Flow field uniformity analysis and structural optimization of coal pyrolysis heat exchanger

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ABSTRACT
The design of heat exchanger structure relies more on experience and lacks a guiding method. In view of this, this paper takes existing heat exchanger design structure as the research object minimizing mean square error of velocity at the outlet of coal pyrolysis heat exchanger. Optimization parameters are selected according to characteristics of model structure, and a three-dimensional model is established for each parameter value. The CFD numerical simulation method is used for trial calculation, and simulation results were numerically analyzed. The optimization parameters are taken as independent variables, numerical analysis results of mean square error are used as dependent variables to fit function curve and obtain function equation. Taguchi method is used to design an orthogonal experiment to obtain the weights of optimization parameters on objective function. Then the function equation is weighted to obtain a weighting function. Finally, genetic algorithm program is developed to find the optimal solution. Qualitative and quantitative analysis show that optimal solution results in a 36% reduction in the mean squared velocity at outlet of heat exchanger. The research results have a great significance to the design of heat exchangers.

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Coal pyrolysis; heat exchanger; Taguchi method; numerical simulation; computational fluid dynamics; genetic algorithm

Nomenclatures

\( \rho \) Density
\( t \) Time
\( \nabla \) Divergence
\( u \) Flow velocity
\( \nu \) Kinematic viscosity
\( \mu \) Dynamic viscosity
\( \rho_0 \) Uniform density
\( \omega \) Internal source term
\( g \) External source
\( k \) Turbulent kinetic energy
\( \varepsilon \) Turbulent energy dissipation rate
\( u_i \) Velocity component in the corresponding direction
\( E_{ij} \) Component of the deformation rate
\( \mu_t \) Eddy viscosity
\( D_x \) Shunt tube diameter
\( D_y \) Distributor inlet diameter
\( D_z \) Hole diameter on the baffle
\( \sigma (\gamma) \) Velocity mean variance at outlet
\( x_i \) Variable
\( \gamma \) Population mean
\( N \) Population number
\( y_i \) Observed value
\( \tilde{y} \) Observed mean value
\( \sigma \) Mean square error
\( \tilde{y}_i \) A predicted value calculated from the coefficient after regression
\( R^2 \) A statistic that will give some information about the goodness of fit of a model
\( K_i \) The sum of \( i \)-th level indicators of the same factor
\( k_i \) The average of the corresponding indicators
\( S \) Extreme deviation
\( \sigma_{inner} \) The velocity mean square error at inner ring outlet of heat exchanger
\( \sigma_{outer} \) The velocity mean square error at outer ring outlet of heat exchanger

σ summary The velocity mean square error at the outlet of heat exchanger

Introduction

In recent years, the prices of traditional energy, such as coal and oil, have continued to rise. Environmental pollution caused by burning of these fossil fuels is becoming more serious. The rational use of energy will greatly promote economic development of society (Hassani, Silva, & Al Kaabi, 2017). On the other hand, the concept of energy conservation and emission reduction
is a necessary requirement for development, and it is also an expression of attention to energy and environmental issues (Sarkar, Omair, & Choi, 2018). Coal airification technology is an important part of clean coal technology and one of the main ways of coal conversion. The goal of airification is to get as much flammable air component as possible (Minchener, 2005).

Heat exchanger is the main static equipment in the coal pyrolysis process. The main function is to transfer heat of hot fluid to cold fluid (He et al., 2017; Jang & Chen, 1997). This process involves heat conduction, heat radiation, convection heat transfer and other processes (Mehendale, Jacobi, & Shah, 2000). From the heat transfer rate equation \( Q = KS\Delta tm \), main factors affecting heat transfer effect of heat exchanger are: heat transfer coefficient \( K \), heat transfer area \( S \) and average temperature difference \( \Delta t \). Poor uniformity of the flow field of heat exchanger will reduce the heat transfer area to a certain extent, which will affect heat transfer efficiency of heat exchanger. Liu and Li (2015) used FLUENT software to optimize the pipe structure in the double-pipe smoke water medium column-tube heat exchanger, aiming at the adverse effect of uneven flow distribution of flue air on heat transfer. Through the analysis of variance, the factors affecting uniformity of smoke flow are obtained. Peng, Peterson, and Wang (1994) based on the forced-flow convection of water through rectangular microchannels having hydraulic diameters of 0.133–0.367 mm and aspect rations of H/W = 0.333–1 was investigated experimentally. The geometric parameters were found to be important variables that could significantly affect the flow characteristics and heat exchanger. Wang, Hao, and Li (2017) based on CFD numerical simulation method, taking the average velocity and flow field uniformity as the indicators, studied the fluid distribution effect of longitudinal section and inlet section of shell of guide tube in the shell-and-tube heat exchanger. It is proved that the reasonable selection of outer guide tube can improve uniformity of shell flow field of heat exchanger.

