

A liquid desiccant-assisted free-cooling system for energy-efficient data centers in hot and humid climates

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ABSTRACT

Air conditioning and cooling are essential to ensure the reliable operation of data centers. Several studies have been conducted regarding the use of free cooling for energy-efficient cooling. However, free cooling faces challenges when used in hot and humid environments, which can lead to a decline in cooling tower performance. To address this issue, we propose a cooling system that utilizes a liquid desiccant to lower the humidity and improve the performance of the cooling tower, thereby reducing the energy consumption of the refrigeration system. Simulations were conducted in areas where the data centers were located, considering hot and humid climates. According to the simulation results, it was possible to achieve an average reduction of 35.12 % in cooling energy consumption by improving the performance of the cooling tower.

Nomenclature

a	surface area-to-volume ratio of packing [m^2/m^3]
cp:	specific heat of air [$\text{kJ}/\text{kg} \text{ } ^\circ\text{C}$]
g	gravitational acceleration [kg/s^2]
DBT	inlet air temperature [$^\circ\text{C}$]
G	gas flow rate [kg/h]
h	enthalpy [kJ/kg]
H	total head [m]
L:	liquid flow rate [kg/h]
m	mass flow [kg/h]
ps	vapor pressure of desiccant solution [kPa]
Q	load [kW]
ρ :	specific weight of liquid [kg/m^3]
T	temperature [$^\circ\text{F}$, $^\circ\text{C}$]
V	air velocity [m/s]
Z	packing height [m]

Greek Symbols

ϵ effectiveness

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ω	humidity ratio [kg/kg]
$\cdot$	function of vapor pressure depression of the desiccant solution to the vapor pressure of pure water $[(P_{water} - P_{solution})/P_{water}]$
η :	efficiency
<i>Subscripts</i>	
a	air
abs	absorber
c	cooling water
eq	equilibrium humidity
i	inlet
o	outlet
r	refrigerant
reg	regenerator
s	solution
server	server of data center
sump:	water, solution sump
<i>Abbreviations</i>	
cond	condenser
CT	cooling tower
evap:	evaporator
HX	heat exchanger
pPUE	Partial power usage effectiveness

1. Introduction

Data centers are infrastructure designed to securely manage exponentially increasing data in the implementation and cultivation of information and communications technology industries such as Internet of Things, cloud computing, big data, and mobile communication [1]. The number of data centers is increasing worldwide, owing to the growing amount of data available. The cloud market in Southeast Asia is expected to expand significantly in line with the regional growth of the Internet economy. Data center occupancy across Southeast Asia averages 60–70 %, with Singapore being a key country with an occupancy rate of over 85 % [2]. In 2020, Singapore's data center investments contributed to the growth in demand for data centers in Indonesia, Malaysia, and other countries. The rapid increase in data center demand is driven by the expansion of hyperscale cloud service providers and internet service providers [2]. However, the increasing demand for data centers has led to higher energy consumption and increased heat-dissipation density [3]. As a result of the analysis of data center energy consumption, cooling accounts for 50 % of the total energy consumption, whereas servers and storage utilize 26 %, energy usage for power conversion amounts to 11 %, network hardware accounts for 10 %, and the remaining 3 % is attributed to lighting within the data center [4]. To achieve energy-efficient data centers, it is essential to reduce energy consumption, specifically for cooling, which is the most energy-intensive aspect. Various cooling technologies have been researched to minimize cooling energy consumption in data centers, most notably free cooling, which involves utilizing outside air for data center cooling [5]. Free cooling often uses water-side economizers connected to cooling towers, evaporative coolers, or dry coolers to remove heat from data centers [6]. It utilizes outside air as a cooling medium or directly supplies it to a data center when the outside air temperature is sufficiently low. One of the advantages of using outside air in free cooling is its lower energy consumption compared to traditional cooling systems that rely on refrigeration units.

INTEL [7] tested an air economizer that used outside air at temperatures up to 32.2 °C to cool production servers, potentially reducing annual operating costs by up to USD 2.87 million for a 10 MW data center. According to Miller [8], Google operates data centers in Belgium using a free-cooling system without refrigeration to improve energy efficiency. According to Amado et al. [9], with the implementation of free cooling systems in 14 states throughout Brazil, cities like Curitiba, São Paulo, Porto Alegre, and Brasília have the capability to utilize both airside and waterside economizer modes for over 3000 h annually; this is achievable even when considering conservative data center thermal operating limits.

Siriwardana et al. [10] implemented airside economizers in certain states in Australia, taking advantage of the cool and dry climate conditions. Consequently, they confirmed cost savings in data center cooling expenses, thus demonstrating the potential of airside economizer usage. Lee and Chen [11] investigated the energy-saving potential of air-side free cooling technology with differential enthalpy control in data centers spanning 17 climate zones. These findings unequivocally establish the viability of the free-cooling system in mixed-humid, warm-marine, and mixed-marine climate zones. Udagawa et al. [12] conducted simulations in four regions of Japan to develop guidelines for implementing free-cooling systems in data centers, which resulted in an 8 % reduction in power consumption when using air-side economizers. Specialized cooling tower installations require space, cost, and capacity considerations, whereas packaged air conditioners with expanded low-pressure ratio control have shown the lowest annual power consumption in all regions. Capzzoli et al. [13] investigated the state-of-the-art and emerging technologies in datacenter cooling systems and performed a

comparative analysis of their advantages and disadvantages. In addition, they examined waste heat technologies and proposed possibilities for improving energy efficiency.

However, free cooling has limitations in regions exhibiting hot and humid climates. The use of cooling towers in free-cooling systems often cannot provide sufficient cooling for the coolant [14]; this is because, when hot and humid outside air enters the cooling tower, the temperature of the incoming cold water and the evaporation cooling process are affected by the high humidity and temperature of the outside air. Consequently, the temperature of the cold water increases, leading to challenges in maintaining effective cooling. Therefore, to apply a free cooling system in Southeast Asia data centers, improvements in the evaporative cooling performance of the cooling towers are required. This involves finding ways to reduce the humidity and temperature of the hot and humid outside air to lower the temperature of the cold water during the evaporation-cooling process.

