Comparison of Two Ventilation Systems in a Chinese Commercial Kitchen

Xiongfeng Wan Likui Yu Huabo Hou
Master Associate professor Master
School of Energy Science and Engineering, Central South University
Changsha P. R. China，410083
xiongfengwan@gmail.com

Abstract: A numerical simulation of an indoor thermal environment in a Chinese commercial kitchen has been carried out using indoor zero-equation turbulence model. Two different ventilation systems in a Chinese commercial kitchen have been simulated. The results calculated for two different models show the airflow, temperature distribution and human thermal comfort index-PMV and PPD value. The simulation results indicate that both methods are capable of enhancing the capture and containment performance of cooking effluent effectively. Under the first condition, not only is unnecessary cooling load reduced and energy consumption of the air conditioning system economized, but a perceived thermal comfort environment can be provided. However, the supply air velocity and air temperature of the spot diffuser are restricted. In contrast, the perceived level of thermal comfort is not improved under the second circumstance. Further, the energy consumption of this case is higher than the former. Finally, the indoor thermal environmental properties have been analyzed.

Key words：Chinese commercial kitchen, kitchen ventilation, indoor thermal environment, numerical simulation, indoor zero equation

1. INTRODUCTION

During the last twenty years, some studies on the commercial ventilation have been continual reported to the ASHRAE Transaction as a special topic. Most of the papers are presented to investigate the ventilation problems in a Western-style kitchen [1-3]. As we known, standards of living and acceptable indoor thermal environment between occidental and oriental are dissimilar with each other evidently. So the design of Chinese commercial kitchen ventilation is different from Western-style kitchen.

Due to prefer to stir-frying foods in high temperature oil in Chinese commercial kitchen, the output of smoke, grease vapors and exothermic rate are so big, furthermore, the chef is badly exposed to buoyancy-driven hot fume and radiant heat from cooking operating face above the appliance directly. However, currently no standards for the optimized design of ventilation systems in Chinese kitchens are available. This may impact on optimizing system design and energy use for kitchen ventilation.

In order to investigate the indoor thermal environment in Chinese commercial kitchen, a three-dimension numerical simulation is applied to a typical commercial kitchen using indoor zero-equation model with a view to indoor surface-to-surface heat radiation [4].

In this paper, two different ventilation systems in Chinese commercial kitchen have been simulated. Spot diffusers are added to the top of operate zone in the first system analyzed which utilizing displacement ventilation. Backdrop plenum (rear discharge) with ceiling diffuser is adopted to the second system analyzed which utilizing mixing ventilation, 70% supply flow rate is brought in through the backdrop plenum without conditioned.

At last, the indoor thermal environmental properties have been analyzed according to calculate data, and some adjusting strategies to air condition system are presented. For instance, based on the theories of demand control ventilation and surging flow, the flow rate and frequency of the spot cooling are controlled depend upon the demand of human respiration and individual thermal comfort. The supply air points of displacement ventilation are optimized allowing of commercial kitchen configuration, etc.

2. NUMERICAL ANALYSIS

2.1 Physical Model

A typical Chinese commercial kitchen was researched in the paper. The ranges and hoods were placed along two jointed walls which formed into L-shape configuration. The nominal outside dimensions of commercial kitchen were 10 m by 6 m by 3.5 m. Two different indoor configurations of commercial kitchens using different ventilation systems were depicted schematically in Fig.1 and Fig.2, respectively. The geometric dimensions of the two different kitchen models are listed in table.1.

