Study on the Influence of Moisture Dissipation on the Thermal and Humidity Environment in the Turbine Floor of Underground Pumped Storage Hydropower Plants

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Abstract The turbine floor of the main plant is an important part in an underground pumped storage hydropower plant. The health of personnel and the safe operation of equipment are endangered due to high space humidity and water seepage through the walls. In this paper, numerical simulations were used to study the thermal and humidity environment of the main plant turbine floor in summer. Two working conditions with and without the dissipation of humidity are computed and compared. The psychrometric chart was also used to analyze the air treatment process in the turbine floor. Results show that the air temperature decreases by 28% and the relative humidity increases by 34% when considering the effect of the moisture dissipation source. The results of temperature and humidity analyzed by the psychrometric chart are similar to those of numerical simulation. And validation of the numerical method is conducted. The research results have a reference significance for predicting the relative humidity distribution of the turbine floor and the arrangement of dehumidification devices at design stage.

1 Introduction

Underground hydropower station has the characteristics of huge space, large depth of burial of hydropower station, complex structure of cavern group, large heat generation and very complicated control of heat and humidity environment of cavern group. There are large differences between the heat and humidity environment design and control in underground hydropower plant and those in the above ground buildings. Underground plants often have humidity problems in summer, and indoor heat and humidity environment is crucial to the safe operation of equipment and the health of staff.

In order to get a suitable ventilation scheme to eliminate the heat generated by underground hydropower plants and electromechanical equipment, several scholars have conducted some studies on the thermal and humid environment of the main plant of underground hydropower plants. Liu et al. [1] proposed an improved K-ε model to study the ventilation system and optimization scheme of underground hydropower station by numerical and experimental methods. Li et al. [2] used a reduced-scale model to conduct experiments under 48 different operating conditions to study the airflow distribution in the Hohhot generatrix floor and to evaluate the effect of airflow distribution by analyzing the dimensionless temperature and dimensionless velocity.

In order to analyze the flow and heat transfer in the tunnel, Xue et al. [3] proposed a new temperature prediction model to predict the temperature distribution in the tunnel with or without dehumidifiers. It is worth noting that the current research on the thermal and humid environment of underground pumped storage power plants is more experimental and lacks simulation studies.

In this paper, the turbine floor of an underground pumped storage power station in Shandong Province is selected as the research object, and computational fluid dynamics would be used to simulate the thermal and humidity environment of the turbine floor in summer.

There are many factors affecting airflow parameters of turbine floor, including air supply volume, the air supply and exhaust outlet, and the distribution of the internal heat source and moisture dissipation source. The main purpose of this study is to discuss the influence of internal moisture dissipation source on the spatial airflow parameters of turbine floor.

Differences in thermal and humidity environment with and without dissipative sources of humidity would be analyzed. The results of this study might be useful for the turbine floor of the main plant design.

2 Numerical simulation

2.1 Physical model

In order to verify the reliability of the model, we applied it to the simulation study of the turbine floor of an underground pumped storage power plant.
A schematic diagram of the turbine floor of a pumped storage power plant is given in Fig. 1. The size of the turbine floor is 108.5 m long, 25.1 m wide and 8.5 m high. The turbine floor has four pier heat sources, which are simplified as cylindrical walls with a radius of 5.7 m and a height of 6.3 m. Upstream set six air supply vents (size 0.9 m × 0.3 m) height from the ground is 0.5 m. The coordinates of the center of the first air supply vent are (4.95, 0, 0.65) when viewed from the positive direction of X-axis, and the distance between the remaining air supply outlets is 20.7 m, 24.9 m, 6.0 m, 17.4 m and 27.3 m respectively. Downstream set six air supply vents (size 1.5 m × 0.9 m) height from the ground is 2.9 m. The coordinates of the center of the first air vent is (20.5, 25.5, 3.35) when viewed from the positive direction of X-axis, and the distance between the rest of the exhaust air outlets are 4.5 m, 21 m, 4.5 m, 20.4 m, 4.5 m, 21 m, 4.5 m.

Fig. 1 Schematic diagram of turbine floor model

The exhaust air outlets are directly connected to the generatrix floor and the air is exhausted into the generatrix floor corridor. According to the design scheme, the lower surface of this floor has four lifting holes and four connection stairways connected to the volute floor. The exhaust air from the volute floor enters the turbine floor through the stalk holes (quantity: 4, size 0.9 m × 0.3 m) and the stairway openings (quantity: 4, 2.4 m × 2.4 m) and is finally exhausted from the generatrix floor.

