Characteristics of Heat and Humidity Transfer Between Aquatic Holding Tanks and Indoor Air

Zhibin Mu[1], Hualing Zhang[2*]

912794715@qq.com[1], hlzhang@cqu.edu.cn[2*]

School of Civil Engineering, Chongqing University, Chongqing 400045 China

Abstract: This study established a comprehensive system to investigate heat and humidity transfer involving aquatic holding tanks' walls and exposed water surfaces and the indoor environment. We manipulated tank water temperature and circulating pool water volume. Measurements included air temperature and humidity within the experimental chamber, tank wall temperatures, condensation levels, and various parameters related to heat exchangers and circulating water in both tubular and shell processes. The analysis characterized heat and moisture transfer coefficients on horizontal and vertical tank surfaces across different water temperatures. Notably, we identified correlations for these coefficients in the literature and established a linear relationship between water vapor diffusion and aquaculture pool water volume. These findings have practical implications for calculating air-conditioning loads in aquatic market buildings and heating/cooling needs in holding tanks.

Keywords: aquatic tanks; heat and moisture transfer; convective heat transfer; wet transfer; vapor diffusion coefficient.

1. Introduction

Improved living standards have led to the proliferation of aquatic markets[1]. However, energy efficiency and emissions reduction challenges in these buildings persist. Currently, load calculations for aquatic market buildings, especially those for tank cooling and heating, rely mainly on estimation, lacking precision. Variations in aquatic product types result in different tank temperatures, typically ranging from 5 to 20°C [2]according to market research. These temperature variations impact the indoor heat and humidity environment. Hence, precise calculations of heat and humidity transfer between tanks and indoor air are vital for aquatic markets.

Various factors impact heat and moisture exchange on staging pond water surfaces, including water and air conditions and disturbances caused by water circulation and airflow. While empirical experiments are essential[3], previous work by Sobin VM[4] provided empirical and analytical methods for these exchanges, albeit under different airflow conditions. Robert L. Stree[5] applied the rough wall defect theory to study direct-contact heat and moisture transfer, though this differs from staging pond water surfaces. Xing Wei[6] used CFD simulations and experiments for uniform air-water heat and moisture transfer but without providing an empirical formula, making it less suitable for engineering calculations. Common formulas for water vapor evaporation under atmospheric pressure[7-8], often used for indoor heat and humidity

IDEE 2023, November 24-26, Zhengzhou, People's Republic of China

DOI 10.4108/eai.24-11-2023.2343352

calculations, require validation for holding tank conditions. Additionally, heat and humidity transfer between wall surfaces and air depends on various factors, including wall angles and airflow directions. Literature [9] offers formulas for natural convection heat transfer but doesn't consider wet surfaces, necessitating formula selection and calibration for glass wall surfaces in holding tanks.

This study refines calculation formulas for heat and humidity exchange coefficients involving the transient pool's open water surface, horizontal and vertical pool walls, and indoor air. Calibration of these formulas is performed using experimental data obtained from heat and humidity transfer tests within the transient pool. The outcomes provide a solid theoretical basis for precise heat and humidity load calculations in both transient pools and aquatic market buildings.

2. Heat Transfer Characteristics Testing in Holding Tanks

2.1 Experimental Test Methods and Operating Conditions

The experimental system for studying heat and humidity transfer in the holding tank (Figure 1) includes a refrigerant preparation system, a heat exchanger, the holding tank, an experimental chamber, temperature and flow measurement equipment, and wall-mounted dew collection units. The refrigerant system consists of a 2 kW frequency-conversion refrigeration unit with 10% sodium chloride aqueous solution as the refrigerant. A GP-125 centrifugal pump circulates water. The heat exchanger uses titanium tubing for refrigerant and a shell for 3.5% sodium chloride pool water. The temporary pool is made of 12mm transparent glass, measuring 1m×0.5m×0.8m, with a volume of 150L. An intelligent control pump, WAP-3000, circulates water in the temporary pool. The experimental chamber measures 2.9m×2.6m×2.8m and is constructed from foam sandwich color steel plates. Dewwater from the pool walls is collected in a simple tank, with separate collection for vertical and horizontal walls. Table 1 lists experimental parameters and instruments.

