International Journal of Advanced Culture Technology Vol.3 No.2 171-178 (2015)
http://dx.doi.org/10.17703/IJACT.2015.3.2.171

IJACT 15-2-20

Numerical Simulation of Thermal Fluctuation of Hot and Cold Fluids Mixing in a Tee Junction

Kai Gao, Tao Lu

School of Mechanical and Electrical Engineering, Beijing University of Chemical Technology, Beijing, China

ABSTRACT

In this work, mixing processes of hot and cold fluids of three different jet types are predicted by large-eddy simulation (LES) on FLUENT platform. Temperature at different positions of internal wall and mixing conditions of T-junctions at different times are obtained, then the simulated normalized mean and root-mean square (RMS) temperature, temperature contour and velocity vector of every case are compared. The results indicate that, the mixing regions in the tee junction is related to the jet type, and temperature fluctuations on the pipe wall in the type of the deflecting jet is the least.

Keywords: *tee junction, mixing, thermal fluctuation, large-eddy simulation*

1. INTRODUCTION

Tee junction is a familiar structure that is universally used in pipeline systems of power plants, nuclear power plants and chemical plants, it often applied to mix hot and cold fluids of main and branch pipes. Because of the temperature difference between two fluids, it results in temperature fluctuations near the piping wall that could cause fluctuant thermal stresses in piping material and subsequent thermal fatigue cracking of the pipes when hot and cold fluids mix in the T-pipe. So far, leakage accidents took place in several light water and sodium cooled reactors due to thermal fatigue. In December 1987, a leakage occurred in high pressure line of Farley nuclear power plant, USA; In June 1988, another accident happened in high pressure safety injection (HPSI) pipeline in Tihange, Belgium[1]; In 1990, sodium leakage happened in the French reactor Superphenix[2]. Therefore, it is significant to study the thermal fatigue of the T-junctions to ensure ordinary running of plants and structural integrity of pipeline systems.

In the analysis of thermal fatigue, temperature fluctuation is a very important evaluation parameter. Kamaya and Nakamura[3] used the transient temperature obtained by simulation to assess the distribution of thermal stress and fatigue when cold fluid flowed into the main pipe from a branch pipe. Numerical simulation of flow in the tee has been carried out by Simoneau et al.[4] to get temperature and its fluctuation curves, and the numerical results were in good agreement with the experimental data. Through the analysis on thermal fatigue stress, it draw the conclusion that the enhanced heat transfer coefficient and the

Manuscript Received: Aug. 22, 2015 / Revised: Oct. 20, 2015 / Accepted: Nov. 25, 2015
Corresponding Author: zhaihao2010@126.com
Tel: +86 10 6443 4820
School of Mechanical and Electrical Engineering, Beijing University of Chemical Technology, Beijing, China

temperature difference between hot and cold fluids were primary factors of thermal fatigue failure of tees.

Many numerical simulations and experiments have been carried out to evaluate the flow and heat transfer in a mixing tee junction. Minoru et al.[5] depended on the momentum ratio and divided jet flow into four categories, among which fluid temperature fluctuation intensity had the maximum in the downstream regions near the piping wall in the cases of wall jet and re-attachment jet. Kimura et al.[6] took advantage of thermocouples and particle image velocimetry for the sake of fluid mixing processes in the tee in the cases of wall jet, impinging jet and deflecting jet.

Large-eddy simulation (LES) is an effective method to predict temperature fluctuation in numerical simulations. Thermal striping phenomena in the tee junction had been numerically investigated using LES by Hu and Kazimi[7] for two different mixing cases, and the simulated normalized mean and root-mean square (RMS) was consistent with experimental results. Large-eddy simulation have been used by Lu et al.[8] to predict the fluctuations of temperature and velocity in a mixing T-junction to understand the cause of thermal fatigue.

As mentioned above, The results of previous researches provide a good reference value for this work that analyses the mixing of fluids and the thermal fluctuation of piping wall by large-eddy simulation. In this work, mixing processes of hot and cold fluids in three different jet types are predicted, then the simulated normalized mean and root-mean square (RMS) temperature, temperature contour and velocity vector of every case are analyzed.

