

The Vibration Analysis of Tube Bundles Induced by Fluid Elastic Excitation in Shell Side of Heat Exchanger

Minle Bao¹, Lu Wang¹, Wenyao Li¹ and Tianze Gao¹

1 Key Laboratory of Ocean Energy Utilization and Energy Conservation of Ministry of Education, Dalian University of Technology, 116023, China
E-mail: baominle@hotmail.com

Abstract. Fluid elastic excitation in shell side of heat exchanger was deduced theoretically in this paper. Model foundation was completed by using Pro / Engineer software. The finite element model was constructed and imported into the FLUENT module. The flow field simulation adopted the dynamic mesh model, RNG k- ϵ model and no-slip boundary conditions. Analysing different positions vibration of tube bundles by selecting three regions in shell side of heat exchanger. The results show that heat exchanger tube bundles at the inlet of the shell side are more likely to be failure due to fluid induced vibration.

1. Introduction

Heat exchanger is one of the main industrial equipment which can improve energy efficiency and achieve heat exchange. It is widely used in various industrial fields, playing an important role in the production [1]. With the rapid development of industrial production and the continuous expansion of production scale, heat exchanger has been toward the development of large-scale and high parameterized [2-4]. But fluid induced vibration has become more and more obvious, because the non-supporting span of the heat exchanger bundles increases which has decreased the rigidity of the tubes. The failure of the heat exchanger bundles poses serious harm to the industrial production [5].

Fluid induced vibration incorporates a wide range of disciplinary knowledge, such as structural mechanics, material mechanics, fluid mechanics, and vibration. It is essential for the design and operation of heat exchangers. Up to now, scholars have done a lot of work around this discipline. At present, it is generally believed that the fluid-induced tube bundle vibration mechanism includes vortex shedding, turbulence buffeting, fluid elastic vibration, acoustic resonance and so on [6-7]. In this paper, we studied fluid elastic vibration in the shell flow field. In the case of the existence of fluid-induced vibration mechanisms, the vibration of any tube bundle changes the flow field around the tube bundle [8]. The change of the flow field causes the change of excitation force acting on the adjacent tubes. Fluid elastic vibration is caused by the interaction between the fluid exciting force and the elastic displacement of the tubes. The amplitude of heat exchanger tube bundles increases when the velocity of the fluid exceeds critical velocity of vibration induced by fluid elastic excitation. If the damping isn't large, the amplitude will increase until the tubes collide with each other [9-11].

Studies have shown that the fluid elastic vibration mechanism took place in the high velocity region mainly. It determines whether the fluid elastic vibration in the heat exchanger causes the tube bundle to vibrate in this paper. The dynamic mesh model in FLUENT software is used to determine the transverse displacement of each tube bundle.

2. Numerical approach



2.1. Mathematical models [12]

In this paper, Lever and Weave's model [13-14] of flow tubes are adopted to explain the instability of fluid elasticity. The assumption in this model is that the vibration of each tube is not affected by the vibration of adjacent tubes and the fluid is regarded flowing in a channel between the heat exchanger. The schematic diagram of the flow tube model is as follows.

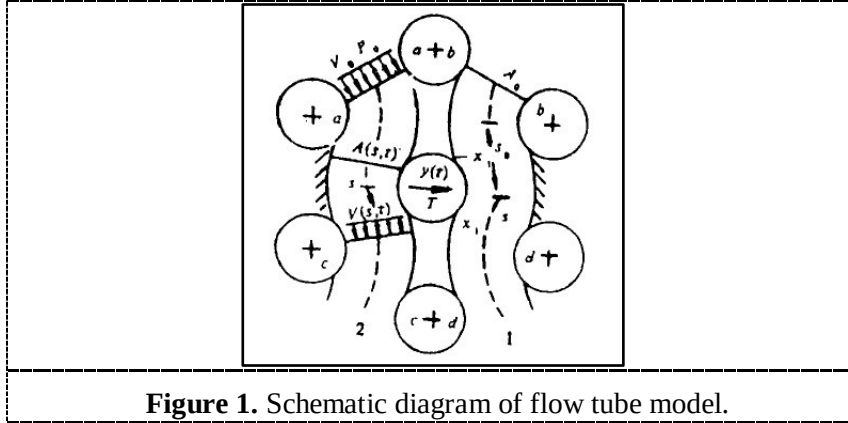


Figure 1. Schematic diagram of flow tube model.