During the design process of heat exchanger, there are optimization problems (Lan, 2012). Li, Chen, and Zhang (2001) improved the inlet distribution characteristics of heat exchanger by optimizing the combined design of the seal structure, the guide vane flow angle and the structural parameters of baffle. Liu (2010) used the U-shaped heat exchanger as research object. The object-oriented programming language for MATLAB programs is used. Tube plate thickness value and a series of thickness variation curves under different design conditions are obtained. It is concluded that the u-tube heat exchanger tube plate thickness and pressure differential are proportional to the relationship, to achieve the optimization of u-tube heat exchanger tube plate design. Li, Li, and Liu (2016) pointed out the uncertainty of low temperature heat transfer in tube during the design of LNG coiled heat exchanger. FLUENT is used to analyze the pressure drop and cooling heat transfer characteristics of large pitch coil in the turbulent state. The new formula suitable for LNG low temperature cooling is fitted to provide a basis for the process calculation of LNG coiled heat exchanger. The linear weighting method is to give corresponding weight coefficients according to the importance of each parameter when multiple parameters determine an evaluation result (Loog, Duin, & Haebumbach, 2001). Li, Guo, and Guo (2015) based on thermal efficiency, energy efficiency, economic performance indicators and heat recovery efficiency, the objective function was established and linear weighted technique was used to present a new comprehensive evaluation index to evaluate and analyze ORC system of different working media. Rao and Patel (2013) introduced and applied a modified version based on the TLBO algorithm to multi-objective optimization of heat exchanger. The results of optimization using the modified TLBO are validated by comparing with those obtained by using the genetic algorithm. In computer science and operations research, a genetic algorithm (GA) is a meta-heuristic inspired by the process of natural selection that belongs to the larger class of evolutionary algorithms (EA). Genetic algorithms are commonly used to generate high-quality solutions to optimize and search problems by relying on bio-inspired operators such as mutation, crossover and selection (Connor, Clement, & Hynynen, 2002; Kopsinis & Mclaughlin, 2007). Zhang, Yang, and Zhou (2010) develop a general three-dimensional distributed parameter model (DPM) for designing the plate-fin heat exchanger (PFHE). The genetic algorithm was employed to conduct the optimization due to its robustness in dealing with complicated problems. Finally, the fin type and fin geometry were selected optimally from a customized fin database.

There are two reasons for this research work that pose considerable challenges. First, there are no models or theories that related the structural parameters of the heat exchanger to the uniformity of the flow field distribution. Second, the complexity of the heat exchanger structure. In view of the above, the author has performed establishing a model to link the structural parameters of heat exchanger with flow field distribution uniformity of heat exchanger. According to the structural characteristics of heat exchanger, the optimization parameters will be selected. The distribution characteristics of the flow field in the heat exchanger are simulated, and the program is programmed to perform a polynomial function fitting on simulation results to obtain a fitting curve and a fitting equation. The orthogonal test was designed by Taguchi method, intuitive analysis of the test results are used to
determine the weight of each optimization parameter on velocity uniformity at the outlet of heat exchanger. The final calculation equation is obtained using the weighting factors. The genetic algorithm program is programmed to solve optimal solution of weighting equation from a set of initial populations. Finally, authenticity of the optimal solution is verified by simulation method.

## Modeling method

In this paper, the motion of internal flow field of heat exchanger is studied. Air phase is treated as a continuous phase. Governing equation is conservation of the local mean variable of mass and momentum on calculation cell. The simulation of internal flow field of heat exchanger is carried out by solving simultaneous control equation. Governing equation is given by:

Mass conservation equation:

$$\frac{D\rho}{Dt} + \rho \nabla \cdot u = 0$$  \hspace{1cm} (1)

where in: $\rho$ is the density; $t$ is time; $\nabla$ is the divergence; $u$ is the flow velocity; $\frac{D\rho}{Dt}$ is a convection term; $\rho \nabla \cdot u$ is a diffusion term.