In this study, we propose the implementation of a cooling system using a liquid desiccant to enhance the performance of cooling towers in hot and humid climate regions, specifically for free cooling purposes. This cooling system incorporates dehumidification and regeneration processes in which the desiccant solution is sprayed to extract moisture from the incoming air in the dehumidification section [15]. The primary research objective is to improve cooling tower efficiency and minimize the energy consumption related to refrigeration, resulting in more effective cooling for data centers. Moreover, an assessment is conducted to evaluate the suitability of this liquid-desiccant-based free-cooling system for deployment in Southeast Asia.

2. System overview

2.1. Reference system

The reference system is shown in Fig. 1. It consists of a chiller, a cooling tower, a heat exchanger, a fan, and a pump [16].

The cooling tower produces coolant through the evaporative cooling of outside air and water. If the coolant temperature does not decrease to the design temperature required to remove heat from the data center, chiller units are employed to further lower the coolant temperature. Heat from the condenser of the chiller unit is dissipated into the atmosphere.

2.2. Proposed system

The cooling system proposed for the data center in this study utilizes a liquid desiccant system. The goal of this system is to lower the wet-bulb temperature to improve the performance of the cooling tower. The proposed system is illustrated in Fig. 2.

The cooling system is composed of a liquid-desiccant unit (absorber and regenerator), two types of cooling towers, a chiller, three heat exchangers, fans, and pumps. During summer, the outdoor air passes through the absorber of the liquid desiccant system. In this process, the inlet air and desiccant solution interact, causing the moisture in the outdoor air to be absorbed by the solution due to the difference in vapor pressure between the air and moisture.

A solution with a lower concentration during the dehumidification process is collected in a sump. Similarly, the diluted solution can be regenerated through a regeneration process to increase its concentration. The regenerated solution is heated to increase its temperature and then flowed into the regenerator unit, where water vapor is removed through a desorption process. To maintain the regeneration and dehumidification performance of liquid-desiccant solutions, it is essential to maintain the appropriate temperature and consistent temperature and concentration levels [17]. There has been extensive research on the ranges of temperature and humidity, and temperature ranges between 20 and 35 °C and humidity levels around 30–45 % have been suggested for optimal dehumidification [18]. The desiccant solution exhibits effective dehumidification performance at low temperatures and high concentrations [19,20]. After the dehumidification process, the solution becomes diluted due to an increase in moisture content. Heating is required to remove the moisture content within the solution, and the heated solution must subsequently be cooled to lower temperatures [21]. In this study, uses the method of recovering waste heat generated from the condenser of the chiller and increasing the concentration of the diluted desiccant solution through a regenerator process. This method utilizes waste heat for heating the solution, allowing for the omission of heating energy. In addition, a cooling tower is used to lower the temperature of the desiccant solution. The coolant and solution discharged from the cooling tower for desiccant solution cooling can be brought to the desired design temperature for dehumidification in heat exchanger 3 (HX3). After passing through the dehumidification process, the outside air experiences a decrease in its wet-bulb temperature. When the outside air with a lowered wet-bulb temperature is introduced into the cooling tower, it encounters the coolant and undergoes free cooling. The coolant, which is discharged from the cooling tower, flows through heat exchanger 1 (HX1) to remove the heat generated by the data center. HX 1 is placed to prevent damage to the equipment in the data center when the coolant flows directly into the data center. If the coolant temperature is not sufficient to remove the heat, the coolant is

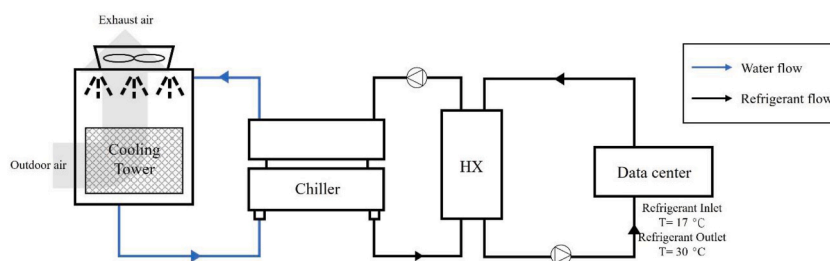


Fig. 1. Schematic diagram of Reference cooling system.

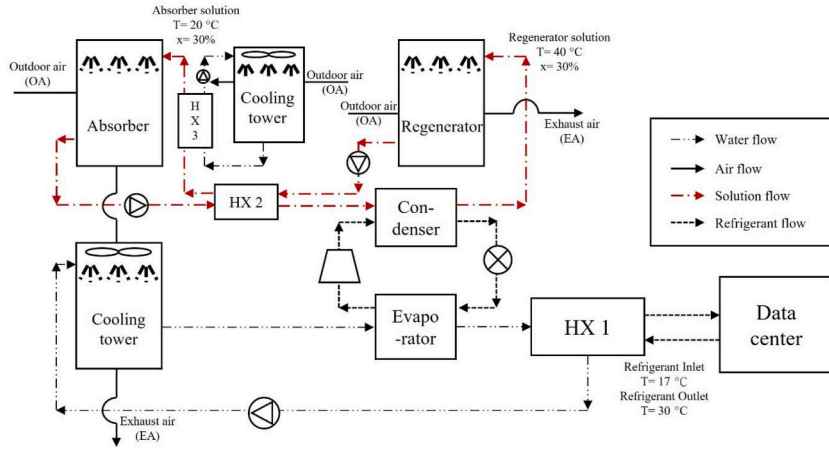


Fig. 2. Schematic diagrams of proposed cooling system.

further cooled in the refrigeration unit to meet the design temperature; however, if the coolant temperature is sufficient to satisfy the design temperature, it can be directly used for heat removal without additional cooling in the refrigeration unit.

3. Simulation overview

3.1. EnergyPlus weather data

Energyplus includes typical weather conditions based on previous years. This study selected weather data for five regions from EnergyPlus to evaluate the dehumidification performance of the proposed system in hot and humid Southeast Asia and compared it with the reference system. The five regions chosen were Singapore (Singapore), Vietnam (Hanoi), Indonesia (Jakarta), Malaysia (Kuala Lumpur), and the Philippines (Manila), where data centers are currently situated. Weather data from June to August were used for this analysis.