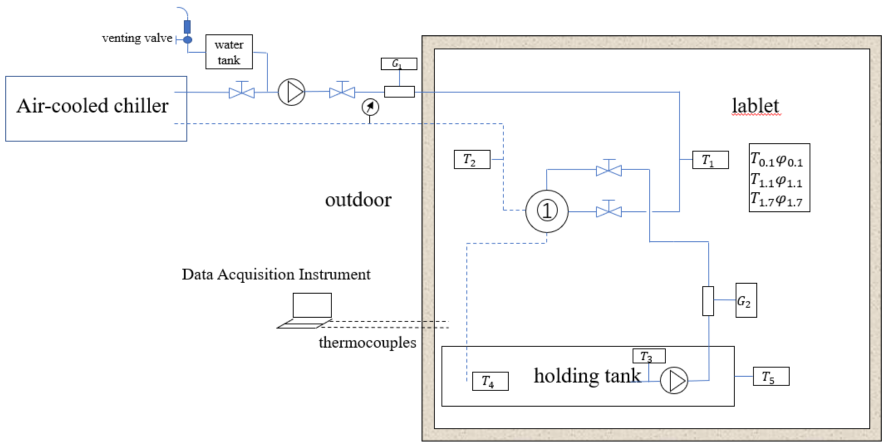


Figure 1 Experimental system and measurement point arrangement

Note the following parameters: ① Heat exchanger (HE); T_1, T_2, T_3, T_4 :Inlet and outlet temperatures for chilled water in the heat exchanger tubes and temporary pool circulating water in the shell process; T_5: Temperature at three points on the temporary pool's outer wall; $T_{0.1}\varphi_{0.1}, T_{1.1}\varphi_{1.1}, T_{1.7}\varphi_{1.7}$: Temperature and humidity measurements at different heights in the experimental chamber; G_1, G_2: Flow rates of chilled water and holding tank circulating water. As show in figure 2.

Figure 2 Staging pond for experimental testin

Table 1 Test parameters and instrumentation

test parameter	Instrument name	Measurement range	accurate
Air temperature, humidity	HOBO data loggers	20~70℃1%~95%RH	±0.21℃±5%
T_1, T_2, T_3, T_4 and T_5	K thermocouple	-40~125℃	±0.5℃
G_1	LZT-15	2~18L/min	±0.2L/min
G_2	Ultrasonic flowmeter	10~60L/min	±0.2L/min

The experimental setup, as outlined in Table 2, involved sealing the experimental chamber and initiating the circulating water pump and refrigeration unit. The refrigerant outlet temperature was meticulously adjusted, gradually reducing the water temperature in the temporary pool until it reached a stable, constant level. These controlled conditions ensured the accuracy of the experiments.

Table 2 Experimental working condition parameters

pool water temperature (℃)	G_1(L/min)	G_2(L/min)
5	16	20, 30, 40, 50
10	16	20, 30, 40, 50
15	16	20, 30, 40, 50
20	16	20, 30, 40, 50

2.2 Methods for Experimental Data Processing

2.2.1 heat balance

The heat exchange capacity of the tube course in the heat exchanger of the holding tank is Φ_1 ,The shell heat exchange is Φ_2 . The formulae are：

$$\Phi_1 = G_1 C_{p1}(T_2 - T_1) \tag{1}$$
$$\Phi_2 = G_2 C_{p2}(T_3 - T_4) \tag{2}$$

If $\left|\frac{\Phi_1 - \Phi_2}{\Phi_1}\right| \times 100\% \leq 5\%$,it is considered that the heat balance error is minimal, thereby validating the reliability of the experimental data. The heat transfer Φ of the heat exchanger is calculated using the following equation.

$$\Phi = \frac{\Phi_1 + \Phi_2}{2} \tag{3}$$

The formula is as follows: Φ: Heat exchange capacity (W). G_1, G_2: Mass flow rates of tube and shell fluids (kg/s). C_{p1}, C_{p2}: Specific heat capacities of tube and shell process fluids (J / (kg · °C)). $T_1, T_2, T_3 and T_4$: Inlet and outlet temperatures of tube and shell fluids (°C).

$$\Phi = Q_1 + Q_2 + Q_3 \tag{4}$$

Equation:

Q_1, Q_2, Q_3:Heat exchange between the open water surface, vertical wall, horizontal wall, and room air of the holding tank (W).

