Numerical Analysis of Influence Factors on Operation Performance of an Axial Exhaust Condenser

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Abstract. Taking the axial exhaust condenser of one gas-steam combined cycle unit as the research object, the steam flow in the shell side of the condenser was simulated under summer operating conditions. The flow distribution of the condenser in the shell side was obtained. Through the extraction of simulation data, the pressure of the condenser was obtained. The simulation results of single factor influence showed that the main factors leading to high operating pressure of the condenser under summer working condition were steam load, circulating water inlet temperature, circulating water flow and vacuum tightness.

1. Introduction
Axial exhaust condenser is one type of condenser which is installed downstream of turbine to receive the axial exhaust. Since the height foundation required for downward exhaust of steam turbine is eliminated, and the design requirements for floor height of powerhouse are reduced, it is more suitable for gas-steam combined cycle units [1, 2].

Like the traditional thermal power unit, the condenser of gas-steam combined cycle unit still plays an important role, and its performance has a great impact on the economy of the whole unit [3]. The gas-steam combined cycle units started late in our country. Most of them were imported units. The early domestic literatures about axial exhaust condenser either focused on the introduction of key points and technologies of design [4, 5], or focused on solution of technical problems in installation [6, 7]. The research on the shell side flow and the performance of axial exhaust condenser has only been reported in recent years. Based on the numerical simulation of shell side flow in reference [8], an axial exhaust condenser for solar heating was designed. The simulation of the shell side flow and the analysis of performance of one import axial exhaust condenser were carried out in reference [9]. In reference [10], the thermal performance of one export condenser was evaluated by simulating the shell side flow. Therefore, compared with the development trend of gas-steam combined cycle units in the future, the research on axial exhaust condenser in China is not enough.

Referring to the research method of shell side flow in downward exhaust condensers [11-13] which is commonly used in traditional thermal power units, the performance of the axial exhaust condenser of one gas-steam combined cycle unit under the actual running operation condition was studied in this paper. The work of the paper could help operators better understand the influencing factors of condenser performance and improve the operation performance of the condenser.
2. Numerical Computation Method

2.1. Mathematical Model
Following the same heat transfer principle with downward exhaust condenser, axial exhaust condenser also exchanges heat between the steam and the cooling water, condenses the steam, and forms low pressure downstream of turbine.

The phenomenon of flow and condensation in the shell side of condenser is complex. In order to realize the numerical simulation, it is necessary to simplify the steam flow reasonably. Reference [14] systematically describes the mathematical model of the steam flow in the shell side of a condenser: the complex flow can be simplified as the two-dimensional steady flow of a two-component mixture of steam and air. The flow and condensation in the tube bundle zone can be modeled by the distributed resistance and the distribution mass sink through porous media. Therefore, in Cartesian coordinates, the steam flow can be described in the unified form of the following equation.

\[
\frac{\partial (\beta \rho \dot{\phi})}{\partial x} + \frac{\partial (\beta \rho \dot{\phi})}{\partial y} = \frac{\partial}{\partial x} \left[ \beta \dot{u} \frac{\partial \dot{\phi}}{\partial x} \right] + \frac{\partial}{\partial y} \left[ \beta \dot{k} \frac{\partial \dot{\phi}}{\partial y} \right] + S_d
\]

(1)

In the equation, \( \beta \) means the porosity of steam flow area. \( \dot{\phi} \) can be 1, \( \dot{u} \), \( \dot{k} \), \( \dot{e} \) or \( \rho_{air} \), which means continuity equation, momentum equation, turbulent flow model equation and air component equation respectively. The steam temperature could be calculated by local steam partial pressure based on the one-to-one correspondence relationship between saturation pressure and saturation temperature. In addition, the necessary formula such as the formula for calculating the outlet temperature of cooling water, the segmented formula of heat transfer coefficient, the empirical formula of flow resistance in the pipe bundle zone and the formula for calculating steam condensation, are included in the mathematical model[14].

2.2. Realization and Verification of Numerical Calculation
The above mathematical model was solved on the FLUENT platform. By using the user-defined program function (UDF) of FLUENT, the distributed resistance and mass sink in porous media area and the changing physical properties of medium with state were realized by compiling and loading UDF.

