Modeling and optimization for regenerator with heat recovery in liquid desiccant dehumidification system

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**MODELING AND OPTIMIZATION FOR
REGENERATOR WITH HEAT RECOVERY IN
LIQUID DESICCANT DEHUMIDIFICATION
SYSTEM**

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SCHOOL OF ELECTRICAL & ELECTRONIC ENGINEERING
2018**

MODELING AND OPTIMIZATION FOR REGENERATOR WITH HEAT RECOVERY IN LIQUID DESICCANT DEHUMIDIFICATION SYSTEM

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SUMMARY

As a broad and complex topic, energy efficiency has drawn an intensive attention in research community. Air Conditioning and Mechanical Ventilation (ACMV) systems in Singapore consume a substantial portion of energy in commercial buildings. Prioritizing high energy-efficiency ACMV system will deliver dramatic energy savings. Compared with conventional mechanical based air dehumidification schemes, liquid desiccant dehumidification system (LDDS) exhibits many advantages and demonstrates its superiority. Dehumidifier and regenerator are two major parts in LDDS, where the dehumidification and regeneration take place. Liquid desiccant solution is diluted in the dehumidifier since it absorbs excess moisture from the air. The diluted solution is re-concentrated in the regenerator. Majority of energy is consumed by regenerator in system operation in order to keep a suitable regenerating rate. Therefore, this thesis tries to put effort on energy saving of regenerator and mainly focus on waste heat recovery, modeling, design and operating optimization and performance evaluation. The main contributions of this thesis are summarized as below:

1. To reduce the amount of energy required to attain demanding regenerating performance, this thesis employs the waste heat recovery technique to reutilize the waste heat in the regenerating process. In the regenerator, the direct emission of the exhausted regenerating air, which holds high temperature and humidity, causes a great loss of heat. Thus, this part of heat can be recovered. Two heat recovery device, Heat Pipe Heat Exchanger (HPHE) and Fixed-plate Heat Exchanger (FPHE) are installed in regenerator to recover heat from the exhausted regenerating air to preheat the incoming air continuously. For the purpose of analyzing the heat recovery process, hybrid heat recovery model is established and validated by the experimental data. The model is simple and does not need iterative computation. It can be used to monitor and optimize the heat recovery performance in regenerator. The heat recovery efficiency

under different working conditions of HPHE and FPHE can be predicted with the hybrid model.

2. To evaluate and compare the regenerator of Liquid Desiccant Dehumidification System (LDDS) without and with heat recovery, the performance analysis is conducted by hybrid heat transfer, mass transfer and heat recovery models, and the simulation results are then validated by the experimental results. Regenerator without heat recovery, with 4 rows HPHE, with 8 rows HPHE and with FPHE are compared to investigate the relationship between heat recovery rate and the additional fan energy consumption caused by the existence of heat recovery device. Many indexes are defined as the comparing criteria. Effects of air mass flow rate on the regenerating and heat recovery performance are also discussed. The results show that the numerical computation is effective and accurate. With heat recovery device, the regenerating performance is in general improved.
3. In order to find out the trade-off way between the regenerating performance and total pressure drop of the system, the multi-objective optimization is performed with pressure drop model, heat recovery model, heat transfer model and mass transfer model. Heat recovery device can recover waste heat and save the energy consumption of regenerator. However, its existence causes additional pressure drop. Other than heat recovery device, the structured packing is filled in regenerator tower to increase the contact area between regenerating air and liquid desiccant. It induces dramatic pressure drop due to the remarkable flow resistance. The dimension of structured packing, such as optimal height and diameter, affects the regenerating performance and pressure drop greatly. Optimal geometric dimension of packing can provide sufficient regenerating performance with acceptable pressure drop. Besides the design parameters in the optimization, operating parameters, especially air flow rate, also plays a significant role in energy saving and pressure drop.

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NOMENCLATURE

A_f	surface area of FPHE (m^2)
$a_{fan,0} - a_{fan,3}$	fan model parameters
a_p	specific surface area of structured packing (m^2/m^3)
b	corrugation base length (m)
$b_{f,1}$ FPHE	unknown parameters of heat recovery model for regenerator with FPHE
$b_{f,2}$ FPHE	unknown parameters of heat recovery model for regenerator with FPHE
$b_{h,d}$ HPHE	unknown parameters of heat recovery model for regenerator with HPHE
$b_{h,e}$ HPHE	unknown parameters of heat recovery model for regenerator with HPHE
$c_{f,1} - c_{f,7}$	unknown parameters of heat and mass transfer models for regenerator with FPHE
c_f FPHE	unknown parameters of heat recovery model for regenerator with FPHE
$c_{h,1} - c_{h,7}$	unknown parameters of heat and mass transfer models for regenerator with HPHE
c_h HPHE	unknown parameters of heat recovery model for regenerator with HPHE
C_p	specific heat capacity of air (J/kg · °C)
$C_{p,s}$	specific heat capacity of air (J/kg · °C)
D	diffusivity (m^2/s)

D_{exp}	experimental data
D_{calc}	calculated data
D_h	hydraulic diameter of FPHE (m)
d	heat pipe diameter (m)
$d_{a,in}$	air humidity ratio at the regenerator inlet (kg/kg.dry.air)
$d_{a,out}$	air humidity ratio at the regenerator outlet (kg/kg.dry.air)
$d_{eq,in}$	air humidity ratio when air status is equilibrium with inlet liquid desiccant (kg/kg.dry.air)
d_{hA}	hydraulic diameter of packing for air (m)
d_p	packing diameter (m)
$d_{p,min}$	the minimum packing diameter (m)
$d_{p,max}$	the maximum packing diameter (m)
f	friction factor
g	gravitational acceleration (m/s ²)
H	Henry's law constant (Pa)
h	corrugation height (m)
$h_{a,in}$	air enthalpy at the regenerator inlet (kJ/kg)
$h_{a,out}$	air enthalpy at the regenerator outlet (kJ/kg)
h_L	liquid holdup
h_{pb}	total packing bed height (m)
h_{pe}	packing element height (m)

K_G	overall mass transfer coefficient in the regenerator (m^2sPa)
k_a	convective mass transfer coefficient of air ($\text{kg}/\text{m}^2\text{sPa}$)
k_s	convective mass transfer coefficient of liquid desiccant solution ($\text{kg}/\text{m}^2\text{sPa}$)
L_m	mean length of the duct of FPHE (m)
m	packing element number
$\dot{m}_a$	mass flow rate of air (kg/s)
$\dot{m}_{a,nom}$	nominal mass flow rate of air (kg/s)
$\dot{m}_s$	mass flow rate of liquid desiccant solution (kg/s)
$\dot{m}_{re}$	regeneration rate (kg/s)
n	number of heat pipe row
$P_{h,preheated}$	water vapor pressure of air preheated by HPHE at the regenerator inlet (Pa)
$P_{h,sat}$	saturated water vapor pressure of air preheated by HPHE at the regenerator inlet (Pa)
$P_{f,preheated}$	water vapor pressure of air preheated by FPHE at the regenerator inlet (Pa)
$P_{s,in}^*$	equilibrium water vapor pressure of liquid desiccant solution at the regenerator inlet (Pa)
Q_f	heat recovery rate of FPHE (W)
Q_h	heat recovery rate of HPHE (W)
$Q_{f,max}$	the maximum heat recovery rate of FPHE (W)
$Q_{h,max}$	the maximum heat recovery rate of HPHE (W)

Q_{hr}	heat recovery rate (W)
Q_{fan}	energy consumption of fan without heat recovery device (W)
$Q_{fan,additional}$	additional energy consumption of fan caused by heat recovery device (W)
$Q_{fan,hr}$	energy consumption of fan with heat recovery device (W)
$Q_{fan,nom}$	nominal energy consumption of fan (W)
Q_{heater}	energy consumption of heater without heat recovery device (W)
$Q_{heater,hr}$	energy consumption of heater with heat recovery device (W)
Q_{pump}	energy consumption of pump (W)
Q_{re}	heat transfer rate during the regenerating process (W)
R	ideal gas constant
RE	relative error between experimental data and predicted data
Re	Reynolds number
S_v	transverse pitch of heat pipe (m)
s	corrugation side length (m)
$T_{h,preheated}$	temperature of air preheated by HPHE at the regenerator inlet (°C)
$T_{h,c,in}$	temperature of cool air at the HPHE inlet (°C)
$T_{h,w,in}$	temperature of warm air at the HPHE inlet (°C)
$T_{f,preheated}$	temperature of air preheated by FPHE at the regenerator inlet (°C)
$T_{f,c,in}$	temperature of cool air at the FPHE inlet (°C)
$T_{f,w,in}$	temperature of warm air at the FPHE inlet (°C)
$T_{s,in}$	temperature of solution at the regenerator inlet (°C)

$T_{s,heater,in}$	temperature of solution at the heater inlet (°C)
$T_{s,heater,out}$	temperature of solution at the heater outlet (°C)
u_{Ae}	effective air velocity (m/s)
u_{Le}	effective liquid velocity (m/s)
u_{As}	superficial gas velocity (m/s)
u_{Ls}	superficial liquid velocity (m/s)
V	volume of structured packing (m^3)
V_f	volume of FPHE (m^3)
ΔP	pressure drop (Pa)
α	corrugation inclination angle of structured packing
$\alpha_0 - \alpha_2$	fitted parameters
α_w	wetted specific surface area (m^2/m^3)
$\beta_0 - \beta_5$	fitted parameters
η_{re}	regenerating effectiveness
η_{th}	regenerating thermal efficiency
μ	viscosity of air (Pa S)
ρ	density of air (kg/m^3)
U	air velocity (m/s)
ε	heat recovery effectiveness

λ	latent heat of vaporization (kJ/kg)
γ	net heat recovery ratio
φ	relative humidity of air (%)
χ	energy saving ratio
ζ	flow resistance coefficient
δ	liquid film thickness (m)
σ	packing porosity

Subscripts

$1,2,3$	section 1, 2 and 3 of FPHE
AL	air-liquid film surface interaction
AA	air-air interaction at the crossing of air flow channel
DC	direction changes at the transitions between packing elements
a	air
c	cool air
d	condenser section of HPHE
e	evaporator section of HPHE
h	HPHE
hr	heat recovery
f	FPHE
eq	air status in equilibrium with liquid desiccant solution
re	regeneration
s	liquid desiccant solution

<i>p</i>	packing
<i>w</i>	warm air
<i>in</i>	inlet
<i>out</i>	outlet

Abbreviations

<i>FPHE</i>	fixed-plate heat exchanger
<i>HPHE</i>	heat pipe heat exchanger

CHAPTER 1 INTRODUCTION

1.1 Background and motivations

As a broad and complex topic, energy crisis is a concern to many researchers. Air Conditioning and Mechanical Ventilation (ACMV) equipment consumes a substantial portion of energy in commercial buildings [1]. Air humidity control is an important aspect of providing comfortable environment for human beings, especially in hot and humid areas. Up to now, most of the air conditioning systems in the world have been based on the conventional vapor compression system as shown in Figure 1. In order to remove the excess moisture from the outdoor air (state A), it has to be cooled down below its dew-point temperature and then reheated to ensure the supplied air (state D) meets the comfort requirement. This dehumidification system has the merits of high effectiveness of heat transfer, mature technology, compact size and convenient operation. However, there are shortcomings in this method. The condensed water makes the cooling coil work under wet condition, which may bring health problems since the wet coil surface tends to be a breeding ground for bacteria. More importantly, the cooling and reheating processes consume a large amount of electricity, the generation of which impairs the environment.

The dehumidification process from outdoor air (state A) to supplied air (state D) of conventional dehumidification system is depicted in a psychometric chart as shown in Figure 2. The outdoor air (state A) is cooled down to its dew point temperature (state B) firstly. The line A-B in Figure 2 represents this process in a simple and straightforward manner. To condense extra water from the outdoor air, the cooling process continues and follows the line B-C in Figure 2. To meet the comfort

requirement, the air is reheated by following the line C-D in Figure 2. The information regarding each air state is given in Table 1. If the mass flow rate of air is given, the total energy required in the conventional dehumidification process can be calculated by:

$$q = q_{cool} + q_{heat} = h_A - h_C + h_D - h_C = 44.3682 \text{ kJ} / \text{kg.d.a} \quad (1-1)$$

Figure 1 Conventional vapor compression dehumidification system

Figure 2 Psychrometric chart of dehumidification process

Table 1 Information of four states in dehumidification process in Figure 1

	Temperature (°C)	Relative humidity (%)	Enthalpy (kJ/kg.d.a)
A	30	70	78.7799
B	24	100	72.3389
C	15	100	42.3103
D	23	60	50.2089

Liquid desiccant dehumidification system (LDDS) is another way of air dehumidification. It uses the vapor pressure difference between process air and liquid desiccant as the driving force to achieve mass transfer, consequently, dehumidifying the process air. LDDS avoids the cooling and reheating processes of air. As shown in Figure 3, the dehumidifier and regenerator are two major components of LDDS where the heat and mass transfer take place. Cooler and heater are employed to provide cool and heat for liquid desiccant so that the mass transfer driving force of dehumidification and regeneration can be enhanced. Pump and fan create liquid and gas flow by consuming energy. The structured packing is filled in the dehumidifier and regenerator to increase the contact area between the air and the liquid desiccant.

Figure 3 Liquid desiccant dehumidification system

As shown in Figure 3, liquid desiccant solution is cooled by cooler to lower its water vapor pressure before entering the dehumidifier. In the dehumidifier, the outdoor air is cooled and dried continually by the low temperature liquid desiccant solution. The heat and mass transfer from the air to the liquid desiccant. The hot and humid outdoor air (state A) is cooled and dehumidified into supplied air (state D). The dehumidification process of LDDS is represented by line A-D in Figure 2. The outdoor air contacts with liquid desiccant and changes from state A to state D directly. Given the same mass flow rate of air, the total energy demand is:

$$q = h_A - h_D = 28.5710 \text{ kJ / kg.d.a} \quad (1-2)$$

The results of Eqs. (1-1) and (1-2) indicate that the dehumidification process of LDDS can save nearly 35.6% of energy compared with the conventional system. The results prove the potential of LDDS for energy saving.

Beside the great potential for energy saving, LDDS can also improve the indoor air quality (IAQ) [2]. It can remove harmful substances in the air, such as virus, bacteria and particulate matter (PM). The commonly-used liquid desiccants are salt solutions, such as LiBr solution, LiCl solution and CaCl_2 solution. The salt solutions have the ability to kill or inactivate the bacteria and virus. And the particulate matter tends to be captured by the liquid film or liquid droplets. Due to the superiorities of LDDS, it has drawn much attention.

As the dehumidification process continues, the liquid desiccant solution is diluted gradually due to the condensed water and its dehumidifying capacity is decreasing. In this case, the existence of regenerator in LDDS is inevitable. Its aim is the re-concentration of the liquid desiccant solution. The diluted liquid desiccant solution

from dehumidifier is pumped to the heater to absorb heat and thus achieve high water vapor pressure. Then it is sprayed into the regenerator. Since the temperature and water vapor pressure of liquid desiccant are both higher than those of air, the heat and moisture transfer from liquid desiccant to air. After the regeneration process, liquid desiccant solution is concentrated and the exhausted regenerating air is discharged to the environment directly.

For re-concentrating the liquid desiccant and using it repeatedly, the regenerator consumes significant amount of energy. The optimization of the regenerator is critical for reducing the amount of energy required to achieve high performance. Many researchers have developed novel regenerators with low energy consumption and high performance, such as internal heated regenerator [3] and electro dialysis regenerator [4]. Besides, the employment of low grade energy in regenerator as heat source, such as solar energy [5] and waste energy [6], has also been investigated. In this way, the energy-consumption type shifts from electricity toward renewable and cheaper fuels.

One method to reduce the energy consumption of the regenerator is through waste heat recovery. However, few attention has been paid on the study of waste heat recovery in regenerator. The exhausted regenerating air in Figure 3 has high temperature and humidity compared with ambient air. When the exhausted regenerating air is discharged directly, large amount of heat will be wasted. By a highly efficient heat recovery device, the waste heat in the exhausted regenerating air can be recycled rather than released to the environment. It can be reused to preheat the incoming air. With high temperature, the incoming air has high moisture

absorptive capability, which is beneficial for regeneration. There are four types of heat recovery devices developed by researchers [7], namely: heat pipe heat exchanger, fixed-plate heat exchanger, rotary wheel recovery and run-around heat recovery. The first three recovery devices have high heat recovery efficiency. However, rotary wheel needs extra power input. Run-round needs intermediate fluid and has the lowest heat recovery efficiency. Thus, heat pipe and fixed-plate heat exchangers highlight themselves in the application of waste heat recovery in regenerator.

For evaluating the regenerating performance of regenerator with heat recovery, experiments need to be conducted. In addition, numerical simulation is also necessary to describe the regenerating and heat recovery process. In order to monitor, predict and optimize the heat recovery process, a simple model is required. However, the existing models are very complicated and unsuitable for the performance estimation and optimization of heat recovery process in regenerator. It is necessary to develop model that is simple yet accurate and can be used to predict and optimize the heat recovery process in regenerator.

Except the energy requirement of regenerating process, the fan and pump also consume energy. The existence of heat recovery device and structured packing in regenerator induces sharp pressure drop due to significant flow resistance [8, 9], which may cause high consumption of fan energy. It is therefore of great essence to determine the best heat recovery device, the best packing size and the optimal air flow rate to achieve high regenerating performance as well as acceptable pressure drop.

1.2 Objectives

To fully utilize the potentials of LDDS, it is necessary to reduce the energy consumption of regenerator. With the purposes of addressing aforementioned issues, the in-depth investigation and optimization of regenerator with heat recovery are required. More specifically, the objectives of this study are:

- Analyze the regenerating and heat recovery performance of regenerator with heat recovery.
- Propose an appropriate model to predict the heat recovery process, which is accurate and simple for engineering practices.
- Acquire optimal results of design and operating parameters of regenerator with heat recovery considering regenerating performance and pressure drop.

1.3 Major contributions

The major contributions of this study include:

- Heat pipe heat exchanger and fixed-plate heat exchanger are selected to recover waste heat from exhausted air in regenerator since these two heat recovery devices can provide high heat recovery efficiency without additional power input.
- The hybrid models for heat recovery processes in regenerator with few unknown parameters are developed. They are derived from the physical governing equations and lump the complex geometric parameters and fluids' thermodynamic coefficients as constants since they have very small changes

during the process operation. The unknown parameters are identified by experimental data.

- A series of experiments and simulations are carried out to evaluate the regenerating performance in regenerator with heat recovery. The numerical simulations are based on the hybrid models developed previously. Performance indexes are defined to assess the regenerating and heat recovery performance. The experimental and numerical results are compared with each other.
- Multi-objective optimizations of regenerator with heat recovery are carried out. The regenerating rate and the total pressure drop are selected as two conflicting objectives. The most optimal values of design and operating parameters of the regenerator with heat recovery are revealed within their permission range.

1.4 Organization of the thesis

This thesis is organized as follow:

A literature review is presented in Chapter 2. This chapter also develops a motivation for this work. Chapter 3 presents the basic working principle of LDDS and a detailed explanation of the experimental platform employed in this study. Chapter 4 describes the hybrid models of the heat recovery devices in the regenerator. The models are validated by the experimental data and they can be used for energy analysis of regenerator. In Chapter 5, the performance of the regenerator with heat recovery is analyzed and compared with a traditional regenerator. Chapter 6 presents the

performance optimization of regenerator with heat recovery. Chapter 7 summarizes the primary conclusion from this work and suggests the future work.

CHAPTER 2 A RESEARCH REVIEW OF PREVIOUS STUDIES ON REGENERATOR OPTIMIZATION IN LDDS

2.1 Introduction

LDDS attracts much attention since it can save more energy compared with other dehumidification systems. Much research has focused on increasing its efficiency and decreasing its energy consumption. This has led to new designs for the dehumidifier and the regenerator and improved components selection for LDDS. The previous studies have been experimental and numerical validated. In this chapter, these studies are summarized.

2.2 New types of LDDS

After many years of development, different types of LDDS have been proposed to enhance the heat and mass transfer and reduce the energy consumption. These include internally cooled/heated dehumidifier/regenerator system, multi-stage dehumidification system, hybrid liquid desiccant dehumidification system, membrane-based liquid desiccant dehumidification system and ultrasonic atomization-based liquid desiccant dehumidification system.