3.2. Data center condition

Determining the total IT load plays a crucial role in guiding data centers to establish the appropriate IT power density for designing the cooling system, as well as in making decisions regarding the cooling method and capacity. For this study, a total IT load of 150 kW was assumed, which represents the accumulated minimum unit IT room configuration, as indicated in prior research [22]. It was further assumed that each rack would generate 15 kW of heat with 10 racks deployed. The R134a refrigerant, introduced for the purpose of dissipating the heat generated in the data center, enters the liquid state at a temperature of 17 °C ($T_{HX1,i,r}$); after heat removal, it undergoes a phase change into the gaseous state and is exhausted at 30 °C ($T_{HX1,o,r}$). It is assumed that a complete phase change from liquid to gas occurs in the R134a refrigerant. The flow rate of the R134a refrigerant can be calculated using the heat generated in the data center and the enthalpy difference between the refrigerant's inlet and outlet ($h_{server,o,r}$, $h_{server,i,r}$). As shown in Eq. (1), considering the amount of heat and enthalpy.

$$m_{server,r} = \frac{Q_{server}}{h_{server,o,r} - h_{server,i,r}} \quad (1)$$

HX 1 is utilized to cool the refrigerant that returns from the data center servers after removing the heat from the servers in the evaporator.

The refrigerant that is cooled in the evaporator and returns from the data center servers, circulating back to the servers for cooling, is assumed to have a temperature of 17 °C, and the efficiency of HX 1 is assumed to be 75 % based on previous research [23]. The capacity of the heat exchanger (Q_{HX1}) can be calculated based on the enthalpy difference between the refrigerant flow entering the heat exchanger ($m_{server,r} = m_{HX1}$) and the inlet and outlet of the R134a refrigerant (Eq. (2)).

$$Q_{HX1} = m_{HX1} (h_{HX1,o} - h_{HX1,i}) \quad (2)$$

The temperature of the coolant in and out of HX 1 can be calculated using the temperature difference between the inlet coolant and the refrigerant, as well as the heat exchanger efficiency (Eqs. (3) and (4)). In this study, the outlet temperature of $T_{HX1,o,c}$ was calculated assuming a coolant outlet temperature of 12 °C for the coolant that is cooled and discharged from the chiller. Additionally, the coolant flow rate ($m_{HX1,c}$) was determined using the heat exchanger capacity, the specific heat of the coolant, and the coolant temperature difference in HX 1 (Eq. (5)).

$$T_{HX1,i,c} = T_{HX1,i,r} - \left(\frac{T_{HX1,i,r} - T_{HX1,o,r}}{\varepsilon_{HX1}} \right) \quad (3)$$

$$T_{HX1,o,c} = T_{HX1,i,c} + \varepsilon_{HX1} (T_{HX1,i,r} - T_{HX1,i,c}) \quad (4)$$

$$m_{HX1,C} = \frac{Q_{HX1}}{cp_{HX1,i,c}} (T_{HX1,o,c} - T_{HX1,i,c}) \quad (5)$$

3.3. Liquid desiccant unit

LiCl solutions are commonly used in liquid desiccant applications [24]. In this study, a LiCl solution was used in the liquid desiccant system. The dehumidification effectiveness of the system was determined using Eq. (6); when absorber inlet and outlet absolute humidity ($\omega_{abs,i}$, $\omega_{abs,o}$) are known in the absorber, the dehumidification effectiveness of the liquid desiccant system is determined as the ratio of the change in actual moisture content of air removing the system to the maximum possible change in the moisture content under a given operating condition [25].

The regeneration efficiency can be calculated using Eq. (7), given the regenerator inlet and outlet absolute humidity ($\omega_{reg,i}$, $\omega_{reg,o}$) states in the regenerator. The equilibrium humidity ratio (ω_{eq}) is determined using Eq. (8) [26], and the vapor pressure (p_s) required to determine can be calculated using Eq. (9) and the coefficients in Table 1.

$$\varepsilon_{abs} = \frac{\omega_{abs,i} - \omega_{abs,o}}{\omega_{abs,i} - \omega_{abs,eq}} \quad (6)$$

$$\varepsilon_{reg} = \frac{\omega_{reg,o} - \omega_{reg,i}}{\omega_{reg,eq} - \omega_{reg,i}} \quad (7)$$

$$\omega_{eq} = 0.622 \left(\frac{p_s}{101.325 - p_s} \right) \quad (8)$$

$$p_s = (a_0 + a_1 T_L + a_2 T_L^2) + (b_0 + b_1 T_L + b_2 T_L^2) C \quad (9)$$

$$+ (c_0 + c_1 T_L + c_2 T_L^2) C_L^2$$

3.3.1. Absorber

In this study, the model proposed by Chung and Luo was used to calculate the dehumidification effectiveness (Eq. (10)) [26]. L_i and G_i are the inlet mass flow rates of air and liquid, respectively, DBT_{G_i} and T_{L_i} denote the inlet air temperature and inlet liquid temperature, respectively. a is the surface-area-to-volume ratio of the packing, and Z is the packing height. π represents a function of the vapor pressure depression of the desiccant solution to the vapor pressure of pure water. Based on the experimental results of this model, approximately 80 % of the data points exhibited a deviation of ± 10 %, and the average error between the predicted and experimental data was approximately 9.96 %. Therefore, the simulation results of this study show an error of approximately 9.96 % [27].

$$\varepsilon_{abs} = \frac{\left(1 - \frac{0.024 \left(\frac{G_i}{L_i} \right)^{0.6} \exp \left(1.057 \frac{DBT_{G_i}}{T_{L_i}} \right)}{(aZ)^{-0.185} \pi^{0.638}} \right)}{\left(1 - \frac{0.195 \exp \left(0.615 \frac{DBT_{G_i}}{T_{L_i}} \right)}{\pi^{-21.498}} \right)} \quad (10)$$

The temperature of the solution entering the absorber must be cooled to within 15–30 °C to minimize the effects of heat generated during the dehumidification process. The absorber solution in the proposed system is cooled using water-side free cooling through a cooling tower, which is an easy way to reduce the cooling energy consumption. Considering the temperature limits of the coolant that can be produced by free cooling during the summer, this study set the target temperature and concentration of the solution used in the absorber to 20 °C and 30 %, respectively [28]. Moreover, the airflow rates of the cooling tower ($m_{ct,i,a}$) and absorber ($m_{abs,i,a}$) both equal

Table 1
Dehumidification and regeneration process constant [26].

