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Energy and Exergy Analyses of a Novel Recirculated Regenerative Rotary Desiccant Wheel-Assisted Dehumidification System

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Energy and exergy analyses of a novel recirculated regenerative rotary desiccant wheel-assisted dehumidification system

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Abstract:

The indoor air humidity and temperature in the marine cabins could be as high as 80% and 40 ℃ - 50 ℃, respectively, which may cause security risks to crews and devices. The incorporation of solid desiccant wheel for improved dehumidification and air quality control would contribute to energy saving in air conditioning system through waste heat recovery for regeneration. In the present work, a novel recirculated regenerative rotary desiccant wheel-assisted dehumidification system was proposed. The energy and exergy performances of the system under variable working conditions were quantitatively studied based on experimental tests. The dehumidification effectiveness, dehumidification performance coefficient (DCOP), sensible energy ratio and coefficient of performance (COP) were introduced as key energetic indicators. The effects of air bypass ratio (50% - 85%), process air temperature (28 ℃ - 40 ℃), relative humidity of process air (60%RH - 80%RH) and regeneration air temperature (130 ℃ - 150 ℃) on the system performances were obtained. It was found the COP of the system ranged between 0.4 and 1.8, while the exergy efficiency was only 4% - 11%. An optimal bypass ratio existed which was determined 55%-60%. Overall, the energy saving and exergy consumption characteristics of the proposed system at various working conditions were explored to prompt its practical application.

Keywords:

Solid desiccant wheel, Dehumidification, Recirculation, Energy analysis, Exergy analysis.

1. Introduction

The indoor air humidity and temperature in closed marine cabins could be extremely high due to long time operation, sometimes as high as 80% and 40 ℃~50 ℃, respectively, which may bring about security risks to crews and devices. Dehumidification is thus energy-intensive but urgent, so as to ensure the normal navigation of the marines in humid environment. Likewise, the air quality control and thermal comfort are especially important for enclosed space with restricted or impossible air exchange with outdoor atmosphere, such as life-support system of submarines, underground shelters, space and aircrafts [1]. Besides, dehumidification is also widely required in various occasions [2], such as flue gas moisture recovery in power plants [3], air conditioning in buildings [4], desalination [5], etc.

The desiccant wheel (DW) is a promising technology for energy-efficient air dehumidification [6]. It is based on the cyclic adsorption and desorption of the desiccant materials for continuous moisture removal from process airstream. In particular, desiccants allow for the decoupling of the thermal load in latent and sensible components, which provides more accurate humidity control and lower energy consumption [7]. Moreover, the ozone depletion potential associated with refrigerants used in conventional vapor compression air conditioning systems can be alleviated by the adoption of desiccant wheel assisted HVAC system [8]. As a result, there have been a variety of commercially available desiccants that have been applied in buildings, food drying, medicine preservation, mobile air conditioning , and so on [9,10].

The incorporation of solid desiccant wheel for improved dehumidification and air quality control would contribute to energy saving in air conditioning system through waste heat recovery for regeneration [11]. Much work has been conducted towards performance optimization of dehumidification systems based on the rotary desiccant wheels [12]. Among them, material screen [13], multistage optimization [14], regeneration process energy saving, and performance variation in different thermal and humid environments have been the focus of concern. There is a limit to material optimization in single-stage desiccant wheels. The performance could be further improved by applying two-stage desiccant dehumidification system as indicated by Tavakol et al. [15] and Liu et al. [16]. Apart from system configuration optimization, reduction of the energy consumption in the regeneration process has always been an important method to improve the dehumidification performance. In particular, the reduction of regeneration energy consumption can be achieved by lowering the regeneration temperature [17], utilizing the industrial waste heat [18] and solar energy [19], and recovering condensation energy by heat pumps. Tu et al. [20] proposed efficient configurations for desiccant wheel cooling systems using different heat sources for regeneration. Moreover, the performance optimization of the rotary desiccant wheel under various thermal and humid environments has also attracted much attention [21]. It was found that the performance of the rotary desiccant wheel dehumidification system was affected by process air temperature, process air humidity, process air ratio, regeneration temperature, rotation speed [22].

Exergy analysis may serve as a useful tool in desiccant-assisted HVAC system optimization [23-25]. Enteria et al. [26] investigated the energetic and exergetic performance of solar-desiccant dehumidification system at summer days. Tu et al. [27] analysed exergy destructions in desiccant wheel based ventilation systems. The regeneration air preheating by electricity would account for 70% of the total exergy destructions as found by Hürdoğan et al. [28], and could be remarkably reduced by the application of heat pump [29]. Uçkan et al. [30] carried out exergy analysis of a novel configuration of desiccant based evaporative air conditioning system, and investigated the effect of operation conditions on the system performance. Rafique et al. [31] conducted energy, exergy and anergy analysis of a solar desiccant cooling system. El-Agouz et al. [32] investigated the performance of desiccant air conditioning system with geothermal energy under different climatic conditions. Singha et al. [33] presented exergy analysis of desiccant assisted evaporative cooling system. La et al. [34] investigated the effect of irreversible processes on the thermodynamic performance of open-cycle desiccant cooling cycles. Vivekh et al. [35] performed experimental performance evaluation of desiccant coated heat exchangers from a combined first and second law of thermodynamics perspective. Gonçalves et al. [36] conducted exergetic analysis of a desiccant cooling system to search for performance improvement opportunities. Tu et al. [37] found from exergy analysis that lowering the regeneration temperature was a sensible pathway for desiccant-based dehumidification system.

