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LARGE-EDDY SIMULATIONS OF MIXING HOT AND COLD FLUIDS IN TEE JUNCTIONS WITH/WITHOUT SINTERED POROUS MEDIUM FOR VARIOUS TEMPERATURE DIFFERENCES

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ABSTRACT

Large-eddy simulation is applied to simulate the mixing of the hot and cold fluids in a T-junction packed with periodic array of sintered cooper spheres based on the fluid-solid coupling method. The numerical results reveal that the thermal striping phenomenon is obvious and the temperature and velocity fluctuations are weakened in porous media region. Compared the results of cases with the same physical models and velocity conditions, the increasing temperature difference between the main inlet and the branch inlet does not impact on the hot and cold fluid flow and heat transfer obviously due to change the hot and cold fluid duct in T-junctions.

INTRODUCTION

T-junctions is main forms of pipeline system in petrochemical and nuclear power plants. The cross flow fluid with different properties (temperature) is mixing in T-junctions. This kind of thermal mixing is one of the causes of thermal fatigue failure. Thermal mixing characterizes the phenomenon where hot and cold flow streams join, mix and result in temperature fluctuations. The temperature fluctuations cause cyclic thermal stresses and subsequent fatigue cracking of the pipe wall[1]. Therefore, thermal fatigue of the structure around a T-junction is a technically important issue for the safety of petrochemical or nuclear power plants [2-7]. However, thermal fatigue is still not fully understood. One of the main obstacles to a full understanding resides in the multi-domain nature of the loading and associated damage, involving three complementary scientific disciplines: thermal-hydraulics, thermo-mechanics and materials science [8]. For the structural analysis of the thermal stress fluctuation and the generation of thermal fatigue cracks, it is necessary to know the temperature fluctuates in processes of mixing of hot and cold fluids in T-junctions. The amplitude and frequency of temperature fluctuation are two key parameters in evaluating the thermal fatigue of a T-junction.

Prediction of thermal fatigue in mixing tees is a challenging subject that is needed for life cycle management of piping systems. Recent studies have shown large-eddy simulations (LES) to be successful in predicting the mixing of hot and cold fluids in a tee. Zhu [4] focus on simulation of mixing processes in T-junctions using the large eddy simulation (LES) with Smagorinsky-Lilly of SGS model. The numerical normalized mean and root mean square temperatures for describing time-averaged temperature and temperature fluctuation intensity agree with experimental data. The results show that the LES is reliable. The effects of varying Reynolds number and Richardson number on the mixing course and thermal fluctuations were analyzed. Kuhn [5] concentrate on the numerical prediction of temperature fluctuations in a T-junction. The investigation includes the mixing in tees made of different materials and different pipe wall thicknesses. The influence of the wall thickness is represented as a damping effect on the temperature fluctuations in radial direction in the pipe wall. The comparison between available experimental data and the numerical results reveals a good agreement. Their study shows the capability of LES to predict thermal fluctuations in turbulent mixing. Lee [6] focus on numerical analyses of the temperature fluctuations and structural response of coolant piping at a mixing tee. The coolant temperature fluctuations obtained from LES are validated by experimental data. An assessment of the accuracy of LES predictions is made by Kuczaj et al. [9] for the applied Vreman SGS model through a direct comparison with the available experimental results. An estimation of the minimal mesh-resolution requirements for LES was examined on the basis of the complementary RANS simulations. They found that in order to obtain numerical solutions close to the experimental findings, the required mesh resolution must resolve the Taylor micro-scale length or should be of the order of Taylor micro-scale obtained from the RANS simulations. Benchmark studies were carried out by Hu and

Kazimi [10] using the LES turbulence model solved by the commercial CFD code FLUENT. Co-current and Collision types of mixing tee were modeled to evaluate the performance of the CFD code. Their simulation results presented in normalized average temperature and normalized fluctuating temperatures are in good agreement with measurements. Simoneau et al. [11] presented the use of LES for calculating turbulent flows in tee in the nuclear field. They focused on the knowledge of temperature or pressure fluctuations, required to address issues such as thermal fatigue or vibrations. The comparison with experimental results highlights the good behavior of LES, in terms of averaged fields, amplitude and frequency of fluctuations.

As above mentioned, Large-eddy simulation is a viable method to predict the fluid flow and heat transfer in the T-junction. Experimental analysis and numerical prediction of temperature fluctuations in processes of mixing of hot and cold fluids in T-junctions are important methods not only to estimate of the thermal fatigue of a T-junction, but also to seek to reduce the temperature fluctuations to improve the structural integrity. In recent years, many researchers are interested in investigations [12-20] on fluid flow and heat transfer in channels packed with porous media. The porous material improves the convective heat transfer due to the high surface area to volume ratio in the system and the enhanced flow mixing caused by the tortuous path through the porous matrix which improves the thermal dispersion [3].

The mathematical models of most studies of the flow and heat transfer in porous medium are based on the volume-averaging method popularized by Whitaker [21]. But the volume-averaging assumption does not reflect the influence of physical structure of porous medium on the fluid flow and heat transfer in porous medium. Fluid-solid coupling model for porous medium is expected to overcome the limitation of the volume-averaging assumption and really describe the relationship of fluid flow and heat transfer between the solid and fluid phases. Fluid flow and convective heat transfer of water in plate channels packed with uniformly sized sintered bronze particle was investigated numerically by Jiang and Lu [19] using solid-fluid coupling model. The permeability and inertia coefficient were calculated numerically according to the modified Darcy's model. The numerical calculation results are in agreement with well-known correlation results. The calculated local heat transfer coefficients on the plate channel surface, which agreed well with the experimental data. The convection heat transfer coefficients between the solid particles and the fluid and the volumetric heat transfer coefficients in the porous medium predicted by the numerical results increase with mass flow rate and decrease with increasing particle diameter. Kuwahara et al. [22] investigate that a flow through a periodic array of square cylinders using one of standard solid-fluid coupling models for a porous medium. The numerical results calculated by LES were processed to extract macroscopic results such as the macroscopic turbulent kinetic energy and the macroscopic pressure gradient. These macroscopic results are compared against those obtained using conventional models of turbulent kinetic energy and its dissipation rate, so as to examine the validity of extending the conventional two equation models of turbulence to the flow in porous medium.