Internally cooled/heated dehumidifier/regenerator can provide good performance of dehumidification and regeneration. Its schematic is shown in Figure 4. Except for the essential cooler and heater, cold fluid and hot fluid can provide additional cold and heat energy for the dehumidifying and regenerating process, which is helpful for keeping the operating temperature in dehumidifier and regenerator. Y.G. Yin et al. [10] designed a new internally-heated regenerator based on the plate-fin heat exchanger. Their conclusion showed that the internally-heated regenerator had

higher performance compared with an adiabatic regenerator. Besides, internally-heated regenerator can also provide better thermal performance. J. Liu et al. [11] designed an internally-cooled dehumidifier made by thermally conductive plastic. Experimental and numerical investigations have been carried out to analyze its performance. The results indicated that this plastic dehumidifier presents good dehumidifying performance.

Figure 4 Schematic of internally cooled/heated dehumidifier/regenerator

A multi-stage dehumidification system can achieve a high moisture removal rate and high heat transfer. As shown in Figure 5, multi-stage dehumidification installs several single dehumidifier in series, in which the liquid desiccant flows one by one and is cooled by separate cooler in each dehumidifier for achieving better performance. Z.Q. Xiong et al. [12] investigated a two-stage solar liquid desiccant dehumidification system. The pre-dehumidifier and the main dehumidifier used CaCl_2 and LiBr respectively. Their results found that the pre-dehumidification effect of CaCl_2 solution is remarkable when the ambient humidity is high. And the desiccant solution investment can be decreased by 53%. The thermal coefficient of

performance and the COP of the system could reach 0.97 and 2.13, respectively. Meanwhile, the authors also developed the exergy analysis based on the second thermodynamic law to study the thermal performance of the system [13]. The results revealed that the thermal coefficient of performance increased from 0.24 to 0.73. And the exergy efficiency increased from 6.8% to 23.0%. When the solution concentration of CaCl_2 and LiCl were at 40%, the useful energy storage capacity of them could reach 237.8 and 395.1 MJ/m^3 , respectively. M.M. Bassuoni [14] performed a theoretical analysis of two-stage air dehumidifier based on two feeding lines of liquid desiccant solution. The theoretical model was used to study different operating parameters. The results revealed that with the increasing inlet air humidity ratio, the respective average increases for both moisture removal rate and dehumidification effectiveness are around 48.9% and 49.2% compared to single stage dehumidifier. Their results implied the significant enhancement in dehumidifying performance by using the proposed system.

Figure 5 Schematic diagram of multi-stage air dehumidifier

Even though internally cooled/heated and multi-stage dehumidification systems can achieve higher mass transfer efficiency, their system arrangement is complex. Compared with them, hybrid liquid desiccant dehumidification system has a relatively simpler structure and lower cost. Y. Chen et al. [15] integrated vapor compression system with liquid desiccant dehumidification system. The condensation heat of vapor compression system is used for liquid desiccant regeneration. Liquid desiccant solution, whose concentration and temperature was both low, was used in the study. Theoretical and experimental analysis of the integrated system under different operating circumstance was conducted. The results revealed that the performance of the novel system was 13.6-116.1% more than conventional system. S. Yamaguchi et al. [16] evaluated a hybrid liquid desiccant air-conditioning system, which combined a traditional liquid desiccant dehumidification system with a vapor compression system. The liquid desiccant solution was lithium chloride. The refrigerant operating in the vapor compression heat pump was R407C. The results revealed that the COP of the system was 2.71 and it increased with the increasing compressor isentropic efficiency.

Membrane based liquid desiccant dehumidification system can resolve the carryover of liquid droplets by the air flow through permeating the water vapor while impermeating liquid desiccant. Nevertheless, membrane, as the core part of membrane dehumidification system, still has drawbacks, such as low permeability, low intensity and high cost, which limit the development of this technology. Zhang et al. [17] presented a compression heat pump driven membrane-based LDDS. The dehumidifier and regenerator are made of two hollow fiber membrane bundles

packed in two shells. A detail model of the heat and mass transfer processes in the membrane was developed and validated by the experiments. Their results showed that the EER of the heat pump was all larger than 3.75. The COP varied from 0.4 to 0.9, which implied the high energy performance of the dehumidification system. Das et al. [18] carried out experimental investigation for different kinds of membrane contactors operating with lithium chloride solution as liquid desiccant. The results showed that vapor flux up to $1300\text{ g}/\text{m}^2\text{h}$ can be obtained in the contactor with polypropylene membranes. However, the dehumidification effectiveness varied from 23% to 45%, which was rather low. The influence of the primary parameters on the performance of transmembrane vapor transport was also studied.

With the ultrasonic atomization technology, the liquid desiccant solution could be atomized into tiny droplets whose diameter are only about $50\mu\text{m}$. Tiny droplets of desiccant solution may largely promote the dehumidifying process since the contact area between liquid and gas is enlarged drastically. Yang et al. [19] pointed out that the tiny atomized droplets tend to be captured easily by the wall of dehumidifier tower, leading to poor performance of mass transfer. They proposed an optimizing dehumidifier chamber to enhance the efficiency of the ultrasonic atomization LDDS. Experiments have been conducted to prove the enhancement by evaluating the dehumidifying efficiency, liquid consumption rate and the air temperature enhancement. The results show that compared with conventional system, the average dehumidification effectiveness of the novel system increased dramatically from 20.78% to 66.5%. Besides, the amount of desiccant solution decreased from 0.4589kg/s to 0.2484kg/s . Moreover, they [20] also investigated the influence of the

inlet parameters of air and liquid desiccant on dehumidifying performances in the ultrasonic atomization LDDS experimentally. Compared with the cross flow packed dehumidifier, the desiccant solution consumption of ultrasonic atomization system reduced sharply from 1.5954kg/s to 0.4157kg/s.

2.3 Regenerator design

In LDDS, for re-concentrating the liquid desiccant and applying it repeatedly, regenerator consumes major energy to provide enough heat for liquid desiccant regeneration [21]. Qi et al. [22] built and tested a single channel internally heated regenerator. Experiments were conducted to investigate the keys that affect the wet area, liquid film thickness and mass transfer performance. Besides, a 3-D model, which was grounded on the experimental results, was developed. With this model, optimized designing as well as operating conditions can be predicted. Liu et al. [23] established a test-bed for investigate the efficiency of a cross flow regenerator. Two indexes, moisture removal rate and regenerator effectiveness, have been discussed. The influence of inlet parameters of air and solution have been compared experimentally. Besides, comparison between present regenerator and others was also carried out.

Since the requirement for energy grade of the regeneration process is not strict, low-grade energy, such as solar energy and waste heat, has attracted much attention. The first choice is solar energy. Typical solar regenerator system is shown in Figure 6. The diluted liquid desiccant is heated by solar thermal energy when it flows through solar collector and then it is sprayed into the regenerator and contacts with the regenerating air. Solar energy is dependent on the weather, thus back-up energy or

energy storage is necessary to ensure the dehumidifying efficiency when solar energy is not available. Waste heat can eliminate the cost. However, this type of energy is not suitable for small size dehumidification system since only factories or supermarkets can discharge waste heat, whose temperature is between 60°C and 140°C [6]. Alosaimy et al. [24] combines solar water heater with regenerator in LDDS and presented theoretical and experimental investigation. The solar water heater comprises a storage tank which is connected to an air-water heat exchanger. The importance of the storage tank was proved for achieving a steady regenerating performance. The results showed that the diluted solution with 30% concentration could be regenerated up to 50% by using solar energy. Cheng et al. [25] reviewed the recent theoretical and experimental works on solar thermal regeneration method and solar PV electro dialysis regeneration method. The results revealed that compared with the solar thermal regenerator, the solar PV electro dialysis regeneration system is more energy-efficient but expensive. For a PV system of 1kW load, the initial cost is nearly \$700. And the initial cost for a typical solar flat plate collector is below \$400.

Figure 6 Typical solar regeneration system

2.4 Structured packing

The regenerator and dehumidifier in LDDS are usually filled with structured packing, which can enhance the contact area for gas and liquid. However, it causes pressure drop. Besides, the packing geometries, liquid load and distribution affects the mass transfer process significantly. Researches have been conducted to study the packing geometry, liquid distribution on the packing surface and the total pressure drop in different types of structured packing.

Zakeri et al. [8] conducted several experimental studies to measure the pressure drop in 3 types of structured packed towers. The total pressure drop of the packing with certain height is function of liquid and gas flow rate. Yang et al. [26] tested eight structured packings with various geometrical parameters. The mass transfer performance for the absorption of CO_2 has been compared. The results showed that the packing geometrical parameters, load of liquid and F-factor of gas affected the effective packing area where mass transfer occurs. Authors also proposed a new

effective packing area based on the load of liquid and F-factor of gas, which agreed the experimental results well. A. Janzen et al. [27] used an ultrafast electron-beam X-ray tomography to investigate the liquid distribution and holdup in a column filled by structured packings. Two packings with different specific surface area were compared to investigate the influence of the packing geometry for the liquid distribution and liquid holdup. Olujic. Z. et al. [28, 29] investigated the liquid distribution in a conventional corrugated sheet structure packing experimentally. Their study showed that both gas load and liquid load didn't exert large influence on the liquid distribution. From a comparison with different experiments, it appeared that the liquid mal-distribution produces an effect on the pressure drop.

2.5 Numerical model of regenerator/dehumidifier

Wang et al. [30, 31] proposed simplified, yet accurate hybrid dehumidifier/regenerator heat and mass transfer model for predicting, control and optimizing the performance in near real-time as shown in Eqs.(2-1) and (2-2). These models lumped the fluids' thermodynamic coefficients and the geometrical parameters and as constants. Only three and four unknown parameters were needed respectively for the heat and mass transfer models during the dehumidifying and regenerating process. The experimental validation proves the effectiveness and accuracy of the proposed models. The hybrid models has been used for the optimization of liquid desiccant dehumidifier [30] and regenerator [31]. The genetic algorithm (GA) and multi-objective particle swarm optimization algorithm have been utilized respectively. Optimal desiccant solution temperature and flow rate were obtained for dehumidifier and regenerator. Compared with the conventional

strategies, the energy consumption can be reduced by 12.2% in the dehumidifying process and 8.55% in the regenerating process. Furthermore, Wang [32] optimized the operating condition in order to reduce energy consumption in LDDS. And the results showed that 18.5% energy could be saved by the optimized operating condition.

$$Q_{de} = \frac{C_1(m_s)^{C_3}}{1 + C_2\left(\frac{m_s}{m_a}\right)^{C_3}}(T_{a,in} - T_{s,in}) , \quad N_{de} = \frac{C_4(m_s)^{C_6}}{1 + C_5T_{a,in}(m_s)^{C_6}(m_a)^{C_7}}(P_{a,in} - P_{s,in}^*) \quad (2-1)$$

$$Q_{re} = \frac{C_1(m_a)^{C_3}}{1 + C_2\left(\frac{m_a}{m_s}\right)^{C_3}}(T_{s,in} - T_{a,in}) , \quad N_{re} = \frac{C_4(m_s)^{C_6}}{1 + C_5T_{a,in}(m_s)^{C_6}(m_a)^{C_7}}(P_{s,in}^* - P_{a,in}) \quad (2-2)$$

Langroudi et al. [33] used response surface methodology (RSM) to access the optimal results of six main operating variables related to the dehumidification mass rate, which can be predicted by a reduced quadratic statistical model. And the dehumidifying process was described by the effectiveness number of transfer unit (NTU) model. The simulated results agreed with the experimental results. Fakhrabadi et al. [34] optimized the design condition for a hybrid liquid desiccant regenerator. The simplified conjugated gradient method (SCGM) was used to identify the optimal point of design variables to maximize the performance. The effects of different design and operating conditions were also discussed for the optimal RCC. Yin et al [3] developed a regenerating model to evaluate the internally-

heated regenerator. The discussion mainly focused on the regenerating rate and regenerating thermal efficiency. Their results indicated that internally-heat regenerator performed better with larger regenerating rate and higher energy efficiency. Moreover, they [10] also explored and disclosed desiccant solution regenerator heated by hot air numerically by a validated model. The influences of main operating variables on regenerating rate and regenerating thermal efficiency were discussed.

2.6 Waste heat recovery

Energy saving technologies and their deployment has a great potentiality for large-scale reduction in energy demand. Kim [35] presented an economic analysis on energy saving technologies and his results indicated that with energy saving technologies, the total energy cost could be saved by 14%. Oh et al. [36] evolved an innovative dehumidifier based on adsorbent and employed indirect evaporative cooling technology. They concluded that this technology could save 21,096 GWh electricity until 2030. As one of the energy saving technology, waste heat recovery system can reutilize a fraction of waste heat. Instead of creating heat, this system reduces heat waste, which makes it more cost-effective. It can reuse the exhausting heat sources to preheat the incoming gas and hence, decrease the heating load and energy demand. Heat recovery system can recover about 60-95% heat from the waste heat and can dramatically increase the energy consumption efficiency in the buildings [7]. Riffat et al. [7, 9] made a review for the application of heat recovery systems in buildings in detail. The investigations on different heat recovery system were summarized. They categorized the heat recovery devices into four types

considering the construction type of heat exchanger, namely fixed-plate heat exchanger, heat pipe heat exchanger, rotary wheel recovery and run-around.

Fixed-plate heat exchanger (FPHE), as shown in Figure 7, has been investigated by many researchers. S. Anisimov et al. [37] presented theoretical analyses for plate heat exchanger employed for waste heat recovery. The heat recovery exchanger was operated under ice formation condition. The most unfavorable operating conditions were established. The impacting factors on temperature effectiveness were evaluated. O.P. Arsenyeva et al. [38] investigated the optimal design of plate heat exchanger. A mathematic model has been firstly developed. The number of passes and plates has been optimized. Besides, the plate type and size have been also discussed. The optimized plate heat exchanger exhibited better heat and hydraulic performance. S. Gendebien et al. [39] developed a model for air-to-air heat exchanger, which described both dry and partially wet regions. Experimental investigation has been conducted for validating the developed model. Effects of air humidity on the heat recovery rate and ways to eliminate freeze have also been studied. X.P. Liu et al. [40] employed a novel analysis way to minimize material cost and fan energy consumption at any given operating condition. Different geometric parameters have been considered. Based on their study, the manufacturing and operating cost can be comprehensively optimized.

Figure 7 Fixed-plate heat exchanger

Heat pipe heat exchanger (HPHE) had a variety of applications in heat recovery area in air condition dehumidification system. It contains a group of heat pipes. As shown in Figure 8, HPHE is divided into evaporator section and condenser section. High and low temperature fluid flows through the evaporator and condenser section of these arranged heat pipes respectively. By these heat pipes, the heat in high temperature fluid transfers to the low temperature fluid. Y.H. Yau [41] performed experiments to set up the baseline performance of heat pipe heat recovery, which was applied in dehumidification and air conditioning system. He also assembled a simulation model of empirical transient systems to predict the efficiency of heat pipe heat recovery [42]. Besides, an overall review [43] on the utilization of heat pipe heat exchanger in air conditioning under tropical weather has been carried out. Wang et al. [44] analyzed the secondary heat recovery efficiency for HPHE air condition system. They compared the newly HPHE air condition system with the traditional HPHE air condition system. The results indicated that heat recovery

efficiency of HPHE air condition system is 21.08% and 39.2% in winter and summer respectively and the secondary heat recovery HPHE air condition system exhibits better performance. Zhu et al. [45] designed and tested the micro heat pipe heat exchanger for heat recovery in residential buildings under normal temperature. Effect of air velocity, temperature difference of indoor and outdoor air and inclination angle of heat pipes were investigated experimentally. Their studies indicates that the heat recovery plays significant roles on the energy savings in residential building.

Figure 8 Heat pipe heat exchanger

Unlike FPHE and HPHE, the rotor of the rotary wheel recovery shown in Figure 9 is driven by motor so that the exhausted air and incoming air can pass through each section alternately. Yamaguchi et al. [46] proposed a mathematical model to analyze rotary desiccant wheels. Experimental results have been used to validate the proposed model. The effects of regenerating air temperature, velocity, thickness of wheel and rotational speed of wheel on the dehumidification performance were discussed. Angrisani et al. [47] pointed out that the rotational speed of the desiccant wheel was a crucial parameter of the desiccant wheel. They optimized rotational

speed under different operating conditions to achieve the best dehumidifying performance. The maximal dehumidification effectiveness fell into the range of 0.50-0.60. Bareschino et al. [48] developed a couple dehumidifying model for desiccant wheel, which considered both thermodynamic and transport properties of air. The model was validated by temperature and humidity results from both literature and experiments. O'Connor et al. [49] designed a novel desiccant rotary wheel. The CFD and experimental test results showed that the novel design has great potentials to enhance incoming air condition and reduce the energy cost. Their studies could achieve the reduction in relative humidity of air up to 55% while the pressure drop value is below 2Pa.

Figure 9 Rotary wheel recovery

Run-around heat recovery system, as presented in Figure 10, consists of two separated recuperative heat exchangers and they are linked by a third fluid. The two separated heat exchangers can be located in different place and the third fluid exchanges energy with hot fluid and cold fluid as shown in Figure 7. These features of run-around system shows its superiority over other heat recovery system when

cross contamination may occur. Vali et al. [50] studied and proposed a steady state model for the heat recovery process in a run-around heat recovery system with two counter/cross heat exchangers. The authors used finite difference method to solve the steady state model and the simulated results were verified with the literatures. The model could be used in performance prediction for engineering designs of run-around heat recovery systems. Wallin et al. [51] summarized the finding of run-around coil heat recovery system and compared three different run-around heat recovery systems. The results showed that with 2 stage heat pump retrofitted, the annual heat recovery rate could be increased by 18%. With variable speed capacity heat pump, the annual heat recovery rate improved from 47% to 66%. Lu et al. [52] proposed a simple and novel method, which was supported by the power law relationship, to assess the performance of run-around heat recovery systems. The air temperature efficiency could be predicted by this method. They also developed a tuning method to measure the power and computed the heat recovery effectiveness of the run-around heat recovery systems.

Figure 10 Run-around heat recovery system

2.7 Summary

This chapter reveals the current state-of-the-art of LDDS, which is an energy-saving as well as environmental-friendly dehumidification method. Researches focusing on system design, energy source, structured packing and model development had been carried out for enhancing the dehumidification performance and reducing the total energy consumption in LDDS. It is also observed that waste heat recovery technique has been widely used in conventional air conditioning and dehumidification system. Many types of heat recovery device have been explored by researchers.

The evaluation of the regenerating process in LDDS discovered that part of the waste heat can be recycled and reused rather than released to the environment. However, the application of waste heat recovery technique in regenerator has not drawn much attention. Insufficient research can be found in liquid desiccant dehumidification areas. Thus, this thesis studied waste heat recovery in regenerator of LDDS and the detailed discussions on the application of waste heat recovery in regenerator will be presented in next few chapters.

CHAPTER 3 WORKING PRINCIPLE AND PLATFORM DESCRIPTION

3.1 Introduction

In LDDS, regenerator re-concentrates the desiccant solution which is diluted by absorbing the excess moisture in the dehumidifier. Majority of energy is consumed by regenerator in the system operation in order to keep a suitable regenerating rate. In this study, to reduce the energy cost, heat recovery technique is employed in regenerator of liquid desiccant dehumidification system (LDDS) as it is a cost-effective way to save energy. Besides, to increase the air and liquid contact area inside the regenerator and meanwhile control the air pressure drop, structured packing is employed in this study. This chapter introduces the working principle of regenerator with heat recovery in LDDS. The experimental platform is also described.

3.2 Working principle of LDDS

Figure 11 is the schematic diagram of LDDS. Dehumidifier and regenerator are two main components of LDDS. In dehumidifier, the liquid desiccant solution is cooled by the cooler and is sprayed into the tower from the top. The outside air enters the dehumidifier tower from the bottom. They contact each other on the surface of structured packing. After dehumidification, the air becomes cool and dry and is supplied to the room. The liquid desiccant solution absorbs the extra moisture of air and becomes diluted. Regenerator can re-concentrate the solution and make sure its recirculation to dehumidifier. After regenerating process, the liquid desiccant drops down to the tank beneath regenerating tower and the warm regenerating air flows to

the heat recovery device, where the waste heat of regenerating air can be re-utilized to preheat the incoming fresh air.

Figure 11 Schematic diagram of LDDS

3.3 Working principle of heat recovery device

Compared with the four types of heat recovery devices introduced in Chapter 2, rotary wheel needs additional power input and run-round has the lowest heat recovery efficiency. Therefore, this study chooses fixed-plate and heat pipe heat recovery systems to recover waste heat in regenerator of LDDS.

