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Numerical Simulation of Air Curtain Used in the Cold Store Through Different Models

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Abstract. Air curtains are commonly used to cut off air flow between the cold store and hot environment, and reduce heat and mass transfer in order to maintain the low temperature in the cold store. CFD can be used to predict operation rule of air curtain used in the cold store intuitively and modelling is the most important part of numerical simulation of air curtain used in the cold store. In this study, different numerical models including standard $\kappa-\varepsilon$ model (with and without boussinesq approximation) and RSM (with and without boussinesq approximation) were used to simulate the temperature field and air flow field in the cold store and operation rule of air curtain used in the cold store after air curtain opened for 60s. Meanwhile, the actual operation of air curtain used in the cold store was tested and the simulation values were compared with the experimental values. The results showed that the velocity of the central mainstream decayed slowly, but the velocity of both sides of air curtain decayed fast. The optimal model used to predict the temperature field and air flow field in the cold store and operation rule of air curtain used in the cold store was standard $\kappa-\varepsilon$ model with boussinesq approximation, the relative error was within 20%. The optimal model can be used to predict the temperature field and air flow field in the cold store during different times in the future research.

Introduction

The cold store is one of the critical links of cold chain. The interior temperature of the cold store will increase due to hot air infiltration into the cold store when door opened for loading and uploading, and the mist caused by the exchange of air will influence the sight of staff and can lead to accidents. The most serious result is the energy consumption of operation and defrosting in the cold store will increase due to the deterioration of heat transfer caused by frosting of evaporation ducts[1]. According to statistics, the average power consumption of cold stores in China is high to 131 kW·h/m^3 every year[2]. In order to improve the operating efficiency of cold stores and realize the energy conservation, the air curtain installed at the entrance of the cold store is used to reduce heat and mass transfer of air inside and outside the cold store to keep temperature stable inside the cold store.

Computational fluid dynamics, CFD has become an effective tool to study the air curtain because the air flow field and temperature field are difficult to measure when the air curtain is working because CFD can be used to analyse physical phenomena including fluid flow and heat transfer, and the air flow field and temperature field of air curtains can be simulated visually and quickly by CFD. The most important part of numerical simulation of air curtain is modelling[3]. Zhao Lin[4] used standard $\kappa-\varepsilon$ model to simulate the influence of cyclic mine air curtain. Cui air flow field and temperature field of the display cabinet air curtain by RSM and the experimental verification was also taken. Jintan[5] and Xu Zhengben[6] applied correction $\kappa-\varepsilon$ model to predict the air flow field of air curtain used in the vertical open display. Xie Zhuo[7] simulated the air flow field and temperature field of air curtain used in the display by RSM. However, the research about modelling of air curtain used in th cold store is still less. In order to simulate the air curtain used in the cold store accurately, the standard $\kappa-\varepsilon$ model (with and without boussinesq approximation) and RSM (with and without boussinesq approximation) were used in this study to simulate the air flow field

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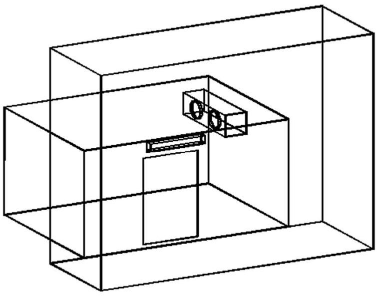
and temperature field, and the optimal model of simulating the air curtain used in the cold store could be obtained by comparing the simulation results for reference of further simulation research in the future.

Materials and methods

Physical model. The research object was an experimental empty cold store with internal dimension of 4.5m (L)×3.3m (W)×2.5m (H) in Shanghai ocean university and the material of this cold store was polystyrene sandwich insulation panel with 0.15m thick. This cold store had an door with 1.2m (W)×2m (H), which was 1m from the ground. The suspended ceiling fan with dimension of 1.75m (L)×0.46m (W)×0.5m (H) in the cold store had two circular air outlets with 0.2m radius and the return air grille was on the back of the fan. The air curtain device (Model FM-3515LY, Fenghao Ltd, Shanghai) was installed centrally above the door outside the cold store, which was 0.15m from the outside surface of the cold store. The return air grille was in front of the air curtain device with area of 1.2m×0.16m=0.192m^2 and the air supply grille was under the air curtain with area of 1.2m×0.04m=0.048m^2, which was 0.2m above the door. The computational domain covered the cold store, air curtain and a volume outside the cold store. This volume was a long hall with dimension of 6m (L)×2.1m (W)×5m (H). It was assumed that the heat and mass transfer between air inside and outside the cold store only happened during door opened in this study. The physical models were showed as follows.