The waste heat generated by the marine devise requires to be recovered promptly and used effectively. Bearing this in mind, the waste heat can be introduced as the heat source for the regeneration of the desiccant wheel to reduce energy consumptions of air-conditioning system. Specifically, the main energy-consuming equipment in the marines includes diesel main engine, diesel generator, auxiliary boiler, etc. The exhaust temperature of the diesel main engine is generally between 260 and 400 ℃, accounting for 30% of the total input heat. Plenty of residual heat in high grade is not effectively utilized. Moreover, the relative humidity of the air in enclosed cabins generally exceeds 80%, which not only affects the crew's comfort, but also easily causes hull corrosion, reduces cable insulation performance, and raises security risks. High air humidity leads to a large proportion of the latent heat load for the air conditioning in the ship, which is one of the main electric consumers in the ships. Its energy consumption accounts for about 20% of the total power consumptions.

Some scholars have tried to optimize the air-conditioning performance in ships by applying rotary desiccant wheel. For instance, Zhu et al. [38] performed energy and exergy performance analysis of a marine rotary desiccant air-conditioning system based on orthogonal experiment. However, dehumidification in underwater enclosed cabins, where natural ventilation cannot be applied, has always been a difficult problem. Conventional sea water direct cooling and air conditioning dehumidification have their merits and drawbacks. The desiccant wheel is a solution to obtain better control of humidity and enhance human comfort. To date, several classic rotary desiccant wheel systems have been proposed [39], including Pennington cycle, Pennington recycling, Dunkle cycle and solid dry cooling circulation system [40]. However, the process air can't be provided as a source of regeneration air for closed compartments such as ships and submarines.

Evidently, the existing cycles are not suitable for application in enclosed cabins. To address the issues of energy efficient dehumidification in the enclosed space with high humidity, a novel recirculated regenerative rotary desiccant wheel-assisted dehumidification system was proposed in our previous work [41], and an experiment system was established. Energy and exergy analyses were carried out based on experimental results to demonstrate the energy saving potential and exergy destruction distribution of the system. The objective of the present work is to investigate the influence of key operating conditions on the dehumidification performance from both energetic and exergetic point of views.

2. Working principle and experimental setup

2.1. Working principle of the proposed system

The diagram of the recirculated regenerative rotary desiccant wheel-assisted dehumidification system is shown in Fig. 1. It is characterized by recycling and utilizing the waste heat of the marine equipment to realize the regeneration of the dehumidifier, and using the air circulation to form the closed structure of the rotary desiccant wheel dehumidification system for the closed environment of cabins in marines. More specifically, the high-humidity air from the enclosed cabins (1) is dehumidified by the dehumidifier attached to the passage through the dehumidification side of the desiccant wheel. The dry and cold air (2) passes through the waste heat recovery devices, in which it is heated. Then, part of the outlet hot air (3) enters the regeneration zone to desorb the water in the dehumidifier for the circulation regeneration of the wheel. The bypass ratio should be determined by heat balance. Finally, both air streams (4 and 5) are cooled by a seawater condenser. Cold and dry air (6 and 7) is recycled to the cabins. The role of the desiccant wheel is to remove moisture from the process airstream and to improve the dehumidification rate because d_2 is lower than d_4.

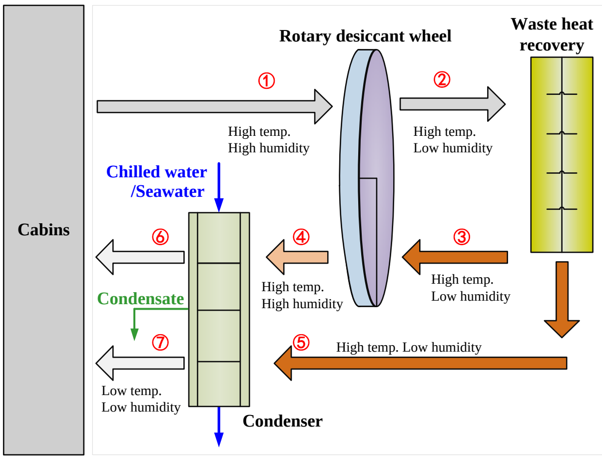
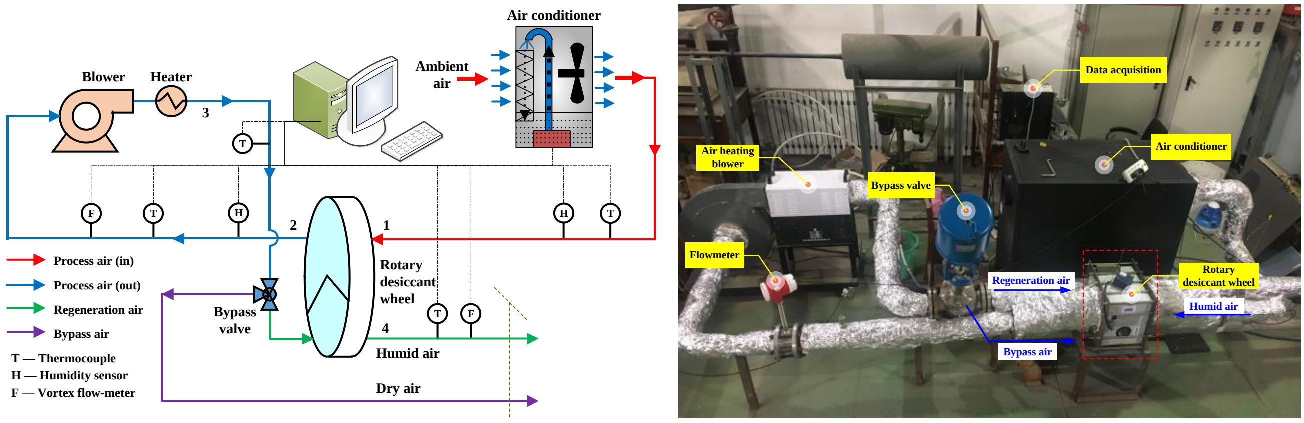