3.3.1 Heat pipe heat exchanger (HPHE)

Heat pipe is high-efficient heat transfer device and thermal conduction and phase transition both exist in it. As Figure 12 shown, it consists of wick, sealed container and refrigerant which is in equilibrium state. Heat pipe composes three parts along its length: evaporator section, adiabatic section and condenser section. An external heat source transfers heat by conduction through pipe wall and wick to the evaporator section, where the working fluid vaporizes. The vapor flows to the condenser via adiabatic section driven by the vapor pressure. In the condenser section, the vapor

condenses and releases its latent heat to the heat sink. The condensed fluid is flowed back to the evaporator section driven by the capillary pressure generated by pipe wick. Heat pipe continuously transfers the latent heat from evaporator section to condenser section by conduction and phase transition. The fins outside the heat pipe will increase the heat transfer efficiency.

Figure 12 Schematic of the wick heat pipe

A group of heat pipes are arranged to form heat pipe heat exchanger and the central part is partitioned to divide the heat source and heat sink. High and low temperature fluid flows through the evaporator and condenser section of these arranged heat pipes respectively as shown in Figure 13. By these heat pipes, the heat in high temperature fluid transfers to the low temperature fluid. These heat pipes can be easily mounted and demounted, and, therefore, the efficiency of the heat exchanger can be varied as required. In LDDS, the HPHE is installed horizontally in the regenerator side to recovery the heat in the exhausted regenerating air and a cross configuration arrangement is adapted for near two rows heat pipes. Each row contains 8 heat pipes. In this study, the conventional type of heat pipe is utilized. The HPHE used in this study is shown in Figure 14. The specifications of HPHE are summarized in Table 2.

Figure 13 Schematic of heat pipe heat exchanger

Figure 14 Picture of heat pipe heat exchanger

Table 2 Specifications of HPHE in the system

Item	Type/Material/Number/Fluid/Range
Heat pipe	Conventional heat pipe
Fin pitch, height, thickness	2mm,10mm,0.2mm
Single heat pipe diameter, length	44mm,1000mm
Dimension of HPHE	1000×500×600 mm
Wall material	Aluminium
Total number of rows	4;8
Number of heat pipes per row	8

Working fluid of heat pipe	R134a
Wick structure of heat pipe	Axial grooves
Working temperature of heat pipes	30-80°C
Arrangement of heat pipe row	Cross configuration arrangement

3.3.2 Fixed-plate heat exchanger (FPHE)

The fixed-plate heat exchanger (FPHE) is consisted of a series of hexagonal parallel aluminium plates. The corrugated flow channel is formed by these parallel plates. It recovers heat from the exhausting regenerating air which would otherwise be wasted and reused it to preheat the incoming supply air. The two air streams are separated by hexagonal plates as shown in Figure 15. FPHE is divided into 3 sections, namely section 1, 2 and 3 as shown in Figure 16. The main region of FPHE (section 2) is in parallel-flow arrangement. The inlet and outlet regions (section 1 & 3) is in cross-flow arrangement. Compared with HPHE, the heat transfer area, i.e. heat recovery rate, of FPHE is fixed. The specifications of FPHE are summarized in Table 3.

(a) External frame

(b) Internal structure

Figure 15 Fixed-plate heat exchanger

Figure 16 Schematic of fixed-plate heat exchanger

Table 3 Specifications of FPHE in the system

Item	Number/Material/Range
Total length A	950mm
Total width B	420mm
Total height C	400mm
Plate pitch	5mm
Frame dimension	1050×500×470 mm
Wall material	Aluminium

3.4 Working principle of regenerator with heat recovery

The schematic diagram of regenerator with heat recovery in LDDS operating with lithium chloride is shown in Figure 17-18. Regenerator with heat recovery mainly consists of four components: regenerator tower, structured packing, heater and heat recovery device. Being as heat recovery device, the functions of HPHE and FPHE are the same. However the implementation mechanisms are different. For HPHE, the heat transfer between warm and fresh air is realized by the phase change of refrigerant inside the heat pipe. For FPHE, warm and fresh air is separated by metal plate and the thermal conductivity of metal is the main reason of the heat transfer.

Figure 17 Schematic diagram of regenerator with HPHE in LDDS

Figure 18 Schematic diagram of regenerator with FPHE in LDDS

The system operating procedure of regenerator with heat recovery device is briefly described below:

1. The ambient air (state C) is drawn into the regenerator by a fan as regenerating air. For HPHE, the air flows through the condenser section. For FPHE, the air flows through the one side of the corrugated channel. Preheated by the heat recovery device, the state of incoming ambient air is changed from state C to state D. The process is depicted as line C-D in the psychometric chart in Figure 19. The humidity ratio of the regenerating air is unaltered while its temperature rises, leading to lower relative humidity and higher moisture absorptive capacity. The preheated regenerating air (state D) enters the regenerator tower from the bottom;
2. The low concentration liquid desiccant is heated by the heater to obtain high temperature and high partial water vapour pressure firstly. After that, it is sprayed into the regenerator tower from the top;

3. The descending liquid desiccant forms liquid film on the structured packing surface while the ascending regenerating air flows over the liquid film. They contact with each other and both heat and mass transfer occurs:
 - a. Heat transfer is caused by temperature difference between liquid and air;
 - b. Mass transfer is driven by the water vapor pressure difference.

As liquid desiccant possesses higher water vapour pressure and higher temperature, its moisture and heat migrate towards the regenerating air. The heat and mass transfer processes in the regenerator are represented simply as line D-A in the psychometric chart in Figure 19;

4. During the regenerating process, the liquid desiccant is cooled and concentrated whereas the regenerating air is heated and humidified. Eventually, the descending liquid desiccant drops down to the tank beneath regenerator. The warm and humid ascending regenerating air (state A) is blown into the heat recovery device, i.e. the evaporator section of HPHE and other side of the corrugated channel of FPHE.
5. Through the heat recovery device, the warm regenerating air transfers its heat to the incoming regenerating air drawn from the environment. The results are the temperature increase of the incoming regenerating air from state C to state D and the temperature decrease of exhausted air from state A to state B. As shown in the psychometric chart in Figure 19, from state A to state B, with the invariable humidity ratio and reducing temperature, the relative humidity of exhausting air is increased.

Heat recovery device recovers the waste energy from exhausted regenerating air to preheat the incoming ambient air. Through this way, the energy waste is avoided and thus the energy utilization efficiency is increased. In addition, preheated regenerating air holds higher moisture absorptive capability, which can enhance the mass transfer efficiency in the regenerator.

Figure 19 Psychrometric chart of hear recovery process of regenerator

3.5 Structured packing

As a popular gas-liquid contacting device, structured packing has been widely used in distillation, absorption, CO₂ capture, dehumidification and many other applications. It consists of an alternating arrangement of corrugated sheets to form intersecting triangular flow channels, which expose large surface area for liquid and gas. With the well-defined geometrics, structure packing has high specific surface area and relatively low pressure drop. As shown in Figure 20, α is the corrugation angle, h_{pe} is the packing element height, b and h are the dimensions of the triangular flow channel.

Figure 20 Geometry of structured packing

Inside the regenerating tower, two adjacent packing elements are stacked together and rotated to each other by 90° , as shown in Figure 21. The rotation between two adjacent packing elements can provide large scale mixing effect for both liquid desiccant and regenerating air when they flow through the transition from layer to layer. However, the abrupt flow path change of regenerating air flow leads to considerable pressure loss.

Figure 21 Stacked structured packing

In this study, evaporative cooling pad, which is shown in Figure 22, has been applied to provide large contact area for air and liquid desiccant. Its packing element height is 200mm. Both dehumidifying and regenerating tower are filled with it. It is made of corrugate fiber paper by special chemical treatment to resist corrosion and to

service long life. The corrugated paper has dense structure and liquid can evenly cover its surface. The cross fluted and corrugated angle design promotes the mixing of air and liquid desiccant for optimum heat and mass transfer process. With these features, evaporative cooling pad exhibits high efficiency, high air velocity and easy maintenance.

Figure 22 Evaporative cooling pad

3.6 System platform description

A hybrid LDDS experimental test-bed, as depicted in Figure 23, is built to conduct experiments. To supply the driving energy for the system, double effect heat pump is integrated with this proposed system. The liquid desiccant solution is $\text{LiCl-H}_2\text{O}$ solution. The dehumidifying and regenerating column is made by polypropylene with a height of 2000mm and is filled with cylindrical structure packing with dimension of 500mm $\times$ 1600mm (diameter $\times$ height). In the column, the air and liquid desiccants interact with each other in a counter flow configuration. The descending solution is distributed by a liquid distributor installed in the top of the column. The ascending air is blown by axial flow fan at position 1 in Figure 23 and the gas flow is paralleled with the axial. This kind of fan is fit for high flow rate but

low pressure. Plate heat exchangers at position 2 in Figure 23 are applied as cooler and heater to provide thermal energy to dehumidifier and regenerator. Since the LiCl solution is corrosive, those two heat exchangers are made of titanium alloy. Plate heat exchangers can provide high heat exchange efficiency, small heat loss and long service life and its install and clean is easy. The heat exchange area is 3 m^2 . The heat recovery device is located at position 3 in Figure 23 in the regenerating side so that the exhausted high temperature regenerating air can flow through and the waste can be recovered to preheated the incoming supply air before it enters into the regenerating tower. Power cabinet and data acquisition system cabinet at position 4 and 5 in Figure 23 are the controllers and data acquisition system. The solution storage at position 6 stores the concentrated solution. The parameters that need to be used in calculation and validation, such as temperature, flow rate, humidity and density, have been measured by sensors installed in the experimental platform. Table 4 list the specifications of sensors. The results of these sensors have been obtained by the data acquisition system.

Figure 23 Experimental platform of LDDS

1- fan; 2 - plate heat exchanger; 3- heat recovery device; 4 - power cabinet ; 5 - data acquisition system cabinet ; 6 - solution storage tank.

In the experimental investigation, the range of air mass flow rate is 0.25-0.41kg/s and the test points are kept the same for simulations and experiments. The test No. of air flow rate is shown in Table 5. The rest input parameters remain the same for simulations and experiments. The experiments are conducted in an indoor lab in Singapore. The temperature and relative humidity of the fresh incoming regenerating air are around 25°C and 40% respectively. The liquid desiccant mass flow rate and inlet temperature are kept at 0.83kg/s and 50°C respectively.

Table 4 Specifications of sensors in the system

Sensors	Type	Model	Accuracy	Range
Solution temperature sensor	PT1000	WZPK-291	0.15°C	0-100°C
Solution flow meter	Magnetic flow meter	SDLDB-25ST2F102	±0.5%	0-50L/min
Air temperature sensor	Probe	EE210	0.1°C	0-60°C
Air humidity sensor	Probe	EE210	±0.5%	0-100%
Air flow meter	Pressure-gradient	C310-BO	±0.5%	0-1200m ³ /h
Power meter	Direct meter	CW240	±0.5%	0-3kW

Table 5 Test No. of air mass flow rate

No.	Air mass flow rate (kg/s)
Case 1	0.25

Case 2	0.27
Case 3	0.31
Case 4	0.35
Case 5	0.41

3.7 Summary

In this chapter, the working principle of regenerator and dehumidifier in liquid desiccant dehumidification system has been introduced. The main components of regenerator with heat recovery, i.e., regenerating tower, heat recovery device and structured packing have also been described, including their working principle and specification. Besides, the experimental platform of LDDS has been discussed in detail.

CHAPTER 4 HYBRID MODEL OF HEAT RECOVERY IN REGENERATOR

4.1 Introduction

As high efficient heat exchangers, HPHE and FPHE have been widely employed in air conditioning and conventional cooling based dehumidification systems for heat recovery. For clearly describe the heat transfer process, many models have been developed for heat exchanger, such as LMTD model, $\varepsilon - NTU$ model, finite difference model and hybrid model.

LMTD model: The LMTD model uses logarithmic temperature difference between hot and cold fluid at inlet and outlet of heat exchanger. Feldman et al. [53] used LMTD model to analyze HPHE. In this study, the performance of horizontal HPHE was examined including the temperature drop due to the vapor flow inside the heat pipes. They found that using staggered tube bundles, larger diameter of heat pipes, smaller fin heights and more fins number could improve the performance of HPHE. Wakiyama et al. [54] developed a LMTD model for an air-to-air HPHE. This study covered both horizontal and vertical operation of HPHE and had a good consistency with the experimental data. Claesson et al [55] investigated the applicability of LMTD approach in a compact brazed plate evaporator. The authors identified a set of equations and introduced a few simplifications. From their analysis, the suggestions on the correction factor under some simplified assumptions have been presented. The application of LMTD approach requires to know the overall heat transfer coefficient, whose determination is the most difficult part in the analysis of heat exchanger. Besides, the outlet temperature of the fluid streams is also required

to know in advance, which indicates the necessity of the iterative computation process based on a guessed initial temperature distribution.

$\varepsilon - NTU$ model: Kays and London [56] proposed $\varepsilon - NTU$ method and defined a scalar parameter named heat transfer effectiveness. It is the ratio of the actual to the maximum heat transfer rate. Browne et al [57] improved the $\varepsilon - NTU$ model and overcame the idealizations of previous models for the modelling of heat exchangers. They used an elemental approach to analyze the heat transfer process. The improved model allows the variable of heat transfer coefficient through the heat exchanger. Tan and Liu [58] used this model to study an air-to-air HPHE by neglecting the internal resistance of heat pipe. It was concluded that decreasing heat capacity ratio would increase the effectiveness of HPHE since more heat could be transferred to the low-temperature fluid. Soylemez [59] proposed a simple algebraic formula using method for HPHE. With this model, the optimal HPHE effectiveness, optimal heat recovery net savings and payback period could be determined. Raju et al [60] presented a closed form expression for $\varepsilon - NTU$ of a heat exchanger with unbalanced flow and longitudinal heat conduction. The accuracy of the proposed expression was verified by numerical results. The restrictions of the application of $\varepsilon - NTU$ are: (1) the heat transfer coefficient is very hard to determine since some key geometric information must be known in advance; (2) the application of $\varepsilon - NTU$ model for online optimization is difficult.

Finite difference model: Finite difference discretizes the computing domain and conducts the iterative computation in each infinitesimal element. During the simulation process, a fine discretization is required, which yields large models and

long computing times. This model is the most frequently-used model in the investigation of heat exchanger in recent years. Jung et al. [61] proposed a row-by-row heat transfer model which was useful for understanding the temperature distribution of each row and could be used to predict the cold-side inlet temperature of HPHE with counter flow. Han et al. [62] established finite difference heat transfer model of HPHE starting from $\varepsilon - NTU$ model. This model was employed to compute the heat flow of each row of heat pipe. B.R. Hughes et al [63] investigated ventilation streams in buildings and utilized heat pipe technology to recovery energy. Analytical model was developed from HPHE model and CFD simulation was conducted to predict the heat recovery rate. Mottaghy et al [64] presented an effective 3D finite difference expression for borehole heat exchanger that coupled heat and flow transport. By considering the heat transfer between fluid and soil, the expression reduced the simulation time significantly. Based on the finite difference simulation, the detailed heat transfer process in heat exchanger could be obtained. However, the deriving and resolving process is often very difficult. Besides, the iterative process usually consumes long times. The drawbacks limit the application of finite difference model in the online optimization.

Hybrid model: Cai et al. [65, 66] developed a simple and accurate hybrid model for heat exchanger operated with single phase fluid, e.g. cooling coil and cooling tower. The comparison results between the experimental data and the simulated data proved the accuracy and robustness of the hybrid model. With this model, it is easier to predict, monitor and control the heat transfer process of heat exchanger. Ding et al. [67, 68] extended the research and proposed the hybrid model for evaporator and

condenser operating with two phase flow. Wang et al [30, 31] proposed the heat and mass transfer hybrid model for dehumidifier and regenerator in LDDS. Started from the energy and mass balance, the model lumps the complex geometric parameters and fluids' thermodynamic coefficients as constants since they have very small changes during the process operation. The resulting model has limited unknown parameters. The hybrid model combines the advantages of both physical and empirical model method as it is based on fluid characteristics and thermodynamic principles and uses the experimental data to fit the unknown parameters which are difficult to measure directly. Compared with the existing models, the hybrid model is very simple, accurate, and does not require iterative computations. A large amount of experimental validation shows that the model is very effective to predict the performance in a wide operating range. The model is expected to find its applications in monitoring, control and optimization. In this thesis, we will employ this technique to develop hybrid model for heat recovery process of regenerator in liquid desiccant dehumidification system.

In this chapter, simple yet accurate hybrid model of heat recovery process in regenerator is proposed for performance monitoring and optimization. The proposed hybrid models are started from Effectiveness-NTU method, whose heat transfer coefficient is very hard to determine. The complex geometric and fluids' thermodynamic parameters have been lumped to unknown coefficients in the hybrid models. Compared with other existing models, the hybrid models don't need iterative computation and are easy to be applied in engineering.

4.2. Heat recovery model

Some assumptions have been made to simplify the derivation process: 1. The heat transfer loss to the ambient is neglected; 2. The heat recovery process is steady-state; 3. The specific heat capacity of air is constant in the heat recovery process; 4. The heat transfer coefficient is constant in the heat recovery process; 5. The air mass flow rate is constant; 6. The air thermal physical properties are constant under temperature variation.

HPHE and FPHE operate as heat recovery device in the study. Their heat transfer rate, which is the heat amount recovered from the exhausting regenerating air per second, is regarded as heat recovery rate. Similarly, the heat transfer effectiveness of HPHE and FPHE during the heat recovery process is considered as heat recovery effectiveness. It is the ratio of the actual to the maximum heat recovery rate [69]:

$$\varepsilon_h = \frac{Q_h}{Q_{h,\max}} = \frac{\max[(T_{h,w,in} - T_{h,w,out}), (T_{h,c,out} - T_{h,c,in})]}{T_{h,w,in} - T_{h,c,in}} \quad (4-1a)$$

$$\varepsilon_f = \frac{Q_f}{Q_{f,\max}} = \frac{\max[(T_{f,w,in} - T_{f,w,out}), (T_{f,c,out} - T_{f,c,in})]}{T_{f,w,in} - T_{f,c,in}} \quad (4-1b)$$

where ε_h , Q_h , $Q_{h,\max}$, $T_{h,w,in}$, $T_{h,w,out}$, $T_{h,c,in}$ and $T_{h,c,out}$ are the heat recovery effectiveness, actual heat recovery rate, the maximum heat recovery rate, the warm air inlet temperature, the warm air outlet temperature, the cool air inlet temperature and the cool air outlet temperature of HPHE, respectively. ε_f , Q_f , $Q_{f,\max}$, $T_{f,w,in}$,

$T_{f,w,out}$, $T_{f,c,in}$ and $T_{f,c,out}$ are the heat recovery effectiveness, actual heat recovery rate, the maximum heat recovery rate, the warm air inlet temperature, the warm air outlet temperature, the cool air inlet temperature and the cool air outlet temperature of FPHE, respectively. If the effectiveness is known, the actual heat recovery rate of HPHE and FPHE can be obtained by:

$$Q_h = \varepsilon_h Q_{h,max} = \varepsilon_h C_p \times \dot{m}_a (T_{h,w,in} - T_{h,c,in}) \quad (4-2a)$$

$$Q_f = \varepsilon_f Q_{f,max} = \varepsilon_f C_p \times \dot{m}_a (T_{f,w,in} - T_{f,c,in}) \quad (4-2b)$$

In Eq. (4-2), $\dot{m}_a$ is the mass flow rate of air; C_p is the specific heat capacity rate of flowing air.