2. PHYSICAL MODEL AND NUMERICAL SIMULATION

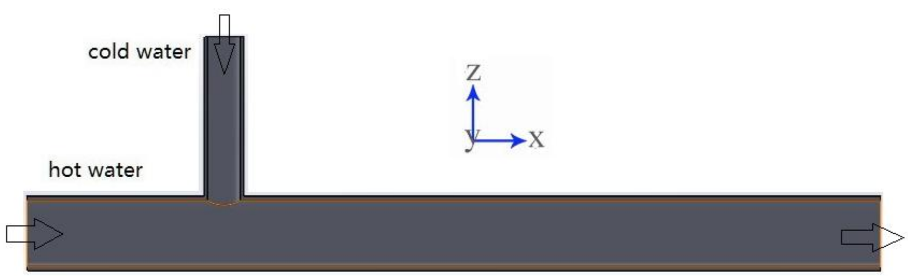


Figure 1. Physical model of the tee junction

Figure 1 shows the physical model of the tee junction, the diameters of the main pipe and branch pipe are respectively 100mm and 50mm. The inlet length of the main pipe is 300mm and the downstream length is 1000mm, the inlet length of the branch pipe is 300mm. Hot water at a temperature of 333.15K enters the inlet of the main pipe while cold water in the branch pipe is 288.15K. The direction of gravity is opponent to the flow of cold water . The velocities of main pipe and branch pipe of different cases are shown in Table I.

Table 1. Parameter of simulated cases

	Velocity of the main pipe (m/s)	Velocity of the branch pipe (m/s)	Jet type
Case 1	1.0	0.5	Wall jet
Case 2	0.5	1.0	Deflecting jet
Case 3	0.5	2.0	Impinging jet

For incompressible flow, the filtered Navier-Stokes and energy equations can be written as:

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

$$\frac{\partial \rho \bar{u}_i}{\partial t}+\frac{\partial \rho \bar{u}_i \bar{u}_j}{\partial x_j}=-\frac{\partial \bar{p}}{\partial x_i}-\rho_0 \beta (T-T_0) g+\frac{\partial}{\partial x_j}\left[(\mu+\mu_t)\left(\frac{\partial \bar{u}_i}{\partial x_j}+\frac{\partial \bar{u}_j}{\partial x_i}\right)\right] \tag{2}$$

$$\frac{\partial \rho \bar{T}}{\partial t}+\frac{\partial \rho \bar{T} \bar{u}_j}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\frac{\lambda}{c_p}\frac{\partial \bar{T}}{\partial x_j}-\rho \overline{T'' u_j''}\right) \tag{3}$$

In these equations, ρ, β, μ, μ_t, λ and c_p represent the density, thermal expansion coefficient, molecular viscosity, turbulent viscosity, thermal conductivity and specific heat capacity, respectively. The Smagorinsky-Lilly model is used for the turbulent viscosity, which is described as:

$$\mu_t=\rho L_s^2 \left|\bar{S}\right| \tag{4}$$

Where L_s is the mixing length of the subgrid scales. L_s and $\left|\bar{S}\right|$ are computed by equations：

$$L_s=\min\left(kd, C_s V^{1/3}\right) \tag{5}$$

$$\left|\bar{S}\right|=\sqrt{2\bar{S}_{ij}\bar{S}_{ji}} \tag{6}$$

$$\bar{S}_{ij}=\frac{1}{2}\left(\frac{\partial \bar{u}_i}{\partial x_j}+\frac{\partial \bar{u}_j}{\partial x_i}\right) \tag{7}$$

Where k is the Von Karman constant of 0.42; d is the distance to the closest wall; C_s is the Smagorinsky constant of 0.1; V is the volume of the computational cell.

During the calculation, the steady result of flow field and heat transfer are obtained by Reynolds stress model (RSM) firstly, and then set them as the initial condition for the LES calculation. The transient temperature of internal wall is extracted with the frequency of 1000Hz, the distribution of sampling points are shown in Figure 2.

