

Article

3E (Energy–Exergy–Environmental) Performance Analysis and Optimization of Seawater Shower Cooling Tower for Central Air Conditioning Systems

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Abstract: The thermodynamic, exergy and carbon reduction potential of seawater shower cooling towers for central air conditioning systems was investigated under various geometric, physical and environmental conditions. A mathematical model of the shower cooling tower was proposed, and the governing equations considering droplet diameter changes were numerically solved during evaporation. The results showed that compared with the downward-spraying tower, the upward-spraying tower achieved 15.59% higher cooling efficiency, 4.89% higher exergy efficiency and 34.58% higher heat dissipation. Increasing droplet diameter significantly weakened the tower's cooling capacity, with heat dissipation decreasing by 78.11% as the diameter increased from 1 mm to 2.25 mm. The cooling efficiency, thermal efficiency and exergy efficiency demonstrated consistent declining trends with increasing droplet diameter. Under high-salinity conditions (105 g/kg), compared with standard salinity (35 g/kg), the average reduction of cooling efficiency was 2.96%, and exergy efficiency decreased by 2.73%. The increase in air velocity from 2.5 m/s to 4.0 m/s led to a 13.76% improvement in cooling efficiency and a 22.44% increase in heat dissipation, and exergy efficiency decreased by less than 2%. Through multi-objective optimization analysis, the cooling efficiency increased by 14.69% and exergy destruction decreased by 37.87%, demonstrating significant potential for energy conservation and carbon emission reduction in central air conditioning applications.



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Keywords: shower cooling tower; central air conditioning; heat and moisture exchange characteristics; exergy analysis; carbon reduction potential

1. Introduction

The central air conditioning system is one of the major energy consumers in buildings, where cooling towers play a crucial role in heat rejection. Compared with the traditional cooling tower, the shower cooling tower lacks the packing part and improves the water distribution area, providing an innovative solution to solve the problem of packing scaling. This is particularly significant for high-salinity seawater applications with poor water quality, where the shower cooling tower demonstrates superior applicability and better handles high-temperature, high-turbidity circulating water that tends to scale [1]. The enhanced droplet–air interfacial contact ensures efficient heat dissipation under elevated thermal loads—critical features validated by recent advancements in hydrodynamic modeling [2] and fouling-resistant designs [3]. It has emerged as a predominant solution in building cooling systems.

Shower cooling towers were first developed and experimentally evaluated by Givoni [4–6]. The research is reflected in the establishment of a reasonable mathematical model, and the numerical simulation results are used to guide the optimization and improvement of experiments and theoretical analysis.

Qi et al. [7–9] proposed a calculation model for thermal characteristics of counter-flow shower cooling towers, which describes the droplet movement and evaporative cooling processes within the tower. The model validation through actual operation results showed relative errors within 4%, demonstrating its capability to predict tower performance under different operating conditions.

Anbazhagan et al. [10] analyzed the influence of droplet diameter on outlet tower water temperature under different initial droplet velocities and air velocities. Asgharia and Kordani [11] analyzed the effect of droplet diameter on dry cold and dry hot. Cui et al. [12–14] considered three types of shower cooling towers with ST-UD (Spray Tower with Upward spraying and Downward gas flow), ST-DU (Spray Tower with Downward spraying and Upward gas flow) and ST-UU (Spray Tower with Upward spraying and Upward gas flow), which were studied to optimize the thermal characteristics of the tower. In terms of an exergic analysis of the shower cooling tower, relevant literature is still lacking. Qi et al. [15,16] proposed the exergic analysis model of the cooling tower to describe the distribution of exergic parameters in the tower and guide the improvement of the performance of the packed tower.

While prior studies have advanced the understanding of evaporative cooling towers, key gaps persist in seawater applications. Belarbi et al. [3] pioneered thermal modeling of freshwater evaporative towers, yet their framework neglects salinity effects—a limitation addressed in this study by introducing salt concentration-dependent heat/mass transfer correlations. Fisenko et al. [17] analyzed natural draft towers but focused on air–water vapor equilibrium, overlooking droplet dynamics, which our work resolves through time-resolved droplet diameter modeling. Klopper and Kröger [18] advanced numerical methods for packed towers, but their assumptions of uniform water distribution and fixed droplet sizes fail for unpacked shower towers, motivating our dynamic droplet trajectory and shrinkage algorithms. Recent work by Teodori et al. [19] optimized hybrid cooling systems but omitted exergy–environmental trade-offs, a gap filled by our 3E (energy–exergy–environmental) framework. Sureshkumar et al. [20,21] experimentally validated freshwater spray cooling but did not explore hypersaline conditions (>35 g/kg) or upward/downward spray configurations—critical contributions of this study.

Although extensive research [22] has been conducted on cooling tower performance, comprehensive analyses of energy efficiency and environmental impacts of shower cooling towers remain limited, particularly for seawater applications. Understanding these aspects is crucial for optimizing central air conditioning systems and reducing their carbon footprint. Therefore, this study aims to bridge this gap by conducting a detailed investigation of shower cooling tower performance under various operating conditions.

In this paper, based on the Poppe model [23], a detailed thermodynamic performance calculation model of the shower cooling tower is established, the seawater and air outlet parameters are numerically solved, and the experimental results are compared and verified. The heat and mass transfer performance and exergy performance of the seawater tower were evaluated by performance evaluation indicators (cooling efficiency, heat dissipation, gas-phase side thermal efficiency and exergy efficiency). The effects of salinity, droplet diameter and air velocity on performance evaluation indexes were analyzed, which provided a reference for the actual operation of the seawater cooling tower.

2. Physical Model

The primary distinction between shower cooling towers and traditional cooling towers is the removal of the packing section, while other components remain similar. In the shower cooling tower, the atomizing nozzle converts cooling water into small droplets, which directly contact with upward-flowing air for heat and mass exchange. The waste heat carried in the cooling water is transferred to the air, and the air is heated and humidified and discharged from the outlet of the tower. The cooled droplets are collected in the reservoir for recirculation, completing the thermal cycle while maintaining a stabilized water temperature through continuous heat rejection. According to the different directions of the water from the nozzle, it can be divided into the upward-spraying shower cooling tower and the downward-spraying shower cooling tower. The schematic diagram of the two is shown in Figure 1.

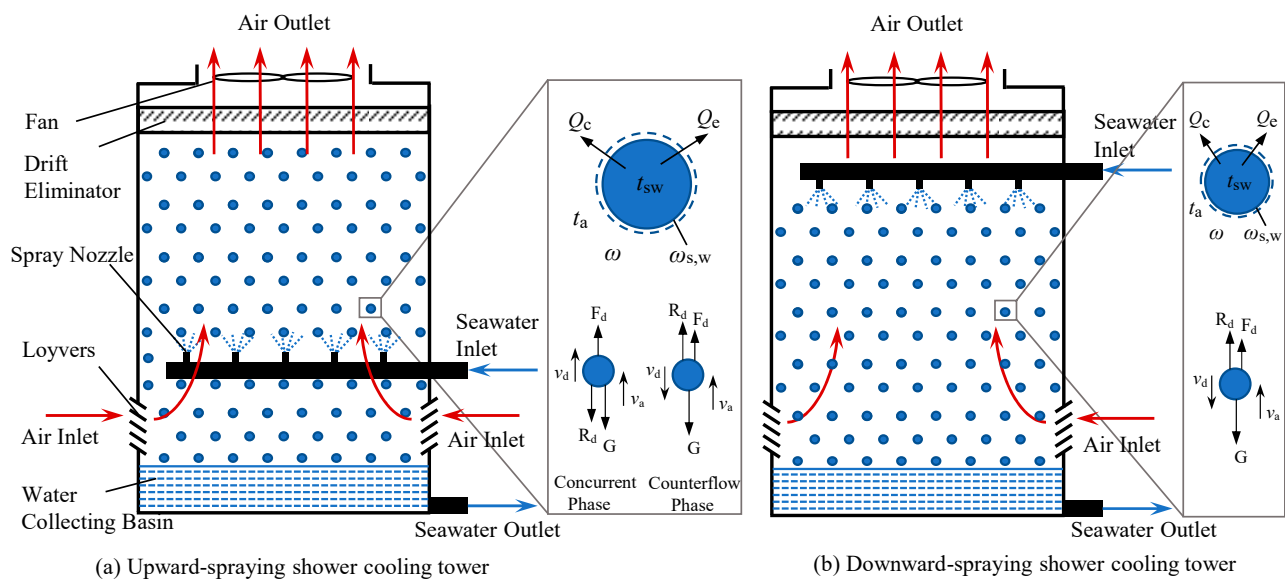


Figure 1. The schematic diagrams of shower cooling towers.

3. Mathematical Model

3.1. Governing Equations

Figure 2 shows the microelement control body of an upward-spraying shower cooling tower; ΔH is the distance between the nozzle and the tower bottom. The overall calculated height, including downstream section H_r and upstream section H_f , is divided into n microelement control bodies with a height of dz from bottom to top along the tower height direction. Based on the conservative law of mass and energy, the control equations for heat and mass transfer are established. The equation forms for the upstream section and downstream section are the same, but different boundary conditions need to be included in the calculation. The established control equations for heat and mass transfer inside the tower are based on the following assumptions [6,7]: (1) the thermal parameters of droplets and air are uniformly distributed on the cross section of the tower, and only change along the height direction of the tower; (2) the radiation heat transfer of the fluid inside the tower and the heat transfer between the fluid inside the tower and the tower wall are not considered; (3) the influence of droplets drifting through the mist eliminator and fans on the heat and mass transfer between water and gas phases is ignored; and (4) the air adjacent to the droplet surface is assumed to be saturated, forming a boundary layer where the humidity ratio corresponds to the droplet temperature.

