



# Operational optimization of a hybrid desiccant cooling system for building air conditioning system

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## Abstract

Hybrid desiccant cooling system has a higher electric coefficient of performance than that of electric heat pump system because evaporative cooler highly contributes to diminishing the cooling load. An abundant waste heat must be secured because of a thermal requirement for regenerating a desiccant rotor. However, the cooling capability of the evaporative cooler is not necessarily proportional to the amount of supplied heat. The optimized regeneration temperature of desiccant cooling system is acquired to maximize the cooling performance of the evaporative cooler. The hybrid desiccant cooling system model developed in our previous study was simulated to determine the optimized operating conditions under actual climate conditions of the summer season worldwide. The optimal value of temperature for desiccant rotor regeneration and the process air flow rate, those can minimize the electric power consumption, are obtained as a based on the outdoor absolute humidity and outdoor temperature. The effect of the room inlet temperature of the process air on the consumption rate of regeneration heat and electric power consumption is captured. The characteristics of the hybrid desiccant cooling system at the minimal and optimal regeneration temperatures were investigated and compared.

**Keywords** Performance optimization · Numerical modeling · Hybrid desiccant cooling system · Actual climate conditions · Sensitivity analysis

## Abbreviations

$A_s$	Area of interfacial contact [ $\text{m}^2$ ]	$\varepsilon$	Adsorption potential [ $\text{kJ} \cdot \text{kmol}^{-1}$ ]
$\Delta H_{evap}$	Enthalpy of water vaporization [ $\text{kJ} \cdot \text{kg}^{-1}$ ]	$\eta$	Isentropic efficiency [-]
$h_1$	Enthalpy of refrigerant at inlet of compressor [ $\text{kJ} \cdot \text{kg}^{-1}$ ]	$\theta$	Water adsorption amount [ $\text{kg} \cdot \text{kg}^{-1}$ ]
$h_2$	Enthalpy of refrigerant at outlet of compressor [ $\text{kJ} \cdot \text{kg}^{-1}$ ]	$\rho$	Density [ $\text{kg} \cdot \text{m}^3$ ]
$h_m$	Mass transfer coefficient of silica-gel [ $\text{kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$ ]	<i>amb</i>	Ambient
$\dot{m}$	Mass flow rate [ $\text{kg} \cdot \text{hr}^{-1}$ ]	<i>cooled</i>	Cooled room condition
$P$	Electric power [kW]	<i>EHP</i>	Electric heat pump
$\dot{Q}_{evap}$	Rate of water vaporization heat [kW]	<i>evap</i>	Evaporation
$R$	Gas constant [ $\text{kJ} \cdot \text{kmol}^{-1} \cdot \text{K}^{-1}$ ]	<i>heat</i>	Heat generation
$RH$	Relative Humidity [%]	<i>in</i>	Inlet
$U_{air}$	Air flow rate [ $\text{m}^3 \cdot \text{s}^{-1}$ ]	<i>out</i>	Outlet
$x$	Absolute humidity of air [ $\text{kg} \cdot \text{kg}^{-1}$ ]	<i>regen</i>	Regeneration
$Z$	Regeneration temperature [ $^{\circ}\text{C}$ ]		

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## 1 Introduction

The cooling demand of commercial and residential buildings air conditioning has increased three-fold from 1990s to 2018, as reported by the International Energy Agency [1]. The energy usage of the data center has grown from 153 terawatt hours in 2005 to 205 terawatt hours by 2018 [2], 30–40% of which is used to cool the server rooms in data centers [3]. Air conditioning system need to be more

efficient to save an electrical energy and reduce a carbon emission [4]. Electric heat pumps (EHPs) are air conditioning systems mostly installed in small residential and commercial buildings. The EHP system consumes a significant amount of electric energy in the summer. The reduction in electric power consumption in the EHP system can greatly diminish the peak power consumption demand in the summer [5–7]. Because the COP of a single-stage EHP has reached a limitation, diverse air conditioning system is suggested which is make use of water vaporization enthalpy. Water evaporation heat applied cooling system has advantage of not using refrigerant. A hybrid desiccant cooling system is a promising alternative cooling system for small residential and commercial buildings that make use of water vaporization enthalpy.

A solid desiccant cooling system expenses approximately two third of the electric power usage of an EHP [8, 9]. The desiccant rotor air conditioning system is composed of a desiccant rotor, heat exchangers, a direct evaporative cooler or an indirect evaporative cooler. If we can use the waste heat of distributed power generation systems such as fuel cell systems or solar thermal collectors, the thermally driven cooling system can be an effective air conditioning system that saves electric energy and increases energy efficiency. Thermally driven cooling systems with solid desiccant rotors and water evaporative coolers are typically installed in a small cooling space for commercial and residential buildings [10]. Because a hybrid desiccant cooling system is composed with few component within air duct, the system can installed in small space [11–13].

The evaporation methods for desiccant cooling system are largely categorized into two methods: direct evaporative desiccant cooling (DEC) system and indirect evaporative desiccant cooling (IEC) system. Both DEC and IEC systems require many thermal energy sources to obtain a satisfactory cooling ability and have insufficient cooling ability in extremely humid and hot climates, while hybrid desiccant cooling (HDC) has abundant cooling ability [14]. The HDC system is composed of a desiccant rotor, a direct and indirect evaporative cooler, and an EHP system. The HDC system has the advantages of both IEC system and EHP system because it can utilize thermal energy and supplement a cooling ability with an EHP in case additional cooling is required [10]. A HDC system has several advantages [13, 14]: 1) It can have cooling capacity with moderate electrical energy usage by using waste heat; 2) It can manipulate the humidity of inlet air by adjusting a water evaporation rate without cooling an air until dew point; 3) Its low noise level and small occupancy promote deployment compared to the liquid absorption chiller.

There are several studies on HDC systems. Numerical model of a hybrid desiccant cooling system is developed by Jia et al. [15]. They present that HDC could diminish

the electric power usage by up to 38% compared to electric heat pump system. Luo et al. [16] investigates the performance of hybrid desiccant cooling system at the humid and hot region of Taiwan by long-term experiments. The performance of the HDC system in the extremely humid and hot climates of northern India was numerically and experimentally captured by Jani et al. [17]. They investigated HDC system characteristics such as the desiccant rotor outlet temperature, dehumidifier effectiveness, and COP by varying humidity, temperature, and air flow rate. Jani et al. [18] analyzed the HDC system performance by experimentally investigating the effect of operational parameters. They investigated the system performance, including the COP, dehumidification efficiency, and indoor air supply temperature, by increasing the outdoor temperature to 31 °C. Heidari et al. [8] proposed a hybrid desiccant cooling system that contained an ejector. Their proposed system can supply comfortable conditions at humid climate conditions with reductions in both electric consumption and CO<sub>2</sub> emissions. Zeng et al. [19] developed a model and simulated a solar thermal driven hybrid desiccant cooling system. They figured out the optimized ratio of fresh air and return air flow rate. Fatouh et al. [20] experimentally analyzed the HDC system and conventional air conditioning system in hot and humid climate conditions by operating the air mixing ratio and humidity. Hwang et al. [21] analyzed the hybrid desiccant cooling system of thermal and electric energy consumption characteristics.