According to the research, the vibration direction of the tube is mainly perpendicular to the flow direction. The tube transverse does simple harmonic vibration, and the displacement equation can be expressed as:

$$y(t) = y_0 \sin \omega t \quad (1)$$

$$A_i(s, t) = A_0 + (-1)^{i+1} A(s) \sin[\omega t + \psi(s)] \quad i=1,2 \quad (2)$$

$$U_i(s, t) = U_0 + (-1)^{i+1} U(s) \sin[\omega t + \varphi(s)] \quad i=1,2 \quad (3)$$

$$P_i(s, t) = P_0 + (-1)^{i+1} P(s) \sin[\omega t + \theta(s)] \quad i=1,2 \quad (4)$$

Among the equations, $A_i(s, t)$ represents the area of the unsteady flow tube. A_0 represents the unit length steady flow tube area at the entrance. $A(s)$ indicates the disturbance quantity of unit length flow tube area. $\psi(s)$ is the flow tube area phase difference. $U_i(s, t)$ indicates an unsteady flow rate. U_0 represents the inlet flow rate. $U(s)$ is the velocity disturbance quantity. $\varphi(s)$ is the phase difference of the velocity. $P_i(s, t)$ represents unsteady pressure. P_0 is the entrance pressure. $P(s)$ indicates pressure disturbance quantity. $\theta(s)$ indicates the pressure difference for the pressure. In those equations, the first one is the steady state term and the second one is the disturbance term.

The continuity equation of one-dimensional unstable incompressible fluid can be expressed as:

$$\frac{\partial A_i(s, t)}{\partial t} + \frac{\partial A_i(s, t) U_i(s, t)}{\partial s} = 0 \quad (5)$$

Derivation of equation (2) is

$$\frac{\partial A_i(s, t)}{\partial t} = (-1)^{i+1} y_0 \omega \cos \left[\omega t + \frac{2s\omega}{U_0} \right] \quad (6)$$

Make equation (5) and (6) joint order, and integral along s :

$$(-1)^{i+1} y_0 \omega \int_{-s_0}^s \cos \left[\omega t + \frac{2s\omega}{U_0} \right] ds + A_i(s, t) U_i(s, t) \Big|_{-s_0}^s = 0 \quad (7)$$

Substituting equation (3) and (4) into equation (7):

$$U(s) = \frac{3U_0 y_0}{A_0} \sin \frac{s+s_0}{U_0} \omega \quad (8)$$

$$\varphi(s) = -\frac{\pi}{2} + \frac{s-s_0}{U_0} \omega \quad (9)$$

The velocity distribution of the flow field can be determined by the former equation.

The differential form of the one-dimensional unstable incompressible fluid energy equation can be expressed as:

$$\frac{d}{dt} \left(\rho \frac{U_i^2}{2} \right) = \rho \left[\frac{\partial}{\partial t} \left(\frac{U_i^2}{2} \right) + U_i \frac{\partial}{\partial s} \left(\frac{U_i^2}{2} \right) \right] = -U_i \frac{\partial P_i}{\partial s} \quad (10)$$

After sorting out the equations, and integrating along s , we can obtain:

$$\begin{aligned} P(s) \sin[\omega t + \theta(s)] &= \frac{3U_0^2 \rho y_0}{4A_0} \sin \omega t \left[3 \cos \left(\frac{2\omega s}{U_0} \right) - 3 \cos \left(\frac{2\omega s_0}{U_0} \right) - \frac{2\omega(s+s_0)}{U_0} \sin \left(\frac{2\omega s_0}{U_0} \right) \right] \\ &+ \frac{3U_0^2 \rho y_0}{4A_0} \cos \omega t \left[3 \sin \left(\frac{2\omega s}{U_0} \right) + 3 \sin \left(\frac{2\omega s_0}{U_0} \right) - \frac{2\omega(s+s_0)}{U_0} \cos \left(\frac{2\omega s_0}{U_0} \right) \right] \end{aligned} \quad (11)$$

According to the pressure distribution on the both sides of the flow tube, the excited force can be obtained, and the equation can be expressed as:

$$F_p(t) = l \int_{-s_1}^{s_2} [P_1(s, t) + P_2(s, t)] ds \quad (12)$$

Among them, the stable phase of P_1, P_2 is uniform, the wave phase has the same size, but opposite direction, then:

$$F_p(t) = F_1 \sin \omega t + F_2 \cos \omega t \quad (13)$$

The vibration equation of the tube can be expressed as:

$$F_p(t) = ml \frac{\partial^2 y}{\partial t^2} + C \frac{\partial y}{\partial t} + ky \quad (14)$$

$$C = \frac{ml\omega\delta_0}{\pi} \quad (15)$$

m indicates the quality per unit length of the heat transfer tube. k is the elastic coefficient. C represents the damping coefficient. δ_0 is the logarithmic decay rate.