4.2.1 Model development of HPHE

In HPHE, the effectiveness of evaporator and condenser section can be respectively written as [69]:

$$\varepsilon_{h,e} = 1 - \exp(-NTU_{h,e}) \quad (4-3a)$$

$$\varepsilon_{h,d} = 1 - \exp(-NTU_{h,d}) \quad (4-3b)$$

$NTU_{h,e}$ and $NTU_{h,d}$ are number of transfer units for evaporator section and condenser section respectively, which can be estimated from:

$$NTU_{h,e} = \frac{U_{h,e} S_{h,e}}{\dot{m}_a C_p} \quad (4-4a)$$

$$NTU_{h,d} = \frac{U_{h,d} S_{h,d}}{\dot{m}_a C_p} \quad (4-4b)$$

where $U_{h,e}$, $U_{h,d}$, $S_{h,e}$ and $S_{h,d}$ are the heat transfer coefficients of evaporator section, the heat transfer coefficients of condenser section, the convection heat transfer area of the evaporator section, the convection heat transfer area of condenser section, respectively. For an individual heat pipe, the effectiveness can be obtained from:

$$\varepsilon_{h,p} = \left(\frac{1}{\varepsilon_{h,\min}} + \frac{1}{\varepsilon_{h,\max}} \right)^{-1} \quad (4-5)$$

where $\varepsilon_{h,\min}$ and $\varepsilon_{h,\max}$ are the lower effectiveness and higher effectiveness between $\varepsilon_{h,e}$ and $\varepsilon_{h,d}$, respectively. For HPHE which consists of a number of rows, the effectiveness is determined by:

$$\varepsilon_h = \frac{n\varepsilon_{h,p}}{1 + (n-1)\varepsilon_{h,p}} \quad (4-6)$$

where n is number of heat pipe rows. For the force convection heat transfer between air flow and heat pipe, the heat transfer coefficient U is affected by the geometrical parameters of heat pipe and physical properties of air. Through dimensional analysis, the following equation has been proposed [70]:

$$\frac{UD}{K} = C \left(\frac{D\rho v}{\mu} \right)^f \left(\frac{C_p \mu}{K} \right)^g \quad (4-7)$$

where K is the thermal conductivity; D is the diameter of on heat pipe; ρ, v and μ are the density, velocity and viscosity of air, respectively; C , f and g are unknown coefficients. The air mass flow rate is:

$$\dot{m}_a = \rho v S \quad (4-8)$$

Substitute Eq. (4-8) into Eq. (4-7), yields

$$U = \frac{CK}{D} \left(\frac{4\dot{m}_a}{\pi\mu D} \right)^f \left(\frac{C_p\mu}{K} \right)^g = b_s \dot{m}_a^f \quad (4-9)$$

where

$$b_s = \frac{CK}{D} \left(\frac{4}{\pi\mu D} \right)^f \left(\frac{C_p\mu}{K} \right)^g$$

Substitute Eqs.(4-9) into (4-4), we obtain

$$NTU_{h,e} = \frac{U_{h,e} S_{h,e}}{\dot{m}_a C_p} = \frac{b_{h,se} \dot{m}_a^f S_{h,e}}{\dot{m}_a C_p} = b_{h,e} \dot{m}_a^{c_h} \quad (4-10a)$$

$$NTU_{h,d} = \frac{U_{h,d} S_{h,d}}{\dot{m}_a C_p} = \frac{b_{h,sd} \dot{m}_a^f S_{h,d}}{\dot{m}_a C_p} = b_{h,d} \dot{m}_a^{c_h} \quad (4-10b)$$

where $b_{h,e} = b_{h,se} S_{h,e} / C_p$; $b_{h,d} = b_{h,sd} S_{h,d} / C_p$, $c_h = f - 1$.

Combining Eqs. (4-5), (4-6) and (4-10), the effectiveness can be rewritten as:

$$\mathcal{E}_h = \frac{n \times \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}}{1 + (n-1) \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}} \quad (4-11)$$

Substituting Eq. (4-11) to (4-2a), the heat recovery rate is determined by:

$$Q_h = \frac{n \times \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}}{1 + (n-1) \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}} \times \dot{m}_a C_p (T_{h,w,in} - T_{h,c,in}) \quad (4-12)$$

In Eq. (12), $T_{h,w,in}$, $T_{h,c,in}$, $\dot{m}_a$ are inputs and they are varied in wide ranges. Q_h is output. $b_{h,e}$, $b_{h,d}$ and c_h are lumping parameters. When the lumping parameters in the hybrid model are identified, the heat recovery effectiveness can be obtained by Eq. (4-11), therefore the outlet temperature of the warm air exhausted to the environment and the outlet temperature of the cool air supplied to the regenerator inlet can be determined by:

$$T_{h,w,out} = T_{h,w,in} - \varepsilon_h \times (T_{h,w,in} - T_{h,c,in}) \quad (4-13a)$$

$$T_{h,c,out} = T_{h,c,in} + \varepsilon_h \times (T_{h,w,in} - T_{h,c,in}) \quad (4-13b)$$

4.2.2 Model development of FPHE

For FPHE, the flow arrangement in section 1 and section 3 is cross flow and the heat recovery effectiveness for these two parts is determined by Eq. (4-14a) [71]. In section 2, the flow pattern is parallel flow and the effectiveness is calculated by Eq (4-14b).

$$\varepsilon_{f,1} = \varepsilon_{f,3} = 1 - \exp \left[\left(NTU_{f,1}^{0.22} \right) \left\{ \exp \left[- \left(NTU_{f,1} \right)^{0.78} \right] - 1 \right\} \right] \quad (4-14a)$$

$$\varepsilon_{f,2} = \frac{1 - \exp \left[-2NTU_{f,2} \right]}{2} \quad (4-14b)$$

The overall effectiveness of the FPHE is given by [13]:

$$\varepsilon_f = \alpha_1 \varepsilon_{f,1} + \alpha_2 \varepsilon_{f,2} + \alpha_3 \varepsilon_{f,3} \quad (4-15)$$

where α_1 , α_2 and α_3 are the respective ratio of the heat recovery areas of section 1, 2 and 3 to the total area of FPHE.

NTU_f is the number of transfer units for FPHE, which can be estimated from:

$$NTU_{f,1} = NTU_{f,3} = \frac{U_{f,1} S_{f,1}}{\dot{m}_a C_p} \quad (4-16a)$$

$$NTU_{f,2} = \frac{U_{f,2} S_{f,2}}{\dot{m}_a C_p} \quad (4-16b)$$

where U_f and S_f are the heat transfer coefficients and the heat transfer area of the FPHE, respectively.

Based on Eq.(4-9), we obtain

$$NTU_{f,1} = NTU_{f,3} = \frac{U_{f,1} S_{f,1}}{\dot{m}_a C_p} = \frac{b_{f,s,1} \dot{m}_a^f S_{f,1}}{\dot{m}_a C_p} = b_{f,1} \dot{m}_a^{c_f} \quad (4-17a)$$

$$NTU_{f,2} = \frac{U_{f,2} S_{f,2}}{\dot{m}_a C_p} = \frac{b_{f,s,2} \dot{m}_a^f S_{f,2}}{\dot{m}_a C_p} = b_{f,2} \dot{m}_a^{c_f} \quad (4-17b)$$

where $b_{f,1} = b_{f,s,1} S_{f,1} / C_p$, $b_{f,2} = b_{f,s,2} S_{f,2} / C_p$, $c_f = f - 1$.

Combining Eqs. (4-14) to (4-17), the heat recovery effectiveness of FPHE can be rewritten as:

$$\varepsilon_f = 2\alpha_1 \left\{ 1 - \exp \left[\left(b_{f,1} \dot{m}_a^{c_f} \right)^{0.22} \left(\exp \left(- \left(b_{f,1} \dot{m}_a^{c_f} \right)^{0.78} \right) - 1 \right) \right] \right\} + \frac{\alpha_2}{2} \left[1 - \exp \left(b_{f,2} \dot{m}_a^{c_f} \right) \right] \quad (4-18)$$

In Eq. (4-18), $\dot{m}_a$ is given in advance. $b_{f,1}$, $b_{f,2}$ and c_f are lumping parameters.

The heat recovery rate of FPHE can be computed by:

$$Q_f = \varepsilon_f \times \dot{m}_a C_p (T_{f,w,in} - T_{f,c,in}) \quad (4-19)$$

where Q_f , $T_{f,w,in}$ and $T_{f,c,in}$ are the heat recovery rate of FPHE, warm air inlet temperature, and cool air inlet temperature, respectively. When the lumping parameters in the hybrid model are identified, the heat recovery effectiveness can be obtained by Eq. (4-18), therefore the outlet temperature of warm air exhausted to the environment and the outlet temperature of cool air supplied to the regenerator inlet can be determined by:

$$T_{f,w,out} = T_{f,w,in} - \varepsilon_f \times (T_{f,w,in} - T_{f,c,in}) \quad (4-20a)$$

$$T_{f,c,out} = T_{f,c,in} + \varepsilon_f \times (T_{f,w,in} - T_{f,c,in}) \quad (4-20b)$$

4.3 Parameter identification

Starting from the physical governing equations and lumping the complex geometric parameters and fluids' thermodynamic coefficients as constants, the proposed hybrid model only has three unknown parameters, which can be identified by the experimental testing data. In the identification process of the three lumping parameters, M points of experimental data, $\{y_{h,i}, x_{h,i}\}_{i=1}^M$ and $\{y_{f,i}, x_{f,i}\}_{i=1}^M$, are chosen

for identification, where

$$y_{h,i} = \{Q_h\}, x_{h,i} = \{T_{h,w,in}, T_{h,c,in}, \dot{m}_a\}, y_{f,i} = \{Q_f\}, x_{f,i} = \{T_{f,w,in}, T_{f,c,in}, \dot{m}_a\} \quad . \quad \text{The}$$

objective function is the sum of the square of the heat recovery rate residual between the model prediction and experimental test. Parameters identification can be viewed as optimization process. Find out the optimal values of three lumping parameters, which can minimize the objective function subject to constraints:

$$\begin{aligned} \min S_h(u_h) &= \sum_{i=1}^M r_{h,i}^2(u_h) \\ &= \sum_{i=1}^M \left[\frac{n \times \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1} \times \dot{m}_a C_p (T_{h,w,in} - T_{h,c,in})}{1 + (n-1) \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}} - Q_{h,exp,i} \right]^2 \\ \text{subject to : } &b_{h,d} > 0; b_{h,e} > 0. \end{aligned}$$

(4-21a)

$$\begin{aligned} \min S_f(u_f) &= \sum_{i=1}^M r_{f,i}^2(u_f) \\ &= \sum_{i=1}^M \left[2\alpha_1 \left\{ 1 - \exp \left[\left(b_{f,1} \dot{m}_a^{c_f} \right)^{0.22} \left(\exp \left(- \left(b_{f,1} \dot{m}_a^{c_f} \right)^{0.78} \right) - 1 \right) \right] \right\} \right. \\ &\quad \left. + \frac{\alpha_2}{2} \left[1 - \exp \left(b_{f,2} \dot{m}_a^{c_f} \right) \right] \times \dot{m}_a C_p (T_{f,w,in} - T_{f,c,in}) - Q_{f,exp,i} \right]^2 \\ \text{subject to : } &b_{f,1} > 0; b_{f,2} > 0. \end{aligned}$$

(4-21b)

In Eq.(4-21), $S(u)$ is the sum of the squares of the residuals between the experimental data and identification data; $r(u) = [r_1(u) \ r_2(u) \ \dots \ r_M(u)]^T$ is the

residuals between the experimental data and identification data; $u_h = [b_{h,d}, b_{h,e}, c_h]$ and $u_f = [b_{f,1}, b_{f,2}, c_f]$ are the parameter vectors to be identified; M is number of fitting points; $Q_{h,\text{exp},i}$ and $Q_{f,\text{exp},i}$ are experimental heat recovery rates and are calculated by Eqs. (4-1) and (4-2) using the experimental data.

Since these equations are non-linear, Levenberg-Marquardt method [72] has been used. The method incorporates a constant or variable scalar number to deal with problems related to singularity and is an effective algorithm for small residual problems. It blends steepest descent and Gauss-Newton method. When the parameters are far from their optimal value, this method acts more like steepest descent method. When the parameters are close to the optimal value, it acts more like Gauss-Newton method.

At the k th step of the iteration, the descent direction $h(u)$ is determined by the following equation:

$$\left(J^{(k)T}(u) J^{(k)}(u) + \lambda^{(k)} I \right) h^{(k)} = -J^{(k)T}(u) r^{(k)}(u) \quad (4-22)$$

where $\lambda^{(k)}$ is damping parameter in the k th iteration and is adjusted at each iteration.

I is the identity matrix of order 3. $J^{(k)}(u)$ is Jacobian matrix in the k th iteration whose entries are the gradient of $r^{(k)}(u)$ with respect to u :

$$J^k(u_h) = \begin{bmatrix} \frac{\partial r_{h,1}^{(k)}}{\partial b_{h,d}} & \frac{\partial r_{h,1}^{(k)}}{\partial b_{h,e}} & \frac{\partial r_{h,1}^{(k)}}{\partial c_h} \\ \frac{\partial r_{h,2}^{(k)}}{\partial b_{h,d}} & \frac{\partial r_{h,2}^{(k)}}{\partial b_{h,e}} & \frac{\partial r_{h,2}^{(k)}}{\partial c_h} \\ \vdots & \vdots & \vdots \\ \frac{\partial r_{h,M}^{(k)}}{\partial b_{h,d}} & \frac{\partial r_{h,M}^{(k)}}{\partial b_{h,e}} & \frac{\partial r_{h,M}^{(k)}}{\partial c_h} \end{bmatrix} \quad (4-23a)$$

$$J^k(u_f) = \begin{bmatrix} \frac{\partial r_{f,1}^{(k)}}{\partial b_{f,1}} & \frac{\partial r_{f,1}^{(k)}}{\partial b_{f,2}} & \frac{\partial r_{f,1}^{(k)}}{\partial c_f} \\ \frac{\partial r_{f,2}^{(k)}}{\partial b_{f,1}} & \frac{\partial r_{f,2}^{(k)}}{\partial b_{f,2}} & \frac{\partial r_{f,2}^{(k)}}{\partial c_f} \\ \vdots & \vdots & \vdots \\ \frac{\partial r_{f,M}^{(k)}}{\partial b_{f,1}} & \frac{\partial r_{f,M}^{(k)}}{\partial b_{f,2}} & \frac{\partial r_{f,M}^{(k)}}{\partial c_f} \end{bmatrix} \quad (4-23b)$$

In each iteration, if the reduction of $S(u)$ is rapid, λ is divided by a factor β ($\beta > 1$) to get a smaller value, bringing the algorithm closer to the Gauss–Newton algorithm, whereas if an iteration gives insufficient reduction in the residual, λ can be increased β times, giving a step closer to the gradient descent direction. The iteration ends if $|u^{(k+1)} - u^{(k)}| < \mu$, where μ is tolerance and is usually in the range of 1×10^{-6} to 1×10^{-5} . The identification process is shown as Figure 24.

Figure 24 Flow chart of the identification process

4.4. Heat recovery model validation

For the reason of evaluate the accuracy of this model, four error indexes are used, that is Relative Error (RE), Mean Relative Error (MRE), Root Mean Square Error (RMSE) and Standard Deviation Relative Error (STD_RE).

$$RE = \frac{|D_{\text{exp}} - D_{\text{calc}}|}{D_{\text{exp}}} \times 100\% \quad (4-24)$$

$$MRE = \frac{\sum_{i=1}^M RE_i}{M} \quad (4-25)$$

$$RMSE = \sqrt{\frac{\sum_{i=1}^M (D_{\text{exp}} - D_{\text{calc}})^2}{M}} \quad (4-26)$$

$$STD_{-RE} = \sqrt{\frac{\sum_{i=1}^M (RE_i - MRE)^2}{M-1}} \quad (4-27)$$

In these equations, D_{exp} and D_{calc} are the experimental and calculated data. The experimental heat recovery rate of HPHE and FPHE can be calculated by Eqs (4-1)-(4-2). The temperature and mass flow rate of air are measured by sensors introduced in Chapter 3.6.

4.4.1 HPHE hybrid model validation

30 data covering different operating condition has been collected from the experiments for the parameters identification. To validate the heat recovery model, 300 points in wide working ranges are tested. The air mass flow rate is range from 0.25 to 0.41 kg/s. The ambient air temperature ($T_{h,c,in}$) is around 22 to 25°C. The temperature exhausted from the regenerator outlet ($T_{h,w,in}$) is range from 36 to 47°C. The heat recovery rate is calculated from the experimental data by Eq. (4-2a) and its range is from 1.7 to 5.5 kW. Based on the identification process introduced in part 4, the results of the parameters for 4, 6 and 8 rows of heat pipes are presented in Table 6.

Table 6 Identification results for three heat pipe row arrangements

Heat pipe row n	$b_{h,d}$	$b_{h,e}$	c_h
4	0.6875	1.7137	0.0192
6	0.5876	5.3596	0.3070
8	6.5070	0.4597	0.5265

The comparison results of heat recovery rates between experimental data and theoretical data predicted by the 4, 6 and 8 rows hybrid models have been shown in Figures 25-27. It is found that under different heat pipe arrangement, the theoretical prediction results are consistent with the experimental data and 100% of 300 data points are falling within the relative error range of $\pm 5\%$. The largest relative error is 4.99% for all 4, 6 and 8 rows.

(a) Heat recovery rate

(b) Relative error

Figure 25 Model prediction of heat recovery rate and relative error (4 rows)

(a) Heat recovery rate

(b) Relative error

Figure 26 Model prediction of heat recovery rate and relative error (6 rows)

(a) Heat recovery rate

(b) Relative error

Figure 27 Model prediction of heat recovery rate and relative error (8 rows)

Figures 28-29 present the comparison between theoretical and experimental results of outlet warm air temperature and outlet cool air temperature when there are 4 rows of heat pipe. In Figure 28, all the relative errors are below 10% and 70.7% of which are below the relative error of 5%. The largest relative error is 9.02%. In Figure 29, it is found the theoretical data has a good consistency with the experimental data. All the relative errors of 300 fitting points are below 2%. The largest relative error is only 1.52%.

(a) Outlet warm air temperature

(b) Relative error

Figure 28 Model prediction of outlet warm air temperature and relative error (4 rows)

(a) Outlet cool air temperature

(b) Relative error

Figure 29 Model prediction of outlet cool air temperature and relative error (4 rows)

Figure 30 shows the heat recovery effectiveness results of 6 rows and 8 rows HPHE calculated from experiment data and predicted by the hybrid model. It is found that the theoretical results are consistent with the experimental results and the discrepancies between them are very small. The relative errors are all below $\pm 5\%$.

(a) 6 rows

(b) 8 rows

Figure 30 Model prediction of heat recovery effectiveness

The error indexes calculated from Eqs. (4-25)-(4-27) for heat recovery effectiveness, heat recovery rate, outlet warm air temperature and outlet cool air temperature are shown in Tables 7-9. These error indexes are all minuscule, which indicates the accuracy of the hybrid model. The model verification shows that the proposed hybrid heat recovery model is accurate enough for the applications in performance monitoring, control and optimization of the heat recovery process in the regeneration of LDDS.

Table 7 Error index for the performance prediction of HPHE (4 rows)

Error index	Heat recovery effectiveness	Heat recovery rate	Outlet warm air temperature	Outlet cool air temperature
MRE	2.58%	2.58%	2.04%	0.67%
RMSE	0.02	0.09	0.91	0.28
STD_RE	1.46	1.46	1.91	0.40

Table 8 Error index for the performance prediction of HPHE (6 rows)

Error index	Heat recovery effectiveness	Heat recovery rate	Outlet warm air temperature	Outlet cool air temperature
-------------	-----------------------------	--------------------	-----------------------------	-----------------------------

MRE	3.59%	3.59%	2.13%	0.97%
RMSE	0.03	0.12	0.89	0.36
STD_RE	1.22	1.22	2.12	0.34

Table 9 Error index for the performance prediction of HPHE (8 rows)

Error index	Heat recovery effectiveness	Heat recovery rate	Outlet warm air temperature	Outlet cool air temperature
MRE	2.11%	2.11%	3.51%	0.61%
RMSE	0.02	0.10	1.33	0.29
STD_RE	1.54	1.54	1.96	0.45

In order to present the general applicability of the proposed model, the experimental data in Ref. [73] has been used for the identification and the validation of the developed hybrid model. In Ref. [73], 4 row HPHE has been modelled by the finite difference method. The comparison results between experimental data, theoretical data by the finite difference model and the developed model have been shown in Figure 31. The consistent tendency of three categories heat recovery rate has been observed in Figure 31(a). Figure 31(b) displays the relative error of two kinds of theoretical data. For the finite difference model in Ref. [73], the largest and smallest relative errors of the hybrid model are 9.01% and 0.17% respectively. As for the developed hybrid model, the largest and smallest relative errors of the hybrid model are 18.21% and 0.06% respectively. The finite difference model provides accurater results than the hybrid model's results. However, the computing process of hybrid model is much simpler and faster than the finite difference model.

(a) Heat recovery rate

(b) Relative error

Figure 31 Comparison of hybrid model with Ref. [73]

4.4.2 FPHE hybrid model validation

The lumping parameters in the hybrid model of FPHE are identified by the 30 real-time experimental data covering different operating condition by using the Levenberg-Marquardt method. And the results are $b_{f,1} = -0.1091, b_{f,2} = 37.7022, c_f = -1.1371$. To validate the heat recovery model of FPHE, 300 points in wide working ranges are tested. The heat recovery rate is calculated from the experimental data by Eq. (4-2b) and its range is from 2.6 to 4.5

kW. The comparison results of heat recovery rate between experimental data and theoretical data predicted by the hybrid model have been shown in Figure 32. It is found that, the theoretical prediction results are consistent with the experimental data and 97% of 300 data points are falling within the relative error range of $\pm 5\%$. The largest relative error is 8.63%.

