A STUDY OF A DESICCANT AIR DEHUMIDIFIER REGENERATED BY HOT AND COLD WATER HEATER

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A THESIS SUBMITTED AS A PART OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF ENGINEERING IN ENERGY TECHNOLOGY AND MANAGEMENT

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ABSTRACT

A desiccant dehumidifier regenerated by hot and cold water heater system has been installed and experiments were carried in a hot and humid climate (Thailand). A Desiccant coated on a heat exchanger system can handle latent and sensible loads by removing the moisture by an adsorption process when supplying cold water from a chiller and desorption by supplying hot water from a heater. Influence of operation parameters including air inlet temperature, air inlet humidity, cycle time, inlet air velocity, and water flow rate on system. Performance are analyzed in term of moisture adsorption (D_{ad}), moisture desorption (D_{de}), thermal coefficient of performance (COP_{th}), moisture removal capacity (MRC), Moisture removal regeneration (MRR), Dehumidification effectiveness (E_{deh}) and regeneration effectiveness (E_{reg}). The system could reduce the temperature of the delivered air by about 8°C while the humidity ratio was reduce by 0.005 kg_w/kg_{da} equivalent to 20% relative humidity reduction.

Keywords: Desiccant dehumidifier, Dehumidification, Air dehumidifier

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SYMBOL	DESCRIPTION	UNIT
q	Heat flux	(w/m^2)
h	Heat transfer coefficient	(w/m^{2}K)
E_b	Emissive power	(W)
R	Thermal resistance	(m^{2}K/W)
E	Solar radiation flux	(w/m^2)
T	Temperature	(°C)
$\dot{m}$	Mass flow rate	(kg/h)
C_p	Specific heat	(kJK/kg)
COP_{th}	Thermal coefficient of performance	
D	Moisture removal rate	(g/kg)
H	Enthalpy	(kJ/kg)
HR	Humidity ratio	(g/kg)
Q	Quantity of heat	(kg/s)
m_{si}	Airflow source in sub-zone I	(kg/s)
h_{ik}	The convection heat transfer coefficient between surface k in zone I	(w/m^{2}K)
α	The solar absorbtance	
Nu_f	Solar thermal gain from the solar collectors	
Pr	The Prandtl number	
Gr	The Grashof number	
K_i	Factor the radiation interchange	
Re_f	The Reynolds number	
k	A constant value of various flow configurations	
a	Air	
ads	Adsorption	
avg	Average	

LIST OF NOMENCLATURES (Cont')

SYMBOL	DESCRIPTION	UNIT
in	inlet	
out	outlet	
w	water	
DCHE	Desiccant coated heat exchanger	
SG	Silica gel	
VC	Vapor compression	

CHAPTER 1
INTRODUCTION

1.1 Rationale/Problem statement

In a tropical climate, a air-conditioning (AC) systems are used to remove sensible and latent heat from building spaces. Conventionally, mechanical dehumidification using cooling coil to cool down the ambient air below it dew-point temperature for remove the moisture in ambient air. However, a cooling coil of an air condition system can handle sensible load and latent load by condensation dehumidification but the coefficient of performance (COP_{th}) of the system is limited to very low. It is energy-inefficient for using this system to dehumidification.

Nodaways, there are many solutions to solve this problem, such as using a rotating desiccant wheel to adsorb moisture from the air. A reheating process is needed to raise again the temperature of the over cooled air before supply air into the building space. The novel desiccant dehumidification method has been proposed. So far, various types of desiccant dehumidifiers such as liquid desiccant, solid desiccant have been developed and widely adopted to dehumidification system.

In this study, the experiments of a desiccant air dehumidifier was regenerated by hot and cold water heater, is proposed and investigated experimentally under the natural hot and humid climate conditions. The system was installed at a laboratory building at King Mongkut’s University of Technology Thonburi, Bang Khun Tien Campus, Thailand. A series of dehumidifier can continuous work in dehumidification and regeneration on the same time. A heater supplies hot water for regeneration process and a chiller supply cold water for dehumidification process. The performance of a system also discussed.

1.2 Literature Review

Nowadays, there are various dehumidifications for the dehumidified humidity of the air before supplying it to the space. Some researchers conducted investigations on different desiccant components, desiccant wheels, etc. for dehumidification. But there are consumed more energy for regenerated desiccant before dehumidifier. However, many researchers developed desiccants coated on heat exchangers (DCHE) to solve this problem.

1.2.1 Performance of cross-flow and counter-flow regenerators

Liu et al. studied the effects of air and desiccant inlet parameters on the regenerator performance and comparisons between present cross-flow regenerators and other counter-flow ones. The comparison results show moisture removal rate increases with increasing air flow rate, desiccant flow rate and desiccant inlet temperature, decreases with air inlet humidity ratio and desiccant inlet concentration, and changes little with air inlet temperature. Regenerator effectiveness increases with desiccant flow rate and inlet concentration, decreases with air flow rate and desiccant inlet temperature, and is affected little by air inlet temperature and humidity ratio. The impacts of air and desiccant inlet parameters show similar tendency with those previously reported for counter-flow regenerators. Dimensionless mass transfer correlation is calculated in the present study, which is correlated by Reynolds number, Schmidt number, flow rate ratio of desiccant to air, and water content of the desiccant. A good agreement is shown between the predicted values and experimental data with the correlation coefficient of 0.962 [1].

1.2.2 Effect of various parameter indices

Abdalla et al. studied the performance of an internally cooled dehumidifier using Triethylene Glycol as a desiccant. During the experimental investigation, the dehumidifier inlet parameters, including air flow rate, humidity ratio, temperature, desiccant flow rate, and temperature are varied. The effect of these variables on the moisture condensation rate and dehumidifier effectiveness was studied. It is found that the moisture condensation rate increases with increasing the inlet air flow rate, inlet air humidity ratio, desiccant flow rate, and desiccant solution concentration. While the dehumidifier effectiveness increases with increasing desiccant flow rate and concentration. The dehumidifier effectiveness decreases with increasing inlet air flow rate and humidity ratios [2].

Jia et al. has been experimentally the dehumidification performances of the composite desiccant wheel was tested and compared with those of the silica gel wheel. The test results indicate that the moisture removal capacity of the new composite desiccant wheel, on average, is bigger than that of the traditional silica gel wheel by 50%, in that high hygroscopic LiCl embedded in the pore channels of the silica gel improves its moisture adsorption capacity. The moisture removal capacity and DCOP are affected by the inlet air humidity, obviously. They increase with an increase in inlet air humidity. Especially the composite desiccant wheel possesses evident dominance at low relative humidity. Higher regeneration temperature results in more moisture removal capacity for the two desiccant wheels. However, there are different optimal regeneration temperatures for them. The optimal regeneration temperature of the composite desiccant wheel is lower than that of the silica gel wheel [12].

1.2.3 Desiccant coated on heat exchanger

Nowadays, a solid desiccant component named “desiccant coated on heat exchanger (DCHE)” is developed. In this component, a solid desiccant material is coated on the surface of a fin-tube sensible heat exchanger, and then cooling water inside the tube can keep cool down the passed process air in dehumidification process. There are many current researches on silica gel coated on heat exchanger and study the influence of major parameters on system performance in term of optimum cycle time and thermal COP.

Ge et al. fabricated two desiccants coated on heat exchangers with silica gel and polymers. It is found that this desiccant-coated fin-tube heat exchanger well overcomes the side effect of adsorption heat which occurs in desiccant dehumidification process, and achieves good dehumidification performance under given conditions, with bigger transient as well as average moisture removal and longer effective dehumidification time [3].

Ge et al. conducted research based on the whole desiccant cooling system utilizing two DCHEs by simulation. It is found the operation time in dehumidification process is a crucial factor for cooling capacity of DCHE system, which can be enhanced by eliminating the initial period with higher outlet air temperature, the largest cooling power of DCHE system increase from 2.6 kW to 3.5 kW by eliminating first 50 s of operation time under ARI summer condition. Also, DCHE system can only provide cooling power after a short operating duration from the initial dehumidification process [4].

Ge et al. studied solar powered desiccant coated heat exchanger cooling systems which can provide satisfied supply air to the conditioned indoor space from 8:00 to 17:00 in June and July, the highest cooling powers are 2.9 kW and 3.5 kW, and corresponding solar COP are 0.22 and 0.24, respectively[5].

Zhao et al. has experimentally set up a desiccant dehumidification unit in which a silica gel coated fin-tube heat exchanger is installed and investigated. In this unit, two silica gel coated heat exchangers (SCHE) are adopted and switched to provide continuous dehumidification capacity. Meanwhile, hot water from vacuum tube solar collector is used to regenerate silica gel. System performance is evaluated in terms of moisture removal mass and thermal COP. Influences of major parameters on system performance are tested and analyzed under Shanghai summer conditions. It was found that system performance is affected significantly by cycle time between dehumidification and regeneration, and optimal cycle time is 600 sec under test conditions [6].

Y. Jiang et al. has been experimentally set up built to test and compare the dynamic performance of SGCHE and CCHE. Influences of main operation parameters including water temperatures and inlet air conditions on system performance are analyzed in terms of average dehumidification capacity (D_{avg}) and thermal coefficient of performance (COP_{th}).Experimental results show that CCHE has better dehumidification performance compared with SGCHE. In addition, precooling before dehumidification process is found to be advantageous to both D_{avg} and COP_{th} [8].

1.3 Research Objectives

The objective of this study is to conduct a theoretical and experimental study on a desiccant dehumidification system that is regenerated by conducted heat from hot water. The specific objectives of the study are:

- to fabricate air dehumidifiers comprising solid desiccants deposited on water to air heat exchangers
- to conduct theoretical and experimental studies of the fabricated dehumidifiers

1.4 Scope of Research Work

In this study, a desiccant system was constructed at a laboratory building at King Mongkut's University of Technology Thonburi, Bang Khun Tien Campus. The experiment was conducted in a real tropical climate.

CHAPTER 2
THEORIES

2.1 Moist Air Properties

The study of desiccant dehumidifier requires knowing properties of process airs under different states. The following summarizes a set of formulas that can be applied for determining at acceptable accurate level properties of the air when two other properties are known. An alternative approach of using Psychometric chart is presented next.

2.1.1 Calculation of Moist Air Properties

Moist air in the atmosphere is a binary mixture of dry air and a small amount of water vapor in equilibrium at a given temperature and pressure. The pressure is relatively low so that the behavior of dry air, water vapor or the mixture can all be examined under ideal gas law.

a) Pressure

According to Dalton's law, the total pressure exerted by a gas mixture is the sum of partial vapor pressure of each component:

$$p = p_a + p_w \tag{2.1}$$

where

p is the total pressure exerted by the mixture

p_a and p_w are partial pressure exerted by dry air and water vapor respectively

In the given volume V (m^3) of the binary mixture at absolute temperature T_{abs} (K), the following relationships are obtained according to the ideal gas law:

$$p_a V = \frac{m_a R T_{abs}}{M_a} \tag{2.2}$$

and,

$$p_w V = \frac{m_w R T_{abs}}{M_w} \tag{2.3}$$

where

m_a is mass of the dry air in the volume, kg

m_w is mass of the water vapor in the volume, kg

R is the universal gas constant, 8.315 kJ/ (kmol·K)

M_a is the molecular weight of dry air, 28.9645 kg/ kmol

M_w is the molecular weight of water vapour, 18.01528 kg/ kmol

b) Humidity ratio

The humidity ratio or the absolute humidity (w) of moist air at a given condition is the ratio of the mass of water vapor to the mass of dry air contained in the same volume:

$$W = \frac{m_w \text{ (kg water vapor in the mixture)}}{m_a \text{ (kg dry air in the mixture)}} \tag{2.4}$$

From equation (2.2) and (2.3), we obtain

$$W = \frac{M_w}{M_a} \times \frac{p_w}{p_a} \tag{2.5}$$

$$W = \frac{18.01528}{28.9645} \times \frac{p_w}{p_a} \tag{2.6}$$

$$W = \frac{0.62198\, p_w}{p - p_w} \tag{2.7}$$

The relationship between the humidity ratio and the relative humidity is expressed as

$$W = \frac{0.62198 \varnothing_w p_{ws}}{p - \varnothing_w p_{ws}} \tag{2.8}$$

where p = 101.325 kPa at standard condition.

c) Specific volume

The specific volume (v), m^3/kg dry air of a moist air mixture is the volume of that mixture per unit mass of dry air:

$$v = \frac{V}{m_a} \tag{2.9}$$

From the equation (2.2), this can also be expressed as

$$v = \frac{RT_{abs}}{p_a M_a} \tag{2.10}$$

Writing $p_a = p - p_v$, and substituting $M_a = 28.9645$ in the last expression gives

$$v = \frac{RT_{abs}}{(28.966)(p - p_w)} \tag{2.11}$$

Substituting the value for the universal gas constant R, and the last expression is reduced to

$$v = \frac{0.2871 T_{abs}}{(p - p_w)}, \ m^3/\text{kg dry air} \tag{2.12}$$

d) Relative humidity

The relative humidity (ϕ_w) of moist air at a given temperature and pressure is the ratio of the existing partial vapor pressure (p_v) to the saturation pressure at the same temperature and pressure (p_{vs}):

$$\emptyset_w = \frac{p_w}{p_{ws}} \tag{2.13}$$

This quantity has no unit and is often expressed in terms of percentage.

e) Saturation vapor pressure

Under a given pressure, the maximum vapor pressure, or saturation vapor pressure is related to the temperature. From ideal gas law this relationship is given as:

$$p_{ws}(T_{abs}) = \exp\left[53.5224 - \frac{6834.27}{T_{abs}} - 5.17\ln(T_{abs})\right], \text{kPa} \tag{2.14}$$

A similar relationship is given by ASHARE (ASHARE, 2009) for temperature range of 0- 200°C as:

$$p_{ws}(T_{abs}) = \exp\left[\frac{C_1}{T_{abs}} + C_2 + C_3T_{abs} + C_4T_{abs}^2 + C_5T_{abs}^3 + C_6\ln(T_{abs})\right], \text{kPa} \tag{2.15}$$

where

C_1 = -5.8002206 x 10^3

C_2 = -5.516256

C_3 = -4.8640239 x 10^{-2}

C_4 = 4.1764768 x 10^{-5}

C_5 = -1.4452093 x 10^{-8}

C_6 = 6.5459673

f) Wet bulb temperature

The wet-bulb temperature of air is measured by a thermometer whose bulb is covered by a muslin sleeve which is kept moist with distilled and clean water, freely exposed to the air and free from radiation.

g) Enthalpy of moist air

The enthalpy of moist air is the sum of the enthalpy of each constituent. The enthalpy of dry air h_a is given as

$$h_a = C_a(T - T_R), \ \frac{\text{kJ}}{\text{kg}} \tag{2.16}$$

where

C_a is the specific heat capacity, kJ/ (kg.K) of dry air

T is its (dry bulb) temperature

T_R is the reference temperature

In SI unit, T_R is at 0°C. For the normal temperature range (0-100°C), the value of C_a is 1.006 kJ/ (kg.K). Therefore, the enthalpy of dry air can be expressed as

$$h_a = 1.006T,\ \frac{kJ}{kg} \tag{2.17}$$

The enthalpy of water vapor h_v comprises the enthalpy of water vapor at the reference temperature h_g = 2501 kJ/ kg, and the enthalpy of superheated vapor at T:

$$h_v = h_g + C_{pv}T,\ \frac{kJ}{kg} \tag{2.18}$$

where C_{pv} is the specific heat capacity of the superheated vapor. Its value is given as 1.805 kJ/(kg.K). The enthalpy of moist air (*h*) is then given as

$$h = h_a + Wh_v \tag{2.19}$$

$$h = 1.006T + W(2501 + 1.805T),\ kJ/kg\ dry\ air \tag{2.20}$$

h) Dew-point temperature

The dew point temperature of a moist air is the saturation temperature reached when the air is cooled down without moisture added or extracted from it. The dew-point temperature can be calculated from the partial vapor pressure p_v of the given moist air using the following relationship (ASHARE, 2009): Between dew points of 0 and 93°C,

$$T_{dp} = 6.54 + 14.526\ln(p_w) + 0.7389[\ln(p_w)]^2 + 0.09486[\ln(p_w)]^3 + 0.4569(p_w)^{0.1984},\ ^\circ c \tag{2.21}$$

2.1.2 Psychrometric Chart

Psychrometric charts as shown in Figure 2.1 are constructed to assist in obtaining value of a property of moist air. If two property values of the moist air are given, so the state of moist air is exactly known. The chart can be used to find all other property values of a given moist air condition by the following:

- Humidity ratio
- Enthalpy
- Specific volume
- Dry-bulb temperature
- Wet-bulb temperature

- Relative humidity

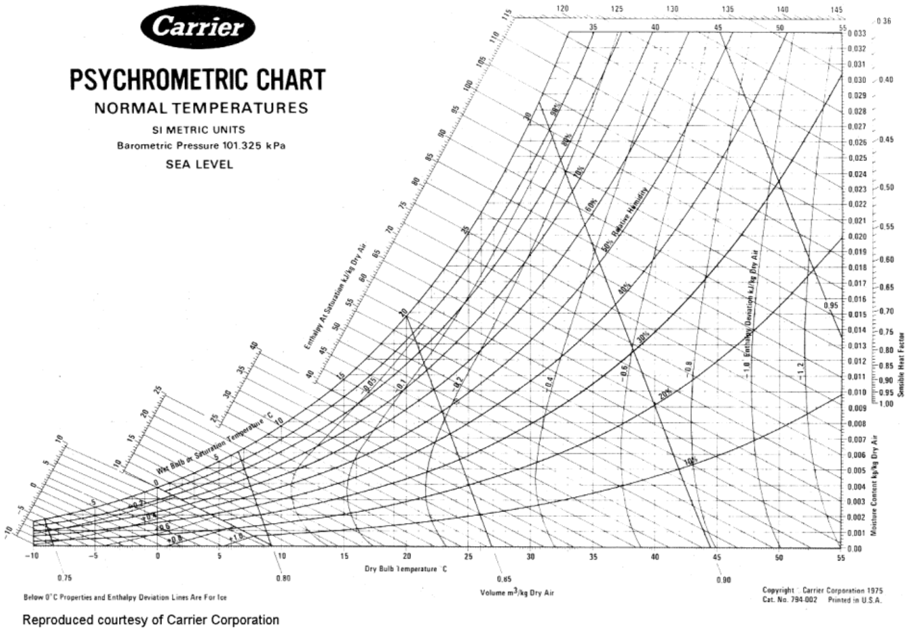


Figure 2.1 Psychrometric chart

2.2 Desiccant Dehumidifier types

Air dehumidification can be achieved by two methods: (I) cooling the air below its dew point and removing moisture by condensation, or (2) sorption by a desiccant material. Desiccants in either solid or liquid forms have a natural affinity for removing moisture. As the desiccant removes the moisture from the air, desiccant releases heat and warms the air, i.e., latent heat becomes sensible heat. The dried warm air can then be cooled to desired comfort conditions by sensible coolers (e.g., evaporator coils, heat exchangers, or evaporative coolers). To re-use the desiccant, it must be regenerated or reactivated through a process in which moisture is driven off by heat from an energy source such as electricity, waste heat, natural gas, or solar energy.

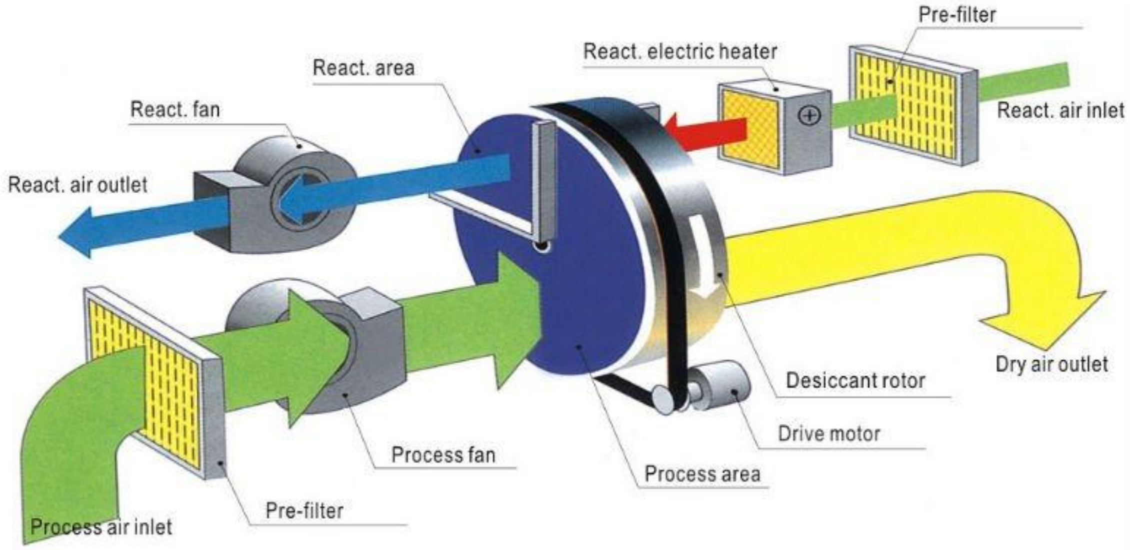


Figure 2.2 Solid-Desiccant-Wheel Dehumidifier (Munters Cargocaire)

For industrial applications, solid desiccant cycles use dual-column packed-bed dehumidifiers; however, the most appropriate dehumidifier configuration for air-conditioning applications is the rotary wheel . The air to be dehumidified enters the system, comes into contact with the desiccant wheel, and exits the dehumidifier hot and dry. The wheel is then rotated so that the desiccant portion that has picked up moisture is exposed to hot reactivation air and its moisture removed.

Since the air leaving the desiccant is heated because of the release of heat adsorption, there is a need for cooling the dried air in cooling applications. This can be accomplished with a sensible heat exchanger such as a heat pipe or with a standard vapor-compression cooling coil. Figure 2.3 shows schematics of a desiccant air conditioner incorporating direct-evaporative coolers and a rotary solid-desiccant wheel.

Figure 2.4 is a schematic of a liquid-desiccant dehumidification system. In a liquid system, there are two separate chambers—one to perform the dehumidification (or conditioning) and the other to reactivate (or regenerate) the desiccant. The processed air from the dehumidification chamber enters into the conditioned space. The desiccant, leaving the dehumidification chamber containing absorbed moisture, goes through a heat exchanger and down to the regenerator, where heat is added to remove the moisture. The liquid desiccant is pumped continually between the two chambers when dehumidification is needed.

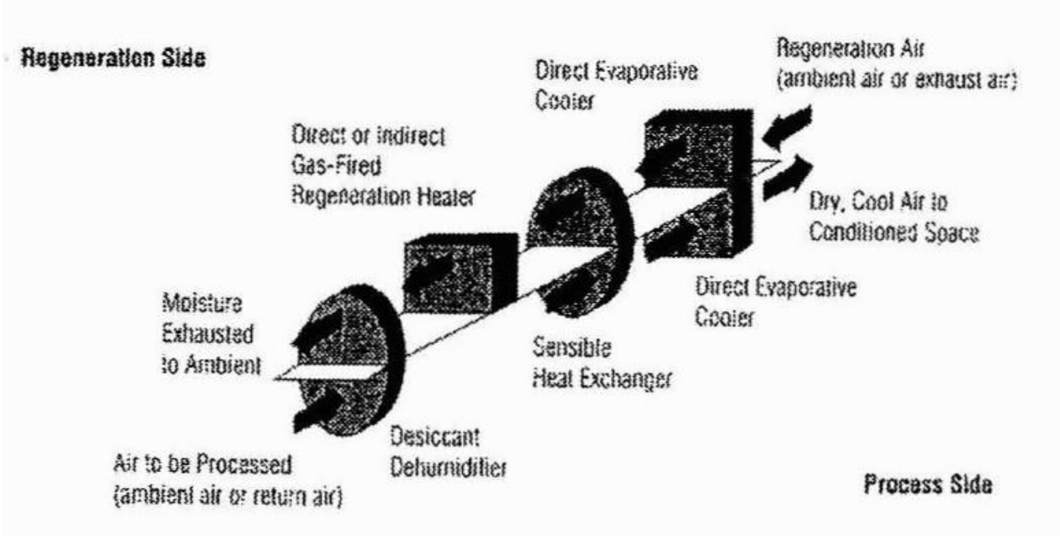


Figure 2.3 Schematic of a Solid-Desiccant Air Conditioner

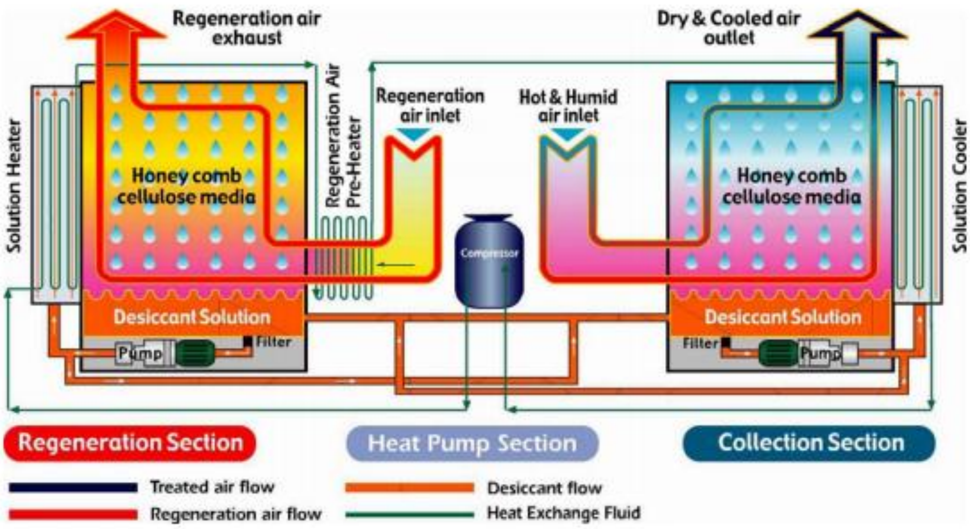


Figure 2.4 Schematic of a Liquid-Desiccant Dehumidification System

2.3 Desiccant Dehumidification Applications

Desiccant systems are especially useful when the latent load is high (i.e., when the latent-to-sensible heat ratio is high), because they remove moisture more economically than they remove sensible heat. Another desirable situation is when the cost of dehumidification with a desiccant is lower than the cost of dehumidification with a refrigeration system. This is where thermal energy comes into the picture: there are instances where desiccant regeneration done by waste heat, natural gas, or off-peak electricity is more economical compared to regular electric refrigeration. Because there is no need for reheating with desiccant dehumidification systems, another appropriate use is when conditioned air must be reheated after coming out of a coil to reach a comfortable dry-bulb temperature. Finally, the use of a desiccant is well-suited to the case where dehumidification is required at levels below freezing dew-point temperatures. For example, an ice arena has is a great deal of humidity, but the cooling coil has to cool below the freezing point. In such an environment, dehumidification with desiccants can play a major role.

2.4 Performances of Desiccant Dehumidification

To maximize benefit for the desiccant system performance and for proper sizing, a clear understanding of the variables that affect the performance is essential. There are eight key parameters which affect the performance of desiccant. The process air expected to dehumidification before entering the room is called process air and reactivation air is used to regenerate desiccant after desiccant surface is humid from adsorbing a lot of water. These parameters include:

- Process air moisture
- Process air temperature
- Process air velocity through the desiccant
- Reactivation air temperature
- Reactivation air moisture
- Reactivation air velocity through the desiccant
- Amount of desiccant presented to the reactivation and process airstreams
- Desiccant sorption-desorption characteristics

In any system, these variables change because of weather, and variations in moisture load moreover, the exact effect of each parameter depends on the type of desiccant dehumidifier. In discussing desiccant dehumidifier performance, it must be made on basic assumption at the start to the dehumidifier is operating at equilibrium. In other words, the total energy on the process side is balanced by the energy in regeneration side. If the system is not in equilibrium, a desiccant dehumidification will not perform in an easily predictable manner. Uncontrolled of airflows and temperatures tend to the system is not in equilibrium.

2.5 Performance indexes

a) Moisture removal rate ($\Delta\boldsymbol{D}$)

$$\Delta D = D_{a,in} - D_{a,out}$$

Where $D_{a,in}$ and $D_{a,out}$ are humidity ratio of process air inlet and outlet, respectively in g/kg

b) Thermal coefficient of performance (COP_{th})

Thermal coefficient of performance (COP_{th}) is used to show the overall energy efficiency. It is defined as the ratio between average enthalpy exchanged of process air in efficiency dehumidification process (Q_{cool}) and average heat exchanged of water in effective regeneration process (Q_{reg}).The electrical power input of the fans is neglected for heat energy is the primary source of energy.

$$\text{COPth} = \frac{Q_{cool}}{Q_{reg}} = \frac{m_a(h_{a,in} - h_{a,out})}{C_p m_w(T_{w,in} - T_{w,out})}$$

Where m_w is the mass flow rate of water (kg/s),C_p is specific heat of water (kJ/kg·K),and T_w is the temperature of water, m_a is the mass flow rate of air (kg/s), h_a is the enthalpy (kJ/kg) of process air.

c) Moisture removal capacity (MRC)

Moisture removal capacity (MRC) shows the amount of moisture removed in the air passing the desiccant wheel. MRC is the calculation of the desiccant wheel sorption rate. The formulation of the MRC is presented as

$$MRC = \dot{m}_{a,deh}(W_{in} - W_{out})$$

Where $\dot{m}_{a,deh}$ is mass flow rate of dehumidification process air in (kg/s), W in term of humidity ratio (kg_w/kg_{da})

d) Moisture removal regeneration (MRR)

Moisture removal regeneration (MRR) is the performance of the desiccant wheel in removal of moisture from its desiccant surface. The formulation of MRR is presented as

$$MRR = \dot{m}_{a,reg}(W_{in} - W_{out})$$

Where $\dot{m}_{a,reg}$ is mass flow rate of regeneration process air in (kg/s), W in term of humidity ratio (kg_w/kg_{da})

e) Dehumidification effectiveness (E_{deh})

E_{deh} represents the ratio between the humidity reduction across the heat exchanger and the inlet humidity ratio, (Mandegari MA., Pahlavanzadeh H.)

$$\text{Dehumidification effectiveness}= \frac{W_{in}-W_{out}}{W_{in}}$$

Where W in term of humidity ratio (kg_w/kg_{da})

f) Regeneration effectiveness (E_{reg})

The calculation of the amount of regeneration heat that consumed for desorbed water is given in the regeneration effectiveness (E_{reg}).

$$\text{Regeneration effectiveness}= \frac{W_{in}-W_{out}}{W_{in}}$$

Where W in term of humidity ratio (kg_w/kg_{da})

CHAPTER 3
METHODOLOGY

This study focuses on a desiccant dehumidification for decreasing cooling the load in a tropical climate. The dehumidification system comprises main equipments of desiccant, heat exchanger, heater, chiller, ventilation fan and pump.

3.1 Dehumidification System

3.1.1 Dehumidification Schematic Diagram

Fig 3.1 shows the schematic of the system. This system consists of two sets of heat exchangers. In addition, a chiller provides cold water for dehumidification process. A heater generates hot water for regeneration process. In the system installed two storage water tanks for store hot water and cold water. The 3-way valves are installed for switched direction between hot and cold water. Timers used for controlling the 3-way valves by time setting. The process of this system divided for two modes dehumidification and regeneration modes. When one part of heat exchanger in dehumidification process supplied by cold water and outlet water return to cold water tank. Meanwhile, other part of heat exchanger is in regeneration mode by supply hot water and water outlet water return to hot water tank.