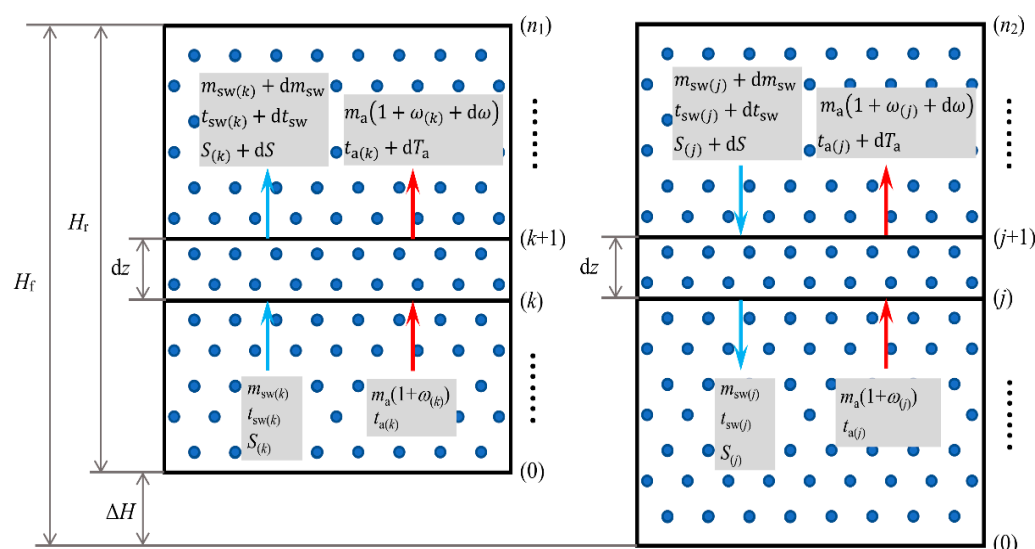


Figure 2. Control volume of upward-spraying shower cooling tower.

The heat and mass transfer processes in the seawater shower cooling tower are governed by the following equations. Key parameters and variables are summarized in Tables 1 and 2.

Table 1. Model parameters and default values.

Parameter	Symbol	Value	Unit	Source/Description
Droplet diameter	d_d	1.0–2.25	mm	Experimental range [5]
Air velocity	v_a	2.5–4.0	m/s	Measured
Seawater salinity	σ	35 (standard)	g/kg	ASTM D1141-98 [24]
Drag coefficient	C_D	0.44 (spherical)	–	Assumed for spherical droplets
Latent heat	h_{fg}	2257	kJ/kg	At 100 °C [16]
Lewis factor	Le	0.865–0.905	–	$Le = hmc_p/\rho_a h_c$

Table 2. Boundary conditions for simulation.

Variable	Symbol	Inlet Value	Outlet Value	Unit
Seawater temperature	$t_{sw,in}$	44	$t_{sw,out}$	°C
Air dry-bulb temperature	$t_{a,in}$	28	$t_{a,out}$	°C
Air wet-bulb temperature	$t_{wb,in}$	21.7	–	°C
Seawater flow rate	q	1000	–	m ³ /h
Air flow rate	G	878,000	–	m ³ /h

(1) Droplet Motion Equation

The vertical motion of droplets is described by Newton's second law. The net force acting on the object is the sum of drag force, buoyancy, and gravitational force, which is expressed as

$$m_d \frac{dv_d}{dt} = \frac{1}{2} C_D \rho_a A_d |v_a - v_d| (v_a - v_d) + m_d g \left(1 - \frac{\rho_a}{\rho_w} \right) \quad (1)$$

where m_d is mass of droplet, kg; v_d is droplet velocity, m/s; t is the time, s; v_a is air velocity, m/s; C_D is the drag coefficient; A_d is the droplet cross-sectional area, m²; ρ_w is the density of water, kg/m³; and ρ_a is the density of air, kg/m³.

(2) Heat Transfer Equations

Heat exchange between droplets and air includes convection and evaporation, which is expressed as

$$Q_{\text{conv}} = h_c A_d (t_d - t_a) \text{ (Convection)} \quad (2)$$

$$Q_{\text{evap}} = h_m A_d (w_s - w_a) h_{fg} \text{ (Evaporation)} \quad (3)$$

$$Q_{\text{total}} = Q_{\text{conv}} + Q_{\text{evap}} \text{ (Total Heat Dissipation)} \quad (4)$$

where Q_{conv} is convective heat transfer rate, W; h_c is convective heat transfer coefficient, W/(m²·°C); Q_{evap} is evaporative heat transfer rate, W; h_m is mass transfer coefficient, m/s; Q_{total} is total heat transfer rate, W; h_{fg} is latent heat of vaporization of water, kJ/kg; w_s is saturated water vapor concentration, kg/m³; and w_a is ambient air water vapor concentration, kg/m³.

(3) Mass and Energy Balances

The mass balance expression for the liquid droplet is as follows:

$$dm_d/dt = -h_m A_d (w_s - w_a) \text{ (Droplet Mass Loss)} \quad (5)$$

The energy balance expression for the air is as follows:

$$m_a dh_a/dt = Q_{\text{total}} \text{ (Air Enthalpy Change)} \quad (6)$$

where m_a is mass of air, kg; and h_a is air specific enthalpy, kJ/kg.

The temperature change of seawater droplets is expressed as

$$m_w c_{p,w} dt_d/dt = -Q_{\text{total}} \text{ (Droplet Temperature Change)} \quad (7)$$

where m_w is mass of seawater, kg; $c_{p,w}$ is specific heat of seawater, kJ/(kg·°C); and t_d is droplet temperature, °C.

The rising height of droplets in the spray tower is shown in Equation (8). Among them, the upper and lower limits of the integration in downstream stage 1 are v_a and v_{dr0} , and the upper and lower limits of the integration in downstream stage 2 are 0 and v_a .

$$H_r = \int_{v_{dr0}}^{v_a} \rho_{sw} v_{dr} \left[(\rho_{sw} - \rho_a)g + \frac{3C_d \rho_a (v_{dr} - v_a)^2}{4d_d} \right]^{-1} dv_{dr} + \int_{v_a}^0 \rho_{sw} v_{dr} \left[(\rho_{sw} - \rho_a)g - \frac{3C_d \rho_a (v_{dr} - v_a)^2}{4d_d} \right]^{-1} dv_{dr} \quad (8)$$

where m_d is the mass of the droplet, kg; g is the acceleration of gravity, 9.8 m/s²; ρ_{sw} is the density of seawater, kg/m³; ρ_a is the density of air, kg/m³; V_d is the volume of the droplet, m³; C_d is the drag coefficient; and d_d is the diameter of the droplet, mm.

The moisture content difference between the saturated air boundary layer on the droplet surface and the external air is the driving force of evaporative heat dissipation, as shown in Equations (9) and (10).

$$q_c = k_c (t_{sw} - t_a) A_d \quad (9)$$

$$q_e = h_v k_d (\omega_{s,w} - \omega) A_d \quad (10)$$

where q_c and q_e are the heat dissipated by the droplets to the surrounding air through convection and evaporation; k_c is the heat transfer coefficient; k_d is the mass transfer coefficient; A_d is the heat exchange area of a single droplet in contact with air; h_v is the specific enthalpy of seawater vapor; t_{sw} and t_a are the droplet temperature and ambient air temperature, respectively; and $\omega_{s,w}$ and ω are the moisture content of the saturated air

on the surface of the droplet and the moisture content of the surrounding air, respectively, kg_v/kg_a .

The heat transfer coefficient can be calculated by the Nusselt number [7], which is expressed as

$$k_c = \frac{Nu\lambda_a}{d_d} \quad (11)$$

$$Nu = 2 + 0.6Re^{1/2}Pr^{1/3} \quad (12)$$

where the Prandtl number $Pr = c_{p,a}\mu_a/\lambda_a$; and λ_a is the thermal conductivity of air, $\text{kW}/(\text{m}\cdot^\circ\text{C})$, which can be estimated by the fitting correlation $\lambda_a = 2.44 \times 10^{-5} + 7.62 \times 10^{-8}t_a$, and the error is less than 0.5%, in the range of $0\text{--}70^\circ\text{C}$ [25].

The mass transfer coefficient can be calculated by the Sherwood number [11], which is expressed as

$$Sh = 2 + 0.6Re^{1/2}Sc^{1/3} \quad (13)$$

where the Schmit number $Sc = \mu_a/\rho_a D$; and D is the diffusion coefficient of water vapor in air, m^2/s ; with $p_0 = 0.1\text{ MPa}$, $T_0 = 298\text{ K}$, and $D_0 = 2.5 \times 10^{-5}\text{ m}^2/\text{s}$, the diffusion coefficient in other air states (T_a, p_a) can be calculated by the Equation $D = D_0(T_a/T_0)^{1.5}(p_0/p_a)$ estimate [25].

The Lewis factor is as follows:

$$Le_f = \frac{k_c}{k_d c_{p,ma}} \quad (14)$$

where $c_{p,ma} = c_{p,a} + \omega c_{p,v}$ is the specific heat of humid air at constant pressure, $\text{kJ}/(\text{kg}\cdot^\circ\text{C})$.

The variation of droplet diameter when the droplet moves in the vertical direction is as follows:

$$\frac{dd_d}{dz} = -\frac{2k_d(\omega_{s,w} - \omega)}{\rho_{sw}v_d} \quad (15)$$

During time dt , the number of droplets passing through the micro-element control body is shown in Equation (16), and the total heat exchange area when the droplets in the body are in direct contact with air is controlled, as shown in Equation (17).

$$N_d = \frac{m_{sw}dt}{m_d} = \frac{m_{sw}dz}{m_d v_d} \quad (16)$$

$$dA = N_d A_d = \frac{m_{sw}\pi d_d^2}{m_d v_d} dz = \frac{6m_{sw}}{\rho_{sw}v_d d_d} dz \quad (17)$$

For the micro-element control body, the mass is conserved, which controls the evaporation and mass transfer of all droplets in the body to the air in contact with it, resulting in an increase in the moisture content of the air:

$$dm_{sw} = k_d(\omega_{s,w} - \omega)dA = m_a d\omega \quad (18)$$

For the micro-element control body, energy is conserved, and all droplets in the body are controlled to transfer heat (the sum of convection heat dissipation and evaporative heat dissipation) to the air in contact with it, resulting in an increase in the specific enthalpy of the air:

$$m_a dh_{ma} = m_{sw}c_{p,sw}dt_{sw} + c_{p,sw}t_{sw}dm_{sw} = N_d(q_c + q_e) \quad (19)$$

where h_{ma} is the specific enthalpy of wet air, kJ/kg ; m_a and m_{sw} are the mass flow rates of air and seawater, respectively, kg/s ; and $c_{p,sw}$ is the specific heat of seawater.