Dehumidification process	a_0	a_1	a_2	b_0	b_1
	4.58208	−0.159174	0.0072594	−18.3816	0.5661
	b_2	c_0	c_1	c_2	
	−0.019314	21.312	−0.666	0.01332	
Regeneration process	a_0	a_1	a_2	b_0	b_1
	16.294	−0.8893	0.01927	74.3	−1.8035
	b_2	c_0	c_1	c_2	
	−0.01875	−226.4	7.49	−0.039	

$m_{HX1,c}$ due to the balanced flow setting of the coolant flow rate of HX 1 ($m_{HX1,c}$) with the cooling tower airflow ($m_{ct,i,a}$). The mass flow rates of the solution and air ($m_{abs,i,a}$, $m_{abs,i,s}$) used in the absorber are set to a liquid-to-gas ratio of 1.0 [29]; thus, the flow rates of the absorber solution ($m_{abs,i,s}$) and air ($m_{abs,i,a}$) are the same. The inlet air temperature ($T_{abs,i,a}$) and absolute humidity ($w_{abs,i,a}$) of the absorber unit are the same as the outdoor temperature (T_{oa}) and absolute humidity (w_{oa}) when outside air is directly introduced.

The dehumidification of air passing through the absorber unit yields the outlet humidity ratio of the processed air, which can be calculated using Eq. (11) by considering the difference between the inlet air humidity ratio and equilibrium humidity, as well as the dehumidification efficiency. Similarly, the outlet air temperature can be calculated using a comparable approach, as expressed in Eq. (12), by considering the difference between the inlet air temperature and inlet solution temperature, along with the effectiveness of the absorber. The description of the parameters indicated in each equation is as shown in Table 2.

$$w_{abs,o,a} = w_{abs,i,a} - \varepsilon_{abs} (w_{abs,i,a} - w_{eq,abs,i,s}) \quad (11)$$

$$T_{abs,o,a} = T_{abs,i,a} - \varepsilon_{abs} (T_{abs,i,a} - T_{abs,i,s}) \quad (12)$$

The difference between the inlet and outlet absolute humidities represents the change in absolute humidity and the amount of dehumidification (Eq. (13), Eq. (14)). The outlet air mass flow can be estimated from the difference between the estimated dehumidification mass flow and the inlet air mass flow (Eq. (15)).

$$w_{dehum} = w_{abs,i,a} - w_{abs,o,a} \quad (13)$$

$$m_{dehum} = m_{abs,i,a} (w_{abs,i,a} - w_{abs,o,a}) \quad (14)$$

$$m_{abs,o,a} = m_{abs,i,a} - m_{dehum} \quad (15)$$

After dehumidification, the outlet mass flow of the solution can be calculated as the sum of the dehumidified and inlet solution mass flows, as shown in Eq. (16). This is because the moisture content within the solution increases after dehumidification, leading to an increase in the mass flow.

$$m_{abs,o,s} = m_{dehum} + m_{abs,i,s} \quad (16)$$

The absorber outlet solution temperature can be derived from the concentration and enthalpy in the EES library. The outlet concentration of the solution is given by Eq. (17), which is determined by the proportionality relationship between the inlet and outlet solution mass flows and concentrations.

$$x_{abs,o,s} = \frac{m_{abs,i,s} x_{abs,i,s}}{m_{abs,o,s}} \quad (17)$$

The enthalpy of the solution can be obtained using the energy balance equation in Eq. (18). This equation relates the dehumidified mass flow, heat of vaporization of water, absorber inlet solution mass flow, absorber inlet solution enthalpy, and outlet solution mass flow.

$$h_{abs,o,s} = \frac{m_{dehum} h_{fg} + m_{abs,i,s} + h_{abs,i,s}}{m_{abs,o,s}} \quad (18)$$

After undergoing the dehumidification process, the solution becomes diluted and is directed towards the absorber sump. From

Table 2
Parameter of equation.

parameter	Explain	parameter	Explain
ε_{abs}	Absorber effectiveness	m_{dehum}	Dehumidification mass flow
G_i	Inlet gas flow rate	$m_{abs,o,a}$	Absorber outlet air mass flow
L_i	Inlet liquid flow rate	$m_{abs,o,s}$	Absorber outlet solution mass flow
DBT_{G_i}	Inlet air temperature	$x_{abs,o,s}$	Absorber outlet solution concentration
T_{L_i}	Inlet liquid temperature	$h_{abs,o,s}$	Absorber outlet solution enthalpy
π	Function of vapor pressure depression of the desiccant solution to the vapor pressure of pure water	$C_{min,HX2}$	Minimum capacitance rate of heat exchanger
a	surface area-to-volume ratio of packing	$CP_{sump,reg}$	Regenerator sump specific heat
Z	packing height	$CP_{sump,abs}$	absorber sump specific heat
$w_{abs,o,a}$	Absorber outlet air humidity ratio	$T_{HX2,o,To,abs}$	Outlet heat exchanger2 temperature (for absorber)
$w_{abs,i,a}$	Absorber inlet air humidity ratio	$T_{sump,reg}$	Regenerator sump temperature
$w_{eq,abs,i,s}$	Absorber inlet solution equilibrium humidity	ε_{HX2}	Heat exchanger 2 effectiveness
$T_{abs,o,a}$	Absorber outlet air temperature	$T_{sump,abs}$	Absorber sump temperature
$T_{abs,i,a}$	Absorber inlet air temperature	$m_{abs,i,s}$	Absorber inlet solution mass flow
$T_{abs,i,s}$	Absorber inlet solution temperature	$T_{HX2,o,To,reg}$	Outlet heat exchanger2 temperature (for regenerator)
w_{dehum}	Absolute humidity	$m_{reg,i,s}$	Regenerator inlet solution mass flow

there, the collected solution is transferred to heat exchanger 2 (HX2). Leveraging the hot, concentrated desiccant released from the regenerator, HX2 efficiently preheats the weak solution flowing into it. The use of HX2 in reduced thermal energy consumption for the regenerator and decreased cooling requirements for the absorber. The two solutions undergo heat exchange with each other. The efficiency of the heat exchanger is assumed to be 0.75, and the capacitance rate is determined by the mass flow and specific heat of the solution in the absorber and regenerator, as shown in Eq. (19). The specific heat of the solution collected in the sumps can be tracked using the temperature and concentration in the EES library. The temperature of the outgoing solution can be determined using Eqs. (20) and (21), based on the determined capacitance rate.

