sustainability

MDPI

Article

Design Selection Method of Exhaust Air Heat Recovery Type Indirect Evaporative Cooler

Chunmei Guo [1], Yu Li [1], Xianli Li [1,*], Ruxue Bai [1] and Chuanshuai Dong [2]

[1] School of Energy and Safety Engineering, Tianjin Chengjian University, Tianjin 300384, China
[2] Key Laboratory of Enhanced Heat Transfer and Energy Conservation of Education Ministry, School of Chemistry and Chemical Engineering, South China University of Technology, Guangzhou 510640, China
* Correspondence: lixianliyn@163.com; Tel.: +022-23085106

Abstract: In order to promote the engineering application of indirect evaporative cooling (IEC) in the field of building air conditioning, as well as reduce air conditioning energy consumption and carbon emissions, this paper proposes a fresh air unit using indirect evaporative cooling to achieve heat recovery from exhaust air, which gives the recommended values of air and spray water operation parameters. The indirect evaporative cooler heat and mass transfer mathematical model and numerical solution procedure were made. In summer outdoor design conditions, the fresh air outlet state parameters, cooling capacity, fresh air cooling load, wet bulb efficiency and enthalpy efficiency were numerically solved for thirty typical cities from five climate zones of China. In addition, also based on the model results for the cities in China, two representative operating conditions points of medium and high humidity were selected. Eight models of fresh air unit coolers in the air volume range of 1000–10,000 m^3/h commonly used in engineering were simulated to obtain the optimal heat transfer area and size selection of ERIEC heat exchangers for fresh air units, and economic analysis was performed. The results show that the wet bulb efficiency ranges from 0.67–0.98, and increases as the outdoor design wet bulb temperature decreases; the enthalpy efficiency ranges from 0.76–1.29, and increases as the outdoor design wet bulb temperature increases; and the fresh air load that the exhaust air heat recovery type indirect evaporative cooler can bear ranges from 55–100%, which could largely decrease the cold load of the matched surface cooler. As demonstrated, the energy-saving effect is remarkable.

Keywords: exhaust air heat recovery type indirect evaporative cooler; cooling capacity; heat exchange area; water consumption; economic analysis

Citation: Guo, C.; Li, Y.; Li, X.; Bai, R.; Dong, C. Design Selection Method of Exhaust Air Heat Recovery Type Indirect Evaporative Cooler. *Sustainability* **2023**, *15*, 7371. https://doi.org/10.3390/su15097371

Academic Editor: Antonio Caggiano

Received: 23 February 2023
Revised: 6 April 2023
Accepted: 25 April 2023
Published: 28 April 2023

1. Introduction

Due to the increasingly prominent problems of energy and the climate environment, energy-saving and emission reduction work has been widely carried out in various industries around the world. According to the China Building Energy Consumption Research Report 2020, energy consumption of buildings in China was 1 billion TCE in 2018, accounting for 21.7% of the country's total energy consumption. The proportion of air conditioning energy consumption is the largest in the building operation stage, so it is necessary to improve the performance of the air conditioning system. Reducing the system energy consumption is of great significance to reduce energy demand. Relative to the traditional compression refrigeration air conditioning, the evaporative cooling air conditioning technology is more environmentally friendly, energy-saving and economical. The direct evaporative cooling technology has been widely used, while the indirect evaporative cooling technology is still in the stage of research and demonstration of application.

The experimental study from Bruno et al. [1] shows that the wet-bulb efficiency of the cross-flow indirect evaporative cooling air conditioner in commercial residential buildings is between 93–106%. For civil residential conditions, the wet bulb efficiency ranges from

118 to 129%. Moshar et al. [2] studied, through numerical simulation, the wet-bulb efficiencies of indirect evaporative cooling air conditioning in different cities from six climate zones are in the range of 62–91%. Wang et al. [3] conducted an experimental study on a fork outflow point indirect evaporative cooler. Under the working condition of 150 m^3/h primary air, the wet bulb was is 80–90%, the dew point efficiency range was 50–75%, and the cooling capacity was 0.27–0.71 kW. Fan et al. [4] designed an indirect evaporative cooler with porous ceramic material and vertical arrangement of heat exchange tubes, and, through experimental study, demonstrated that the evaporative cooler could drop 6.3 °C at the highest and the wet bulb efficiency could reach 60%. Ala Hasan et al. [5] studied three kinds of two-stage evaporative coolers (two-stage countercurrent, parallel flow and parallel regeneration combined flow) and a single-stage countercurrent regeneration cooler, and the wet bulb efficiencies were, respectively, 126%, 109%, 131% and 116%. Chen et al. [6] studied the application of a heat recovery indirect evaporative cooler in an air-conditioning system of a fresh market in a high temperature and humidity area, which had significant energy-saving effect, and the maximum COP (coefficient of performance) of the indirect evaporative cooler could reach 9.0. You et al. [7] experimentally demonstrated that the wet-bulb efficiency of indirect evaporative cooling energy heat recovery systems in high temperature and high humidity regions is in the range of 52% to 72%, with energy efficiency ratios in the range of 8 to 25. Zheng et al. [8] developed a two-dimensional mathematical model of an indirect evaporative cooling system considering primary-side condensation, and showed that the wet bulb efficiency of the indirect evaporative cooling system can reach up to 0.83 in the non-condensing condition, while it decreases to 0.5 in the condensing condition. You et al. [9] experimentally analyzed two different hydrophilic materials for indirect evaporative cooler systems and developed a three-dimensional mathematical model to study the effect of hydrophilic materials on the enthalpy efficiency, wet bulb efficiency, expansion coefficient and COP of the IEC system under different boundary conditions. Chen Qian et al. [10] presented the energy savings of a hybrid indirect evaporative cooling-mechanical vapor compression (IEC-MVC) cycle in the Saudi Arabia region, where IEC can take about 60% of the summer cooling load in arid cities and reduce energy consumption by up to 50%. Throughout the year, IEC contributes 50% of the total cooling capacity, reducing energy consumption by 40% in dry cities and only 15–25% in humid cities. The above research shows that the indirect evaporative cooler has a remarkable energy saving effect, which is worthy of further study, optimization and promotion.

In air conditioning, refrigeration and energy recovery systems using evaporative cooling technology, the design selection of evaporative cooler has a great impact on the total energy efficiency, capacity, cost and size of the heating ventilating and air conditioning (HVAC) and refrigeration system. Mao et al. [11] proposed a tubular indirect evaporative cooler made of porous ceramic materials, and designed and calculated it from the aspects of air volume, air state parameters and structure. She et al. [12] theoretically analyzed the application of the cross-flow indirect evaporative cooling air conditioning unit in the data room, optimized the unit and proposed the operation scheme. Wang et al. [13] made a comparative analysis of several different engineering application cases of indirect evaporative coolers and obtained the significant characteristics of energy saving and environmental protection of indirect evaporative coolers. Wan Yangda et al. [14] proposed a hybrid evaporatively enhanced cooling system under commercial operating conditions. The temperature and humidity of outdoor air cooled by the indirect evaporative cooler unit reduced by 6–10 °C and 2–11 g/kg, thus improving the efficiency of the cooler. Abdalazeem et al. [15] investigated the effect of different plate types on the performance of indirect evaporative coolers based on the CDF method, showing that complex plate types contribute to higher air flow rates and heat transfer rates.

The above studies have shown that the indirect evaporative cooler has great potential for air conditioning energy conservation and emissions reduction. However, the application guidance is lacking, especially for the exhaust air heat recovery type indirect evaporative cooler (energy recovery indirect evaporate cooling, ERIEC) fresh air unit. It has high energy efficiency; however, due to the imperfections in thermal performance, design selection, product development and other aspects, the use of this product is still a research pilot stage. This paper describes the design selection method of the product and analyzes its economy to promote the application of the product.

2. Structure and Working Process of ERIEC Air-Conditioning Fresh Air Unit

In order to meet the requirements of human health, a certain amount of fresh air needs to be introduced in the comfort air conditioning. At the same time, to meet the requirements of pressure balance in the building, there will be a certain amount of indoor air discharged to the outdoor environment. Whether in the cooling season or heating season, the introduction of fresh air will bring fresh air load and increase the energy consumption of air conditioning. ERIEC fresh air conditioning units provide evaporative cooling for indoor exhaust air in summer, while exchanging heat between exhaust air and fresh air to cool and dehumidify the fresh air. This paper proposes an air conditioning fresh air unit with indirect evaporative cooling and heat recovery function, together with its design and selection method. The unit structure and the working process of the air system and water system are shown in Figure 1.