Substituting Equations (9), (10), and (16)–(18) into Equation (19), we can get

$$\begin{aligned} \frac{dh_{ma}}{dz} &= \frac{6m_{sw}}{m_a\rho_{sw}v_d d_d} [k_c(t_{sw} - t_a) + h_v k_d(\omega_{s,w} - \omega)] \\ &= \frac{6m_{sw}k_d}{m_a\rho_{sw}v_d d_d} [Le_f(h_{mas,w} - h_{ma}) + (1 - Le_f)(\omega_{s,w} - \omega)h_v] \end{aligned} \quad (20)$$

where $h_{mas,w}$ is the specific enthalpy of saturated humid air at seawater temperature.

From Equations (17) and (18), we can get

$$\frac{dm_{sw}}{dz} = \frac{6m_{sw}k_d}{\rho_{sw}v_d d_d} (\omega_{s,w} - \omega) \quad (21)$$

$$\frac{d\omega}{dz} = \frac{6m_{sw}k_d}{m_a\rho_{sw}v_d d_d} (\omega_{s,w} - \omega) \quad (22)$$

From Equations (18)–(19), we can get

$$\frac{dt_{sw}}{dz} = \frac{m_a}{m_{sw}} \left[\frac{1}{c_{p,sw}} \frac{dh_{ma}}{dz} - t_{sw} \frac{d\omega}{dz} \right] \quad (23)$$

The differential equation of the salinity change in seawater droplets in the shower cooling tower is

$$dS = -\frac{Sm_a}{m_{sw}} d\omega, \quad \frac{dS}{dz} = -\frac{6Sk_d}{\rho_{sw}v_d d_d} (\omega_{s,w} - \omega) \quad (24)$$

The above is the control equation of water–gas two-phase heat and mass transfer in the unpacked seawater tower, which reflects the changes in seawater droplet diameter, salinity, air moisture content and air specific enthalpy along the tower height direction in the tower.

3.2. Exergy Calculation

The exergy values of seawater and air are calculated as shown in Equations (19) and (20), by substituting the relevant parameters of the seawater droplets and air at the entrance and exit of the upward-spraying shower cooling tower and the downward-spraying shower cooling tower, respectively. The seawater/air inlet exergy and seawater/air outlet exergy of the seawater shower cooling tower are obtained, and then the total exergy loss and exergy efficiency are calculated.

$$\begin{aligned} E_{air} &= m_a \left\{ (c_{p,a} - \omega c_{p,v}) \left(T_a - T_0 - T_0 \ln \frac{T_a}{T_0} \right) \right. \\ &\quad \left. + R_a T_0 \left[(1 + 1.608\omega) \ln \frac{1+1.608\omega_0}{1+1.608\omega} + 1.608\omega \ln \frac{\omega}{\omega_0} \right] \right\} \end{aligned} \quad (25)$$

$$E_{sw} = m_{sw} \left[c_{p,sw} \left(T_{sw} - T_0 - T_0 \ln \frac{T_{sw}}{T_0} \right) - R_v T_0 \ln RH_0 \right] \quad (26)$$

The basis of exergy analysis is to establish the exergy balance equation. Since the system does not do external work in the process of water–gas two-phase heat and mass transfer, the process is considered to be adiabatic. The energy balance is referred to as energy flowing into the system = energy flowing out of the system. The exergy balance can be expressed as “total exergy flowing into seawater tower = total exergy flowing out of seawater tower + total exergy loss” [25,26]:

$$E_{a,in} + E_{sw,in} = E_{a,out} + E_{sw,out} + E_D \quad (27)$$

4. Cooling Tower Performance Evaluation Index

4.1. Cooling Efficiency

Cooling efficiency describes the ratio of the actual cooling capacity of the seawater tower to the theoretical maximum cooling capacity [23,25]. Cooling efficiency is a function of cooling water temperature difference and cooling amplitude.

$$\eta_{\text{cooling}} = \frac{t_{\text{sw,in}} - t_{\text{sw,out}}}{t_{\text{sw,in}} - t_{\text{wb}}} = \frac{1}{1 + \frac{t_{\text{sw,out}} - t_{\text{wb}}}{t_{\text{sw,in}} - t_{\text{sw,out}}}} \quad (28)$$

4.2. Thermal Efficiency

The cooling efficiency refers to the flow process of the water in the tower, while the thermal efficiency of the gas phase side describes the process of warming and humidifying the humid air in the tower, which is defined as the ratio of the actual enthalpy change of the humid air to the maximum possible enthalpy change [23,25,27]; the ideal situation where the maximum possible enthalpy change occurs in a cooling tower is when the humid air at the tower outlet is saturated at the inlet water temperature.

$$\eta_{\text{air}} = \frac{h_{\text{ma,out}} - h_{\text{ma,in}}}{h_{\text{mas,win}} - h_{\text{ma,in}}} \quad (29)$$

4.3. Exergy Efficiency

Cooling efficiency and gas-phase thermal efficiency quantify heat rejection performance from energy conservation principles, whereas exergy efficiency evaluates thermodynamic irreversibilities, providing complementary insights into system optimization. The main function of the seawater tower is to transfer the energy of the hot fluid (seawater) to the cold fluid (wet air). The ratio of changes in air exergy to changes in seawater exergy reflects the degree of exergy transmission.

$$\eta_{\text{exergy}} = \frac{E_{\text{ma,out}} - E_{\text{ma,in}}}{E_{\text{sw,in}} - E_{\text{sw,out}}} = 1 - \frac{E_D}{E_{\text{sw,in}} - E_{\text{sw,out}}} \quad (30)$$

4.4. Heat Dissipation

The total heat dissipation from the hot fluid (seawater) to the cold fluid (humid air) in the cooling tower is shown in Equation (27) [28,29]. If the changes in water volume and physical parameters in the water cooling process are ignored, the heat dissipation is numerically equal to the product of the circulating water volume and the cooling water temperature difference and is also equal to the product of the air flow and the air enthalpy change.

$$Q = c_{\text{psw,in}} m_{\text{sw,in}} t_{\text{sw,in}} - c_{\text{psw,out}} m_{\text{sw,out}} t_{\text{sw,out}} = m_{\text{a}} (h_{\text{ma,out}} - h_{\text{ma,in}}) \quad (31)$$

5. Solution and Verification of the Mathematical Model

5.1. Solution Methodology

Based on the finite difference method, the calculation model in differential form is discretized into an algebraic equation, and the numerical method of iterative calculation is used to solve it from the bottom.

Considering the computational cost comprehensively, take the iteration step as m . For the top-spray tower, the numerical solution is carried out in two parts: the specific solution process is shown in Figure 3, and the solution of the bottom-spray tower is similar to the top-spray counter-flow section.

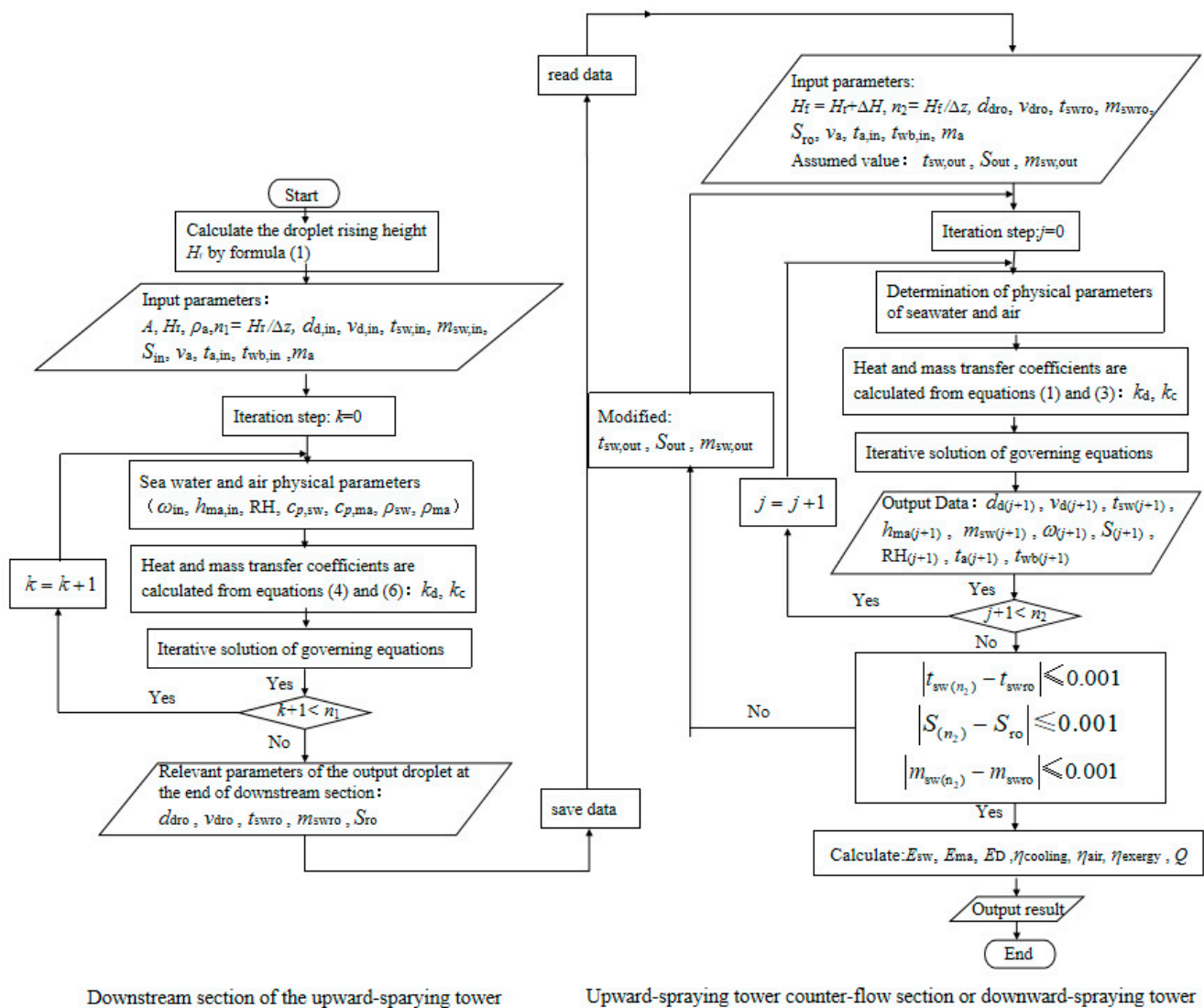


Figure 3. Flow chart of numerical solution for upward-spraying tower/downward-spraying tower.