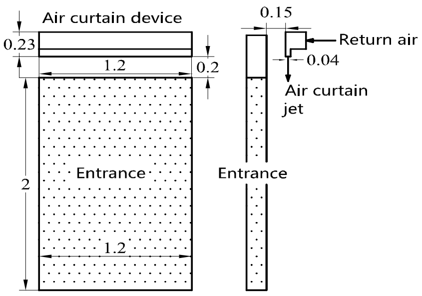


Fig.1 Computation model of experimental cold store Fig.2 Front and left elevation of the air curtain

Numerical model. The governing equations were written as Reynolds Average Navier-Stokes (RANS) equations. Thus the instantaneous value of each variable was given by the sum of its time averaged mean value. In order to solve this equation, the standard $\kappa-\varepsilon$ model was used and the wall-function approach must be used[1] because the Reynolds number in the viscous sublayers near the wall was low. The turbulence kinetic energy κ equation, the kinetic energy's rate of dissipation ε equation and the turbulent viscosity coefficient were written as follows[8]:

$$\rho u_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + u_i \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \rho\varepsilon$$

$$\rho u_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \frac{C_1\varepsilon}{k} - u_i \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - C_2\rho\frac{\varepsilon^2}{k}$$

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$$

All values in the above-mentioned equations were given in Table 1.

Table 1 Coefficients in $k-\varepsilon$ model

C_1	C_2	σ_T	σ_k	σ_ε	C_μ
1.44	1.92	0.9~1.0	1.00	1.30	0.09

The general differential equations could be written by solving the continuity, momentum, energy, turbulence kinetic energy and kinetic energy's rate of dissipation equations simultaneously in the rectangular coordinates:

$$div(\rho V\varphi) = div(\Gamma grad\varphi) + S$$

Where φ is universal variables, Γ is generalized diffusion coefficient corresponding to φ, S is generalized source corresponding to φ.

To compare results of different models, the Reynolds stress model (RSM) was also used in this study. RSM was transport equation to solve each component of Reynolds stress tensor. The form could be written as follows:

$$\frac{\partial}{\partial t}(\rho\overline{u_i u_j})+\frac{\partial}{\partial x_k}(\rho U_k\overline{u_i u_j})=-\frac{\partial}{\partial x_k}\left[\rho\overline{u_i u_j u_k}+\overline{p(\delta_{kj}u_i+\delta_{ik}u_j)}\right]+\frac{\partial}{\partial x_k}\left[\mu\frac{\partial}{\partial x_k}\overline{u_i u_j}\right]-\rho\left(\overline{u_i u_k}\frac{\partial U_j}{\partial x_k}+\overline{u_j u_k}\frac{\partial U_i}{\partial x_k}\right)$$

$$-\rho\beta\left(g_i\overline{u_j\theta}+g_j\overline{u_i\theta}\right)+\overline{p\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)}-2\mu\overline{\frac{\partial u_i}{\partial x_k}\frac{\partial u_i}{\partial x_k}}-2\rho\Omega_k\left(\overline{u_j u_m}\varepsilon_{ikm}+\overline{u_i u_m}\varepsilon_{jkm}\right)$$

The heat and mass transfer between air inside and outside the cold store happened caused by density difference of air when door was opened. The boussinesq approximation model was used when the buoyancy effect was considered. The boussinesq approximation meant that the densities were constant in equations except for the buoyancy term in the momentum equation[9].

Boundary conditions. The wall of the cold store was chosen following the second boundary condition of constant heat flux, the wall heat transfer coefficient K_w was 0.192W/m^2·K and the computational temperature inside the cold store was -5°C[1]. The walls of the volume outside the cold store were modelled as no slip walls where the velocity of the fluid at the walls was zero and the value of temperature was set to 15°C following the actual measurement. The density of heat flux q was 3.84W/m^2.

The air supply outlet of the fan in the cold store was modelled as velocity inlet condition, the values of velocity and temperature of the air were set to 3.5m/s and -5°C following actual measurements. The turbulence intensity and the hydraulic diameter were 5% and 0.4m according to the characteristics of the fan. The return air grille of the fan was modelled as outflow condition.