Based on a literature survey, few studies have investigated the HDC characteristics under extremely hot and humid conditions. Nie et al. [22] designed an electric heat pump integrated with a solid desiccant cooling system and the physical system was experimentally measured. The model was validated and analyzed at outdoor temperatures of 21–38 °C. G. Angrisani et al. [23] experimentally analyzed the performance of the HDC system at outdoor temperatures of 30 °C and 35 °C. Sheng et al. [24] experimentally analyzed the performance of a HDC system heated by a high-temperature EHP condenser without an additional heat supply at outdoor temperatures of 28–40 °C. Henning et al. [25] simulated and compared various designs of HDC systems, and a selected pilot system was installed and obtained electricity savings of more than 30% at the outdoor temperature of 35 °C. Tu et al. [26] developed a two-stage HDC system model and validated the system performance under the Beijing summer condition of a maximum outdoor temperature of 40 °C. Panaras et al. [27] developed a desiccant cooling system model, which is validated by desiccant wheel experimental results. The model is simulated to identify the effects of the parameters: flow rate, humidity, and outdoor temperature of 32–37 °C. These studies only focused on capturing the cooling performance of HDC systems at outdoor temperatures

below 40 °C, while the world climate conditions are becoming increasingly hotter [28].

There are some literatures that offering an optimized operating conditions of evaporative cooler desiccant cooling system. Chung et al. [29] optimized a rotation speed (50–250 s/rev) and size (0.01–0.2 m) of the desiccant wheel at various regeneration temperature (50–150 °C). Ahmed et al. [30] conducted an experiment and simulation of desiccant rotor performance for optimizing the air flow rate, rotor thickness, rotor distribution ratio, rotor porosity, and rotor rotation speed. Goldworthy and Whilte [31] optimized a wheel rotation speed and air velocity of indirect evaporative cooler desiccant cooling system. These literatures only focused on the optimization of the desiccant cooling system. There is no previous research that established the optimal operating conditions for HDC system balancing an electrical and thermal energy consumption.

When the HDC system is used for the residential air conditioning system, the system operates to meet the demand cooling load by manipulating the cooling rate of indirect evaporative cooler and EHP cooler respectively. In order EHP to significantly reduce the electrical consumption, the desiccant rotor should be maximally operated under massive thermal demand. Because plenty of thermal energy is consumed by regenerating the desiccant rotor, abundant heat must be secured. If there is no applicable waste heat nearby, the maximum regeneration temperature operating desiccant rotor strategy is inefficient. Even though plenty amount of heat is secured, the cooling performance of the evaporative cooler is not necessarily proportional to the amount of supplied heat. Because high temperature of regeneration increases an air temperature as well as dehumidification. The optimized regeneration temperature of desiccant rotor should be established to maximize the cooling performance of the water evaporative cooler, and to prevent overly thermal supply for a desiccant rotor.

The objective of this study is to identify the optimal operating conditions of the HDC system under actual climate conditions in the summer season worldwide. To capture the optimal operating conditions, a one-dimensional steady-state model of a HDC system developed in our previous study was utilized for the simulation [32]. The HDC system model is simulated to optimize the operating conditions under various climate conditions, which represent the actual climate conditions of cities worldwide and covers the extreme climate conditions of a maximum outdoor temperature of 50 °C. The optimal value of both regeneration temperature and process air flow rate, which can minimize the electric power consumption, has been obtained as a polynomial regression equation based on the outdoor humidity and outdoor temperature. The effect of the temperature of supply air into the room on the consumption rate of regeneration heat and electric power consumption is captured. The consumption

rate of regeneration heat, electric power consumption, room supply temperature of the process air, process air flow rate, thermal COP, electric COP, and total COP at the minimal regeneration temperature and at the optimal regeneration temperature were investigated and compared.

## 2 Hybrid desiccant cooling system

The HDC system is composed of a direct and an indirect evaporative cooler, a desiccant rotor, a heat exchanger, and an EHP. Figure 1(a) is the schematic diagram of the HDC system [32]. Before entering the desiccant rotor, the process air from the indoor room is added up with outdoor air (1 → 2) to supplement the extracted air. The process air is heated and dehumidified throughout the desiccant rotor, which is filled with silica gel (2 → 3), which considerably facilitates evaporative cooling. Then, the process air is cooled throughout the direct evaporative cooler (3 → 5). 30% of the air is extracted in the indirect evaporative cooler and exhausted (7). When both indirect evaporative cooler and directive evaporative cooler have deficient cooling performance, the EHP operates to satisfy the target cooling load (5 → 6). During the process air dehumidification (2 → 3), water molecules of the process air are removed by silica gel. To continuously remove water from the process air through the desiccant rotor, silica gel must be dehumidified, which is referred to as the regeneration process. The outdoor air is slightly heated throughout the condenser of the EHP (8 → 9) and mainly heated at the heat exchanger by an external source to reach a target regeneration temperature (9 → 10). Finally, the regeneration high temperature air enters the desiccant rotor to dehumidify the rotor (10 → 11).

### 2.1 Hybrid desiccant cooling system model

The HDC system is composed of desiccant rotor, direct and indirect evaporative cooler, EHP (evaporator, condenser), heat exchanger and room thermal model. The system model is developed in 0-D steady state that each component simulates outlet control volume parameter of temperature, flow rate, humidity, and enthalpy. The simplified model is described as follows, and the detailed modeling is described in previous work [32].

#### 2.1.1 Desiccant rotor model

Desiccant rotor model is developed in a simplified regression equation and validated with experiment results from reference Nia [33]. A water adsorption amount ( $\theta$ ) in a silica-gel is acquired by adsorption potential ( $\epsilon$ ) by the Polanyi theory as follows:



$$(x_{room} - x_{cooled}) \cdot m_{air} = \int (x_{in} - x_{out}) \cdot \dot{m}_{air} dt, \quad (6)$$

where  $x_{room}$ ,  $x_{cooled}$  are the absolute humidity of room before air conditioning and after air conditioning cooled room, respectively.  $x_{in}$ ,  $x_{out}$  are absolute humidity of inlet air by HDC system and outlet air of room. The temperature of air conditioned room is balance of the energy conservation equation as follows:

$$\dot{m}_{air} h_{in} - \dot{m}_{air} h_{out} + \dot{Q}_{heat} + \dot{Q}_{amb} = 0, \quad (7)$$

where  $h_{in}$ ,  $h_{out}$  are inlet and outlet enthalpy of air, and  $\dot{Q}_{gen}$  is the heat generation rate of the room and  $\dot{Q}_{amb}$  is the natural heat transfer rate by ambient temperature.