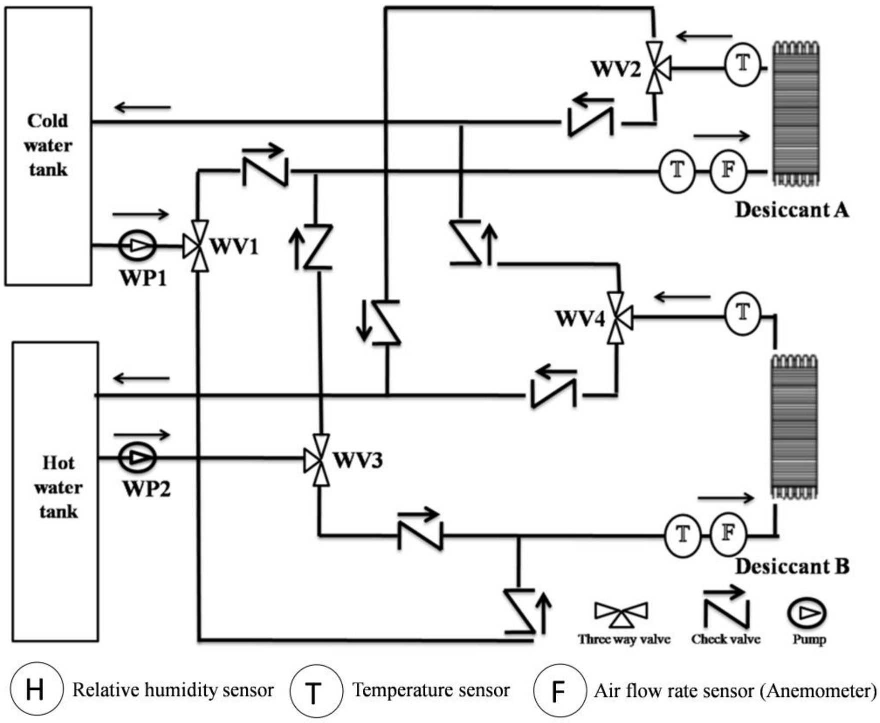


Figure 3.1 Schematic diagram of the desiccant dehumidification system

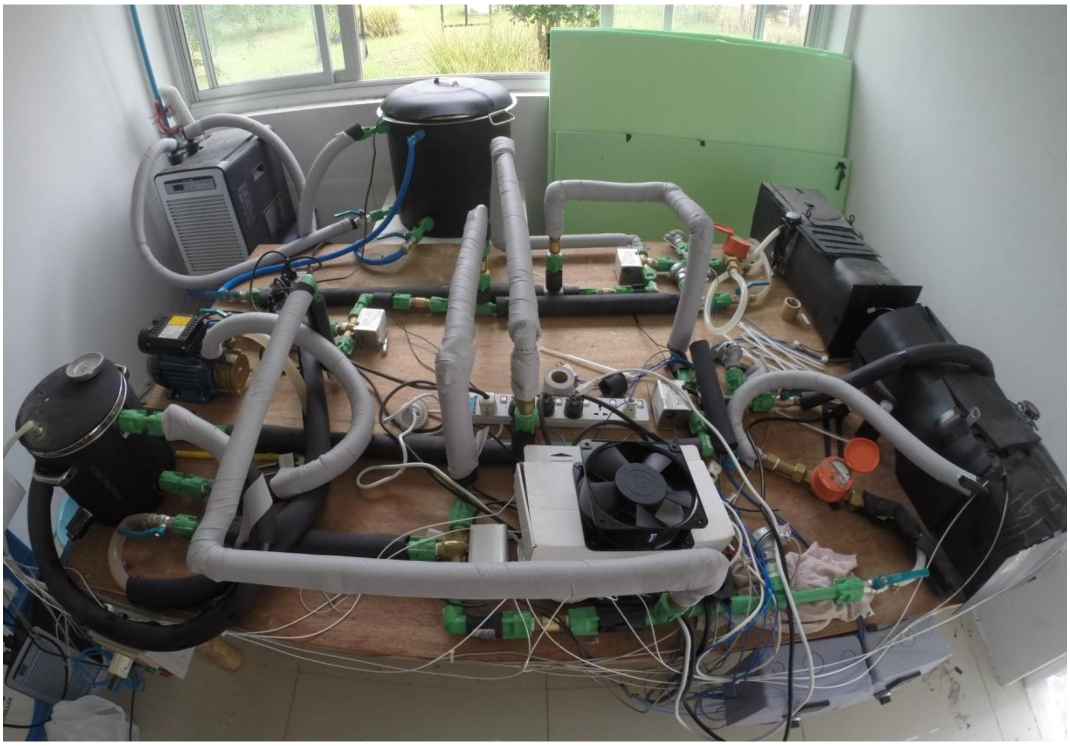


Figure 3.2 Pictorial view of the desiccant dehumidification system

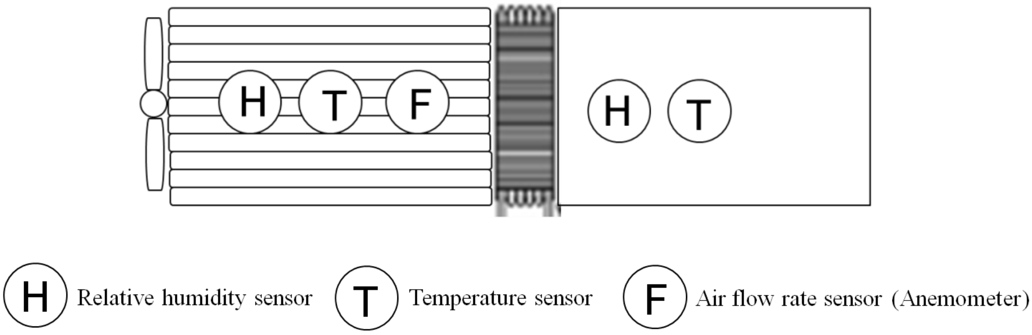


Figure 3.3 Experiment setup of desiccant heat exchanger A

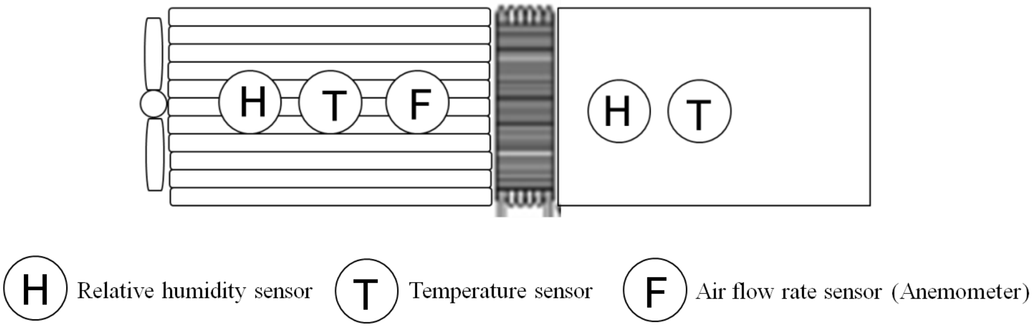


Figure 3.4 Experiment setup of desiccant heat exchanger B

Figure 3.2 shows the pictorial view of the dehumidification system and the position of the devices in the system, such as heater, chiller, hot water tank, cold water tank and etc. And Figures 3.3 and 3.4 shows the composition of the desiccant heat exchanger A and B, and the position of the measurement sensor installed in this system.

3.1.2 Heat Exchanger Coated with Silica-gel

a) Silica gel

The silica gel come from a Thai powerdry company .The chemical properties of silica sand is the same as silica gel. The only difference is size of silica sand is slightly smaller than silica gel. Silica sand has diameter of 0.1- 0.7 mm. Silica sand is available in porous, granular, and amorphous from, synthetically manufactured from the chemical reaction between sulfuric acid and sodium silicate. The internal structure of Silica sand is composed of a vast network of interconnected microscopic pores which attract and hold water, alcohol, hydrocarbons and other chemicals by the phenomena known as physical adsorption and capillary condensation.

Advantage of Silica-gel

Silica gel has the ability to adsorb up to one third of its own weight in water vapor. This adsorption efficiency is approximately 35% greater than typical desiccant clays, making Silica-gel the preferred choice where weight or efficiency are important factors.

Figure 3.5 Silica-gel (0.1-0.7mm) before and adsorbed humidity after 1 day

b) Ball mill

For mashed silica-gel to small particle used Ball mill planetary mill from Fritsch company type planetary mill pulverisette 5

Figure 3.6 Equipment for mashed (planetary mill)

c) Heat exchanger

In this experiment, used heat exchanger from hot coil heat exchanger from motorcycle of Honda CBR model. The heat exchanger size is 0.25x0.18x0.025m.The overall heat transfer coefficient is 10 w/m^2K. And the area of the heat exchanger is 0.42 m^2.

Figure 3.7 Heat exchanger

d) Adhesive spray

Adhesive Spray used in this experiment should have high temperature resistance.

Figure 3.8 Adhesive Spray 3M

e) Spray color

For coating silica-gel on the heat exchanger, used spray color for pressing the small size particles of silica-gel to the surface of heat exchanger. After coated silica-gel the arrangement of silica-gel on the heat exchanger is regularly.

Figure 3.9 Spray color and Air compressor

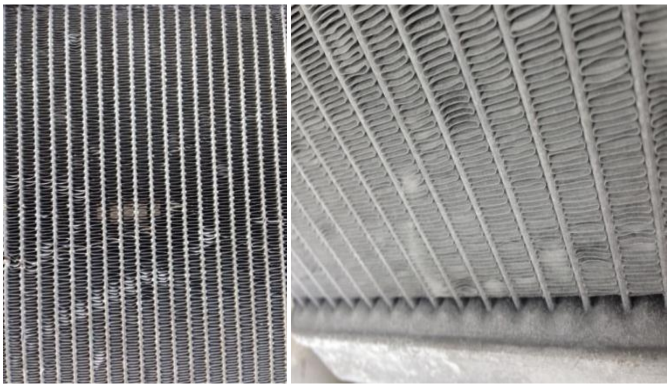


Figure 3.10 Heat exchanger before and after coated silica-gel

3.1.3 Hot and cold water system

a) Heater

Define:	Area of a fin (front and back sides)	= 0.42 m^2
	Cross-sectional area of air outlet	= 0.034 m^2
	Maximum of water flow rate	= 16 l/min
	Maximum of air flow rate	= 146.86 m^3/h
	Maximum of air velocity	= 1.2 m/s
	Specific heat of water at 80°C	= 4194 J/kg°C

Calculation of the required heat transfer at the water flow rate of 16 l/min

$$Q = \dot{m}C_p\Delta T$$

$$Q = 0.267\,\frac{kg}{s} \times 4194\,\frac{J}{kg°C} \times (1.5)°C$$

$$Q \approx 1{,}679\ W$$

Heating capacity of the heater must be more than 1.68 kW.

Calculate volume of hot water tank by:

$$Q = m_w C_p \Delta T$$

$$1679\ W = m_w \times 4194 j/(kg°C) \times (90\text{-}30)°C$$

$$m_w = 0.007\ kg$$

from $\rho = m/v$

$$1000\ (kg/m^3) = (0.007\ kg)/v$$

$$V \approx 7\ liter$$

Hot water tank is 7 liter

The temperature of hot water for regeneration required is relatively (50-80 °C)

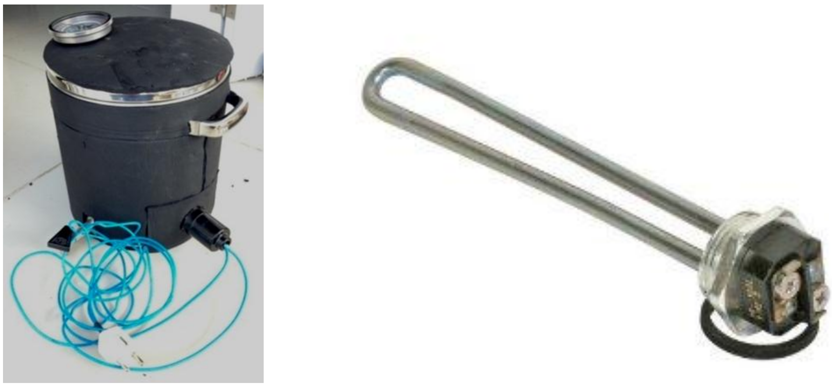


Figure 3.11 Heater

b) Chiller

Calculation of the required chiller capacity by:

$$Q = \dot{m} C_p \Delta T$$

$$Q = 0.167 \times 4181 \frac{j}{kg°C} \times (1)°C$$

$$Q = 696\ W$$

Chiller capacity for cold water is 696W

Calculate volume of cold water tank by:

$$Q = m_w C_p \Delta T$$

$$696\ W = m_w \times 4181\ j/(kg°C) \times (30\text{-}25)°C$$

$$m_w = 0.033\ kg$$

from $$\rho = m/v$$

$$1000\ (kg/m^3) = (0.033\ kg)/v$$

$$V \approx 33\ liter$$

cold water tank is 33 liter

Figure 3.12 Chiller

c) Hot and cold water pumps

In this experiment, used 2 water pumps for hot and cold water to supply both heat exchangers.

Figure 3.13 Hot and cold water pump

After calculating mass the flow rate of the pump, then selected pump have water flow rate 17.5 l/min for hot water and 12 l/min for cold water. The size of pump was higher water flow rate than calculated because it can protect pressure drop in this system.

d) Ventilation Fan

For this experiment, installed 2 sets of ventilation fan in this system. Both are responsible to drive the ambient air for dehumidification and regeneration processes Figure 3.22 shows the ventilation fan of system.

Figure 3.14 Ventilation fan

3.1.4 System Controllers

a) Time Controller

Digital timer was installed to control the cycle time for dehumidification and regeneration processes.

Figure 3.15 Digital timer

b) Three -way valve

In this experiment, the direction of hot and cold water was controlled by electricity three way valves before supplying to the heat exchanger. The three way valves were controled by a digital timer.

Figure 3.16 Three-way valve

3.2 Measuring instrument

3.2.1 Air velocity and humidity sensor

The air velocity, temperature and humidity ratio of the inlet and outlet air were measured by four set high accuracy and multi-functional digital hot wire anemometer (type: AM4224SD produced by Lutron Instruments).The measurement range of air velocity 0.2-25m/s with accuracy of ±5% The measurement range of temperature is 0-50°C with accuracy of ±0.8°C and measurement range of relative humidity is 5-98%RH with accuracy of ±1%RH.

Figure 3.17 Air velocity and humidity sensor

3.2.2 Water temperature sensor

In this system, the temperature of the hot and cold water were measured by PT-100RTD, with an accuracy of ±0.15°C.

Figure 3.18 PT-100

3.2.3 Hot and cold water flow meter

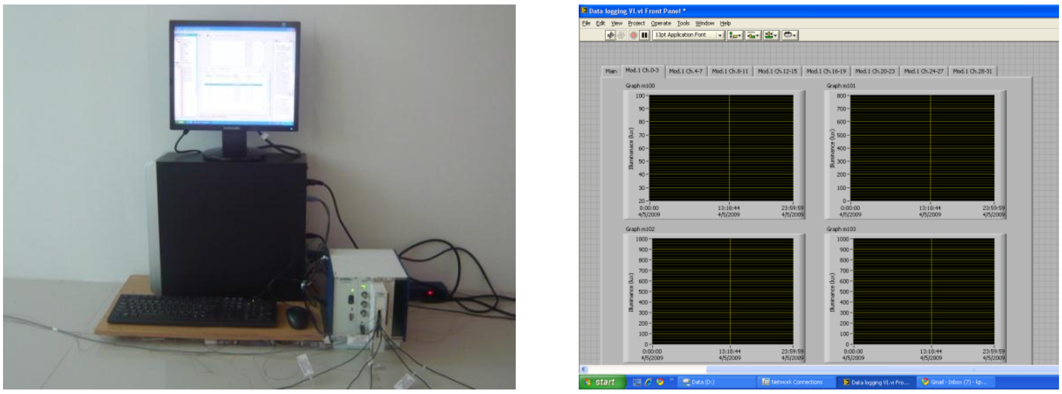
Two sets of water flow meters were installed to measure the water flow rate of hot and cold water for each case in this experiment. Water flow meter can resist high temperature of water.

Figure 3.19 Water flow meter

3.2.4 Data logging system

In the experiment building, a National Instrument (NI) data logging system was provided to serve intensive measurements of ambient temperature, hot water temperature, cold water temperature, and humidity ratio inlet and outlet of the air.

The software program of the computer controller was developed by the use of the data acquisition toolbox of National Instrument (NI) Lab View software. Figure 3.20(a) is a photograph of the NI computer data controller with a SCXI 1000 chassis and a signal conditioner SCXI-1102 mounted with a front-end SCXI-1503. Figure 3.20(b) exhibits a front panel of the real time data monitoring. All the measured data from the sensors are recorded onto the computer hard disc every ten seconds.



(a) Data acquisition (DAQ) system

(b) Font panel the DAQ system

Figure 3.20 Data acquisition system for the experiment

3.3 Experimentation and Facilities

The experiments of a desiccant air dehumidification were conducted at Bangkuntien Campus of King Mongkut's University of Technology, Thonburi. The campus is located in the suburban, southern part of Bangkok (latitude 14.7°N and longitude 100.5°E). The main facilities, a laboratory building, and a meteorological station. The details of each facility including the experimental procedure are described below.