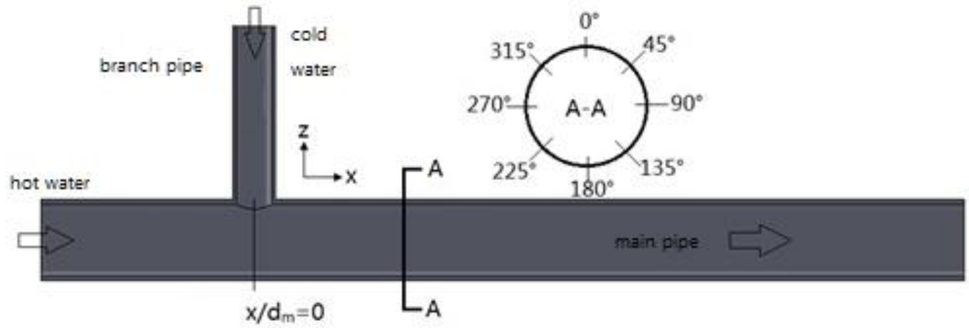


Figure 2. The distribution of sampling points on the planes

3. RESULTS AND DISCUSSION

The normalized mean and root-mean square temperature are used to describe the time-averaged temperature and temperature fluctuation intensity. The normalized temperature is defined as:

$$T_i^* = \frac{T_i - T_c}{T_h - T_c} \tag{8}$$

N is the total number of sample times.

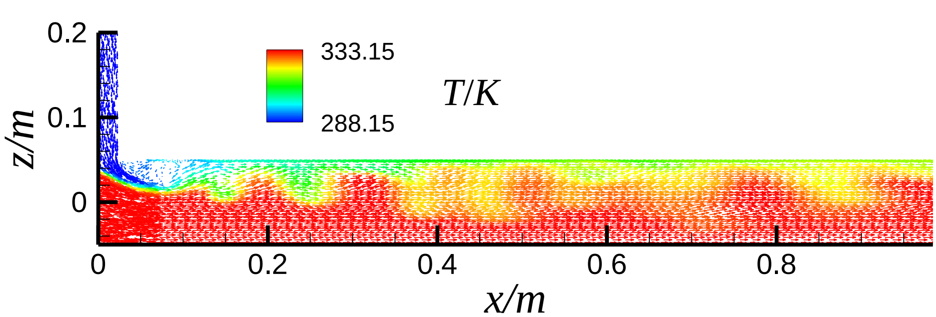
$$\overline{T^*} = \frac{1}{N}\sum_{i=1}^{N} T_i^* \tag{9}$$

Where T_i is the transient temperature, T_c is the cold fluid inlet temperature and T_h is the hot fluid inlet temperature.

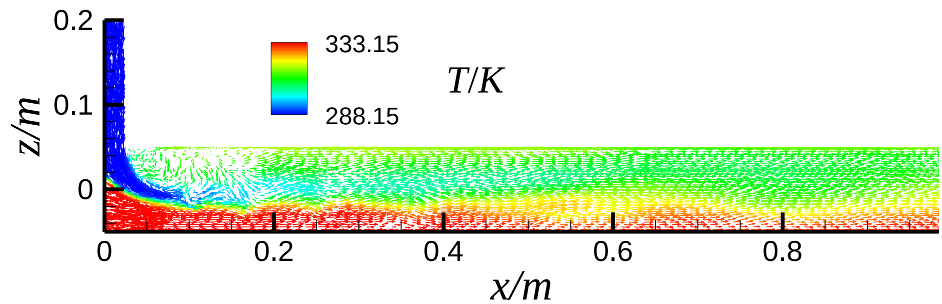
The root-mean square (RMS) of the normalized temperature is defined as:

$$T_{rms}^* = \sqrt{\frac{1}{N}\sum_{i=1}^{N}\left(T_i^* - \overline{T^*}\right)^2} \tag{10}$$