Figure 1. Working principle of the proposed rotary desiccant wheel-based dehumidification system

2.2. Experimental system setup

Fig. 2 discloses the experimental system of the rotary desiccant wheel-based air dehumidification system. It consists of an air conditioner which could provide constant temperature and humidity of air as desired, a rotary desiccant wheel, an air heating blower, a bypass valve, four thermocouples, two temperature and humidity sensors, and two vortex flow-meters. The air is heated and humidified by the air conditioner to simulate high-humidity air from the ship cabin. Then it is dehumidified by the rotary desiccant wheel. After being heated by the heating blower, a part of the air is used for the regeneration of the desiccant materials, and the other part is exhausted. By measuring the air humidity before and after the desiccant wheel, the dehumidification rate of the system can be obtained. By measuring the volume flow rates of the process air and regeneration air, the improvement in dehumidification rate of the system can be quantitatively analysed. The measuring range and accuracy for main instruments are listed in Table 1.



(a) Schematic of the experiment system *(b) Photographic view of the experiment rig*

Figure 2. Experimental system of the rotary desiccant wheel dehumidification system.

Table 1. The measuring range and accuracy for main instruments.

Measuring instrument	Measuring range	Measuring accuracy
Air conditioner	temperature: 20-65 °C	temperature: ±2 °C
	relative humidity: 40-95%RH	relative humidity: ±4%RH
Thermocouple	temperature: -20°C-400 °C	temperature: ±0.5 °C
Temperature and humidity sensor	temperature: -40°C-60 °C	temperature: ±0.3 °C
	relative humidity: 0-100%RH	relative humidity: ±3 %RH
Vortex flow-meter	air flow: 80-800 m^3h^{-1}	flow rate: ±1.5%

The parameters of the solid desiccant dehumidifier used in the present work are listed in Table 2. The channels of the desiccant wheel are fabricated in honeycomb shape structure which is coated with desiccant materials made of metal silicate.

Table 2. Summary of the desiccant dehumidifier constructional parameters.

Item	Unit	Value
Desiccant material	Synthesized metal silicate	
Producer	Desiccant Rotors International	
Area ratio		3:1
Desiccant wheel dimension	diameter/depth, mm	260/200
Channel shape	honeycomb	
Rotary speed	r/h	8-16
Regeneration temperature	°C	100-180
Volume flow rate	m^3/h	300
Drive		220-240V/AC/50Hz

3. Performance parameters

3.1 Energetic analysis indicators

The bypass ratio in experiments, γ_{DW}, can be obtained by:

$$\gamma_{\mathrm{DW}} = \frac{q_{\mathrm{tot}} - q_{\mathrm{reg}}}{q_{\mathrm{tot}}} \quad (1)$$

where q_{tot} is the process air volume flow rate, $m^3 \cdot h^{-1}$ and q_{reg} is the regeneration air volume flow rate, $m^3 \cdot h^{-1}$.

The dehumidification effectiveness of the desiccant wheel, $\varepsilon_{\mathrm{DW}}$, which denotes the moisture removal capacity, is derived as [11,39,42]:

$$\varepsilon_{\mathrm{DW}} = \frac{\left(d_{\mathrm{pi}} - d_{\mathrm{po}}\right)}{d_{\mathrm{pi}}} \times 100\% \quad (2)$$

where d_{pi} is the humidity of the process air, $g \cdot kg^{-1}$ and d_{po} is the humidity of the process air after passing through the desiccant wheel.

The dehumidification coefficient of performance (DCOP), which denotes the energy utilization efficiency in dehumidification process, is calculated as the ratio between the thermal power due to air dehumidification and the regeneration thermal power [42]:

$$\mathrm{DCOP} = \frac{q_{\mathrm{pro}} h_v \left(d_{\mathrm{pi}} - d_{\mathrm{po}}\right)}{q_{\mathrm{reg}} c_{\mathrm{p,reg}} \left(t_{\mathrm{reg}} - t_{\mathrm{reg,o}}\right)} \quad (3)$$

where h_v is the latent heat of water vapor (J/kg), c_{p} is the specific heat capacity of humid air, J (kg K)$^{-1}$. Evidently, higher value of DCOP indicates better performance of the rotary wheel.