The vibration equation can be expressed as:

$$ml \frac{\partial^2 y}{\partial t^2} + \left(C - \frac{F_2}{\omega y_0} \right) \frac{\partial y}{\partial t} + \left(k - \frac{F_1}{y_0} \right) y = 0 \quad (16)$$

Heat transfer tubes vibrate under the action of fluid elastic excitation. The vibration equation is:

$$\ddot{x}_j^n + 2\delta_0 f \dot{x}_j^n + 4\pi^2 f^2 x_j^n = \frac{F_j^n}{m_0} \quad (17)$$

In this equation, x represents the displacement of the tube vibration. f represents the natural frequency of the tube. F indicates the lift and drag force of the tube under fluid action. m is the quality of the tube. $j=1, 2$.

The displacement equation of the tube produced by the force of fluid elastic excitation is:

$$\ddot{x}_j^n = \frac{F_j^n - kx_j^n}{m_0} - 2\delta \dot{x}_j^n \quad (18)$$

$$\dot{x}_j^{n+1} = \dot{x}_j^n + \ddot{x}_j^n \times \Delta t \quad (19)$$

$$x_j^{n+1} = x_j^n + \frac{1}{2}(\dot{x}_j^n + \dot{x}_j^{n+1})\Delta t + \frac{1}{2}\ddot{x}_j^n \times \Delta t^2 \quad (20)$$

2.2. Computational model

There are many boundary deformation and motion problems in engineering simulation. The dynamic mesh model in FLUENT software can be used to simulate the flow problem of calculation area change caused by boundary deformation or motion [15]. The feature of dynamic mesh model is that mesh can be reconstructed according to the needs of the different processes in the solution. FLUENT software can be used to observe the function relationship between the transverse displacement and the time history to reflect the strength of elastic excitation of the heat transfer tubes in the flow field.

Use Pro / Engineer software to create the model, and then initial meshing. In order to analyse vibration of the tube at different positions, we select three regions as objects of the analysis of numerical simulation: the shell side of the entrance, the shell side of the central and shell side of the exit. In order to be closer to the actual situation, the nine adjacent tubes of each region are also taken into account (figure 2). The finite element model is figure 3.

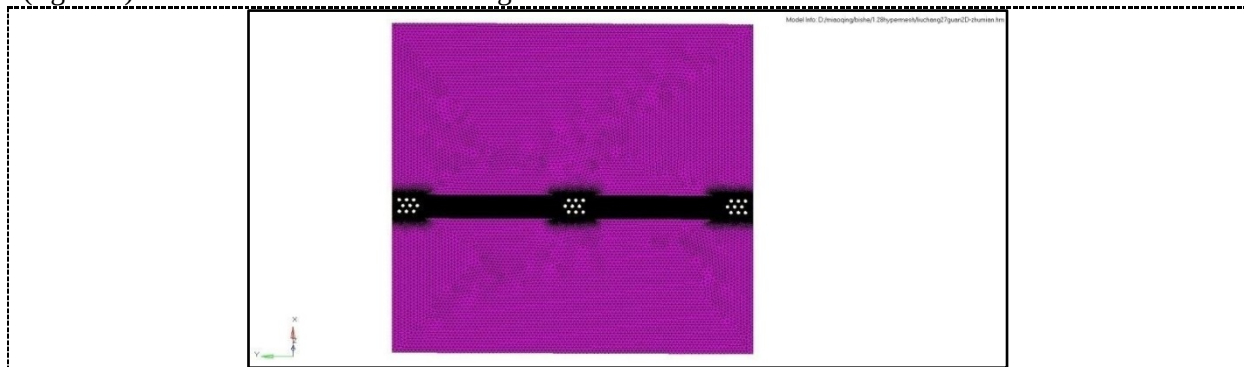


Figure 2. Finite element model for the numerical simulation of the flow around a cylinder in a heat exchanger.