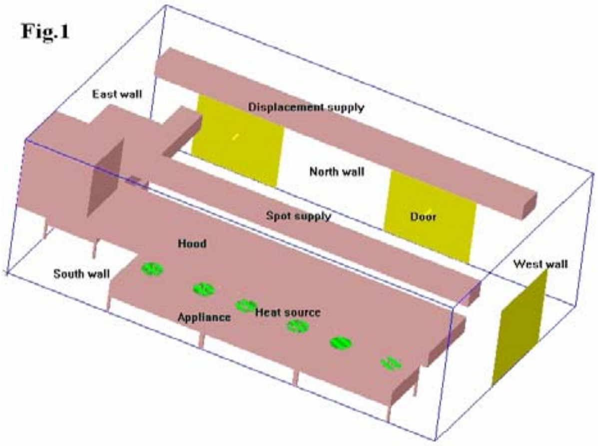


Fig.1 Schematic of kitchen model 1

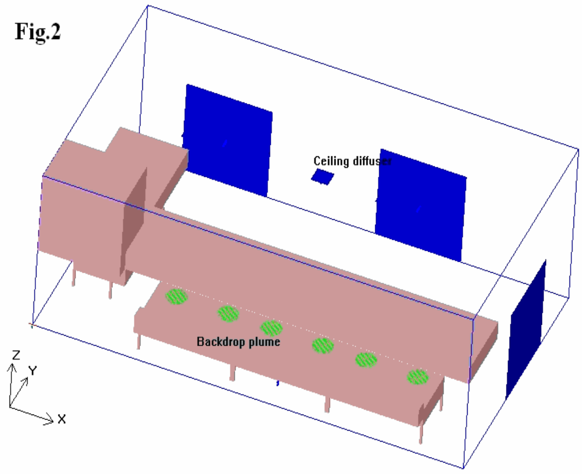


Fig.2 Schematic of kitchen model 2

2.2 Mathematical Model

Indoor zero-equation turbulence model was used in the paper, which is a new Zero-Equation model for numerical simulation of indoor airflow derived from the results of directly numerical simulation（DNS） of natural convection and mixed convection in a room.

In order to calculate indoor nonisothermal current when the Rayleigh number range from 2.6 to 3.0×10^{10}, a single algebraic function was developed to express the turbulence viscosity, which a function of the fluid density, ρ , local velocity magnitude, V, and the distance from the nearest wall, L : $\mu_t = 0.03874\rho VL$. Compared with other models, this model did not take all seven differential equations into account but only solving five differential equations in relate to mass, momentum and energy conservation. So when doing the simulation of fluid flow and heat transfer in the HVAC（Heat, Ventilation and Air Condition）field, it can obtain a convergence solution sooner, design and analysis indoor airflow more accurately[5]. Some presuppositions to facilitate the problems are setup as follows.

Tab.1 Geometric dimensions of kitchen models

Name	Model 1	Model 2
Room	10×6×3.5	10×6×3.5
Appliance	2.2×1.2×0.8*4	2.2×1.2×0.8*4
heat source	Φ0.44*8	Φ0.44*8
Door	2×2*3	2×2*3
Local hood	8×1.6×0.4*1 1.6×2.6×0.4*1	8×1.6×0.4*1 1.6×2.6×0.4*1
Exhaust opening	0.8×1.2*8	0.8×1.2*8
Spot supply opening	0.2×0.2*8	——-
Displacement supply opening	0.4×0.4*6	——-
Ceiling diffuser	——-	0.4×0.4*5
Backdrop plenum	——-	6.6×0.3×0.4*1 0.3×2.2×0.4*1
Backdrop plenum opening	——-	6.6×0.3*1 0.3×2.2*1

- The fluid flow is assumed to be steady, turbulence, incompressible ideal gas according with boussinesq assumption;
- Radiation Modeling for Objects Using the Surface-to-Surface Model;
- Gas tightness of commercial kitchen is well.
- Ignore lights and others little dispersive heat sources.

2.2.1 Governing equations

Mass continuity:

$$\frac{\partial V_i}{\partial X_i} = 0 \qquad (1)$$

Where V_i represents mean velocity component in X_i -direction; X_i corresponds to three perpendicular axes.

Momentum:

$$\frac{\partial}{\partial t}(\rho V_i) + \frac{\partial}{\partial X_j}(\rho V_i V_j) = \rho\beta(T_0 - T)g_i$$

$$-\frac{\partial P}{\partial X_i} + \frac{\partial}{\partial X_j}\left[\mu_{eff}\left(\frac{\partial V_i}{\partial X_j} + \frac{\partial V_j}{\partial X_i}\right)\right] \qquad (2)$$

Where ρ represents air density; V_j represents velocity component in X_j-direction; P represents pressure; μ_{eff} represents effective viscosity; β represents thermal expansion coefficient of air; T_0 represents temperature of a reference point; T represents temperature; g_i represents gravity acceleration in i-direction. And the effective viscosity, μ_{eff} , equals the sum of the turbulent viscosity, μ_t ,and the viscosity of the fluid, μ .

$$\mu_{eff} = \mu_t + \mu \qquad (3)$$

Energy:

$$\frac{\partial \rho T}{\partial t} + \frac{\partial \rho V_j T}{\partial X_j} = \frac{\partial}{\partial X_j}\left(\Gamma_{T,eff}\frac{\partial T}{\partial X_j}\right) + \frac{q}{C_P} \qquad (4)$$

Where $\Gamma_{T,eff}$ represents effective turbulent diffusion coefficient for T; q represents thermal source; C_p represents specific heat. The effective turbulent diffusion coefficient for T in Eq.（4）, $\Gamma_{T,eff}$, is determined by:

$$\Gamma_{T,eff} = \frac{\mu_{eff}}{Pr_{eff}} = \frac{\mu_{eff}}{0.9} \qquad (5)$$

Where the effective Prandtl number, Pr_{eff} , is 0.9.