The numerical simulation of one condenser, which was reformed, was carried out using the above method. The condenser pressure obtained by numerical simulation was 4499.5 Pa. It was in good agreement with the measured value of 4440 Pa. The verification of numerical calculation of the shell side flow in condenser was shown in reference [15].

3. Introduction of Research Object

3.1. Research Object
The research object of this paper was the axial exhaust condenser of one gas-steam combined cycle unit. The condenser was a surface heat exchanger with single flow and single shell. The tube layout of the condenser was showed in Figure 1. The relevant parameters of the condenser under design conditions were presented in Table 1.

3.2. Design Performance Evaluation
The design performance of the condenser was evaluated by numerical method in Reference [9]. The evaluation result showed that the condenser pressure under the design condition was about 3960pa, which nearly met the design requirements.

3.3. Actual Operation Situation
The actual operation condition of the above axial exhaust condenser deviated from the design condition greatly. The steam load of condenser seriously exceeded the design data. The vacuum tightness of condenser was poor. The flow rate of the cooling water could not guarantee the need of the condenser.
The operating parameters of the condenser under summer condition were given in Table 2. The data showed that: (1) The steam load was 160% higher than the design value; (2) The vacuum drop rate of 1.6 kPa/min was much higher than the requirement of “excellent” tightness evaluation (0.133 kPa/min in general); (3) The velocity of cooling water was 1.88 m/s, which is lower than the design value; (4) The inlet temperature of cooling water is 10 °C higher than the design temperature due to seasonal reasons. It was precisely because of the deviation of the above four operating conditions that the operating pressure of the condenser was much higher than the design pressure.

Table 1. The design data of an axial exhaust condenser

| Parameters                      | Unit | Value |
|---------------------------------|------|-------|
| Exhaust quantity of turbine     | t/h  | 92.3  |
| Exhaust quantity of shaft seal   | t/h  | 0.5   |
| Exhaust enthalpy of turbine     | kJ/kg| 2348.1|
| Exhaust enthalpy of shaft seal   | kJ/kg| 1912.0|
| Design pressure                 | Pa   | 3884  |
| Inlet temperature of circulating water | °C  | 20    |
| Velocity of circulating water   | m/s  | 2.28  |
| Number of cooling tubes         | pcs  | 4480  |
| Effective length of cooling tube| m    | 7.01  |
| Specifications of cooling tube  | mm/mm| 25.4/0.71 |
| Material of cooling tube        | --   | Titanium |

Table 2. The operating parameters of the axial exhaust condenser under summer condition

| Condenser Pressure (Pa) | Steam Load (t/h) | Vacuum Drop Rate (kPa/min) | Flow Rate of Cooling Water (t/h) | Inlet Temperature of Cooling Water (°C) |
|-------------------------|------------------|-----------------------------|---------------------------------|----------------------------------------|
| 13994                   | 151              | 1.6                         | 13000                           | 30                                    |

4. Numerical Simulation and Result Analysis

4.1. Meshing and Calculation Condition Setting
The two-dimensional geometric model was made for the flow area of steam of the condenser. And the geometric region was meshed with quadrilateral mesh. The total number of meshes was set as 56874 after mesh independence verification.

The inlet boundary condition of condenser was set as the mass inlet boundary condition. The mass fraction of air at the inlet of condenser was determined according to the vacuum drop rate\(^{[16]}\). The outlet boundary of condenser was set as the pressure outlet boundary. The flow in shell side of the condenser under different extraction pressure conditions could be simulated by changing the outlet boundary pressure. The RNG k-ε Turbulence model was used. In order to ensure the convergence of the calculation, the mass conservation of the air at the inlet and outlet of the condenser was taken as the convergence condition.

4.2. Simulation Result under Summer Condition
The flow in the shell side of the condenser was numerical simulated under summer condition. The distributions of all flow parameters with different extraction pressure were obtained. Figure. 2 only showed the distribution of velocity vector, heat transfer coefficient and pressure in the shell side of condenser when the extraction pressure was 13886 Pa.