Figure 3.21 Experiment house

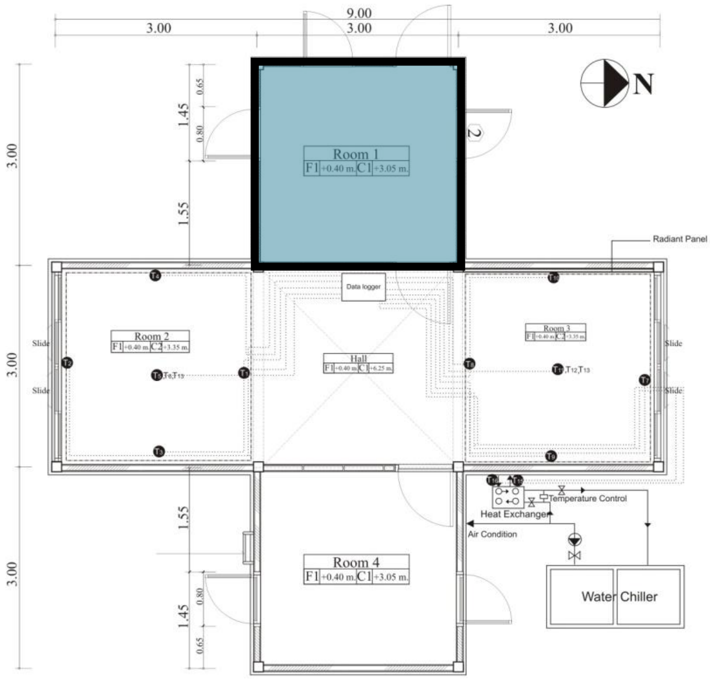


Figure 3.22 Floor plan of the experimental building

3.4 Experiment schemes

As presented in Table 3.1, different schemes of the experiments were set to investigate the characteristics and performance of a desiccant air dehumidification system. Table 3.1 shows the regeneration temperature, the dehumidification temperature, air velocity and hot and cold water flow rate are the parameters to be varied each of which way set for three particular values. Regeneration temperatures of hot water are varied for 55, 65, 75 and 80°C. Air velocities were varied for 0.5, 0.8, and 1.2 m/s. Lastly, the water flow rate of the hot and cold water flow rate were varied for 10, 13, 17 l/min and 5, 7, 10 l/min, respectively.

Table 3.1 Total experimental conditions of a desiccant air dehumidification system

Case	Regeneration temperature (°C)	Dehumidification temperature (°C)	Air velocity (m/s)	Hot water flow (l/min)	cold water flow (l/min)
1-3	75	25	0.5 0.8 1.2	10	5
4-6	75	25	0.5 0.8 1.2	13	7
7-9	75	25	0.5 0.8 1.2	17	10
10	75	15	1.2	13	7
11	75	20	1.2	13	7
12	75	30	1.2	13	7
13	55	25	1.2	13	7
14	65	25	1.2	13	7
15	80	25	1.2	13	7

CHAPTER 4
RESULTS AND DISCUSSIONS

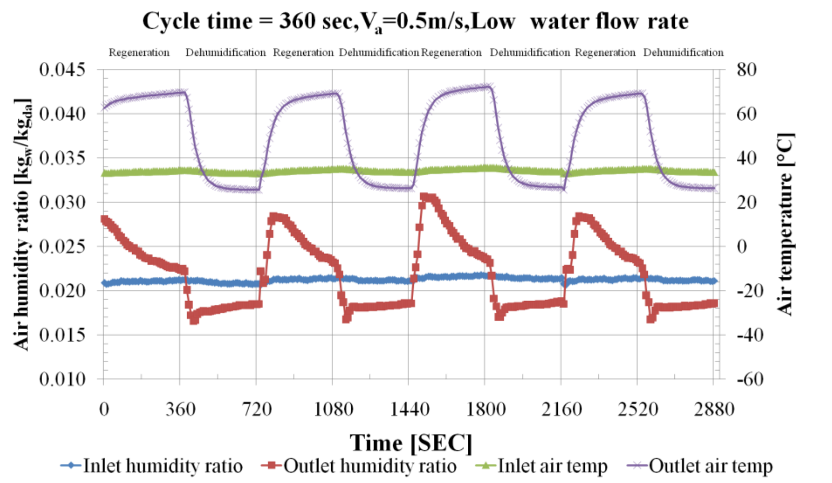
4.1 Experimental results

In this experiment, the measured data was divided into 9 conditions test exclude the effect of hot and cold water temperature. There are three conditions for water flow rate and three condition for air velocity in process 0.5, 0.8, and 1.2 m/s, respectively. The cycle times in this experiment varied for 360, 480, 600, 720 and 900 second. The measured parameters in this experiment include temperature, humidity ratio and flow rate of air inlet and air outlet.

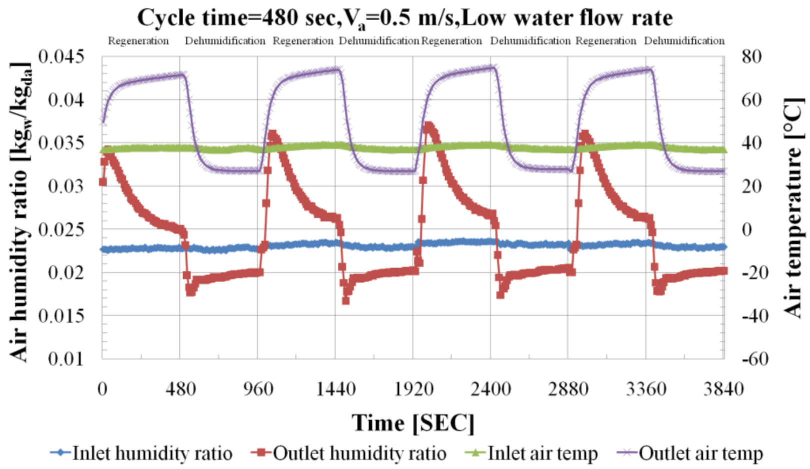
The measured data are moisture adsorption from the dehumidification process (D_{ads}) and the moisture desorption from the regeneration process (D_{des}). The effective of the system represented by the Thermal coefficient of performance (COP_{th}), Moisture removal capacity (MRC), Moisture removal regeneration ratio (MRR), Dehumidification effectiveness (E_{deh}), and regeneration effectiveness (E_{reg}).The results are shown in Table 4.1 below.

Table 4.1 The result of an experimental performance at the dehumidification and regeneration systems under difference conditions

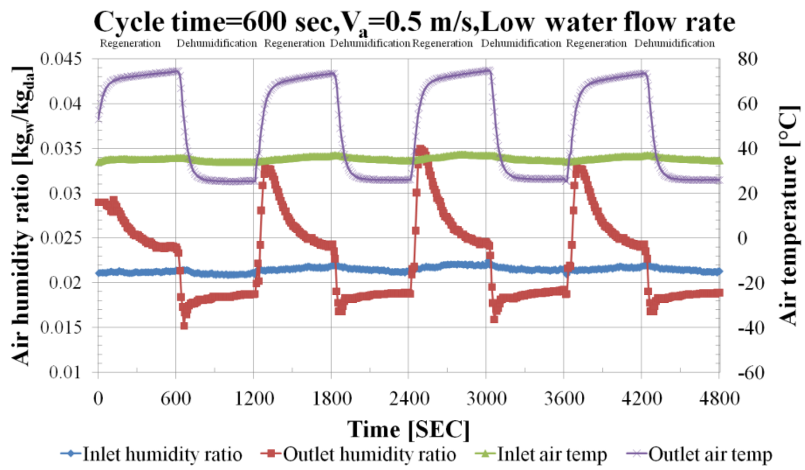
Water Flow rate	Air velocity (m/s)	Cycle time (sec)	Average moisture adsorption (g_w/kg_{da}) Dehumidification	Average moisture desorption (g_w/kg_{da}) Regeneration	COP_{th}	MRC (kg/h)	MRR (kg/h)	E_{deh}	E_{reg}
Low	**0.5**	**360**	3.00	4.25	0.45	0.22	0.26	0.14	0.20
		480	3.30	6.05	0.53	0.23	0.37	0.14	0.26
		600	2.94	5.56	0.50	0.22	0.34	0.14	0.26
		720	3.00	5.23	0.43	0.21	0.32	0.13	0.24
		900	2.70	5.72	0.39	0.19	0.35	0.12	0.26
		Avg	**2.99**	**5.36**	**0.46**	**0.21**	**0.33**	**0.13**	**0.24**
	0.8	**360**	3.60	6.13	0.40	0.42	0.60	0.15	0.26
		480	2.90	5.21	0.37	0.34	0.51	0.13	0.23
		600	2.79	4.39	0.50	0.32	0.43	0.13	0.20
		720	2.90	4.29	0.40	0.34	0.42	0.13	0.19
		900	2.91	2.76	0.36	0.33	0.27	0.13	0.12
		Avg	**3.02**	**4.55**	**0.41**	**0.35**	**0.45**	**0.13**	**0.20**
	1.2	**360**	3.80	6.39	0.76	0.65	0.92	0.13	0.33
		480	3.60	6.18	0.70	0.63	0.89	0.14	0.27
		600	3.50	5.07	0.69	0.60	0.73	0.14	0.20
		720	3.00	5.21	0.67	0.52	0.75	0.13	0.22
		900	2.90	5.00	0.65	0.50	0.72	0.13	0.22
		Avg	**3.48**	**5.57**	**0.71**	**0.60**	**0.82**	**0.14**	**0.26**
Medium	**0.5**	**360**	4.40	2.29	0.48	0.32	0.14	0.19	0.10
		480	4.50	1.80	0.40	0.32	0.11	0.19	0.08
		600	4.90	2.45	0.43	0.35	0.15	0.20	0.10
		720	4.90	3.43	0.62	0.35	0.21	0.20	0.13
		900	4.40	2.12	0.54	0.32	0.13	0.18	0.09
		Avg	**4.62**	**2.42**	**0.49**	**0.33**	**0.15**	**0.19**	**0.10**
	0.8	**360**	3.50	2.25	0.47	0.41	0.22	0.15	0.10
		480	3.70	1.94	0.42	0.43	0.19	0.16	0.08
		600	3.80	3.68	0.41	0.44	0.36	0.15	0.15
		720	3.80	1.94	0.39	0.44	0.19	0.16	0.08
		900	3.80	1.02	0.47	0.43	0.10	0.15	0.04
		Avg	**3.72**	**2.17**	**0.43**	**0.43**	**0.21**	**0.15**	**0.09**
	1.2	**360**	2.80	2.71	0.37	0.49	0.39	0.13	0.10
		480	2.60	2.92	0.40	0.45	0.42	0.12	0.14
		600	3.15	3.26	0.41	0.54	0.47	0.14	0.14
		720	2.90	2.08	0.36	0.47	0.30	0.12	0.09
		900	2.60	2.78	0.39	0.45	0.40	0.12	0.12
		Avg	**2.81**	**2.75**	**0.39**	**0.48**	**0.40**	**0.13**	**0.12**
High	**0.5**	**360**	4.20	7.19	0.44	0.30	0.44	0.18	0.32
		480	3.90	7.84	0.41	0.28	0.48	0.17	0.35
		600	3.64	7.68	0.54	0.26	0.47	0.17	0.37
		720	3.80	7.03	0.62	0.28	0.43	0.18	0.33
		900	3.60	7.19	0.44	0.26	0.44	0.17	0.34
		Avg	**3.83**	**7.39**	**0.49**	**0.28**	**0.45**	**0.17**	**0.34**
	0.8	**360**	5.30	2.66	0.65	0.61	0.26	0.22	0.11
		480	4.30	2.86	0.55	0.49	0.28	0.17	0.12
		600	3.58	4.29	0.54	0.41	0.42	0.17	0.20
		720	4.30	4.39	0.64	0.50	0.43	0.18	0.19
		900	4.00	4.49	0.78	0.46	0.44	0.16	0.18
		Avg	**4.30**	**3.74**	**0.63**	**0.49**	**0.44**	**0.18**	**0.16**
	1.2	**360**	5.00	5.42	0.49	0.87	0.78	0.19	0.18
		480	5.00	5.35	0.72	0.88	0.77	0.20	0.17
		600	5.23	5.76	0.61	0.90	0.83	0.21	0.23
		720	4.80	4.86	0.70	0.78	0.70	0.19	0.19
		900	5.00	5.97	0.79	0.87	0.86	0.20	0.21
		Avg	**5.01**	**5.47**	**0.74**	**0.86**	**0.79**	**0.20**	**0.20**



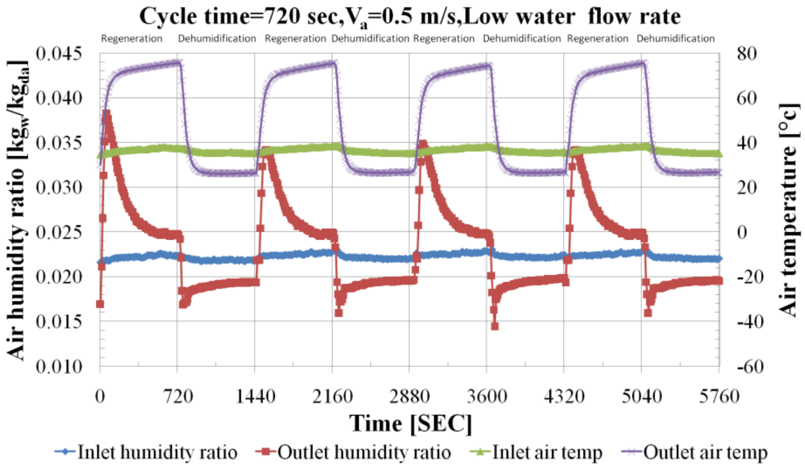
(a) Cycle time 360 seconds



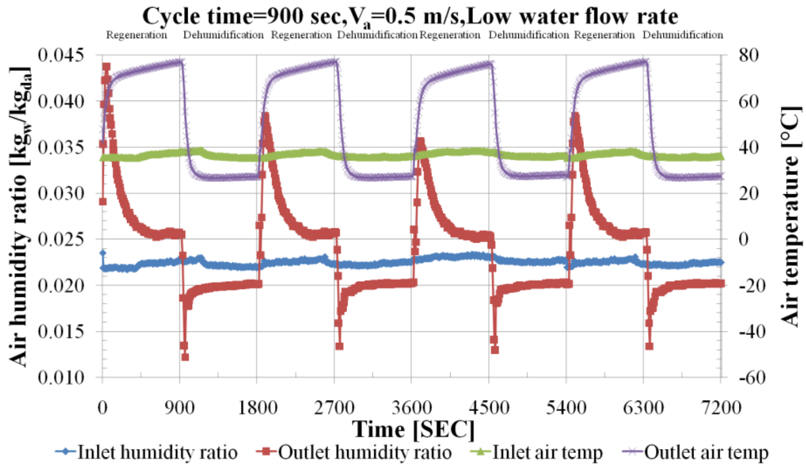
(b) Cycle time 480 seconds



(c) Cycle time 600 seconds



(d) Cycle time 720 seconds



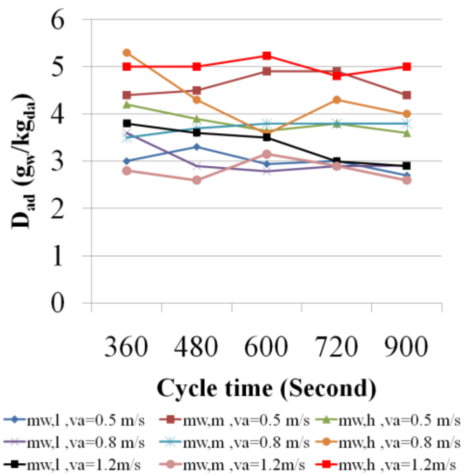
(e) Cycle time 900 seconds

Figure 4.1 Air humidity ratio and air temperature conditions of air velocity 0.5 m/s at low water flow rate

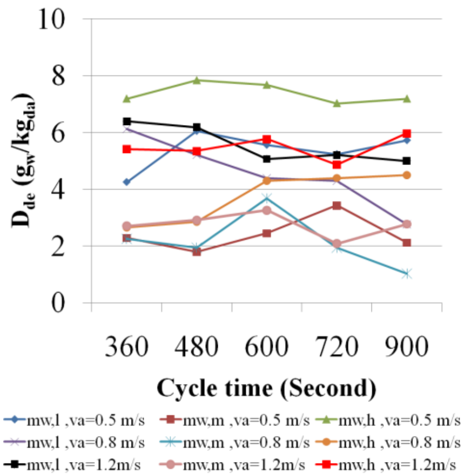
Figures 4.1(a)-(e) show the relation between air humidity ratio and air temperature on various cycle times of air velocity 0.5 m/s at low water flow rate. For this experiment, the temperature of the cold water supplied was set at nearly 25°C for dehumidification process and hot water supplied was nearly 75°C for regeneration process. At the regeneration time, the humidity ratio of air was higher than air inlet humidity ratio because the moisture desorbed from silica-gel which coated on the heat exchanger and released to the air. When changed mode for dehumidification time the humidity ratio of the air outlet dropped to lower than air humidity inlet and slightly increased to equilibrium condition with nearly to air humidity inlet.