Q_2 and Q_3 can be determined using the wall material's thermal conductivity formula, and this enables the calculation of the convective heat transfer coefficient between the wall and indoor air. For Q_1, the coefficient for water evaporation or condensation at the pool water-air interface is computed through heat balance.

2.2.2 Wet volume balance

Indoor air humidity balance:

$$\Delta W = W_1 + W_2 + W_3 \tag{4}$$

In the formula:

ΔW: represents changes in indoor air humidity (g/h). W_1, W_2, W_3 indicate moisture exchange amounts between the open water surface, vertical wall, horizontal wall, and indoor air of the holding tank (g/h).

$$\Delta W = m(d_1 - d_2) \tag{5}$$

Equation: m represents indoor air quality (kg). d_1 and d_2 denote the calculated initial and final indoor room humidity content over the specified time period, respectively (g/kg). For humid conditions, W_2 and W_3 were collected from the side wall and bottom sinks of the holding tank, W_1 and was calculated as ΔW minus the sum of W_2 and W_3. For dry conditions, W_2 and W_3 were both set to zero.

3. Theoretical calculation model

Heat and moisture exchange in the holding tank involves both the tank's open water surface and its walls.

3.1 Open water surface

Figure 3 presents a physical model demonstrating heat and moisture exchange in open water.

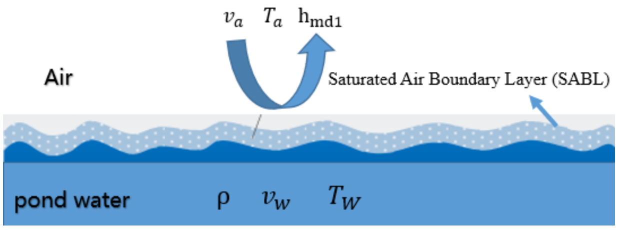


Fig. 3: Physical Model of Heat and Moisture Exchange on an Open Water Surface.

The heat (Q_1) and moisture (W_1) exchange between the open water surface and the indoor air are computed as follows.

$$Q_1 = h_{md1}(i - i_i)A_1 \tag{6}$$

$$W_1 = h_{md1}(d - d_i)A_1 \tag{7}$$

Equation (6) defines the heat exchange (Q_1) between the open water surface and indoor air in watts (W). It involves crucial parameters:A_1:The contact area between the open water surface and indoor air in square meters (m^2). h_{md1}: The moisture transfer coefficient in kg/(m^2-s).i and i_i:Enthalpy of indoor air and saturated air at water temperature in J/kg. W_1:The wet exchange volume in g/s, representing moisture transfer rate. The formula for W_1 is $(\alpha + 0.00013v)(P_{q\cdot b} - P_q)A_1\frac{B}{B'}$, In this formula:

$P_{q\cdot b}$ and P_q: Water vapor partial pressures in Pa, accounting for moisture pressure.

B and B': Atmospheric pressure in kPa, addressing pressure variations.v:Air flow velocity, assumed as 0.3 m/s. α:Water vapor diffusion coefficient, set at 0.00017 kg/(m^2-h-Pa)[7]. d and d_i—Humidity content in g/kg, indicating moisture levels.

3.2 Walls of holding tanks

When the temporary pool wall temperature is below the indoor air's dew point, condensation occurs (wet condition). When the wall temperature is above the dew point, only heat exchange transpires (dry condition).

3.2.1 Dry working conditions

Figure 4 depicts the simplified physical model for heat transfer calculations on the dry condition of the holding tank wall. Assumptions include one-dimensional thermal conductivity[9] through the tank wall and the inner wall's temperature being the same as the pool water, neglecting convective thermal resistance between the inner wall and the water.

