Field test and numerical study on effective dehumidifying zones for a dehumidifier

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Abstract. The problem of moist conditions has always been a challenge for humid underground spaces. Field test and numerical simulations of two plants in the underground space are performed, one of which has independent dehumidifiers. The results reveal that the relative humidity can be 12.94% lower, and its average temperature is increased by 1.5°C with the independent dehumidifier added. The thermal and humid environment can be improved to a certain extent. A definition for the effective dehumidifying zone radius of the dehumidifier is presented. Its effective dehumidifying zone is studied. The maximum effective dehumidifying zone radius is approximately 5.0 m at the studied conditions. This outcome can aid in the design of dehumidifiers for HVAC engineers.

1 Introduction

In some humid areas, the annual average relative humidity can range between 70%~80%, and sometimes it can reach 95%~100%. The typical climate is characterized by high relative humidity and a long dehumidification period throughout the whole year [1]. As revealed, bacteria breed the fastest when the relative humidity is higher than 65% or lower than 38% [2]. Therefore, it is necessary to control the humidity.

Jarek Kurnitski [3] found that the ground moisture evaporation and pressure conditions in scrawl spaces were important in affecting humidity. Elmroth [4] analyzed the moisture behavior of crawl spaces in 1975 and gave recommendations for calculating ground moisture evaporation with constant moisture transfer coefficients. However, in industrial plants, there are typically large wet surfaces resulting in a large amount of energy consumption if still using existing ventilation and air conditioning systems to remove waste heat and moisture. Therefore, it is better to utilize stand-alone desiccant system equipment. Wei Xiaodai et al. [5] introduced a desiccant technique in underground construction. Wang Yifei [6] discovered that the CH1800RB type of dehumidifier can be more effective while meeting the need for large storerooms to lower relative humidity. S. Lowrey, G. Carrington et al. [7] investigated the potential for improving the low-temperature dehumidification capacity of a domestic dehumidifier using an evaporator economizer, with particular emphasis on the limitations caused by evaporator frosting. Li Yiwen [8] et al. built a steady mathematical model by analyzing a dehumidifier whose nominal dehumidification capacity was 10 kg/h and studied the variation trend of its desiccant effect under different air supply conditions. However, they did not analyze it quantitatively.

This study conducts field tests and numerical simulations of two plants in one underground space. The results show that the thermal and humid environment can be improved to some extent with added independent dehumidifying equipment. Furthermore, a definition for the effective dehumidifying zone radius of the dehumidifier is given. And its effective dehumidifying zone is studied.

2 Field test and numerical simulation of an underground space

2.1 Field test

Field tests were made for the temperature and moisture distribution of two plants (here named Plant A and Plant B) in an underground space in Guangdong, China. The sketch map of the two plants is shown in Fig. 1. The layout of the measuring points is shown in Fig. 2. The OC direction is the direction of the supply air jet flow of the dehumidifier, and the OA and OB directions are in the same plane as the OC at an angle of 45°.
2.2 Numerical simulation

2.2.1 Mathematical model

In order to simplify the mathematical model, the air flow in the space is assumed as follows:

The airflow is incompressible and homogeneous, and its density is constant. The viscous dissipation is neglected.

1) The continuity equation

\[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \]  

(1)

2) The momentum equation

The increase rate of fluid momentum in the micro element equals the sum of various forces acting on it. By introducing Newton shear stress formula and Stokes expression, the momentum equation of three velocity components can be obtained:

\[ \frac{\partial (\rho u)}{\partial t} + \text{div}(\rho u U) = \text{div}(\eta \text{grad} u) + S_u - \frac{\partial p}{\partial x} \]  

(2)

\[ \frac{\partial (\rho v)}{\partial t} + \text{div}(\rho v U) = \text{div}(\eta \text{grad} v) + S_v - \frac{\partial p}{\partial y} \]  

(3)

\[ \frac{\partial (\rho w)}{\partial t} + \text{div}(\rho w U) = \text{div}(\eta \text{grad} w) + S_w - \frac{\partial p}{\partial z} \]  

(4)

Where \( S_u, S_v, S_w \) are the general source terms. For incompressible fluid with constant viscosity, \( S_u = S_v = S_w = 0 \).

3) The energy equation

\[ \frac{\partial T}{\partial t} + \text{div}(U T) = \text{div}(\lambda \text{grad} T) + \rho c_p \frac{\partial T}{\partial t} \]  

(5)

4) Because the humidity of air is considered, the mass conservation equation as follows should be added:

\[ \frac{\partial (\rho c_s)}{\partial t} + \text{div}(\rho U c_s) = \text{div}[D_s \text{grad}(\rho c_s)] \]  

(6)

Notation used refers to: \( u, v, w \) are the velocity components in \( x, y \), and \( z \) directions respectively; \( \rho \) is the density of the fluid; \( P \) is the pressure; \( \eta \) is the viscosity coefficient; \( \lambda \) is the thermal conductivity; \( T \) is the temperature; \( c_p \) is the specific heat at constant pressure; \( T_F \) is the internal heat sources; \( c_s \) is the volume concentration of component \( S \); \( \rho c_s \) is the mass concentration; \( D_s \) is the diffusion coefficient of the component.

2.2.2 Physical model

CFD simulation is used to study the relative humidity and temperature distribution of the two plants tested above. The models of the two plants established are the same as those shown in Fig. 1.

2.2.3 Boundary conditions

For simplified, the boundary conditions are defined according to the measured data of the space wall to replace the water droplet gasification and steam condensation. Therefore, make the calculation of moisture load fixed in CFD when using the component transport equation.