4.1.1 Effect of cycle time

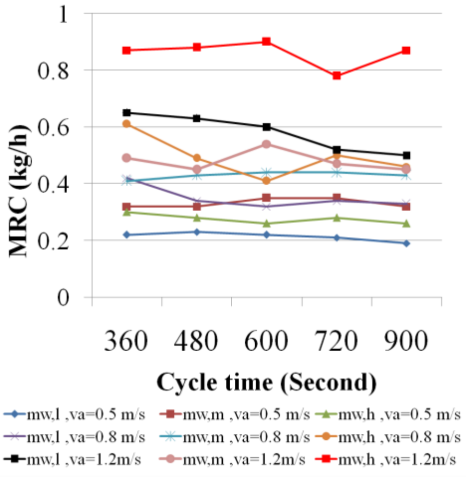
From Table 4.1, the performance indices of system from each case seem to be not significantly different except on cycle time 360 sec to 900 sec. The valves of performance average moisture adsorption, average moisture desorption, COP_{th}, MRC, MRR, E_{deh}, and E_{reg} are nearly same every cycle time in the same condition.



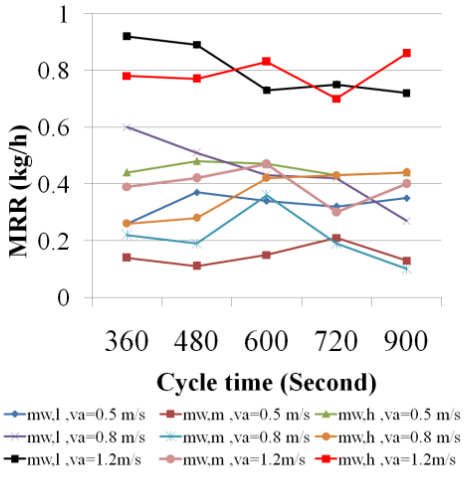
(a) Moisture adsorption (D_{ad})



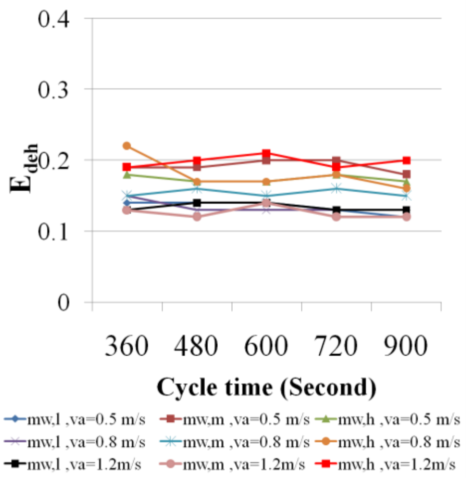
(b) Moisture desorption (D_{de})



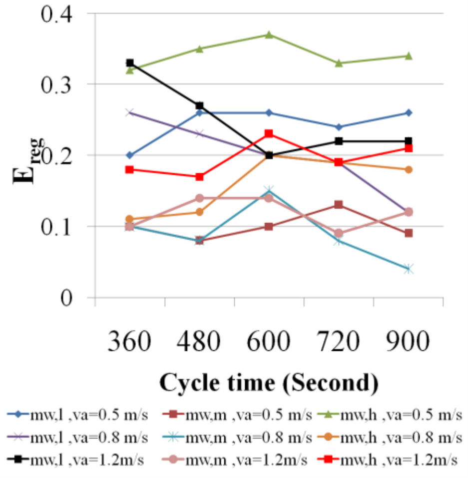
(c) Moisture removal capacity (MRC)



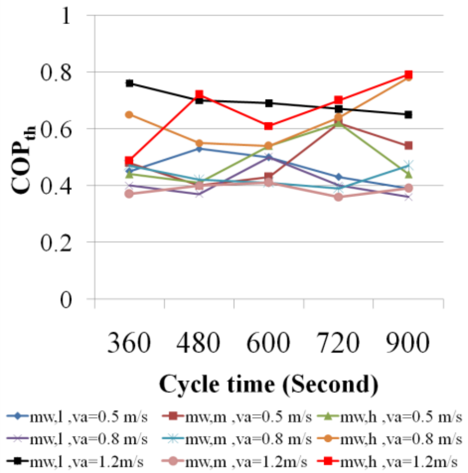
(d) Moisture regeneration ratio (MRR)



(e) Dehumidification effectiveness (E_{deh})



(f) Regeneration Effectiveness (E_{reg})



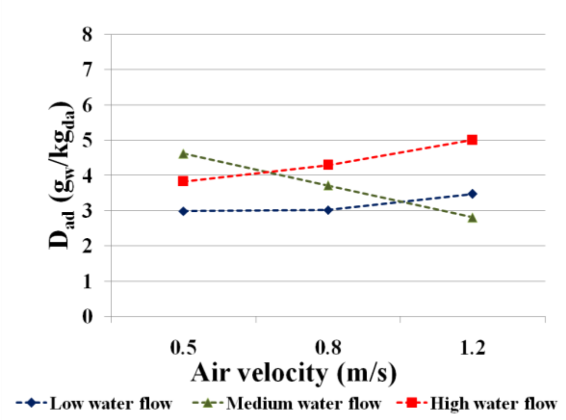
(g) Coefficient performance (COP_{th})

Figure 4.2 The results from the parametric desiccant dehumidifier system

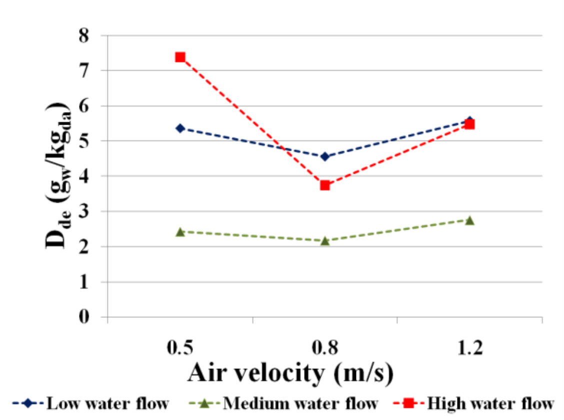
Figures 4.2(a)-(g) are illustrate all of parameters which represent the performance of the system. Figure 4.2(a) shows the moisture adsoption from 2.81 to 5.01 g/kg. Figure 4.2(b) shows the moisture desorption of the system those increased from 2.17 to 7.39 g/kg. MRC of the system is highest at high water flow rate and velocities of air 1.2 m/s. The values of parameter indices each of case are significantly different.MRR of the system is highest at low water flow rate and velocities of air 1.2 m/s condition, which nearly high water flow rate and velocity of air 1.2 m/s condition. Figure 4.2(e) illustrates the effectiveness of dehumidification (E_{deh}) process are from 0.13 to 0.22.The highest regeneration effectiveness (E_{reg}) is 0.34 at high water flow rate and velocity of air 0.5 m/s.

4.1.2 Effect of air velocity

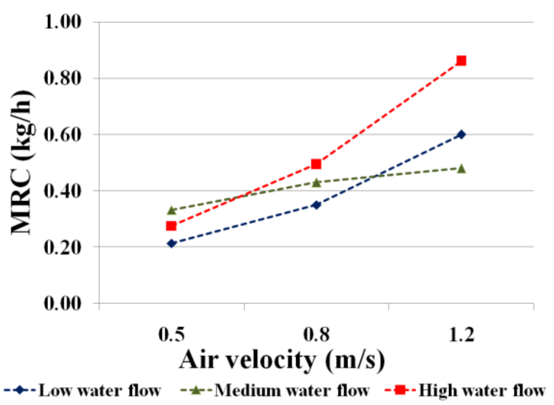
In this part of the experiment, the air velocity before being supplied to the air dehumidifier is separated to 3 valves such as 0.5 m/s, 0.8 m/s and 1.2 m/s, respectively. To evaluate the effect of air velocity to dehumidification and regeneration of the system.



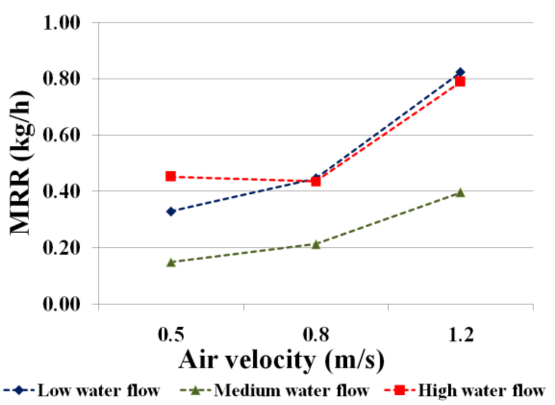
(a) D_{ad} of dehumidification section



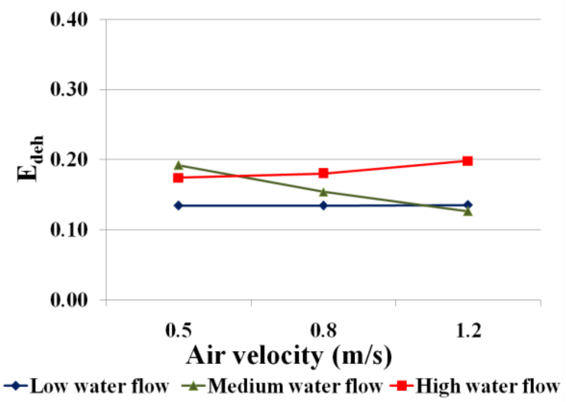
(b) D_{des} of regeneration section



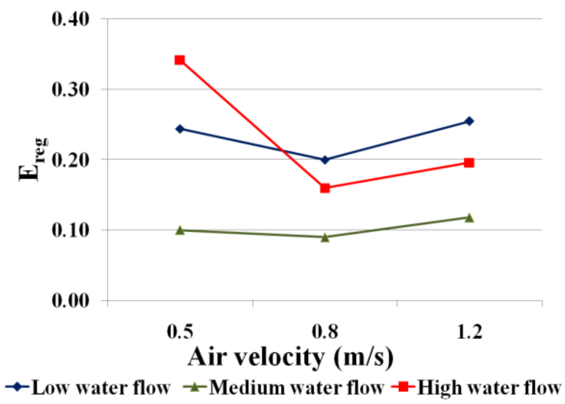
(c) MRC of dehumidification section



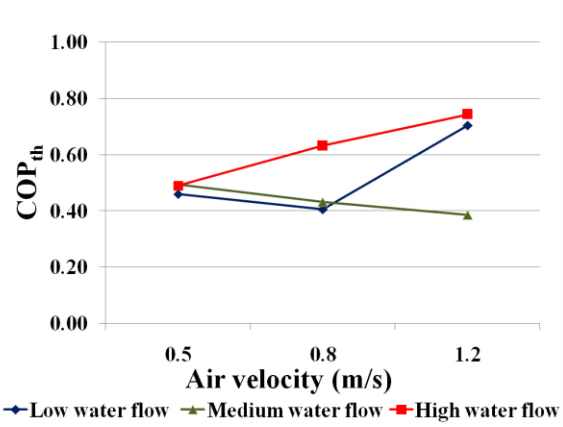
(d) MRR of regeneration section



(e) E_{deh} of dehumidification section



(f) E_{reg} of regeneration section



(g) COP_{th} of system

Figure 4.3 Effect of air velocity on the system

Table 4.2 Average heat exchanger rate under different inlet air velocities

Velocity of inlet air (m/s)	Q_a(kW)			Q_w(kW)		
	Low	**Medium**	**High**	**Low**	**Medium**	**High**
0.5	0.17	0.38	0.27	0.37	0.78	0.55
0.8	0.35	0.56	0.56	0.85	1.3	0.89
1.2	0.65	0.69	0.9	0.92	1.75	1.22

Figures 4.3(a)-(g) illustrate the experimental results on the effect of air velocity (v_a) on the system performance. The ranges of air velocity are 0.5, 0.8 and 1.2 m/s, respectively. Figure 4.3(a) shows the moisture adsorption increase when increasing air velocity at low water flow rate and high water flow rate. On the other hand, the moisture adsorption decrease when increasing air velocity on medium water flow rate.

Figure 4.3(b) shows the influences of moisture desorption in various air velocities. The results indicate that low air velocity preferred higher moisture desorption.

Figures 4.3(d)-4.3(e) show the influences of MRC and MRR on the effect of air velocity. When increase air velocity MRC and MRR also increasing. The range of MRC and MRR are 0.20 -0.85 kg/h and 0.18-0.82 kg/h, respectively.

Lastly, the COP_{th} of the system decreases when increasing air velocity is at low and medium water flow rate at 0.5,0.8 m/s and 0.5,0.8 and 1.2 m/s, respectively. However, the COP_{th} of high water flow trend to be increase from 0.49 to 0.74 because the temperature and humidity of ambient air cannot control as present on Figure 4.4. That should be effect to COP_{th} of system.

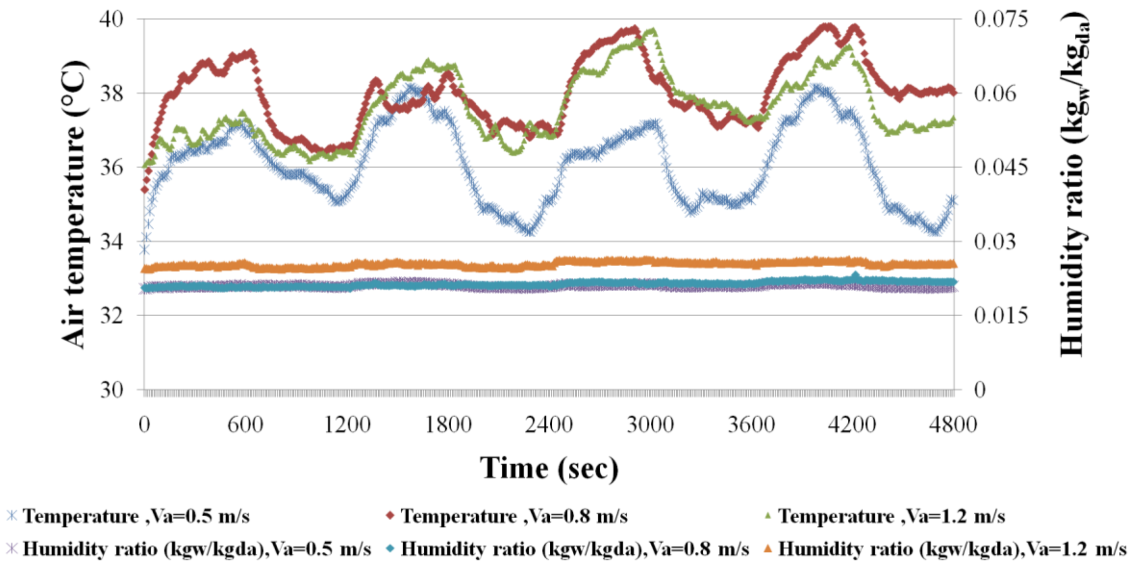


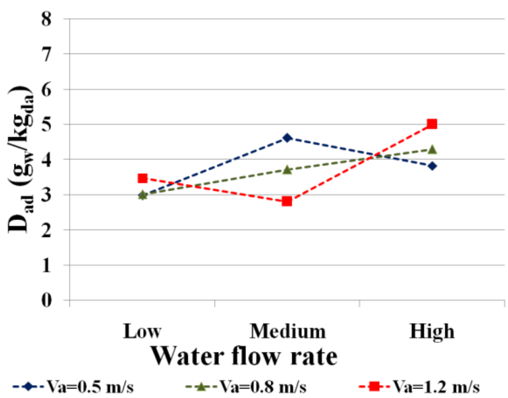
Figure 4.4 Sample of air inlet temperature and air humidity ratio of the experiment

4.1.3 Effect of water flow rate

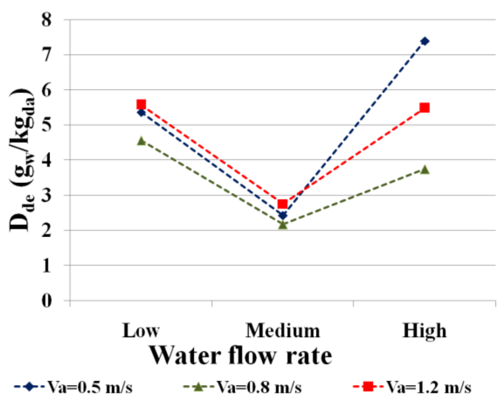
The experiments of a desiccant air dehumidifier at different water flow rates under various air velocites. The value of water flow rate of hot and cold water as shown in Table 4.3.