The sensible energy ratio (ε_{s}) is defined as the ratio between the thermal power due to the process air heating though the wheel and the regeneration thermal power [42]:

$$\varepsilon_s = \frac{q_{\mathrm{pro}} c_{\mathrm{p,pro}} \left(t_{\mathrm{po}} - t_{\mathrm{pi}}\right)}{q_{\mathrm{reg}} c_{\mathrm{p,reg}} \left(t_{\mathrm{reg}} - t_{\mathrm{pi}}\right)} \quad (4)$$

The coefficient of performance (COP) of the entire system is expressed as [6]:

$$\mathrm{COP} = \frac{Q_{\mathrm{cc}}}{E_{\mathrm{tot}}} \quad (5)$$

where Q_{cc} is the cooling capacity of the proposed system and E_{tot} is the total electricity consumption.

3.2 Exergetic analysis indicators

Apart from energy, exergy is also an essential factor to evaluate the potential efficiency of a process. Exergy is defined as the maximum amount of work which can be produced by a stream of matter, heat or work as it comes to equilibrium (thermal, mechanical, and chemical) with a specified reference environment through reversible processes [43]. In the present work, exergy analysis was implemented to reveal the mechanism of working principle and optimization of the system [44].

The specific exergy of humid air is calculated by: [45]

$$ex = \left(c_{p,a} + dc_{p,v}\right) T_0 \left[\frac{T}{T_0} - 1 - \ln\frac{T}{T_0}\right] + R_{\mathrm{a}} T_0 \left[\left(1 + 1.608d\right) \ln\frac{1 + 1.608 d_0}{1 + 1.608 d} + 1.608 d \ln\frac{d}{d_0}\right] \quad (6)$$

where the reference state is taken as: T_0=308.15 K, d_0=37.1 g/kg.

The exergy destruction in the desiccant wheel, $E_{\mathrm{d,DW}}$, is written as:

$$E_{\mathrm{d,DW}} = E_{\mathrm{w,DW}} + E_{\mathrm{pi}} + (1\text{-}\gamma_{\mathrm{DW}}) E_{\mathrm{reg}} - E_{\mathrm{po}} - E_{\mathrm{reg,o}} \quad (7)$$

where E_{pi}, E_{reg}, E_{po}, and $E_{\mathrm{reg,o}}$ are the exergy rates of the process air in, regeneration air, process air out and regeneration air out, respectively, $E_{\mathrm{w,DW}}$ is the power consumption of the desiccant wheel.

The exergy destruction in the air heater, $E_{\mathrm{d,AH}}$, is calculated by:

$$E_{\mathrm{d,AH}} = E_{\mathrm{e,AH}} + E_{\mathrm{po}} - E_{\mathrm{reg}} \quad (8)$$

where $E_{e,AH}$ is the power consumption of the air heater.

The exergy destruction in the chilled water condenser, $E_{d,COND}$, is obtained by:

$$E_{d,COND} = E_{reg,o} + \gamma_{DW} E_{reg} - E_{ca} - E_{cond,out} - \Delta E_{cw} \quad (9)$$

where E_{ca} denotes the exergy of cooling air after water condenser, $E_{cond,out}$ represents the exergy of condensed water leaving the system, and ΔE_{cw} is the exergy increase of the cooling water.

The total exergy destruction of the system is the sum of exergy destructions in the components:

$$E_{d,tot} = E_{d,DW} + E_{d,AH} + E_{d,COND} \quad (10)$$

The exergy efficiency of the overall system is obtained by the product exergy (dehumidified air) divided by the exergy provided by the regeneration air and electricity. It represents the irreversibility of the process [34].

$$\eta_{ex} = \frac{E_p}{E_f} = 1 - \frac{E_{d,tot}}{E_f} \quad (11)$$

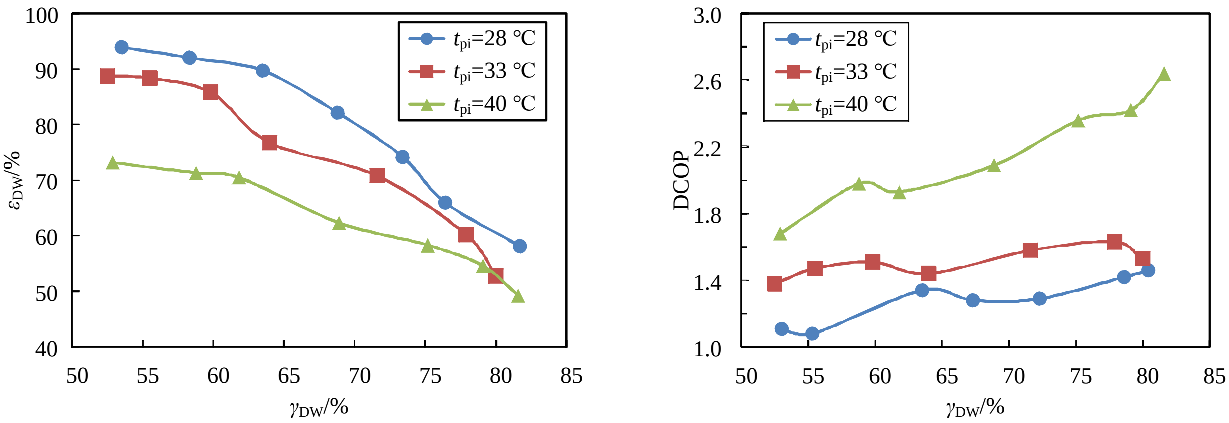
where E_p represents the exergy of product air and E_f is the total exergy input.