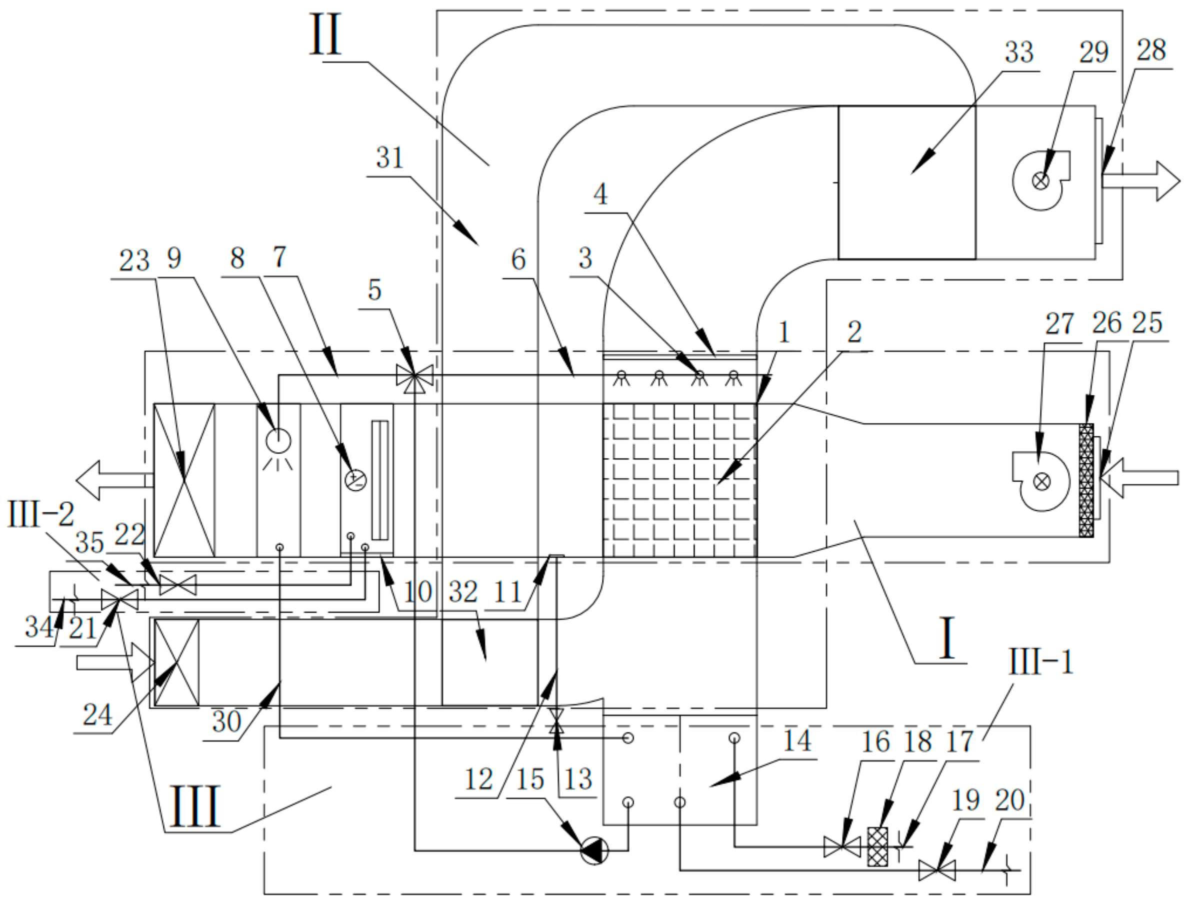

Figure 1. Schematic diagram of energy recovery indirect evaporate cooling (ERIEC) air-conditioning fresh air system.

1. Heat exchanger shell; 2. exhaust air heat recovery type indirect evaporative cooler; 3. spray water mouth; 4. manger; 5. three-way valve; 6. spray water pipe; 7. spray room water pipe; 8. surface cooler; 9. spray chamber; 10. condensate pan of surface cooler; 11. condensate tray of indirect evaporative cooling heat exchanger; 12. condensate pipe; 13. first stop valve; 14. circulating water tank; 15. circulating water pump; 16. second stop valve; 17. hydration tube; 18. tap water filter; 19. third stop valve; 20. drainpipe;

21. meter cooler return water shut-off valve; 22. meter cooler water supply shut-off valve; 23. fresh air supply air outlet; 24. exhaust air inlet; 25. fresh air inlet; 26. fresh air filter hood; 27. fresh air fans; 28. exhaust air outlet; 29. exhaust fans; 30. spray room return pipe; 31. side ventilation pipe; 32. duct insert reversing valve; 33. duct insert reversing valve; 34. cold water return pipe; 35. cold water supply pipe.

I. Fresh air system; II. exhaust system; III. water distribution system; III-1. circulating water tank water distribution loop; III-2. surface cooler circulation water distribution loop.

Fresh air system workflow: outdoor fresh air under the action of fresh air fan (27), flows through the fresh air outlet (25) and air filter (26) into the exhaust air heat recovery type indirect evaporative cooler (2), and different fresh air state points may occur in the cooling or cooling dehumidification two air treatment process. The fresh air is cooled by indirect evaporation of circulating water from the circulation tank (14) and heat exchange with the air conditioning exhaust air. If the air supply design state point can be reached, it is sent to the air conditioning room through the air supply pipe through the fresh air outlet (23). If the air supply state point is not reached, it is further cooled by the table cooler (8) dehumidification after reaching the design requirements.

Working process of exhaust system: under the action of the exhaust fan (29), the exhaust air in the air-conditioned room enters the exhaust air heat recovery type indirect evaporative cooler (2) through the exhaust inlet (24) and exhaust pipe, and the heat is transferred to the fresh air of the air conditioner while the spray water evaporates and cools, and then discharged to the outdoors.

Water distribution system workflow: the second stop valve (16) on the water replenishment pipeline is in the open state in the direction of the tap water pipe (17), and the filter (18) is in the working state during the water replenishment process. The water in the circulating water tank (14) is pressurized by the circulating water pump (15), and the first three-way valve (5) is in the open state in the direction of the spray water pipe (6), which is sprayed from the spray water nozzle (3), so that the air in the exhaust air channel can achieve evaporative cooling effect in the indirect evaporative cooler (2). The spray water will be blocked by the water baffle (4), and the spray water without evaporation will fall into the circulating water tank under the action of gravity for the reciprocal circulation process. The third shut-off valve (19) is open in the direction of the drainage pipe (20) in case the circulating water tank needs to be drained. When the surface cooler (8) is reprocessing the fresh air, the surface cooler water supply shut-off valve (22) is open in the direction of the cold water supply pipe (35), and the surface cooler water return shut-off valve (21) is open in the direction of the cold water return pipe (34). At this time, the refrigerant in the water distribution loop of the table cooler is circulated by connecting with the refrigeration unit. In summer, the condensate produced by the indirect evaporative cooler and the condensate produced by the surface cooler are discharged from the condensate tray (10 and 11) to the circulating water tank (14) through the condensate pipe (12), and at this time, the stop valve (13) on the condensate pipe (12) is in the open state.

3. ERIEC Design Methodology

3.1. Structure and Operating Parameters Design

The core component of the ERIEC air conditioning unit is an exhaust air heat recovery type indirect evaporative cooler, the main body of which consists of a number of plate heat exchanger sheets stacked at 90° to each other to form the primary and secondary air channels of the heat exchanger. The exhaust air heat recovery type indirect evaporative cooler is generally square, channel spacing is 5 mm, and heat exchanger fin material is aluminum foil, with a thickness of 0.15 mm. Zhao et al. [16] have studied the heat transfer performance of different filler forms, and demonstrated that aluminum foil is superior in heat transfer capacity, durability, compatibility with hydrophilic coating, cleanliness, cost, etc., to other forms of filler such as fiber, ceramic and carbon materials, and has advantages over other metallic materials. Compared with other metal materials, aluminum foil is often chosen as the material for the production of heat exchange fins. Figure 2 shows a 3D model of the developed exhaust air heat recovery type indirect evaporative cooler.

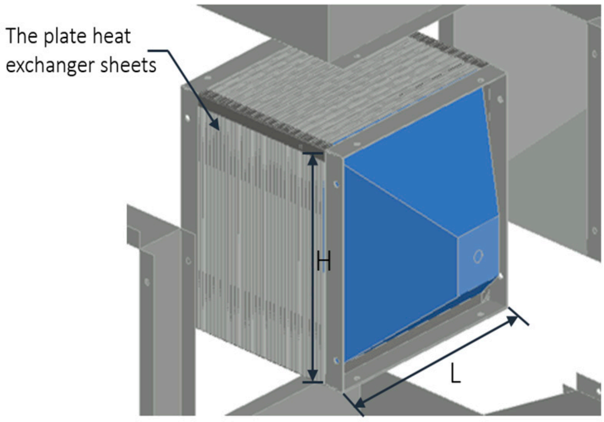

Figure 2. Physical model.

In the exhaust air heat recovery type indirect evaporative cooler with a spray device above the secondary air channels, a nozzle aperture of 1.5 mm was selected to ensure the effect of atomization, along with a nozzle installation height of 300 mm from the top of the cooler, a spray angle of 50°, and the nozzle uniformly arranged above the cooler, according to which the number of nozzle installations can be selected to achieve full coverage of the spray on the secondary channel of the cooler. Indoor exhaust air flows through the secondary side and then the heat and mass transfer occurs with the spray water in the opposite direction and the cross flowing fresh air. The flow direction of each fluid during operation is shown in Figure 3.

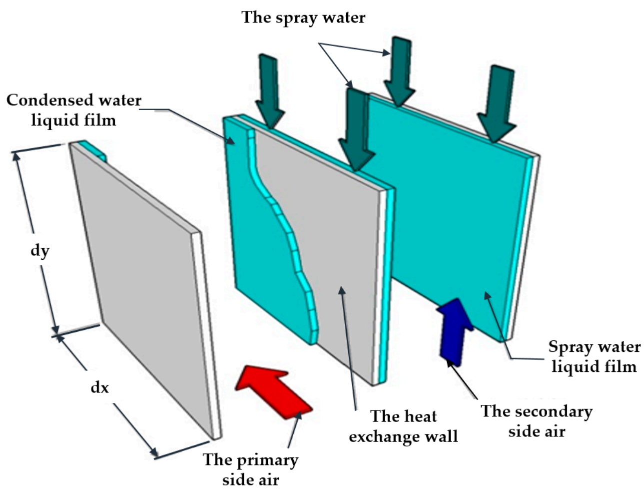

Figure 3. Fluid flow model.