Thermal, electrical, and total COPs are determined as follows [32]:

$$COP_{th} = \frac{\dot{Q}_{cool}}{\dot{Q}_{reg}} \quad (8)$$

$$COP_{elec} = \frac{\dot{Q}_{cool}}{P_{fan} + P_{comp}} \quad (9)$$

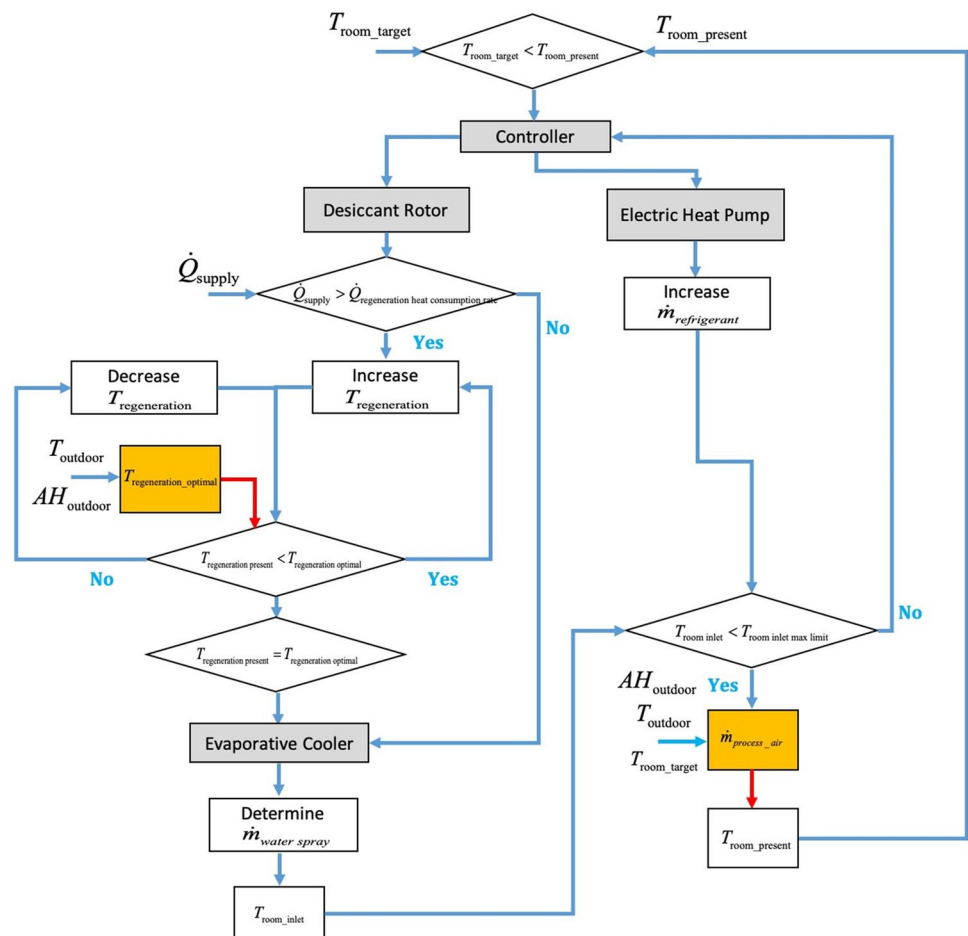
$$COP_{tot} = \frac{\dot{Q}_{cool}}{\dot{Q}_{reg} + P_{comp} + P_{fan}} \quad (10)$$

### 3 Simulation results

#### 3.1 Parameters for operational optimization

The HDC system is composed of three cooling devices: a direct evaporative cooler, an indirect evaporative cooler, and an EHP. Figure 2 shows the HDC system control algorithm. The regeneration temperature is controlled to rise until the optimal regeneration temperature when a room temperature is not reached a target temperature of 24 °C. When sufficient regeneration heat is not secured and needs a more cooling capacity, the EHP should be operated to supplement the cooling performance because of the short cooling performance of indirect evaporative cooler. In the opposite cases, the evaporative cooler can fully achieve the target cooling performance, which can save the electric power consumption by the HDC system, since the EHP does not need to be

Fig. 2 Control algorithm of the HDC system



**Table 1** Parameters for the operational optimization

Parameter	Unit	Value		
Outdoor temperature	[°C]	30	40	50
Outdoor humidity]	[%]	13, 46, 79	6, 16.5, 27	4, 7, 9
Regeneration temperature	[°C]	50, 60, 70, 80, 90	50, 60, 70, 80, 90	57, 60, 70, 80, 90
Supply temperature	[°C]	9.5, 10.5, 11.5, 12.5	9.5, 10.5, 11.5, 12.5	9.5, 10.5, 11.5, 12.5
Supply air flow rate	[kg/hr]	970~1230	1350~1700	1720~2180
Target temperature in indoor room	[°C]	24	24	24
Target humidity in indoor room	[%]		30–40	

operated. In severe hot and humid conditions, although there are sufficient external heat sources, HDC cannot satisfy the target cooling performance without EHP operation.

In this study, the optimal operating conditions of the HDC system to maximize the electric COP under various ambient conditions are studied. The dominant operating parameters of HDC system such as the regeneration temperature, room inlet temperature of the process air, and process air flow rate were selected and used for the sensitivity analysis. Table 1 shows the values of operating parameters: regeneration temperature, room inlet temperature of the process air, and process air flow rate. The regeneration temperature and room inlet temperature of the process air are 50–90 °C and 9.5–12.5 °C, respectively. The process air flow rate is 970–2180 kg/hr. The minimal process air flow rate is determined by the room inlet temperature of the process air. The combination of the process air flow rate and supply temperature of air into the room is obtained to maintain the indoor room at 24 °C.