Momentum equation:

$$\frac{\partial u}{\partial t} + (u \cdot \nabla)u - \nu \nabla^2 u = -\nabla \omega + g$$  \hspace{1cm} (2)

where: $u$ is the flow velocity; $\nabla$ is the divergence; $t$ is time; $\nu = \frac{\mu}{\rho_0}$ is called the kinematic viscosity; $\omega$ is the specific(with the sense of per unit mass) thermo $\mu$ dynamic work, the internal source term; $\frac{\partial u}{\partial t}$ is variation term; $(u \cdot \nabla)u$ is convection term; $\nabla^2 u$ is diffusion term; $-\nabla \omega$ is internal source term; $g$ is external source term.

Turbulence model:

This paper uses standard K-epsilon model, which is a two-equation model that uses two transport equations (PDEs) to describe turbulence.

For turbulent kinetic energy, $k$:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho ku_i) = \frac{\partial}{\partial x_j} \left[ \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right] + 2\mu_t E_{ij} E_{ij} - \rho \varepsilon$$  \hspace{1cm} (3)

For turbulent energy dissipation rate, $\varepsilon$:

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left[ \frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_j} \right]$$

$$+ C_{1\varepsilon} \frac{\varepsilon}{k} 2\mu_t E_{ij} E_{ij} - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k}$$  \hspace{1cm} (4)

where: $u_i$ represents velocity component in the corresponding direction; $E_{ij}$ represents component of the deformation rate; $\mu_t$ represents eddy viscosity, $\mu_t = \rho C_{\mu} \frac{k^2}{\varepsilon}$.

These equations also include five adjustable constants $C_{\mu}, \sigma_k, \sigma_\varepsilon, C_{1\varepsilon}, C_{2\varepsilon}$ (Drainy, Saqr, & Aly, 2009). The values of these constants are derived from large amounts of turbulent data repeated fitting air shown in Table 1 below.

In this paper, based on existing design of heat exchanger for flow field analysis and structural optimization, the heat exchanger has two fluid domains, referred to herein as the inner and outer rings. The air enters from inlet with a diameter of 2,600 mm, and is subjected to secondary distribution at the inlet of distributor with a diameter of 968 mm. The air will enter inner-ring flow field and outer-ring flow field, respectively. A baffle having a thickness of 6 mm is provided in the middle of inner ring flow field. The baffle has two holes with a diameter of 250 mm and has three holes with a diameter of 150 mm, and the lower part of inner ring is connected with 44 tubes with a diameter of 86 mm. The outer ring flow field is connected to a shunt tube with a diameter of 420 mm. The air chamber height is 580 mm, inner arc radius is 1300 mm, and outer arc radius is 2,290 mm. The air enters the air chamber through shunt tube and then flows out through 88 small tubes with a diameter of 86 mm.

Figure 1 shows the calculation domain and grid arrangement of heat exchanger. The calculation domain is divided into small calculation cells, and the total number of grid cells is 16,935,151. In order to avoid the influence of grids number on the calculation results, the grid-independent verification is performed to verify the influence of different grids numbers on calculation results. The model is divided into three different numbers of meshes: 16,935,151, 19,376,126 and 27,856,412. The results show that the change of grid number has little effect on the calculation results, and in order to save computing time and calculation load, the number of grids is selected as 16,935,151.

Finite Difference Method is used to solve the differential equation. These data are calculated using semi-implicit method for pressure-linked equations (SIMPLE). The air enters calculation domain at a certain velocity, and finally flows out of outlet through the flow field secondary distribution. The simulation conditions determined based on experimental operating conditions is shown in Table 2. The simulation conditions were set according to description in Table 2, and calculations were

### Table 1. Constant in the turbulence model.

| $C_{\mu}$ | $\sigma_k$ | $\sigma_\varepsilon$ | $C_{1\varepsilon}$ | $C_{2\varepsilon}$ |
|----------|-----------|---------------------|-------------------|------------------|
| 0.09     | 1.00      | 1.30                | 1.44              | 1.82             |
performed in Fluent. According to the paper (Arvay, Ahmed, Peng, & Kannan, 2012), monitoring surface is placed at the inlet and outlet of heat exchanger model to monitor the mass flow at the inlet and outlet of heat exchanger. When the mass flow value of the inlet is close to the value at the outlet, then the residual curve is observed. The residual curve tends to be stable and the calculation is regarded as converged.