3.2. Recommended Design Air Speed

Regarding the recommended design wind speed, several scholars have conducted studies, and the results are shown in Table 1; as such the recommended primary and secondary wind speed is 2–3 m/s.

Table 1. Indirect evaporate cooling (IEC) primary and secondary wind speed studies.

Research Literature	Heat Exchanger Type	Research Methods	Recommended Wind Speed (m/s)
Xing et al. [17]	Plate type indirect evaporative cooler	Theoretical Simulation	Primary wind speed of 2.232
Zang et al. [18]	Folded plate indirect evaporative cooler	Experimental Research	Primary wind speed of 2.5 Secondary wind speed 2.5–3
Li et al. [19]	Curved plate type indirect evaporative cooler	Measured calculations	Primary wind speed 3.5–5
Liu et al. [20]	Folded plate indirect evaporative cooler	Experimental Research	Primary wind speed 1.5–2.5 Secondary wind speed 2–3
Huang et al. [21]	Metal foil plate type indirect evaporative cooler	Experimental Research	Primary wind speed 2.2–2.8

3.3. Recommended Design Air Speed

The spray water flow rate needs to completely cover the heat exchange surface of the secondary air channels of the ERIEC heat exchanger, and the circulating water flow rate of the ERIEC system can be calculated according to Equation (1). Among them, according to the conclusions of the experimental study in the literature [21], the best average liquid film thickness is between 0.5–0.55 mm, and the recommended spray water density per unit length Γ is 15–20 kg/(m·h).

$$m_w = \Gamma \cdot (n+1) \cdot L/3600 \tag{1}$$

4. ERIEC Cooling Capability

4.1. ERIEC Heat Transfer Efficiency

ERIEC heat transfer efficiency can be used for both performance evaluation of heat exchangers and as a basis for equipment design selection, and is often expressed in terms of enthalpy efficiency and wet-bulb efficiency. Wet-bulb efficiency is used as a performance evaluation index for direct evaporative coolers, while for indirect evaporative cooling, enthalpy efficiency is more appropriate.

In the process of indirect evaporative cooling, a large amount of latent heat exchange is generated due to the evaporation of air on the secondary side, which reduces the wall temperature and thus improves the heat transfer performance of the system. In the case that condensation does not occur on the primary side, the wet-bulb efficiency can be used to express the heat and mass transfer efficiency of the indirect evaporative cooler. It is expressed by using the ratio of the temperature difference between the primary side air inlet and outlet and the difference between the primary side air inlet dry bulb temperature and the secondary side air inlet wet bulb temperature [22,23], as shown in Equation (2).

$$\eta_{wb} = \frac{t_{1,in} - t_{1,out}}{t_{1,in} - t_{2,wb,in}} \tag{2}$$

In the case where condensation occurs on the primary side, the enthalpy efficiency is used to express the heat transfer efficiency of the indirect evaporative cooler, which is expressed as the ratio of the enthalpy difference between the primary air inlet and outlet states and the enthalpy difference between the primary air inlet state point enthalpy and the limit state point enthalpy, as shown in Equation (4).

$$i = c_{p,g} \cdot t + (2500 + c_{p,v} t) \cdot d \tag{3}$$

$$\eta_i = \frac{i_{1,in} - i_{1,out}}{i_{1,in} - i_{\text{limit}}} \tag{4}$$

in the formula:
$c_{p,g}$—Specific heat capacity of dry air, kJ/(kg·°C);
$c_{p,v}$—Specific heat capacity of water vapor, kJ/(kg·°C);

i_{limit}—Secondary air limit enthalpy, the limiting state point is (t_{limit} = $t_{2,wb}$, d_{limit} = $d_{1,in}$), kJ/kg;

In the case of no condensation on the primary side, only the sensible heat exchange term remains in the numerator of the enthalpy efficiency, so the specific heat capacity of the numerator and denominator is eliminated, and the expression of the enthalpy efficiency is as in Equation (5).

$$\eta_i = \frac{t_{1,in} - t_{1,out}}{t_{1,in} - t_{2,wb,in}} \quad (5)$$

The limiting case of the enthalpy efficiency is also the cooling of the fresh air to the wet bulb temperature on the secondary side, so the denominator is the same as that in the wet bulb efficiency. At this time, the new efficiency index and wet bulb efficiency expression is exactly the same, proving that the wet bulb efficiency is a special case of the enthalpy efficiency expression (no condensation).

4.2. Numerical Solution of ERIEC Cooling Capacity

According to the air conditioning outdoor design state parameters, the fresh air outlet state parameters after ERIEC processing are calculated to obtain the fresh air load that ERIEC can bear.

According to the formula of enthalpy efficiency, the enthalpy of fresh air inlet and outlet should be calculated first; that is, the temperature and moisture content of fresh air inlet and outlet should be calculated. When establishing the indirect evaporative cooling heat and mass transfer differential equation, it is first assumed that there is a condensed liquid film on the heat transfer wall of the primary side channel, and assumed that the primary side channel heat exchanger wall surface exists as condensate film; the liquid film condensation area ratio is σ. σ = 0 indicates no condensation, σ = 1 indicates all condensation, and 0 < σ < 1 indicates partial condensation.

As shown in Figures 2 and 3, through the analysis of the heat and mass transfer process of the indirect evaporative cooling system, the set of differential equations is established as shown in Equations (6)–(12) [24], including the energy balance equation and the mass balance equation in the heat and mass transfer process of air convection on the primary and secondary sides; the energy balance equation of heat and mass transfer process of condensate film and spray water film; and the energy balance equation of heat transfer process on the heat exchange wall.

Energy balance equation for the convective heat and mass transfer process of air in the primary channel (6):

$$\frac{m_1}{H}dy\frac{\partial i_1}{\partial x}dx = h_1(T_c - T_1)\sigma dxdy + h_1(T_{pl} - T_1)(1-\sigma)dxdy + rh_{m1}(d_{1,w} - d_1)\sigma dxdy \quad (6)$$

Mass balance equation for air in the primary channel (7):

$$\frac{m_1}{H}dy\frac{\partial d_1}{\partial x}dx = h_{m1}(d_{1,w} - d_1)dxdy \quad (7)$$

Energy balance equation for the convective heat and mass transfer process of air in the secondary channel (8):

$$\frac{m_2}{L}dx\frac{\partial i_2}{\partial y}dy = h_2(T_w - T_2)dxdy + rh_{m2}(d_{2,w} - d_2)dxdy \quad (8)$$

Mass balance equation for air in the secondary channel (9):

$$\frac{m_2}{L}dx\frac{\partial d_2}{\partial y}dy = h_{m2}(d_{2,w} - d_2)dxdy \quad (9)$$

Heat exchange equation between condensed water film and heat exchange wall in primary channel (10):

$$\frac{m_c}{H}dyC_{pc}\frac{\partial T_c}{\partial x}dx = \frac{\lambda_c}{\delta_c}(T_{pl} - T_c)\sigma dxdy - [h_1(T_c - T_1)\sigma dxdy + rh_{m1}(d_{1,w} - d_1)\sigma dxdy] \quad (10)$$

The heat transfer equation of the spray water liquid film and heat exchange wall in the secondary channel (11):

$$\frac{m_w}{L}dxC_{pw}\frac{\partial T_w}{\partial x}dy = \frac{\lambda_w}{\delta_w}(T_{pl} - T_w)dxdy - [h_2(T_w - T_2)dxdy + rh_{m2}(d_{2,w} - d_2)dxdy] \quad (11)$$

Energy balance equation of heat exchange wall (12):

$$\frac{\partial[\lambda_{pl,x}\frac{\partial T_{pl}}{\partial x}(\delta_{pl}dy)]}{\partial x}dx + \frac{\partial[\lambda_{pl,y}\frac{\partial T_{pl}}{\partial y}(\delta_{pl}dx)]}{\partial y}dy = [\frac{\lambda_c}{\delta_c}(T_c - T_{pl})\sigma dxdy + h_1(T_1 - T_{pl})(1-\sigma)dxdy] - \frac{\lambda_w}{\delta_w}(T_{pl} - T_w)dxdy \quad (12)$$

In the cross-flow plate ERIEC, the length and height of the heat exchange wall are equal ($L = H$). In the simplification of theoretical differential equations, dimensionless coordinates are introduced $\hat{x} = x/L$, $\hat{y} = y/H$. In addition, the thermal conductivity in the length and height direction of the heat exchange wall can be determined to be consistent, i.e., $\lambda_{pl,x} = \lambda_{pl,y}$. The set of control equations for the indirect evaporative cooling heat and mass transfer process with condensation can be obtained by simplifying Equations (6)–(12) as Equations (13)–(19).