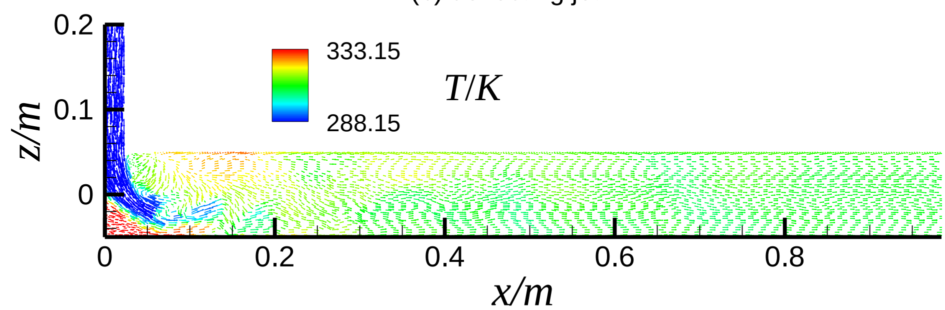
3.1 Velocity vector and temperature contour



(a) wall jet



(b) deflecting jet

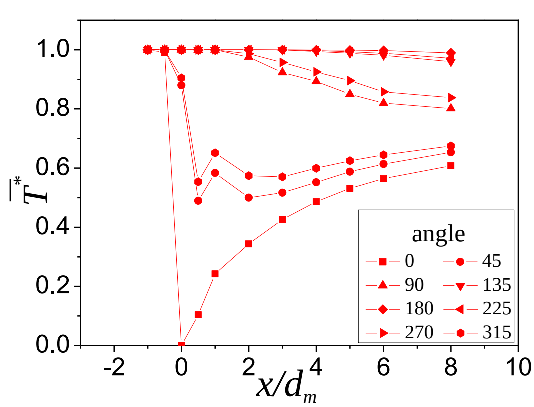


(c) impinging jet

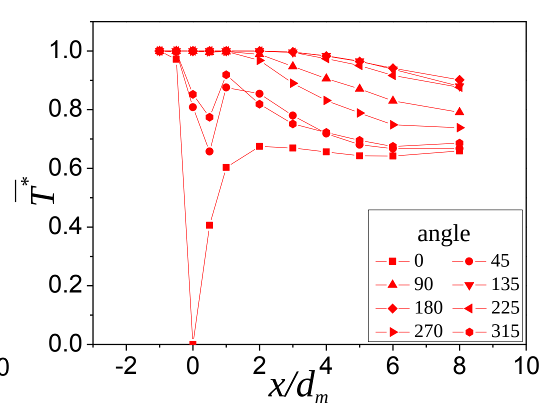
Figure 3. Temperature contour and velocity vector in the plane of y/dm= 0 in different cases

Figure 3 shows the temperature contour and velocity vector in the plane of y/d_m= 0 of different cases. As is shown in the illustration, in the wall jet, the mixing of two fluids concentrates in the upper part of the main pipe, while for the deflecting jet it moves to the middle part and for the impinging jet the mixing occurs in the lower part. It is because with the increasing of the momentum of the branch pipe, the effect of the horizontal momentum of the main pipe decreases, the branch fluid could reach to the deeper regions of the main pipe.