(a) Heat recovery rate

(b) Relative error

Figure 32 Model prediction of heat recovery rate and relative error

Figures 33-34 present the comparison between theoretical and experimental results of outlet warm air temperature and outlet cool air temperature. In Figure 33, all the relative errors are below 10% and 98% of which are below the relative error of 5%.

The largest relative error is 5.76%. For the outlet cool air temperature in Figure 34, it is found the theoretical data has a good consistency with the experimental data. 99.6% of the relative errors of 300 fitting points are below 5%. The largest relative error is only 5.32%.

The error indexes calculated from Eqs. (4-25)-(4-27) for heat recovery effectiveness, heat recovery rate, outlet warm air temperature and outlet cool air temperature are shown in Table 10. These error indexes are also very small. Thus the proposed hybrid model is proved to be effective for monitoring, control and optimization of heat recovery by FPHE in regenerator of LDDS.

(a) Outlet warm air temperature

(b) Relative error

Figure 33 Model prediction of outlet warm air temperature

(a) Outlet cool air temperature

(b) Relative error

Figure 34 Model prediction of outlet cool air temperature

Table 10 Error index for the performance prediction of FPHE

Error index	Heat recovery rate	Outlet warm air temperature	Outlet cool air temperature
MRE	1.51%	1.91%	0.71%
RMSE	0.02	0.97	0.35
STD_RE	1.38	0.64	1.87

4.5. Summary

In this chapter, the hybrid models for HPHE and FPHE used for heat recovery in regenerator of LDDS were developed. Starting from the physical governing

equations, the proposed hybrid model lumped complex geometric and fluids' thermodynamic parameters and only had three unknown coefficients. These three parameters were identified by the Levenberg-Marquardt method by using experimental data covering different operating condition. The identifying results were validated by 300 experimental fitting points. According to results of validation, the heat recovery effectiveness, heat recovery rate, outlet warm air temperature and outlet cool air temperature predicted by the model were consistent with the experimental data over a wide operating range. The hybrid models for HPHE and FPHE don't need iterative computation and can be used for performance monitoring and evaluation of heat recovery process in regenerator. It should be note that for long operating time, the unknown parameters in the hybrid models may change greatly and should be reevaluated periodically to make sure the accuracy of the models for the performance prediction.

The existence of heat recovery device increases the temperature of ascending regenerating air in regenerator, which will exert great impact on the heat and mass transfer process in the regenerator of LDDS. The energy saving by heat recovery devices and regenerating performance analysis of regenerator with heat recovery will be discussed in the following chapters.

CHAPTER 5 PERFORMANCE ANALYSIS OF REGENERATOR WITH HEAT RECOVERY

5.1 Introduction

To evaluate and compare the regenerator of Liquid Desiccant Dehumidification System (LDDS) without and with heat recovery, this chapter conducts performance analysis by hybrid heat transfer, mass transfer and heat recovery models, and the simulation results are validated by the experimental results. The heat recovery performance of HPHE and FPHE are compared to investigate the relationship between heat recovery rate and the additional fan energy consumption caused by the existence of heat recovery device. Effects of air mass flow rate on the regenerating and heat recovery performance are also discussed.

5.2 Modeling of Regenerator with heat recovery

Assumptions for the modeling of regenerator with heat recovery are: 1. Heat loss to the environment is negligible; 2. Liquid desiccant vaporization is neglected in the regenerator; 3. The heat and mass transfer processes in regenerator with heat recovery is steady-state; 4. The mass flow rate of regenerating air and liquid desiccant solution are constant in the regenerator with heat recovery; 5. The heat transfer coefficient is constant; 5. The thermal physical properties of the air and liquid are constant.

5.2.1 Modeling of regenerator with HPHE

The hybrid regenerating model, heat recovery model and fan model have been combined together to assess the performance of regeneration with heat recovery. Simple and accurate hybrid models for heat and mass transfer in regenerator and fan

energy consumption have been developed by the previous study [31, 74]. The hybrid heat recovery model for HPHE has been proposed in Chapter 4.

$$Q_{re} = \frac{c_{h,1} \dot{m}_a^{c_{h,3}}}{1 + c_{h,2} \left(\frac{\dot{m}_a}{\dot{m}_s} \right)^{c_{h,3}}} (T_{s,in} - T_{h,preheated}) \quad (5-1)$$

$$\dot{m}_{re} = \frac{c_{h,4} (\dot{m}_s)^{c_{h,6}}}{1 + c_{h,5} T_{c,out} (\dot{m}_s)^{c_{h,6}} (\dot{m}_a)^{c_{h,7}}} (P_{s,in}^* - P_{h,preheated}) \quad (5-2)$$

$$Q_h = \frac{n \times \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}}{1 + (n-1) \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}} \times \dot{m}_a C_p (T_{h,w,in} - T_{h,c,in}) \quad (5-3)$$

$$Q_{fan} = Q_{fan,nom} \left(a_{fan,3} \left(\dot{m}_a / \dot{m}_{a,nom} \right)^3 + a_{fan,2} \left(\dot{m}_a / \dot{m}_{a,nom} \right)^2 + a_{fan,1} \left(\dot{m}_a / \dot{m}_{a,nom} \right) + a_{fan,0} \right) \quad (5-4)$$

In Eq.(5-1), Q_{re} is the heat transfer rate in regenerator. $\dot{m}_s$ and $\dot{m}_a$ are the mass flow rate of liquid desiccant and air respectively. $T_{s,in}$ is the temperature of inlet liquid desiccant solution in the regenerator. $T_{h,preheated}$ is the preheated air temperature at the condenser outlet of HPHE, i.e. the regenerator inlet.

In Eq. (5-2), $P_{s,in}^*$ is the equilibrium water vapor pressure of the inlet liquid desiccant.

It can be calculated by:

$$P_{s,in}^* = \beta_0 + \beta_1 T_{s,in} + \beta_2 \omega_s + \beta_3 T_{s,in}^2 + \beta_4 \omega_s^2 + \beta_5 T_{s,in} \omega_s \quad (5-5)$$

where ω_s is the concentration of liquid desiccant solution. $\beta_0 \sim \beta_5$ are fitted parameters and their values are shown in Table 11.

Table 11 Fitted parameters of Eq. (5-5)

β_0	β_1	β_2	β_3	β_4	β_5
-2.6434273	0.20955349	5.2451548	0.0054591075	61.771904	-1.5411157

$P_{h,preheated}$ is the water vapor pressure of the inlet air in the regenerator. It can be determined by the relative humidity of air $\phi_{h,preheated}$:

$$P_{h,preheated} = \phi_{h,preheated} P_{h,sat} \quad (5-6)$$

where $P_{h,sat}$ is the saturated water vapor pressure of air and can be obtained from the following equation:

$$P_{h,sat} = \alpha_0 T_{h,preheated}^2 + \alpha_1 T_{h,preheated} + \alpha_2 \quad (5-7)$$

in which $\alpha_0 \sim \alpha_2$ are fitted parameters and their values are shown in Table 12.

$T_{h,preheated}$ is the temperature of air at the regenerator inlet and iterative computation is needed to determine this value.

Table 12 Fitted parameters of Eq. (5-7)

α_0	α_1	α_2
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13.099	-688.46	14009
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Eq. (5-3) is the hybrid heat recovery model of HPHE that is developed in Chapter 4. In this equation, $T_{h,w,in}$ is the warm air temperature at the evaporator inlet, i.e. the regenerator outlet. $T_{h,c,in}$ is the cool air temperature at the condenser inlet. n is the number of heat pipe row. C_p is the specific heat capacity of air. In these three equations, $c_{h,1} \sim c_{h,7}$, $b_{h,d}$, $b_{h,e}$ and c_h are the lump parameters that need to be identified by the operating data. In Eq. (5-4), Q_{fan} is the energy consumption of fan. $Q_{fan,nom}$ is the nominal fan energy consumption. $\dot{m}_{a,nom}$ is the nominal mass flow rate of regenerating air. $a_{fan,0} \sim a_{fan,3}$ is the unknown parameters. Based on the energy conservation, the energy supplied by the heater Q_{heater} is used to heat solution and this part of energy should be transferred to the enthalpy change of air:

$$Q_{heater} = C_{p,s} \dot{m}_s (T_{s,heater,out} - T_{s,heater,in}) = \dot{m}_a (h_{a,out} - h_{a,in}) \quad (5-8)$$

where $C_{p,s}$ is the specific heat capacity of liquid desiccant solution. $T_{s,heater,out}$ and $T_{s,heater,in}$ are the respective solution temperature at the heater outlet and inlet. $h_{a,in}$ and $h_{a,out}$ are the enthalpy of air at the inlet and outlet of regenerator respectively.

In the identification process, M points of experimental data, $\{y_i, x_i\}_{i=1}^M$, are chosen for identification, where $y_i = \{Q_{re}, \dot{m}_{re}, Q_h, Q_{fan}\}$, $x_i = \{T_{s,in}, T_{h,c,in}, \dot{m}_a\}$. The objective

function is the square of the residual between the model predicted and experimental test results. The optimal values of $c_{h,1} \sim c_{h,7}$, $b_{h,d}$, $b_{h,e}$, c_h and $a_{fan,0} \sim a_{fan,3}$ are identified to minimize the objective function subject to constraints:

$$\min S_h(U_h) = \sum_{i=1}^M R_{h,i}^2(U_h) = \begin{bmatrix} \sum_{i=1}^M r_{h,1,i}^2(u_{h,1}) \\ \sum_{i=1}^M r_{h,2,i}^2(u_{h,2}) \\ \sum_{i=1}^M r_{h,3,i}^2(u_{h,3}) \\ \sum_{i=1}^M r_{h,4,i}^2(u_{h,4}) \end{bmatrix} \quad (5-9)$$

subject to: $c_{h,1} > 0, c_{h,2} > 0, c_{h,4} > 0, c_{h,5} > 0, b_{h,d} > 0, b_{h,e} > 0$

where

$$r_{h,1,i}^2(u_{h,1}) = \left[\frac{c_{h,1} \dot{m}_a^{c_{h,3}} (T_{s,in} - T_{h,preheated}) - Q_{re,exp,i}}{1 + c_{h,2} \left(\frac{\dot{m}_a}{\dot{m}_s} \right)^{c_{h,3}}} \right]^2 \quad (5-10a)$$

$$r_{h,2,i}^2(u_{h,2}) = \left[\frac{c_{h,4} (\dot{m}_s)^{c_{h,6}} (P_{s,in}^* - P_{h,preheated}) - \dot{m}_{re,exp,i}}{1 + c_{h,5} T_{h,preheated} (\dot{m}_s)^{c_{h,6}} (\dot{m}_a)^{c_{h,7}}} \right]^2 \quad (5-10b)$$

$$r_{h,3,i}^2(u_{h,3}) = \left[\frac{n \times \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}}{1 + (n-1) \left(\frac{1}{1 - \exp(-b_{h,d} \dot{m}_a^{c_h})} + \frac{1}{1 - \exp(-b_{h,e} \dot{m}_a^{c_h})} \right)^{-1}} \right. \\ \left. \times \dot{m}_a C_p (T_{h,w,in} - T_{h,c,in}) - Q_{h,exp,i} \right]^2 \quad (5-10c)$$

$$r_{h,4,i}^2(u_{h,4}) = \left[Q_{fan,nom} \left(a_{fan,3} \left(\dot{m}_a / \dot{m}_{a,nom} \right)^3 + a_{fan,2} \left(\dot{m}_a / \dot{m}_{a,nom} \right)^2 + a_{fan,1} \left(\dot{m}_a / \dot{m}_{a,nom} \right) + a_{fan,0} \right) - Q_{fan,exp,i} \right]^2 \quad (5-10d)$$

In Eq. (5-9), $S_h(U_h)$ is the sum of the squares of the residuals between the experimental data and identification data; $R_h(U_h)$ is the residuals between the experimental data and identification data; U_h is the parameter vector to be identified and $u_{h,1} = [c_{h,1} \quad c_{h,2} \quad c_{h,3}]$, $u_{h,2} = [c_{h,4} \quad c_{h,5} \quad c_{h,6} \quad c_{h,7}]$, $u_{h,3} = [b_{h,d} \quad b_{h,e} \quad c_h]$, $u_{h,4} = [a_{fan,3} \quad a_{fan,2} \quad a_{fan,1} \quad a_{fan,0}]$; M is number of fitting points; $Q_{re,exp}$, $\dot{m}_{re,exp}$, $Q_{h,exp}$ and $Q_{fan,exp}$ are the experimental heat transfer rate, mass transfer rate, heat recovery rate and fan energy consumption of regenerator with HPHE respectively.

5.2.2 Modeling of regenerator with FPHE

The hybrid heat and mass transfer model developed by the previous study [31] has been combined with the hybrid heat recovery model of FPHE proposed in Chapter 4 to investigate the performance of regenerator with FPHE.

$$Q_{re} = \frac{c_{f,1} \dot{m}_a^{c_{f,3}}}{1 + c_{f,2} \left(\frac{\dot{m}_a}{\dot{m}_s} \right)^{c_{f,3}}} (T_{s,in} - T_{f,preheated}) \quad (5-11)$$

$$\dot{m}_{re} = \frac{c_{f,4} (\dot{m}_s)^{c_{f,6}}}{1 + c_{f,5} T_{c,out} (\dot{m}_s)^{c_{f,6}} (\dot{m}_a)^{c_{f,7}}} (P_{s,in}^* - P_{f,preheated}) \quad (5-12)$$

$$Q_f = 2\alpha_1 \left\{ 1 - \exp \left[\left(b_{f,1} \dot{m}_a^{c_f} \right)^{0.22} \left(\exp \left(- \left(b_{f,1} \dot{m}_a^{c_f} \right)^{0.78} \right) - 1 \right) \right] \right\} + \frac{\alpha_2}{2} \left[1 - \exp \left(b_{f,2} \dot{m}_a^{c_f} \right) \right] \times \dot{m}_a C_p (T_{f,w,in} - T_{f,c,in}) \quad (5-13)$$

$$Q_{fan} = Q_{fan,nom} \left(a_{fan,3} \left(\dot{m}_a / \dot{m}_{a,nom} \right)^3 + a_{fan,2} \left(\dot{m}_a / \dot{m}_{a,nom} \right)^2 + a_{fan,1} \left(\dot{m}_a / \dot{m}_{a,nom} \right) + a_{fan,0} \right) \quad (5-14)$$

Eqs. (5-11) and (5-12) are the heat transfer rate and mass transfer rate of regenerator with FPHE. These two equations are similar with Eqs. (5-1) and (5-2). $T_{f,preheated}$ is the preheated air temperature at the FPHE outlet, i.e. the regenerator inlet. $P_{f,preheated}$ is the water vapor pressure of the inlet regenerating air and can be calculated by Eq. (5-6) and (5-6). Eq. (5-13) is the hybrid heat recovery model of FPHE that is developed in Chapter 4, where $T_{f,w,in}$ and $T_{f,c,in}$ are the warm air inlet temperature and cool air inlet temperature of FPHE, respectively. In these three equations, $c_{f,1} \sim c_{f,7}$, $b_{f,1}$, $b_{f,2}$ and c_f are the parameters that need to be identified by the operating data.

In the identification process, M points of experimental data, $\{y_i, x_i\}_{i=1}^M$, are chosen for identification, where $y_i = \{Q_{re}, \dot{m}_{re}, Q_f, Q_{fan}\}$, $x_i = \{T_{s,in}, T_{f,c,in}, \dot{m}_a\}$. The objective function is the square of the residual between the model predicted and experimental test results. The optimal values of $c_1 \sim c_7$, $b_{f,1}$, $b_{f,2}$, c_f and $a_{fan,0} \sim a_{fan,3}$ are identified to minimize the objective function subject to constraints:

$$\min \quad S_f(U_f) = \sum_{i=1}^M R_{f,i}^2(U_f) = \left[\begin{array}{c} \sum_{i=1}^M r_{f,1,i}^2(u_{f,1}) \\ \sum_{i=1}^M r_{f,2,i}^2(u_{f,2}) \\ \sum_{i=1}^M r_{f,3,i}^2(u_{f,3}) \\ \sum_{i=1}^M r_{f,4,i}^2(u_{f,4}) \end{array} \right] \quad (5-15)$$

subject to: $c_{f,1} > 0, c_{f,2} > 0, c_{f,4} > 0, c_{f,5} > 0, b_{f,1} > 0, b_{f,2} > 0$

where

$$r_{f,1,i}^2(u_{f,1}) = \left[\frac{c_{f,1} \dot{m}_a^{c_{f,3}}}{1 + c_{f,2} \left(\frac{\dot{m}_a}{\dot{m}_s} \right)^{c_{f,3}}} (T_{s,in} - T_{f,preheated}) - Q_{re,exp,i} \right]^2 \quad (5-16a)$$

$$r_{f,2,i}^2(u_{f,2}) = \left[\frac{c_{f,4} (\dot{m}_s)^{c_{f,6}}}{1 + c_{f,5} T_{c,out} (\dot{m}_s)^{c_{f,6}} (\dot{m}_a)^{c_{f,7}}} (P_{s,in}^* - P_{f,preheated}) - \dot{m}_{re,exp,i} \right]^2 \quad (5-16b)$$

$$r_{f,3,i}^2(u_{f,3}) = \left[2\alpha_1 \left\{ 1 - \exp \left[(b_{f,1} \dot{m}_a^{c_f})^{0.22} \left(\exp \left(- (b_{f,1} \dot{m}_a^{c_f})^{0.78} \right) - 1 \right) \right] \right\} + \frac{\alpha_2}{2} \left[1 - \exp(b_{f,2} \dot{m}_a^{c_f}) \right] \times \dot{m}_a C_p (T_{f,w,in} - T_{f,c,in}) - Q_{f,exp,i} \right]^2 \quad (5-16c)$$

$$r_{f,4,i}^2(u_{f,4}) = \left[Q_{fan,nom} \left(a_{fan,3} (\dot{m}_a / \dot{m}_{a,nom})^3 + a_{fan,2} (\dot{m}_a / \dot{m}_{a,nom})^2 + a_{fan,1} (\dot{m}_a / \dot{m}_{a,nom}) + a_{fan,0} \right) - Q_{fan,exp,i} \right]^2 \quad (5-16d)$$

In Eq. (5-15), $S_f(U_f)$ is the sum of the squares of the residuals between the experimental data and identification data; $R_f(U_f)$ is the residuals between the

experimental data and identification data; U_f is the parameter vector to be identified

and $u_{f,1} = [c_{f,1} \ c_{f,2} \ c_{f,3}]$, $u_{f,2} = [c_{f,4} \ c_{f,5} \ c_{f,6} \ c_{f,7}]$, $u_{f,3} = [b_{f,1} \ b_{f,2} \ c_f]$,

$u_{f,4} = [a_{fan,3} \ a_{fan,2} \ a_{fan,1} \ a_{fan,0}]$; M is number of fitting points; $Q_{re,exp}$, $\dot{m}_{re,exp}$,

$Q_{f,exp}$ and $Q_{fan,exp}$ are the experimental heat transfer rate, mass transfer rate, heat

recovery rate and fan energy consumption of regenerator with FPHE respectively.

5.2.3 Parameter identification and simulation

The unknown parameters in Eqs. (5-1)-(5-4) and Eqs. (5-11)-(5-14) are identified by 30 experimental data covering different operating condition. The experimental regenerating rate $\dot{m}_{re,exp}$ and heat transfer rate during the regenerating process $Q_{re,exp}$ are calculated by the following Eqs (5-17)-(5-18). The experimental heat recovery rate of HPHE $Q_{h,exp}$ and FPHE $Q_{f,exp}$ are calculated by Eqs (4-1)-(4-2). The experimental fan energy consumption $Q_{fan,exp}$ is measured by power meter whose details are presented in Table 3. The temperatures, humidity ratio and mass flow rate of air used in Equations (4-1), (4-2), (5-17) and (5-18) are all obtained by sensors introduced in Chapter 3.6.

$$\dot{m}_{re,exp} = \dot{m}_a (d_{a,out} - d_{a,in}) \quad (5-17)$$

$$Q_{re,exp} = \dot{m}_a (h_{a,out} - h_{a,in}) - \lambda \dot{m}_{re} \quad (5-18)$$

The flowchart of the numerical simulation is shown in Figure 35. The unknown parameters of heat transfer model, mass transfer model, heat recovery model and fan

model are identified by the Levenberg-Marquardt algorithm [72] as introduced in Chapter 4. After obtaining the unknown parameters, iterative computation is conducted to get the performance comparison indexes of regenerator with heat recovery. In the iterative computation, the initial value of the warm air temperature at the regenerator outlet, i.e. inlet of heat recovery device, $T_{w,in}$, is assumed firstly as it is one of the input of heat recovery model and needs to be known in advance. With the assumed warm air temperature, the preheated air temperature at the regenerator inlet $T_{preheated}$ is predicted by the heat recovery model. And then based on the heat and mass transfer model of regenerating process, a new value of warm air temperature is predicted. The difference between the new warm air temperature and the former one is called residual and if the residual is less than ε , the iterative process is ended. Otherwise, the new warm air temperature is employed in the heat recovery model and another iteration is started.