$$\frac{\partial T_2}{\partial \hat{y}} = NTU(T_w - T_2) \quad (13)$$

$$\frac{\partial d_2}{\partial \hat{y}} = \frac{NTU}{Le_{f2}}(d_{2,w} - d_2) \quad (14)$$

$$\frac{\partial T_1}{\partial \hat{x}} = \frac{k_{1,2}}{M}NTU[(T_c - T_1)\sigma + (T_{pl} - T_1)(1-\sigma)] \quad (15)$$

$$\frac{\partial d_1}{\partial \hat{x}} = \frac{k_{1,2}}{M}\frac{NTU}{Le_{f1}}(d_{1,w} - d_1)\sigma \quad (16)$$

$$\frac{\partial T_w}{\partial \hat{y}} = k_{w,2}C_w^*NTU(T_{pl} - T_w) - C_w^*NTU[(T_w - T_2) + \frac{r}{C_pLe_{f2}}(d_{2,w} - d_2)] \quad (17)$$

$$\frac{\partial T_c}{\partial \hat{x}} = k_{c,2}C_c^*NTU(T_{pl} - T_c)\sigma - k_{1,2}C_c^*NTU[(T_c - T_1)\sigma + \frac{r}{C_pLe_{f1}}(d_{1,w} - d_1)\sigma] \quad (18)$$

$$(\frac{\partial^2 T_{pl}}{\partial \hat{x}^2} + \frac{\partial^2 T_{pl}}{\partial \hat{y}^2}) = \frac{A}{\delta_{pl}^2}\{[k_{c,pl}(T_c - T_{pl})\sigma + k_{1,pl}(T_1 - T_{pl})(1-\sigma)] - k_{w,pl}(T_{pl} - T_w)\} \quad (19)$$

In the state of no condensation, there is no mass balance equation for the air condensation process in the primary side, and hence no relevant energy balance equation, and the three quantities, namely σ, d_1, T_c, are reduced. The simplified set of control equations for the indirect evaporative cooling heat and mass transfer process under no condensation is shown in Equations (20)–(24). And the expressions for defining parameters are shown in Table 2.

$$\frac{\partial T_2}{\partial \hat{y}} = NTU(T_w - T_2) \quad (20)$$

$$\frac{\partial d_2}{\partial \hat{y}} = \frac{NTU}{Le_{f2}}(d_{2,w} - d_2) \quad (21)$$

$$\frac{\partial T_1}{\partial \hat{x}} = \frac{k_{1,2}}{M}NTU(T_{pl} - T_1) \quad (22)$$

$$\frac{\partial T_w}{\partial \hat{y}} = k_{w,2} C_w^* NTU(T_{pl} - T_w) - C_w^* NTU[(T_w - T_2) + \frac{r}{C_p Le_{f2}}(d_{2,w} - d_2)] \quad (23)$$

$$(\frac{\partial^2 T_{pl}}{\partial \hat{x}^2} + \frac{\partial^2 T_{pl}}{\partial \hat{y}^2}) = \frac{A}{\delta_{pl}^2}[k_{1,pl}(T_1 - T_{pl}) - k_{w,pl}(T_{pl} - T_w)] \quad (24)$$

Table 2. Expressions defining parameters.

Parameter Meaning	Expression of Parameters
the ERIEC heat exchanger number of transfer units	$NTU = \frac{h_2 A}{m_2 c_p}$
Lewis coefficients in condensation and evaporation processes	$Le_{f1} = \frac{h_1}{h_{m1} c_p}$ $Le_{f1} = \frac{h_2}{h_{m2} c_p}$
the heat capacity ratios of secondary air to spray water and condensed water	$C_w^* = \frac{m_2 c_p}{m_w c_{pw}}$ $C_c^* = \frac{m_2 c_p}{m_c c_{pc}}$
the secondary air mass flow ratio	$M = \frac{m_1}{m_2}$
the ratios of heat transfer coefficients of spray water liquid film, condensed water liquid film and primary air to heat exchange wall	$k_{w,pl} = \frac{\lambda_w/\delta_w}{\lambda_{pl}/\delta_{pl}}$ $k_{c,pl} = \frac{\lambda_c/\delta_c}{\lambda_{pl}/\delta_{pl}}$ $k_{1,pl} = \frac{h_{s1}}{\lambda_{pl}/\delta_{pl}}$
the ratios of heat transfer coefficients of primary air, spray water liquid film and condensate water liquid film to secondary air	$k_{1,2} = \frac{h_1}{h_2}$ $k_{w,2} = \frac{\lambda_w/\delta_w}{h_2}$ $k_{c,2} = \frac{\lambda_c/\delta_c}{h_2}$

The set of indirect evaporative cooling heat and mass transfer with condensation and full condensation control equations is a two-dimensional set of partial differential equations, consisting of seven partial differential equations and four boundary conditions. The set of non-condensing control equations consists of five partial differential equations and three boundary conditions. In the numerical solution, the partial differential equation toolbox is used to carry out programming calculation based on finite element analysis method (FEM).

In the process of computer numerical solution, the heat exchange wall of ERIEC is set as a square with a side length of 1, and divided into several micro-elements. This paper sets the micro-element size to 0.025 in order to ensure that the calculation results are not affected by the size of micro-element and have a higher computer speed [24].

In order to provide guidance for ERIEC unit design, the parameters that can be entered in the parameter input section of this model are the primary and secondary air inlet temperature and humidity, primary and secondary airflow, and heat exchanger size parameters, etc., which facilitate the simulation and analysis of ERIECs with different operating conditions and different models.

In this paper, the automatic solution logic was established [25], as shown in Figure 4. The model sets the judgment error of the condensation area ratio σ as 0.01, which requires a judgment of the condensation state in the heat exchanger channel. In the calculation of the solution, before assuming that the primary side of the channel in the heat exchanger wall condensation state is the full condensation state and σ = 1, In the calculation of the solution, firstly assuming that the heat exchanger wall in the primary side of the channel is in the full condensation state, i.e., σ = 1, and then using the condensation model to solve the moisture content. When its inlet value of is less than or equal to the outlet value, the ERIEC heat exchanger can be judged at as the state of non-condensation state. And the non-condensation control equation solver can be better solved for primary and secondary air temperature, moisture content distribution results and heat transfer channels within the spray water film and heat transfer wall temperature field distribution state. On the contrary, when the moisture content of the inlet value is greater than the outlet value, it can be judged that the state of the ERIEC heat exchanger is condensation, and if the solved σ value is greater than or equal to 1, it can be further judged that the state of the ERIEC is full condensation, and the solution result is the predicted value of the model in this case. If the solved value of σ is less than 1, it is proved that the state of ERIEC is partial condensation at this time, and the solved condensate film area ratio is compared with the assumed value. If the error between the solved value and the assumed value meets the calculation error requirement, the condensation area ratio is considered correct. If the error between the two exceeds the allowable range of calculation error, the condensation state should be reassumed for calculation (gradually reducing the assumed σ value) until the error reaches within the allowable range.

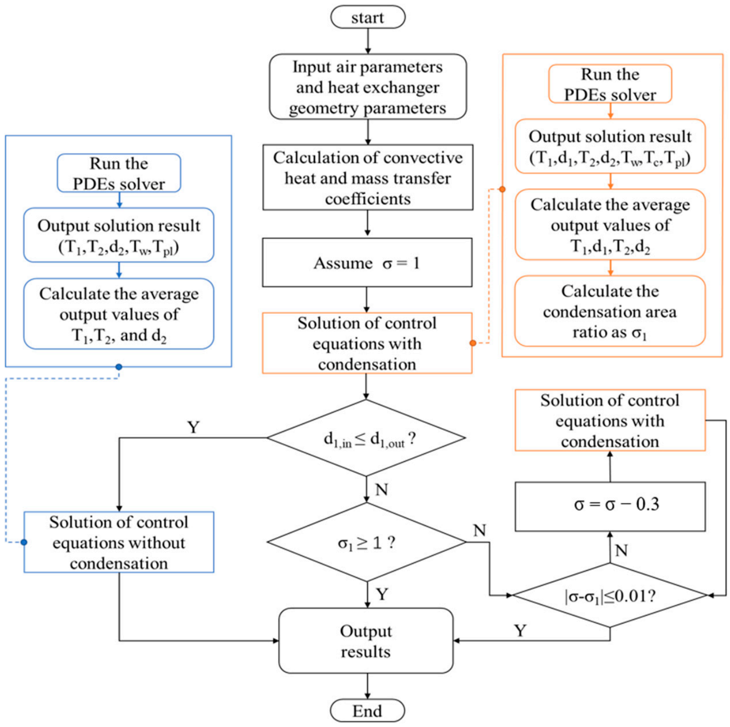

Figure 4. Flow chart of solving optimized ERIEC governing equations.

The model has been validated against experimental data, and the theoretical calculations of temperature and moisture content of primary air differ from the experimental values by a maximum of 8.6% and an average of 2.87% at four conditions of 30%, 50%, 70% and 90% relative humidity, respectively [24].

4.3. Calculation of ERIEC Cooling Capacity for Summer Air Conditioning Design Conditions in Typical Chinese Cities

Based on the meteorological data in the literature [26], thirty typical cities in five climate zones in China, namely, severe cold, cold, mild, hot summer and cold winter, and hot summer and warm winter, were selected to calculate the fresh air outlet temperature and humidity, ERIEC wet bulb efficiency, enthalpy efficiency, and fresh air cooling load that ERIEC can bear after the treatment of fresh air units based on the developed indirect evaporative cooling heat and mass transfer theoretical model under the calculated dry bulb temperature and wet bulb temperature state of outdoor air conditioning in summer. In the calculation, the temperature and humidity of fresh air into the air-conditioning room were 18 °C and 90%, and that of exhaust air out of the air-conditioning room were 26 °C and 50%. The fresh air volume and exhaust air volume were taken as 400 m^3/h, the size of the heat exchanger was 400 × 400 mm with a primary and secondary channel number of 25, and the calculation results are shown in Table 3.