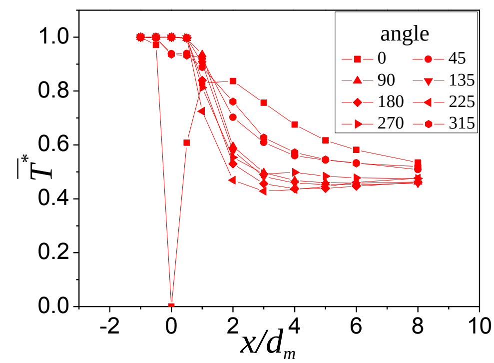
3.2 Normalized mean and RMS temperature



(a) wall jet



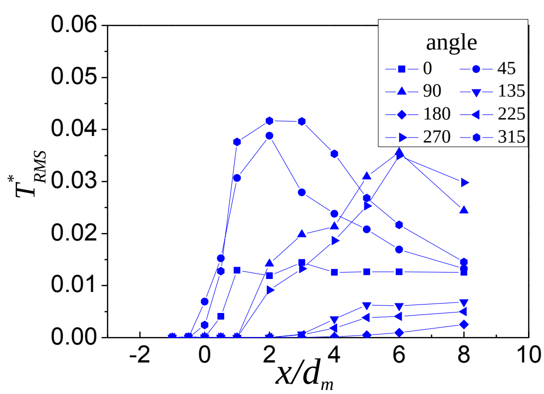
(b) deflecting jet



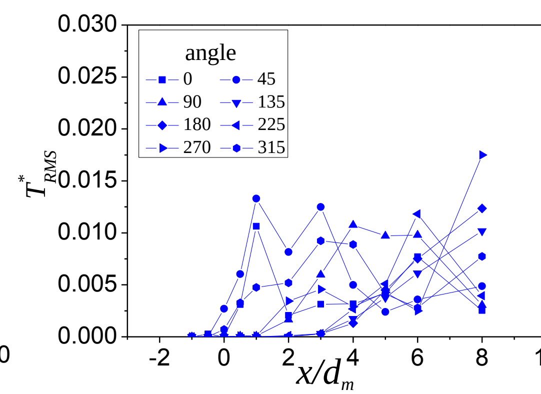
(c) impinging jet

Figure 4. Normalized mean temperatures of different cases

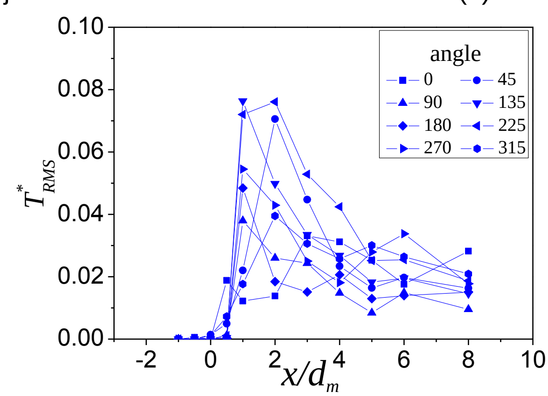
Figure 4 shows the normalized mean temperatures of sampling points of different cases. In the wall jet, the temperature variations along the direction of x axis is larger at the angles of 0°，45° and 90°, these angles are located in the upper part of the main pipe wall. In the case of deflecting jet, the temperature variations of 0°，45° and 90° decrease while the temperature variations of other angles increase, it shows that the mixing of hot and cold fluids moves downward because of the larger velocity of branch pipe. For the impinging jet, the larger temperature variations occurs at the angles of the lower part of the main pipe wall, they are from 90° to 270°. With the increasing of the axial length, the mean temperature difference of each angle decrease, it shows the mixing of fluids becomes fuller.



(a) wall jet



(b) deflecting jet



(c) impinging jet

Figure 5. Normalized RMS temperatures of different cases

Figure 5 shows the normalized RMS temperatures of sampling points of different cases. In these pictures, it can be seen that there are temperature fluctuations at every point because of the mixing of fluids, but the positions of the maximum value of each case is different. For the wall jet, it appears at the angles of 45° and 315° that are located in the upper part of the main pipe wall, while in the case of deflecting jet, it occurs at the point of 270° that is on the central plane. For the impinging jet, the maximum temperature fluctuations are mainly in the area of the lower part such as 135° and 225°. By comparing the above three figures, it can be seen the temperature fluctuations of the deflecting jet is lower than the others, it is because the mixing of the deflecting jet is concentrates in the internal, so there are less effect of the mixing on the pipe wall.

4. CONCLUSION

When hot and cold fluid mix in tee junctions in industrial pipeline systems, it results in temperature fluctuation near the piping wall that could cause thermal fatigue. In this work, mixing processes of hot and cold fluids in the tee junction in three cases are predicted by large-eddy simulation (LES). The simulated normalized mean and root-mean square (RMS) temperature, velocity vector and temperature contour of every case are compared. Conclusions are as follows:

1. The mixing regions are different with the change of jet type in the tee junction, it moves to the lower part of the main pipe as the velocity of the branch pipe increases.
2. There are temperature fluctuations on the pipe wall because of the mixing of fluids and the effect of the mixing on the pipe wall in the type of the deflecting jet is the least.

ACKNOWLEDGEMENTS

This work was supported by projects of the National Natural Science Foundation of China (Project No. 51276009), the Program for New Century Excellent Talents in Universities (No. NCET-13-0651), and the National Basic Research Program of China (Project No. 2011CB706900).

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