5.2. Stability and Accuracy Analysis for the Model

The numerical stability and accuracy of the steady-state model were rigorously evaluated to ensure reliable predictions. Convergence was achieved when normalized residuals of seawater temperature (t_{sw}), air temperature (t_a) and humidity ratio (wa) fell below a tolerance of $\epsilon = 10^{-4}$, balancing computational efficiency and precision. Under-relaxation factors $\alpha = 0.3$ – 0.6 for thermal variables and $\alpha = 0.1$ – 0.2 for droplet diameter were applied to stabilize iterative updates and mitigate oscillations in strongly nonlinear systems. A grid independence study confirmed spatial discretization adequacy, with $N = 200$ grids yielding a seawater outlet temperature error of $<0.05\%$ compared to $N = 500$ while maintaining computational efficiency. Residual histories exhibited monotonic decay, achieving convergence within 50–100 iterations for typical cases, with no dependence on initial guesses due to the elliptic nature of the steady-state equations. Model accuracy was validated through energy and mass balance checks, showing errors $< 2\%$ and $< 1\%$, respectively. For robust simulations, $N \geq 200$ grids, $\epsilon = 10^{-4}$ and $\alpha = 0.4$ are recommended as defaults, complemented by the post-solution validation of conservation laws. This approach ensures reliable performance predictions for engineering applications while adhering to computational best practices.

5.3. Mathematical Model Validation

Experimental data on cooling towers reported in the existing literature depart from the type of tower treated here. So, the use of practical full-scale towers is preferred. The experiment was developed in a shower cooling tower in Jiangsu Seagull Cooling Tower Co., Ltd., Changzhou city, China. A schematic diagram of the test rig is shown in Figure 4. The picture of the test shower cooling tower is shown in Figure 5.

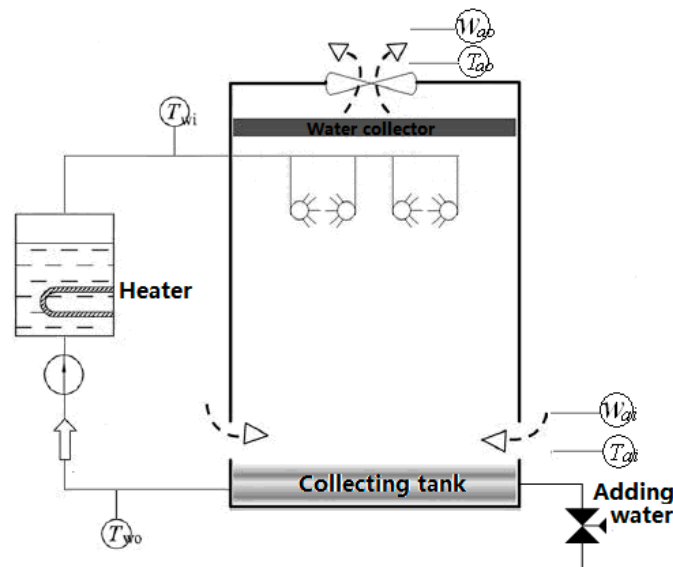


Figure 4. Schematic diagram of the test shower cooling tower.



Figure 5. Pictorial diagram of the test shower cooling tower.

The experimental data were collected under steady-state conditions. All parameters were measured simultaneously when the system reached stability. Measurements were repeated five times at regular intervals. Then, the average data were considered to be the valuable data. The measurements were considered valid when the thermal balance error between air and water remained below 5%. Then, the original experiment data were recorded. The experimental setup utilized a shower cooling tower with the following key dimensions and design parameters shown in Table 3.

Table 3. Geometric specifications of the experimental shower cooling tower.

Parameter	Symbol	Value	Unit
Tower height (nozzle to basin)	H	3.2	m
Cross-sectional area	A	1.8×1.8	m ²
Nozzle arrangement	N	4×4 grid	–
Nozzle spacing	S	0.45	m
Adjustable spray height	H_{spray}	1.5–2.5	m
Basin volume	V_{basin}	0.8	m ³
Compliance standard	–	ASHRAE 133-2021 [30]	–

The shower cooling tower was tested under various air intake conditions. The measurement parameters and their locations are detailed in Table 4. Ten sets of experimental conditions were selected, as shown in Table 5.

Table 4. Measurement parameters and locations.

Measurement Point	Measured Parameters	Instrument	Accuracy
Air Inlet	Dry-bulb temperature ($t_{a,in}$)	Vane anemometer	± 0.5 °C
	Wet-bulb temperature ($t_{wb,in}$)	Psychrometer	$\pm 3\%$ FS
	Air velocity (va)		
Air Outlet	Dry-bulb temperature ($t_{a,out}$)	Hygrometer (capacitive sensor)	$\pm 1.5\%$ RH
	Humidity ratio (wa,out)		± 0.2 g/kg
	Temperature ($t_{sw,in}$)		
Seawater Inlet	Salinity (σ_{in})	RTD sensor	± 0.1 °C
	Flow rate (q)	Conductivity meter	± 0.5 g/kg
	Temperature ($t_{sw,out}$)	RTD sensor	± 0.1 °C
Seawater Outlet	Salinity (σ_{out})	Conductivity meter	± 0.5 g/kg
	Air temperature ($t_a(z)$)	Thermocouples	± 0.3 °C
Tower Mid-Height ($z = 1.6$ m)	Droplet diameter ($dd(z)$)	High-speed camera	± 0.05 mm

Table 5. Operating conditions of shower cooling tower test bench.

Experimental Conditions										
	1	2	3	4	5	6	7	8	9	10
Spray height (m)	1.7	1.7	1.7	1.7	1.7	2.5	2.5	2.5	2.5	2.5
$t_{sw,in}$ (°C)	34.5	35.0	32.2	32.9	33.5	50.4	40.5	43.6	45.5	43.4
$t_{a,in}$ (°C)	28.5	28.2	29.4	27.5	27.9	33.1	32.8	32.6	32.0	31.8
$t_{wb,in}$ (°C)	20.5	20.5	19.5	19.6	19.7	28.6	28.0	28.0	27.9	27.8
q (l/min)	124.0	124.0	140.0	155.0	155.0	107.0	107.0	132.5	162.5	162.5
v (kg/s)	2.34	2.34	2.21	2.20	2.20	1.16	0.81	2.08	2.54	3.12
d_d (mm)	1.3	1.3	1.0	1.3	1.3	2.2	2.2	2.0	1.8	1.8
v_d (m/s)	6.58	6.58	7.43	8.22	8.22	2.90	2.90	3.50	4.30	4.30
v_a (m/s)	1.81	1.81	1.72	1.70	1.70	1.00	0.70	1.80	2.20	2.70

Figure 6 shows the comparison between the experimentally measured values of the tower exit parameters (water temperature, air dry and wet bulb temperature) and the numerical simulation results of the computational model. The maximum relative errors between experimental and calculated values were 1.67%, 2.53% and 2.89%, while the average relative errors were 0.45%, 1.73% and 1.30%, respectively. The calculation errors are within the acceptable range. This shows that the calculation model established in this paper has a good prediction effect on the thermodynamic parameters of droplets and air and can be used for the subsequent study of heat and moisture exchange characteristics of unpacked seawater towers.

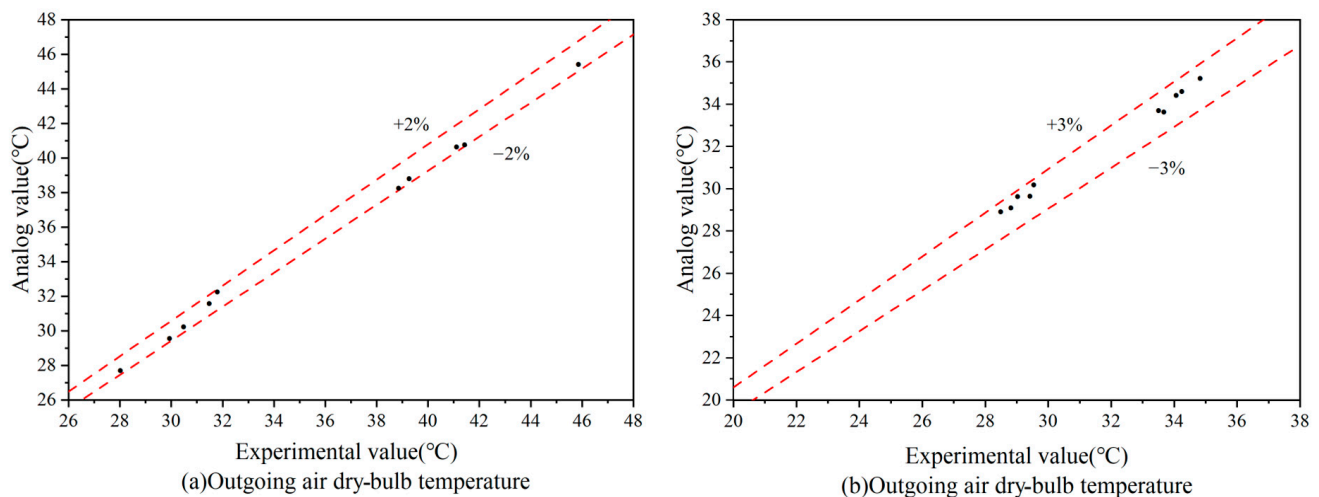


Figure 6. Comparison between experimental results and numerical simulation results.