| Table 2. Simulation condition. |
|--------------------------------|
| Fluid Type | Air               |
| Density (kg/m³) | 1.2              |
| Import speed (m/s) | 5.5              |
| Type of export | Outflow          |
| Turbulence intensity (%) | 5                |
| Hydraulic diameter (mm) | 1295             |

**Experimental design**

Heat exchanger geometry model is shown in Figure 2. According to the structure of heat exchanger and fluid flow characteristics, it can be seen that:

(1) Since the change of shunt tube diameter will affect heat exchanger outer ring flow field resistance. The distribution rate of air flow secondary distribution in the inner ring and outer ring flow field of heat exchanger will be changed. Therefore, shunt tube diameter is selected as an optimization variable, which is recorded as $D_x$. The original size of shunt tube diameter is 420 mm.

(2) When air enters the distributor, a secondary distribution is performed. Due to the change of inlet distributor diameter, air distribution rate of inner ring and outer ring flow fields is affected. And there are not too many constraints with other structures in actual design. Therefore, distributor inlet diameter is selected as an optimization variable, which is recorded as $D_y$. The original size of distributor inlet diameter is 968 mm.

(3) There are two big holes and three small holes on the baffle, and diameter of the holes affects inner ring flow resistance. Therefore, the hole diameter is selected as an optimization variable, which is recorded as $D_z$. The original size of the hole diameter is 250 mm.
Figure 3. Influencing factors.

Table 3. The simulation size of optimized parameters.

| Dx (mm) | 340  | 370  | 390  | 420  | 450  | 460  | 470  |
|---------|------|------|------|------|------|------|------|
| Dy (mm) | 700  | 800  | 900  | 968  | 1000 | 1120 | 1200 |
| Dz (mm) | 180  | 200  | 220  | 250  | 270  | 300  | 320  |

Table 4. Simulation results about $D_x$, $D_y$, and $D_z$.

| $D_x$ (mm) | 340  | 370  | 390  | 420  | 450  | 460  | 470  |
|------------|------|------|------|------|------|------|------|
| $\sigma_{D_x}$ | 1.119 | 1.445 | 1.370 | 1.317 | 1.173 | 1.038 | 0.801 |
| $D_y$ (mm) | 700  | 800  | 900  | 968  | 1000 | 1120 | 1200 |
| $\sigma_{D_y}$ | 1.068 | 1.276 | 1.233 | 1.317 | 1.219 | 0.930 | 1.289 |
| $D_z$ (mm) | 180  | 200  | 220  | 250  | 270  | 300  | 320  |
| $\sigma_{D_z}$ | 1.256 | 1.035 | 1.203 | 1.317 | 1.031 | 0.996 | 1.196 |

Due to the constraints on model size, the constraints optimization parameters $D_x$, $D_y$, and $D_z$ are [340, 470] mm, [700, 1200] mm, [180, 320] mm. Two parameters are fixed, other parameter is changed, the value is taken within the constraint range, and tests are performed to determine the influence of the three parameters on flow field of heat exchanger (Figure 3).

According to simulation size of the optimization parameters in Table 3, the simulation model is established, and the solution is set in Fluent according to simulation conditions described in Table 2. Then, numerical analysis is carried out to obtain velocity mean variance at outlet as shown in Table 4. Data in the table are calculated by equation (5).

$$
\sigma(\gamma) = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (x_i - \gamma)^2}
$$

where: $x_i$ is the variable; $\gamma$ is the population mean; $N$ is the population number.

Using the value of optimization parameters in the Table 4 as independent variable, the numerical analysis results of mean square error are used as dependent variables to calculate polynomial function of optimized parameters by using MATLAB. The fitted image is shown in Figure 4.