### 3.2 Optimization of a regeneration temperature

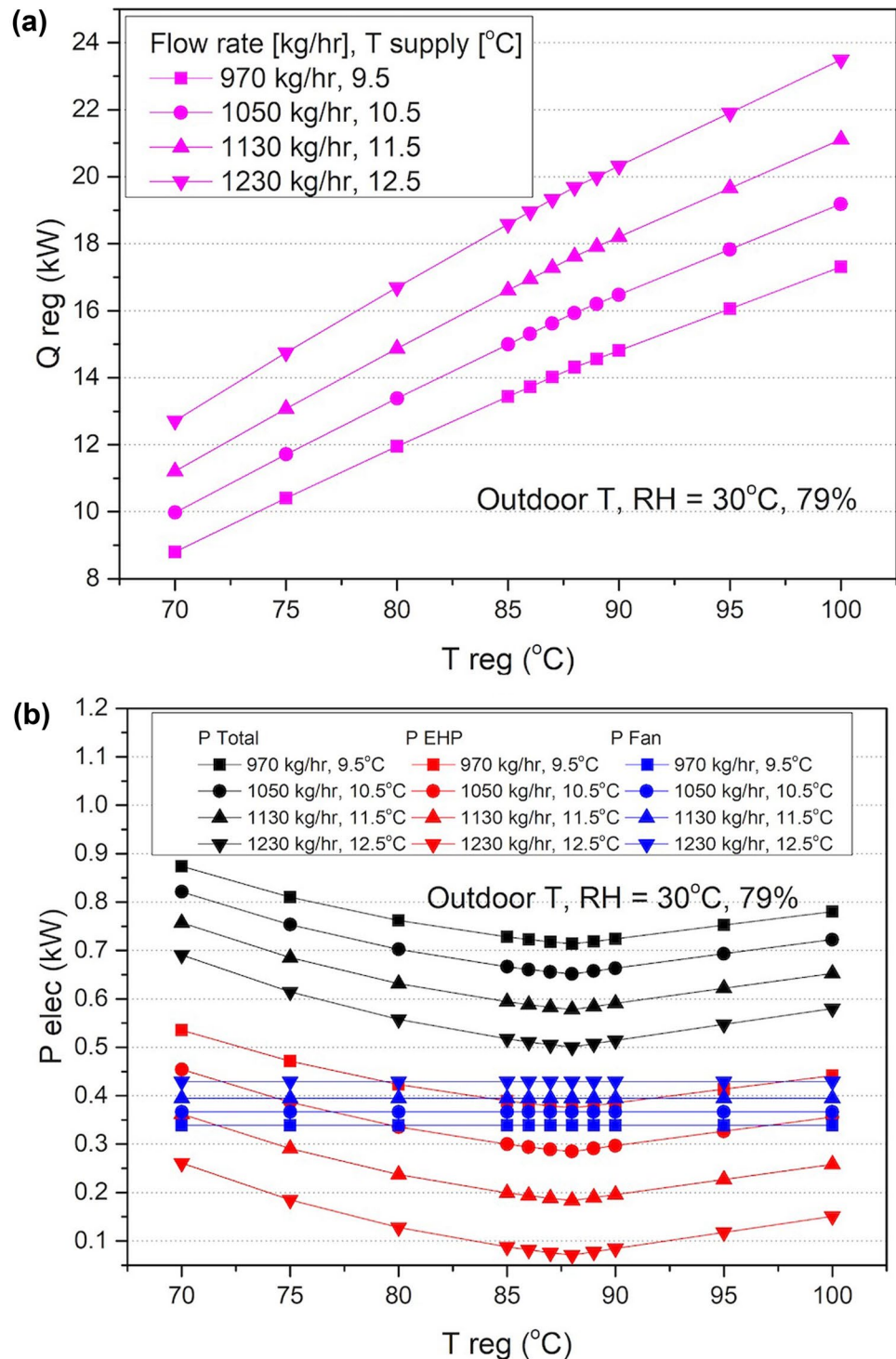
The regeneration temperature is the most dominant factor in determining the dehumidification rate by the desiccant rotor. A high regeneration temperature increases the drying performance of the rotor and cause low humidity of a process air, which low humidity air enhances the cooling performance of the direct and indirect evaporative cooler. Too high temperature of the regeneration makes an overheating of desiccant rotor and process air that can worsen a cooling performance of HDC system. The optimized regeneration temperature is figured out by the simulation that can maximize a performance of desiccant rotor and evaporative cooler. The effects of the regeneration temperature, room inlet temperature of the process air, and process air flow rate on the consumption rate of regeneration heat and electric power consumption under an outdoor temperature of 30 °C and a relative humidity of 79% are presented in Fig. 3(a) and (b), respectively. As expected, the consumption rate of regeneration heat increased with increasing regeneration temperature. A decrease in room supply temperature of the process air results in an increase in consumption rate of regeneration heat because the

regeneration air stream can be more heated by flowing through the condenser. The EHP electric power consumption of the HDC system is minimized at the regeneration temperature of 88 °C. Although the magnitudes of the fan power consumption are generally higher than those of the EHP power consumption, the difference in EHP power consumption at various room supply temperatures of the process air is much higher than that of the fan power consumption. Consequently, the total power consumption of the HDC system highly depends on the variations in the EHP power consumption. The variations in the room inlet temperature of the process air from 9.5 °C to 12.5 °C have no effect on the optimal regeneration temperature. Figures 4(a) and (b) present the changes in consumption rate of regeneration heat and electric power consumption with varying process air flow rate, room inlet temperature of the process air, and regeneration temperature under various ambient conditions. As shown in Fig. 4(b), there is an optimal regeneration temperature according to outdoor temperature and humidity condition. The highest regeneration temperature is 88 °C when the outdoor temperature is 30 °C and the outdoor relative humidity is 79%. The lowest optimal regeneration temperature is 70 °C when the outdoor temperature is 30 °C and the outdoor relative humidity is 13%. As shown in Figs. 3(b) and 4(b), when the regeneration temperature is too high over the optimized temperature, excessed heating to the desiccant rotor induces more electric power usage of the EHP system because the positive effect of the dehumidification of the desiccant rotor on water evaporative cooling becomes saturated, and the temperature increase of the desiccant rotor negatively affects the cooling load. The optimal regeneration temperatures of the HDC system under various outdoor temperature and outdoor humidity are presented in Table 2. The regeneration temperature regression by the outdoor temperature and the outdoor absolute humidity based on a nonlinear polynomial is conducted as follows:

$$Z = z_0 + aT_{outdoor} + bx_{outdoor} + cT_{outdoor}^2 + dx_{outdoor}^2 + fT_{outdoor}x_{outdoor} \quad (11)$$

where  $z_0, a, b, c, d, f$  are coefficient;  $T_{outdoor}, x_{outdoor}$  are the outdoor temperature and outdoor absolute humidity, respectively. The values for the coefficients in Eq. 1 are presented in Table 3.

**Fig. 3** Variations in **a** consumption rate of regeneration heat and **b** electric power consumption with varying process air flow rate, room inlet temperature of the process air, and regeneration temperature at the outdoor temperature of 30 °C, relative humidity of 79%