Figure 35 Flowchart of the numerical calculation

5.3 Performance comparison indexes

Performance indexes are defined to assess the regenerating and heat recovery performance. Their values are calculated by the numerical model and compared with the experimental data. The Relative Error (RE), as defined in Eq. (5-19), is calculated to evaluate the discrepancies between the numerical and experimental results.

$$RE = \frac{|D_{\text{exp}} - D_{\text{calc}}|}{D_{\text{exp}}} \times 100\% \quad (5-19)$$

The environmental air firstly enters the heat recovery device and is preheated by recovery heat. The preheated air temperature $T_{preheated}$ and relative humidity $\phi_{preheated}$ reveal the heat recovery results and will impose great influence on the following regeneration. In the regenerator, the preheated air absorbs heat and mass from liquid desiccant. Three evaluating parameters, i.e. regenerating rate, regenerating effectiveness and regenerating thermal efficiency, are considered in this study to assess the strengthening performance of regeneration with heat recovery.

Regenerating rate $\dot{m}_{re}$ is the amount of water removed from the desiccant solution:

$$\dot{m}_{re} = \dot{m}_a (d_{a,out} - d_{a,in}) \quad (5-20)$$

where $d_{a,in}$ and $d_{a,out}$ are the humidity ratio of air at the inlet and outlet of regenerator respectively.

The regenerating effectiveness η_{re} is the ratio of the actual humidity ratio variance of the air passing through the regenerator to its ideal variance:

$$\eta_{re} = \frac{d_{a,out} - d_{a,in}}{d_{eq,in} - d_{a,in}} \quad (5-21)$$

where $d_{eq,in}$ is the humidity ratio of air which is in vapor pressure equilibrium with the liquid desiccant solution at the regenerator inlet.

Regenerating thermal efficiency η_{th} is defined to indicate the energy consumption efficiency for liquid desiccant regeneration:

$$\eta_{th} = \frac{\dot{m}_a (d_{a,out} - d_{a,in}) \lambda}{Q_h} = \frac{\dot{m}_a (d_{a,out} - d_{a,in}) \lambda}{\dot{m}_a (h_{a,out} - h_{a,in})} = \frac{(d_{a,out} - d_{a,in}) \lambda}{h_{a,out} - h_{a,in}} \quad (5-22)$$

where λ is latent heat of vaporization.

The heat recovery rate and heat recovery effectiveness are important indexes in evaluating heat recovery process. The heat recovery rates of HPHE and FPHE are respectively determined by:

$$Q_h = \varepsilon_h Q_{h,max} = \varepsilon_h C_p \times \dot{m}_a (T_{h,w,in} - T_{h,c,in}) \quad (5-23a)$$

$$Q_f = \varepsilon_f Q_{f,max} = \varepsilon_f C_p \times \dot{m}_a (T_{f,w,in} - T_{f,c,in}) \quad (5-23b)$$

The heat recovery effectiveness of HPHE and FPHE is computed by:

$$\varepsilon_h = \frac{Q_h}{Q_{h,max}} = \frac{\max[(T_{h,w,in} - T_{h,w,out}), (T_{h,c,out} - T_{h,c,in})]}{T_{h,w,in} - T_{h,c,in}} \quad (5-24a)$$

$$\varepsilon_f = \frac{Q_f}{Q_{f,max}} = \frac{\max[(T_{f,w,in} - T_{f,w,out}), (T_{f,c,out} - T_{f,c,in})]}{T_{f,w,in} - T_{f,c,in}} \quad (5-24b)$$

The application of heat recovery device should comprehensively consider the energy saving and the additional air pressure drop, i.e. additional fan energy consumption. The net heat recovery rate is the actual heat recovery rate subtracted by additional fan energy consumption. The net heat recovery ratio γ is the ratio of net heat recovery rate and total energy consumption of regenerator with heat recovery as shown below:

$$\gamma = \frac{Q_{hr} - Q_{fan,additional}}{Q_{fan,hr} + Q_{pump} + Q_{heater,hr}} \quad (5-25)$$

where Q_{hr} is the heat recovery rate of HPHE or FPHE. $Q_{fan,additional}$ is the additional fan energy consumption caused by the heat recovery device. $Q_{fan,hr}$ and $Q_{heater,hr}$ are the respective fan and heater energy consumptions when the heat recovery device is employed. Q_{pump} is the energy consumption of pump and this value is invariant whether the heat recovery device is applied or not.

The incoming regenerating air is preheated by the heat recovery device. With higher ascending air temperature, the sensible heat transfer between air and solution in regenerator is reduced, which decreases the energy demand of heater to reheat the solution. Therefore, the energy consumption of heater is reduced. Energy saving ratio χ is defined as Eq. (5-26) to quantify the energy save amount.

$$\chi = \frac{(Q_{fan} + Q_{heater}) - (Q_{fan,hr} + Q_{heater,hr})}{Q_{fan} + Q_{heater} + Q_{pump}} \quad (5-26)$$

where Q_{fan} and Q_{heater} are the respective fan and heater energy consumptions when the heat recovery device is not employed.

5.4 Results and discussion

The identifying results in Eqs. (5-1)-(5-4) and Eqs. (5-11)-(5-14) are presented in Table 13-16. With the identified parameters, the regenerating and heat recovery processes of regenerator with heat recovery can be investigated numerically. The

numerical results are compared with the experimental results. The uncertainties of the experimental results are calculated and the error bars are given in the following figures. The accuracy of the numerical model is presented and analyzed in the following part.

Table 13 Identified results of model parameters for regenerator without heat recovery

$c_{h,1} / c_{f,1}$	$c_{h,2} / c_{f,2}$	$c_{h,3} / c_{f,3}$	$c_{h,4} / c_{f,4}$	$c_{h,5} / c_{f,5}$	$c_{h,6} / c_{f,6}$	$c_{h,7} / c_{f,7}$
0.1932	0.8740	0.3196	1.2126	0.8684	1.0009	-0.7920

Table 14 Identified results of model parameters for regenerator with HPHE

	$c_{h,1}$	$c_{h,2}$	$c_{h,3}$	$c_{h,4}$	$c_{h,5}$	$c_{h,6}$	$c_{h,7}$	$b_{h,d}$	$b_{h,e}$	c_h
4	0.6293	0.5018	0.6383	1.2220	0.8389	1.0010	-0.7654	6.1401	0.5094	0.4777
8	0.2914	0.7944	0.3736	1.2326	0.8275	1.0009	-0.7767	0.4057	0.3976	0.3081

Table 15 Identified results of model parameters for regenerator with FPHE

	$c_{f,1}$	$c_{f,2}$	$c_{f,3}$	$c_{f,4}$	$c_{f,5}$	$c_{f,6}$	$c_{f,7}$	$b_{f,1}$	$b_{f,2}$	c_f
FPHE	0.5936	0.5448	0.5716	1.2572	0.8146	1.0009	-0.7577	1.7288	1.7309	1.3247

Table 16 Identified results of fan model parameters

	$a_{f,3}$	$a_{f,2}$	$a_{f,1}$	$a_{f,0}$
Without heat recovery	0.0629	868.5680	1.0000	16.2636
4 rows HPHE	0.0760	912.9687	1.0000	111.7901
8 rows HPHE	0.0861	996.8390	1.0000	221.0210
FPHE	0.0016	1.1405	1.0000	21.2871

Figure 36 presents the regenerator inlet air temperature and relative humidity with the increasing air flow rate. It is clear that compared with HPHE, FPHE can provide

higher regenerator inlet air temperature and lower relative humidity. And for HPHE, more rows heat pipe can generate higher regenerator inlet air temperature and lower relative humidity. As the air flow rate increases, more waste heat is recovered, thus the preheating air temperature raises with the increasing air flow rate. As presented in Figure 36(a), FPHE, 4 row HPHE and 8 rows HPHE can raise the air temperature from 25°C to around 38°C, 34°C and 35.5°C, respectively. Since the humidity ratio of air is invariant, the relative humidity decreases with the increasing air temperature as shown in Figure 36(b). The air relative humidity is reduced from 40% to nearly 28%, 33.5% and 31.5% by FPHE, 4 rows HPHE and 8 rows HPHE, respectively. Figure 36 indicates that the discrepancy between the predicted and experimental results is small. The respective largest REs for air temperature and relative humidity are only 1.09% and 2.06%.

(a) Temperature $T_{preheated}$

(b) Relative humidity $\varphi_{preheated}$

Figure 36 Regenerator inlet air state with increasing air flow rate

Figure 37 reflects the variation trend of regenerating rate with the increasing regenerating air mass flow rate. For case 1-5, the regenerating rate of regenerator with heat recovery outnumbers that of regenerator without heat recovery. Based on Figure 36, the existence of heat recovery device increases the temperature and decrease the relative humidity of incoming air, which implies higher moisture absorption capability. Thus, the existence of heat recovery improves the regenerating performance. Since FPHE can recover more heat compared with HPHE, its employment can provide better regenerating performance. Based on the results in Figure 37, the regenerating rate of regenerator with 4 rows HPHE is increased by 11.51% maximally compared with regenerator without HPHE. And that values for regenerator with 8 rows HPHE and FPHE are 17.12% and 31.26%.

In Figure 37, with the increasing air mass flow rate, the regenerating rate follows the same trend for regenerator without heat recovery and with heat recovery. It decreases firstly and hits its lowest value at Case 2. Afterwards, it starts to increase with the

upward air mass flow rate. Higher air mass flow rate could enhance the mass transfer coefficient between air and desiccant. However, higher air mass flow rate would absorb more sensible heat from the liquid desiccant, which leads to faster reduction of mass transfer driven force. Therefore, the effect of air mass flow rate on the regeneration combines the variation of both mass transfer coefficient and driven force. High air mass flow rate offers high mass transfer coefficient as well as low driven force. Figure 37 reveals that for Case 1 and Case 2, the effect of decreasing driven force dominates, thus the total regenerating rate is lowered. Conversely, for the Case 3, 4 and 5, the regenerating rate increases because the effect of increasing mass transfer coefficient overwhelms the decreasing driven force.

Figure 37 Regenerating rate ($\dot{m}_{re}$) with increasing air mass flow rate

It can be drawn from Figure 38 that for cases 1-5, regenerator with heat recovery present higher regenerating effectiveness than that of regenerator without heat recovery. When the air mass flow rate is low, the regeneration effectiveness of regenerators with FPHE, 4 rows HPHE and 8 rows HPHE are almost close. As the flow rate arising, regenerating effectiveness goes down, though the falling speed of

the regenerator with FPHE is slower than those of regenerator without heat recovery and with HPHE. The regenerating effectiveness of regenerator is improved by 11.38% when it works with 4 rows HPHE, by 15.83% with 8 rows HPHE and by 18.51% with FPHE to the largest extent. The regenerating effectiveness follows a downward trend with the upward air mass flow rate. This value is only affected by the air humidity ratio at the regenerator outlet. As the air flow rate increases, the mass transfer coefficient increases while the vapor pressure difference decreases, leading to the reduction of the air humidity ratio at the regenerator outlet.

Figure 38 Regenerating effectiveness (η_{re}) with increasing air mass flow rate

Figure 39 Regenerating thermal efficiency (η_{th}) with the increasing air mass flow rate

Figure 39 shows the regenerating thermal efficiency under different air mass flow rate. It is found that the thermal efficiency decreases with the increasing regenerating air flow rate. The latent heat and the heater energy consumption increase simultaneously when air flow rate raises. However, the increment of heater energy overwhelms that of latent energy, causing the reduction of thermal efficiency. As Figure 39 depicts, the presence of heat recovery slows down the drop rate of thermal efficiency. Regenerator with FPHE exhibits the highest as well as the most stable regenerating thermal efficiency. The first reason is the regenerator with FPHE has high regenerating rate as shown in Figure 37, which means the latent heat is high as well. The second reason is the decrease of the energy demand of heater. Since the ascending regenerating air is preheated by FPHE, it absorbs less sensible heat from the solution. The solution temperature maintains high in the regenerating process, leading to stable mass transfer process. Besides, the heat demand of heater is also reduced in the following reheating process.

After the heat and mass interaction with liquid desiccant, the regenerating air flows through the heat recovery device with high temperature and high humidity ratio. The heat recovery device recovers part of thermal energy from the regenerating air. After being recovered, the thermal energy of air is reduced. And it is directly released to the environment. The results of heat recovery rate and effectiveness with increasing regenerating air mass flow rate are shown in Figures 40-41. As depicted in Figure 40, the heat recovery rate goes up proportionally to the increasing air flow rate, while the heat recovery effectiveness raises slightly. Compared with 4 rows HPHE, 8 rows

HPHE and FPHE raises the heat recovery rate by 10.61% and 14.65% respectively in case 2 and they raise the effectiveness by 5.49% and 11.89% respectively in case 4. Figures 40-41 prove that for HPHE, more heat pipe rows can recover more energy with higher effectiveness. However the existence of heat pipe adds resistance to air flow and induces additional fan energy consumption. Balance decision has to be made for compromising the heat recovery rate and fan energy consumption.

Figure 40 Heat recovery rate Q_{hr} with increasing air mass flow rate

Figure 41 Heat recovery effectiveness ε with increasing air mass flow rate

Figure 42 presents the net heat recovery ratio of HPHE and FPHE. It is found that nearly 14-19% of the total regenerator energy consumption can be recovered by HPHE and FPHE. When the air flow rate is low, the net heat recovery ratio of 4 rows HPHE is much lower than that of 8 rows HPHE. As the air flow rate increases, the additional pressure drop caused by heat pipe increases and the net heat recovery ratios of regenerator with 8 rows HPHE gradually decreases while that value gradually increases for regenerator with 4 rows HPHE. When the air mass flow rate reaches 0.41kg/s, the net heat recovery ratio of regenerators with 4 and 8 rows HPHE are almost the same. As for FPHE, it provides the highest heat recovery rate as well as pressure drop. At case 1, its net heat recovery ratio nearly equals to 8 rows HPHE's. When the air flow rate increases, the heat recovery rate and pressure drop of FPHE are both increasing. As Figure 42 shows, the neat heat recovery ratio of FPHE decreases firstly and then increases. At case 5, this value of FPHE is larger than 8 rows HPHE's.

Figure 43 indicates that the application of heat recovery in regenerator can save nearly 5-19% energy compared with regenerator without heat recovery. When the air flow rate is lower than 0.31kg/s, the energy saving ratio gradually increases for both 4 rows HPHE and 8 rows HPHE. However, when the air flow rate is larger than 0.34kg/s, the energy saving ratio drops down dramatically for regenerator with 8 rows HPHE. Contrarily, it still increases slightly for the regenerator with 4 rows HPHE. When the air flow rate comes to 0.41kg/s, the energy saving ratio of regenerator with 4 and 8 rows HPHE nearly equals to each other. At case 1 and case

5, the energy saving ratio of regenerator with FPHE is highest compared with regenerator with HPHE.

The analyses of both net heat recovery ratio and the energy saving ratio for regenerator with heat recovery indicate that at case 1, the employment of FPHE in regenerator exhibits the best performance in terms of regeneration and heat recovery as well as energy consumption. When the air flow rate is not that high, the application of 8 rows HPHE becomes the best choice compared with 4 rows HPHE and FPHE. However, with the continuous growth of air flow rate, the superiority of 8 rows HPHE diminished gradually due to the high air pressure drop and low heat recovery rate. And at case 5, FPHE becomes the preferable choice compromising the regenerating performance as well as pressure drop.

Figure 42 Net heat recovery ratio γ with increasing air mass flow rate

Figure 43 Energy saving ratio χ with increasing air mass flow rate

The predicted results by the proposed model in Figures 37-43 show good agreement with the experimental data. Tables 17-19 summarize all the REs between the predicted values and the experimental data. It is found that for all the performance indexes, the REs are all below 11% and the largest RE is 10.87%. The low RE reveals that the hybrid regenerating and heat recovery model is accurate and effective to predict the regenerating and heat recovery performance of regenerator with heat recovery.

Table 17 RE of regenerating rate, effectiveness and thermal efficiency (%)

	Case 1			Case 2			Case 3			Case 4			Case 5		
	$\dot{m}_{re}$	η_{re}	η_{th}	$\dot{m}_{re}$	η_{re}	η_{th}	$\dot{m}_{re}$	η_{re}	η_{th}	$\dot{m}_{re}$	η_{re}	η_{th}	$\dot{m}_{re}$	η_{re}	η_{th}
0	6.59	2.42	7.02	3.66	3.18	5.98	2.75	1.29	8.36	4.65	4.48	8.12	2.79	3.55	6.41
4	2.94	0.42	1.63	0.98	0.31	6.26	0.05	1.75	4.13	0.06	1.82	6.56	2.62	3.39	2.84
8	5.72	2.91	0.84	2.18	2.61	0.43	0.24	1.35	1.59	0.62	3.56	0.65	4.36	1.57	2.12
F	9.77	0.21	1.09	0.93	0.76	1.06	0.11	0.12	0.82	2.32	1.02	1.19	2.09	1.14	5.59

Table 18 RE of heat recovery rate and effectiveness (%)

	Case 1		Case 2		Case 3		Case 4		Case 5	
	Q_{hp}	ε	Q_{hp}	ε	Q_{hp}	ε	Q_{hp}	ε	Q_{hp}	ε
4	5.72	2.57	0.38	1.71	4.04	1.66	7.04	4.81	7.67	4.37
8	9.59	8.73	3.15	3.05	3.98	4.21	4.43	4.01	3.32	2.28
F	6.54	2.47	4.93	2.08	3.16	2.03	4.85	3.84	2.87	2.28

Table 19 RE of net heat recovery ratio and energy saving ratio (%)

	Case 1		Case 2		Case 3		Case 4		Case 5	
	γ	χ	γ	χ	γ	χ	γ	χ	γ	χ
4	3.37	8.97	3.67	7.36	3.34	10.87	2.59	9.71	3.88	6.54
8	3.11	10.46	3.59	6.79	1.88	9.63	0.75	5.49	1.84	10.60
F	10.03	8.22	8.05	1.01	6.09	0.68	8.59	8.95	2.68	7.42

In order to consider the comprehensive performance improvement of regenerator with HPHE and FPHE, the numerical computation is conducted by the proposed model to investigate the heat recovery and energy saving when the regenerating rate maintains the same for regenerator with 4 rows HPHE, 8 rows HPHE and FPHE. Supposing the humidity ratios of the inlet and outlet regenerating air are 12g/kg and 30g/kg respectively, the numerical predicted results of net heat recovery ratio and energy saving ratio are given in Figure 44 (a) and (b). When the regenerating rate keeps the same, 4 rows HPHE can recover 25% from the total energy consumption maximally. Accordingly, 26.5% of the energy usage can be reduced compared with regenerator without heat recovery. For 8 rows HPHE and FPHE, the maximal reutilization of 26.5% and 27% energy contributes to almost 27% and 26.9% of the energy saving respectively.

Figure 44 Numerical predicted results

5.6 Summary

In this chapter, the regenerator with heat recovery in LDDS has been investigated. The heat pipe heat exchanger (HPHE) and fix-plate heat exchanger (FPHE) were employed as the heat recovery device. Regenerators without heat recovery, with 4 rows HPHE, with 8 rows HPHE and with FPHE were compared in terms of regenerating performance and heat recovery performance. The results were predicted by the numerical computation and were validated by the experimental results. The main results are summarized below:

- The numerical results predicted by the models were consistent with the experimental results. All the relative errors were less than 11% and the largest RE is only 10.87%. The low RE value indicated the accuracy and availability of the numerical computation to predict the regenerating and heat recovery performance of regenerator with heat recovery.