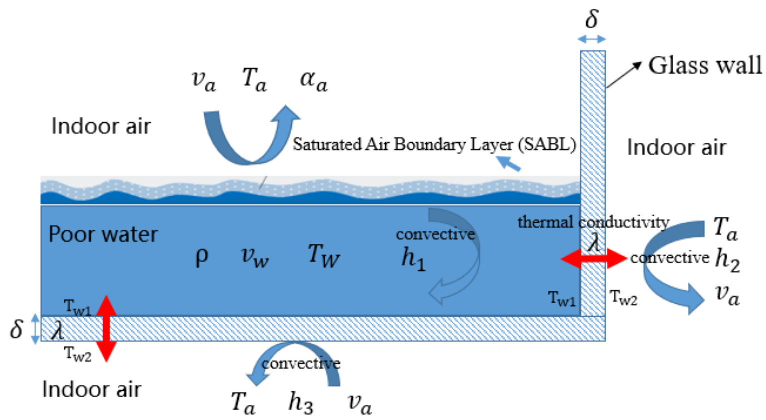


Fig. 4 Physical model for dry condition heat transfer calculation

The heat transfer from the walls of the holding tank was calculated as follows [9]:

$$Q = h_1(T_w - T_{w1})A = \frac{\lambda}{\delta}(T_{w1} - T_{w2})A = h(T_{w2} - T_a)A \tag{8}$$

In the equation:

Q—Heat transfer from the pool wall, W;

λ—Thermal conductivity of the temporary pool wall, W/(m-K);

δ—Thickness of the holding tank wall, 0.012m;

A—Heat transfer area of the pool wall, m2;

T_w—Temperature of the holding tank's pool water, °C;

T_{w1}, T_{w2}—Wall temperature inside and outside the holding tank, °C;

T_a—Indoor air temperature, °C;

h_1, h—Convective heat transfer coefficients on the temporary pool wall water-side and air-side, W / (m2- °C). The convective heat transfer coefficient h on the air-side of the temporary pool wall needs to be distinguished between vertical and horizontal walls. For vertical walls, h is calculated using equations (14) and (14)[10], while for horizontal walls, h is calculated using equations (10) and (14)[11]. The mean temperature is represented as $t_m = (T_{w2} + T_a)/2$.

$$Nu = \left\{0.825 + \frac{0.387Ra^{\frac{1}{6}}}{\left[1+\left(^{0.492}/_{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2 \tag{9}$$

$$Nu = \begin{cases} 0.54 * Ra^{\frac{1}{4}} & 2 \times 10^4 < Ra < 8 \times 10^6 \\ 0.15 * Ra^{\frac{1}{3}} & 8 \times 10^6 < Ra < 10^{11} \end{cases} \tag{10}$$

$$R_a = Gr \cdot Pr \tag{11}$$

$$Gr = \frac{g\alpha\Delta T l^3}{\nu^2} \tag{12}$$

$$Pr = \frac{\nu}{\rho} \tag{13}$$

$$\mathrm{h} = \frac{Nu}{l}\lambda \tag{14}$$

Equation: Nu : Nusselt number; Pr : Prandtl number; Gr : Grashof number; Ra: Rayleigh number; g : gravitational acceleration; α : coefficient of volumetric thermal expansion; ν : kinematic viscosity; l : characteristic length, 0.3 m for vertical walls and 0.75 m for horizontal walls; ΔT : temperature difference; ρ : air density; λ: thermal conductivity.

3.2.2 Wet working conditions

Figure 5 depicts the physical model employed for heat transfer calculations under wet conditions of the temporary pool wall. To streamline theoretical calculations, several assumptions are made in line with scientific conventions:1) The thermal resistance of the water film on the wall's surface is disregarded; 2) The temperature difference between the wall temperature of the outer wall of the holding tank and the surface of the water film is neglected; 3)The Lewis relation is applied in accordance with the experimental conditions of the holding tank, as follows:

$$\frac{h}{h_{md}} = c_p \tag{15}$$

In the equation:

h:Air convection heat transfer coefficient of the wet temporary pool wall surface, W/(m²·°C). h_{md}: Wet transfer coefficient of the wet temporary pool wall surface, kg/(m²·s).c_p: Specific heat of air, J/(kg·°C).

