Simulation and Optimization of Flow Field in Jujube Hot Air Drying Room

Rui LIU1,*, Ji-xian DONG1,2, Dong WANG1,2, Jia-hao WEN1 and Yi WANG1

1School of Mechanical and Electrical Engineering, Shaanxi University of Science and Technology, Xi’an 710021, China;
2Light Industrial Xi’an Mechanic Design Research Institute, Xi’an 710086, China
757967607@qq.com

Abstract. Numerical simulation of the flow field of chrysanthemum hot air drying chamber before and after optimization was carried out by using porous media model. To solve this problem, by setting the optimization scheme of the turbulence fan, the wind speed non-uniformity coefficient M in multiple sections was reduced from 125.51%, 149.27%, 144.81%, 135.63% and 106.45% to 90.44%, 116.15%, 86.92%, 77.49% and 76.67%, respectively. After flow field optimization, airflow velocity in the middle and rear of drying chamber increases significantly, and airflow uniformity in the whole space is improved.

1. Introduction

In the field of product processing, hot air drying room is an important way to reduce postpartum rotting loss. But there are still defects in the drying process of hot air drying room. The wind speed is an important factor affecting the drying characteristics of materials. When the hot air enters the drying room and is not distributed evenly, the materials are not uniformly heated by the wind, which greatly affects the drying quality [1-4]. Therefore, the application of numerical simulation method to optimize the flow field in the drying room to improve the uniformity of airflow distribution in the drying room is of great significance to ensure the drying quality of the hot air drying room. In view of the above problems, this paper takes 0.5t/batch drying room of a company with more market sales as the research object, establishes the three-dimensional model of the drying room of jujube, and uses the porous medium model to simulate the indoor flow field in the drying process of materials based on FLUENT software. The indoor flow field in the drying process was studied. By installing disturbing fan, the flow field in the drying chamber was optimized and the drying quality was improved.

2. Flow Field Simulation of Hot Air Drying Room

2.1. Physical Model and Meshing

The drying room of the hot air drying room mainly includes air inlet, air return outlet, dehumidification fan and four drying vehicles, each of which is equipped with 11 material plates. The size of the drying room is 2.4m×2.77m×2.2m, the wall is made of polyurethane sandwich board with thickness of 0.05m, the diameter of the air inlet is 0.45m, and the size of the air outlet is 0.93m×0.4m. Solid Works software was used to build a three-dimensional model of the hot air drying room. After the modeling was simplified, the dryer was removed, and the material plate and materials
were simplified together into the material layer. Finally, ICEM software was imported to extract the fluid domain of the drying room and conduct grid division (as shown in Figure 1).

Figure 1. Drying room physical model  Figure 2. Grid independence validation

The variation of the average velocity of the bottom tuyere surface of different grids was monitored and the meshless verification was conducted (the verification results were shown in Figure 2. Based on the verification results and factors such as calculation accuracy, the overall number of grid of the drying room finally selected is 337w, and the grid division is shown in Figure 3.

Figure 3. Grid division of drying room

2.2. Mathematical model

The air in the drying room is a steady-state viscous flow, and the gas can be approximately an incompressible ideal gas due to its low velocity and small density change. The flow state satisfies the conservation of mass equation and conservation of momentum equation. Any fluid flow must follow the conservation of mass equation, also known as the continuity equation.

\[
\frac{\partial u_i}{\partial x_i} = 0
\]  

(1)

For incompressible fluids with constant viscosity, the N-S equation can be simplified as:

\[
\rho \frac{\partial (u_i u_j)}{\partial x_j} = \frac{\partial}{\partial x_i} \left( \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu \delta_i j \frac{\partial u_i}{\partial x_j} \right) - \frac{\partial P}{\partial x_i}
\]  

(2)

Where is the fluid density, kg/m³; \( u_i \) and \( u_j \) (i, j=1,2,3) are the average velocity components at each time, m/s; \( P \) for fluid pressure, N/m²; \( \mu \) for fluid dynamic viscosity, N-s/m²; \( x_i \) and \( x_j \) are coordinate components; \( \delta_i j \) is the function, and when \( i=j \), \( \delta_i j =1 \), and \( i\neq j \), \( \delta_i j =0 \).

RNG k-\( \varepsilon \) turbulence model can predict medium complex flows more accurately. RNG K-\( \varepsilon \) turbulence model was used for flow field simulation before and after optimization. Kinetic energy \( K \) and diffusion of gas turbulence fluctuation \( \varepsilon \) are as follows:

\[
\frac{\partial (u_i u_j)}{\partial t} + u_i \frac{\partial (u_j u_i)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \alpha_j \frac{u_i}{\rho} \frac{\partial u_i}{\partial x_j} \right] + \frac{G_i}{\rho} + \varepsilon
\]  

(3)

\[
\frac{\partial \varepsilon}{\partial t} + u_i \frac{\partial \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \alpha_j \frac{u_i}{\rho} \frac{\partial \varepsilon}{\partial x_j} \right] + C_{\mu} \frac{\varepsilon}{k} G_i - C_{\varepsilon} \frac{\varepsilon^2}{k}
\]  