Table 3. ERIEC cooling capacity of typical cities.

Climate Zones	City	W		W′		Wet-Bulb Efficiency	Enthalpy Efficiency	ERIEC Percentage %
		$t_{1,in}$ (°C)	$t_{1,wb,in}$ (°C)	$t_{1,out}$ °C	$d_{1,out}$ g/kg			
Severe cold regions	Xining	26.4	16.6	19.7	7.8	0.87	0.89	100.00
	Urumqi	33.4	18.3	20.3	7.0	0.89	0.90	100.00
	Hohhot	30.7	21.0	21.1	11.7	0.80	0.81	100.00
	Harbin	30.6	23.8	22.1	15.3	0.71	0.85	55.74
	Changchun	30.4	24.0	22.2	15.4	0.70	0.90	55.50
	Shenyang	31.4	25.2	22.7	16.1	0.69	1.04	55.60
Cold regions	Lhasa	24.0	13.5	18.8	5.4	0.98	0.97	100.00
	Lanzhou	31.3	20.1	20.8	10.2	0.83	0.85	100.00
	Yinchuan	31.3	22.2	21.5	13.1	0.78	0.80	84.30
	Taiyuan	31.6	23.8	22.1	15.3	0.74	0.78	55.74
	Xi’an	35.1	25.8	22.9	16.4	0.74	0.89	55.85
	Beijing	33.6	26.3	23.1	16.7	0.70	1.07	55.90
	Shijiazhuang	35.2	26.8	23.4	17.0	0.72	1.03	55.48
	Tianjin	33.9	26.9	23.4	17.1	0.69	1.14	55.56
	Ji’nan	34.8	27.0	23.4	17.2	0.71	1.08	55.17
	Zhengzhou	35.0	27.5	23.6	17.5	0.70	1.16	55.69
Hot summer and cold winter regions	Chengdu	31.9	26.4	23.1	16.8	0.67	1.23	55.97
	Chongqing	36.3	27.3	23.6	17.3	0.72	1.05	55.86
	Guilin	34.2	27.3	23.6	17.4	0.68	1.18	55.09
	Hangzhou	35.7	27.9	23.8	17.7	0.70	1.17	55.68
	Hefei	35.1	28.1	23.9	17.9	0.68	1.24	55.26
	Nanjing	34.8	28.1	23.9	17.9	0.68	1.26	55.26
	Nanchang	35.6	28.3	24.0	18.0	0.69	1.24	55.64
	Wuhan	35.3	28.4	24.0	18.0	0.68	1.29	56.22
	Changsha	36.5	29.0	24.3	18.5	0.69	1.28	55.42
Hot summer and warm winter regions	Guangzhou	34.2	27.8	23.8	17.7	0.67	1.26	55.18
	Fuzhou	36.0	28.1	23.9	17.9	0.70	1.21	56.87
	Haikou	35.1	28.1	23.9	17.9	0.68	1.24	55.26
Mild regions	Kunming	26.3	19.9	20.7	15.7	0.74	0.74	100.00
	Guiyang	30.1	23.0	21.8	17.3	0.73	0.76	57.24

In the summer design condition, the dry bulb temperature distribution range of the inlet fresh air for each city is 24.3–36 °C, and the dry bulb temperature distribution range of the outlet fresh air from ERIEC is reduced to 15.7–26.3 °C, which has a significant cooling effect. Wet bulb efficiencies of 0.67–0.98 for each city and a decreasing trend for each climate zone as the outdoor design wet bulb temperature increases. Enthalpy efficiency ranges from 0.76 to 1.29, with an increasing trend in each climate zone as the outdoor design wet bulb temperature rises. ERIEC can bear between 53% and 100% of the fresh air load, then the fresh air unit table cooler only needs to bear the rest of the fresh air load. The outdoor fresh air status points of thirty typical cities in five climate zones are subject to the treatment process of ERIEC fresh air units, as shown in Figures 5–9.

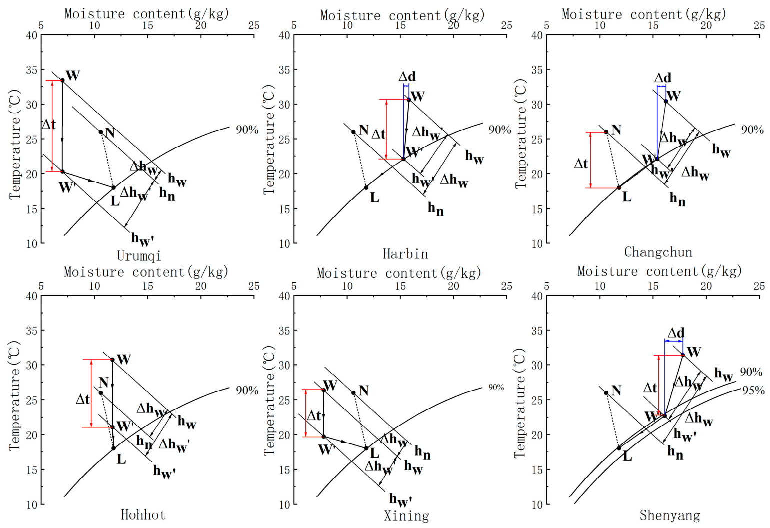

Figure 5. Psychrometric chart of fresh air treated by ERIEC in severe cold regions.

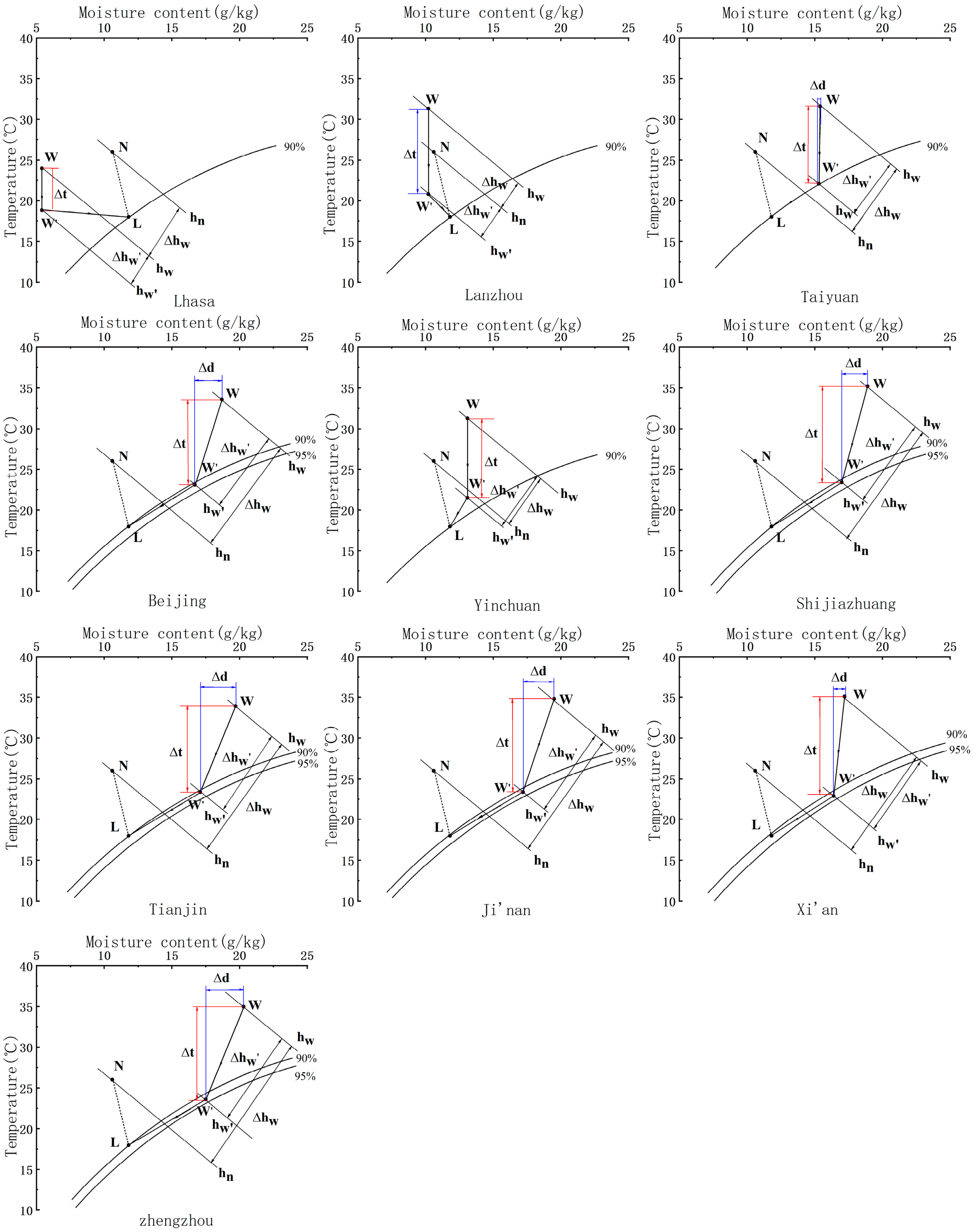

Figure 6. Psychrometric chart of fresh air treated by ERIEC in cold regions.