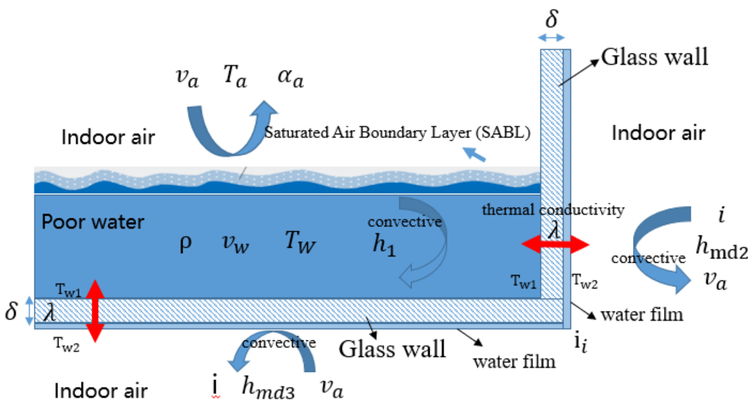


Fig. 5 Physical model for wet working conditions

1. Calculation of heat transfer in wet working condition

The air-side convective heat transfer coefficients at the pool wall for wet conditions are approximated as for dry conditions[3,12-15] .

2. Dehumidifying quantity calculation

Based on the equations $h_{md2} = \frac{h_2}{c_p}$ and $h_{md3} = \frac{h_3}{c_p}$, we can calculate h_{md2} and h_{md3}. Subsequently, we can determine the dehumidification amount of the vertical wall, W_2, and the dehumidification amount of the horizontal wall, W_3, using equations (16) and (17):

$$W_2 = h_{md2}(d - d_i)A_2 \tag{16}$$

$$W_3 = h_{md3}(d - d_i)A_3 \tag{17}$$

Equation:

W_2 and W_3 denote the dehumidification rates of the vertical and horizontal walls (g/s). Additionally, d and d_i represent indoor air moisture content and moisture content at the wet surface (g/kg), while A_2 and A_3 refer to the surface areas of the vertical and horizontal wet surfaces (m²).

4. Analysis of Holding Tank Heat Transfer Characteristics

Figure 6 shows the time points (A, B, C, D) when the pool walls transition from dry to wet conditions at different pool water temperatures, based on the experimental data of pool water temperature and chamber conditions.

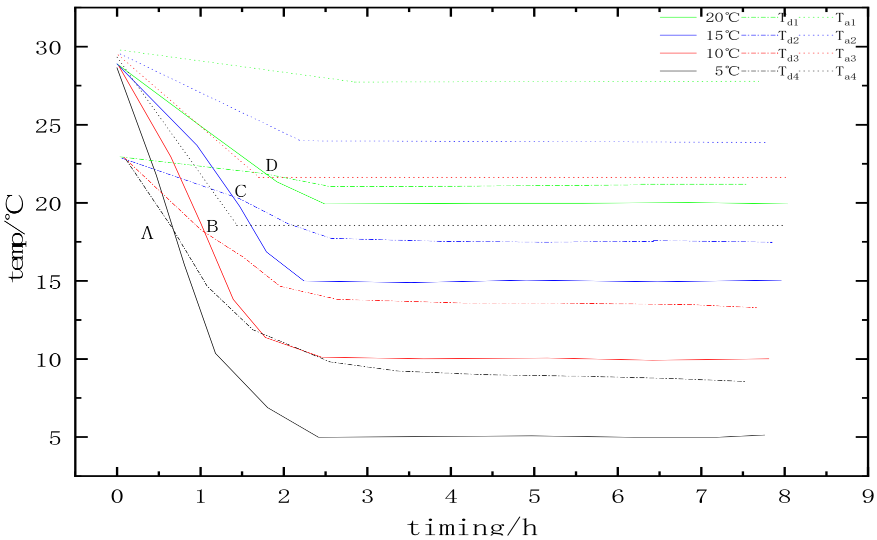
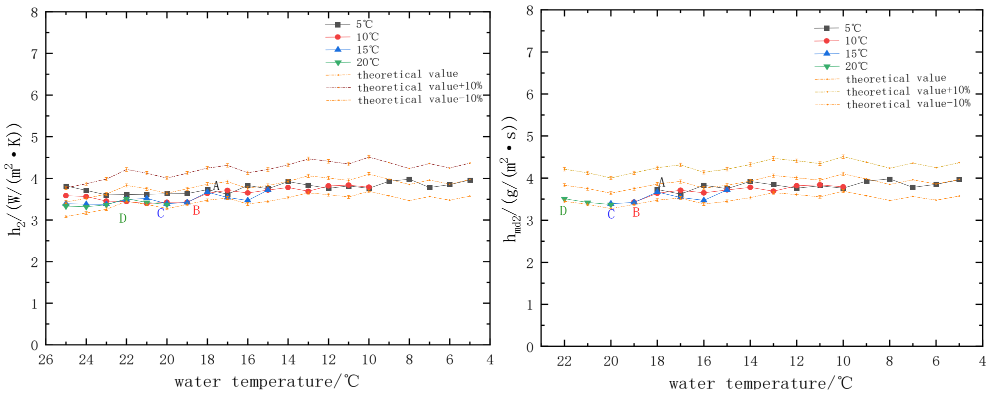