(4)
Among them, 
\[ C'_{\nu} = C_\nu - \frac{\eta(1-\frac{\eta}{\eta_0})}{1+\rho_q} \eta = (2E_k - E_\nu)^{1/2} E_\nu = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \]

\[ E_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \]

Among them, 
\[ k \] is turbulent kinetic energy, m$^2$/s$^2$; \( \varepsilon \) is the dissipation rate of turbulent kinetic energy, m$^2$/s$^3$. \( \alpha_k \) and \( \alpha_\varepsilon \) are the Number of Prandtl of \( k \) and \( \varepsilon \) respectively. \( \alpha_k=1.39 \). \( C_k \) and \( C_\varepsilon \) are turbulence model coefficients, which are determined by empirical values. \( C_k=1.42 \) and \( C_\varepsilon=1.68 \). \( \eta \) is a dimensionless parameter. \( E_{ij} \) time average strain rate; \( G_k \) is the generation term of turbulent kinetic energy \( k \) caused by average velocity gradient:

\[ G_k = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \]

(5)

Among them, the \( \mu_t \) for the turbulent viscosity, \( \mu = \rho C_\mu \frac{k^2}{\varepsilon} \) ; \( \mu_{eff} \) equivalent viscosity coefficient, \( \mu_{eff}=\mu+\mu_t \); \( C_\mu, \eta_0, \beta \) are constant, \( C_\mu=0.0845 \), \( \eta_0=4.377 \), \( \beta=0.012 \).

In the drying room, jujube is piled into the material layer with irregular shape, and there are pores between jujube. The material plate and the material layer are simplified into a porous medium model instead. The air flow in the pore of the material layer is affected by jujube resistance, i.e. viscous resistance and inertia resistance [5]. The porous media model is generated by adding the momentum equation source term into the momentum equation, and the momentum equation source term is composed of two parts: viscous loss term and inertia loss term:

\[ S_i = \sum_j \frac{D_{ij}}{\rho} \left( v_j \right) \]

(6)

Where: Momentum source term in the direction of \( S_i \) —i (x, y, z); \( v \) —magnitude of velocity; \( D \) and \( C \)—the specified coefficient matrix. For a simple and isotropic porous medium, it can also be expressed as:

\[ S_i = \frac{\mu}{\alpha} v_j + C_2 \frac{1}{2} \rho |v_j| \]

(7)

Where: \( \alpha \)—permeability; \( C_2 \)—Coefficient of inertia resistance.

Viscosity resistance coefficient and inertia resistance coefficient can be calculated according to the following formula:

\[ \alpha = D_p^2 \frac{\phi}{150 (1-\phi)^3} \]

(8)

\[ C_2 = \frac{3.5 (1-\phi)}{D_p \phi^2} \]

(9)

In the formula: \( D_p \)—the average diameter of the chrysanthemum; \( \phi \)—Porosity of porous media.

2.3. The boundary conditions

| Table 1. Setting conditions of simulation |
|-----------------------------------------|
| Parameter                  | Value         |
|----------------------------|---------------|
| Material of fluid          | Air           |
| Inlet velocity/(m/s)       | 7             |
| Inlet temperature/(°C)     | 60            |
| Inlet turbulent intensity/%| 3.58          |
| Inlet hydraulic diameter/m | 0.45          |
| Viscous resistance[6]      | 479653        |
| Inertial resistance[6]      | 714           |
| Fluid porosity[6]          | 0.425         |
| Outlet                     | Outflow       |
The air inlet is set as the velocity inlet, and the inlet air temperature, velocity, inlet turbulence intensity and hydraulic diameter are set. The jujube layer is set as porous media domain, and viscosity resistance coefficient, internal resistance factor and porosity are given. The physical parameters of jujube were experimentally measured in literature [6]. No slip adiabatic wall was selected for wall boundary. The return air outlet is set as natural outflow. The parameters of specific boundary conditions in Fluent software simulation are shown in Table 1.

3. Results and Discussion

3.1. Evaluation of Uniformity of Flow Field
By introducing wind speed non-uniformity coefficient M as the evaluation index of air distribution uniformity [7]:

\[
M = \frac{\sigma_v}{\bar{v}} \times 100\% = \frac{1}{n} \sum_{i=1}^{n} (v_i - \bar{v})^2 \times 100\% \tag{10}
\]

Where: \(\sigma_v\) is the standard deviation of wind speed values at all measured points in the plane; \(\bar{v}\) is the average wind speed value of all measuring points in the plane; \(n\) is the number of measured points uniformly distributed in the plane. (Take \(11 \times 13 = 143\) measuring points uniformly distributed on ZX section and \(11 \times 13 = 143\) measuring points on XY section)

3.2. Analysis of Original Structure Flow Field
FIG. 4 shows the overall flow diagram of the flow field of the drying chamber. It can be seen from the figure that after being heated, the airflow enters the air inlet of the drying chamber with a speed of about 7m/s. After most of the airflow passes through the jujube layer of the first two drying vehicles, it gradually diffused. The other part flows horizontally through the jujube layer of the last two vehicles and flows upward along the wall of the drying room and then flows out. A small part passes through the jujube layer from top to bottom, generating eddy in the lower part of the drying chamber, and flowing out along the wall where the drying chamber inlet and outlet are located.