Table 4.3 The values of the water flow rate

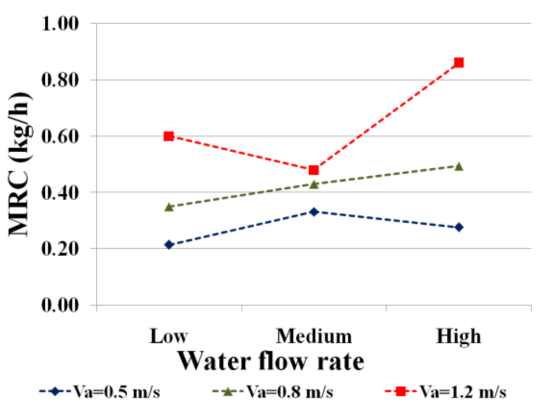
Water flow	Hot water flow rate (l/min)	cold water flow rate (l/min)
Low	10	5
Medium	13	7
High	16	10



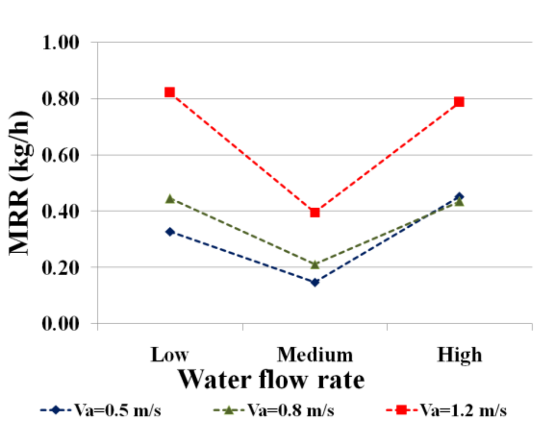
(a) D_{ad} of dehumidification section



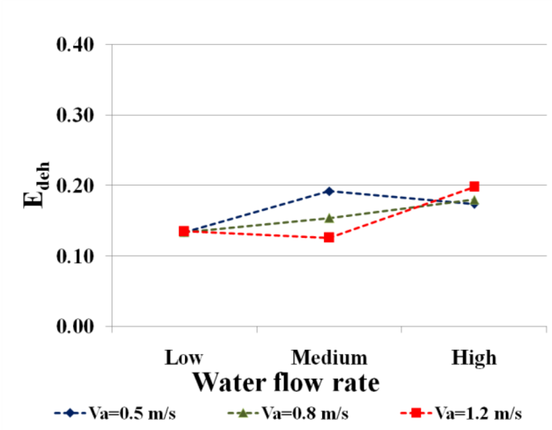
(b) D_{de} of regeneration section



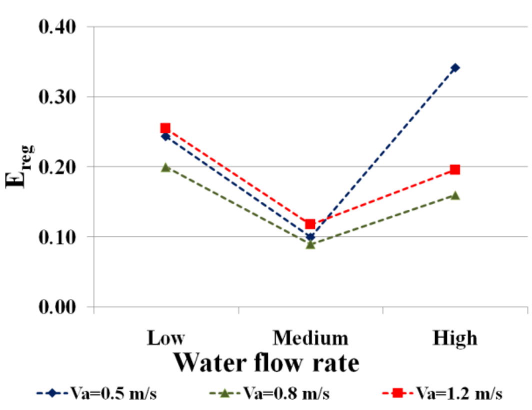
(c) MRC of dehumidification section



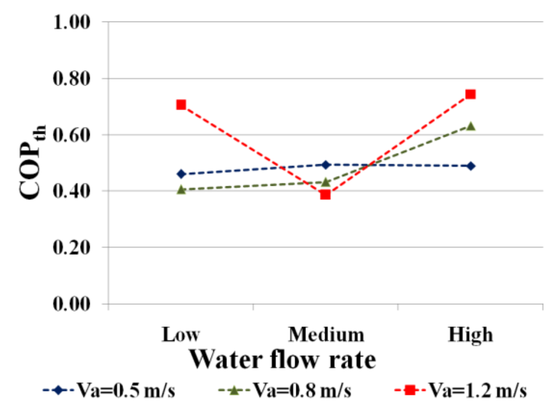
(d) MRR of regeneration section



(e) E_{deh} of dehumidification section



(f) E_{reg} of dehumidification section



(g) COP_{th} of system

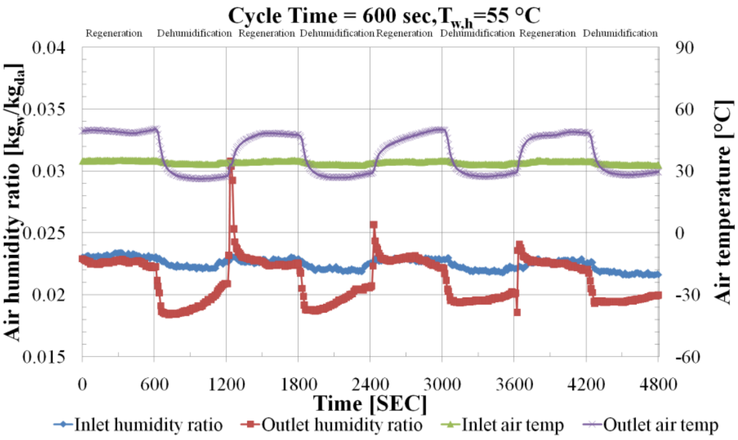
Figure 4.5 Effect of water flow rate

Figure 4.5(a) illustrates the range of humidity ratio of low, medium, and high water flow rates are 2.99 g/kg-3.48 g/kg, 2.81 g/kg-4.62 g/kg, and 3.83 g/kg-5.01 g/kg, respectively. Figure 4.5(b) shows the moisture desorption of the system released from

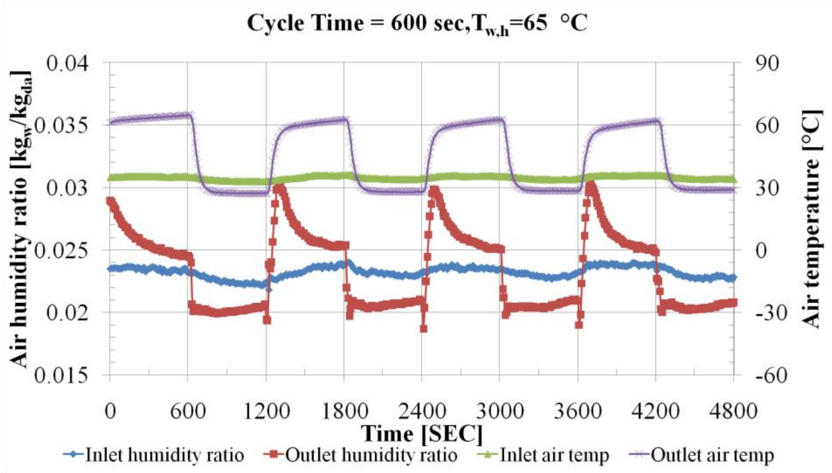
2.17g/kg-7.39 g/kg at different water flow rates. The highest moisture desorption in case of high water flow rate and air velocity is 0.5 m/s. Observing from the plot in Figure 4.5(c) ,the moisture removal capacity (MRC) of the system increase when increasing water flow rate. Figure 4.5(d) illustrates the moisture regeneration removal (MRR) of the system are fluctuate and highest MRR located on low water flow rate and air velocity is 1.2 m/s. The influence reason is from D_{de}. High moisture desorption (D_{de}) and high velocity of air are directly effect to moisture regeneration removal (MRR). The effectiveness of dehumidification and regeneration of the system increase when increasing flow rate of water as shown in Figures 4.5(e) and 4.5(f), respectively. However, the thermal coefficient of performances of the system (COP_{th}) fluctuate and the highest COP_{th} of the desiccant dehumidification system occurs at high water flow rate at air velocity 1.2 m/s.

4.1.4 Effect of hot water temperature

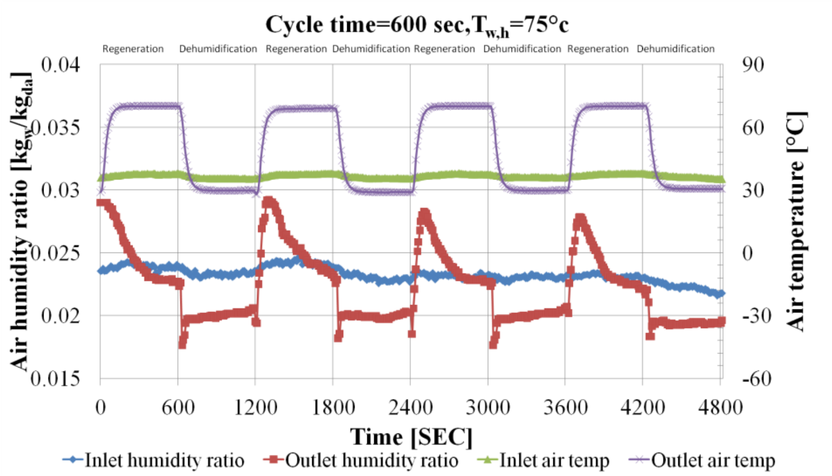
This part of the experiment to evaluate the effect of hot water supplied to the system for regeneration of the desiccant coated on the heat-exchanger. The values of the hot temperature are 55 ºC, 65 ºC, 75 ºC and 80 ºC.



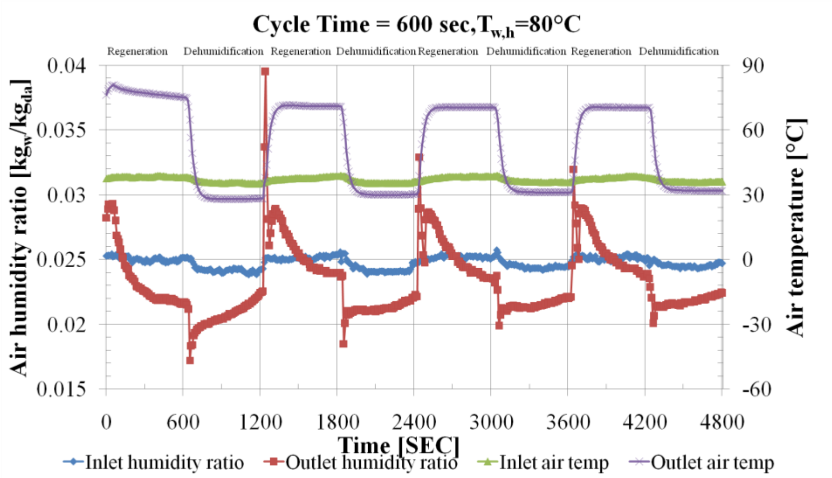
(a) Hot water temperature 55 °C



(b) Hot water temperature 65 °C



(c) Hot water temperature 75 °C



(d) Hot water temperature 80 °C

Figure 4.6 Result from an experimental set up of a desiccant air dehumidification system on different hot water temperatures for the regeneration process

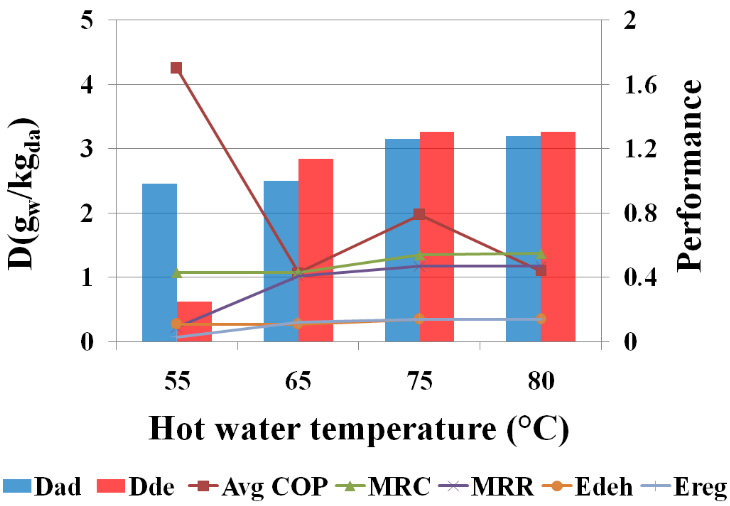


Figure 4.7 Effect of hot water temperature ($T_{w,h}$)

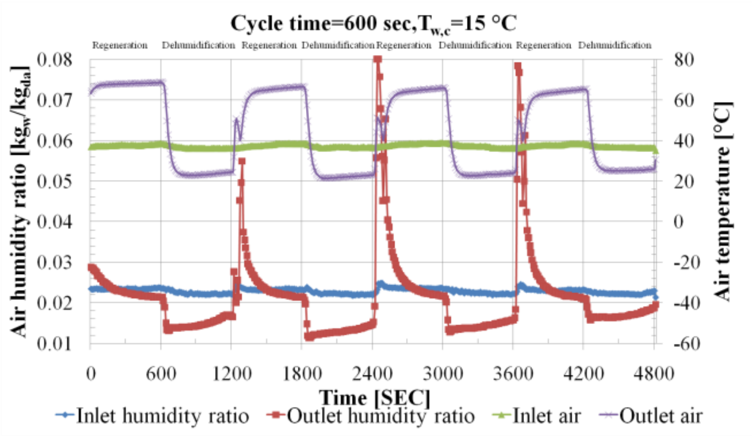
Table 4.4 Performance of a desiccant air dehumidifier under different hot water temperatures

Temp.of hot water (°C)	Average moisture adsorption (g_w/kg_{da})	Average moisture desorption (g_w/kg_{da})	Avg COP_{th}	MRC (kg/h)	MRR (kg/h)	E_{deh}	E_{reg}
55	2.46	0.63	1.7	0.43	0.09	0.11	0.03
65	2.5	2.85	0.43	0.43	0.41	0.11	0.12
75	3.15	3.26	0.79	0.54	0.47	0.14	0.14
80	3.2	3.26	0.44	0.55	0.47	0.14	0.14

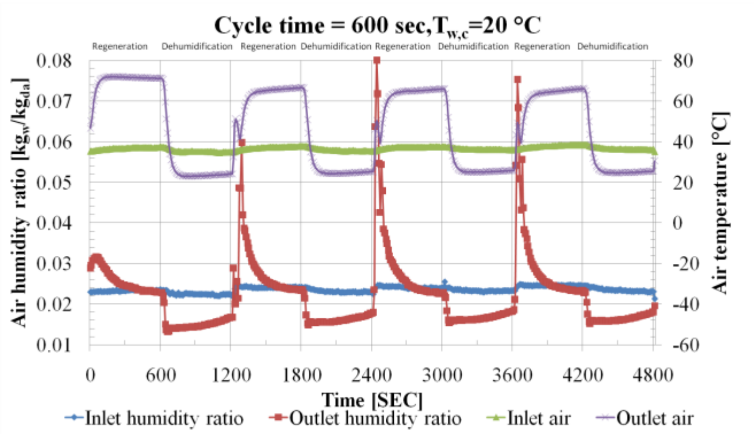
Figure 4.6(a) shows the effects of hot water temperature at 55°C. The moisture desorption by supplied hot water is very low. It should be not effective for regenerate the moisture at this temperature. Figure 4.7 shows the effect of hot water temperature on the system .The water temperature range ($T_{w,h}$) 55, 65, 75 and 80°C respectively. The hot temperature is crucial operating parameter which effects to system performance significantly. At regeneration water temperature of 55°C the average moisture desorption is lowest and COP_{th} of this condition is highest. When water regeneration temperature increase from 55-80°C average moisture adsorption and desorption are increase from 2.46-3.2 (g_w/kg_{da}) and 0.63 to 3.26 (g_w/kg_{da}) respectively. On the other hand, COP_{th} of the systems fluctuate and the suitable condition is hot water temperature 75°C because high COP_{th} and more moisture adsorbed and desorbed from desiccant.