In the present paper, Large-eddy simulation is applied to simulate the mixing of the hot and cold fluids in a T-junction packed with periodic array of sintered cooper spheres based on the fluid-solid coupling method. The normalized mean and root mean square temperatures were used to describe the time-averaged temperature and temperature fluctuation intensity, as well as velocity. The numerical results of cases with or without porous media were compared to reveal the effect of porous media weaken temperature and velocity fluctuations caused by mixing of hot and cold fluids in T-junctions. The numerical results of cases with different Richardson number were compared to reveal the influence of the temperature difference between hot and cold fluids on mixing process in T-junctions.

PHYSICAL MODEL AND GOVERNING EQUATIONS

Physical model

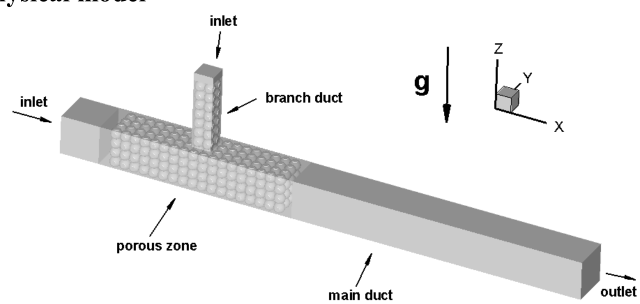


Figure 1 Schematic diagram of physical model

Schematic diagram of the mixing of hot and cold fluids in a tee with sintered copper sphere porous medium was shown in Figure 1. The main duct is arranged in horizontal direction, while the branch duct is arranged in vertical direction. The intersection of the centerline of the two ducts is assumed to be the coordinate system origin. The T-junction without porous medium is constructed of square ducts [2]. The main duct side, d_m, is 100 mm while the branch duct side, d_b , is 50 mm. The porous medium zone is in part of the main duct from x/d_b=-4.5 to x/d_b=4.5 and in part of the branch duct of z/d_b<4.5. The porous medium zone is packed homogeneous, uniform-sized solid particles made of the sintered copper sphere with diameter of 28 mm. The sintered copper sphere is periodic array of 18×4×4 in main duct and 7×4×4 in branch duct. The center distance between each two spheres is 25 mm. The porosity of the sintered porous medium is about 30.8%. Hot and cold water flows through the porous media, and the mixing region in the T-junction is from x=0 to the outlet in main duct.

Governing equations

The three-dimensional fluid flow and heat transfer problem in T-junction was solved using the CFD software FLUENT. The LES turbulence model was used in the numerical calculations for the turbulent flow with the sub-grid scale (SGS) Smagorinsky-Lilly (SL) model. The SIMPLE algorithm was used to couple the pressure and velocities. The second-order upwind advection model was used in the momentum, turbulent kinetic energy, and turbulent energy dissipation equations.

As shown in Figure 1, no-slip and adiabatic conditions were imposed at the walls as the velocity and thermal boundary conditions. The inlet velocity boundary conditions for both the main duct and the branch duct are specified as the inlet flow boundary conditions with the out flow boundary condition used at the main duct outlet. The operating pressure was ambient pressure, 101,325 Pa. The water properties such as the dynamic viscosity, the specific heat at constant pressure, and the thermal diffusivity were assumed to be constants in the simulations.

The filtered LES equations for isothermal incompressible flows with a passive scalar θ transport are [23-26]:

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho \overline{u}_i}{\partial x_i}=0 \tag{1}$$

$$\frac{\partial \rho \overline{u}_i}{\partial t}+\frac{\partial \rho \overline{u}_i \overline{u}_j}{\partial x_j}=-\frac{\partial \overline{p}}{\partial x_i}-\rho_0 \beta (T-T_0) g+\frac{\partial}{\partial x_j}\left(2\mu \overline{S}_{ij}-\tau_{ij}\right) \tag{2}$$

$$\frac{\partial \overline{\theta}}{\partial t}+\frac{\partial \overline{\theta} \overline{u}_j}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\alpha \frac{\partial \overline{\theta}}{\partial x_j}-q_j\right) \tag{3}$$

Where $\overline{u_i}$ are the velocity components, $\overline{p}$ is the pressure, ρ is the density, β is the thermal expansion coefficient, and ρ_0 is the reference density at the reference temperature, T_0. The molecular dynamic viscosity is denoted by μ. α is diffusivity.

The strain rate tensor is

$$\overline{S_{ij}}=\frac{1}{2}\left(\frac{\partial \overline{u_i}}{\partial x_j}+\frac{\partial \overline{u_j}}{\partial x_i}\right) \tag{4}$$

The sub-grid scale stress and scalar flux are

$$\tau_{ij}=\rho \overline{u_i u_j}-\rho \overline{u}_i \overline{u}_j \tag{5}$$

$$q_j=\overline{\theta u_j}-\overline{\theta}\overline{u}_j \tag{6}$$

The fluid mechanics are combined with the energy transport in the energy equation [27]

$$\frac{\partial \overline{T}}{\partial t}+\frac{\partial \left(\overline{u_i}\overline{T}\right)}{\partial x_i}=\frac{1}{R_e P_r}\frac{\partial^2 \overline{T}}{\partial x_i^2}-\frac{\partial q_i}{\partial x_i} \tag{7}$$

Where R_e represents the Reynolds number and P_r is the molecular Prandtl number. q_i represents sub-grid scale heat flux, which is modeled by the SGS model.

The energy equation of the sintered copper spheres is

$$\frac{\partial}{\partial t}(\rho c_p \overline{T})=\frac{\partial}{\partial x_i}(\lambda \frac{\partial \overline{T}}{\partial x_i}) \tag{8}$$

The buoyancy force due to the temperature difference is incorporated by adding a body force term to the Navier-Stokes equations based on the Boussinesq approximation. The effects of buoyancy are related to the Richardson number defined as

$$Ri=\frac{g\beta\left(T_{cold}-T_{hot}\right)d_b}{u_m^2} \tag{9}$$

The LES governing equations of fluid-solid coupling method are Equation (1), (2), (3) and (4). Equation (1) is the conservation of mass. Equation (2) is the Navier–Stokes equations which represent the momentum conservation. Equations (7) and (8) represent the energy conservation. The four governing equations for the flow and energy together boundary conditions were solved for the simulations.