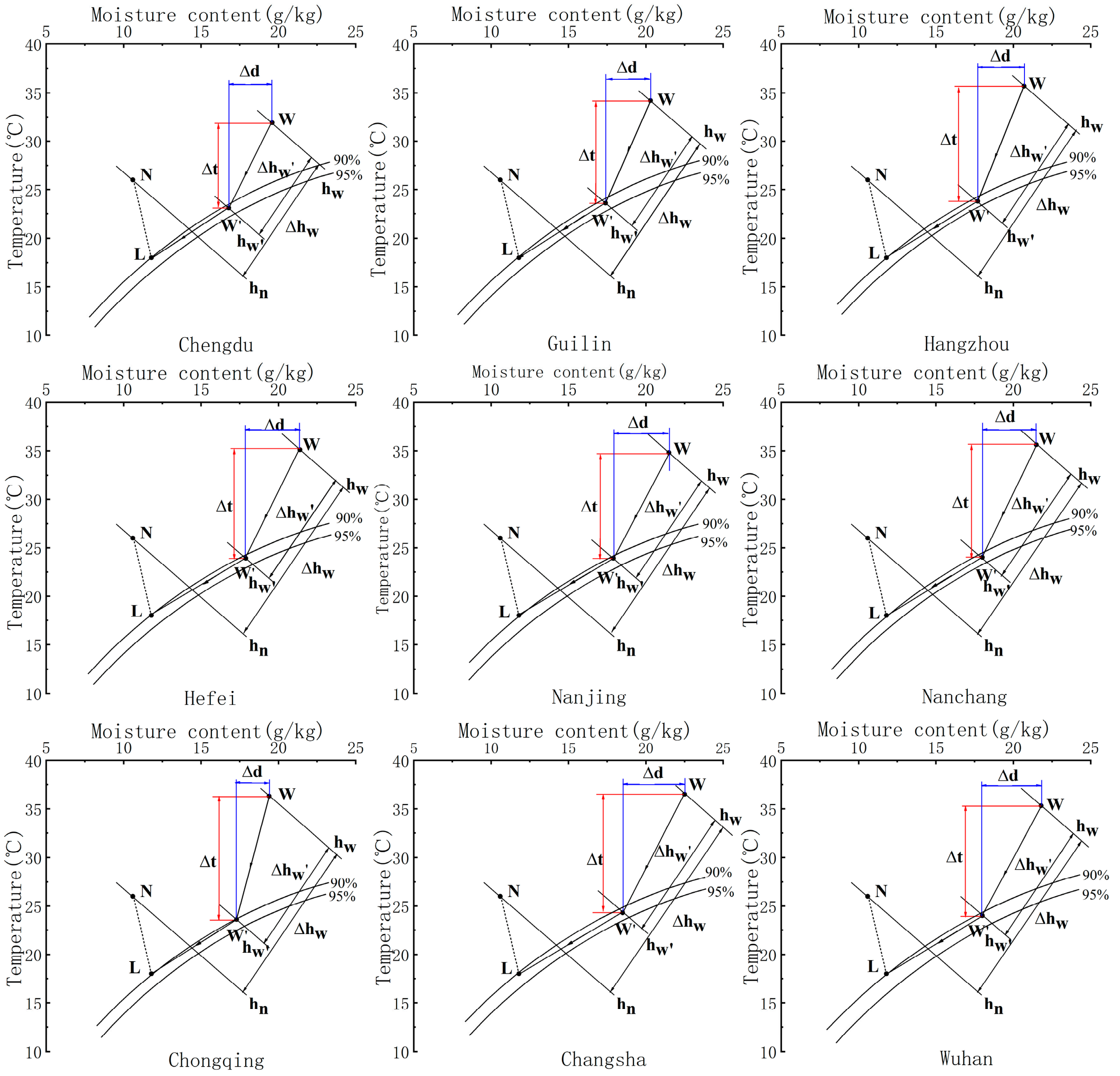

Figure 7. Psychrometric chart of fresh air treated by ERIEC in hot summer and cold winter regions.

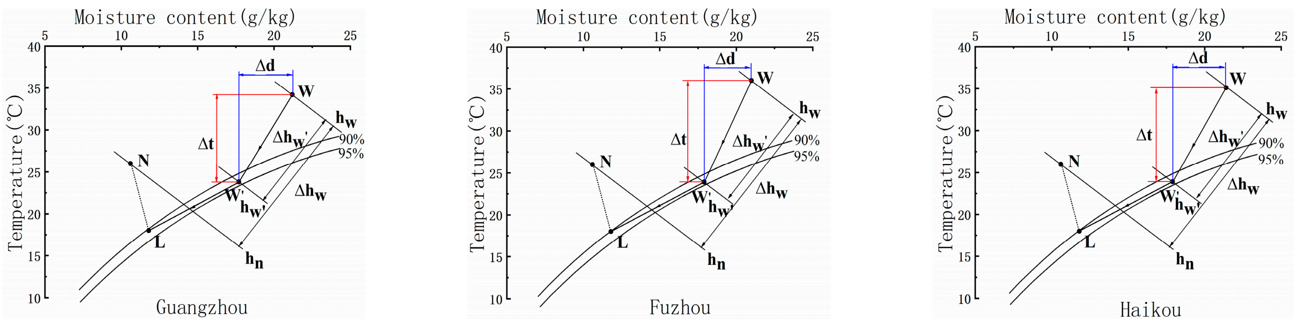

Figure 8. Psychrometric chart of fresh air treated by ERIEC in hot summer and warm winter regions.

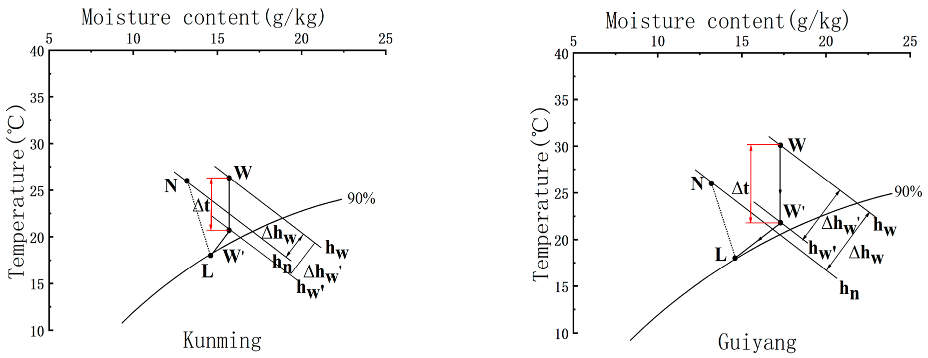

Figure 9. Psychrometric chart of fresh air treated by ERIEC in mild regions.

In Figures 5–9, W is the state point of outdoor fresh air, W′ is the state point of fresh air treated by ERIEC unit, L is the state point of fresh air treated by the ERIEC unit and spray chamber or surface cooler, and N is the state point of indoor air. $\Delta h_{W'}$ is the difference in enthalpy of fresh air before and after treatment by the ERIEC unit. Δh_W is the difference in enthalpy between the fresh outdoor air and the indoor air. The magnitude of the ratio between $\Delta h_{W'}$ and Δh_W reflects the size of the fresh air load borne by the fresh air unit to the total load. As shown in Figure 5, in severe cold regions, because of the lower outdoor wet bulb temperature in summer compared to other regions, ERIEC mainly bears the sensible heat load of fresh air. As shown in Figures 6–8, in cold regions, hot summer and cold winter regions, and hot summer and warm winter regions, as the moisture content of fresh outdoor air gradually increases in summer, condensation occurs on the primary side and the moisture content of fresh air treated by ERIEC units is reduced more often. This will lead to an increase in the fresh air latent heat load borne by the ERIEC unit, but the cooling capacity of EERIEC does not vary much, and is in the range of 55% to 57%. As shown in Figure 9, the fresh air load in the mild region is relatively small, and there is no reduction in moisture content after the ERIEC treatment.

5. Economic Analysis of ERIEC Design and Selection

The theoretical design process of ERIEC is complex, and to facilitate faster design by designers according to the samples of evaporative cooling fresh air units provided by various manufacturers, evaporative cooling heat exchangers can be divided into eight models, while corresponding air volumes are, respectively, 1000 m^3/h, 2000 m^3/h, 3000 m^3/h, 4000 m^3/h, 5000 m^3/h, 6000 m^3/h, 8000 m^3/h and 10,000 m^3/h. Based on the recommended air velocity of 2–3 m/s, the heat transfer area is calculated. However, the size of the heat transfer area will have a large impact on the economy of the system, thus it is necessary to analyze in terms of economy and recommend the heat transfer area and size selection of the ERIEC heat exchanger.

The paper defines an economic indicator *EI*, which means the economic benefit generated by using the ERIEC with the recommended heat transfer area and plate type at common air volumes. The positive parameter E_{ERIEC} is the mechanical refrigeration electricity cost saved by ERIEC; the negative parameters are the pump electricity cost E_{pump}, the ERIEC water consumption cost E_{water}, and the initial investment in heat exchanger area E_{pl}.

$$EI = E_{ERIEC} - E_{water} - E_{pump} - E_{pl} \tag{25}$$

The outdoor medium humidity point ***a*** and high humidity point ***b*** were chosen. The temperatures are both 33 °C. The humidities are, respectively, 50% and 90%. The secondary air inlet temperature and humidity is taken as 26 °C and 50%. the above model is used to calculate the economic value of the variables when using the two points ***a*** and ***b*** as the inlet state points, respectively.