4.1.5 Effect of cold water temperature

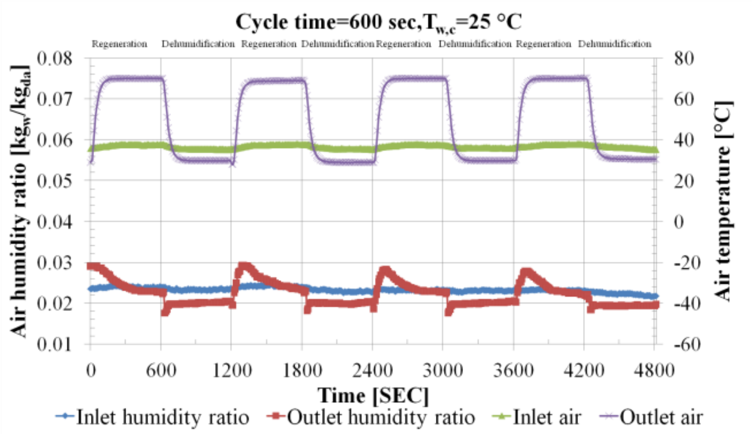
This part of the experiment is to evaluate the effects of cold water supplied to the system for dehumidification mode. The values of cold water temperature are 15ºC, 20 ºC,25 ºC, and 30 ºC, respectively . The results are show in the graphs below.



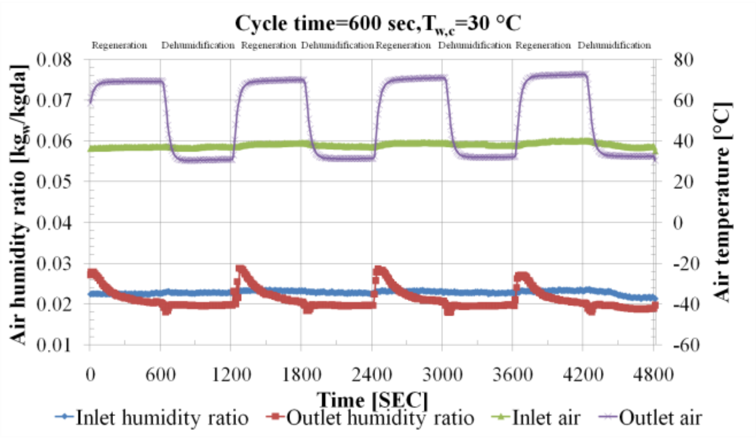
(a) cold water temperature 15 °C



(b) cold water temperature 20 °C



(c) cold water temperature 25 °C



(d) cold water temperature 30 °C

Figure 4.8 Results from an experimentals set up of desiccant air dehumidification system at different cold water temperature for dehumidification process

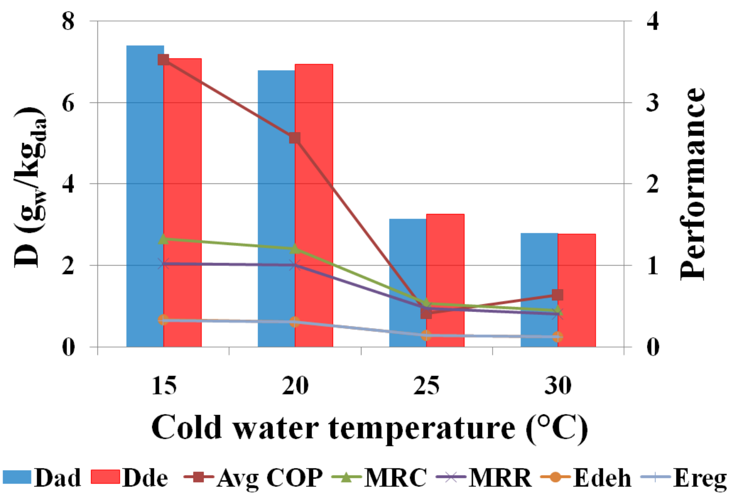


Figure 4.9 Effect of cold water temperature ($T_{w,c}$)

Table 4.5 Performance of a desiccant air dehumidifier under different cold water temperatures

Temp.of cold water [°C]	Average moisture adsorption [g_w/kg_{da}]	Average moisture desorption [g_w/kg_{da}]	Avg COP_{th}	MRC (kg/h)	MRR (kg/h)	E_{deh}	E_{reg}
15	7.4	7.08	3.53	1.33	1.02	0.33	0.32
20	6.8	6.94	2.57	1.21	1.00	0.3	0.3
25	3.15	3.26	0.41	0.54	0.47	0.14	0.14
30	2.8	2.78	0.64	0.45	0.4	0.12	0.12

In this experiment, the cold temperature variation ranges are15, 20, 25 and 30 °C. Figure 4.9 shows the influence of cold water temperature $T_{w,c}$ on system, D_{ads} , D_{des} , COP_{th}, MRC, MRR, E_{deh}, and E_{reg} . The plot illustrates D_{ads} and COP_{th} of system decrease with increasing cold water temperature. When $T_{w,c}$ increase from 15 to 30 °C, D_{ads} of system decrease from 7.4 g/kg to 2.8 g/kg, respectively. At the same time, COP_{th} of system decrease from 3.53 to 0.41. As can be seen from Figure 4.9(a) large different moisture adsorption between cold water temperature 20°C and 25°C. So, better dehumidification temperature should be lower than 25°C is effective to this system.

CHAPTER 5
CONCLUSION

5.1 Conclusions

In this study, a desiccant-coated on a heat exchanger with using silica-gel driven by a heater and chiller are investigated. A system was conducted in tropical climate (Thailand). The dehumidifier supply by realistic ambient air. High temperature and humidity effect to the performance of the system. It cannot control the ambient air condition. The main finding can be summarized as;

- Different cycle times significantly influences the performance of the desiccant dehumidification system. The results show that both average moisture removal and COPth increase as cycle time increases .And suitable cycle time of this experiment is 600 sec, It can consume low heat capacity.
- Cold water temperature influences the performance of the system. Dad and COPth increase with lower cold water temperature. This experiment cold water temperature 20°C can consume more performance of dehumidification process.
- Hot water temperature can enhance Dad and Dde, but COPth decreases. The suitable temperature for hot water is 75°C or higher than this temperature for regeneration process.
- The air velocity influences the adsorption and desorption processes. The effective air velocity is 0.5 m/s. Because mass transfer can be more efficient, it can enhance Dad and Dde of dehumidification and regeneration process of system.

5.2 Suggestion

The existing desiccant dehumidifier system currently uses electricity to produce heat for desiccant regeneration and dehumidification. For further study, a solar water heating system and cooling tower should be integrated into the system for providing hot water and cold water, respectively.

REFERENCES

1 Liu, X.H., Jiang, Y., Chang, X.M., Yi, X.Q., (2007). Experimental investigation of the heat and mass transfer between air and liquid desiccant in a cross-flow regenerator. Renew. Energy , 32, 1623-1636.

2 K.N.Abdalla, A.M.Ahmed, (2011). Experimental investigation of moisture removal rate and dehumidification effectiveness of an internally cooled liquid desiccant air dehumidifier.UofKEJ Vol.1 , Issue1, 25-31.

3 T.S. Ge, Y.J. Dai, R.Z. Wang, et al., (2010). Experimental comparison and analysis on silica gel and polymer coated fin-tube heat exchangers, Energy, 35, 2893-2900.

4 T.S. Ge, Y.J. Dai, R.Z., (2011), Wang, Performance study of silica gel coated fin-tube heat exchanger cooling system based on a developed mathematical model, Energy Convers. Manage, 52, 2329-2338.

5 Ge, T.S., Dai, Y.J., Li, Y., Wang, R.Z., (2012). Simulation investigation on solar powered desiccant coated heat exchanger cooling system. Appl. Energy, 93, 532-540.

6 Zhao, Y., Ge, T.S., Dai, Y.J., Wang, R.Z., (2014). Experimental investigation on a desiccant dehumidification unit using fin-tube heat exchanger with silica gel coating. Appl. Therm. Eng, 63(1), 52-58.

7 M.M. Bassuoni, (2011), An experimental study of structured packing dehumidifier/ regenerator operating with liquid desiccant, Energy, 36, 2628-2638.

8 Y. Jiang, T.S. Ge, R.Z. Wang, L.M. Hu., (2015). Experimental investigation and analysis of composite silica-gel coated fin-tube heat exchangers. Int. J. Refrigeration, 51, 169-179.

9 Banks, N.J., (1992), Desiccant Dehumidifiers in Ice Arenas, Desiccant Cooling and Dehumidification, ASHREA. Atlanta, GA, USA, pp. 69–71.

10 Vineyard, E.A., Sand, J.R., Durfee, D.J., (2000), Parametric analysis of variables that affect the performance of a desiccant dehumidification system. ASHRAE Tran., 106, 87–94.

11 Dhar, P.L., Singh, S.K., (2001). Studies on solid desiccant based hybrid air-conditioning systems. Appl. Therm. Eng, 21, 119-134.

12 Jia, C.X., Dai, Y.J., Wu, J.Y., Wang, R.Z., (2006). Experimental comparison of two honeycombed desiccant wheels fabricated with silica gel and composite desiccant material. Energy Convers. Manag, 47, 2523-2534.

13 Jia, C.X., Dai, Y.J., Wu, J.Y., Wang, R.Z., (2007), Use of compound desiccant to develop high performance desiccant cooling system. Int. J. Refrigeration , 30, 345-353.

14 Lazzarin, R.M., Castellotti, F., (2007). A new heat pump desiccant dehumidifier for supermarket application. Energy Build, 39, 59-65.

15 Weixing, Y., Yi, Z., Xiaoru, L., Xiugan, Y., (2008), Study of a new modified cross-cooled compact solid desiccant dehumidifier. Appl. Therm. Eng. , 28, 2257-2266.

16 Yin, Y., Zhang, X., Chen, Z., (2007), Experimental study on dehumidifier and regenerator of liquid desiccant cooling air conditioning system. Build. Environ, 42, 2505-2511.

17 Y.J. Dai, R.Z. Wang, H.F. Zhang, (2001), Parameter analysis to improve rotary desiccant dehumidification using a mathematical model, Int. J. Therm. Sci. , 40, 400-408.

18 D. La, Y. Dai, Y. Li, T. Ge, et al., (2010), Study on a novel thermally driven air conditioning system with desiccant dehumidification and regenerative evaporative cooling, Build. Environ, 45, 2473-2484.

19 P. Majumdar, M.K. Sarwar, (1994), Performance of a desiccant dehumidifier bed with mixed inert and desiccant materials, Energy , 19,103-116.

20 W.X. Yuan, Y. Zheng, X.R. Liu, et al., (2008), Study of a new modified cross-cooled compact solid desiccant dehumidifier, Appl. Therm. Eng, 28, 2257-2266.

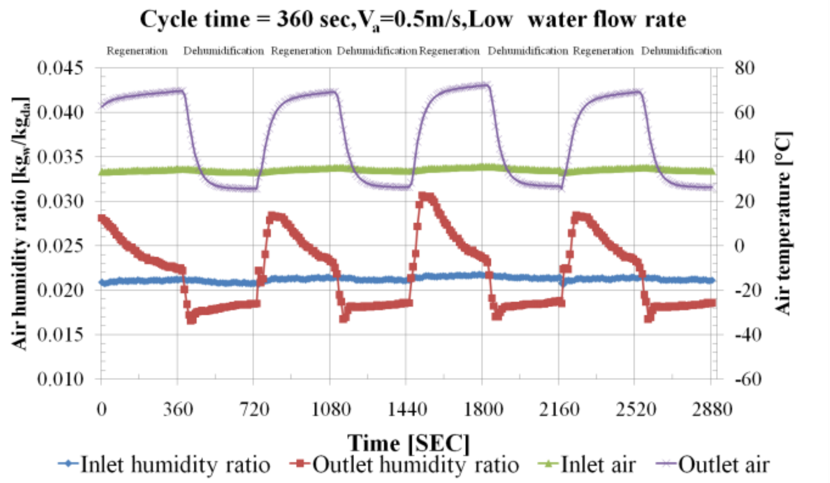
21 Daou, K., R.Z. Wang, and Z.Z. Xia, (2006), Desiccant cooling air conditioning: a review. Renewable and Sustainable Energy Reviews, 10(2), p. 55-77.

23 Google Inc., Desiccant Chart Comparisons-Sorbent Systems [Online], Available : http://www.sorbentsystems.com/desiccants_charts.html [2015, March 21]

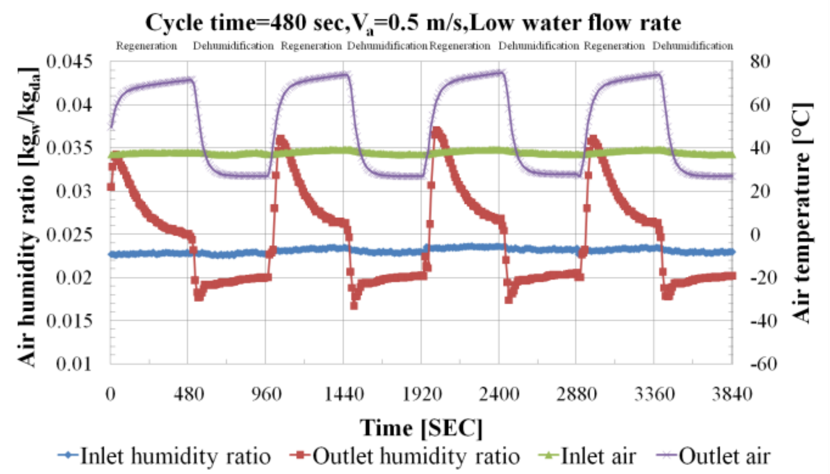
APPENDIX A

Graphs of the results of the experiment

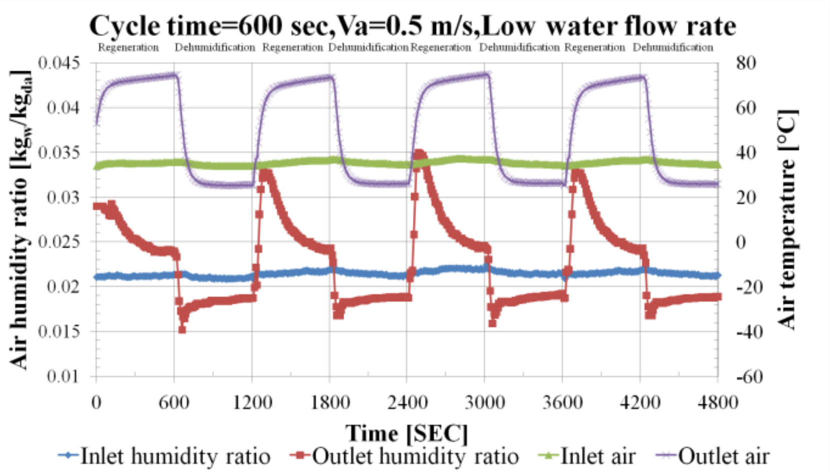
a) Low water flow rate and velocity of air 0.5 m/s