The door was modelled as constant heat flux when it was closed. All walls of the air curtain were modelled as no slip walls when it was closed. However, when the door and air curtain device were opened, the door was modelled as interior condition and the air supply grille was modelled as velocity inlet condition. The values of velocity and temperature were set to 9m/s and 15°C following actual measurements and the direction of the velocity was normal to the boundary. The turbulence intensity were 10%[10] and the hydraulic diameter was 0.05m according to the dimension of velocity inlet. The return air grille was modelled as outflow condition. All walls of the air curtain device were modelled as no slip walls except for the air supply grille and return air grille of the air curtain.

Discretization. The air in the cold store was incompressible steady turbulence[11] because the air was circulated by the fan when the door was closed, so the first step was to simulate the steady operation of the cold store. SIMPLEC method was used to solve the coupling of pressure and velocity and the second-order upwind scheme was also used because it was more accuracy than the first-order upwind scheme. When the door and air curtain device were opened, PISO method was used to solve the coupling of pressure and velocity and the second-order upwind scheme was also used.

Experimental measurement. To verify the reliability of the numerical simulation, the air flow field of the air curtain and the temperature field inside the cold store were tested when the door was opened. The experimental devices included hot wire anemometer (Model KA22, KANOMAX Ltd, Japan) and multipoint temperature collection instrument (Model NetDAQ32, Fluke Ltd, America). The velocity values of each measuring point of the air curtain were measured by the hot wire anemometer and velocity measuring points were set under the air supply grille of the air curtain (Fig 3). The temperature values of each measureing point inside the colds store were measured by thermocouples with accuracy of ±0.5°Cand the multipoint temperature collection instrument. The temperature measuring points were set at four wall corners near the door and the central point of the cold store (Fig 4). In the experimental operation, the door was closed and the fan started after each measuring point was set up. When the temperature inside the cold store decreased to -5°C, the door and the air curtain device were opened, and the temperatures of each measuring points were measured every 1s by multipoint temperature collection instrument. The experiment was completed after 120s and the experimental values of 60s were recorded.

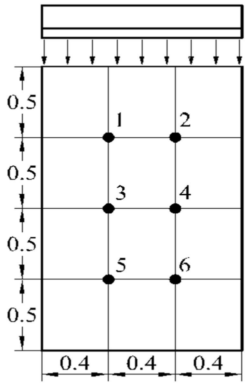


Fig.3Arrangement of velocity measuring points

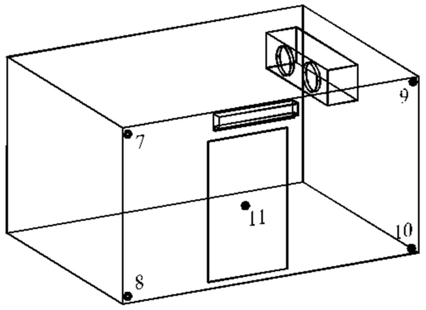


Fig.4Arrangement of temperature measuring points

Results and discussions

Experimental results

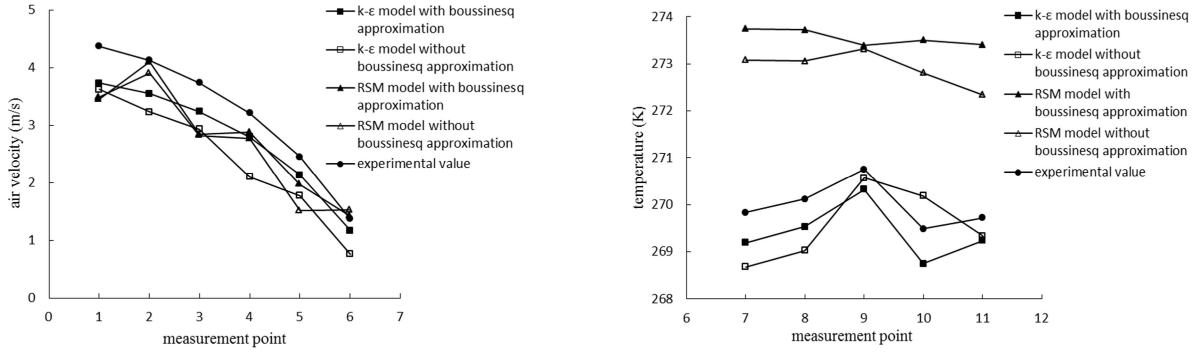


Fig.5 Comparison of predicted velocities and temperatures with experimental values

The comparison of predicted velocities and temperatures with experimental values was shown in Figure 5. The results reflected that the experimental velocity values of each point were higher than the simulation values and the velocity values simulated by the standard $\kappa-\varepsilon$ model with boussinesq approximation were closest to the experimental values, the absolute errors were within 1m/s and the relative errors were within 20%. The temperature values simulated by the standard $\kappa-\varepsilon$ model were closer to the experimental values than those simulated by RSM, and the absolute errors were within 20%. The experimental results showed that the optimal model for simulation of air curtain used in the cold store was the standard $\kappa-\varepsilon$ model with boussinesq approximation.