The fitting equations are:

$$
f(x) = -834.9015 + 8.2922x - 0.0307x^2 + 5.0664 \times 10^{-5}x^3 - 3.1238 \times 10^{-8}x^4
$$

$$
f(y) = -931.1533 + 5.1423x - 0.0112x^2 + 1.2263 \times 10^{-5}x^3 - 6.6262 \times 10^{-9}x^4 + 1.4216 \times 10^{-12}x^5
$$

$$
f(z) = 697.6243 - 13.9811x + 0.1106x^2 - 0.0004x^3 + 8.3148 \times 10^{-7}x^4 - 6.3184 \times 10^{-10}x^5
$$

$R^2$ is derived from $y_i - \bar{y} = y_i - \tilde{y_i} + \tilde{y_i} - \bar{y}$. In a linear regression model, $R^2$ coefficient of determination is a statistical measure of how well the regression predictions approximate the real-data points. $R^2 = 1$ indicates that the regression predictions perfectly fit the data (Yaseen, Sulaiman, Deo, & Chau, 2019).

In general, the larger value of $R^2$, the better fitting effect of model (Yang, 2015). $R^2$ of the $D_x$, $D_y$ and $D_z$ fitting curves obtained by equation (9) as follows: $R_x^2 = 0.9927$, $R_y^2 = 0.9821$, $R_z^2 = 0.9757$.

$$
R^2 = 1 - \sum_{i=1}^{n} \frac{(y_i - \tilde{y}_i)^2}{\sum_{i=1}^{n} (y_i - \bar{y})^2}
$$

where: $y_i$ is the observed value; $\bar{y}$ is the observed mean value; $\tilde{y}_i$ is the predicted value calculated from the coefficient after regression.

It can be judged that the fitting polynomial can well reflect the relationship between optimization parameters and mean square error, and provide powerful data support for subsequent solving equation (Ardabili et al., 2018; Baghban, Jalali, Shafiee, Ahmadi, & Chau, 2019).

Taguchi method is a local optimization design method that uses orthogonal table to select test conditions and arrange test methods for tests. The advantage is that the number of tests can be effectively reduced, and an optimal combination of multiple design parameters can be
obtained with a minimum test data. The specific implementation process is shown as follows.

Firstly, optimization parameters and targets are selected. Secondly, the level of each optimization parameter is selected; generally take 3–5 (Table 5). And establishing an orthogonal table, which is a table designed according to orthogonal rules (Wu, Zhao, & Zhang, 2018). Then, the orthogonal matrix was solved by finite element analysis software. The test data are processed and analyzed. Finally, the optimization scheme is determined and verified according to the optimization target.
Table 5. Factor level table.

|  | $D_x$ (mm) | $D_y$ (mm) | $D_z$ (mm) |
|---|---|---|---|
| 1 | 340 | 800 | 200 |
| 2 | 420 | 968 | 250 |
| 3 | 470 | 1120 | 300 |

Table 6. Heat exchanger experimental scheme and orthogonal design-direct analysis.

|  | $D_x$ | $D_y$ | $D_z$ |
|---|---|---|---|
| 1 | 1 | 1 | 1 |
| 2 | 1 | 2 | 2 |
| 3 | 1 | 3 | 3 |
| 4 | 2 | 1 | 2 |
| 5 | 2 | 2 | 3 |
| 6 | 2 | 3 | 1 |
| 7 | 3 | 1 | 3 |
| 8 | 3 | 2 | 1 |
| 9 | 3 | 3 | 2 |
| $K_1$ | 4.93 | 3.60 | 3.74 |
| $K_2$ | 3.32 | 4.08 | 3.93 |
| $K_3$ | 2.97 | 3.54 | 3.55 |
| $k_1$ | 1.64 | 1.20 | 1.24 |
| $k_2$ | 1.10 | 1.37 | 1.32 |
| $k_3$ | 0.99 | 1.17 | 1.17 |
| $S$ | 0.65 | 0.20 | 0.15 |

There are $3^3 = 27$ combinations in Table 5. Only nine test analyses are required by orthogonal test design.

In order to determine influence level of different optimization parameters on the flow field uniformity, the orthogonal test results were analyzed (Xia, Zhang, & Wen, 2016) by orthogonal design-direct analysis method as shown in Table 6. $S$ is the extreme deviation that reflects the influence of this factor on the mean square error of heat exchanger outlet velocity, $S_i = \max k_i - \min k_i$, $(i = 1, 2, 3)$; $k_i$ are the average of the corresponding indicators, $k_i = \frac{K_i}{S_i}$; $K_i$ are the sum of $i$-th level indicators of the same factor. A fitting weight equation is established using the $S$ values as the weight.