NUMERICAL METHOD AND RESULTS

In the present paper, the inlet velocity of three cases of Case A, B and C is uniform shown in Table 1. The result of Case A is obtained by LES so as to compare to the previous experimental data without porous media [2]. The temperature difference, Δ T, where $\Delta T = T_b - T_m$, between inlet of branch duct and main duct of Case B and C with porous medium is -5.89 ℃ and 50 ℃ so as to reveal the effect of porous medium weaken temperature and velocity fluctuations and compare influence of temperature difference. Note that positive values of the temperature difference correspond to the case where the temperature at the branch inlet is higher than that of the main inlet, such as Case C in Table 1. On the contrary, the case with negative values of the temperature difference, such as Case B, is referred to the case where the temperature at the branch inlet is lower than that of the main inlet.

Table 1 Temperature and velocity of inlet of main and branch duct

	Main duct				Branch duct					
	Um(m/s)	Tm(℃)	Re	Pr	Ub(m/s)	Tb(℃)	Re	Pr	ΔT(℃)	Ri
Case A	0.1	16.89	8790	7.5	0.2	11	7500	8.8	-5.89	-0.093
Case B	0.1	16.89	8790	7.5	0.2	11	7500	8.8	-5.89	-0.093
Case C	0.1	16.89	8790	7.5	0.2	66.89	23200	2.6	50	0.517

The normalized mean temperatures and normalized temperature fluctuations are used to describe the time-averaged temperatures and time-averaged temperature fluctuation intensities [10, 28]. The normalized temperature at a given location of Case A and B is defined as

$$T_i^*=\frac{T_i-T_{cold}}{T_{hot}-T_{cold}} \tag{10}$$

where T_i is the instantaneous temperature at a given location, T_{cold} is the cold fluid inlet temperature, and T_{hot} is the hot fluid inlet temperature. The normalized temperature of Case A and B is zero at the main inlet and unity at the branch inlet, whereas that of Case C obtained using Equation (10) is contrary because the temperature of the main inlet of Case C is lower than that of branch duct. Therefore, The normalized temperature of Case C is defined as

$$T_i^* = -\frac{T_i - T_{hot}}{T_{hot} - T_{cold}} \quad (11)$$

The time-averaged normalized mean temperature at a given location is

$$\overline{T^*} = \frac{1}{N}\sum_{i=1}^{N} T_i^* \quad (12)$$

where N is the total number of sampling points.

The normalized temperature fluctuation is defined as the root-mean square (RMS) of the instantaneous temperature at a given location

$$T_{rms}^* = \sqrt{\frac{1}{N}\sum_{i=1}^{N}\left(T_i^* - \overline{T^*}\right)^2} \quad (13)$$

Similarly, the time-averaged normalized mean velocity and the normalized root mean square of velocity in the x direction is

$$\overline{u_x} = \frac{1}{N}\sum_{i=1}^{N}\frac{u_{i,x}}{u_{\overline{m,x}}} \quad (14)$$

$$u_{x,rms} = \sqrt{\frac{1}{N}\sum_{i=1}^{N}\left(\frac{u_{i,x}}{u_{\overline{m,x}}} - \overline{u_x}\right)^2} \quad (15)$$

where $u_{\overline{m,x}}$ is the mean velocity of different x sections in main duct.

Validation of LES model for predicting the mixing in T-junctions

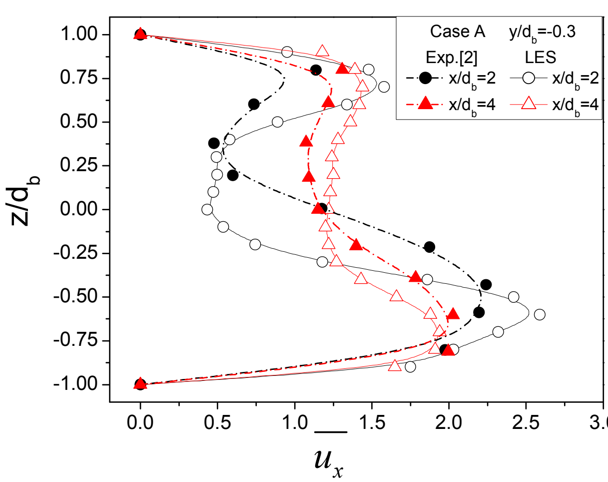


Figure 2 Normalized mean velocity distributions in different x planes at y/d$_b$ =-0.3

The instantaneous velocities of Case A at the intersections between the plane of y/d$_b$ = 0.3 and the plane of x/d$_b$ = 2 and 4 along the z direction were abstracted from the velocity fields in the T-junction using LES to compare with experimental data without the porous media [2]. The normalized mean velocity in x direction, $\overline{u_x}$, and the normalized velocity fluctuation, $u_{x,rms}$ were obtained using Equation (14) and (15). The predicted normalized mean velocity and normalized velocity fluctuations along these two lines are compared with the experimental data in Figure 2 and 3. The numerical results of Case A in Figure 2 for the normalized mean velocity at different intersections agree well with the experimental data, which verifies the validity of the LES model for predicting the mixing of hot and cold fluids in a tee junction.

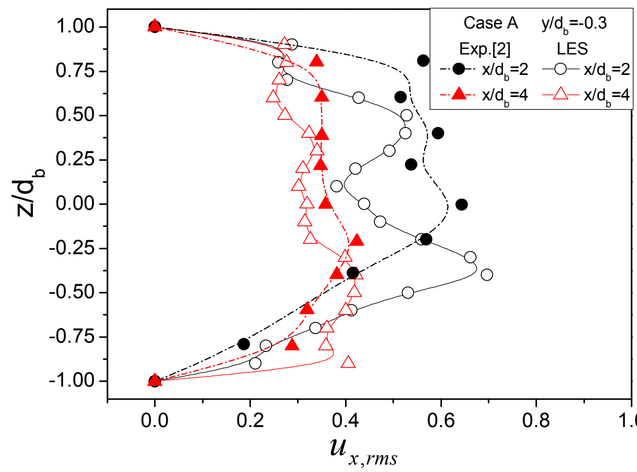
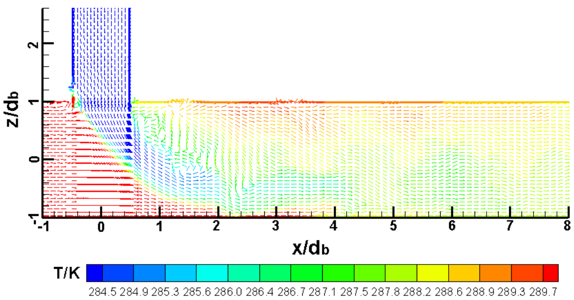