Simulation results

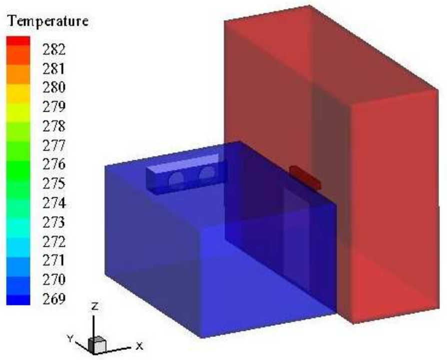


Fig.6 Temperature field of the cold store and volume

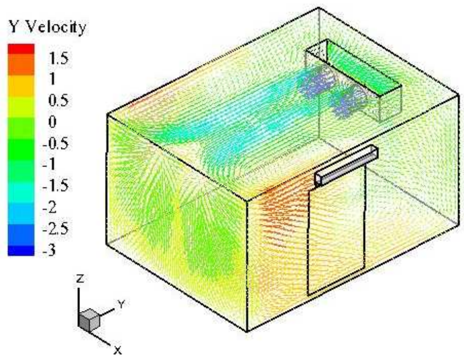
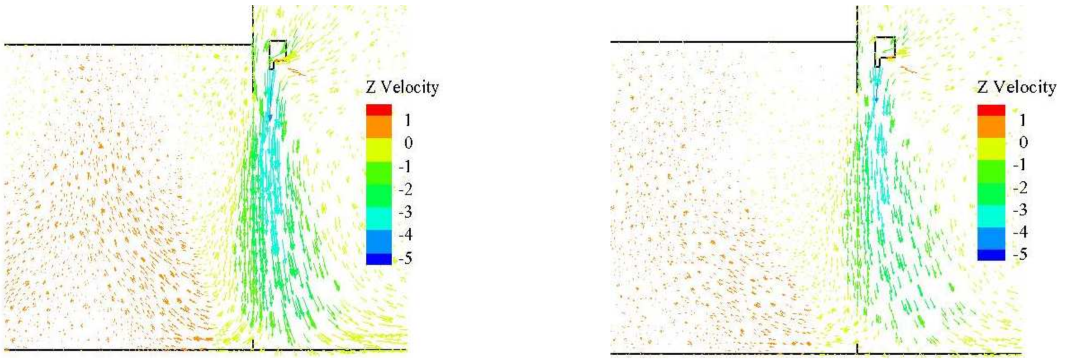
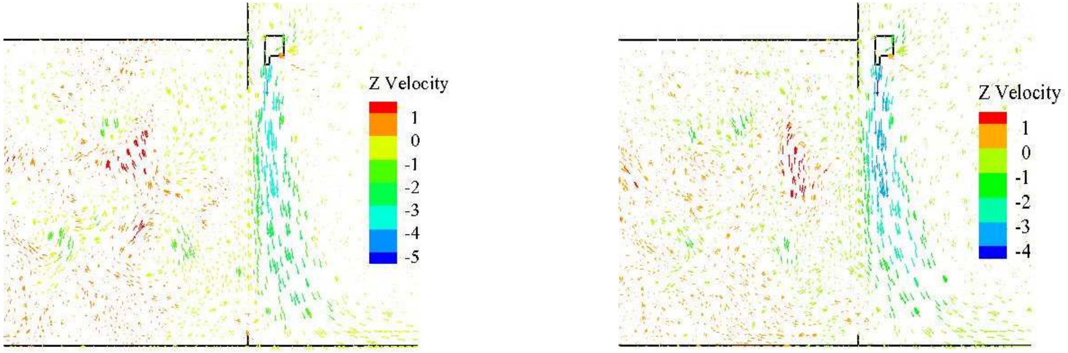


Fig.7 Air flow field of the cold store

Figure 6 and Figure 7 showed the simulation results of cold store in the steady operation. The blue and the red domain reflected the temperature fields of the cold store and the volume outside cold store, respectively. The results reflected that the air flow field and the temperature field in the cold store were stable when the door was closed.