4. Results and discussion

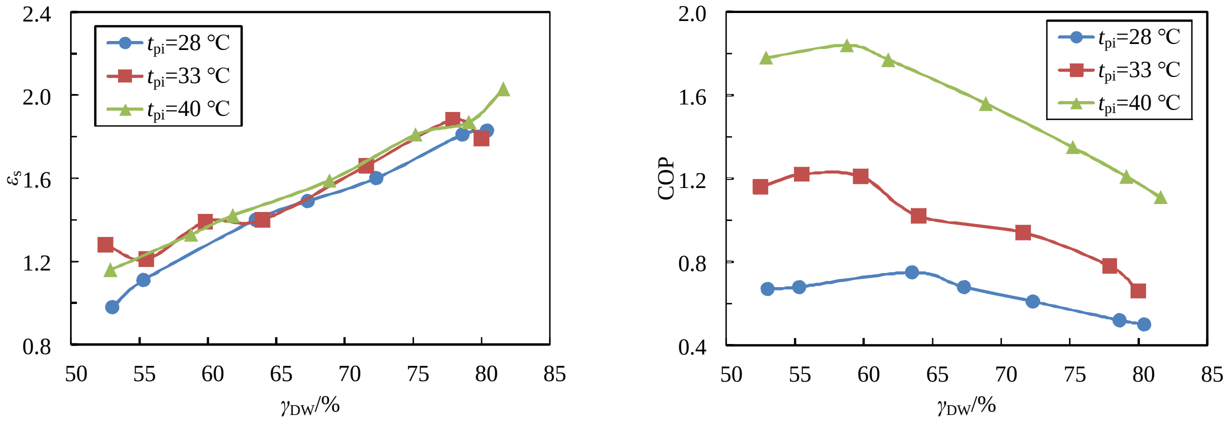
4.1. Energetic analysis of the system

4.1.1. Influence of the process air temperature t_{pi}

During the navigation of ships, the air temperature will change in the cabin, which will affect the performance of wheel dehumidification system. In the experiments, the relative humidity of the process air was 85% and the regeneration air temperature was 150 °C. The influence of the bypass ratio on the dehumidification effectiveness of the desiccant wheel (ε_{DW}) under different process air temperatures is shown in Fig. 3(a). It can be found that the ε_{DW} decreased significantly with the increasing bypass coefficient and process air temperature. It ranged from 50% to 92% under experiment conditions. The influence of bypass ratio on the DCOP under different process air temperatures is shown in Fig. 3(b). It can be found that the DCOP increased with the increasing bypass coefficient and process air temperature. It ranged from 1.0 to 2.8 under experiment conditions. The sensible energy ratio increased linearly with the bypass ratio as shown in Fig. 3(c). It ranged between 1.0 and 2.0. While higher values of ε_{DW} and DCOP represent better dehumidification performances, a higher value of ε_s is unfavorable, as it means a higher temperature increase of the process air through the wheel and therefore a higher cooling load on the cooling device [42]. As a result, the COP increased with the process air temperature (Fig. 3(d)). More importantly, there existed an optimal COP (0.7-1.8) when the bypass ratio was 50%-60%. It is indicated that the proposed system had higher energy utilization under high humidity air conditions.