2.2 Control equations

The airflow in the turbine floor is incompressible low-speed turbulent flow, which contains complex heat and moisture transfer processes internally. The corresponding control equation is:

\[
\frac{\partial}{\partial t} (\rho \Phi) + \nabla (\rho \vec{V} \Phi) = \nabla \left[ \Gamma_\Phi \text{grad}(\Phi) \right] + S_\Phi \tag{1}
\]

where \( \rho \) is the airflow density, \( t \) is the time, and \( \vec{V} \) is the velocity vector, \( \Gamma_\Phi \) is the effective diffusion coefficient, \( S_\Phi \) is the source term. The expressions of the control and source terms corresponding to the different parameters are given in the literature [4, 5]. In this study, the standard k-\( \varepsilon \) turbulence model was used for the calculations.

2.3 Source terms and boundary conditions

According to the design scheme, the total value of heat generation is 111.12 kW. In order to simplify the calculation, these heat sources are evenly distributed on the four piers heat sources. According to the heat dissipation and the geometric size of the machine pier, the heat source is simulated by the volumetric heat source method.

The amount of moisture dissipation is uniformly converted into the moisture dissipation of the wall surface. It can be calculated by the following:

\[
W_i = w_s \times S \tag{2}
\]

Where \( w_s \) is the amount of moisture dissipation per unit area of rock wall seepage, \( S \) is the surface area of the surrounding rock wall, and \( W_i \) is the amount of moisture dissipated by the walls.

According to the design scheme, \( w_s \) is 0.5 g/(m\(^2\) • h), \( S \) is calculated as 2262.7 m\(^2\) (excluding the roof and ground), \( W_i \) is calculated as 0.314 g/s.

Considering the moisture dissipation to the internal open water ground, cold water pipes, and drains, the amount of moisture dissipation is multiplied by 1.2 times on the basis of dispersion of moisture in the rock wall.

The amount of moisture dispersed by the human body can be calculated by the following:

\[
W = w_p \times p \tag{3}
\]

Where \( w_p \) is the per capita moisture dissipation amount, \( p \) is the number of people.

According to the design scheme, \( w_p \) is 109 g/h, \( p \) is 5, and the moisture dissipation amount \( W \) is calculated as 0.151 g/s. The total moisture dissipation inside the turbine floor is about:

\[
W = 1.2W_i + W_p = 0.5278 \text{ g/s} \tag{4}
\]

Air inlet: Assignment of velocity boundary conditions. According to the design scheme of the project, the air supply to the turbine floor is from the generatrix floor with a partial exhaust air volume of \( 3 \times 10^4 \text{ m}^3/\text{h} \). In addition, \( 7 \times 10^4 \text{ m}^3/\text{h} \) exhaust air volume from the volute floor enters the turbine floor through the stalk holes and stairwell. The air supply temperature and relative humidity are determined according to the design conditions.

Exhaust air outlet: Assigning pressure to the outlet boundary, the air pressure is deduced from the area of the exhaust air outlet and the exhaust air volume.

Wall conditions: The solid wall surface adopts no-slip boundary conditions [6]. In the near-wall region of the flow field wall function that is to use the logarithmic distribution to solve the standard function method of turbulent viscosity coefficients for processing. The temperature difference between the upper and lower sides of the ceiling and floor of the turbine floor is small, so the floor and ceiling are treated as adiabatic boundary.

2.4 Grid division and convergence criteria

The non-uniform grid method is used for area division. The grid size near the entrance and exit was set to 0.2 m, and the grid size in the middle of the vertical tall space was set to 0.5 m.

During the computational iterations, the residual values of all equations are set to \( 10^{-3} \) except for the convergence standard residual value of the energy equation, which is set to \( 10^{-6} \).
2.5 Simulated conditions

The main plant is fully fresh air supply in summer, with a supply air temperature of 15°C and a supply air relative humidity RH of 90%. After the preliminary study, the air supply from the upstream side of the turbine floor (from the generator floor exhaust) has a temperature of 15.2°C and a relative humidity of 65.0%. The temperature of the supply air from the volute floor is 16.5°C and the relative humidity is 87.0%.

In order to consider the influence of moisture dissipation source on airflow parameters of turbine floor. Two simulated working conditions are set for the study of environmental parameters in the turbine floor:

Case 1: The thermal and humid environment of the turbine floor is simulated without considering the moisture dissipation from the wall surface and the personnel.

Case 2: The thermal and humid environment of the turbine floor is simulated by considering the moisture dissipation on the wall and the personnel.

3 Results and discussion

According to the design specifications [7], the main plant turbine floor summer parameters were given the following requirements: temperature $t \leq 30^\circ$C and relative humidity RH $\leq 80%$.