2.2.4 Verification to the grid independence

In most cases, when the grid number is large enough, the change in the simulated results is quite small, but the time for calculation increases substantially. Therefore, a relatively appropriate grid density should be considered.

In this study, the changes in temperature and humidity are of greater concern. Moreover, the temperature and humidity are closely related to the air velocity, so the velocity distribution of a line on the cross-section is chosen to validate the grid independence. The specific grid division is shown in Table 1.

| Grid division | Grid size of the air supply outlet (m) | Grid size of the whole body (m) | Grid number |
|---------------|--------------------------------------|--------------------------------|-------------|
| A             | 0.2×0.2                              | 0.7×0.7×0.7                    | 142885      |
| B             | 0.1×0.1                              | 0.5×0.5×0.5                    | 379467      |
| C             | 0.05×0.05                            | 0.3×0.3×0.3                    | 1674334     |
| D             | 0.03×0.03                            | 0.3×0.3×0.3                    | 2102592     |

The air velocity distribution chosen above under
different kinds of grid divisions is shown in Fig. 3.

Fig. 3. Air velocity under different grid conditions.

As shown in Fig. 3, when the grid number increases from 1.67 million to 2.10 million, the change in the air velocity is very small. This shows that the grid number of 1.67 million can be regarded as grid-independent. The kind of grid C not only provides reasonable results but also meets the requirements of the grid number not being too large. Therefore, a grid number of 1,674,334 is selected.

2.2.5 Simulated results and analysis

The section of \(Z=1.7\) m, which is the height of the respiration zone and the plane of the air supply outlet of the dehumidifier, is taken for analysis, and the velocity, relative humidity, and temperature distribution are shown in Fig. 4.

Fig. 4. Air velocity, relative humidity, and temperature distribution of the two plants in section \(Z=1.7\) m: the upper is for Plant A, and the down is for Plant B.

It can be seen that the relative humidity in Plant A is approximately 83%~85%, and it is over 83% in most areas. But in Plant B, it is approximately 60%~86% and in the middle part of the plant, it is about 78%~82%. This is nearly 8% lower than that of Plant A on average. Therefore, the relative humidity in Plant B is controlled to some extent with the added dehumidifiers. However, a stagnation point still exists in some areas where the relative humidity is high. Therefore, it is necessary to understand the effective dehumidifying zone of the dehumidifiers to better control the moisture in the workshop.

3 Determination of the effective dehumidifying zone radius

3.1 Definition of the effective dehumidifying zone radius

According to the numerical simulations performed in section 2, the relative humidity along the axial direction of the air outlet of #5 dehumidifier in Plant B is selected to compare with the humidity at the same position of Plant A. The results are shown as follows.

Fig. 5. Comparison of relative humidity of Plant A and B along the axial direction of the air supply outlet of dehumidifier #5.
It shows that when the air velocity decreases to less than 10% of the supply air velocity of the dehumidifier, the relative humidity increases to more than 80%, which exceeds the value of relative humidity approved in the HVAC design manual for underground buildings [9]. Therefore, the point where the relative humidity reaches 80% is regarded as the farthest dehumidifying point of the dehumidifier. The effective dehumidifying range is regarded as the affected zone when the average air velocity is less than 10% of the supply air velocity. The distance from the effective dehumidifying range to the air supply outlet is regarded as the effective dehumidifying zone radius.

### 3.2 Field test of air distribution of dehumidifiers

Tests were made for the air distribution of three dehumidifiers (#1, #2 and #3 dehumidifiers) in Plant B. The set of test points in the section of Z=1.7 m is like in Fig. 2. The measured values of the three dehumidifiers studied above are shown in Tables 2–4.

#### Table 2 Measuring values of air velocity in three directions for #1 dehumidifier.

| Point | Poin t| Poin t| Poin t| Poin t| Poin t| Poin t |
|-------|-------|-------|-------|-------|-------|-------|
| OA n   | 6.18  | 2.24  | 0.72  | 0.33  | 0.29  | 0.08  |
| OB dir ctio n | 6.18  | 2.37  | 0.76  | 0.36  | 0.18  | 0.08  |
| OC dir ctio n | 6.18  | 4.60  | 3.20  | 2.70  | 1.58  | 0.70  |

The data show that the air velocity in OA and OB directions reduce faster than that in OC direction. This may be due to the air jet flow entrains the surrounding air and mixes with it, resulting in the rapid attenuation of the air velocity at the edge of the jet flow.

For #1 dehumidifier, in the axial direction of the air jet flow, that is, OC direction, the point where the air jet velocity decreases to 10% of the supply air velocity of the dehumidifier is between 5-6 m. And it’s about 5.61 m from the outlet by calculating with linear interpolation. That means the maximum effective dehumidifying zone of the dehumidifier is about 5.61 m. Similarly, the maximum effective dehumidifying zone is about 4.26 m and 5.13 m for #2 dehumidifier and #3 dehumidifier, respectively. Overall, the maximum effective dehumidifying zone of the dehumidifier at the studied conditions is approximately 5.0 m.

#### 4 Conclusions

1) The relative humidity of Plant B with added independent dehumidifying equipment was 12.94% lower than that of Plant A. The average temperature in Plant B was increased by 1.5°C. The overall thermal environment of plant B was improved to a certain extent.

2) A definition for the effective dehumidifying zone radius of the studied equipment is given. Moreover, the effective dehumidifying zone is studied. The radius is confirmed to be approximately 5 m according to this study, which means that within approximately 5 m, the reduction in the relative humidity is greater than that of other parts of the space.

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