- Compared with regenerator without heat recovery, recovering and reutilizing the waste heat could lead to improvement of regenerating performance with higher regenerating rate, effectiveness and thermal efficiency. The model predicted results reveal that the maximal net heat recovery ratios were 25%, 26.5% and 27% , which contributed to 26.5%, 27% and 26.9% maximal energy saving for regenerator with 4 rows HPHE, with 8 rows HPHE and with FPHE respectively.
- When regenerating air mass flow rate increased from 0.25kg/s to 0.41kg/s, the regenerating rate decreased firstly for the first two cases and thereafter increased with the increasing air flow rate for regenerator without and with heat recovery device. Yet for the regenerating effectiveness and thermal efficiency, they both decreased all the way with the increasing air flow rate. The heat recovery rate and effectiveness for 4 rows HPHE, 8 rows HPHE and FPHE increased with the air flow rate.
- When the air flow rate is low, 8 rows HPHE exhibited the best performance for regeneration compared with 4 rows HPHE and FPHE. When the air flow rate is high, FPHE became preferable choice compromising the heat recovery performance and the energy consumption. The comparison of 4 rows and 8 rows HPHE showed that with the continuous increase of air flow, the superiority of 8 rows HPHE diminished while the performance of 4 rows HPHE improved gradually.

CHAPTER 6 PERFORMANCE OPTIMIZATION OF REGENERATOR WITH ENERGY RECOVERY USING GENETIC ALGORITHM

6.1 Introduction

Energy saving and regenerating performance are two important aspects of regenerator. Earlier in this thesis, it is proved that waste energy recovery provides a cost-efficient way to save energy for the regenerator in LDDS even though additional pressure drop is induced. For promoting regenerating performance, the structured packing is filled in regenerator to increasing the contact area of regenerating air and liquid desiccant. However, the remarkable flow resistance of structured packing induces dramatic pressure drop. The optimal geometric dimension of packing, such as optimal height and diameter, can provide sufficient regenerating performance with acceptable pressure drop. Besides, the operating parameters, especially air flow rate, also plays a significant role in energy saving and pressure drop.

As the design and operating parameters have appreciable influence on the performance and the total pressure drop of heat exchanger and structured packing, it is of great essence to determine the optimal points. Many researchers have evaluated the optimal condition of heat exchanger in terms of design and operating parameters. Based on different optimization algorithms, the optimal design condition of heat exchanger, which has high performance with acceptable cost, has been determined.

Sanaye et al [75] presented thermal model and design optimization to estimate the pressure drop and thermal effectiveness of heat exchanger. They minimized the total annual cost and maximized the heat transfer effectiveness which were estimated by

e -NTU method. Six design parameters were optimized by NSGA-II. Najafi et al [76] considered the air-air plate-fin heat exchanger and optimized several geometric parameters within some constraints. The total heat transfer rate and the total annual cost were defined as two conflicting optimization objectives. A set of optimal solutions were obtained by multi-objective genetic algorithm. Yin et al [77] optimized the structural parameters of a water-water plate-fin heat exchanger employed in an air-conditioning system by using genetic algorithm. The single-objective and multi-objective optimization were both considered in their reports. Wen et al [78] optimized the configuration parameters of shell and tube heat exchanger with helical baffles by an improved algorithm, which combines the Kriging response surface and the multi-objective genetic algorithm. The optimization objectives were heat transfer rate and the total cost. The optimization results were compared with the CFD simulation results and the error was small. Besides, the authors [79] were also investigated and optimized the configuration of serrated fin in plate-fin heat exchanger by using genetic algorithm. Two conflicting optimization objectives were maximum j factor and minimum f factor. The optimization results showed that the heat transfer rate increased by 145w and the energy consumption decreased by 48.5%. Yin et al [80] installed a heat recovery exchanger to preheat the water and to reduce the heat loss of the kiln shell. By using the Matlab genetic algorithm toolbox, the design parameters of the heat recovery exchanger were optimized to obtain the minimum heat transfer area, pressure drop and entropy generation. Zeng et al [81] investigated the plate-fin air preheater

experimentally under different conditions. The genetic algorithm was employed to obtain the critical Reynolds number and the heat transfer coefficients.

In the multi-objective optimization problem, two or more objective functions are to be simultaneously optimized. For example, the criteria in manufacturing a product could be cost minimization and efficiency maximization. Many reports [82, 83] suggested that genetic algorithm (GA) is the most popular approach for multi-objective optimization problems. It has a strong ability of evolutionary search and combinational optimization. And it can optimize and predict the thermal problems successfully.

Genetic algorithms (GAs) are adaptive heuristic search method inspired by the evolutionary theory of natural selection. This method was first proposed by Holland [84] and further developed by Goldberg [85]. The first GA for multi-objective optimization, vector evaluated GA (VEGA), was developed by Schaffer [86]. Thereafter, many multi-objective GAs were proposed by researchers, including Niched Pareto GA (NPGA) [87], Weight-based GA (WBGA) [88], Nondominated Sorting GA (NSGA) [89], Strength Pareto GA (SPGA) [90] and improved SPGA (SPGA2) [91], Fast Nondominated Sorting GA (NSGA II) [92], et al. Generally, these multi-objective GAs have different fitness method, elitism and diversification approaches [93]. GAs are widely used to find high-quality solutions of optimization and search problems. The terminologies used in genetic algorithm are explained as following:

Chromosome: In GA, a solution vector is called a chromosome. It consists of discrete units called genes, which are usually binary digits.

Encoding: A mapping mechanism connects the solution space and the chromosomes.

Population: A collection of chromosomes. It is normally randomly initialized. As evolution continues, the solution in population gets fitter based on the fitness function and eventually gets convergence.

Crossover: Genetic operator to produce new solutions from existing ones. In crossover, two parental chromosomes combine together to generate new offspring chromosomes. The parental chromosomes are selected from the existing chromosomes based on the fitness function to obtain the fitter offspring. As the iterative computation goes on, the fitter chromosomes are frequently appearing in the population and finally lead to the convergent solution.

Mutation: Genetic operator to maintain genetic diversity of the population. The mutation operator randomly alters one or more gene values in the chromosome. Mutation occurs during the evolution process based on the mutation probability defined by the user. It prevents the chromosomes in the population from being too similar to each other.

Genetic algorithms can deal with highly non-linear complex equations and constraints. The reason is that GA does not have to solve the equations when finding the optimal variables. Instead, it searches the permissible range of the variables, evaluates the equations, calculates the values of fitness function, and decides the

optimal results of the variables based on the maximal fitness values. All these works are under the constraints. Genetic algorithms are simple and easy to be implemented.

In the procedure of genetic algorithm, optimization begins with an initial population. The value of fitness function of the initial population is computed and the best chromosomes are selected as parents. Afterwards, reproduction which is called offspring is generated. If the fitness value of individual is high, more offspring can be bred. On the contrary, if the fitness value of individual is low, less offspring will be bred, or even be eliminated. The optimization procedure will stop when the criteria are satisfied.

In this chapter, the multi-objective optimization of regenerator with heat recovery device has been formulated and genetic algorithm has been used to tackle the optimization problem. The optimization objective is to find the optimal packing dimensions and air flow rate to maximize the regenerating rate and minimize the total pressure drop of regenerator with heat recovery. The pressure drop models of heat recovery device and structured packing and the mass transfer model have been applied to search the optimal values.

6.2 Problem formulation

Although structured packing has high efficiency and is well established for the interaction of gas and liquid, its performance is very sensitive to the design condition. The structured packing in the regenerator consists of a number of stacked packing elements with certain height and each packing element rotates to each other by 90° to enhance the mixing effect of gas and liquid. The objectives of the optimization for

regenerator with heat recovery are to maximize the regenerating rate and minimize the pressure drop caused by structured packing and heat recovery device.

Practically, the packing size is limited by the regenerator tower. The air flow rate is controlled by the performance of fan. In order to ensure the feasibility of the optimization results, some constraints are imposed so that the optimization results fall within the reasonable design and operational region.

1. Packing element number m : The total packing bed height h_{pb} is limited by the regenerator tower. When the packing element height h_{pe} is certain, the total packing height is decided by the packing element number m . The reasonable range of m is:

$$m_{\min} \leq m \leq m_{\max} \quad (6-1)$$

where $m_{\min}$ and $m_{\max}$ are the minimum and maximum packing element number, respectively.

2. Packing diameter d_p : The structured packing is filled in the regenerator tower and its diameter d_p must be within its limitation:

$$d_{p,\min} \leq d_p \leq d_{p,\max} \quad (6-2)$$

where $d_{p,\min}$ and $d_{p,\max}$ are the minimum and maximum packing diameter, respectively.

3. Heat pipe row number n : Since the heat pipe in HPHE is easy to mount and demount, the heat pipe row is changeable and therefore the heat recovery rate can be changed. Based on the HPHE design, the reasonable range of n is:

$$n_{\min} \leq n \leq n_{\max} \quad (6-3)$$

where $n_{\min}$ and $n_{\max}$ are the minimum and maximum heat pipe row number, respectively.

4. Air flow rate $\dot{m}_a$: According to the fan performance, the air flow rate is restricted by:

$$\dot{m}_{a,\min} \leq \dot{m}_a \leq \dot{m}_{a,\max} \quad (6-4)$$

where $\dot{m}_{a,\min}$ and $\dot{m}_{a,\max}$ are the minimum and maximum air flow rate, respectively.

Consequently, the optimization problem of regenerator with HPHE can be formulated as:

$$\begin{aligned} \min \quad & f_1 = \Delta P_{total} = \Delta P_p + \Delta P_h \\ & f_2 = 1/\dot{m}_{re} \\ \text{subject to:} \quad & m_{\min} \leq m \leq m_{\max} \\ & d_{p,\min} \leq d_p \leq d_{p,\max} \\ & n_{\min} \leq n \leq n_{\max} \\ & \dot{m}_{a,\min} \leq \dot{m}_a \leq \dot{m}_{a,\max} \end{aligned} \quad (6-5)$$

where ΔP_{total} , ΔP_p and ΔP_h is the total pressure drop, pressure drop caused by packing and pressure drop caused by HPHE, respectively. $\dot{m}_{re}$ is regenerating rate

of regenerator. m is the number of packing elements. d_p is the packing diameter. n is the heat pipe row number of HPHE. $\dot{m}_a$ is the mass flow rate of air.

Unlike with HPHE, whose heat pipe row can be mounted and demounted easily and results in alterable heat transfer area, the structure and heat transfer area of FPHE is fixed. Thus the optimizing parameter of regenerator with FPHE is less than that of regenerator with HPHE. The optimization problem of regenerator with FPHE can be expressed as follows:

$$\begin{aligned}
 \min \quad & f_1 = \Delta P_{total} = \Delta P_p + \Delta P_f \\
 & f_2 = 1/\dot{m}_{re} \\
 \text{subject to:} \quad & m_{\min} \leq m \leq m_{\max} \\
 & d_{p,\min} \leq d_p \leq d_{p,\max} \\
 & \dot{m}_{a,\min} \leq \dot{m}_a \leq \dot{m}_{a,\max}
 \end{aligned} \tag{6-6}$$

where ΔP_f is pressure drop of FPHE.

6.3 Model and validation

6.3.1 Pressure drop model of HPHE

The HPHE used in this study consists of multi-row heat pipe with staggered tube rows. The pressure drop caused by the HPHE ΔP_h is given [94]:

$$\Delta P_h = \frac{2f_h v_{\max}^2 n}{\rho_A} \left(\frac{\mu_{Aw}}{\mu_{Ab}} \right)^{0.14} \tag{6-7}$$

where $v_{\max}$ is the air velocity at the minimum flow area; n is the heat pipe row in HPHE; μ_{Aw} and μ_{Ab} are the viscosity at bulk temperature and pipe wall of air

stream respectively; ρ_A is density of air; f_h is friction factor and it is given by the following equation:

$$f_h = \left\{ 0.25 + \frac{0.118}{[(S_v - d)/d]^{1.08}} \right\} \text{Re}_{\max}^{-0.16} \quad (6-8)$$

where S_v is transverse pipe pitch. d is heat pipe diameter. $\text{Re}_{\max}$ is the Reynolds number based on air velocity at the minimum flow area.

6.3.2 Pressure drop model of FPHE

Since FPHE is divided into three sections, the total pressure drop corresponds to the sum of the pressure drop of sections 1, 2 and 3. In section 1 of FPHE, the pressure drop $\Delta P_{f,1}$ can be obtained by [39]:

$$\Delta P_{f,1} = \frac{f_1 \cdot \rho_a \cdot L_{m,1} \cdot v_{a,1}^2}{2 \cdot D_{h,1}} \quad (6-9)$$

where $L_{m,1}$ is the mean length of the duct considered to section 1. $v_{a,1}$ is the air velocity in section 1. f_1 is friction factor of section 1. $D_{h,1}$ is the hydraulic diameter of section 1 and is defined as below:

$$D_h = \frac{4V_{f,1}}{A_{f,1}} \quad (6-10)$$

where $V_{f,1}$ and $A_{f,1}$ are the core volume and total surface area of section 1 of the FPHE. For sections 2 and 3, the pressure drop can be calculated by the same equation. Thus, the total pressure drop of FPHE is:

$$\Delta P_f = \Delta P_{f,1} + \Delta P_{f,2} + \Delta P_{f,3} \quad (6-11)$$

The friction factor f_1 can be obtained from the interpolation of data by Nelis and Klein [95].

6.3.3 Pressure drop model of structured packing

According to the Delta model [96-99], the pressure drop caused by structured packing consists three parts: pressure loss caused by air-liquid interaction, by the air-air interaction at the crossing of the air flow channel and by the direction change of packing elements:

$$\Delta P_{\text{Packing}} = \Delta P_{AL} + \Delta P_{AA} + \Delta P_{DC} = F_{load} (\zeta_{AL} + \zeta_{AA} + \zeta_{DC}) \frac{\rho_A u_{Ae}^2}{2} \quad (6-12)$$

where subscripts AL, AA and DC respectively refer to pressure losses associated with the air-liquid film surface interaction, air-air interaction at the crossing of air flow channel and direction changes at the transitions between packing elements and in the wall zones. u_{Ae} is effective air velocity, ρ_A is air density and F_{load} is a pressure drop enhancement factor in the loading region. ζ_{AL} , ζ_{AA} and ζ_{DC} represent corresponding flow resistance coefficients. The effective air velocity is calculated by:

$$u_{Ae} = \frac{u_{As}}{\sin \alpha (\sigma - h_L)} \quad (6-13)$$

where u_{As} is superficial air velocity. α is the corrugation inclination angle of structured packing. σ is the packing porosity. h_L is the liquid holdup of structured packing and is defined by:

$$h_L = a_p \delta \quad (6-14)$$

in which a_p is the specific surface area of structured packing. δ is liquid film thickness on the packing surface and can be predicted by the Nusselt formula [27]:

$$\delta = \left(\frac{3\mu_L u_{Ls}}{\rho_L g a_p \sin \alpha} \right)^{1/3} \quad (6-15)$$

where μ_L and ρ_L are the viscosity and density of liquid desiccant solution. u_{Ls} is superficial liquid velocity. g is the gravitational acceleration.

The pressure drop enhancement factor F_{load} is determined by:

$$F_{load} = 1453u_{Ls} + \frac{1.17}{\cos \alpha} - \frac{4294u_{Ls} + \frac{2.93}{\cos \alpha}}{u_{Ae}} \quad (6-16)$$

The flow resistance coefficient between air and liquid ζ_{AL} is equaled to:

$$\zeta_{AL} = \tau \xi_{AL} \frac{h_{pb}}{d_{hA} \sin \alpha} \quad (6-17)$$

In the above equation, h_{pb} is the total packing bed height as shown in Figure 45(a).

d_{hA} is the hydraulic diameter for air. τ represents the wetted perimeter of the triangular air flow channel. ξ_{AL} is the friction factor. d_{hA} , τ and ξ_{AL} are determined

by Eqs. (6-18) to (6-20). In Eqs (6-18) and (6-19), b , s and h are dimensions of packing micro-scale representative element as shown in Figure 45(b).

$$d_{hA} = \frac{(bh - 2\delta s)^2 / bh}{\left[\left(\frac{bh - 2\delta s}{2h} \right)^2 + \left(\frac{bh - 2\delta s}{b} \right)^2 \right]^{0.5} + \frac{bh - 2\delta s}{2h}} \quad (6-18)$$

$$\tau = \frac{2s}{b + 2s} \quad (6-19)$$

$$\xi_{AL} = \left\{ -2 \log \left[\frac{(\delta/d_{hA})}{3.7} - \frac{5.02}{Re_{Arv}} \log \left(\frac{(\delta/d_{hA})}{3.7} + \frac{14.5}{Re_{Arv}} \right) \right] \right\}^{-2} \quad (6-20)$$

(a) Stacked structured packing

(b) Micro-scale representative element of packing

Figure 45 Geometry and dimension of structured packing

In Eq. (6-20), Re_{Arv} stands for a Reynolds number based on relative air phase velocity:

$$Re_{Arv} = \frac{\rho_A (u_{Ae} + u_{Le}) d_{hA}}{\mu_A} \quad (6-21)$$

in which u_{Le} is the effective liquid velocity:

$$u_{Le} = \frac{u_{Ls}}{a_p \delta} \quad (6-22)$$

Air-air interaction ζ_{AA} is related to the open side of the air flow channel and it proved to be a strong function of corrugation inclination angle α :

$$\zeta_{AA} = (1 - \tau) \xi_{AA} \frac{h_{pb}}{d_{hA} \sin \alpha} = 0.722 (1 - \tau) (\cos \alpha)^{3.14} \frac{h_{pb}}{d_{hA} \sin \alpha} \quad (6-23)$$

The corresponding overall coefficient for direction change related losses ζ_{DC} is described as;

$$\zeta_{DC} = \frac{h_{pb}}{h_{pe}} (\xi_{bulk} + \Psi \xi_{wall}) \quad (6-24)$$

where h_{pe} is packing element height as shown in Figure 45(a). ξ_{bulk} and ξ_{wall} are the characteristic form drag coefficients for bulk and wall zone, respectively. Ψ is the ratio of the number of gas flow channels in the bottom cross section of a packing element or layer, i.e. the fraction of the total number of gas flow channels ending at column walls. ξ_{bulk} , ξ_{wall} and Ψ can be determined by Eq. (6-25)-(6-27).

$$\Psi = \frac{2h_{pe}}{\pi d_p^2 \tan \alpha} \left(d_p^2 - \frac{h_{pe}^2}{\tan^2 \alpha} \right)^{0.5} + \frac{2}{\pi} \arcsin \left(\frac{h_{pe}}{d_p \tan \alpha} \right) \quad (6-25)$$

$$\xi_{bulk} = 1.76 (\cos \alpha)^{1.63} \quad (6-26)$$

$$\xi_{wall} = \frac{4092 u_{Ls}^{0.31} + 4715 (\cos \alpha)^{0.445}}{Re_{Ae}} + 34.19 u_{Ls}^{0.44} (\cos \alpha)^{0.779} \quad (6-27)$$

In Eq. (6-25), d_p is the packing column diameter as shown in Figure 45(a). In Eq. (6-27), Re_{Ae} is a Reynolds number based on the effective air velocity u_{Ae} :

$$Re_{Ae} = \frac{u_{Ae} \rho_A d_{hA}}{\mu_A} \quad (6-28)$$

6.3.4 Mass transfer computation of regenerator with heat recovery

In the regenerating process, transferring process of water from the liquid desiccant to the regenerating air overcomes three resistances: resistance of desiccant solution, resistance at the interface of solution and air and resistance of regenerating air. Since the mass transfer process at the phase interface is very fast compared with processes

within air and solution, the resistance at the interface can be neglected. Consequently, the mass transfer process in the regenerator driven by the water vapor pressure difference can be described by:

$$\dot{m}_{re} = K_G \left(P_{s,re,in}^* - P_{a,re,in} \right) \quad (6-29)$$

where $\dot{m}_{re}$ is regenerating rate. K_G is overall mass transfer coefficient in the regenerator with heat recovery and can be determined by the following equation:

$$K_G = \frac{1}{\frac{1}{k_a} + \frac{1}{Hk_s}} \quad (6-30)$$

where H is Henry's law constant. k_a and k_s are the convective mass transfer coefficient of air and liquid desiccant solution respectively. The respective correlations of k_a and k_s were proposed by Onda [100]:

$$k_a = 5.23 \left(\frac{4\dot{m}_a}{a_p \mu_a \pi D^2} \right)^{0.7} \left(\frac{\mu_a}{\rho_a D_a} \right)^{1/3} (\alpha_t D_p)^{-2} \frac{V D_a}{R T_a} \quad (6-31)$$

$$k_s = 0.0051 \left(\frac{4\dot{m}_s}{\alpha_\omega \mu_s \pi D^2} \right)^{2/3} \left(\frac{\mu_s}{\rho_s D_s} \right)^{-1/2} (\alpha_t D_p)^{0.4} \left(\frac{\rho_s}{\mu_s g} \right)^{-1/3} a_p V \quad (6-32)$$

where a_p and α_ω are the specific surface area and wetted specific surface area respectively. μ_a , ρ_a and D_a are the viscosity, density and diffusivity of air respectively. μ_s , ρ_s and D_s are the viscosity, density and diffusivity of solution

respectively. D and D_p are the packing diameter and nominal size of packing respectively. R and g are the ideal gas constant and gravitational constant respectively. V is the volume of the structured packing.