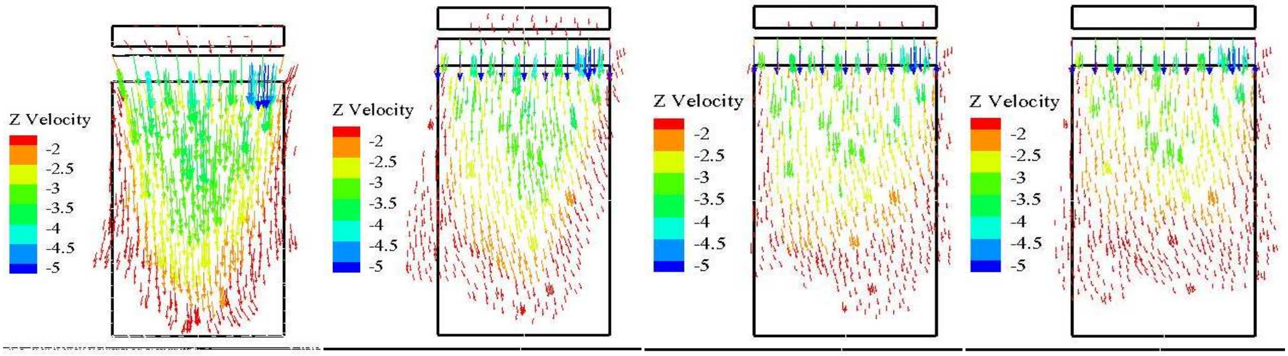
$\kappa-\varepsilon$ model with boussinesq approximation $\kappa-\varepsilon$ model without boussinesq approximation



RSM with boussinesq approximation RSM without boussinesq approximation

Fig.8 Velocity vectors after door and air curtain opened for 60s, left elevation

Figure 8 showed the velocity vectors when the door and air curtain device was opened for 60s on the basis of the steady simulation results of cold store being modelled as the initial condition of the unsteady simulation. The air flow fields of the air curtain and the cold store simulated by standard $\kappa-\varepsilon$ model were more uniform. However, the results simulated by RSM reflected that many vortexes existed in the cold store and the air flow in the cold store was confused because the air curtain was destroyed by the exchange of air inside and outside the cold store. This was because RSM could be used to predict the changes of the vortex, rotating and bending in the simulation of the exchange of the air curtain and air inside and outside the cold store. The comparison results reflected that the results simulated by the standard $\kappa-\varepsilon$ model were better than those simulated by RSM, which coincided with the experimental results. Meanwhile, the buoyancy effect on the air flow field in the cold store should be considered because the results of $\kappa-\varepsilon$ model with boussinesq approximation were better than those without boussinesq approximation, which also coincided with the experimental results.

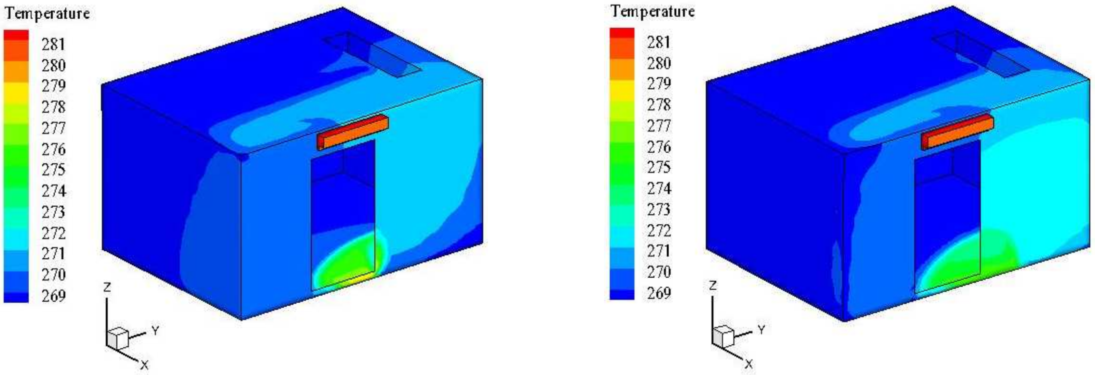


$\kappa-\varepsilon$ model with boussinesq $\kappa-\varepsilon$ model without boussinesq RSM with boussinesq RSM without boussinesq