5.4. Numerical Simulation Calculation Conditions

Taking the upward-spraying shower cooling tower in the literature [1] as the research object, the operating conditions of the tower are shown in Table 2, which are used as the numerical simulation calculation conditions, and the four salinity conditions of the circulating water are also considered. That is, $S_{in} = 0$ g/kg (fresh water), $S_{in} = 35$ g/kg (seawater standard salinity) and $S_{in} = 105$ g/kg (three times the standard salinity). The heat and mass transfer and exergy transfer process between droplets and air in the tower were numerically simulated by using the C language program, and the heat and moisture exchange characteristics were studied. If the position of the nozzle is raised to the vicinity of the water eliminator and the spraying direction is changed, it can be used as a down-spray unpacked tower. The conditions are also selected according to Table 6.

Table 6. The conditions for simulation calculation.

Working Conditions	Parameter Value	Working Conditions	Parameter Value
p	102,100 Pa	d_d	1.5 mm
q	1000 m ³ /h	v_d	8.3 m/s
g	878,000 m ³ /h	v_a	3 m/s
$t_{sw,in}$	44 °C	λ	1.0
$t_{a,in}$	28 °C	$t_{sw,out}$	30 °C
$t_{wb,in}$	21.7 °C		

6. Analysis and Discussion

6.1. Comparison of Heat and Moisture Exchange Characteristics Between Upward-Spraying Tower and Downward-Spraying Shower Cooling Tower Under Different Salinity Conditions

Based on the numerical simulation calculation conditions in Table 2, when the seawater salinity increases linearly from 0 g/kg to 105 g/kg at intervals of 17.5 g/kg, the heat and moisture exchange characteristic parameters of the top-spray /bottom-spray tower are studied.

As shown in Figure 7, the outlet water temperature of both tower types increased with increasing salinity, while the cooling efficiency demonstrated a corresponding decrease. When the salinity increased from 0 g/kg to 105 g/kg, the cooling efficiency of the upper-spray tower decreased from 65.47% to 60.09%, a decrease of 5.38%; the cooling efficiency of the downward-spray tower decreased from 48.38% to 45.95%, a decrease of 2.43%. In addition, it can be found that the cooling efficiency of the upper-spray tower is about 15.59% higher, on average, than that of the downward-spray tower at a given salinity condition.

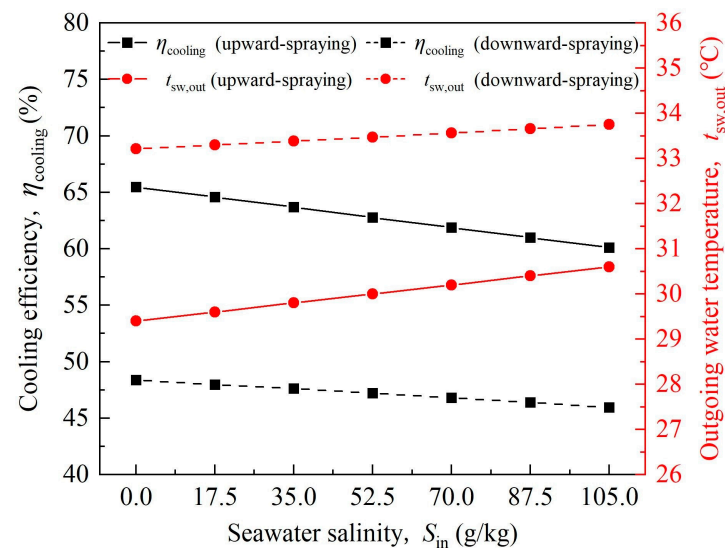


Figure 7. Variations of cooling efficiency and outlet water temperature with salinity for upward-spraying and downward-spraying towers.

An analysis of Figure 8 shows that the heat dissipation of the upper-spray /downward-spray tower is inversely proportional to the salinity, and the heat dissipation of the upper-spray tower is greater than that of the downward-spray tower. When the salinity increased from 0 g/kg to 105 g/kg, the heat dissipation of the upper-spray tower and the lower-spray tower decreased by 6.57% and 8.67%, respectively. Under a given salinity condition, the heat dissipation of the upper-spray tower is about 34.58% higher than that of the downward-spray tower, on average.

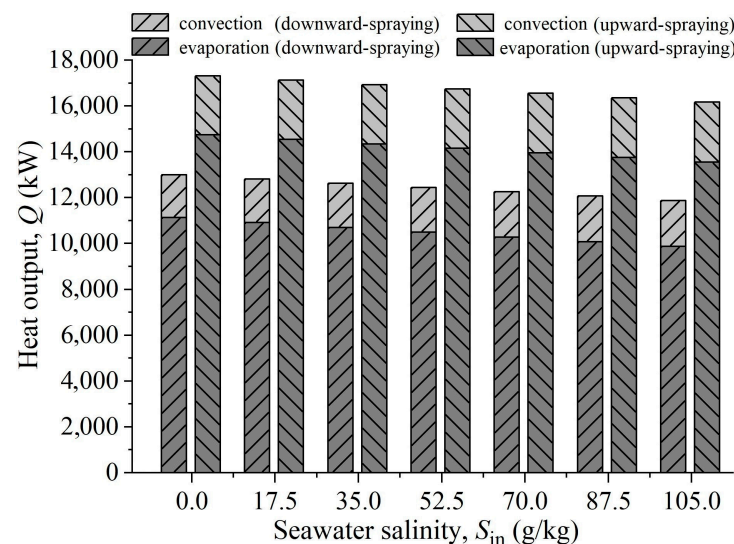


Figure 8. The variation of total heat dissipation with salinity for upward-spraying and downward-spraying towers.

Figure 9 shows that when the salinity is increased from 0 g/kg to 105 g/kg, the specific enthalpy of the outlet air of the upward-spraying tower and the downward-spraying tower decreases by 4.11% and 3.47%, respectively. Under the given salinity conditions, the cooling efficiency of the upper-spraying tower is about 8.62% higher on average than that of the downward-spraying tower. As shown in Figure 10, the increase in salinity aggravated the irreversible degree of heat and mass transfer between droplet and air, resulting in an increase in overall exergic loss and a decrease in exergic efficiency. For the

top-spray tower, when the salinity increased from 0 g/kg to 105 g/kg, the exergy efficiency decreased from 39.72% to 35.88% but was still about 4.89% higher on average than that of the downward-spraying tower. This shows that under the same tower specifications and operating conditions, the upper-spray tower has higher cooling efficiency, gas side thermal efficiency, exergy efficiency and heat dissipation, so its heat and moisture exchange characteristics are better than those of the downward-spraying tower.

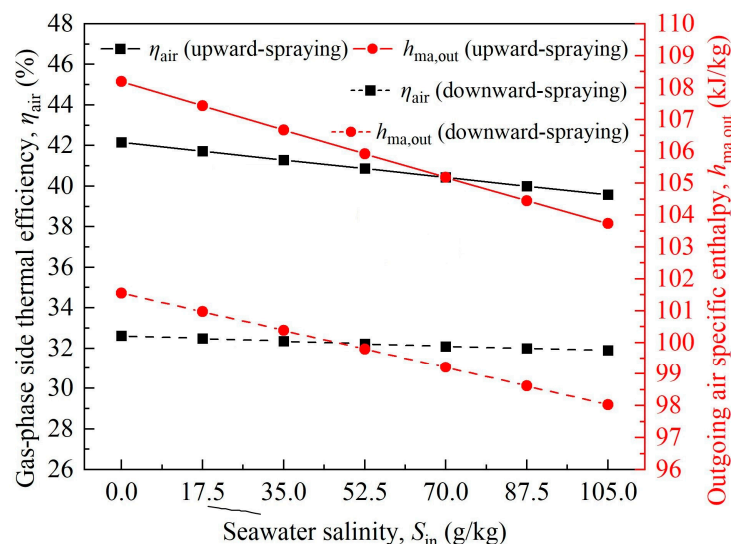


Figure 9. Variations of thermal efficiency and outlet air specific enthalpy with salinity for upward-spraying and downward-spraying towers.

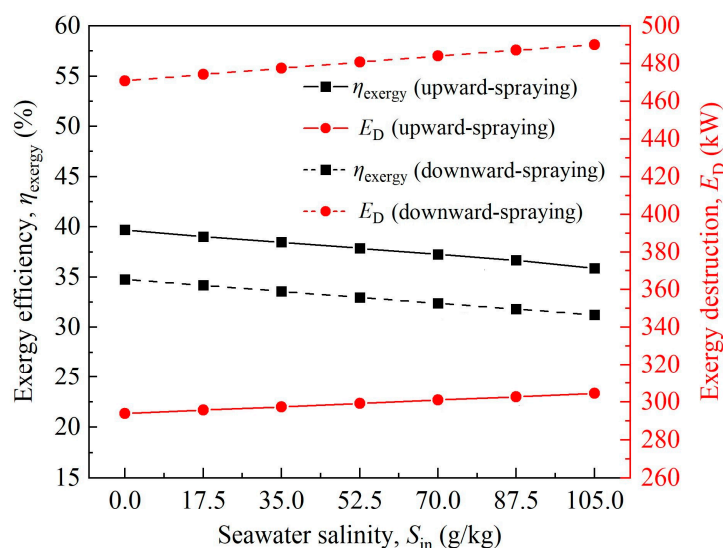


Figure 10. Variations of exergy efficiency and exergy destruction with salinity for upward-spraying and downward-spraying towers.