Figure 3 Normalized velocity fluctuations in different x planes at y/d$_b$ = -0.3

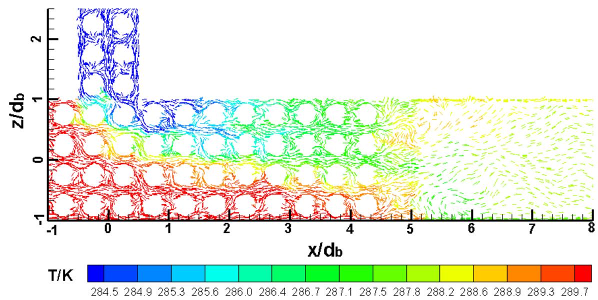
Effect of the porous medium

The temperature and velocity fields both with and without the porous media were calculated using the LES simulations with the SL SGS model. The combined temperature and velocity distributions on the y/d$_b$ = 0 plane at 13s are shown in Figure 4 for three cases of Case A, B and C. The temperature and velocity distributions of Case A in Figure 4(a) are quite different from those of Case B in Figure 4(b) due to effect of the porous medium. On the one hand, the thermal striping phenomena can be seen in Figure 4(a) and Figure 4(b). The cold fluid in branch duct of Case A injects into the main duct along about 45 degrees with x direction and is surrounded by the hot fluid. The situation of the hot and cold fluid in Figure 4(a) is similar to "deflecting jet" or "impact jet", while it in Figure 4(b) is similar to the "wall jet" [10] because the top wall of the main duct in the mixing region (0<x/d$_b$<2 and 0.6<z/d$_b$<1) is hold by the cold fluid. Three horizon layers with different temperature from top to bottom along the z direction in the region of porous medium of Case B is the cold fluid layer, the mixing layer and the hot layer, respectively. The temperature and velocity distribution with the porous medium is more uniform than that without the porous medium because the porous medium restricts the flow.

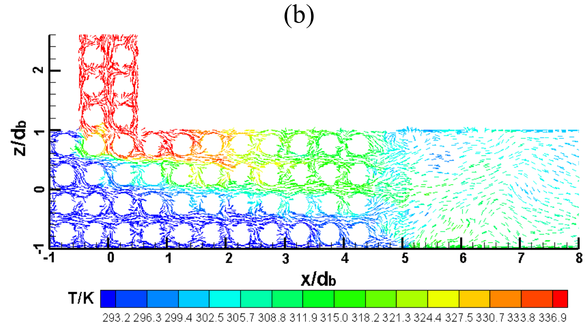
On the other hand, for Case B with the porous medium, compared with the temperature of the fluid around the sintered copper spheres, the temperature of the spheres is higher in the cold fluid layer, is uniform in the mixing layer, and is lower in hot fluid layer because of heat transfer from the hot fluid upstream and bottom to the cold fluid downstream and top resulting from the solid skeleton having a higher conductivity than the fluid.



(a)



(b)



(c)

Figure 4 Combined temperature and velocity vector distributions in the y/d_b=0 plane at 13s. (a) Case A without porous medium, *Ri*= -0.093; (b) Case B with porous medium *Ri*= -0.093; (c) Case C with porous medium, *Ri*= 0.517.

In addition, the numerical results of Case A show that the buoyancy influences the mixing of the hot and cold fluids in the tee junction in Figure 4(a), with the hot fluid in the main duct rising upwards into the branch duct and mixing with the cold fluid in the branch duct. But this phenomenon does not occur in Figure 4(b) and Figure 4(c) due to the higher flow resistance in the main duct packed with the porous media reducing the influence of the buoyancy.

