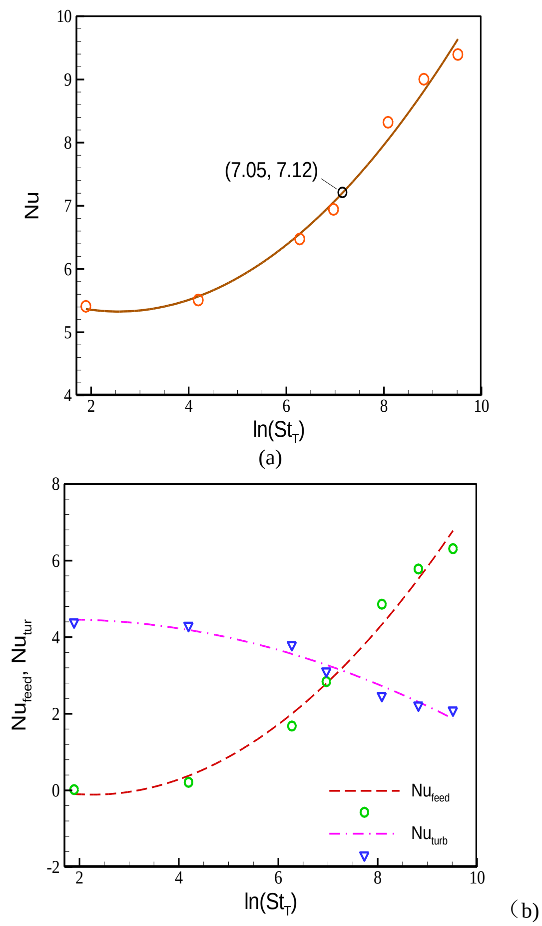


Figure 4. The relationship between Nusselt number and thermal stokes number: *(a)* Total Nusselt number; *(b)* Turbulent Nusselt number and Feedback Nusselt number

Fig. 5 shows the instantaneous fluid temperature fluctuations contours located at a distance $y^{+}=4.0$ from hot wall. The thermal streaks in the near-wall region are similar to the fluid velocity streaks. Due to the presence of the particles, the low-temperature streaks become stronger, more distinct and coherent. Simultaneously, the spanwise intervals between the high-temperature streak and the low-temperature streak become wider. Observing Fig.5, we notice that the thermal streaks is characterized by the high-temperature structures for case ($C_{p,p}/C_{p,f}$ =0.5), and yet the low-temperature streaks is mainly presented for ($C_{p,p}/C_{p,f}$ =8) and cases ($C_{p,p}/C_{p,f}$ =50).

SUMMARY AND CONCLUSIONS

In this paper, we performed a direct numerical simulation of particle-laden turbulent channel flow with heat transfer, and mainly studied in the influence of the inertial particles with different specific heat on the wall-normal heat flux in turbulent channel flow. A analysis is completed to reveal the important of full (momentum and thermal) two-way coupling models to investigate the effect on heat transfer produced by injection of inertia particles with high thermal capacity in turbulent flow.

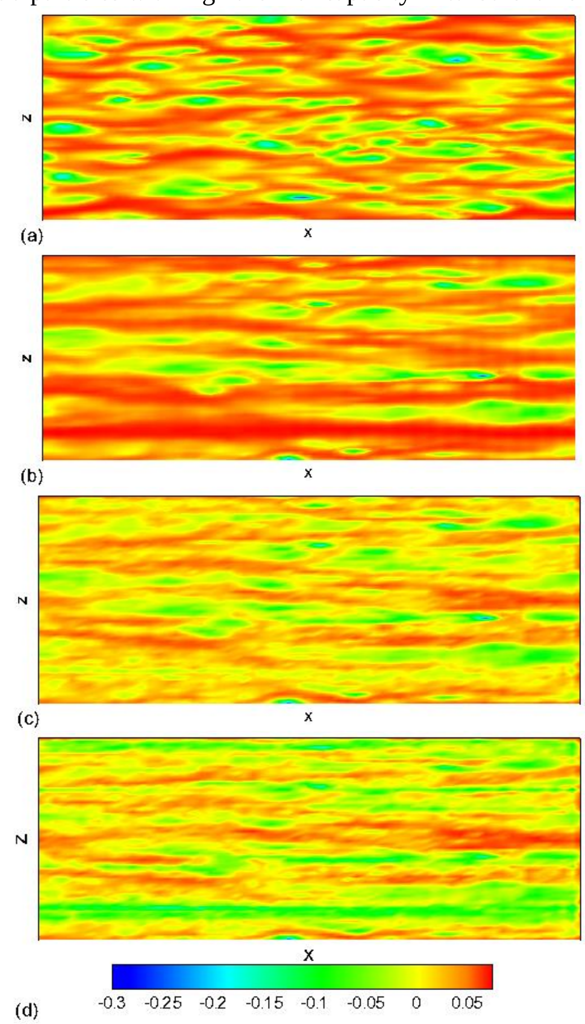


Figure 5. Instantaneous temperature fluctuations contours : *(a)*Unladen flow; *(b)* $c_{p,p}/c_{p,f}=0.5$; *(c)* $c_{p,p}/c_{p,f}=8$; *(d)* $c_{p,p}/c_{p,f}=50$.

The fluid temperature fluctuations and the heat fluxes are gradually decreased with the increasing of the particle specific heat. However, the thermal feedback of the particles is significant more. The reason for this is that the relative temperature between the particles and surrounding flow is approximately proportional to the particle specific heat. Thus, the thermal interaction between particles and turbulence temperature field becomes more severely as the particle specific heat larger. In addition, the Nusselt number is also affected due to the presence of the particles. The total Nusselt number and its particle feedback part (Nufeed) are monotonically increasing with the increase of the particle thermal stokes number, while an opposite trend is observed for the turbulent part. It means that thermal feedback heat flux plays a dominant role in total heat flux. When the particle thermal stokes number reaches a certain value, the ability of heat transfer is enhanced compared to unladen flow.

ACKNOWLEDGEMENT

This work was supported by the Natural Science Foundation of China (11272198).

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