(a) ε_{DW} (b) DCOP

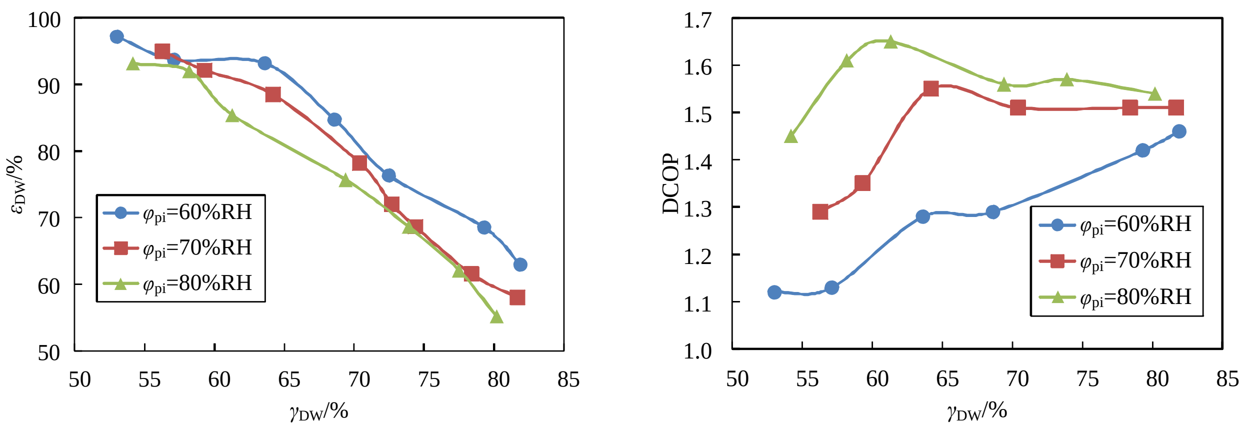


(c) ε_s (d) COP

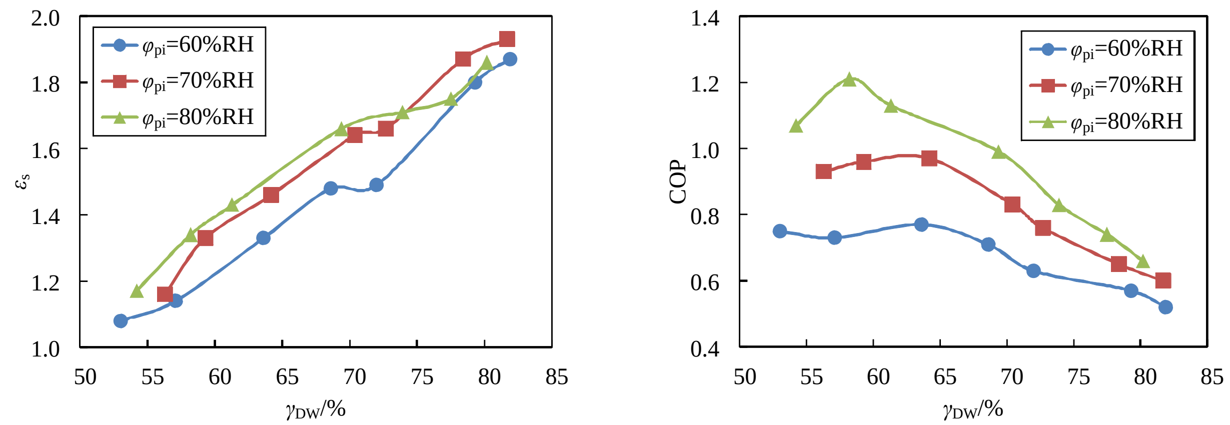
Figure 3. Effect of the process air temperature (t_{pi}) and bypass ratio (γ_{DW}) on the energy performance indicators.

4.1.2. Influence of the process air humidity φ_{pi}

As typical experimental conditions, the process air temperature was 33 °C and regeneration air temperature was 150 °C. The influence of the bypass ratio on the system energy indicators under different process air humidity is shown in Fig. 4. The COP ranged between 0.5 and 1.2. The humidity of process air had slight impact on the dehumidification effectiveness (Fig. 4(a)) and sensible energy ratio (Fig. 4(c)). The DCOP increased with the process air humidity (Fig. 4(b)), which indicated that the desiccant wheel was more suitable for drying air with high humidity. This is also confirmed by the increase of system COP with the process air humidity (Fig. 4(d)).



(a) ε_{DW} (b) DCOP



(c) ε_s (d) COP

Figure 4. Effect of the process air humidity (φ_{pi}) and bypass ratio (γ_{DW}) on the energy performance

indicators.

4.1.3. Influence of the regeneration air temperature t_{reg}

The waste heat from marine equipment is always fluctuating, which causes the regeneration air temperature to change accordingly and has a significant impact on the performance of wheel dehumidification system. Fig. 5 shows the effect of different regeneration air temperature on the energy consumption characteristics of the system when the process air temperature was 28 °C and the relative humidity was 85%. It can be found that the dehumidification effectiveness was significantly enhanced by increasing the regeneration air temperature (Fig. 5(a)), without affecting the sensible energy ratio too much (Fig. 5(c)). The DCOP was also improved in higher regeneration temperatures (Fig. 5(b)). However, the COP of the system slightly reduced with the regeneration temperature, which can be explained by higher electrical consumption in the air heater. Moreover, there existed the maximum COP, around 0.85, when the bypass ratio was 55%-60% (Fig. 5(d)).