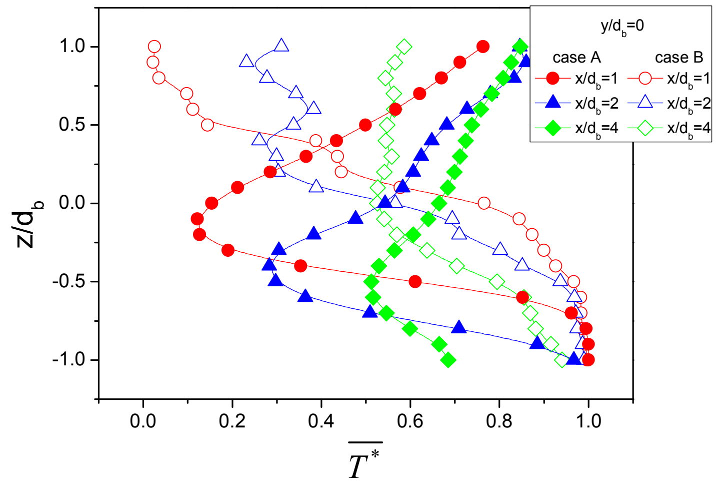


Figure 5 Normalized temperature distributions in different x planes at y/d_b = 0

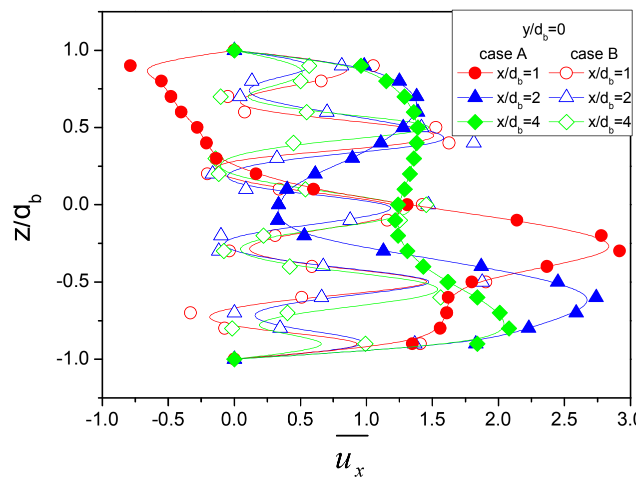
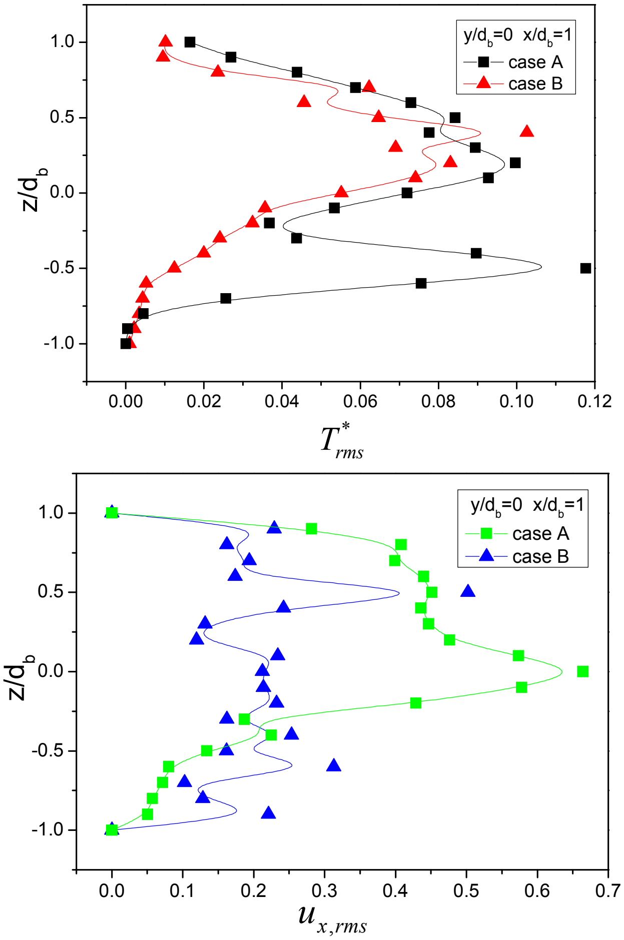
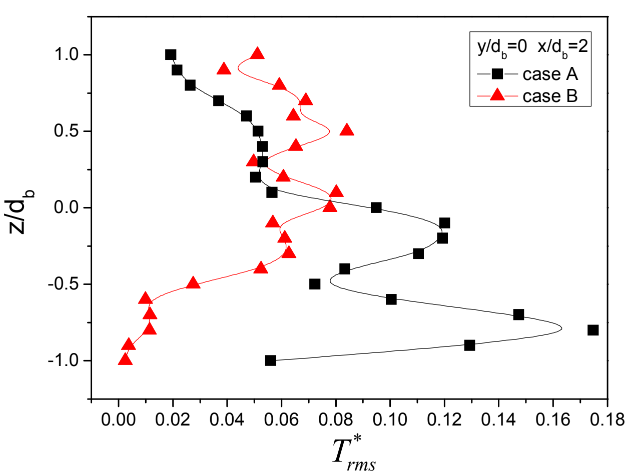


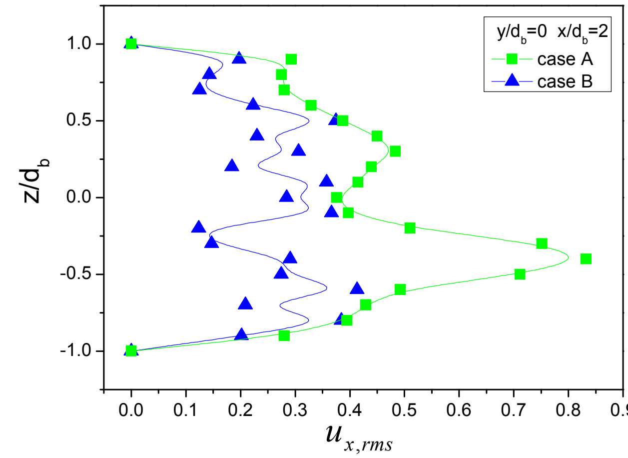
Figure 6 Normalized time-averaged velocity distributions in different x planes at y/d_b = 0

The normalized mean temperature and time-averaged velocity distributions in different x planes at y/d_b=0 for the two cases with and without porous media are quit different shown in Figure 5 and 6, respectively. From top to bottom along z direction, the normalized mean temperature of Case B with porous media is from 0 to 1, which indicates the temperature on top of the main duct is close to that of the cold fluid and the temperature on bottom of the main duct is close to that of the hot fluid. For Case A without porous media, however, the temperatures on top and bottom of the main duct are both higher, and that on the middle height is lowest. The temperature distributions for the case with the porous medium are smoother than those in the absence of the porous medium, which indicates that the high conductivity of the sintered porous medium affords a more uniform temperature. Comparison of the velocity distributions of two cases, that of the Case B with the porous medium is more regular than Case A without porous medium in Figure 6. The velocity distributions at three intersections on the plane y/d_b = 0 for the Case B are similar to each other. The three velocity curves are zigzag lines with five peaks and four troughs corresponding to the location of the clearance and the sintered sphere, because the sintered sphere reduce the flow area and change the flow direction. The numerical results also illustrate that it is really describe influence of physical structure of porous medium on the fluid flow in porous medium using the fluid-solid coupling method.

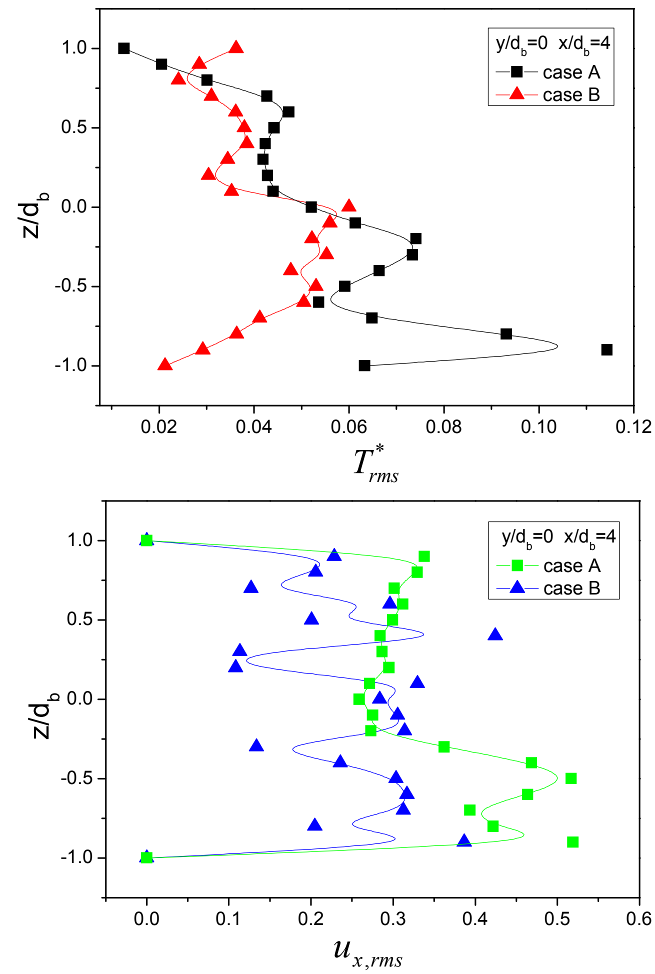


(a)





(b)



(c)

Figure 7 Normalized temperature and velocity fluctuations in different x planes at y/db = 0. (a) x/db=1; (b) x/db= 2; (c) x/db=4.