6.2. Influence Rule of Droplet Diameter of Upper-Spray Tower on Heat and Moisture Exchange Characteristics

Keeping $t_{a,in} = 28\text{ }^{\circ}\text{C}$, $t_{wb,in} = 21.7\text{ }^{\circ}\text{C}$, $t_{sw,in} = 21.7\text{ }^{\circ}\text{C}$, $v_d = 8.3\text{ m/s}$ and $v_a = 3\text{ m/s}$ unchanged, under the standard salinity condition ($S_{in} = 35\text{ g/kg}$) and high-salinity condition ($S_{in} = 105\text{ g/kg}$), the droplet diameter is from 1 mm to 0.25 mm. When the interval linearly increases to 2.25 mm, the influence law on the heat and moisture exchange characteristics of the tower is shown in Figures 11–14.

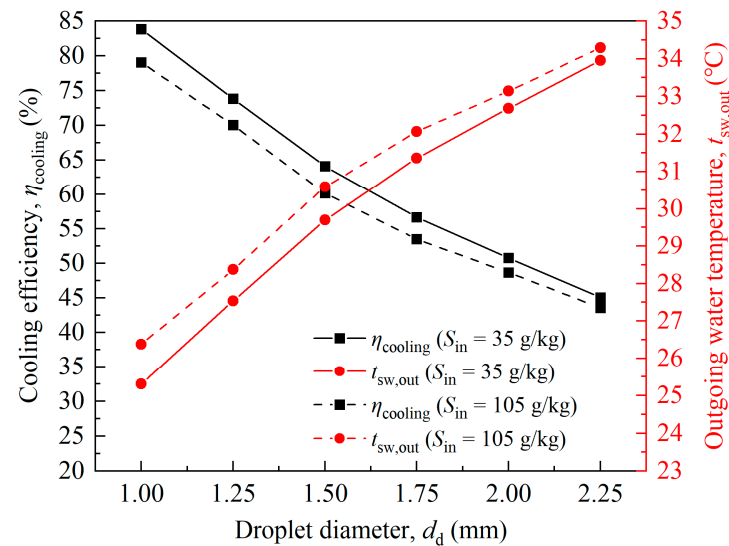


Figure 11. Variations of cooling efficiency and outlet water temperature with droplet diameter.

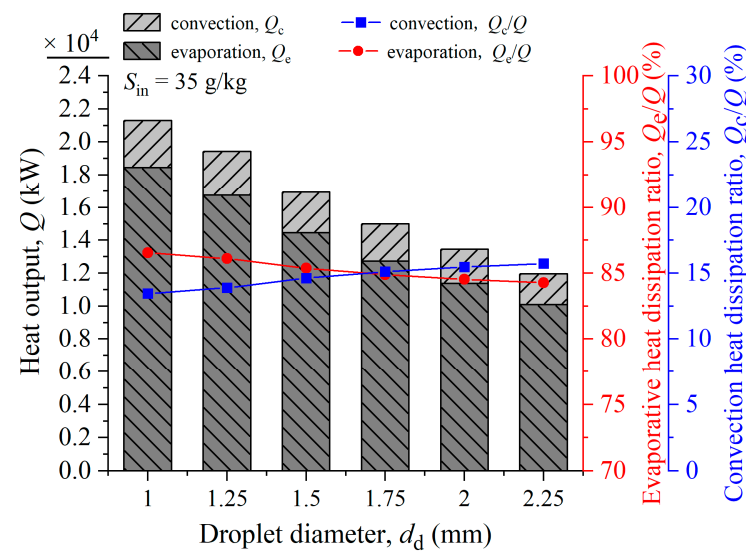


Figure 12. Total heat dissipation and its composition proportion with different droplet diameters.

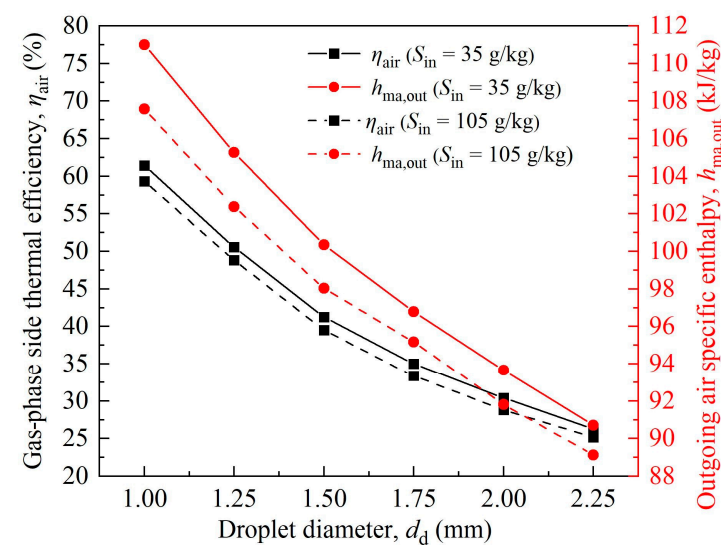


Figure 13. Variations of thermal efficiency and outlet air specific enthalpy with droplet diameter.

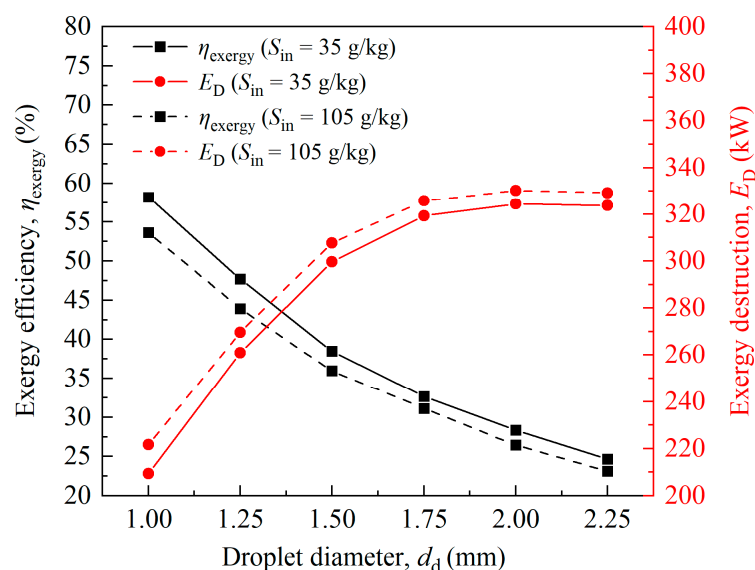


Figure 14. Variations of exergy efficiency and exergy destruction with different droplet diameters.

Figure 11 shows that $t_{\text{sw,out}}$ increases and gradually becomes slower as d_d increases. Taking the standard salinity condition as an example, when d_d is 1 mm, the minimum value of $t_{\text{sw,out}}$ is 25.31 °C, and as d_d increases to 2.25 mm, $t_{\text{sw,out}}$ increases to 33.95 °C, an increase of 34.14%. Since $t_{\text{sw,in}}$ and $t_{\text{wb,in}}$ remain constant, the increase in $t_{\text{sw,out}}$ leads to the decrease in cooling water temperature difference and the increase in cooling amplitude, so η_{cooling} decreases from 83.81% to 45.05%, a decrease of 38.76%.

Figure 12 shows that when $S_{\text{in}} = 35$ g/kg, d_d increases from 1 mm to 2.25 mm, and the heat dissipation Q of the whole tower decreases from 21,300.60 kW to 11,959.03 kW, a decrease of 78.11%. This indicates that an increase in droplet diameter weakens the cooling capacity of the tower to a greater extent. As the droplet diameter increases, the ratio of evaporative heat dissipation decreases while the proportion of convection heat dissipation increases, though the variation remains within 4%.

As shown in Figure 13, under the standard salinity condition, when d_d is 1 mm, $h_{\text{ma,out}}$ is at most 110.99 kJ/kg, the corresponding η_{air} is 61.43%, and as d_d increases to 2.25 mm, $h_{\text{ma,out}}$ decreases to 90.71 kJ/kg, η_{air} decreases to 26.30%, decreasing by 18.27% and 35.13%, respectively. Under high-salinity conditions, when d_d increases from 1 mm to 2.25 mm, $h_{\text{ma,out}}$ and η_{air} decrease by 34.13% and 17.16%, respectively, which are slightly lower than those in standard salinity conditions.

It can be seen from Figure 14 that with the increase in d_d , under the standard salinity condition, when it increases from 1 mm to 2.25 mm, E_D increases from 209.39 kW to 323.74 kW, an increase of 54.61%, and η_{exergy} decreases from 58.20% to 24.67%, a decrease of 33.53%. This shows that the larger the droplet diameter, the greater the irreversibility of the heat and mass transfer process between the water and gas phases in the tower, and the smaller the effective energy available for use and transfer. Under the high-salinity condition, when d_d was increased from 1 mm to 2.25 mm, the changes in E_D and η_{exergy} were 48.49% and 30.55%, respectively, which were lower than in the standard salinity condition. This shows that the change in d_d has a great influence on the heat and moisture exchange characteristics of the tower, the change trends of η_{cooling} , η_{air} , η_{exergy} and Q are consistent and the change range gradually decreases with the increase in diameter.

6.3. Influence of the Air Velocity of the Upward-Spray Tower on the Heat and Moisture Exchange Characteristics

The effects of air velocity on the heat and mass transfer characteristics were investigated under both standard and high-salinity conditions. The air velocity was varied from 2.5 m/s to 4.0 m/s with increments of 0.25 m/s, while other parameters remained constant. The experimental results are shown in Figures 15–18.