3.1 Numerical airflow in Case 1

Fig.2 gives the simulation results of the working area parameters of the turbine floor (1.8m high level section from the ground). It can be seen that the local low-temperature zone appears in some areas on both sides of the upstream and downstream, as the heat source is approximately concentrated near the central pier. Local low-temperature zone mainly occurs near the low-temperature air supply outlets and exhaust air outlets. However, the average temperature of the whole section is about 26.5°C, which meets the specification requirements.

The relative humidity distribution in the working area of the turbine floor is shown in Fig. 2(b). Since the dissipation of moisture from the wall is not considered, the area with high relative humidity mainly appears in the working area at the location of the downstream side exhaust. Due to the increase in moisture content after mixing the exhaust air from the generatrix floor and the return air from the volute floor (with high moisture content), the relative humidity in the area of the exhaust air outlet increases, while the relative humidity value is smaller for the exhaust air from the generatrix floor on the inlet side. However, the average relative humidity value of the entire work area section is about 44.8%, which meets the specification requirements.

3.2 Numerical airflow in Case 2

Fig.3 shows the simulation results of temperature and humidity in the working area plane after adding the moisture dissipation source to the turbine floor (considering the moisture dissipation on the wall and the moisture dissipation by personnel). From the simulation results, it can be seen that the temperature in the working area is reduced compared to that without the addition of a dissipative moisture source. The average temperature of the working area is about 19.1°C and the relative humidity of the working area is maintained in the range of 65%-74%. The combination of lower air temperature and increased absolute moisture content leads to a significant increase in relative humidity values.
3.3 Analysis of the simulation results of case 2 based on the psychrometric chart

This study proposes a method for the relative humidity analysis of the turbine floor based on the psychrometric chart. According to the design scheme, the total moisture dissipation in the turbine floor is $W=0.5278\, \text{g/s}$ and the total heat dissipation is $Q=111.12\, \text{kW}$.

The heat and humidity ratio of the turbine floor is:

$$\xi = \frac{Q}{W} = 21068.5$$  \hspace{1cm} (5)

Based on the heat dissipation in the turbine floor and the energy transfer, then the following expression is given:

$$Q = Gc_p(t_4 - t_3)$$  \hspace{1cm} (6)

where $G$ is the inlet air mass flow of the turbine floor in kg/s, $c_p$ is the specific heat of air in kJ/kg, $t_3$ is the air temperature at the mixing point, $t_4$ is the end-state point temperature in the chamber.

The psychrometric chart of the turbine floor air treatment process is given in Fig.4. According to the exhaust air parameters of the generatrix floor (temperature is 15.2℃, RH is 65.0%, corresponding to state point 1) mixed with the exhaust air from the volute floor (temperature is 16.5℃, RH is 87%, corresponding to state point 2) in the previous study, the ratio of the two parts of air volume is 3:7. The state point 3 after mixing can be determined in the psychrometric chart. The temperature rise is calculated by Eq. (4), and dry bulb temperature value of the point 4 is derived from $t_3$ and the temperature rise, and then combined with the heat and humidity ratio line can determine the location of point 4. The temperature and relative humidity values at point 4 are 19.0℃ and 69.0%.

Comparing the relative humidity values of point 4 with the numerical simulation results of working condition 2, it can be seen from Fig.3(b) that the relative humidity in most areas of the working area is in the range of 65% to 75%, with an average value of about 68.0%. In addition, the results also show the reasonableness and accuracy of the simplified treatment of moisture dispersion used in the numerical simulation process.

Using numerical simulation methods and psychrometric chart to study the influence of moisture dissipation source on the airflow parameters in the turbine floor of pumped storage power plants, the following conclusions were obtained:

(1) For the working condition of the main plant with the supply air temperature of 15℃ and relative humidity of 90%, the average temperature of the working area plane of the turbine floor is about 19.1℃ and the relative humidity is in the range of 65%-70% considering the influence of the dissipative moisture source. It meets the requirements of environmental parameters (summer temperature $\leq 30\, ^\circ\text{C}$, relative humidity $\leq 80\%$) of relevant specifications.

(2) For the air supply parameters of the main plant keeping constant, when considering the influence of its internal moisture dissipation, the relative humidity will increase for the turbine floor. When the air supply relative humidity is kept at 90%, the higher the air supply temperature of the main plant, the higher the absolute moisture content of the turbine floor, and the higher the relative humidity of the working area plane of the turbine floor. For the 20℃ air supply working condition in summer, the dehumidifier is needed to prevent the condensation.

4 Conclusion