In Figure 7, the normalized temperature and velocity fluctuations with the porous media in all three planes of x/d_b=1, 2 and 4 are less than without the porous media, which shows that the porous media effectively reduces the temperature and velocity fluctuations for hot and cold mixing in the T-junction. The largest normalized temperature fluctuations at each location near the middle height are reduced as the flow moves downward, especially with the porous media. Compared to Case A without porous media, the change region of the normalized velocity fluctuations with porous media is more uniform about 0.2 to 0.4. Furthermore, the fluctuation maximum are located in z/d_b=0.5, 0, -0.5 corresponding to the location of maximum flow area in porous media, which is similar to the normalized velocity distributions in Figure 6.

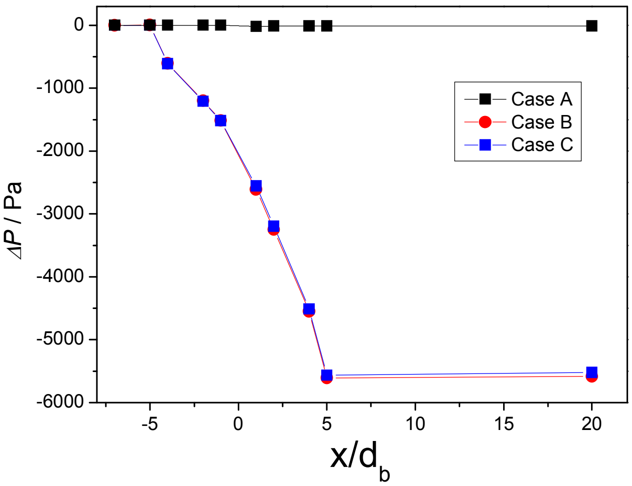


Figure 8 Mean pressure drop in x direction

Figure 8 shows the area averaged pressure drops in the x direction for three cases. The pressure drop of Case B and C with the porous media in the mixing zone is much greater due to the flow resistance in the porous media than that of Case A without the porous media. The pressure of Case B and C decreases rapidly in the porous media region ($-4.5<x/d_b<4.5$), while the pressure remains constant in the region without porous media ($-4.5> x/d_b$ or $x/d_b<4.5$).

Comparison of the temperature fluctuations for the different Richardson number

The effects of buoyancy are related to the Richardson number. *Ri* number defined by Equation (9) increases with the temperature difference increasing when the duct diameter and velocity are uniform. Figure 4(b) and (c) shows the temperature and velocity distributions of Case B, *Ri*= -0.093, and Case C, *Ri*= 0.517. The temperature and velocity distributions of two cases are very similar because the same physical models with porous media and velocity conditions of both cases are apply to calculation using LES model.

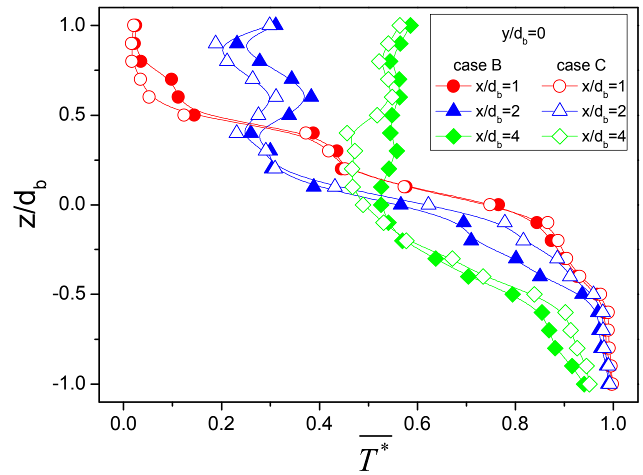


Figure 9 Normalized temperature distributions in different x planes at $y/d_b = 0$

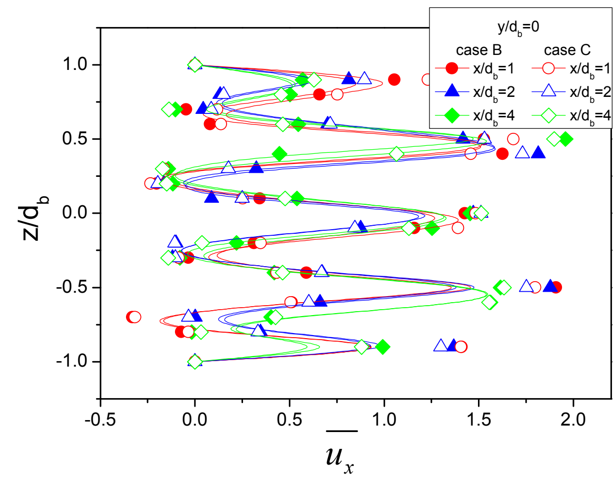


Figure 10 Normalized time-averaged velocity distributions in different x planes at $y/d_b = 0$

The normalized temperature of Case B in Figure 9 is zero at the main inlet and unity at the branch inlet, whereas that of Case C obtained using Equation (10) is contrary because the temperature of the main inlet of Case C is lower than that of branch duct. Therefore, the normalized temperature of Case C in Figure 9 is defined as Equation (11). As for its physical significance of the normalized temperature of two cases in Figure 9, zero or unity mean that the temperature of sampling point is equal to the temperature of the main duct inlet or branch duct inlet, respectively.