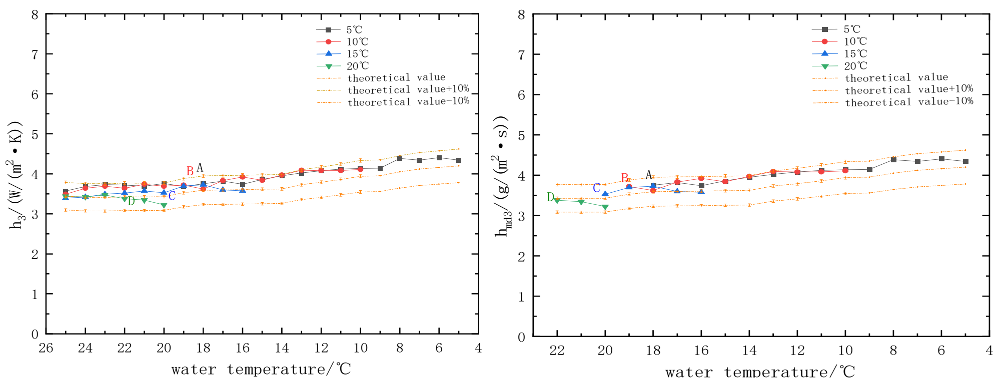
Fig. 6 Variation of pool water temperature with room air dew point temperature and dry bulb temperature

Figures 7, 8, and 9 present the determined convective heat transfer coefficients for the vertical and horizontal walls of the holding tank, along with the water vapor diffusion coefficients of the open water surface, based on the comparative analysis of experimental data and theoretical calculations.



a) h_2 b) h_{md2}

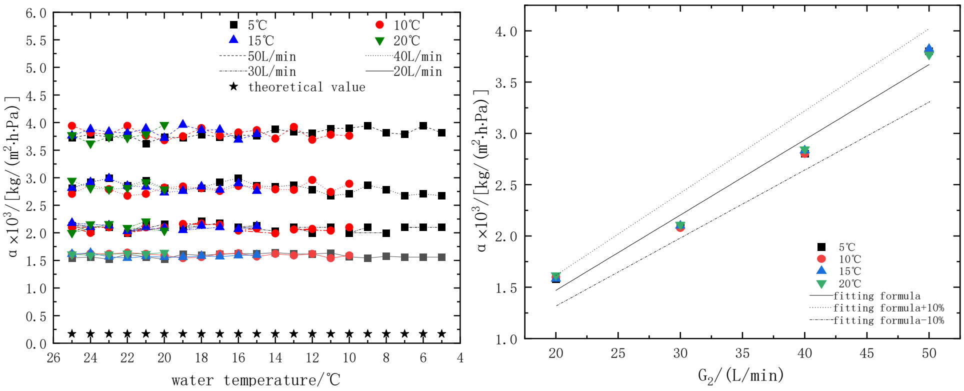
Fig. 7: Comparison of h_2 and h_{md2} Values on Vertical Wall Surface



a) h_3 b) h_{md3}

Fig. 8: Comparison of h_3 and h_{md3} Values on Vertical Wall Surface

From Figures 7 and 8, it is evident that, despite the holding tank having the same water temperature under different operating conditions, the wall's qualitative temperature varies. When the theoretical values of h_2, h_3, h_{md2}, and h_{md3} exhibit minimal variations, they closely align with the experimental values, with errors within ±10%. Consequently, equations (4), (9), and (10) can be utilized to calculate h_2 and h_{md2} for the vertical wall of the holding tank, while equations (5), (9), and (10) can be applied to determine h_3 and h_{md3} for the horizontal wall.