$$C_{min,HX2} = \min (m_{abs,i,s} CP_{sump,reg}, m_{reg,i,s} CP_{sump,abs}) \quad (19)$$

$$T_{HX2,o,To,abs} = T_{sump,reg} - \left(\frac{\varepsilon_{HX2} (C_{min,HX2} (T_{sump,reg} - T_{sump,abs}))}{m_{abs,i,s} CP_{sump,reg}} \right) \quad (20)$$

$$T_{HX2,o,To,reg} = T_{sump,abs} + \left(\frac{\varepsilon_{HX2} (C_{min,HX2} (T_{sump,reg} - T_{sump,abs}))}{m_{reg,i,s} CP_{sump,abs}} \right) \quad (21)$$

3.3.2. Regenerator

The absorber of the liquid desiccant system operates at a low solution temperature range to lower the vapor pressure and increase dehumidification performance, while the regenerator operates at a high solution temperature range of 40–80 °C to increase regeneration performance [30–32]. The outlet air condition of the regeneration unit can be calculated assuming an efficiency of 40 % for the regenerator. The inlet air temperature and humidity ratio of the regenerator are the same as those of the outside air, and the regeneration process of the regenerator unit is similar to that of the dehumidifier. This process can be considered as the inverse of the dehumidification process, which relies on the desorption and adsorption relationship between the solution and the air. The outlet temperature and humidity ratio of the regenerator are calculated using the temperature difference, equilibrium humidity ratio difference, and efficiency, as shown in Eqs. (22) and (23). The regenerated air flow rate can be calculated using Eq (24). This is derived from the product of the humidity ratio difference between the air entering and leaving the regenerator part and the air flow rate entering the regenerator part.

The air mass flow that exits after regeneration is determined using Eq. (25), which considers the increased water vapor mass flow due to regeneration and the inflow rate. The regenerator outlet solution conditions are as follows: The outlet solution mass flow is equal to the difference between the inlet solution mass flow and the increased water vapor mass flow after regeneration (Eq. (26)). The solution concentration can be derived from the proportional relationship between the inlet and outlet solution concentrations and their respective flow rates (Eq. (27)). The enthalpy is determined using the energy balance (Eq. (28)).

$$w_{reg,o,a} = w_{reg,i,a} + \varepsilon_{reg} (w_{eq,reg,i,s} - w_{reg,i,a}) \quad (22)$$

$$T_{reg,o,a} = T_{reg,i,a} + \varepsilon_{reg} (T_{reg,i,s} - T_{reg,i,a}) \quad (23)$$

$$m_{reg} = m_{reg,i,a} (w_{reg,o,a} - w_{reg,i,a}) \quad (24)$$

$$m_{reg,o,a} = m_{reg,i,a} + m_{reg} \quad (25)$$

$$m_{reg,o,s} = m_{reg,i,s} - m_{reg} \quad (26)$$

$$x_{reg,o,s} = \frac{m_{reg,i,s} x_{reg,i,s}}{m_{reg,o,s}} \quad (27)$$

$$h_{reg,o,s} = \frac{m_{reg,i,s} h_{reg,i,s} + m_{reg} + h_{fg}}{m_{reg,o,s}} \quad (28)$$

3.4. Cooling tower unit types

In this study, two types of cooling towers were utilized: Cooling tower 1 (CT1), which provided coolant for heat removal, and Cooling tower 2 (CT2), which cooled the liquid desiccant solution. Both cooling towers used the same type of tower with the same performance. For CT 1, the coolant flow rate and cooling tower airflow were set to a balanced flow, resulting in the same airflow and flow rate as the absorber, which can be calculated using Eq. (1). It was assumed that the inlet coolant temperature of CT 1 equaled $T_{HX1,o,c}$ without any heat loss. An approach temperature of 2 °C was used for the cooling tower analysis. The outlet coolant temperature of CT 1 can be determined using Eq. (29); based on this, the range was calculated using Eq. (30).

$$Approach_{ct} = T_{CT1,o,w} - wbt_{CT1,i,a} \quad (29)$$

$$Range_{ct} = T_{CT1,o,w} - T_{CT1,i,w} \quad (30)$$

To analyze the cooling tower (CT 2) for reducing the solution temperature, the following steps were performed. The inlet water

mass flow to CT 2 was assumed to be the same as that to CT 1, and the inlet air mass flow was calculated by assuming a liquid-to-gas ratio (LG) of 1 using Eq. (31). It was assumed that the water mass flowing in and out of the cooling tower circulated at a constant flow rate without any reduction.

$$m_{CT2,i,a} = m_{CT2,i,w} LG \quad (31)$$

The temperature of the inlet water equaled the outlet temperature after heat exchange with the solution at HX 3. As a result of this, the solution is cooled down to a target temperature of 20 °C at HX 3. The outlet water temperature of CT 2 can be determined using the wet-bulb temperature and approach, which is assumed to be the same as CT 1 at 2 °C (Eq. (32)). The range of CT 2 is calculated using Eq. (33), which computes the difference between the outlet and inlet temperatures of the coolant from CT 2.

$$T_{CT2,o,w} = wbt_{CT2,i,a} + Approach_{CT} \quad (32)$$

$$Range_{CT2} = T_{CT2,o,w} - T_{CT2,i,w} \quad (33)$$

3.5. Chiller unit

In this study, a chiller was used when the coolant leaving the cooling tower was not at a sufficiently low temperature to remove heat. This is because the hot and humid outside air in Southeast Asia has peak times during which it cannot be adequately dehumidified through the liquid desiccant process. However, by lowering the chilled-water temperature through dehumidification, the cooling energy used by the refrigeration system can be reduced compared to the chilled-water temperature from conventional free cooling. Various refrigerants are used in chiller systems; in this study, the widely used refrigerant R410A was selected. The chiller system comprises an evaporator, condenser, compressor, and expansion valve, with the evaporator being the key component in this study. Based on previous research, the efficiencies of the evaporator, condenser, and compressor were set to 75 %. This study focuses on determining the temperature of the evaporator using the efficiency of the evaporator and coolant that exits CT 1 and enters HX 1 (Eq. (34)).