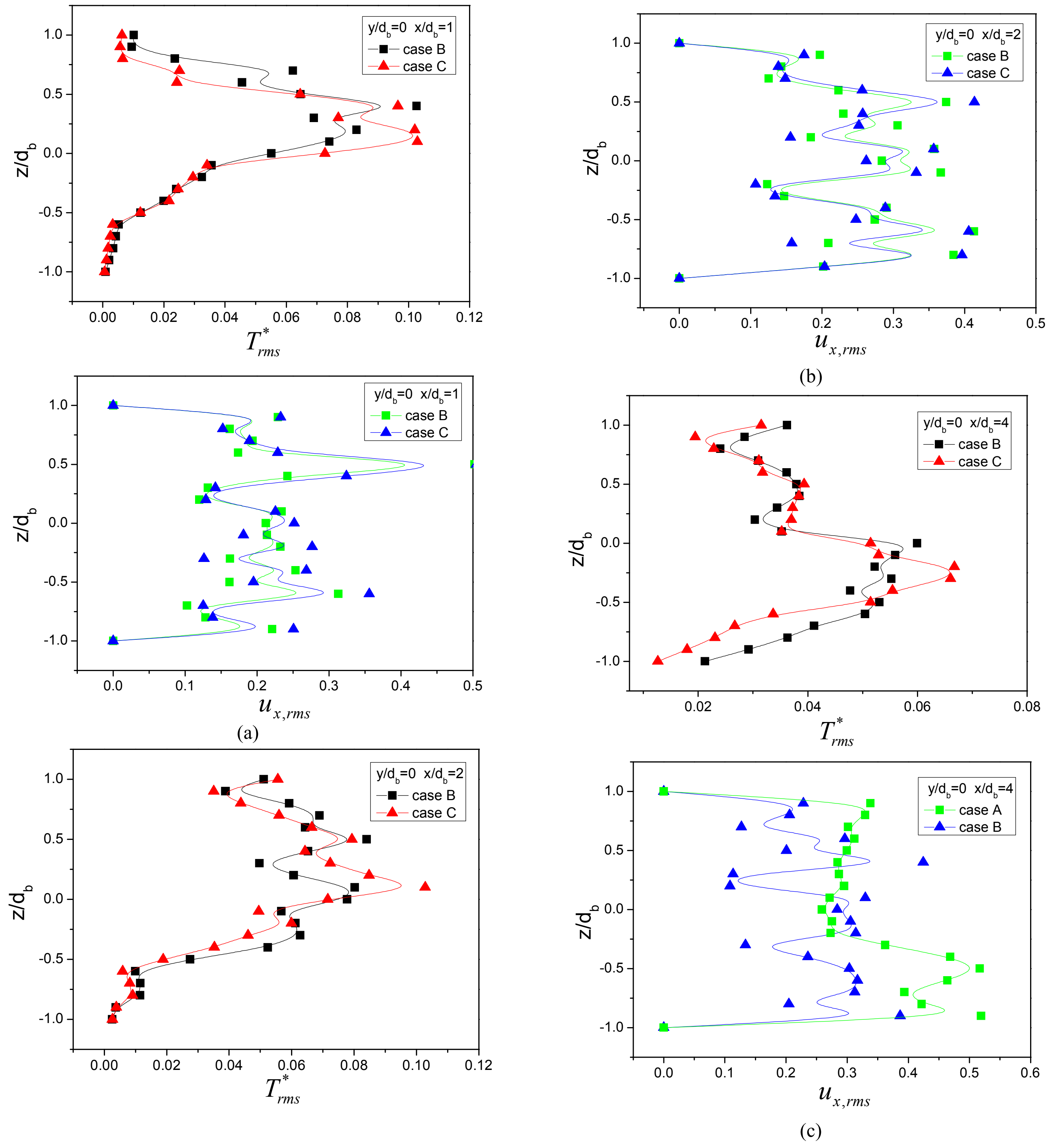


Figure 11 Normalized temperature and velocity fluctuations in different x planes at $y/d_b = 0$. (a) $x/d_b=1$; (b) $x/d_b= 2$; (c) $x/d_b=4$.

Moreover, compared Case B with Case C, the mean pressure drop, the normalized temperature, the normalized time-averaged velocity, normalized temperature and velocity fluctuations in different x planes in Figure 8, Figure 9, Figure 10 and Figure 11 are similar. The numerical results with the same physical models and velocity conditions indicate that the increasing temperature difference does not impact on the hot and cold fluid flow and heat transfer obviously due to change the hot and cold fluid duct in T-junctions.

CONCLUSIONS

Three cases of the temperature and velocity field in T-junctions with and without porous media and different inlet temperature are calculated using LES and the sub-grid scale (SGS) Smagorinsky-Lilly (SL) model. Fluid-solid coupling method is used to establish the governing equations in porous medium. The temperature and velocity distributions were predicted, and the normalized mean and fluctuating temperatures were analyzed. The numerical results showed that:

(1) The LES model can capture the instantaneous turbulent fluctuations for the flow without the porous media in the mixing tee as shown by the good agreement between the numerical results with the corresponding experimental data.

(2) In the porous media region, the temperature and velocity fluctuations are weakened in the T-junction and the thermal striping phenomena is obvious that is similar to the “wall jet”. With fluid flow out of the porous media region, the layer disappears and temperature and velocity fluctuations trend to increase.

(3) Compared the results of cases with the same physical models and velocity conditions, the increasing temperature difference between the main inlet and the branch inlet does not impact on the hot and cold fluid flow and heat transfer obviously due to change the hot and cold fluid duct in T-junctions.

(4) The numerical results reflect the influence of the physical structure and thermal conductivity of sintered copper spheres on the fluid flow and heat transfer in porous medium using the fluid-solid coupling method. This method can be used to further study of the mixing of hot and cold fluid in T-junctions for structural properties, such as diameter, porosity, shape, array of the porous medium.

ACKNOWLEDGEMENTS

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REFERENCES

[1] Naik-Nimbalkar, V.S., A.W. Patwardhan, I. Banerjee, G. Padmakumar, and G. Vaidyanathan, Thermal mixing in T-junctions, *Chemical Engineering Science*, Vol. 22, 2010, pp. 5901-5911.

[2] Fukushima, N., K. Fukagata, and N. Kasagi. Numerical and experimental study on turbulent thermal mxing in a t-junction flow, in *The 6th ASME-JSME Thermal Engineering Joint Conference*, Hawaii, USA, 2003.