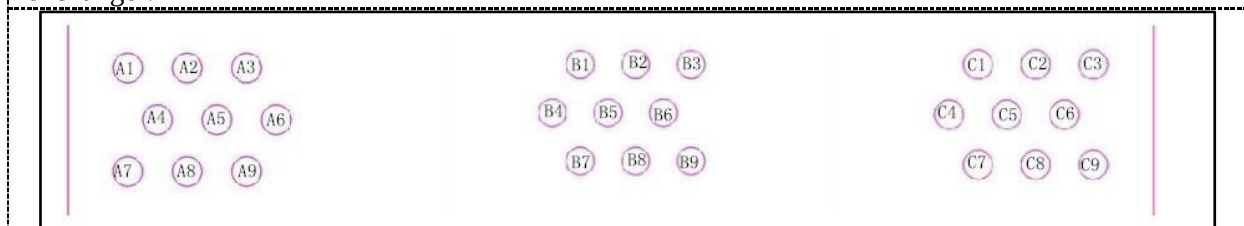


Figure 3. Schematic diagram of the position number of the heat exchange tubes.

2.3. The boundary conditions of the simulation

The finite element model was imported to the FLUENT software. The basic solver is set to the pressure based, transient, implicit, absolute and so on. Selecting the RNG k- ϵ model as calculation model. Set the material parameters for the medium, the entry and exit boundary conditions, velocity inlet. And input the incoming fluid speed. Select pressure outlet, input static pressure. Set shell side, tube board surface, partition surface and the surface of heat transfer tube as wall. The wall is a non-slip. And activate the dynamic mesh model, select the mesh update model as the spring smoothing model and local region reconstruction; deploy dynamic mesh fields, assume that the movement of the rigid body uses UDF to define the movement of the boundary, the motion equation of the heat transfer tube

under the fluid excitation force obtained above defines the movement of the boundary of the heat transfer tube. Set the time step to 0.00001s, 1000 times, iterate 20 times at every turn.

3. Results and discussion

Obtain velocity distribution after simulating (figure 4).

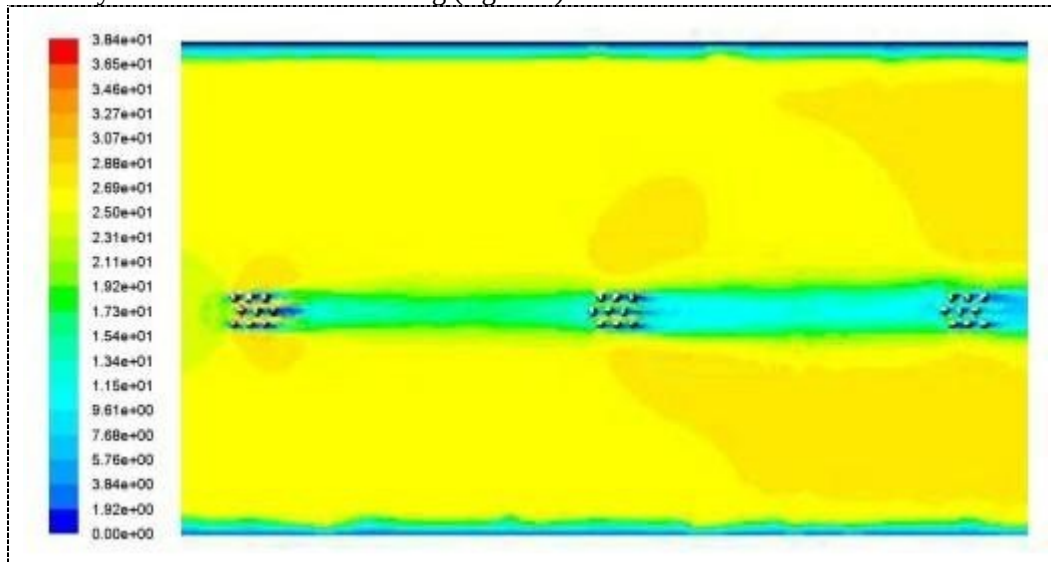


Figure 4. Velocity distribution of the heat exchanger.