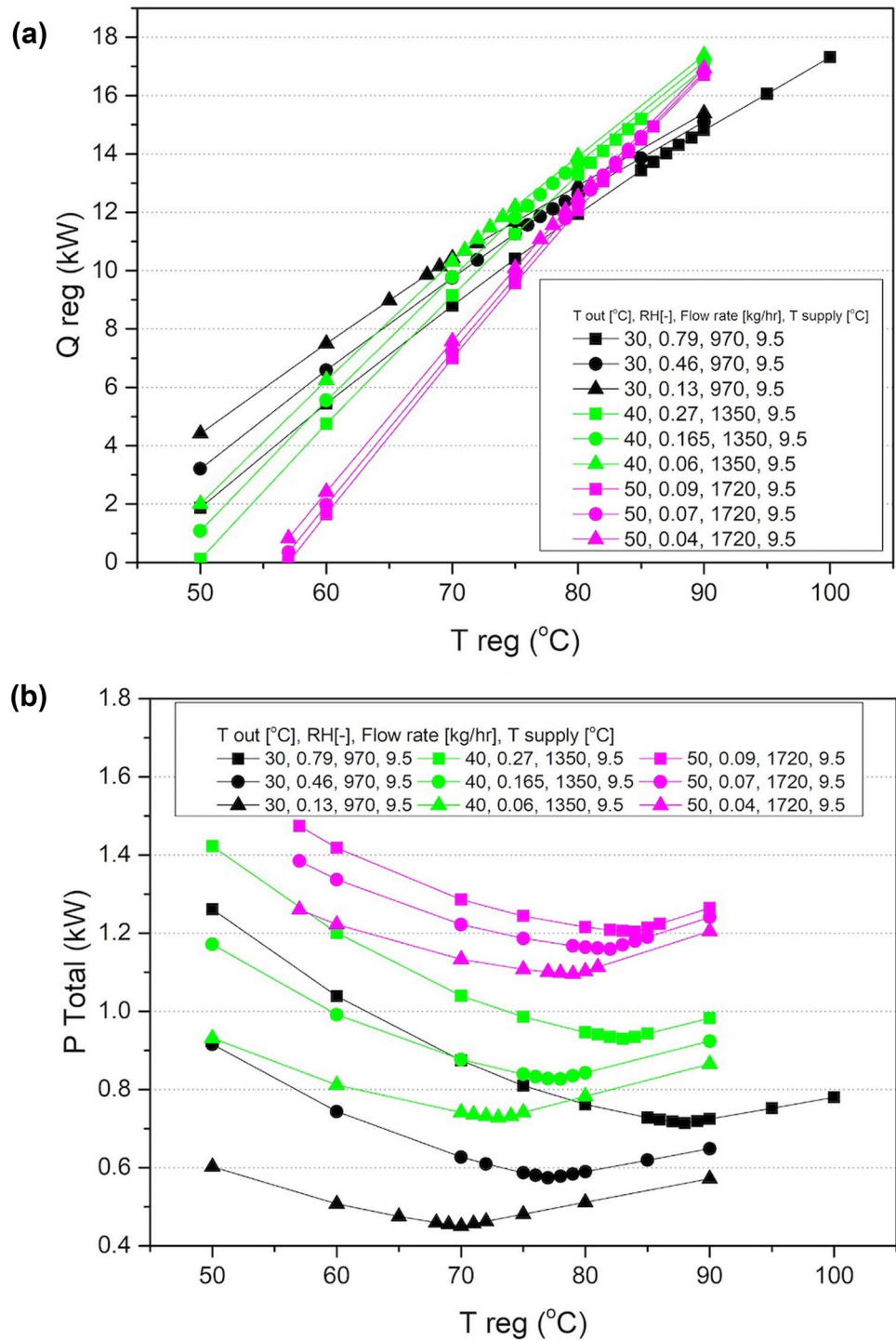


### 3.3 Optimization of the process air flow rate

The air flow rate of the HDC system depends on the supply temperature into to room and outdoor conditions. When the temperature of the supply air into room is set to 9.5 °C by supplementing the EHP operations, the air flow rate is a function of only the outdoor temperature. When the HDC

system is operated at the minimal regeneration temperature, the optimal process air flow rate can be calculated by an outdoor temperature, room inlet temperature, as shown in Table 4. When the outdoor temperature increases, the optimal process air flow rate increases because of the increase in cooling load. Meanwhile, when the HDC system is operated at the optimal regeneration temperature, the optimal process

**Fig. 4** Variations in **a** consumption rate of regeneration heat and **b** electric power consumption with varying process air flow rate, room inlet temperature of the process air, and regeneration temperature under various ambient conditions



air flow rate is determined by the supply temperature into the room, outdoor temperature, and outdoor absolute humidity, as presented in Table 5. The room inlet temperature of the process air is affected by both outdoor temperature and outdoor absolute humidity. Consequently, the optimal process

air flow rate is determined by the outdoor temperature and outdoor absolute humidity. Simulation results are used to obtain the correlations for the optimized process air flow rate at the optimal regeneration temperature based on the outdoor temperature and outdoor absolute humidity as follows:



**Table 2** Optimal regeneration temperature under various ambient conditions

T outdoor [°C]	RH outdoor [%]	AH outdoor [kg/kg]	T reg [°C]	Regression results [°C]	Mean squared error
30	13.0	0.00340	70	69.85	1.5e-3
30	46.0	0.01219	77	77.02	3.5e-5
30	79.0	0.02124	88	87.82	2.1e-3
35	9.0	0.00312	71	71.23	3.6e-3
35	31.0	0.01087	78	78.26	4.5e-3
35	53.0	0.01882	88	88.11	8.2e-4
40	6.0	0.00273	73	73.04	1.2e-4
40	16.5	0.00756	78	77.64	8.6e-3
40	27.0	0.01246	83	83.32	6.9e-3
45	8.0	0.00474	78	77.77	3.4e-3
45	11.0	0.00654	80	79.75	4.3e-3
45	14.0	0.00835	82	81.87	1.1e-3
50	4.0	0.00305	79	79.34	7.8e-3
50	7.0	0.00535	82	82.02	2.7e-5
50	9.0	0.00690	84	83.94	2.2e-4

$$Z = z_0 + AT_{outdoor} + Bx_{outdoor} + CT_{outdoor}^2 + Dx_{outdoor}^2 + FT_{outdoor}x_{outdoor} \quad (12)$$

where  $z_0, A, B, C, D, F$  are coefficients, whose values are presented in Table 6.

### 3.4 Sensitivity analysis with the room inlet temperature of the process air

Figure 5(a) and (b) present the changes in consumption rate of regeneration heat and electric power consumption when the supply temperature into room changes at the optimal regeneration temperature (red) and minimal regeneration temperature (blue). Increasing the supply temperature into room increases the consumption rate of regeneration heat because decreasing the condenser operation diminishes the heating magnitude to the regeneration air stream. Increasing the regeneration temperature increases the consumption rate of regeneration heat. At the minimal regeneration heat, the thermal consumption rate for regeneration decreases

with increasing outdoor temperature and outdoor relative humidity because of a thermal assisted by EHP condenser operation. While, the low EHP supplement by HDC operating in optimized regeneration temperature makes the system consume a lot of heat consumption and it proportionally increases with outdoor condition of temperature and humidity. As shown in Fig. 5(b), regardless of the regeneration temperature, electric power consumption of HDC system increases at the outdoor condition of temperature and humidity rise. The increase in supply temperature into room at the minimal regeneration temperature condition cause the air flow rate rise and it induces the increase of thermal consumption for regeneration. While, at the optimal regeneration temperature, the HDC system consumes less electric power as increase of the supply temperature into the

**Table 3** Coefficients of the regeneration temperature regression Eq. (8)

Constants	Value
$z_0$	73.22263
$a$	-0.53653
$b$	-261.949
$c$	0.0117
$d$	21,215.28
$f$	24.89356

**Table 4** Optimal process air flow rate with varying outdoor temperature and room inlet temperature of the process air at the minimal regeneration temperature

T outdoor [°C]	T supply [°C]	Flow rate [kg/hr]
30	9.5	970
30	10.5	1050
30	11.5	1130
40	9.5	1350
40	10.5	1450
40	11.5	1560
50	9.5	1720
50	10.5	1850
50	11.5	2000

**Table 5** Optimal process air flow rate with varying outdoor temperature and outdoor absolute humidity at the optimal regeneration temperature

T outdoor [°C]	RH outdoor [%]	AH [kg/kg]	T supply [°C]	Air flow rate [kg/hr]	Regression results [kg/hr]	Mean squared error
30	13.0	0.00340	10.56	1050	1049.57	0.01
30	46.0	0.01219	11.72	1160	1161.61	0.2
30	79.0	0.02124	12.5	1230	1223.45	2.9
35	9.0	0.00312	10.92	1290	1287.75	0.3
35	31.0	0.01087	11.97	1400	1402.97	0.6
35	53.0	0.01882	12.5	1470	1479.77	6.4
40	6.0	0.00273	11.25	1540	1539.72	5.1e-3
40	16.5	0.00756	11.93	1620	1625.08	1.7
40	27.0	0.01246	12.5	1700	1695.97	1.1
45	8.0	0.00474	11.96	1850	1856.12	2.5
45	11.0	0.00654	12.22	1890	1890.15	1.5e-3
45	14.0	0.00835	12.48	1940	1922.21	21.1
50	4.0	0.00305	12.15	2110	2115.12	1.7
50	7.0	0.00535	12.49	2180	2164.63	15.7
50	9.0	0.00690	12.5	2180	2195.88	16.8

room because the cooling performance of the evaporative cooler is maximized and the EHP supplement is not much necessary for high supply temperature.