[3] Lu, T., P.X. Jiang, Z.J. Guo, Y.W. Zhang, and H. Li, Large-eddy simulations (LES) of temperature fluctuations in a mixing tee with/without a porous medium, *International Journal of Heat and Mass Transfer*, Vol. 21-22, 2010, pp. 4458-4466.

[4] Zhu, W.-y., T. Lu, P.-x. Jiang, Z.-j. Guo, and K.-s. Wang, Large eddy simulation of hot and cold fluids mixing in a T-junction for predicting thermal fluctuations, *Applied Mathematics and Mechanics*, Vol. 11, 2009, pp. 1379-1392.

[5] Kuhn, S., O. Braillard, B. Niceno, and H.-M. Prasser, Computational study of conjugate heat transfer in T-junctions, *Nuclear Engineering and Design*, Vol. 6, 2010, pp. 1548-1557.

[6] Lee, J.I., L.-w. Hu, P. Saha, and M.S. Kazimi, Numerical analysis of thermal striping induced high cycle thermal fatigue in a mixing tee, *Nuclear Engineering and Design*, Vol. 5, 2009, pp. 833-839.

[7] Frank, T., C. Lifante, H.M. Prasser, and F. Menter, Simulation of turbulent and thermal mixing in T-junctions using URANS and scale-resolving turbulence models in ANSYS CFX, *Nuclear Engineering and Design*, Vol. 9, 2010, pp. 2313-2328.

[8] Chapuliot, S., C. Gourdin, T. Payen, J.P. Magnaud, and A. Monavon, Hydro-thermal-mechanical analysis of thermal fatigue in a mixing tee, *Nuclear Engineering and Design*, Vol. 5, 2005, pp. 575-596.

[9] Kuczaj, A.K., E.M.J. Komen, and M.S. Loginov, Large-Eddy Simulation study of turbulent mixing in a T-junction, *Nuclear Engineering and Design*, Vol. 9, 2010, pp. 2116-2122.

[10] Hu, L.-W. and M.S. Kazimi, LES benchmark study of high cycle temperature fluctuations caused by thermal striping in a mixing tee, *International Journal of Heat and Fluid Flow*, Vol. 1, 2006, pp. 54-64.

[11] Simoneau, J.-p., J. Champigny, and O. Gelineau, Applications of large eddy simulations in nuclear field, *Nuclear Engineering and Design*, Vol. 2, 2010, pp. 429-439.

[12] Huang, P.C. and K. Vafai, Analysis of forced convection enhancement in a channel using porous blocks, *Journal of Thermophysics and Heat Transfer*, Vol. 3, 1994, pp. 563-573.

[13] Zhang, J.M., W.H. Sutton, and F.C. Lai, Enhancement of heat transfer using porous convection-to-radiation converter for laminar flow in a circular duct, *International Journal of Heat and Mass Transfer*, Vol. 1, 1996, pp. 39-48.

[14] Ould-Amer, Y., S. Chikh, K. Bouhadef, and G. Lauriat, Forced convection cooling enhancement by use of porous materials, *International Journal of Heat and Fluid Flow*, Vol. 3, 1998, pp. 251-258.

[15] Huang, P.C., C.F. Yang, J.J. Hwang, and M.T. Chiu, Enhancement of forced-convection cooling of multiple heated blocks in a channel using porous covers, *International Journal of Heat and Mass Transfer*, Vol. 3-4, 2005, pp. 647-664.

[16] Zhang, B. and Y. Zhao, A numerical method for simulation of forced convection in a composite porous/fluid system, *International Journal of Heat and Fluid Flow*, Vol. 4, 2000, pp. 432-441.

[17] Alvarez, G., P.-E. Bournet, and D. Flick, Two-dimensional simulation of turbulent flow and transfer through stacked spheres, *International Journal of Heat and Mass Transfer*, Vol. 13, 2003, pp. 2459-2469.

[18] Jiang, P.-X. and X.-C. Lu, Numerical simulation and theoretical analysis of thermal boundary characteristics of convection heat transfer in porous media, *International Journal of Heat and Fluid Flow*, Vol. 5, 2007, pp. 1144-1156.

[19] Jiang, P.-X. and X.-C. Lu, Numerical simulation of fluid flow and convection heat transfer in sintered porous plate channels, *International Journal of Heat and Mass Transfer*, Vol. 9-10, 2006, pp. 1685-1695.

[20] Jiang, P.-X., M. Li, T.-J. Lu, L. Yu, and Z.-P. Ren, Experimental research on convection heat transfer in sintered porous plate channels, *International Journal of Heat and Mass Transfer*, Vol. 10-11, 2004, pp. 2085-2096.

[21] Whitaker, S., Simultaneous Heat, Mass, and Momentum Transfer in Porous Media: A Theory of Drying, *Advances in Heat Transfer*, Vol. 13, 1977, pp. 119-203.

[22] Kuwahara, F., T. Yamane, and A. Nakayama, Large eddy simulation of turbulent flow in porous media, *International*

Communications in Heat and Mass Transfer, Vol. 4, 2006, pp. 411-418.
[23] Wegner, B., Y. Huai, and A. Sadiki, Comparative study of turbulent mixing in jet in cross-flow configurations using LES, *International Journal of Heat and Fluid Flow*, Vol. 5, 2004, pp. 767-775.
[24] Temmerman, L., M.A. Leschziner, C.P. Mellen, and J. Fröhlich, Investigation of wall-function approximations and subgrid-scale models in large eddy simulation of separated flow in a channel with streamwise periodic constrictions, *International Journal of Heat and Fluid Flow*, Vol. 2, 2003, pp. 157-180.
[25] Pope, S.B., *Turbulence Flow*, 2000, Cambridge: Cambridge University Press.
[26] Majander, P. and T. Siikonen, Large-eddy simulation of a round jet in a cross-flow, *International Journal of Heat and Fluid Flow*, Vol. 3, 2006, pp. 402-415.
[27] Dong, Y.-H. and X.-Y. Lu, Large eddy simulation of a thermally stratified turbulent channel flow with temperature oscillation on the wall, *International Journal of Heat and Mass Transfer*, Vol. 10-11, 2004, pp. 2109-2122.
[28] Kimura, K., Thermal striping in mixing tees with hot and cold water (Type A: Characteristics of flow visualization and temperature fluctuations in collision type mixing tees with same pipe diameter), in *NTHAS3: Third Korea-Japan symposium on Nuclear Thermal Hydraulics and Safety*, Korea, 2002.