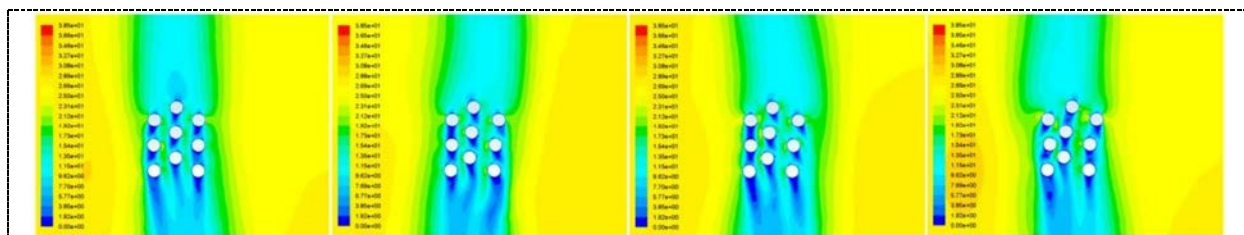


Figure 5. Schematic diagram of vortex shedding in heat exchanger's shell side.

The nine tubes in the central area was taken to observe the velocity distribution as preliminary analysis. Founding that the boundary of the heat transfer tubes is constantly changing and vortex shedding phenomenon obviously. This proves that it is correct that using the dynamic mesh model in the solution condition. It can also be found that the flow field near the outer edge of the tube is changing more intense than the flow field at the edge of the inner tube. With the increase of time, the area of wake is more obvious. But the area of wake is not large. If the area of wake is large, it is shown that the heat transfer tube is more likely to produce fluid induced vibration, and the vibration first caused by the vortex shedding and turbulence buffeting. These vibrations cause partial displacement of the heat transfer tube bundles. And then with the continuous increase of flow velocity, fluid elastic instability is likely to induce heat transfer tube bundles vibration. As the outer flow field changes more intense, the outer tube bundles displacement changes more obvious than the inner tube bundles displacement, and outer tube bundles are more likely to produce flow induced vibration.

We found that the locations of different tubes are constantly changing, and the changing trend is different. Such as figure 5, the right side of a row of three heat exchanger tubes. The connection of these three heat exchanger tubes is nearly a straight line at the first picture, but it becomes a curve at the fourth picture. And based on the second and the third picture, it is unable to analyse the variation tendency and law of relative position of heat exchanger tubes. Therefore, it is necessary to use FLUENT to monitor and record the displacement of each heat exchanger tube boundary separately.

After monitoring the displacement of all the heat exchanger tubes by FLUENT, The variation of the lateral displacement of 27 heat exchangers with time is obtained. Comparing with the heat exchanger tubes at different locations in different areas, we get the following results as figure 6 and figure 7. According to figure 6, the displacement of the heat exchanger tubes at the same position is quite different at different region, the heat exchanger tubes displacement of region A changed with time is much larger than B and C, the maximum displacement reached 3 units during the monitoring period (one unit is 0.001mm). It shows that the intensity of the heat exchange tube in region A is the strongest. It also shows that at position 4, the heat exchanger tubes vibration is less significant at A, B, C three regions, the displacements are all within 1 unit. It means that in different areas, position 4 is not easy to produce a strong vibration. figure 7 shows that region A is more prone vibration and the vibration in the position 4, position 5, and position 6 in region A are less intense. The vibration in position 5 and position 6 in region B is relatively stronger, which is the primary vibration location in the region B. Only position 6 vibration is more significant in the region C. In general, the heat exchanger tubes vibration of region A (the shell side of the entrance) is relatively stronger when the heat exchanger tubes at the same position in different region. And heat transfer tubes vibration in different position in the same region show different characteristics.

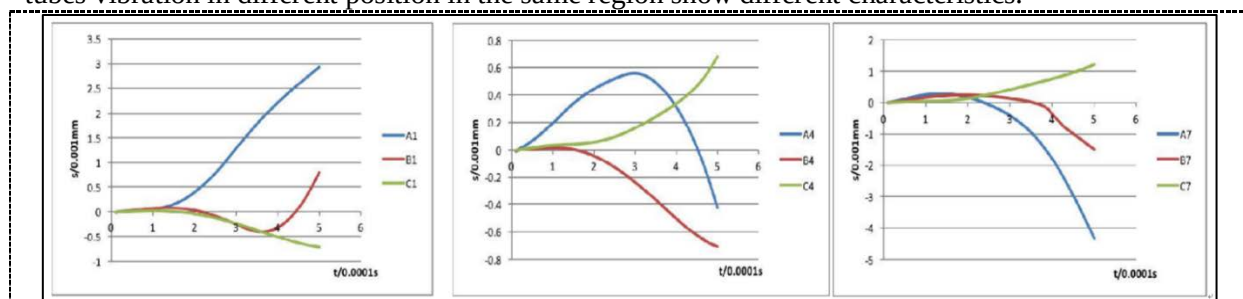


Figure 6. Displacement time curve of tube at same locations in different areas.