In Eq. (6-29), $p_{s, re, in}^*$ is the equilibrium water vapor pressure of inlet liquid desiccant solution and it can be calculated by Eq. (5-5) introduced in Chapter 5. $p_{a, re, in}$ is the water vapor pressure of regenerating air at the regenerator inlet. It can be calculated by Eqs. (5-6)-(5-7) introduced in Chapter 5. Based on Eqs. (5-6) and (5-7), the calculation of $p_{a, re, in}$ needs the temperature of regenerating air which is preheated by the heat recovery device. This value is calculated by the hybrid heat recovery model of HPHE and FPHE proposed in Chapter 4 and the hybrid heat transfer model proposed in [44] is used. Since the warm air temperature at the regenerator outlet is one of the input of hybrid heat recovery model, its value is obtained by the heat and mass transfer model of the regenerating process. So iterative computation is conducted. The computation procedure is shown in Figure 46. The initial value of warm air temperature at the regenerator outlet is assumed firstly. With the assuming value, the preheated regenerating air temperature at the regenerator inlet is calculated by the heat recovery model. And then the new result of warm air temperature at the regenerator outlet is obtained by the heat and mass transfer model of regenerating process. When the residual between the old warm air temperature and new warm air temperature is small enough, the iterative computation is ended.

Figure 46 Procedure of mass transfer computation of regenerator with heat recovery

6.3.5 Mass transfer computation validation

The mass transfer computation of regenerator with heat recovery for the optimization are validated by the experimental data. As shown in Figure 47, the results of mass transfer computation for regenerator with 4 rows HPHE and with FPHE are consistent with the experimental results. And the relative errors are presented in Table 20. It is found that the largest relative error between the computation results

and the experimental data is 10.36%, which indicates the accuracy of the mass transfer computation of regenerator with heat recovery. Therefore, the mass transfer computation can be employed for the optimization problems of regenerator with heat recovery.

(a) Mass transfer computation validation for regenerator with 4 rows HPHE

(b) Mass transfer computation validation for regenerator with FPHE

Figure 47 Validation result of mass transfer computation

Table 20 RE of mass transfer computation

	Case 1	Case 2	Case 3	Case 4	Case 5
4 rows HPHE	9.61%	10.19%	1.29%	10.27%	4.97%

FPHE	8.10%	10.36%	8.43%	5.80%	5.28%
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6.4 Optimization algorithm

6.4.1 Implementation of genetic algorithm

To solve the nonlinear multi-objective optimization problem, a controlled elitist multi-objective genetic algorithm, which is a variant of NSGA-II and has better convergent property, is implemented. An elitist GA randomly creates the initial population and then the next generation of the population is bred by using the non-dominated sorting. Through controlling the elite members of the population in the computing process, the controlled elitist GA can help to maintain the diversity of the population even when the fitness value is low.

The solving procedure of multi-objective GA consists five parts, i.e. initialization, encoding, selection, evolution and termination.

1. Initialization: The initial population is created randomly and covers the entire range of possible solutions in the search space.
2. Encoding: The independent variables in the nonlinear equations are converted to binary strings. All these strings combine together to form chromosome.
3. Selection: Individuals among the existing population is selected as the parents to breed the offspring through fitness based process. The fitter individuals measured by the fitness function are more likely to be selected.

4. Evolution: The evolution process of GA includes crossover and mutation. After the individuals are selected, single point crossover takes place. And single bit mutation operator is performed right after the crossover. The crossover and mutation ultimately lead to offspring that is different from the previous generation. The mutation and crossover probability have a great impact on the performance of GA.
5. Termination: The GA computation stops when one of the following criterion is satisfied:
 - 1). Fixed generation number is reached.
 - 2). The variation of the optimal fitness value over several steps is smaller than the predetermined value.
 - 3). All the chromosomes converge to nearly the same fitness results.

By the genetic algorithm, the pareto-optimal results can be found. It is a set of non-dominated solutions, being chosen as optimal results of the optimizing problem. The plot of all the pareto-optimal results is called pareto curve. It is helpful for finding the trade-offs solution of the optimizing problem.

6.4.2 Optimization procedure

The procedure of GA to optimize the regenerator with energy recovery is depicted as Figure 48 and described as follows:

1. Obtain the ambient air temperature and the solution temperature at the regenerator inlet. Determine the unknown parameters in all models of regenerator with energy recovery.
2. Determine the parameters in GA, i.e. population size, crossover and mutation probability and maximal number of generation.
3. Encoding and initiation. Set $gen=1$. Randomly form the first population and calculate the fitness of the first population.
4. Select certain solutions from the population based on their fitness. Generate offspring population by crossover and mutation operators.
5. If the stop criterion is satisfied, terminate the computation and return the current population. Otherwise, set $gen=gen+1$ and go to step 4.
6. Find the individuals with largest fitness among population. Decode and obtain the optimal setting points.

Figure 48 Flow chart of GA procedure

6.5 Optimization results and discussion

The optimization problem includes four decision variables for regenerator with HPHE and three decision variables for regenerator with FPHE. These decision variables are optimized by using multi-objective genetic algorithm. As introduced earlier, the GA is a population based search algorithm, thus the population number

is one of the critical step. If the population has not been selected suitably, it will be difficult to obtain the appropriate results of the optimization problem. Besides, the performance of GA largely depends on the crossover and mutation operators [101]. Therefore, adapting suitable value for the probability of crossover and mutation is very important since the probability controls the search process and maintains the diversity in the population [102]. If the probability of crossover is high, new strings is introduced more quickly into the population. And if this value is low, stagnation may be caused because of the low exploration rate. Mutation is usually applied with low probability, typically within the range 0.01 and 0.1 [103]. For the selection method, the Roulette-Wheel method [104] is employed in this thesis. The parameters of GA are listed in Table 21. The range of the optimization variables of regenerator with HPHE and FPHE are given in Table 22-23.

Table 21 The tuning parameters in optimization using genetic algorithm

Tuning parameters	Value
Population size	100
Maximum number of generations	200
Probability of crossover	0.7
Probability of mutation	0.01
Selection process	Roulette-wheel

Table 22 The optimization variable range of regenerator with HPHE

Variables	Minimal value	Maximal value
Packing element number m	2	10
Packing diameter d_p	0.4	1.0
Heat pipe rows n	1	8

Air flow rate $\dot{m}_a$	0.25	0.41
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Table 23 The optimization variable range of regenerator with FPHE

Variables	Minimal value	Maximal value
Packing element number m	2	10
Packing diameter d_p	0.4	1.0
Air flow rate $\dot{m}_a$	0.25	0.41

6.5.1 Optimization results of regenerator with HPHE

The optimization results of the Pareto-optimal curve for regenerator with HPHE are shown in Figure 49, which clearly reveals the conflict between two optimization objectives, i.e. the regenerating rate and the total pressure drop. Any change of optimization parameters that increases the regenerating rate results in an increase in the total pressure drop and vice versa, which shows the necessity for multi-objective optimization algorithm in optimal design of regenerator with HPHE. It is shown in Figure 49 that the maximum regenerating rate exists at point A, while the total pressure drop is the largest at this point. On the other hand the minimum total pressure drop occurs at point G, with a minimum regenerating rate value at this point. Point A is the optimal condition at which, the regenerating rate is the only optimization objective, while point G is the optimum situation at which the total pressure drop is the only optimization objective. The values of different decision parameters for points A–G on the Pareto-optimal front are reported in Table 24.

Figure 49 Pareto-optimal points total pressure drop and regenerating rate

Table 24 The selected Pareto-optimal values of regenerator with HPHE

Points	m	d_p (m)	n	$\dot{m}_a$ (kg/s)	ΔP_{total} (Pa)	$\dot{m}_{re}$ (g/s)
A	10	0.4	5	0.40	758.29	8.58
B	10	0.4	4	0.35	579.31	7.87
C	10	0.4	3	0.29	429.44	6.83
D	8	0.4	3	0.26	276.68	5.06
E	5	0.4	2	0.26	170.93	3.11
F	3	0.5	1	0.27	107.94	1.73
G	2	0.9	1	0.25	56.97	0.42

The distribution of various pareto-optimal points of optimization parameters in their permissible range are shown in Figure 50. Figure 50(a) shows that the distribution of packing element number is mostly spread close to the upper and lower limit. The optimal values of heat pipe row number, as shown in Figure 50(c), are mostly below than 4. For packing diameter and air flow rate as shown in Figure 50(b) and (d), their optimal values are close to the lower limit.

(a) Distribution of packing element number

(b) Distribution of packing diameter

(c) Distribution of heat pipe row number

(d) Distribution of air flow rate

Figure 50 The distribution of optimization variables for the Pareto-optimal points

To determine the optimal point for regenerator with heat recovery, two conflicting optimization objectives have been combined into a single objective by weighted sum approach [105], which is the simplest way to solve multi-objective optimization problem. The optimization problem of regenerator with HPHE can be stated as:

$$\begin{aligned}
 \min \quad & f = \omega_1 f_1 + \omega_2 f_2 \\
 \text{subject to:} \quad & m_{\min} \leq m \leq m_{\max} \\
 & d_{p,\min} \leq d_p \leq d_{p,\max} \\
 & n_{\min} \leq n \leq n_{\max} \\
 & \dot{m}_{a,\min} \leq \dot{m}_a \leq \dot{m}_{a,\max}
 \end{aligned} \tag{6-33}$$

where ω_1 and ω_2 are nonnegative weight for objective function 1 and 2. And they are restricted by:

$$\omega_1 + \omega_2 = 1 \tag{6-34}$$

In the optimization of regenerator with heat recovery, since the main objective of regenerator is concentrate the diluted liquid desiccant solution from the dehumidifier,

the regenerating rate is essential and needs to be considered firstly. The total pressure drop can be viewed as the cost for solution concentration. Therefore, the weights of objective functions 1 and 2 are chose as: $\omega_1 = 0.2$ and $\omega_2 = 0.8$. Then the results of the single optimization objective of regenerator with HPHE is obtained as shown in Figure 51. The smallest results, as indicated in red point in Figure 51, is 13.08. And the results of the corresponding optimization variables are: $m = 2$, $d_p = 0.7$, $n = 4$ and $m_a = 0.25$.

Figure 51 Weighted objective of regenerator with HPHE

6.5.2 Optimization results of regenerator with FPHE

The optimization results of the Pareto-optimal curve for regenerator with FPHE are shown in Figure 52. Similar with regenerator with HPHE, two optimization objectives are mutually conflicting where the regenerating rate increases with the increase of total pressure drop. As shown in Figure 52, the maximum regenerating rate and total pressure drop exists at point A. On the other hand, the minimum regenerating rate and total pressure drop occurs at point G. Point A is the optimal condition when the regenerating rate is single objective, while point G is the

optimum situation when the total pressure drop is single objective. The values of different decision parameters for points A–G on the Pareto-optimal front are reported in Table 25.

Figure 52 Pareto-optimal points total pressure drop and regenerating rate

Table 25 The selected Pareto-optimal values of regenerator with FPHE

Points	m	d_p (m)	$\dot{m}_a$ (kg/s)	ΔP_{total} (Pa)	$\dot{m}_{re}$ (g/s)
A	10	0.4	0.39	814.12	9.57
B	10	0.4	0.33	630.48	8.68
C	10	0.4	0.28	465.52	7.81
D	8	0.4	0.26	323.93	5.94
E	5	0.5	0.29	218.77	3.71
F	3	0.7	0.25	121.69	2.26
G	2	0.9	0.26	69.42	0.42

Figure 53 presents the pareto-optimal points distribution of three optimization parameters. As shown in Figure 53(a), the optimal points of packing element number for regenerator with FPHE are close to the upper and lower limit. Figure 53(b) and

(c) prove that for packing diameter and air flow rate, most pareto-optimal points fall within their lower permission range.

(a) Distribution of packing element number

(b) Distribution of packing diameter

(c) Distribution of air flow rate

Figure 53 The distribution of optimization variables for the Pareto-optimal points

By the weighted sum approach [105], the optimization problem of regenerator with FPHE can be stated as:

$$\begin{aligned}
 \min \quad & f = \omega_1 f_1 + \omega_2 f_2 \\
 \text{subject to:} \quad & m_{\min} \leq m \leq m_{\max} \\
 & d_{p,\min} \leq d_p \leq d_{p,\max} \\
 & \dot{m}_{a,\min} \leq \dot{m}_a \leq \dot{m}_{a,\max}
 \end{aligned} \tag{6-35}$$

Similar with regenerator with HPHE, the weights for objective 1 and 2 are: $\omega_1 = 0.2$ and $\omega_2 = 0.8$. Then the results of the single optimization objective of regenerator with FPHE is obtained as shown in Figure 54. The smallest results, as indicated in red point in Figure 54, is 15.54. And the results of the corresponding optimization variables are: $m = 4$, $d_p = 0.6$ and $m_a = 0.27$.

Figure 54 Weighted objective of regenerator with FPHE

6.6 Summary

In this chapter, multi-objective genetic algorithm has been conducted to optimize the design and operating parameters of the regenerator with heat recovery device. The regenerating rate and total pressure drop have been selected as two conflicting objectives. Pressure drop model and mass transfer model have been used in the optimization process. The main results are summarized below:

- The pareto-optimal points of regenerator with HPHE and FPHE have been obtained. For regenerator with HPHE, the distribution of packing element number is mostly spread close to the upper and lower limit. Most of the optimal values of heat pipe row are below 4. For packing diameter and air flow rate, their optimal values are close to the lower limit. For regenerator with FPHE, the optimal points of packing element number are close to the upper and lower limit. And most optimal values of packing diameter and air flow rate fall within their lower permission range.

- The weighted sum approach has been employed to combine two conflicting objectives into one single objective. The weights for two objectives, i.e. total pressure drop and regenerating rate, are 0.2 and 0.8 respectively. For regenerator with HPHE, the smallest results of the single objective is 13.08. And the corresponding values of the optimization variables are: $m=2$, $d_p=0.7$, $n=4$ and $m_a=0.25$. And for regenerator with FPHE, the smallest results of the single objective is 15.54. The corresponding values of the corresponding optimization variables are: $m=4$, $d_p=0.6$ and $m_a=0.27$.

CHAPTER 7 CONCLUSIONS AND FUTURE WORK

7.1 Conclusions

Since majority of energy is consumed by regenerator in Liquid Desiccant Dehumidification System, heat recovery technique has been employed in it. The intention of this thesis is to investigate thoroughly the regenerator with heat recovery, considering how heat recovery process affects the regenerating process and energy consumption. Besides, optimization has been conducted to find the optimal design and operating condition that can provide high regenerating rate as well as low pressure drop for regenerator with heat recovery. The following contributions have been achieved in this thesis:

- Hybrid model for heat recovery process in regenerator of LDDS has been developed. Starting from the physical governing equations, the proposed hybrid model lumped complex geometric and fluids' thermodynamic parameters and only had three unknown coefficients. These three parameters were identified by the Levenberg-Marquardt method by using experimental data. According to the results of validation, the proposed hybrid heat recovery model was accurate and effective and could be used for monitor, prediction and optimization of heat recovery process.
- Numerical simulation has been conducted for the evaluation of regenerator with two heat recovery devices, i.e. heat pipe heat exchanger (HPHE) and fixed-plate heat exchanger (FPHE). The numerical results predicted by the simulation were consistent with the experimental results. Recovering and

reutilizing the waste heat led to the improvement of regenerating performance with higher regenerating rate, effectiveness and thermal efficiency, compared with regenerator without heat recovery.

- According to the experimental and numerical results, the regenerating rate decreased firstly and then increased with the increasing air flow rate for regenerator without and with heat recovery device. Yet for the regenerating effectiveness and thermal efficiency, they both decreased all the way with the increasing air flow rate. The heat recovery rate and effectiveness for 4 rows HPHE, 8 rows HPHE and FPHE increased with the increasing air flow rate.
- The numerical results showed that when the air flow rate is low, 8 rows HPHE exhibited the best performance for regeneration compared with 4 rows HPHE and FPHE. When the air flow rate is high, FPHE became preferable choice compromising the heat recovery performance and the energy consumption. The comparison of 4 rows and 8 rows HPHE shown that with the continuous increase of air flow, the superiority of 8 rows HPHE diminished while the performance of 4 rows HPHE improved gradually.
- The design and operating parameters of regenerator with heat recovery have been optimized and the pareto-optimal points of regenerator with HPHE and FPHE have been obtained. The distribution of decision variables in their whole permissible range have been discovered. Two optimization objectives, i.e. regenerating rate and total pressure drop, were conflicting with each other and the trade-off solution was determined by the weighted sum approach.

The weights for regenerating rate and total pressure drop were 0.8 and 0.2, respectively.

7.2 Future work

Based on the conclusions in this thesis, heat recovery technique can result in a great improvement in regenerating performance and reduction in energy consumption. It is very worthwhile and valuable to make many efforts to reduce the energy consumption of regenerator. There are many researches arising from this thesis which should be pursued:

- The temperature and concentration of liquid desiccant solution on the performance of regenerator with heat recovery needs to be analyzed. Keeping the solution temperature as high as possible is important for the regenerating process. The existence of heat recovery device decreases the sensible heat transfer between air and solution, which is beneficial for maintaining the solution temperature. Therefore it is necessary to investigate the performance of regenerator with heat recovery under different solution temperature and concentration. Initial experiments have been conducted. In the experiments, the flow rate of air was kept at 0.41kg/s. The flow rate of liquid desiccant changed from 0.35kg/s to 0.63kg/s. Two inlet temperatures of liquid desiccant, i.e. 50°C and 55°C have been investigated initially. The inlet concentration of liquid desiccant was kept at 26%. The initial experimental results of regenerating rate with increasing liquid flow rate are presented in Figure 55. The results shows the regenerating rate increases with higher

liquid inlet temperature. And the regenerating rate also increases when the HPHE is installed in regenerator.

Figure 55 Regenerating rate with increasing liquid mass flow rate

- Structured packing is an important component in regenerator. Its dimension has significant impact on the regenerating rate and pressure drop. This thesis only optimized the packing diameter and total height. Other parameters of packing, such as corrugation angle and flute height, also needs to be optimized. Simple and accurate hybrid model for structured packing can also be developed for its optimization. This model will start with physical governing equations of structured packing. And then the operating parameters that are related on the air and liquid will be lumped as unknown parameters that need to be identified. The design parameters of structured packing will be viewed as input of the model. Experiments will be carried out with different types of structured packing. The experimental results will be used to identify the unknown parameters in the hybrid model of structured packing.

- Other than heat recovery technique, the application of solar energy in regenerator to instead the traditional heater is helpful in reduction of energy consumption. In tropical country, with largely available solar radiation, the utilization of hybrid solar system is viable option to reduce the energy consumption of regenerator. With the concept of hybrid solar regenerator and heat recovery technique, the total energy consumption of regenerator must be much lower than traditional regenerator. The regenerating performance and the energy consumption of solar regenerator with heat recovery is worth to be investigated. In the future work, solar collector will be installed in regenerator with heat recovery to provide heat for the liquid desiccant instead of traditional heater. The regenerating performance, total energy consumption and the system cost will be considered comprehensively.

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Author's Publications

1. **Shen Suping**, Cai Wenjian, Wang Xinli, et al. Hybrid model for heat recovery heat pipe system in Liquid Desiccant Dehumidification System. *Applied Energy* 2016; 182:383-393, 2016.
2. **Shen Suping**, Cai Wenjian, Wang Xinli, et al. Investigation of liquid desiccant Regenerator with heat recovery heat pipe system. *Energy and Building* 2017; 146: 353-363.
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4. **Shen Suping**, Cai Wenjian, Yon Haoren, et al. CFD simulation of structured packing in Liquid Desiccant Dehumidification System (LDDS), ICIEA 2016 IEEE 11th Conference.
5. **Shen Suping**, Cai Wenjian, Wu Qiong, et al. Hybrid model for heat recovery system in Liquid Desiccant Dehumidification System (LDDS), ICIEA 2017 IEEE 12th Conference.