### 3.5 System optimization

Figure 6(a), (b) and (c), present the regeneration temperature, supply temperature of air into the room, and process air flow rate at the minimal regeneration temperature and optimal regeneration temperature, respectively. The minimal regeneration temperature is set to 50 °C at outdoor temperatures of 30 °C and 40 °C. In the condition of outdoor temperature 50 °C, the 60 °C is the minimal regeneration temperature because there should be a temperature gradient between the temperature of the regeneration air and the regeneration temperature. At the minimal regeneration temperature, the room inlet temperature of the process air is set to 9.5 °C, which is generally used for residential air conditioning systems. As mentioned above, the process air flow rate only depends on the outdoor temperature unrelated to humidity when the HDC system is operating at the minimal regeneration temperature. The process air flow rate is determined to

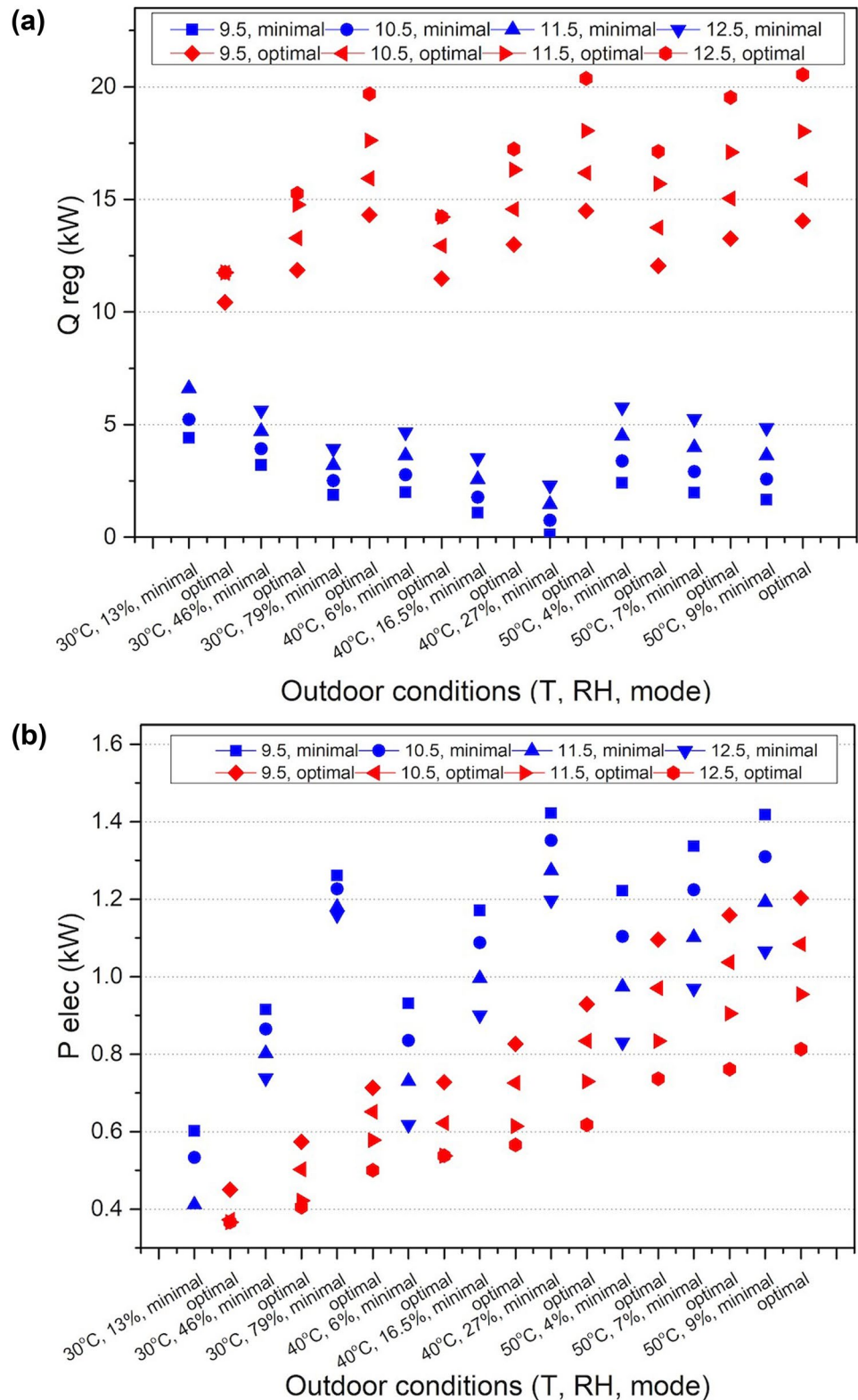
be 970, 1,350, and 1,720 kg/hr at outdoor temperatures of 30, 40, and 50 °C, respectively. When the HDC system is operated at the optimal regeneration temperature, achieving the maximum cooling capacity of the evaporative cooler, the process air flow rate is calculated with outdoor temperature and humidity as presented in Table 5 and Fig. 6(a). At certain operating conditions, 30 °C-13%, 30 °C-46%, 40 °C-6%, 40 °C-16.5%, 50 °C-4%, and 50 °C-7%, the EHP is not operated because the evaporative cooler has adequate cooling capacity to meet the total cooling load. At these operating conditions, the room inlet temperature of the process air is determined by the cooling capacity of the evaporative cooler, which is 10.56, 11.72, 11.25, 11.93, 12.15, and 12.49 °C at the outdoor temperature-outdoor relative humidity of 30 °C-13%, 30 °C-46%, 40 °C-6%, 40 °C-16.5%, 50 °C-4%, and 50 °C-7%, respectively. Under the outdoor temperature-outdoor relative humidity of 30 °C-79%, 40 °C-27%, and 50 °C-9%, because the total cooling load exceeds the maximum cooling performance of the evaporative cooler, EHP is operated to cool down a supply air before entering the room conditioned space at 12.5 °C.

Figure 7(a) presents the changes in consumption rate of regeneration heat and electric power consumption at the optimal regeneration temperature and minimal regeneration temperature under various outdoor conditions. The red and black columns of Fig. 7(a) indicate the consumption rate of regeneration heat and electric power consumption, respectively. Figure 7(b) presents the changes in cooling capacity of the evaporative cooler and EHP at the optimal regeneration temperature and minimal regeneration temperature under various outdoor conditions. The red and black columns of Fig. 7(b) indicate the cooling capacity of the