$$T_{evap} = T_{sump,CT1,old} - \frac{T_{sump,CT1,old} - T_{HX1,i,c}}{\epsilon_{evap}} \quad (34)$$

To improve the energy efficiency, this study proposes a method for recovering the heat generated by the condenser during the cooling cycle. The temperature at which the condenser releases heat is determined by Eq. (35), which considers the solution temperature at HX2 when it flows to regenerator $T_{HX2,To,reg}$, the target temperature for heating the solution ($T_{reg,i,s,target}$) to 40 °C, and the condenser efficiency.

$$T_{cond} = T_{HX2,To,reg} + \left(\frac{(T_{reg,i,s,target} - T_{HX2,To,reg})}{\epsilon_{cond}} \right) \quad (35)$$

3.6. Fan and pump

3.6.1. Fan

The proposed system comprises four fans, including the dehumidification and regeneration sections of the liquid desiccant, as well as two fans for the cooling tower. The energy consumption of these fans can be determined using Eqs. (36) and (37), where Eq. (37) calculates the design fan power and air volume flow rate. The partial load ratio (PLR) represents the ratio of the real-time airflow rate to the design airflow rate. In this study, the PLR for each component was assumed to be 1 while assuming that the ratio of the real-time airflow rate to the design airflow rate was the same. The fan efficiency was set to 0.5, and the pressure rise for the absorber, regenerator, and cooling tower fans was set to 120 kPa based on the airflow rate [33].

$$fan_{design} = \left(\frac{V}{3600} \right) \frac{0.12}{0.5} \quad (36)$$

$$fan = (fan_{design} (0.0013 + 0.1470PLR + 0.9506PLR^2 - 0.0998PLR^3)) \quad (37)$$

3.6.2. Pump

In this study, seven pumps were used: the absorber pump, regenerator pump, absorber-to-regenerator pump, HX1 pump, HX2 pump, HX3 pump, and solution cooling tower pump. In this simulation, it is assumed that the pumps are constant-flow-rate pumps, and their output is determined using Eq. (38), which considers the fluid flow rate (V), a pump efficiency (η_{pump}) of 0.6, and a head (H) of 20 m [34].

$$pump = \frac{\rho g V H}{1000 \eta_{pump}} \quad (38)$$

3.7. Partial power usage efficiency (pPUE)

When assessing the thermal performance of a data center's cooling system, ASHRAE recommends using the power usage effectiveness (PUE) metric. However, PUE is a comprehensive concept that encompasses all electricity used in a data center, not just that of the IT servers. In this study, partial PUE (pPUE) was used to evaluate the efficiency of a cooling system using a liquid desiccant system.

The pPUE is defined as the total energy of the IT equipment and cooling energy within the boundary divided by the IT equipment energy within the boundary [35]. In this context, $w_{cooling}$ represents $IT_{cooling}$ load, which is the sum of the compressor power of the chiller and the total load of the fan and pump. The formula for calculating the pPUE is as follows Eq. (39):

$$pPUE = \frac{IT_{equipment\ power} + w_{cooling}}{IT_{equipment\ power}} \quad (39)$$

4. Simulation result and discussion

4.1. The process air dehumidification effect and cooling tower water temperature

Energy simulations were conducted under hot and humid conditions during the cooling period of the simulation year. Under these conditions, the sum of the dehumidification effect, the cooling tower outlet temperature, and the energy consumed were added to the results. Fig. 3 shows the change in the outlet air conditions passing through the absorber. The red square (T_{oa}) represents the outdoor air temperature and humidity ratio, and the blue square ($T_{abs,o,a}$) represents the temperature and humidity ratio of the dehumidified air.

Table 3 shows the average wet-bulb temperature and humidity ratio of the outdoor air before passing through the absorber in the