Species concentrations

$$\frac{\partial \rho C}{\partial t} + \frac{\partial \rho V_j C}{\partial X_j} = \frac{\partial}{\partial X_j}\left(\Gamma_{C,eff}\frac{\partial C}{\partial X_j}\right) + S_C \qquad (6)$$

Where C represents species concentration; $\Gamma_{C,eff}$ represents effective turbulent diffusion coefficient for C; S_C represents source term of C. The effective turbulent diffusion coefficient for C in Eq.（6）, $\Gamma_{C,eff}$, is determined by :

$$\Gamma_{T,eff} = \frac{\mu_{eff}}{Sc_{eff}} = \frac{\mu_{eff}}{1.0} \qquad (7)$$

Where effective Schmidt number, Sc_{eff} , is 1.0.

2.2.2 Boundary conditions

The boundary conditions for the considered problems are determined as follows.

- Walls : Wall surface temperatures of the room are defined as 37.4℃, 33.7℃, 38.3℃, 34.5℃ respective representing east wall, south wall, west wall, north wall. The outdoor weather parameters are defined by the summer meteorologic parameters of Changsha, China.
- Inlets: for a spot-supply, V_i= V_j= 0,V_k= 2m/s; $T_{spot\text{-}supply}$= 19 ℃ representing the boundary conditions; for a displacement supply, the boundary conditions are: V_i=V_j= 0, V_k= 2.0m/s, $T_{displacement\text{-}supply}$= 28 ℃ ;for a ceiling diffuser, the boundary conditions are: V_i= －1.2m/s ,V_j = 0,V_k = －1.6m/s for j-direction diffusers and V_i= 0,V_j = －1.2m/s,V_k = － 1.6m/s for i-direction diffusers, T_{supply}=19℃; for Backdrop plenum, V_i= V_j= 0,V_k= －0.6m/s, $T_{backdrop\text{-}plenum}$= 28 ℃ ; for a door , the boundary conditions are: T_{intake}= 28 ℃; V_i= V_k = 0, V_j = －0.1 m/s for the doors on the west wall, and V_i= －0.1 m/s ,V_j = V_k = 0 for the doors on the south wall. The intake from the doors is regarded as natural inleakage.
- Outlets : For the exhaust, the boundary conditions are: V_i=V_j= 0, V_k= 0.5 m/s. Where the V

represents hood-face velocity according to capture velocity guideline (0.25 to 0.5 m/s) (ASHRAE 1991, chap.27).The two exhaust hood are connected by air chimney.

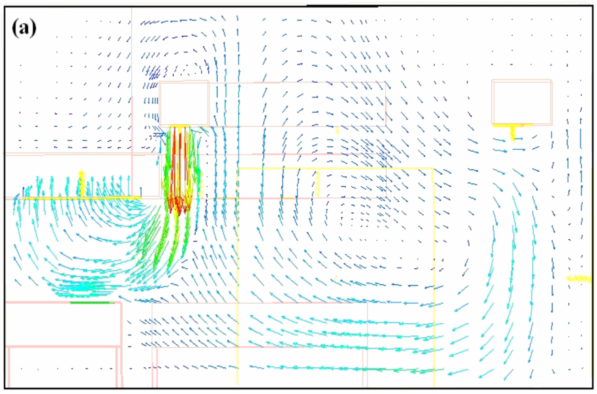
- Air rate: The total intake air rate Q_{in} of both model kitchens equals 12,000 m^3/h, while the total draft capacity Q_{draft} of both model kitchens equals 14,000m^3/h. the ratio between Q_{in} and Q_{draft} is 85 percent.
- heat sources: The total power of two-burner commercial open-top gas range is 93.4 Kw. The radiant heat gain to the space resulted from the range is 37 Kw, which was approximately 40% of the appliance energy consumption rate. This percentage is assumed according to ventilation performance curves of the test results in 1995 (Vernon A Smith et al.1995) [1].