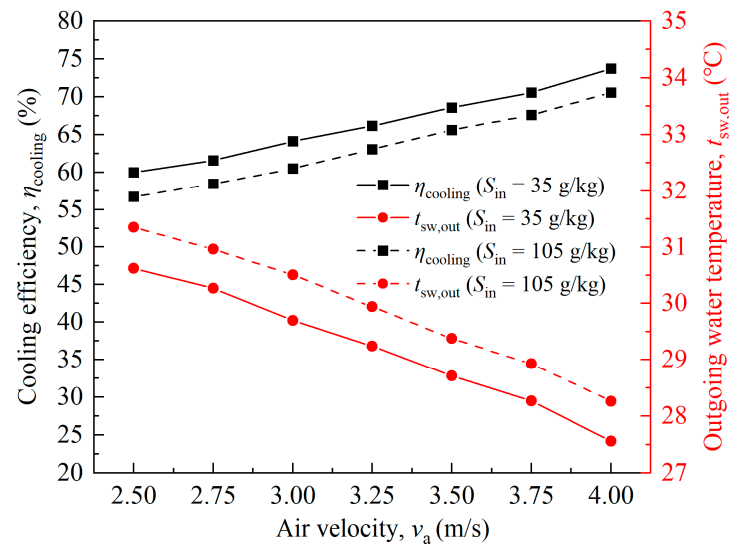


Figure 15. Variations of cooling efficiency and outlet water temperature with different air speeds.

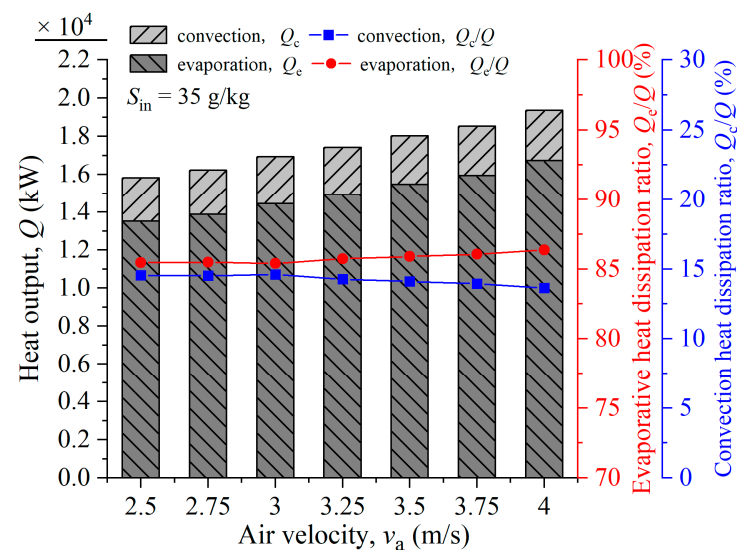


Figure 16. Total heat dissipation and its composition proportion with different air speeds.

As can be seen from Figure 15, $t_{sw,out}$ decreases as v_a increases. Under the standard salinity condition, when v_a is 2.5 m/s, $t_{sw,out}$ is the highest at 30.63 °C, and as v_a increases to 4 m/s, $t_{sw,out}$ decreases to 27.56 °C, a $t_{sw,out}$ decrease of 10.02%. Since $t_{sw,in}$ and $t_{wb,in}$ remain constant, when $t_{sw,out}$ decreases, the cooling water temperature difference increases and the cooling amplitude decreases, so that $\eta_{cooling}$ increases with the increase in v_a , and $\eta_{cooling}$ increases from 59.98% to 73.74%, an increase of 13.76%. Under high-salinity conditions, $t_{sw,out}$ increased by 2.46% on average compared to standard salinity conditions, and $\eta_{cooling}$ decreased by 2.96% compared to standard salinity conditions.

Figure 16 shows that under the standard salinity condition, when v_a increases from 2.5 m/s to 4 m/s, the heat dissipation of cooling tower Q increases from 15,808.58 kW

to 19,355.55 kW, an increase of 22.44%. In addition, it can also be found that both the evaporative heat dissipation Q_e and the convection heat dissipation Q_c have increased, but the proportions (Q_e/Q and Q_c/Q) of each of the total tower heat dissipation hardly change with the change in the air flow rate and are basically maintained at 85.77% and 14.23%.

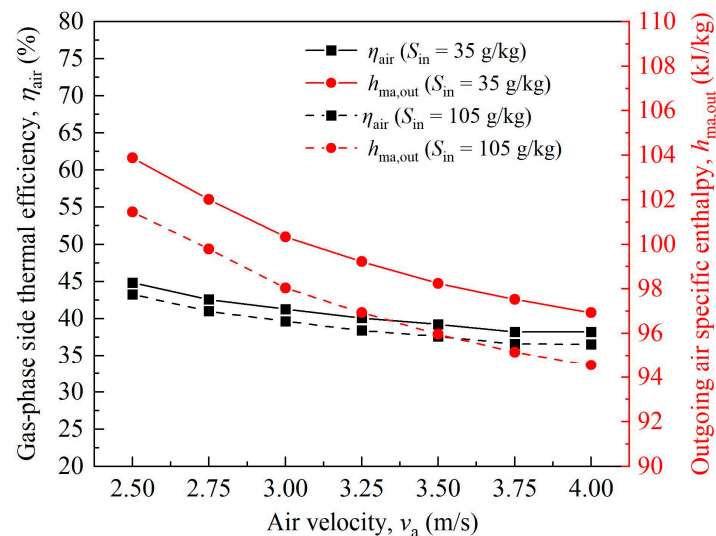


Figure 17. Variations of thermal efficiency and outlet air specific enthalpy with different air speeds.

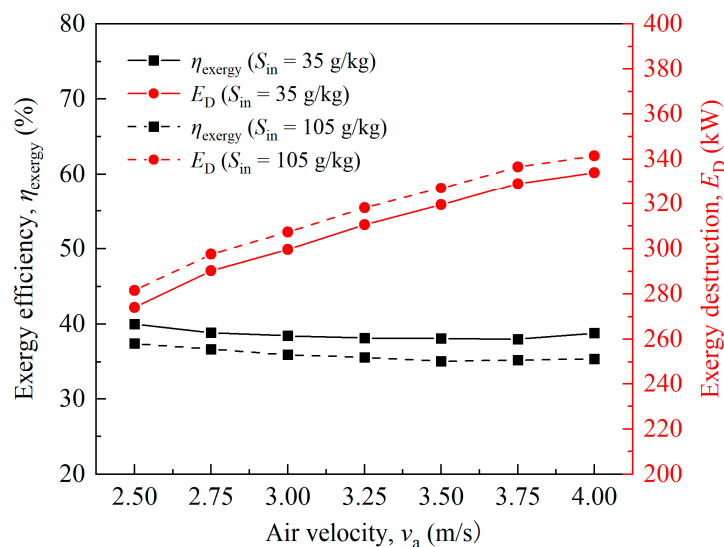


Figure 18. Variations of exergy efficiency and exergy destruction with different air speeds.

It can be seen from Figure 17 that both the thermal efficiency η_{air} and the specific enthalpy $h_{\text{ma,out}}$ of the outlet air at the gas-phase side decrease with the increase in the air flow rate v_a . Under standard salinity conditions, when v_a is 2.5 m/s, $h_{\text{ma,out}}$ is 103.88 kJ/kg and η_{air} is 44.84%. As v_a increases to 4 m/s, $h_{\text{ma,out}}$ becomes 96.94 kJ/kg, and η_{air} becomes 38.18%, decreasing by 6.94 kJ/kg and 6.66%, respectively. Under high-salinity conditions, $h_{\text{ma,out}}$ is reduced by an average of 2.34% compared to standard salinity conditions, and η_{air} is reduced by an average of 1.63% compared to standard salinity conditions. Figure 18 shows the curves of E_D and η_{exergy} as a function of v_a . Taking the standard salinity condition as an example, when v_a increases from 2.5 m/s to 4 m/s, E_D increases from 274.10 kW to 333.91 kW, an increase of 21.82%, while η_{exergy} changes by less than 2%. Under the high-salinity condition, E_D increased by 2.47%, on average, compared to the standard salinity condition, and η_{exergy} decreased by 2.73% compared to the standard salinity condition.

6.4. Optimal Analysis of Up-Spray Cooling Tower

The above operating conditions ($d_d = 1.5$ mm, $v_d = 8.3$ m/s, $v_a = 3$ m/s, $t_{sw,in} = 44$ °C) of the spray-type unpacked seawater tower are used as reference conditions. The value of each evaluation index is $\eta_{cooling} = 64.13\%$, $E_D = 299.67$ kW, $Q = 16,390$ kW, $t_{sw,out} = 29.7$ °C, the cooling water temperature difference $\Delta t_{sw} = 14.3$ °C and the cooling height is 8 °C. When the same heat dissipation (heat load) Q is satisfied, the maximum cooling efficiency $\eta_{cooling}$ and the minimum E_D are the optimization goals. The response optimizer of Minitab software 20.3 is used to directly solve the multi-objective optimization problem, and the optimal operating conditions of the top-spray unpacked seawater cooling tower under the current meteorological conditions are determined. The optimization results are shown in Table 7. The composite desirability of the optimized condition is 0.9999. Compared with the reference condition, when Q is basically the same, the optimized results make all response values reach the optimal state. Considering the actual operation of the cooling tower and taking $d_d = 1$ mm, $v_d = 6.6$ m/s, $v_a = 3.3$ m/s and $t_{sw,in} = 39.8$ °C for numerical simulation, the comparison shows that the numerical calculation results are not much different from the optimization results, which verifies the reliability of the optimization using the response surface method. Taking this combination of working condition parameters as the best operating condition of the cooling tower, the optimized cooling efficiency is increased by about 14.69% compared with the reference condition, the exergy destruction is reduced by about 37.87%, the outlet water temperature is about 25.53 °C (the cooling water temperature difference is 14.27 °C; the cooling amplitude is high at 3.83 °C) and the cooling effect of the cooling tower is optimal under the current meteorological conditions.

Table 7. Multi-objective optimization results.