The distribution of velocity vector in Figure. 2 showed that partial steam flowed directly into the tube bundle. The rest flowed around the tube bundle with a relatively high velocity along the upper, middle and lower steam channels, and then flowed into tube bundle from periphery area of tube bundle.
With the deepening of the flow in the tube bundle, the steam condensed, and the velocity of steam decreased. Finally, non-condensed steam entered the air-cooling zone for further condensing and cooling. As shown by the distribution of the heat transfer coefficient, the heat transfer coefficient was large in the periphery region of the tube bundle. It indicated that the heat transfer in these regions was good, especially the upstream tube bundle region. However, in the tube bundle region on both sides of middle steam channel, the heat transfer coefficient was small because of less steam inflow. It showed the tubes in the region didn’t play a good role in heat transfer. The distribution of the pressure displayed the pressure decreased with the steam condensation. At the same time, the pressure distribution also showed the average pressure in the shell side of the condenser was about 13900Pa.

Figure 2. The distribution of flow parameters of an axial exhaust condenser in at shell side (under summer condition).

A series of numerical simulation work for the flow in the shell side of the condenser was implemented when the extraction pressures was different. Then the mass of the non-condensed steam and the condenser pressure, which was the parameter concerned by the operators, were extracted. And the performance curve varying with the extraction pressure of condenser was drawn in Figure. 3. After matching with the performance of the extraction pump, it could be known that the condenser pressure was 13928Pa under summer condition. Compared with the data in Table 2, the result of numerical simulation calculation was close to the operation pressure of the condenser (13994 Pa).

Figure 3. The performance curve of an axial exhaust condenser with the change of extraction pressure

4.3. Influence Factor Analysis
According to the calculation result of the numerical simulation, the condenser’s operation pressure differed from the designed pressure under summer condition largely, due to the factors such as steam
load, the flow rate of cooling water, vacuum tightness and inlet temperature of the cooling water. Furthermore, the influence of single factor deviating from the design value was analysed one by one.

Table 3. The value of influencing factors and calculation results of condenser pressure

| Influence Factor                        | Steam Load (t/h) | Mass Fraction of Air at inlet (%) | Flow Rate of Cooling Water (m/s) | Inlet Temperature of Cooling Water (℃) | Condenser Pressure (Pa) | Deviation of Condenser Pressure from the Designed Value (Pa) |
|----------------------------------------|------------------|-----------------------------------|----------------------------------|----------------------------------------|-------------------------|-------------------------------------------------------------|
| Steam Load                             | 151              | 5.63×10^{-3}                      | 2.28                             | 20                                     | 6930                    | 3046                                                         |
| Vacuum Tightness                       | 92.8             | 6.77×10^{-2}                      | 2.28                             | 20                                     | 4653                    | 769                                                          |
| Flow Rate of Cooling Water             | 92.8             | 5.63×10^{-3}                      | 1.88                             | 20                                     | 4650                    | 766                                                          |
| Inlet Temperature of Cooling Water     | 92.8             | 5.63×10^{-3}                      | 2.28                             | 30                                     | 5243                    | 1359                                                         |

During the numerical calculation, the setting values of steam load, vacuum drop rate, the flow rate of cooling water and the inlet temperature of cooling water were given in Table 3. When considering one factor, its value was set as the value of running parameter, and other parameters were equal to the design data.

Figure 4 showed the numerical result of the pressure distribution within the shell side when condenser was under the working condition considering one of influence factors. The condenser pressure extracted from the numerical result and the deviation from the designed pressure were also shown in Table 3. According to calculation result in Table 3, the steam load was the most primary factor for increasing the condenser pressure. About 60% increase of steam load greatly aggravated the condensing task of condenser. Since the heat transfer area of condenser was not enough to transfer so much heat, the heat released from steam condensation could only be reduced by increasing the condenser pressure to maintain heat balance. Another main factor leading to the increase of condenser pressure in summer was the inlet temperature of cooling water. Because the saturation pressure (4246.7pa) corresponding to the inlet water temperature had exceeded the design pressure of the condense, it was inevitable to increase the condenser pressure to maintain the temperature difference between steam and cooling water. In comparison, vacuum tightness and flow rate of cooling water were the two weaker factors leading to ing the rise of the condenser pressure in summer. Although the vacuum tightness of the condenser was unqualified, the mass fraction of air at the inlet was still very small, about at the order of magnitude of 10^{-4}. The limited increase of air content didn’t lead to the significant increase in the condenser pressure. The flow rate of cooling water was not more than 20% lower than the design value. It also didn’t lead to the significant increase in the condenser pressure.