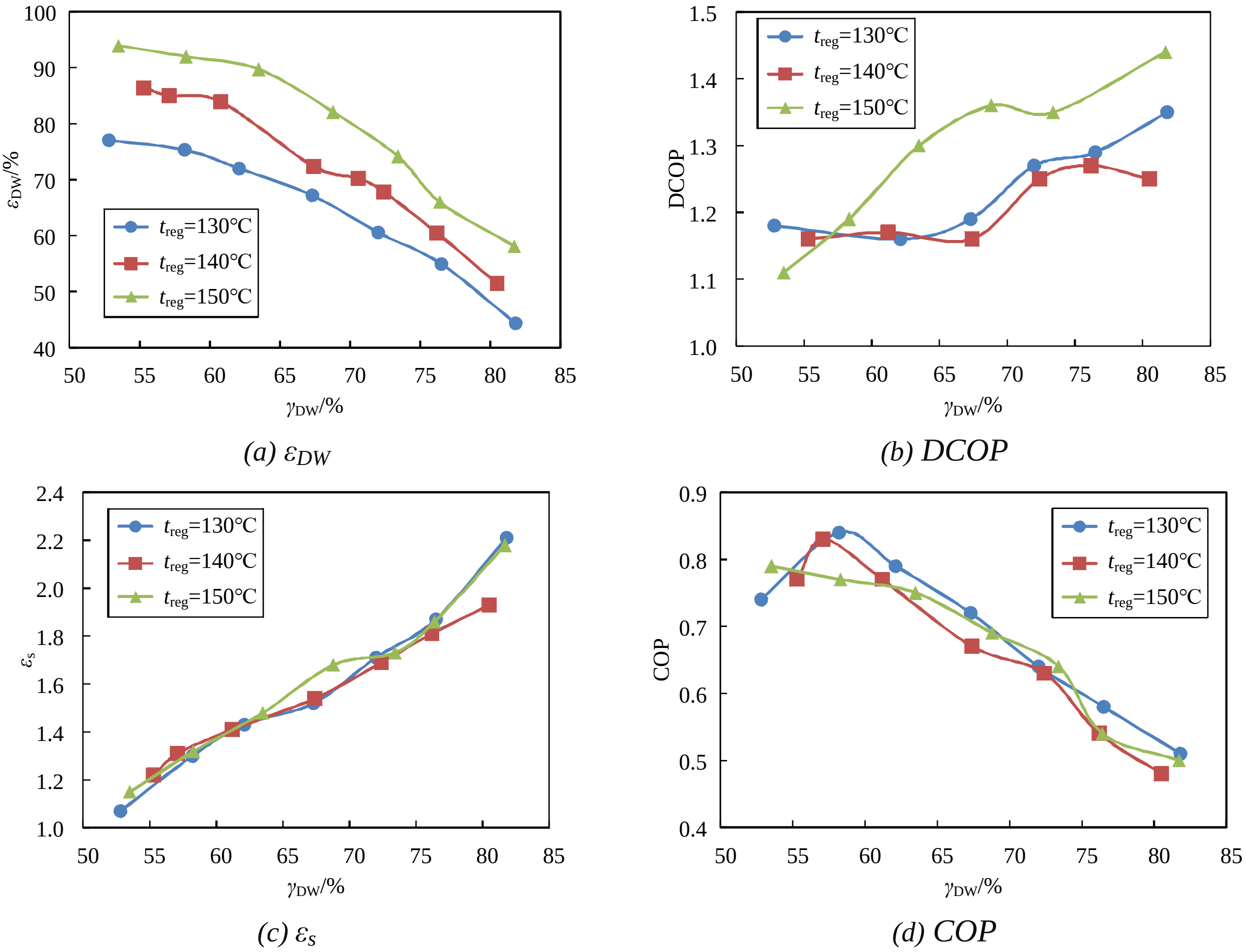
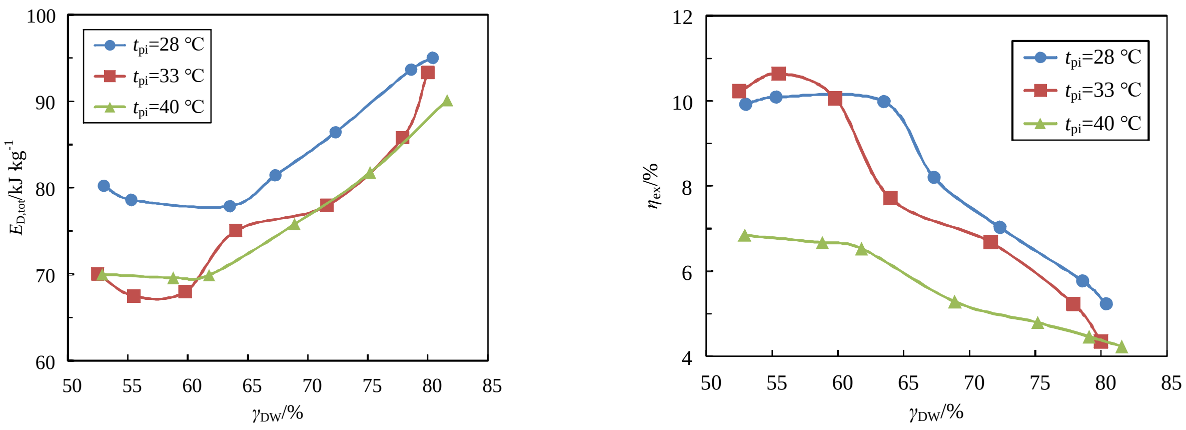


Figure 5. Effect of the regeneration air temperature (t_{reg}) and bypass ratio (γ_{DW}) on the energy performance indicators.

4.2. Exergetic analysis of the system

4.2.1. Influence of the process air temperature t_{pi}

The influence of process air temperature on the exergy destruction rate and efficiency of the system is shown in Fig. 6. The process air humidity was 85% and the regeneration air temperature was 150 °C. The exergy destruction rate increased with the bypass (Fig. 6(a)) because of more regeneration air flowing through wheel. The exergy efficiency decreased with the increase of process air temperature and bypass, which was between 4% and 12% (Fig. 6(b)). There existed an optimal exergy efficiency, which is similar to the findings of COP. The bypass ratio was around 55%. The highest exergy efficiency was only 10.5%.