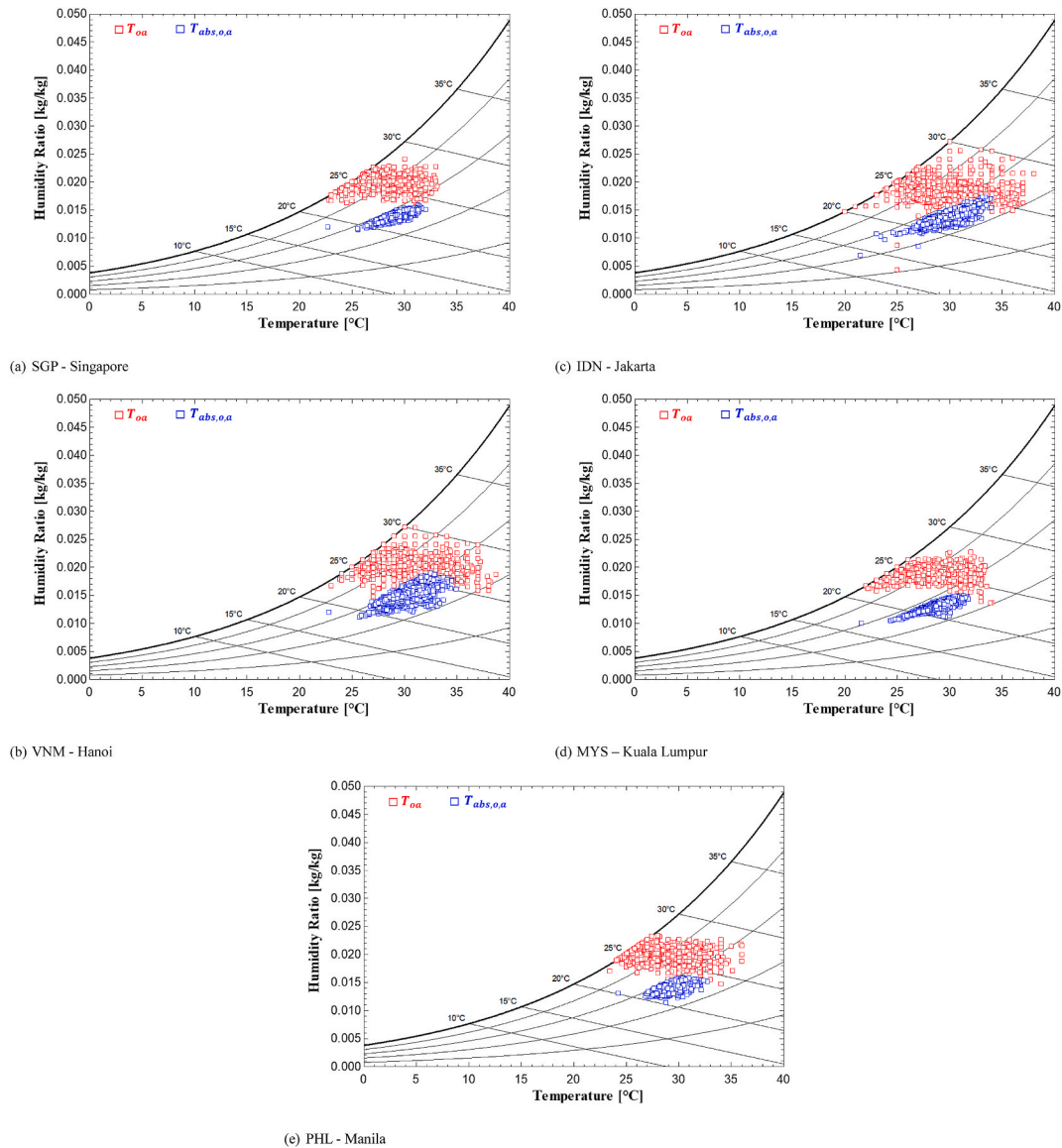


Fig. 3. Change in outdoor air temperature before and after liquid desiccant process in (a) Singapore (SGP - Singapore), (b) Vietnam (VNM - Hanoi), (c) Indonesia (IDN - Jakarta), (d) Malaysia (MYS - Kuala Lumpur), (e) the Philippines (PHL - Manila).

selected regions in Southeast Asia.

The humidity ratio of the hot and humid air passing through the absorber was confirmed to be reduced to an average of 0.0138 kg/kg. After the absorber process, the average wet-bulb temperatures in the five regions decreased from 25.69 to 22.08 °C. Fig. 4 and Table 4 show the variation in the cooling-tower outlet temperature after outdoor air dehumidification. This decrease in wet-bulb temperature confirmed, using a relational approach, that the average outlet temperature of the cooling tower in each region was 24.14, 25.05, 23.66, 23.26, and 24.29 °C respectively.

4.2. Cooling energy consumption and pPUE

The changes in the cooling energy usage and pPUE of a data center cooling system with the application of a liquid desiccant system is shown in Fig. 5. The cooling energy consumption in the reference system for each region was 46.05, 47.91, 44.39, 43.84, and 46.34 kW for Singapore (Singapore), Vietnam (Hanoi), Indonesia (Jakarta), Malaysia (Kuala Lumpur), and the Philippines (Manila), respectively. However, the proposed system with the application of the liquid desiccant system resulted in an average cooling energy consumption of 34.77 kW for Singapore (Singapore), 37.04 kW for Vietnam (Hanoi), 33.59 kW for Indonesia (Jakarta), 32.64 kW for Malaysia (Kuala Lumpur), and 35.11 kW for the Philippines (Manila), showing an average improvement of 24.26 %. This improvement was achieved by reducing the energy consumption of the refrigeration system by an average of 35.12 %.

The pPUE results used to assess the cooling energy efficiency of the data center are presented in Fig. 6.

When comparing the reference system with the proposed system, an average pPUE reduction of 5.82 % was observed across the five regions. The simulation results showed that the proposed system had increased energy consumption owing to the higher number of fans and pumps, as well as the increased airflow in the regeneration section. By improving the free-cooling performance of the cooling tower with dehumidified outdoor air, the proposed system achieved a lower coolant production, thereby reducing the usage of the compressor in the refrigeration system and enabling energy savings.

5. Conclusion

This study aims to achieve two main objectives. The first objective is to enhance the performance of free-cooling systems, and the second objective is the efficient energy consumption of these systems in hot and humid regions, particularly in Southeast Asia. To fulfill these goals, the proposed system combines a liquid-desiccant system with a free-cooling system.

To evaluate the efficiency and applicability of the liquid-desiccant-free cooling system in hot and humid climates, outdoor air data from five Southeast Asia regions, namely Singapore (Singapore), Vietnam (Hanoi), Indonesia (Jakarta), Malaysia (Kuala Lumpur), and the Philippines (Manila), were used. Simulation results demonstrated that the outdoor air, after passing through the liquid desiccant dehumidification section, achieved an average reduction of 42.79 % in humidity ratio. The decreased humidity facilitated heat exchange with water in the cooling tower, resulting in lower cooling-tower outlet temperatures. The calculated cooling tower inlet temperature was 25.6 °C, while the outlet water temperatures after free cooling in the reference system were 27.70, 28.52, 27.44, 26.96, and 27.84 °C for Singapore (Singapore), Vietnam (Hanoi), Indonesia (Jakarta), Malaysia (Kuala Lumpur), and the Philippines (Manila), respectively, while the improved outlet temperatures in each region were 24.14, 25.04, 23.66, 23.26, and 24.29 °C, respectively. By using dehumidified outdoor air and employing evaporative cooling with water in the cooling tower, the proposed system produces coolant at lower temperatures than the traditional free-cooling system, reducing the cooling load on the refrigeration system. Consequently, the energy consumption for data center cooling was reduced by an average of 26.02 %, leading to an average pPUE reduction of 6.11 % and achieving the two main objectives of this study. The integration of liquid-desiccant dehumidification with free cooling demonstrates the potential for energy savings in Southeast Asia.

With the increasing energy consumption of data centers, many engineers are considering the implementation of free-cooling systems for efficient energy usage. However, such systems demonstrate ideal efficiency under colder outdoor conditions, and face challenges in hot and humid climates, resulting in higher cooling energy consumption. The proposed liquid-desiccant free-cooling system aims to address the limitations of traditional free-cooling systems by reducing the cooling energy consumption and improving the free-cooling performance in hot and humid climates. Furthermore, this study was conducted using simulations to assess the feasibility of implementing a data center cooling system with liquid desiccation and to determine trends in energy savings. The simulation results obtained in this study will be validated later by comparing them with experimental operating results using the system that will be constructed.