5.1. Power Consumption of ERIEC Fans

Fan energy consumption includes primary and secondary side fan energy consumption, as calculated in Equation (26) [27]. In the formula, ma is the air mass flow rate, ΔP is the pressure loss, η_0 is the internal efficiency (range 0.7 to 0.8), η_1 is the mechanical efficiency (range is 0.85 to 0.95), K is the kinetic energy coefficient, f_{Re} is the friction loss factor, de is the equivalent diameter, Re is Reynolds number, and α is the absolute roughness of the wall surface.

$$P_{fan} = \frac{m_{\mathrm{a}} \cdot \Delta P}{3600 \cdot 1000 \cdot \eta_0 \cdot \eta_1} \cdot K \tag{26}$$

$$\Delta P = 2 f_{\mathrm{Re}} \rho u^2 \cdot \frac{L}{d_e} \tag{27}$$

$$(f_{\mathrm{Re}})^{-\frac{1}{2}} = 1.74 - 2\log\left(\frac{2\alpha}{d_e} + \frac{18.7}{\mathrm{Re}\sqrt{f_{\mathrm{Re}}}}\right) \tag{28}$$

$$d_e = 2 \cdot \theta \cdot H/(\theta + H) \tag{29}$$

$$R_e = \mu \cdot d_e / v \tag{30}$$

5.2. Power Consumption of ERIEC Pumps and ERIEC Water Consumption

The energy consumption of the spraying pump can be estimated by Equation (31), where m_w is the spraying water volume, H_{total} is total pressure loss of water pump, and K is the kinetic energy coefficient (range is 1.05 to 1.1).

$$P_{pump} = m_w \cdot g \cdot H_{total} \cdot K \tag{31}$$

Total pressure loss H_{total} includes: nozzle pressure loss H_{nozzle}, gravity pressure head loss $H_{gravite}$, and valve pressure loss H_{valve}; units are m, such as in Formula (32).

$$H_{total} = H_{nozzle} + H_{gravity} + H_{valve} \tag{32}$$

In the indirect evaporative cooling process, the secondary side of the heat transfer wall spray water will enter the secondary side of the air through the evaporation process and eventually be discharged and consumed; the part of the spray water consumption is called the system theoretical minimum water consumption, and can be calculated using the Formula (33).

$$w_{mc} = \frac{3600 \cdot m_2 (d_{2,out} - d_{2,in})}{\rho_2} \cdot 1000 \tag{33}$$

In order to prevent salt concentration in the water from forming scale, continuous or periodic drainage is required to maintain the salt concentration from exceeding the limit by replenishing a certain amount of new water. Therefore, the water consumption of indirect evaporative cooling is the sum of the minimum water consumption and the drainage water consumption [28].

$$w_{total} = w_{mc} + w_d = (1 + \frac{1}{R-1}) w_{mc} \tag{34}$$

5.3. Power Consumption Saved by ERIEC

Using the numerical model in Section 2 to take two points of medium humidity point a and high humidity point b to calculate the cooling capacity of the ERIEC, the difference in enthalpy between the primary side inlet and outlet at different air volumes is the reduced cooling capacity for the mechanical refrigeration system. In terms of cooling capacity, the electric cooling source integrated cooling performance coefficient SCOP should be not less than 3.6 for air-conditioning systems in cold regions, hot summer and east-cooling regions and hot summer and warm winter regions [29], and the power consumption per unit time is calculated from Equation (35), which is the power consumption saved by ERIEC.

$$P_{ERIEC} = \frac{m_a(i_{1,in} - i_{1,out})}{SCOP \cdot 3.6 \cdot 10^6} \tag{35}$$

The life span of a central air conditioning system is 15 to 20 years [30]. Assuming a lifetime of 20 years for the evaporative cooled air conditioning fresh air unit, the ERIEC is assumed to operate 13 h per day from June 1 to September 30 in each year, and the ERIEC is also assumed to operate at a fixed cooling capacity. Therefore, the expected operating time of the system is 31,200 h, and thus the total electricity consumption saved by ERIEC is calculated.

5.4. Economic Calculation of ERIEC Recommended Selection

The area of the aluminum plate used in the plate evaporative cooling heat exchanger is shown in Equation (36).

$$A_{pl} = n \cdot k_{pl} \cdot H \cdot L \tag{36}$$

where k_{pl} is the value of the area increase due to the special structure of the aluminum plate, such as corrugation, etc.

The price of 0.35 mm thick aluminum plate q_{pl} is 18.78 RMB/m^2, the electricity cost q_{elec} is taken as 0.55 RMB/kWh, and the water cost q_{water} is taken as 5.55 RMB/m^3. Both the pump electricity cost, pump water cost, the price of aluminum plate and the air conditioning electricity cost saved are calculated according to Equations (37)–(40).

$$E_{ERIEC} = P_{ERIEC} \cdot \tau_{total} \cdot q_{elec} \tag{37}$$

$$E_{water} = w_{total} \cdot \tau_{total} \cdot q_{water} \tag{38}$$

$$E_{pump} = P_{pump} \cdot \tau_{total} \cdot q_{elec} \tag{39}$$

$$E_{pl} = A_{pl} \cdot q_{pl} \tag{40}$$

The economic indicators under the heat exchange area of different air volumes are calculated according to Equation (25), as shown in Table 4.

Table 4. The final economic indicators.

Air volume (m^3/h)		1000	2000	3000	4000	5000	6000	8000	10,000
Recommended heat exchange area (m^2)		14.00	33.48	59.29	77.91	111.36	133.76	223.00	279.00
Recommended board type (mm)		500 × 500	600 × 600	700 × 700	700 × 700	800 × 800	800 × 800	1000 × 1000	1000 × 1000
E_{pl} (RMB)		262.92	628.75	1113.47	1463.15	2091.34	2512.01	4187.94	5239.62
P_{pump} (W)		59	118	179	235	294	354	472	590
E_{pump} (RMB)		1012.44	2024.88	3071.64	4032.60	5045.04	6074.64	8099.52	10,124.40
w_{total}	*a*	6.8	14.2	22.6	30.0	39.2	47.0	66.8	83.6
(L/h)	*b*	12.5	25.7	40.4	53.7	69.6	83.5	117.0	146.0
E_{water}	*a*	1177.49	2458.87	3913.42	5194.80	6787.87	8138.52	11,567.09	14,476.18
(RMB)	*b*	2164.50	4450.21	6995.66	9298.69	12,051.94	14,458.86	20,259.72	25,281.36
P_{ERIEC}	*a*	2.24	4.63	7.25	9.63	12.50	15.00	21.04	26.30
(KW)	*b*	3.19	6.59	10.33	13.74	17.78	21.33	29.93	37.41
S_{ERIEC}	*a*	3.84	7.94	12.44	16.52	21.45	25.74	36.10	45.12
(10^4 RMB)	*b*	5.48	11.31	17.73	23.58	30.51	36.61	51.35	64.19
EI	*a*	3.59	7.43	11.63	15.45	20.06	24.07	33.71	42.13
(10^4 RMB)	*b*	5.14	10.60	16.61	22.10	28.59	34.31	48.10	60.13

6. Conclusions

1. Based on the ERIEC heat and mass transfer analytical solution model and numerical solution procedure developed by our research team, the cooling capacity of ERIEC fresh air units of thirty typical cities in five climate zones of China: severe cold, cold, mild, hot summer and cold winter, and hot summer and warm winter were calculated to obtain the fresh air outlet state parameters, wet bulb efficiency, enthalpy efficiency

and the fresh air load borne under the summer outdoor design conditions. In severe cold regions, cold regions, hot summer and cold winter regions, and hot summer and warm winter regions, as the moisture content of fresh outdoor air will gradually increase in summer, condensation occurs on the primary side and the moisture content of fresh air treated by ERIEC units is reduced more often. This will lead to an increase in the fresh air latent heat load borne by the ERIEC unit.

2. The results of numerical calculations applying ERIEC to 30 cities show that the wet bulb efficiency of each city is 0.67–0.98, which increases as the outdoor design wet bulb temperature decreases; the enthalpy efficiency is 0.76–1.29, which increases as the outdoor design wet bulb temperature increases; and the ERIEC is able to bear between 53% and 100% of the fresh air load; as such, the matching surface cooler only needs to bear a small amount of residual cold load.
3. In this paper, the design parameters of the ERIEC air conditioning fresh air unit structure, primary and secondary air velocity, heat exchange area, water consumption of spray water, plate type and other design parameters are given by calculation for eight models of a fresh air volume of 1000–10,000 m^3/h commonly used in engineering, and their economic analysis is carried out under the recommended parameters.
4. In addition to the structure design and size, the performance of the fresh air system using indirect evaporative cooling would be affected by the outdoor dry bulb temperature and the moisture content. In the paper, the analysis and discussion are for the regions of the outdoor dry bulb temperature from 24.3 °C to 36 °C, and the moisture content from 5.4 g/kg to 22.5 g/kg. As such, when the system is applied in the same or similar weather conditions, the discussion results are applicable. However, further research is needed for the selection design in areas where the weather parameters differ greatly from those studied.

Author Contributions: Conceptualization, C.G. and Y.L.; methodology, C.G.; software, Y.L.; validation, X.L., C.G. and R.B.; formal analysis, Y.L.; resources, C.G.; data curation, Y.L. and C.D.; writing—original draft preparation, Y.L.; writing—review and editing, C.G. and X.L.; supervision, C.G. and C.D.; project administration, C.G. and C.D.; funding acquisition, C.G. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the National Natural Science Foundation of China (No. 51678385).