Results and discussion

Constraints of optimization parameters $D_x$, $D_y$, and $D_z$ are given by the model structure constraints, [340, 470], [700, 1200], [180, 320], respectively, and the unit is mm. The new weighting equation is obtained by weighting the equations (6), (7), and (8) with range $S$ as the weight, as shown below:

$$F = 0.65f(D_x) + 0.20f(D_y) + 0.15f(D_z)$$  \hspace{1cm} (10)

Genetic algorithms provide a general framework for solving complex system optimization problems. In this paper, genetic algorithm is used to find optimal structure of heat exchanger (Kopsinis & Mclaughlin, 2007). The algorithm starts from a set of initial populations. Each set of solutions is encoded in binary form, a fitness function is selected, the fitness of each solution is evaluated, and individuals with good fitness are selected for selection, crossover, mutation, and finally the optimal individual is selected. The specific optimization solution design method is: In each iteration of the genetic algorithm, the current population obtains a new population generation through replication, crossover and mutation. Then the new generation population is brought into the objective function to calculate the adaptation value. Observed whether new population generation meets preconditions, and if not, repeated the above genetic operations (replication, crossover and mutation) with the new generation population as the current population, and proceed to the next iteration of the loop until the pre-set conditions are met (Kerkhoff & Ling, 2007; Safikhani, Khalkhali, & Farajpoor, 2011).

In this paper, $D_x = 420$ mm, $D_y = 968$ mm, and $D_z = 250$ mm are taken as the initial model. Optimal solution design scheme is obtained by writing the genetic algorithm program in MATLAB to solve the weighting equation. The optimal design is as follows: $D_x = 470$ mm, $D_y = 1138$ mm, $D_z = 290$ mm. The mean square error under optimal design is 0.8434.

According to the optimal result values of $D_x = 420$ mm, $D_y = 968$ mm, and $D_z = 250$ mm, the geometric model is built, the grid is divided, and the simulation is calculated by Fluent. Numerical analysis results are as follows: $\sigma_{inner} = 0.3755$, $\sigma_{outer} = 1.0208$, $\sigma_{summary} = 0.8492$.

After comparing of genetic algorithm results and simulation results, mean square error difference is about $-0.0058$. Considering test error, the test can prove authenticity of the optimal result. Therefore, the minimum value of mean square error at outlet of heat exchanger can be obtained as 0.8492. Optimal solution is: the diameter of shunt tube, $D_x = 470$ mm; the inlet of distributor diameter, $D_y = 1138$ mm; the diameter of hole on the baffle, $D_z = 290$ mm.

Conclusion

This study simplifies the actual working conditions, mainly simulates the internal air flow field distribution of the coal pyrolysis heat exchanger. We could establish a model to link the structural parameters of heat exchanger with its flow field distribution uniformity. And the author got following conclusion.

(1) Within the constraint range, as the value of shunt tube diameter ($D_x$) increases, velocity mean square error at the outlet of heat exchanger flow field...
increases first and then decreases; As distributor inlet diameter ($D_y$) increases, mean square error at heat exchanger outlet shows a fluctuation trend; As the diameter of hole on the baffle ($D_z$) increases, velocity mean square error at the outlet of heat exchanger flow field also show a fluctuation trend.

(2) The influence weights of factors on the velocity uniformity at the outlet of heat exchanger flow field are as follows: $S_{Dx} > S_{Dy} > S_{Dz}$. The extreme values $S$ are: 0.65, 0.20, and 0.15, respectively.

(3) Optimal solution result of the weighting equation is obtained by using MATLAB to write genetic algorithm program are: $D_x = 470$ mm, $D_y = 1138$ mm, $D_z = 290$ mm; optimal mean square error result is: 0.8434. The results of genetic algorithm are compared with simulation results to prove the reliability of genetic algorithm. Through the numerical analysis of simulation results, the minimum velocity uniformity at the outlet of heat exchanger is 0.8492.

This paper does not consider the influence of particle phase and heat on the flow field uniformity of heat exchanger. In future, the author will consider adding solid particles in the heat exchanger calculation domain and using thermal model to simulate coal pyrolysis. The actual operating conditions of heat exchanger will be simulated.

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Disclosure statement

No potential conflict of interest was reported by the authors.

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