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Table 3
Variation in humidity temperature and absolute humidity after dehumidification.

	SGP (Singapore)		VNM (Hanoi)		IDN (Jakarta)		MYS (Kuala Lumpur)		PHL (Manila)	
	T_{oa}	$T_{abs,o,a}$	T_{oa}	$T_{abs,o,a}$	T_{oa}	$T_{abs,o,a}$	T_{oa}	$T_{abs,o,a}$	T_{oa}	$T_{abs,o,a}$
Average wet-bulb temperature [°C]	25.70	22.14	26.52	23.05	25.44	21.66	24.96	21.26	25.84	22.29
Average humidity ratio [kg/kg]	0.0199	0.0139	0.0208	0.0149	0.0188	0.0131	0.0188	0.0129	0.0200	0.0141

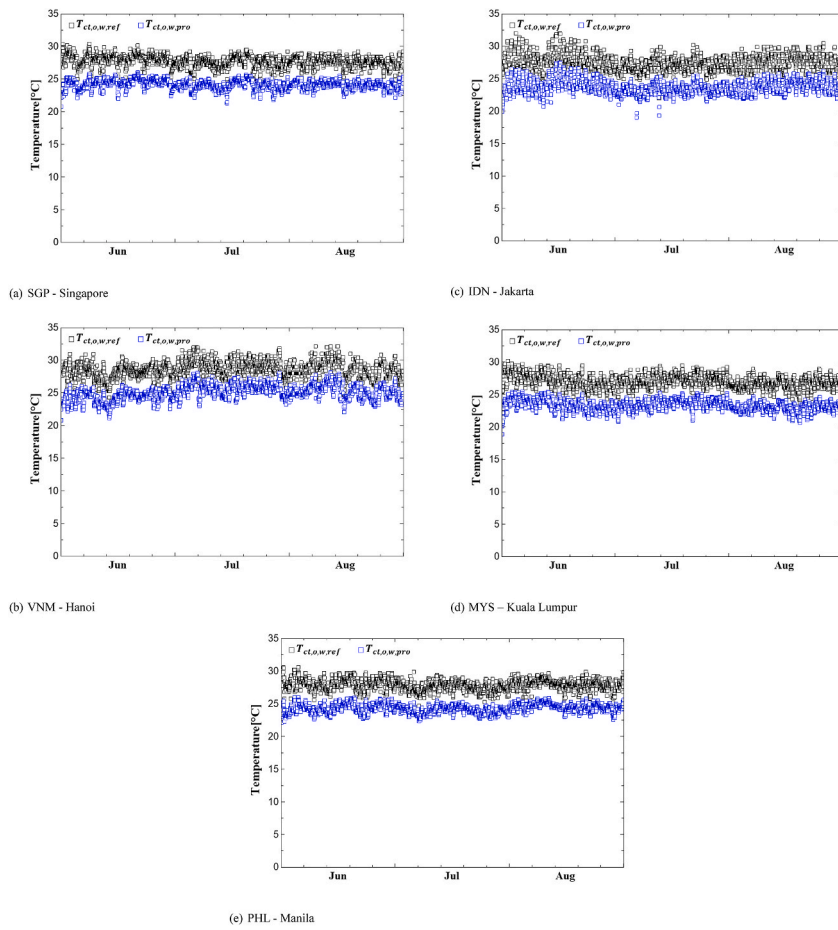


Fig. 4. Cooling tower outlet temperature variations in (a) SGP - Singapore, (b) VNM - Hanoi, (c) IDN - Jakarta, (d) MYS - Kuala Lumpur, (e) PHL - Manila.

Table 4

Average variation in cooling tower outlet temperature.

SGP (Singapore)		VNM (Hanoi)		IDN (Jakarta)		MYS (Kuala Lumpur)		PHL (Manila)	
$T_{ct,o,w,ref}$	$T_{ct,o,w,pro}$	$T_{ct,o,w,ref}$	$T_{ct,o,w,pro}$	$T_{ct,o,w,ref}$	$T_{ct,o,w,pro}$	$T_{ct,o,w,ref}$	$T_{ct,o,w,pro}$	$T_{ct,o,w,ref}$	$T_{ct,o,w,pro}$
27.70	24.14	28.52	25.05	27.44	23.66	26.96	23.26	27.84	24.29

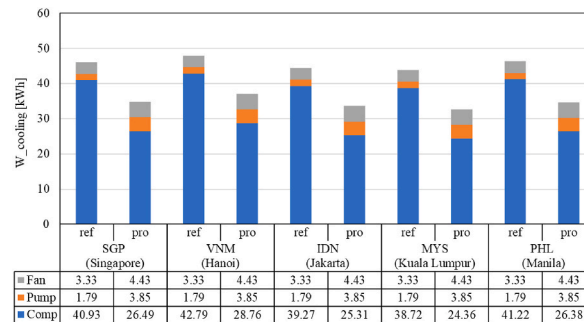


Fig. 5. Energy consumption for Southeast Asia region.

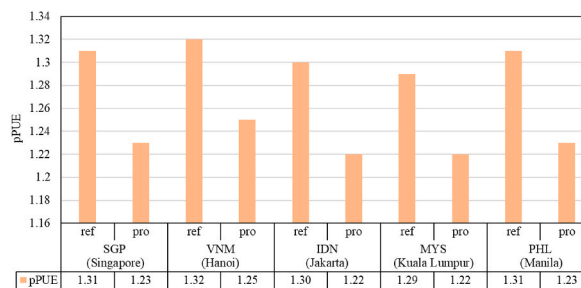


Fig. 6. pPUE analysis of Southeast Asia regions.

CRediT author statement

Hyo-Lim Park: Conceptualization, Methodology, Data curation, Writing original draft preparation. **Soo-Jin Lee:** Methodology, Validation. **Jae-Hee Lee:** Methodology, Validation. **Jae-Weon Jeong:** Supervision, Validation, Reviewing and Editing.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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