**Table 6** Coefficients for the process air flow rate regression Eq. (9)

Constants	Value
$z_0$	-79.43
A	25.62
B	8369.47
C	0.337
D	-330,899
F	317.65

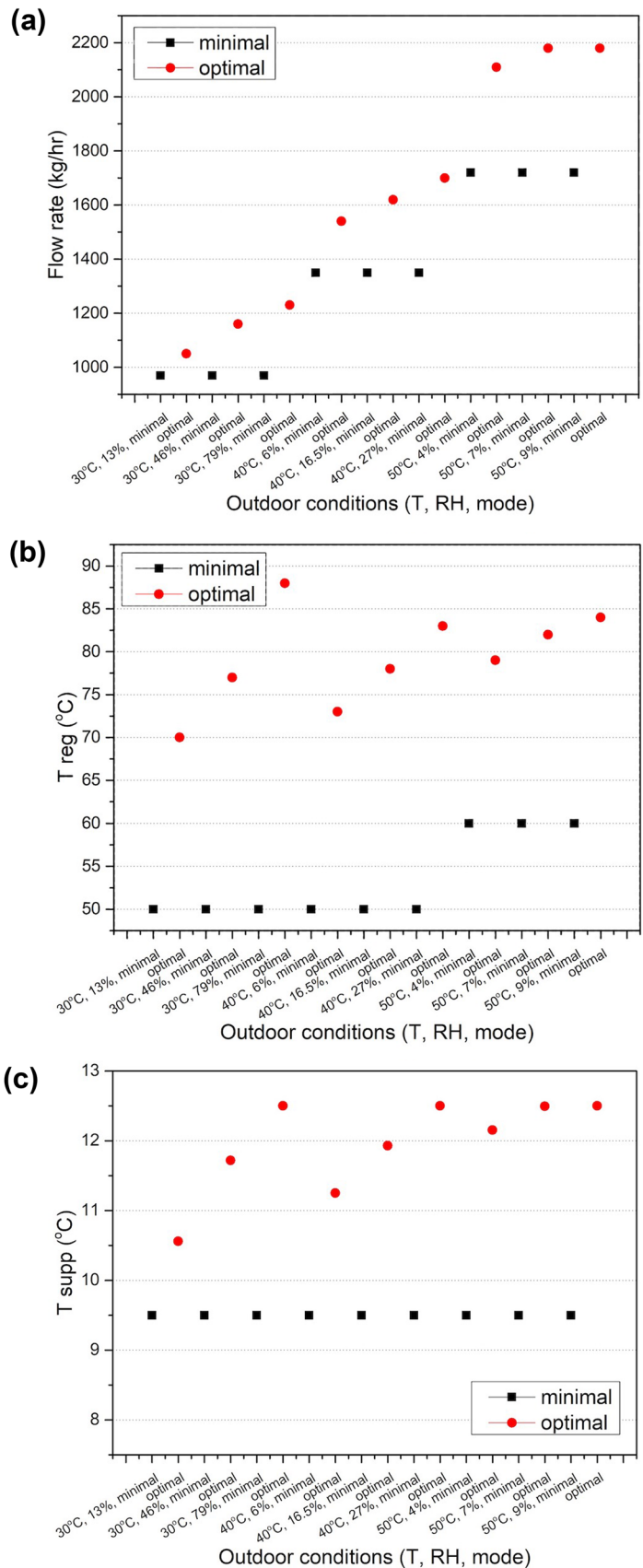
**Fig. 5** Effects of the room inlet temperature of the process air on **a** the consumption rate of regeneration heat and **b** EHP power consumption at the optimal regeneration temperature (red) and low regeneration temperature (blue)

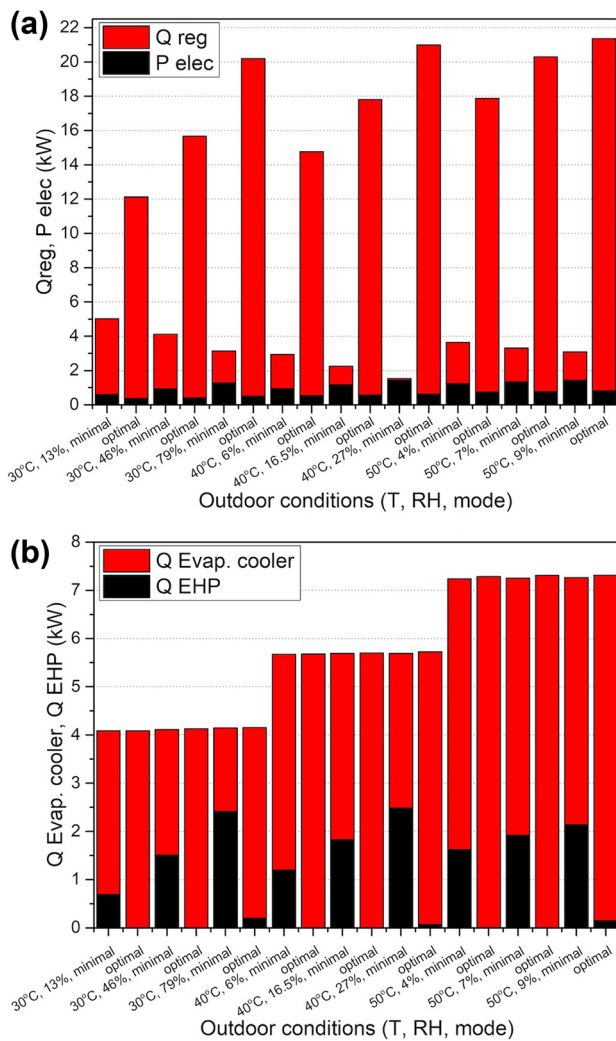


evaporative cooler and the EHP, respectively. As described, the consumption rate of regeneration heat increases due to an increase in both regeneration temperature and process air flow rate. When the HDC is operated at the minimal

regeneration temperature, the consumption rate of regeneration heat slightly decreases when the outdoor temperature increases from 30 to 40 °C due to the decrease in difference between outdoor temperature and regeneration temperature

**Fig. 6** Variations in **a** the process air flow rate, **b** regeneration temperature, and **c** room inlet temperature of the process air at the optimal and minimal regeneration temperatures under various outdoor conditions