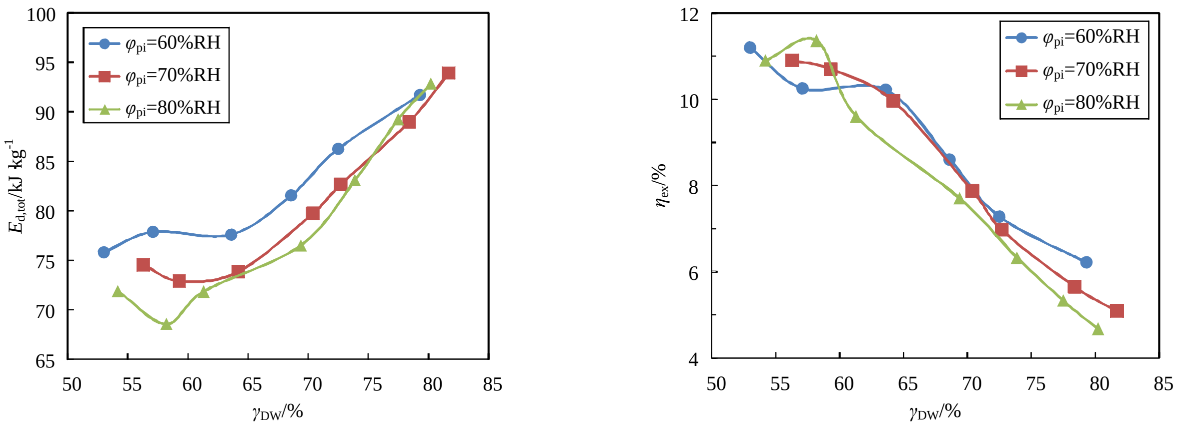


(a) exergy destruction rate *(b) exergy efficiency*

Figure 6. Effect of the process air temperature (t_{pi}) and bypass ratio (γ_{DW}) on the exergy performance.

4.2.2. Influence of the process air humidity φ_{pi}

As typical experimental conditions, when process air temperature was 33 °C and regeneration air temperature was 150 °C, the effect of process air humidity on exergy characteristics of the system is shown in Fig. 7. The exergy destruction rate increased with the bypass ratio while decreased with the process air humidity as shown in Fig. 7(a). As a result, the overall exergy efficiency decreased with the bypass ratio, but showed marginal difference with different process air humidity (Fig. 7(b)). The highest efficiency was found at 11%.

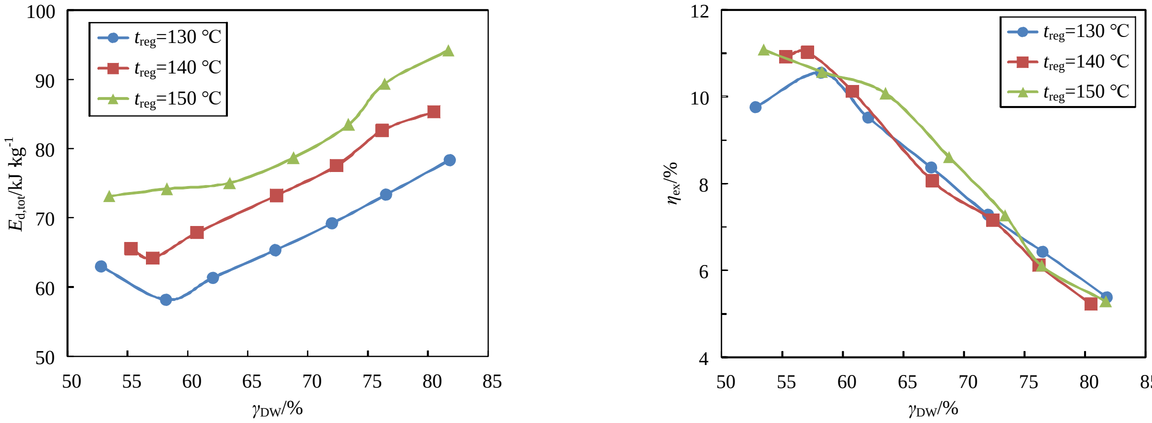


(a) exergy destruction rate *(b) exergy efficiency*

Figure 7. Effect of the process air humidity (φ_{pi}) and bypass ratio (γ_{DW}) on the exergy performance.

4.2.3. Influence of the regeneration air temperature t_{reg}

The effect of regeneration air temperature on exergy characteristics of the system is shown in Fig. 8. The process air temperature was 33 °C and humidity was 85%. The exergy destruction rate increased significantly with the regeneration air temperature, leading to remarkable drop in the overall exergy efficiency. The exergy efficiency ranged between 4% and 11%. The optimal exergy efficiency of the system was just around 11%, which indicated that the direct usage of high temperature air for regeneration should be improved. Through the adjustment of the bypass ratio, the dehumidification performance [41], energy performance (as indicated by COP) and the exergy efficiency (which shows the irreversibility of the process) could be maximized.



(a) exergy destruction rate *(b) exergy efficiency*

Figure 8. Effect of the regeneration air temperature (t_{reg}) and bypass ratio (γ_{DW}) on the exergy performance.

5. Conclusions

In the present work, energy and exergy analyses on a novel recirculated regenerative rotary desiccant wheel-assisted dehumidification system were carried out based on experimental results. The impact of key process parameters and air conditions on the energetic and exergetic performances was investigated. The following conclusions can be drawn:

1) The DCOP of the investigated desiccant wheel ranged between 1.0 and 2.8 in different working conditions, and increased with the air humidity. The COP of the proposed system was between 0.4 and 1.8. There existed the maximum COP when the bypass ratio was 55%-60%. The proposed system was more suitable for working in high humidity environment.

2) Exergy analysis of the system demonstrated that the highest exergy efficiency was only 11%. The exergy destruction rate increased with the bypass ratio and the regeneration air temperature, while decreased with process air temperature and humidity. The electrical air heater should be optimized through waste heat recovery in cabins to improve the system performance.

Acknowledgments

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