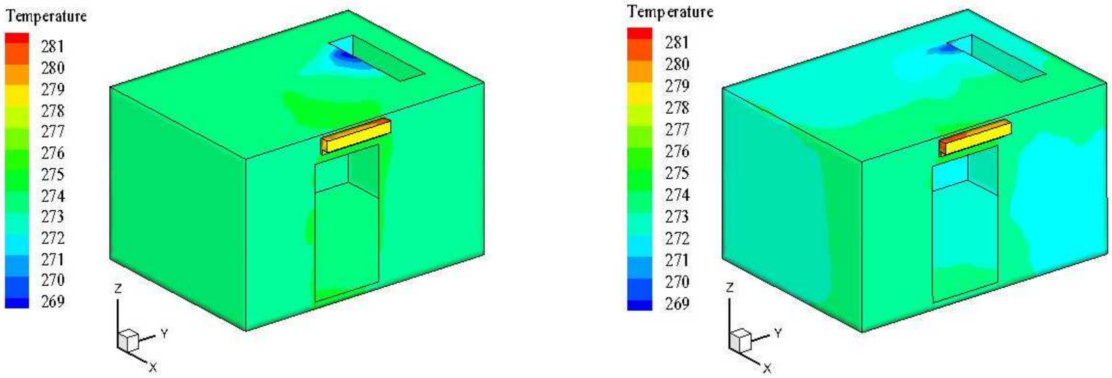
Fig.9 Velocity vectors at velocity more than 2m/s of air curtain after door and air curtain opened for 60s, front elevation

Figure 9 showed that the air curtain predicted by the standard $\kappa-\varepsilon$ model could basically covered the entrance of the cold store, and the hot air easily infiltrated into the cold store from the bottoms of both sides of the entrance because the velocity of the central mainstream decreased more slowly than that of both sides of air curtain. However, the decrease process of velocity of air curtain could be predicted by RSM, but the the velocity distributions were confused and the air curtain couldn't cover the door, which was because the exchange of air inside and outside the cold store destroyed the air curtain. Therefore, the comparison results of Figure 9 reflected that the results predicted by the standard $\kappa-\varepsilon$ model with boussinesq approximation were better than those predicted by other models, which was coincide with the experimental results.

Figure 10 showed 3-d temperature fields in the cold store after door and air curtain opened for 60s. The simulation results simulated by the standard $\kappa-\varepsilon$ model reflected that the temperature of the area close to the door and the bottom of the cold store was higher than that of the area away from the door. As time went by, hot air began to diffuse toward the top and inside of the cold store caused by density differences, thus destroying the uniform temperature field of the cold store. In addition, the temperature of the area close to the fan was higher than that of the area far away from the fan because the area far away from the fan was influenced by the cold air blown from the fan in the cold store. However, the temperature field predicted by the RSM was much different from that predicted by the standard $\kappa-\varepsilon$ model. The initial temperature field of the cold store had been destroyed, which was coincide with the former velocity field predicted by the RSM. This was because many vortexes in the cold store caused by the exchange of air inside and outside the cold store destroyed the uniform temperature field, thus increasing the temperature in the cold store. Therefore, the comparison results of Figure 10 reflected that the results predicted by the standard $\kappa-\varepsilon$ model were better than those predicted by RSM, which was coincide with the experimental results.



$\kappa-\varepsilon$ model with boussinesq approximation $\kappa-\varepsilon$ model without boussinesq approximation



RSM with boussinesq approximation RSM without boussinesq approximation

Fig.10 3-d temperature fields in the cold store after door and air curtain opened for 60s

Conclusions

The standard $\kappa-\varepsilon$ model (with and without boussinesq approximation) and RSM (with and without boussinesq approximation) were used in this study to simulate the air flow field and temperature field of the cold store and operation rules of the air curtain when the air curtain device was opened, and the experiments were also taken to verify the simulation results. Conclusions were drawn as follows by comparing the simulation results and experimental results:

（1）The simulation results predicted by the standard $\kappa-\varepsilon$ model were better than those predicted by the RSM through the comparison of the simulation results and experimental results. Furthermore, the optimal model used to predict the temperature field and air flow field in the cold store and operation rule of air curtain used in the cold store was standard $\kappa-\varepsilon$ model with boussinesq approximation, the relative error was within 20%.

（2）The velocity of the central mainstream decreased slowly, but the velocity of both sides of air curtain decreased fast. Therefore, the hot air easily infiltrated into the cold store from the bottoms of both sides of the entrance, thus influencing the temperature in the cold store.

Acknowledgements

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