2.3 Numerical procedure

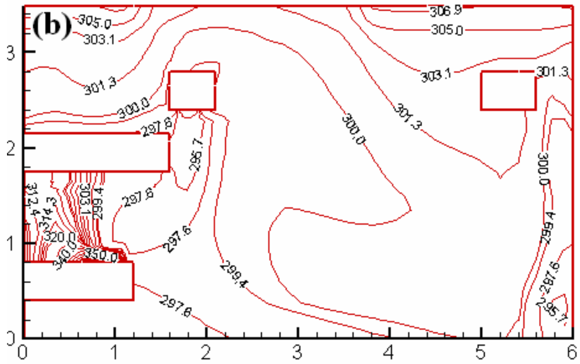
The above governing equations together with the corresponding boundary conditions are solved by A CFD program,AIRPAK2.1.The space is discretized into non-uniform computational cells using a finite volume method（FVM）,and the discretized equations are solved with SIMPLE algorithm. The computations were carried out with the same grid system 0.5m by 0.3m by 0.175m, which are the max sizes for different coordinates by default setup. The convergence criterions of all flow variables and all energy variables for this problem are setup as 1.0 E-03 and 1.0 E-06, respectively. In order to investigate the PMV and PPD values accurately, two parameters among the six variables related to thermal comfort are specified. The clothing worn (CLO) and the metabolic rate (MET) are defined to 0.7 and 2.0, respectively.

3. RESULTS AND DISCUSSION

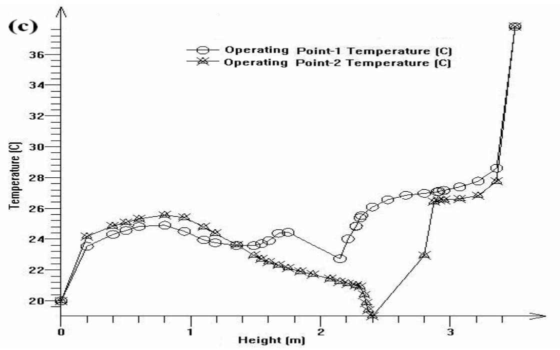
The airflow, temperature distribution and human thermal comfort index-PMV and PPD value of two different ventilation systems have been predicted. Then the energy consumption of the different cases have been calculated and compared.



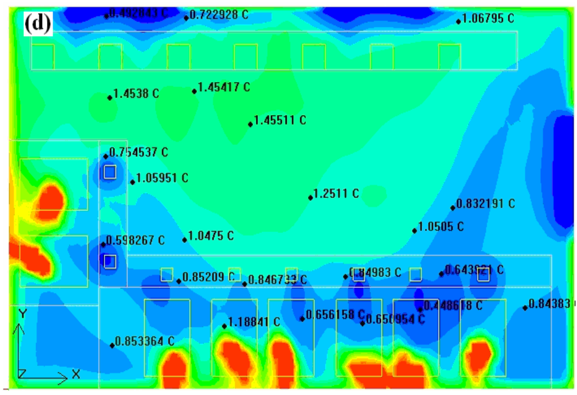
(a)Velocity Vectors in an y-z Plane (x=5.0m)



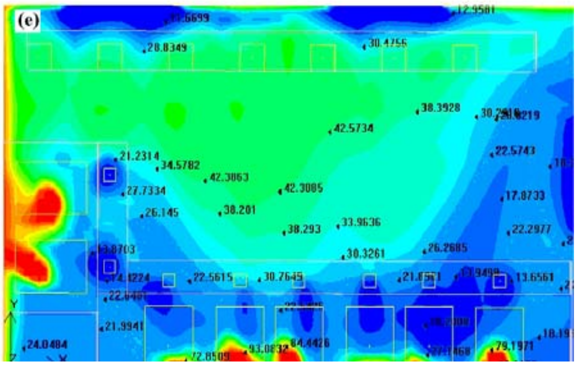
(b)Temperature Contours in an y-z Plane (x=5.0m)



(c)Temperature of operating point



(d)PMV Contours in an x- y Plane (z=1.1m)



(e) PPD Contours in an *x- y* Plane (*z=1.1m*)

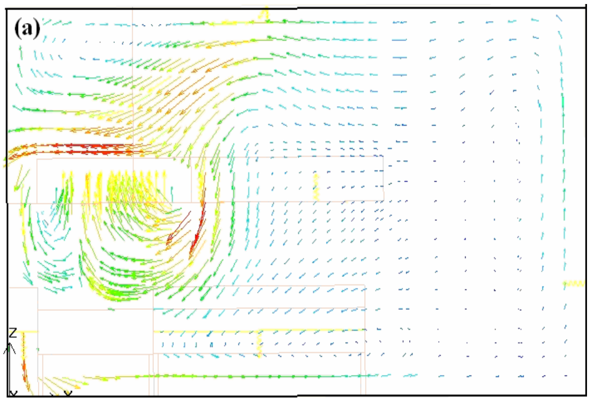
Fig.3 Velocity vectors, temperature contours, temperature of cooking operation points, PMV and PPD for case 1.