3.3. To Optimize the Flow Field

Figure 5. Comparison of M values of ZX section  
Figure 6. Comparison of M values of XY section
In view of the uneven distribution of air distribution in the front and rear space along the X direction and the upper and lower space along the Y direction of the original drying chamber, two spoiler fans were added to the back and upper part of the drying chamber to optimize the airflow uniformity. Eight sections of drying room $Y=0.665m, 0.815m, 0.965m, 1.115m, 1.265m, 1.415m, 1.565m$ and $1.715m$ were selected, which respectively correspond to the middle section of the 4th, 5th, 6th, 7th, 8th, 9th, 10th and 11th layers from the bottom to the top of the jujube layer. After the optimized simulation results, the velocity inhomogeneity coefficient $M$ in the 8 sections was compared with the original structure, as shown in FIG.5. The velocity inhomogeneity coefficient $M$ in the optimized scheme was basically the same as the original structure on the sections $Y=0.665m, 0.815m$ and $0.965m$, while the other 5 sections were all lower than the original structure. The $m$ values of $1.115m, 1.265m, 1.415m, 1.565m$ and $1.715m$ sections decreased from $125.51\%$, $149.27\%$, $144.81\%$, $135.63\%$ and $106.45\%$ to $90.44\%$, $116.15\%$, $86.92\%$, $77.49\%$ and $76.67\%$, indicating a significant decrease in airflow uniformity. Figure 6 shows the comparison between the $M$ value of the optimized scheme and the traditional structure on the sections $Z=0.7m$ and $Z=1.8m$. The distribution of air distribution in these planes can reflect the air distribution uniformity in the upper and lower space of the drying chamber to a great extent. It can be seen from the figure that, compared with the original structure, the $M$ value decreases from $102.18\%$ and $103.75\%$ to $76.85\%$ and $78.74\%$, both of which have a large decrease after optimization.

![Figure 7. Velocity distribution contour of airflow on XY cross section](image)

FIG. 7 Contrast the original structure with the optimal scheme $Z=0.7m$ velocity cloud. As a result, the airflow velocity in the middle and rear space of the drying room increases significantly, the airflow retention phenomenon in the horizontal flow passage between the middle and rear trays is improved, the airflow velocity in the horizontal air passages between the trays is increased, and the overall airflow velocity in the drying room tends to be uniform.

4. Conclusion

Based on Fluent software, using RNG's $k-\varepsilon$ turbulence model and porous media model, the steady-state distribution of the internal flow field before and after the optimization of the drying chamber was simulated. After optimization by adding a fan to the drying chamber, the following conclusions were obtained: In the original drying room, the wind speed values in the X direction on the back of the space and up and down in the Y direction are too small, the multi-plane wind speed unevenness coefficient $M$ is large, and the uniformity of the airflow distribution is poor. After optimization, the wind speed inhomogeneity coefficient in multiple ZX sections decreased significantly from $1115m, 1.265m, 1.415m, 1.565m$ and $1.715m$ sections to $90.44\%, 116.15\%, 86.92\%, 77.49\%$ and $76.67\%$ from $125.51\%, 149.27\%, 144.81\%, 135.63\%$ and $106.45\%$ from $125.51\%, 116.15\%, 77.49\%$ and $76.67\%$ from $102.18\%$ and $103.75\%$ to $76.85\%$ and $78.74\%$. The airflow distribution uniformity of drying chamber is improved after optimization, which has a good effect on improving the drying equipment performance and drying quality of drying chamber.

References

[1] XIAO H W, GAO Z J, LIN H, et al. Air impingement drying characteristics and quality of carrot cubes [J]. Journal of Food Process Engineering, 2010,33(5):899-918.

[2] Gao Zhenjiang. Experimental study on drying mechanism and parameters of particulate matter
impinged by gas jet [D]. Beijing: China Agricultural University, 2000: 1–7.

[3] Wang Lihong, Gao Zhenjiang, Xiao Hongwei, et al. Gas jet impingement drying kinetics of Saint Nugo [J]. Journal of Jiangsu University (Natural Science Edition), 2011, 32(5): 540–544.

[4] Mou Guoliang, Zhang Xuejun, Shi Zenglu. Improved design of circulating dryer based on Fluent [J]. Acta Agriculturae Zhejiangensis, 2015, 27(04): 684–689.

[5] Huo Erguang. Simulation and Optimization Study on airflow organization in Chrysanthemum drying room [D]. Nanchang: Nanchang University, 2016: 31–39.

[6] Mou Guoliang. Research on Airflow Field Simulation in the Circulating Dryer of Chinese Jujube [D]. Urumqi: Xinjiang Agricultural University, 2014.

[7] Wang Jian, Dong Jixian, Wang Dong, et al. Numerical simulation and structure optimization of air distribution chamber in fruit and vegetable drying box [J]. Journal of Shaanxi University of Science and Technology, 2019, 37(1): 128–134.