a) α Comparison of experimental and theoretical values

b) α and G_2 fitting formula

Fig. 9 Water vapour diffusion coefficient on open water surface

Fig. 9 shows that the experimental α significantly deviates from the literature [7]. Water temperature doesn't affect α much in varying working conditions, but α increases notably with higher holding tank circulating water flow rates (G_2). Linear regression with G_2 yielded this equation:

$$\alpha \times 10^3 = 0.0734G_2 + 0.00235 \tag{13}$$

R^2=0.97. The regression equations were well fitted.

5. Conclusion

In this paper, through the experimental method, we studied the heat and humidity transfer between the open water surface, the wall and the indoor air of the holding tank with a water volume of 150L in different water temperatures (5~25℃) and different circulating water flow rates (20~50L/min), as well as the indoor air temperature in the range of 17~28℃ and the humidity in the range of 60~80%, and we analysed the experimental data to obtain the following research conclusions:

(1) The heat exchange coefficients between the vertical walls of the holding tank and the air can be obtained by equations (4) and (9), while the moisture exchange coefficients can be calculated by equations (4), (9) and (10).

(2) The heat exchange coefficients between the horizontal walls of the holding tank and the air can be obtained by equations (5) and (9), while the wet exchange coefficients can be obtained by equations (5), (9) and (10).

(3) The water vapour diffusion coefficient on the surface of the pool water was significantly and linearly correlated with the circulating water volume of the holding tank, the total water volume of the holding tank was 150 L. The circulating water volume and the water vapour diffusion coefficient can be estimated by Eq. (13).

The research results can provide theoretical basis for the calculation of cooling design load of aquatic market holding tanks and the calculation of heat and humidity load of aquatic market buildings.

References

[1] <2019 China Fisheries Statistics Yearbook>[J]. world agriculture, 2020, (03): 2.

[2] Aquaculture should master the requirements for water temperature [J].

[3] Lian ZW. Heat and Mass Exchange Principles and Equipment. China Construction Industry Press, Beijing.

[4] Sobin V M, Dashkov G V. Heat and mass transfer under cooling of flowing water sheets by an air counter flow[J]. Journal of Engineering Physics and Thermophysics,2005,78(4):651—661.[J].

[5] Robert L.Street. Turbulent heat and mass transfers across a rough, air-water interface: A simple theory[J]. International Journal of Heat and Mass Transfer, 1979, 22: 885-899.

[6] Xing Wei,Bingbing Duan,Xuejun Zhang, et al. Numerical simulation of computational fluid dynamics for air-water direct contact heat and mass transfer[J]. Procedia Engineering, 2017, 205: 2537-2544.

[7] Uniform Technical Measures for HVAC Design of Civil Buildings 2022 (no watermark)[J].

[8] Wei WY. Swimming pool air-conditioning design [M]. Machinery Industry Press, Beijing.

[9] Zhang XM. Heat Transfer Theory. China Construction Industry Press , Beijing.

[10] Int. J. Heat Mass Transfer, Vol.18, pp.1323-1329,1975.[J].

[11] J. Heat Mass Transfer, Vol. 15, p. 755, 1972.[J].

[12] Cha XB, Zhang L, Zhang XS. Optimization of evaporative cooling condensation dehumidification combined fresh air system. Journal of Southeast University (Natural Science Edition), 2018, 48(04): 646-653.

[13] Yang TQ. Calculation of wet working conditions of a surface cooler using the wet bulb temperature method[J]. Journal of Refrigeration, 1979, (02): 30-41.

[14] Zhang EZ, Chi YT. Exploration of numerical simulation of thermal performance of the surface cooler[J]. Energy Research and Information, 2001, (04): 232-238.

[15] Zhou G, Zhou XQ, He SQ. Simplified Prediction Method for Heat Transfer Performance of Surface Coolers in Wet Conditions [J]. HVAC, 2019, 49(08): 120-126.