3.1 Airflow, temperature distribution and human thermal comfort

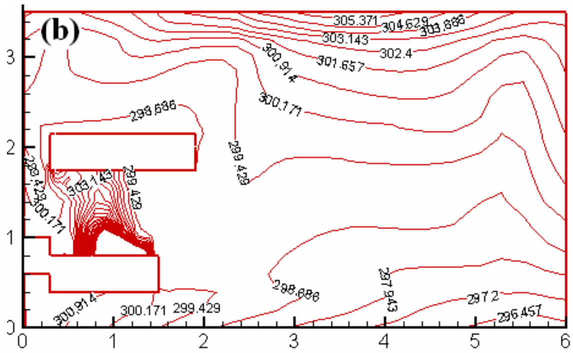
Fig.3 shows the principal characteristics of the flow field and human thermal comfort for the first case. As illustrated by the velocity vectors in fig.3a, a big vortices situated in the mid *y-z* Plane between spot supply and displacement supply, rotates in the anti-clockwise direction. Because of the hood's air draft capability, short circuit evidence is appeared between the appliance and the hood. The capture and containment performance of the exhaust hood is strengthened by the short circuit; furthermore, the exhaust airstreams leakage from the hood can be decreased. Fig.3b shows the kitchen temperature can meet with design requirement. In general, the operating zone is defined as the distance range from 0.3m (operating point-1) to 0.6m (operating point-2) from the appliance. The temperature of operating zone illustrated by Fig.3c is 23.6℃~25.6℃.

The contours of PMV as shown in Fig.3d shows PMV value of the operating zone is 0.64~1.05.Finnally, Fig.3e shows that Predicted Percent Dissatisfied(PPD) of operating zone is 22.5%~33.96%.

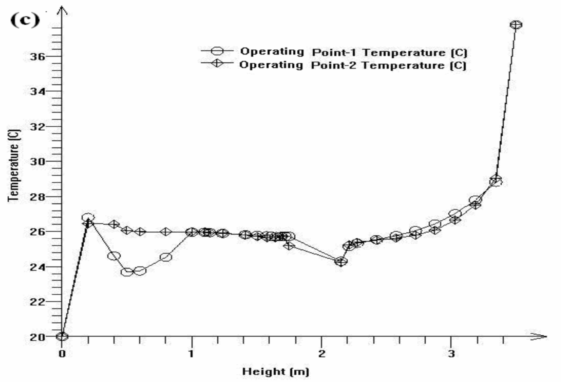
The predicted results of case 2 are showed in Fig.4.Two vortices is created which driven by the three supply air streams and the exhaust air. The first one is situated between the ceiling and hood. The second one, a big vortices situated in the middle of the *y-z* Plane illustrated by Fig.4a, rotates in the anti-clockwise direction. Two divided currents sweep over the appliance surface, enhancing the removal of the cooking fume. The air velocity around the chef's



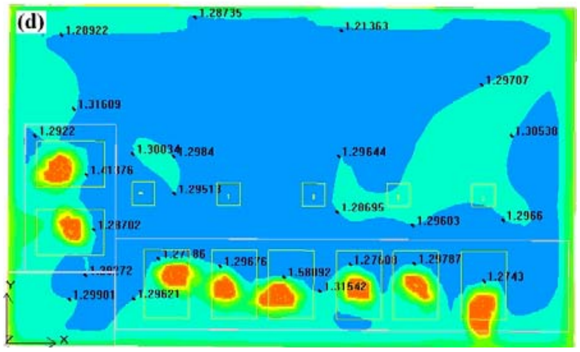
(a)Velocity Vectors in an *y-z* Plane (*x=1.5m*)