	Impact Factor				Response Variable					
	d_d (mm)	v_d (m/s)	v_a (m/s)	$t_{sw,in}$ (°C)	$\eta_{cooling}$ (%)	E_D (kW)	Q (kW)	$t_{sw,out}$ (°C)	Δt_{sw} (°C)	$t_{sw,out} - t_{wb}$ (°C)
Reference condition	1.5	8.3	3	44	64.13	299.67	16,930	29.70	14.30	8.00
Optimized conditions	1	6.59	3.2525	39.8182	78.33	183.82	16,932.67	—	—	—
Simulation verification	1	6.6	3.3	39.8	78.82	186.18	17,055.67	25.53	14.27	3.83

6.5. Implications and Applications for Central Air Conditioning Systems

Based on the comprehensive investigation of shower cooling towers under various operating conditions, several significant implications for central air conditioning systems have been identified. The experimental results indicate that optimal droplet diameters range from 1.00 mm to 2.25 mm, with cooling efficiency decreasing by 38.76% as droplet diameter increases across this range. This finding provides crucial guidance for nozzle selection in practical applications. For seawater cooling systems, when salinity increased from 0 to 105 g/kg, the outlet seawater temperature increased by 34.14%, and the total heat transfer rate decreased by 78.11%. These results suggest the necessity of performance compensation strategies in high-salinity applications.

The established correlations between operating parameters and cooling performance enable more precise control strategies for central air conditioning systems. The comparison between upward-spraying and downward-spraying configurations showed that the cooling efficiency difference was 15.59% under identical conditions, indicating potential energy savings through optimal spray direction selection. Furthermore, the identified trade-off between evaporative and convective heat transfer can be utilized to optimize system per-

formance under different climate conditions, particularly in regions with varying humidity levels throughout the year.

The mathematical model developed and validated in this study, with maximum relative errors of 1.67%, 2.53% and 2.89% for the three key parameters, respectively, provides a reliable tool for predicting system performance under various operating conditions. This enables system operators to make real-time adjustments based on environmental conditions and optimize water and air flow rates to achieve maximum efficiency. The model's accuracy in predicting performance degradation due to salinity effects is particularly valuable for facilities using seawater as a cooling medium, allowing preventive measures to be implemented before significant efficiency losses occur.

From an economic and environmental perspective, the implementation of these research findings can significantly improve the overall performance of central air conditioning systems. The optimized design parameters and operating strategies can lead to reduced energy consumption through improved cooling efficiency. The comprehensive understanding of salinity effects contributes to more effective maintenance scheduling and reduced operational costs. Moreover, the precise performance prediction capabilities enable better system reliability and water conservation through optimized operation. These improvements are particularly significant for coastal areas where seawater cooling applications are prevalent, offering both economic and environmental benefits through enhanced system performance and resource utilization.

6.6. Carbon Emission Reduction Potential

The environmental benefit is quantified by comparing the CO₂-equivalent emissions of the optimized seawater cooling tower to conventional freshwater systems. Emissions are calculated using

$$E_{CO_2} = \left(\frac{Q_{\text{fresh}} - Q_{\text{sea}}}{\text{CEF}_{\text{power}}} \right) + \Delta \text{CO}_2 \quad (32)$$

where Q_{fresh} and Q_{sea} is the energy consumption of freshwater and seawater systems, respectively, kW; $\text{CEF}_{\text{power}}$ is the carbon emission factor of grid electricity (0.85 kg CO₂/kWh); and ΔCO_2 is emissions saved by avoiding freshwater extraction (0.15 kg CO₂/m³ [2]).

The optimized upward-spraying tower reduces energy consumption by 18.6% compared to freshwater systems, yielding 142 tonnes CO₂/year reduction for a 1000 m³/h cooling load. Seawater utilization eliminates freshwater intake, saving 12,600 m³/year and avoiding 1.89 tonnes CO₂/year from water treatment.

6.7. Brine Discharge and Salinity Impacts

High-salinity brine discharge (>70 g/kg) poses ecological risks. We evaluate the salinity concentration outlet (C_{out}) and dilution requirements to meet marine discharge standards (<40 g/kg) as follows:

$$C_{\text{out}} = \frac{m_{\text{sw,in}}\sigma_{\text{in}} - m_{\text{evap}}\sigma_{\text{evap}}}{m_{\text{sw,in}} - m_{\text{evap}}} \quad (33)$$

where $m_{\text{sw,in}}$ is the seawater mass inlet, kg; m_{evap} is the evaporated water mass, kg; σ_{in} is the salinity of the seawater inlet, g/kg; and σ_{out} is the salinity of the seawater outlet, g/kg.

The mitigation strategy involves blending fresh water to reduce brine salinity. The following example(s) illustrate this: at $\sigma_{\text{in}} = 105$ g/kg, blending 30% fresh water reduces C_{out} to 38 g/kg, meeting regulatory limits.

Based on calculation and analysis, the environmental performance metrics of the seawater spray cooling tower are presented in Table 8.

Table 8. Environmental performance metrics.

Indicator	Upward-Spraying Tower	Downward-Spraying Tower	Freshwater Tower
CO ₂ emissions (tonnes/year)	624	732	766
Freshwater use (m ³ /year)	0	0	12,600
Brine discharge salinity (g/kg)	38 (with dilution)	45 (with dilution)	–

7. Conclusions

Based on the calculation model, the effects of various parameters on the heat and moisture exchange characteristics of the upward-spraying tower were investigated, including water salinity, droplet diameter, air flow rate and initial droplet conditions. The following conclusions can be drawn:

(1) The cooling efficiency, exergy efficiency, gas-phase thermal efficiency and heat dissipation decreased with increasing salinity for both upward-spraying and downward-spraying configurations. However, compared with the downward-spraying tower, the upward-spraying configuration demonstrated superior performance with increases of 15.59%, 4.89%, 8.62% and 34.58% in cooling efficiency, exergy efficiency, gas-phase thermal efficiency and heat dissipation, respectively. These results indicate that the upward-spraying configuration exhibits better heat and mass transfer characteristics.

(2) The droplet diameter exhibits a significant impact on cooling performance. As the droplet diameter increased from 1.00 mm to 2.25 mm, the cooling efficiency decreased by 38.76%, the heat dissipation decreased by 78.11% and the exergy efficiency decreased by 33.53% under the standard salinity condition. This inverse relationship between droplet size and cooling performance is attributed to the reduced surface area-to-volume ratio and shortened air–water contact time of larger droplets. These findings provide quantitative guidance for optimal nozzle selection in practical applications.

(3) With increasing air velocity, the cooling performance of the tower improved significantly under standard salinity conditions. The enhanced heat and mass transfer can be attributed to the increased turbulent mixing and extended contact time between air and water droplets. The increase in air velocity from 2.5 m/s to 4.0 m/s led to a 13.76% improvement in cooling efficiency and a 22.44% increase in heat dissipation, and exergy efficiency decreased by less than 2% under the standard salt condition. The results revealed that higher air velocity promotes both convective and evaporative heat transfer processes, leading to better overall cooling efficiency.

(4) Compared to air flow rate, droplet diameter exhibited a more significant influence on cooling efficiency, exergy efficiency and heat dissipation under the investigated conditions. Under high-salinity conditions, although all performance indicators decreased compared to standard salinity conditions, the reduction remained within 6%. For the upward-spraying tower under high-salinity conditions, the cooling efficiency and exergy efficiency showed slightly higher values than the gas-phase thermal efficiency when compared to standard salinity conditions. Moreover, the cooling efficiency and exergy efficiency demonstrated more substantial decreases at smaller droplet diameters.

(5) Based on the optimization analysis of the seawater cooling tower, selecting the best operating conditions can significantly improve the heat and moisture exchange characteristics and carbon reduction potential of the seawater cooling tower. The optimization results revealed that the cooling efficiency increased by 14.69% and exergy destruction decreased by 37.87%, demonstrating improved energy performance and environmental benefits for central air conditioning applications.

(6) By optimizing seawater cooling tower design and implementing strategic brine discharge management, the proposed system demonstrates significant environmental

benefits, including an 18.6% reduction in energy consumption, an annual CO₂ emissions reduction of 142 tonnes, water savings of 12,600 m³ and effective mitigation of high-salinity discharge risks through targeted freshwater blending, thereby simultaneously addressing energy efficiency, carbon emissions, water conservation and marine ecological protection.

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Conflicts of Interest: The authors declare no conflicts of interest.

Nomenclature

Symbol	Description	Unit
A	Heat transfer area	m ²
A_d	Droplet cross-sectional area	m ²
C_D	Drag coefficient	-
$c_{p,a}$	Specific heat of air at constant pressure	kJ/(kg·°C)
$c_{p,w}$	Specific heat of seawater	kJ/(kg·°C)
D	Diffusion coefficient of water vapor in air	m ² /s
d_d	Droplet diameter	mm
Ex	Exergy	kW
g	Gravitational acceleration	m/s ²
h	Specific enthalpy of humid air	kJ/kg
h_{fg}	Latent heat of vaporization	kJ/kg
H	Tower height	m
Le	Lewis factor	-
m_a	Mass of air	kg
m_d	Mass of droplet	kg
m_w	Mass of seawater	kg
Nu	Nusselt number	-
Pr	Prandtl number	-
Q	Heat dissipation	kW
q	Volumetric flow rate of seawater	m ³ /h
Re	Reynolds number	-
Sc	Schmidt number	-
Sh	Sherwood number	-
t	Temperature	°C
$t_{sw,in}$	Inlet seawater temperature	°C
$t_{sw,out}$	Outlet seawater temperature	°C
$t_{a,in}$	Inlet air dry-bulb temperature	°C
$t_{wb,in}$	Inlet air wet-bulb temperature	°C

v_a	Air velocity	m/s
v_d	Initial droplet velocity	m/s
η_c	Cooling efficiency	%
η_{ex}	Exergy efficiency	%
η_{th}	Gas-phase thermal efficiency	%
λ	Thermal conductivity of air	kW/(m·°C)
ρ_a	Air density	kg/m ³
ρ_w	Seawater density	kg/m ³
σ	Salinity of seawater	g/kg
ΔH	Distance between nozzle and tower bottom	m
ΔT	Cooling water temperature difference	°C

Abbreviations

ST-UD	Upward-spraying tower with downward airflow
ST-DU	Downward-spraying tower with upward airflow
ST-UU	Upward-spraying tower with upward airflow

Subscripts

a	Air
c	Contact
d	Droplet
e	Evaporation
ex	Exergy
in	Inlet
ma	Moist air
out	Outlet
sw	Seawater
th	Thermal
v	Vapor
wb	Wet-bulb

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