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Not applicable.

Acknowledgments: The work described in this paper was supported by the School of Energy and Safety Engineering, Tianjin Chengjian University.

Conflicts of Interest: The authors declare no conflict of interest. The funders had no role in the design of the study; in the collection, analyses, or interpretation of data; in the writing of the manuscript; or in the decision to publish the results.

Abbreviations

A	heat transfer area (m^2)
H	cooler height (m)
L	cooler length (m)
C_c^*	heat capacity ratio of condensation water to secondary air
C_w^*	heat capacity ratio of spray water to secondary air
Le_f	Lewis factor
Pr	Prandtl number
Re	Reynolds number

$d_{1,w}$	the moisture content of air in equilibrium with condensation water (kg/kg)
$d_{2,w}$	moisture content of air in equilibrium with spray water (kg/kg)
x,y	coordinate direction
$\hat{x}$,$\hat{y}$	dimensionless coordinate direction
m	mass flow rate of air (kg/s)
m_w	spray water volume (kg/s)
i	enthalpy of air (kJ/kg)
h	heat transfer coefficient (kJ/(m^2·°C))
h_m	mass transfer coefficient (kg/(m^2·s))
r	latent heat of water evaporation (kJ/kg)
T	thermodynamic temperature (K)
t	dry bulb temperature (°C)
d	moisture content of air (kg/kg)
C_p	Specific heat at constant pressure kJ/(kg·K)
n	number of heat exchange channels
NTU	number of heat transfer units
E	fees (RMB)
R	circulating water concentration ratio
P	energy consumption (W)
Le	Lewis number
h_n	Enthalpy of indoor state point (kJ/kg)
h_w	Enthalpy of outdoor fresh air state point (kJ/kg)
$h_{w'}$	Enthalpy of the state point of fresh air treated by ERIEC unit (kJ/kg)
w_d	discharge water consumption (m^3/h)
τ_{total}	the total operating time of the ERIEC fresh air unit (h)
Greek symbols	
λ	thermal conductivity, W/m·°C
σ	condensation area ratio
Γ	Spray water mass per unit length kg/(m·s)
η	efficiency
θ	channel spacing (m)
δ	thickness (m)
ρ	density (kg/m^3)
ω	water consumption (m^3/h)
τ	operating time
Δ	difference value
Subscripts	
1	primary/fresh air
2	secondary air
w	water
in	inlet
out	outlet
pl	wall
$total$	total amount
s	saturation vapor pressure
c	condensate water
w_b	wet bulb

References

1. Bruno, F. On-site experimental testing of a novel dew point evaporative cooler. *Energy Build.* **2011**, *43*, 3475–3483. [CrossRef]
2. Moshari, S.; Heidarinejad, G.; Fathipour, A. Numerical investigation of wet-bulb effectiveness and water consumption in one-and two-stage indirect evaporative coolers. *Energy Convers. Manag.* **2016**, *108*, 309–321. [CrossRef]
3. Wang, Y.G.; Pei, X.Y. Experimental study on the cooling performance of fork-flow type dew point indirect evaporative cooler. In Proceedings of the 6th National Conference on Building Environment and Equipment Technology Exchange, Guangzhou, China, 11 December 2015; Volume 213–217.
4. Fan, L.J.; Huang, X. Design and performance analysis of porous vertical tube type indirect evaporative cooler. *Fluid Mach.* **2019**, *47*, 85–88.
5. Hasan, A. Indirect evaporative cooling of air to a sub-wet bulb temperature. *Appl. Therm. Eng.* **2010**, *30*, 2460–2468. [CrossRef]

6. Chen, Y.; Yang, H.; Luo, Y. Experimental study of plate type air cooler performances under four operating modes. *Build. Environ.* **2016**, *104*, 296–310. [CrossRef]
7. You, Y.; Jiang, J.; Guo, C. Application and Analysis of Indirect Evaporative Cooling Air Conditioning Unit in Hot and Humid Areas. *Fluid Mach.* **2019**, *47*, 71–75.
8. Zheng, B.; Guo, C.; Chen, T.; Shi, Q.; Lv, J.; You, Y. Development of an experimental validated model of cross-flow indirect evaporative cooler with condensation. *Appl. Energy* **2019**, *252*, 113438. [CrossRef]
9. You, Y.; Wang, G.; Yang, B.; Guo, C.; Ma, Y.; Cheng, B. Study on heat transfer characteristics of indirect evaporative cooling system based on secondary side hydrophilic. *Energy Build.* **2022**, *257*, 111704. [CrossRef]
10. Chen, Q.; Burhan, M.; Shahzad, M.W.; Ybyraiymkul, D.; Oh, S.; Cui, X.; Ng, K.C. Long-term performance of a hybrid indirect evaporative cooling-mechanical vapor compression cycle: A case study in Saudi Arabia. *Front. Built Environ.* **2022**, *8*, 1032961. [CrossRef]
11. Mao, X.; Huang, X.; Wen, L. Design calculation of porous ceramic tube type dew point indirect evaporative cooler. *J. Xi'an Eng. Univ.* **2009**, *23*, 88–92.
12. She, J.; Huang, X.; Lv, W.; Liu, K. Application solutions of cross dew point indirect evaporative cooling air-conditioning unit to data center. *Refrig. Air-Cond.* **2017**, *17*, 35–39.
13. Wang, X.; Huang, X.; Qiu, J.; Song, X.; Wang, J. Engineering experimental analysis and applicantion exploration of different indirect evaporative cooler. *Refrig. Air-Cond.* **2015**, *15*, 92–97.
14. Wan, Y.; Huang, Z.; Soh, A.; Chua, K.J. On the performance study of a hybrid indirect evaporative cooling and latent-heat thermal energy storage system under commercial operating conditions. *Appl. Therm. Eng.* **2023**, *221*, 119902. [CrossRef]
15. Abdalazeem, A.; Dong, H.; Weifeng, H.; Qile, S.; Junjie, C.; Hong, Z. The influences of the plate shape on the performance of the indirect evaporative cooler based on the CFD approach. *J. Braz. Soc. Mech. Sci. Eng.* **2023**, *45*, 123.
16. Zhao, X.; Liu, S.; Riffat, S.B. Comparative study of heat and mass exchanging materials for indirect evaporative cooling systems. *Build. Environ.* **2008**, *43*, 1902–1911. [CrossRef]
17. Xing, T.; Li, Y.; Liu, X. Optimization study of cross-flow plate type indirect evaporative heat recovery device. *Refrig. Air-Cond.* **2009**, *9*, 36–41.
18. Zang, D.Y. Design and Research of Folded Plate Indirect Evaporative Cooler in Textile Air Conditioning. Master's Thesis, Zhongyuan Institute of Technology, Foshan, China, 2011.
19. Li, F. The Performance of the Curved Plate Type Indirect Evaporative Cooler. Master's Thesis, Zhongyuan Institute of Technology, Foshan, China, 2014.
20. Liu, L.; Zhou, Y.D.; Yang, R.L. Main influencing factors of folded plate indirect evaporative cooler. *Cryog. Supercond.* **2014**, *42*, 65–68. [CrossRef]
21. Huang, X. *Theory and Application of Evaporative Cooling Air Conditioning*; China Construction Industry Press: Beijing, China, 2010.
22. Cui, X.; Chua, K.J.; Islam, M.R.; Ng, K.C. Performance evaluation of an indirect pre-cooling evaporative heat exchanger operating in hot and humid climate. *Energy Convers. Manag.* **2015**, *102*, 140–150. [CrossRef]
23. ASHRAE. American society of heating, refrigerating and air–conditioning engineers, volume SI edition. In *ASHRAE Handbook Fundamentals*; ASHRAE: Peachtree Corners, GA, USA, 2009; Volume 4.18, Chapter 4.
24. Guo, C.; Liu, Q.; Zheng, B.; You, Y.; Li Y. Development of model based on condensation area ratio and effect on heat transfer capacity of indirect evaporative cooling. *Appl. Therm. Eng. Des. Process. Equip. Econ.* **2020**, *164*, 114557. [CrossRef]
25. Liu, Q.; Guo, C.; Wu, Z.; You, Y.; Li, Y. Heat and mass transfer model optimization and annual energy efficiency analysis for energy recovery indirect evaporative cooling. *Build. Simul.* **2022**, *15*, 1353–1365. [CrossRef]
26. Huang, X. *Design Guide for Evaporative Cooling Ventilation and Air Conditioning Systems*; China Construction Industry Press: Beijing, China, 2016.
27. Peng, M. Exergy Efficiency Evaluation and Analysis of Indirect Evaporative Cooling in Plate Heat Exchangers. Master's Thesis, Hunan University, Changsha, China, 2005.
28. Dong, X.; Huang, X.; Lu, W.; Song, X. Derivation and analysis of water consumption calculation formula for evaporative cooling air conditioning unit. *J. Xi'an Polytech. Univ.* **2016**, *30*, 294–299. [CrossRef]
29. *GB 50189-2015*; Energy-Saving Design Standards for Public Buildings. China Construction Industry Press: Beijing, China, 2015.
30. *GB 50736-2012*; Design Code for Heating, Ventilation and Air Conditioning in Civil Buildings. China Construction Industry Press: Beijing, China, 2012.