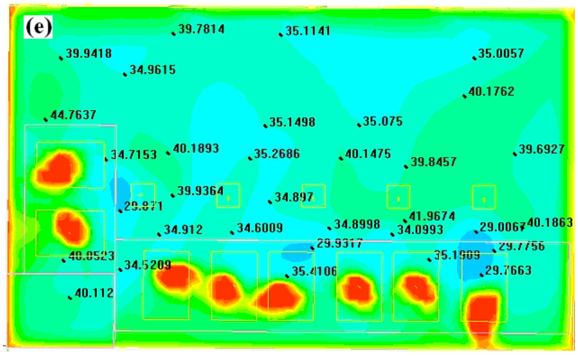
(b) Temperature Contours in an *y-z* Plane (*x=1.5m*)



(c)Temperature of operating point



(d)PMV Contours in an *x- y* Plane (*z=1.1m*)



(e) PPD Contours in an *x*- *y* Plane (*z=1.1m*)

Fig.4 Velocity vectors, temperature contours, temperature of cooking action spots, PMV and PPD for case 2.

head shown in Fig.3a is lower than that under the first condition. The chilling draft feeling of the chef is resulted from high air velocity or low temperature. However, the distance between supply air points and the chef is 0.6cm; the unreasonable air velocity of conditioned supply sharpens the chilled draft feeling. In order to improve the chilling draft feeling in addition to satisfying the chef's comfort and respiration demand, demand control ventilation, surging flow and spot cooling at frequent intervals can be applied in the kitchen ventilation. The temperatures in Fig.4b indicate the air temperatures are much higher than shown in Fig.3b, especially in the operating zone. The temperatures of the operating points shown in Fig.4c are approximately 2℃ higher than that shown in case 1. PMV Contours shown in Fig.4d indicate the PMV value of the operating zone is approximately 1.30, bigger than that shown in Fig.3d.

Under this circumstance, human thermal sensation is little hotter than slightly warm evaluated by P O Fanger's seven-point thermal sensation scale [3]. On the contrary, human thermal sensation is relatively comfort when human works in the former thermal environment. A comparison of Fig.4e and Fig.3e shows that PPD value of operating zone in case 2 is approximately 10 percent higher than that in case 1.

3.2 Energy consumption analysis

The intake air is usually serving three functions：(1) Make-up conditioned air satisfies with thermal comfort requirement of the chef or their respiration demand, and (2) the oxygen demand of the fuel combustion and (3) provides replacement air for that exhausted air rate, and keeps reasonable negative pressure(approximately -5Pa) in the room. The results of calculation indicate that the conditioned air supply rates of two different cases are 2,304 m^3/h and 5,760 m^3/h, respectively. Furthermore, the total intake air rate $\boldsymbol{Q_{in}}$ and the total draft capacity $\boldsymbol{Q_{draft}}$ are kept changeless. So energy saving of the former is as high as 60% compared with the latter.

4. CONCLUSION

CFD simulations of indoor thermal environmental properties in Chinese commercial kitchen have been successfully carried out using indoor zero-equation model. The results have been obtained for two different ventilation systems and indicate the comparative conclusions as follows:

- Two ventilation systems are good to enhance capture and containment performance of cooking effluent effectively the spot cooling can provide more comfortable thermal environment for the chef, but supply air parameters of the spot diffuser are restricted.
- Some new methods such as demand control ventilation, spot cooling at frequent intervals with adjustable location are commended to improve comfort level of the operating zone.
- The Mathematical model should be farther modified combined with the fact; furthermore, more properties under different conditions should be simulated, seeking for effective methods to enhance the performance of ventilation system.

REFERENCES

[1] Smith Vernon A, Swierczyna Richard T, Claar Charles N. Application and enhancement of the standard test method for the performance of commercial kitchen ventilation systems[J], ASHRAE Transactions, 1995,101(2):594-605

[2] ASHRAE.1991.1991 ASHRAE handbook --HVAC applications, SI ed. Atlanta: American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc.

[3] Derek W Schrock, Quantifying kitchen comfort using a thermal mannequin [J], ASHRAE

Transactions,2002, 108(1):971-977

[4] Shiming DENG. Ventilation for Chinese kitchens in hotels in Hong Kong, Energy and the Environment----Proceedings of the International Conference on Energy and the Environment [C], Shanghai, 2003

[5] Qingyan Chen,Weiran Xu, A zero-equation turbulence model for indoor airflow simulation[J],Energy and Buildings,1998, 28 (2):137-144

[6] Chengwu LI. Ventilation design of the kitchen in hotels and restaurants [J].Heating ventilation&Airconditioning,1997, 7(4):56-5