Figure 4. The pressure distribution in the shell side of an axial exhaust condenser

From the above analysis, it can be seen that the condenser studied in this paper has the defect of insufficient heat transfer area due to the heavy load of steam to be condensed. Therefore, the first
measure to solve the high pressure of the condenser is to expand the heat users, so as to reduce the heat transfer burden of the condenser. Due to the influence of climate, the influence of inlet temperature of cooling water is inevitable. Then, the other measures are checking the air leakage point of condenser, improving the circulating water system.

5. Conclusions
(1) The numerical simulation results showed that the numerical simulation could accurately obtain the performance of the axial exhaust condenser.
(2) The analysis results of the influence factors on condenser pressure showed that the main reasons for the high condenser pressure in summer are the high steam load, then the high temperature and the small flow rate of cooling water, and the poor vacuum tightness.
(3) According to the research of this paper, for the axial exhaust condenser studied in this paper, the measures to solve the high pressure are firstly to reduce the heat load of the condenser, secondly to reduce the air leakage and increase the flow rate of cooling water flow.

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References
[1] Wang, B. (2007) Introduction to Gas-Steam Combined Cycle Unit and Design of Its Condenser. Power Station Auxiliary Equipment, 28: 10-12.
[2] Wu, C., Li, H., Jiang, N. (2006) Technical Characteristics of Gas Turbine Generating Unit. Power Station Auxiliary Equipment, 27: 13-15.
[3] Wang, C. (2019) Influence of Condenser Vacuum on Economic Efficiency in Gas-Steam Combined Cycle Unit. Shanghai Energy Conservation, 38: 821-823.
[4] Dai, S., Gu, X., Xiao Y. (2015) Technical Research for Siemens Axial Exhaust Condenser Based on Patent Analysis. Journal of Shanghai Electric Technology, 8: 63-66.
[5] Gong, C., Yang, Z., He, R. (2019) Introduction to the Design and Research of Axial Exhaust Condenser. Power Station Auxiliary Equipment, 40: 27-29.
[6] Fang, W. (2016) Analysis and Research of Expansion Joint of Axial Exhaust Condenser. Power Station Auxiliary Equipment, 37: 18-20.
[7] Wang, J. (2017) Support Loan Analysis of Axial Exhaust Condenser. Equipment Manufacturing Technology, 45: 43-45.
[8] Zuo, Y. (2017) Solar Photovoltaic Design for Axial Exhaust Condenser. Science and Technology Innovation, 14: 4804.
[9] Qiang, Y., (2019) Zhang, Li., Zhu, Y. Numerical Simulation and Analysis for the Performance of Axial Exhaust Condenser. Dongfang Turbine, 11: 23-27.
[10] Jiang, T. (2020) Shell Side Numerical Simulation of Axial Exhaust Condenser. Power Station Auxiliary Equipment, 41: 41-44.
[11] Yu, M., Yao X., Wang, G. (1995) Numerical Analysis for Vapor Phase Flow and Heat Transfer Characteristics of High-power Turbine Condenser. Power Engineering, 15: 42-48.
[12] Huang, X., W. D., Wang, R. (2003) Influence of Pipe Bundle Resistance on Flow and Heat Transfer Properties of Condenser, Journal of Shanghai Jiaotong University, 50: 1035-1039.
[13] Zeng, H., Meng, Ji., Li, Z. (2011) Influence of Turbulence Parameter on Condenser Numerical Simulation, Journal of Engineering Thermophysics, 32: 1707-1710.
[14] Wang, G. (2010) Thermal Performance Value of Power Station Condenser and the Application Concerned. China Electric Power Press, Beijing.
[15] Zhang, L., Zeng, S., Cheng, H. (2016) Numerical Simulation of Condenser Pipe Bundle Layout. Thermal Power Generation, 45: 110-114.
[16] Ma, T, Jiang, A., Qie, Y. (2009) Quantitative Relationship between Vacuum Tightness and In-Leaking Air Flow Rate. Thermal Power Generation, 38: 65-67.