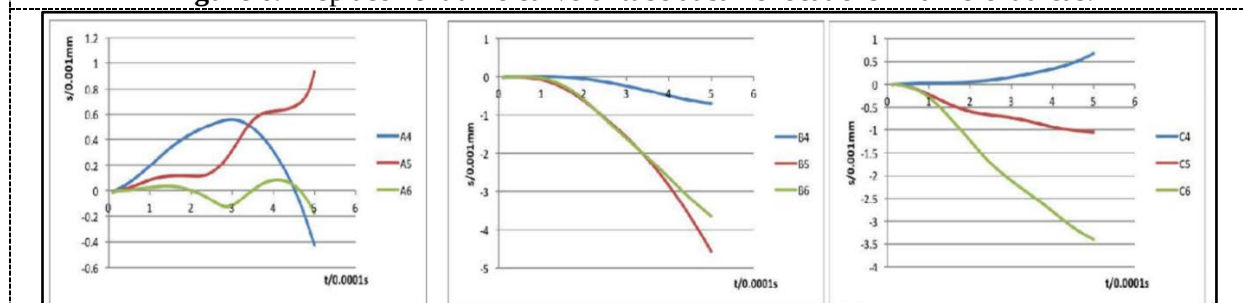


Figure 7. Displacement time curve of tube at different locations in same areas.

4. Conclusions

The background is the failure of tube bundles in air heat exchanger in our paper. We use the FLUENT software dynamic grid model to simulate the flowing status of the flow field around the cylindrical group. Kaman vortex can be seen and the fluid flow is observed to cause the cylinder moving (vibration response). We control the variable, comparing with the different parts of the heat exchanger tube at the same location on the displacement. It shows that the shell side of the tube generated by the vibration displacement is greater than the other location of the tube. So the tubes in the shell side of the entrance are susceptible to failure due to fluid induced vibration.

5. References

- [1] Qian S W 2002 *Handbook for Heat Exchanger Design* (Beijing: China Chemical Industry Press)
- [2] Costa A L H and Queiroz E M 2008 *Appl Therm Eng* **28**
- [3] Fesanghary M, Damangir E and Soleimani I 2009 *Appl Therm Eng* **29** 5-6
- [4] Ravagnani M A S S, Silva A P, Biscaia E C and Caballero J A 2009 *Ind Eng Chem Res* **4**

- [5] Feng G 2012 *Chem Ind Eng Prog* **31**, 3
- [6] Su Y C 2012 *A Study on the Flow-Induced Vibration and Heat Transfer Characteristic of Elastic Tube Bundle* (Shandong: Shandong University)
- [7] Qian S W 1988 *Flow Induced Vibration of Heat exchanger Foundation* (Wuhan: Huazhong College of Technology Press)
- [8] Jiang C Y 2012 *Study of Calculation Method of Flow Induced Vibration in Tube and Shell Heat Exchanger* (Shanghai: Fluid machinery and Engineering, East China University of Science and Technology)
- [9] Li A J, Xing G J and Zhou L W 2007 *Liaoning Chemical Industry* **36**
- [10] Bo H L 1994 *J Tsinghua Univ: Nat Sci Ed* **34** 6
- [11] Bo H L 1995 *Chin J Nucl Sci Eng* **15** 1
- [12] Nie Q D 1996 *Acta Mech Sinica* **28** 2
- [13] Lever J H, Weaver D S 1982 *J. Pressure Vessel Technol* **104** 3
- [14] Lever J H, Weaver D S 1986 *J Sound Vib* **107** 3
- [15] Zhu H J 2011 *Fluent 12 Fluid Analysis and Engineering Simulation* (Beijing: Tsinghua University Press)

Acknowledgments

This work was supported by "The Fundamental Research Funds for the Central Universities" (No. DUT15QY35) and "Dalian University of Technology Innovative Undertaking Management Platform" (No. 2016101410837).