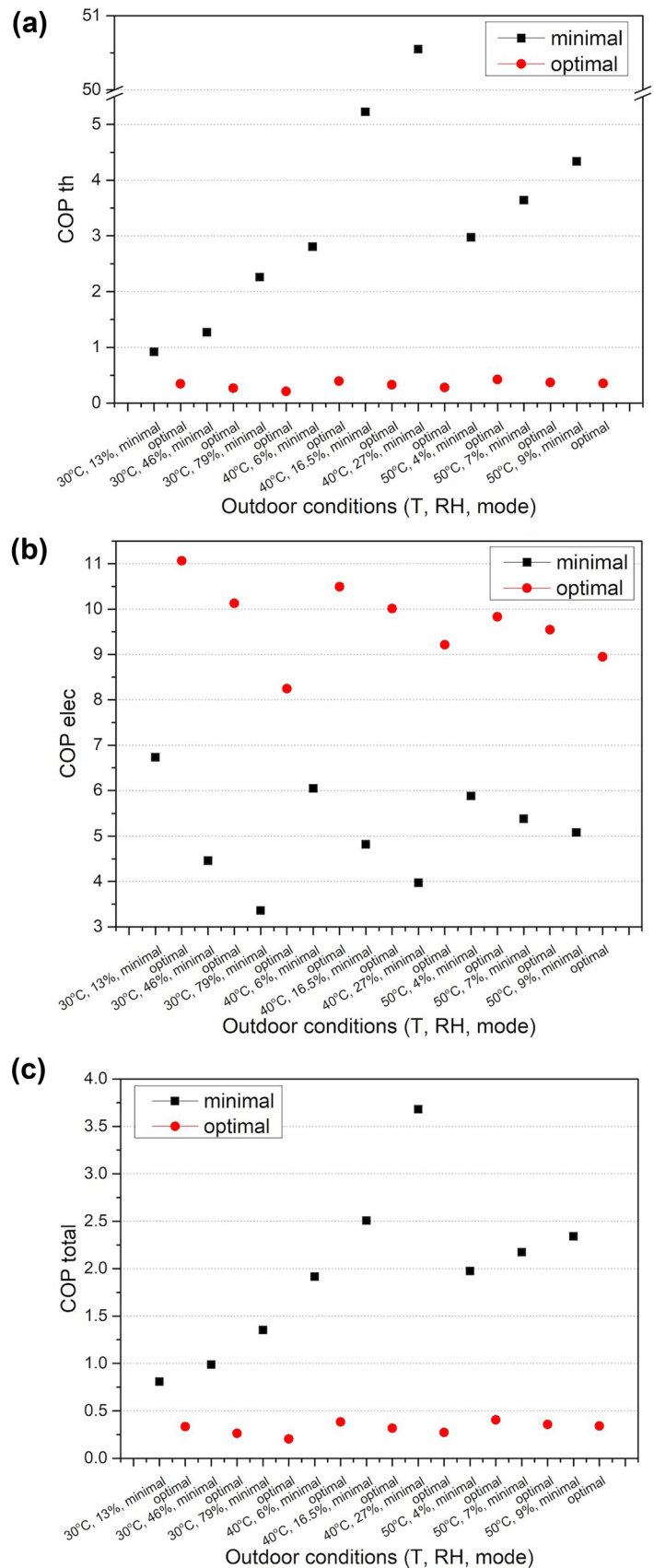
**Fig. 7** Variations in **a** consumption rate of regeneration heat and electric power consumption; **b** cooling capacity of the evaporative cooler and EHP capacity at the optimal regeneration temperature and minimal regeneration temperature under various outdoor conditions

of 50 °C. When the outdoor temperature increases to 50 °C, heat consumption rate for regeneration slightly increases compared to that at 40 °C because the regeneration temperature instantly increases to 60 °C. When the HDC system is operated at the minimal regeneration temperature, the evaporative cooler does not have sufficient cooling capacity, and the EHP operation is increased to meet the target cooling load, as presented in Fig. 7(b). Consequently, the consumption rate of regeneration heat decreases and the electric power consumption enhances at hot and humid conditions,

because the regeneration air absorbs more heat by flowing through the condenser, as presented in Fig. 7(a). When the HDC system is operated at the optimal regeneration temperature, because the total cooling load is mostly achieved by the evaporative cooler, the EHP is slightly operated only at high outdoor temperature and high outdoor relative humidity: 30 °C-79%, 40 °C-27%, and 50 °C-9%, as presented in Fig. 7(b). When the regeneration temperature and process air flow rate increase, as presented in Fig. 7(a), the consumption rate of regeneration heat increases with increasing outdoor temperature and humidity, as shown in Fig. 7(a). However, the electric power consumption is not highly increased by increasing the outdoor temperature and humidity.

Figure 8(a)–(c) present the thermal, electric, and total COP of the HDC system at the minimal and optimal regeneration temperatures, respectively. When the HDC system is operated at the minimal regeneration temperature, the respective thermal COP and electric COP decrease with increasing outdoor temperature and humidity because of decrease in consumption rate of regeneration heat and the increase in electric power consumption. The thermal COP and electric COP are 0.92–50.55 and 3.36–6.73, respectively. In particular, when the outdoor temperature is 40 °C and relative humidity is 27%, the thermal COP instantly increases to the maximum value of 50.55 because the consumption rate of regeneration heat reaches a minimum value of 0.11 kW. When the HDC system is operated at the optimal regeneration temperature, the thermal COP slightly changes from 0.21 to 0.42 because the consumption rate of regeneration heat is much higher than that of the total cooling capacity. The electric COP highly changes from 8.24 to 11.10. Regardless of the regeneration temperature variation, the electric COP decreases when both outdoor humidity and outdoor temperature increase because the EHP supplement should be increased to meet the total cooling load increase. The electric COP at the optimal regeneration temperature is very higher than that at the minimal regeneration temperature due to the electric power consumption variation. Although the electric COP highly varies with the change in outdoor humidity and outdoor temperature, the total COP mainly depends on the thermal COP because the consumption rate of regeneration heat is much higher than that of electric power consumption. The total COP at the optimal regeneration temperature and minimal regeneration temperature varies from 0.81 to 3.68 and from 0.20 to 0.40, respectively. The total COP reaches a maximum value of 3.68 at an outdoor temperature of 40 °C and an outdoor relative humidity of 27%.

**Fig. 8** Variations in **a** thermal COP, **b** electric COP, and **c** total COP at the optimal and minimal regeneration temperatures at various outdoor conditions



## 4 Conclusion

In this study, the optimal operating conditions of the hybrid desiccant cooling system under actual climate conditions in the summer season worldwide were captured. The 1-D steady model of the hybrid desiccant cooling system developed in our previous study was utilized for the sensitivity analysis. The hybrid desiccant cooling system model was simulated to achieve operational optimization under various climate conditions, which represented the actual climate conditions of cities worldwide and covered the extreme climate conditions of a maximum 50 °C outdoor temperature. The summary of this study and its findings are as follows:

1. There is an optimal regeneration temperature for the evaporative cooler and the desiccant rotor to get a maximized cooling performance of the evaporative cooler at various outdoor humidity and outdoor temperature. The optimal regeneration temperature for the hybrid desiccant cooling system is between 70–88 °C when the outdoor humidity is 13–79% at an outdoor temperature of 30 °C. The regression of the optimal regeneration temperature by varying the outdoor temperature and outdoor humidity was obtained as a polynomial regression equation based on the outdoor temperature and outdoor humidity.
2. When the hybrid desiccant cooling system is operated with the minimal regeneration temperature, the outdoor humidity does not affect the optimal process air flow rate. When the hybrid desiccant cooling system is operated with the optimal regeneration temperature, the room inlet temperature of the process air is highly affected by the outdoor temperature and outdoor humidity. The regression of the optimal process air flow rate by varying the outdoor temperature and outdoor humidity has been obtained as a polynomial equation based on the outdoor humidity and outdoor temperature.
3. When the hybrid desiccant cooling system is operated at the minimal regeneration temperature, the respective thermal coefficient of performance and electric coefficient of performance decrease with increasing outdoor temperature and humidity because of the decrease in consumption rate of regeneration heat and increase in electric power consumption. When the hybrid desiccant cooling system is operated at the optimal regeneration temperature, the thermal coefficient of performance is slightly changed from 0.21 to 0.42 because the consumption rate of regeneration heat is much higher than that of the total cooling capacity. The electric coefficient of performance is highly changed from 8.24 to 11.10. Although the electric coefficient of performance highly varies with the change in outdoor humidity and outdoor

temperature, the total coefficient of performance mainly depends on the thermal coefficient of performance because the consumption rate of regeneration heat is much higher than that of electric power consumption. The total coefficient of performance at the optimal regeneration temperature and minimal regeneration temperature varies from 0.81 to 3.68 and from 0.20 to 0.40, respectively.

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## Declarations

**Conflict of interests** 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.

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