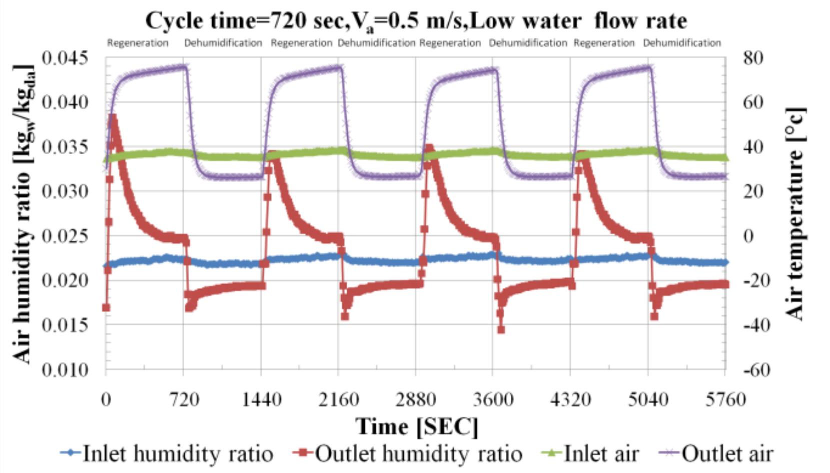
(a) Cycle time 360 seconds



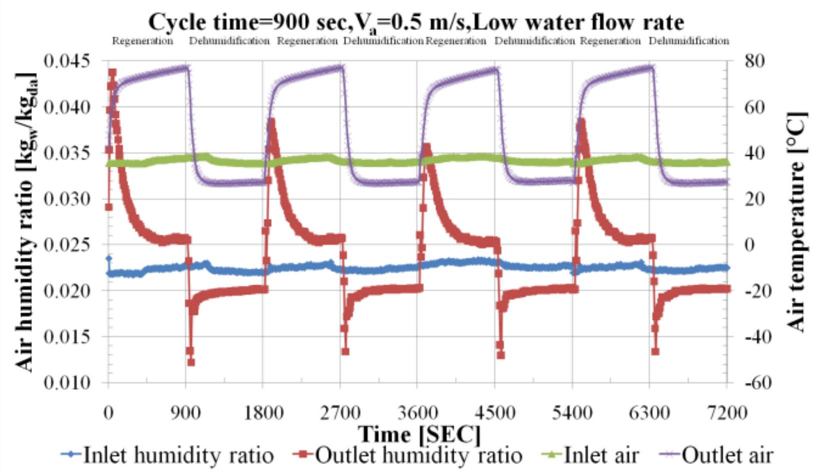
(b) Cycle time 480 seconds



(c) Cycle time 600 seconds



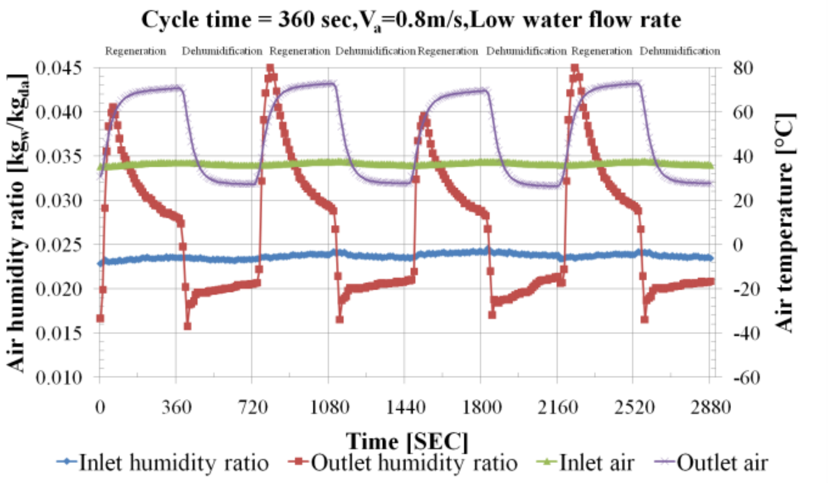
(d) Cycle time 720 seconds



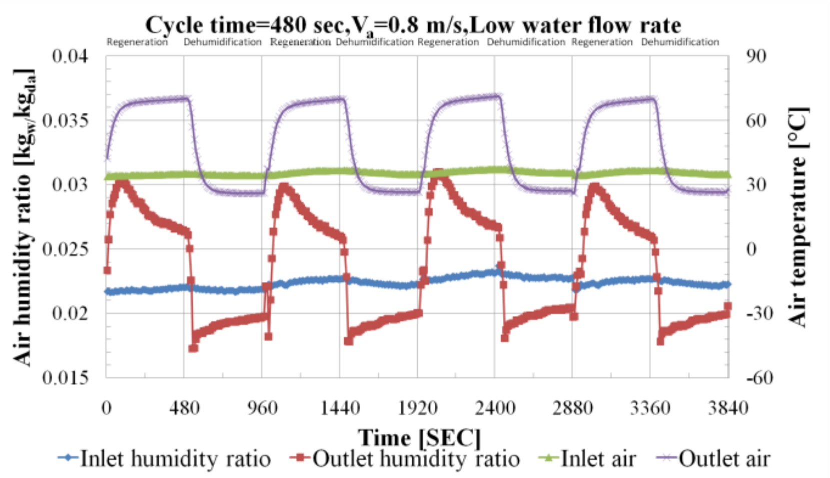
(e) Cycle time 900 seconds

Figure A.1 The result of low water flow rate and velocity of air 0.5 m/s

b) Low water flow rate and velocity of air 0.8 m/s



(a) Cycle time 360 seconds



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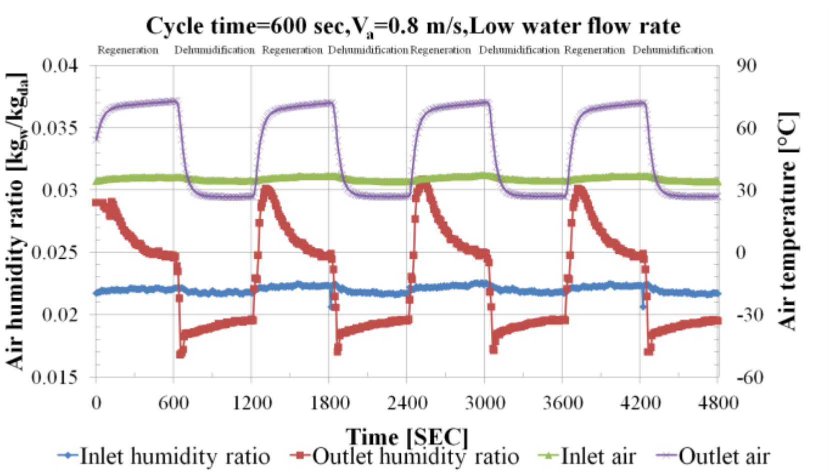
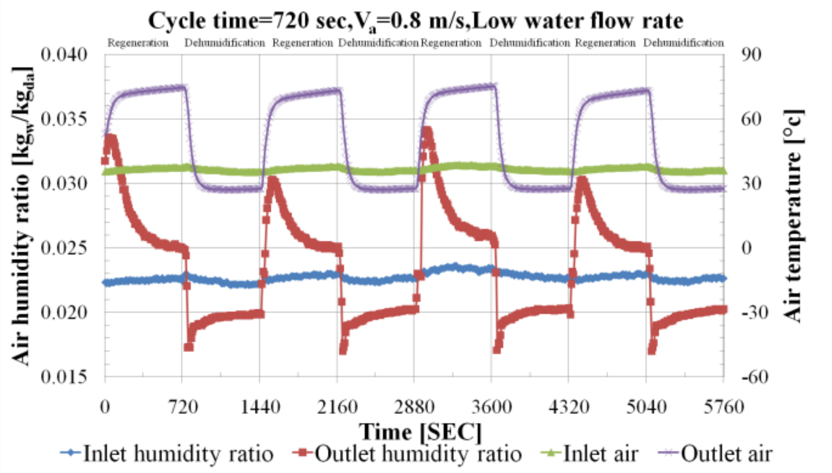
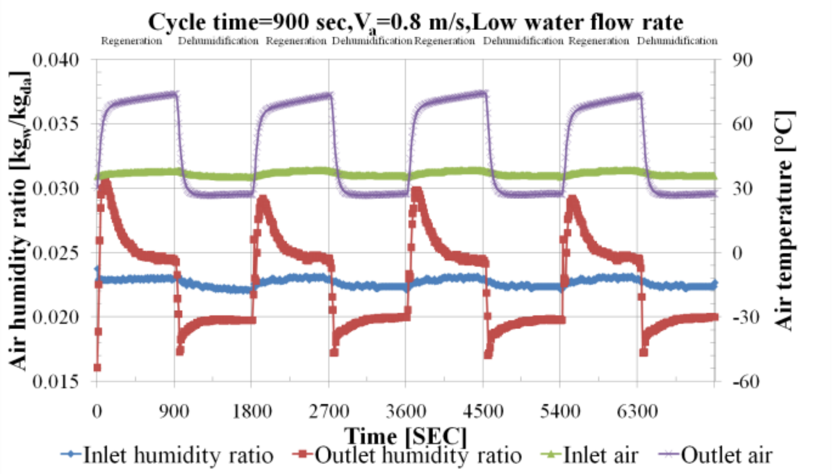


(c) Cycle time 600 seconds



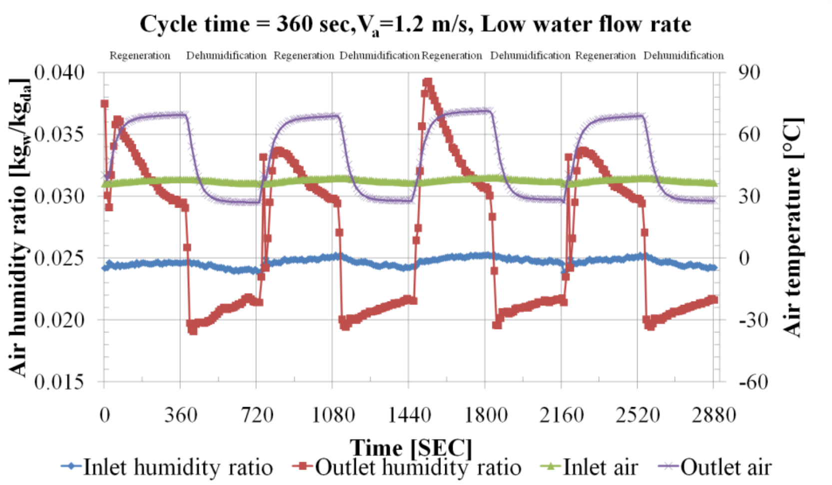
(d) Cycle time 720 seconds



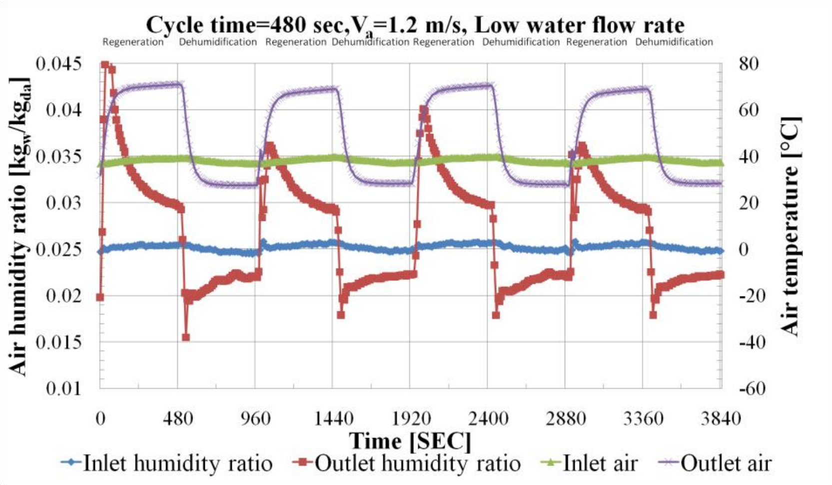
(e) Cycle time 900 seconds

Figure A.2 The result of low water flow rate and velocity of air 0.8 m/s

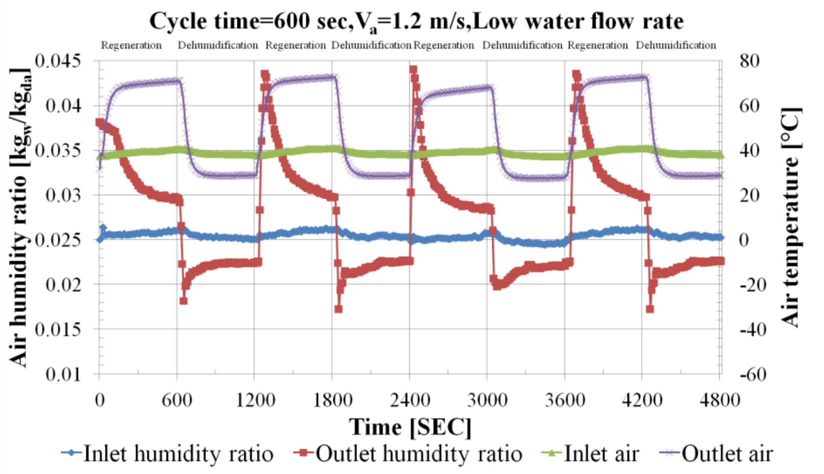
c) Low water flow rate and velocity of air 1.2 m/s



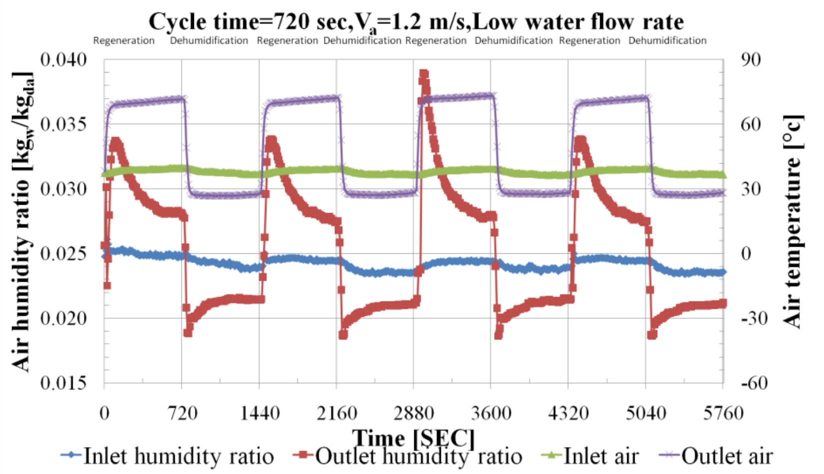
(a) Cycle time 360 seconds



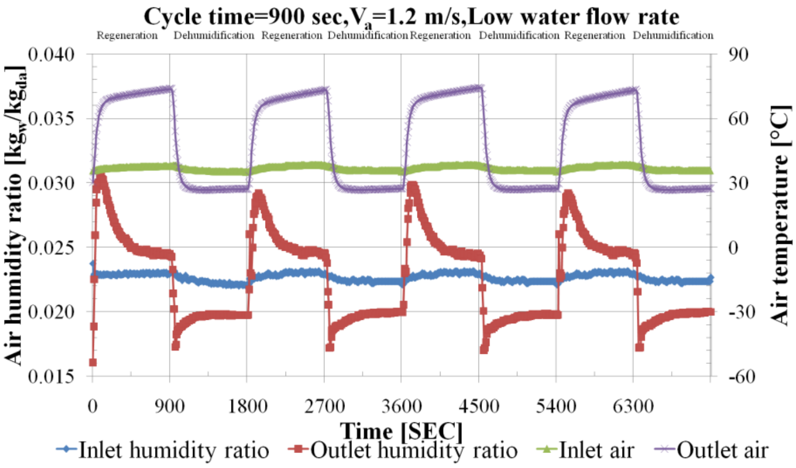
(b) Cycle time 480Seconds



(c) Cycle time 600 seconds



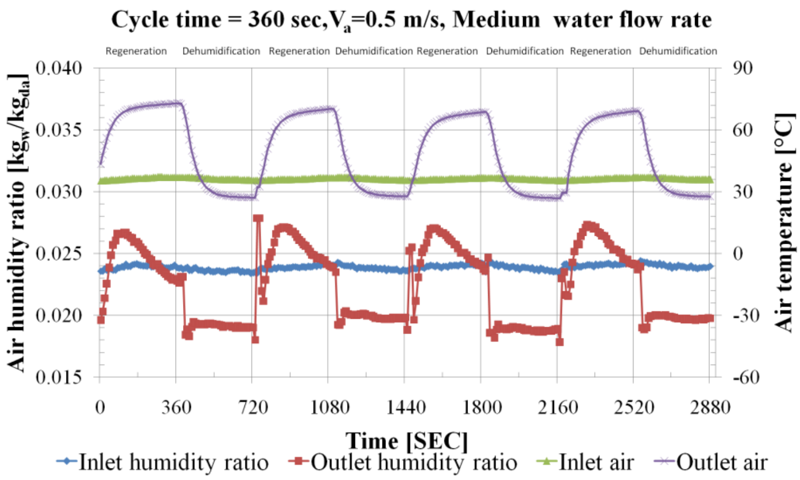
(d) Cycle time 720 seconds



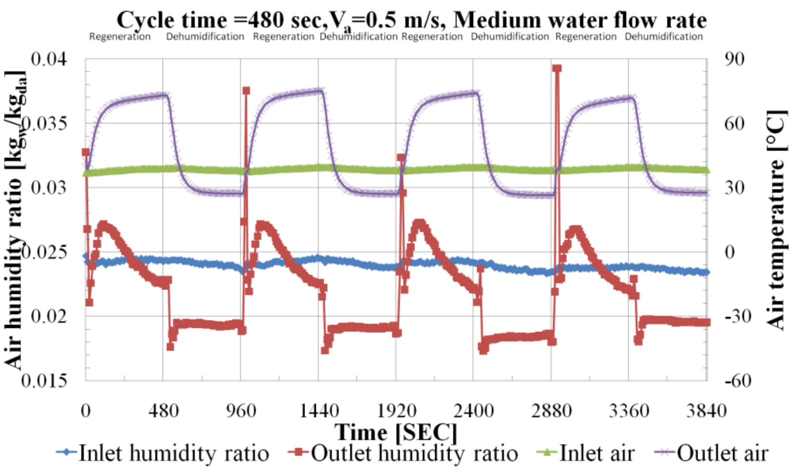
(e) Cycle time 900 seconds

Figure A.3 The result of low water flow rate and velocity of air 1.2 m/s

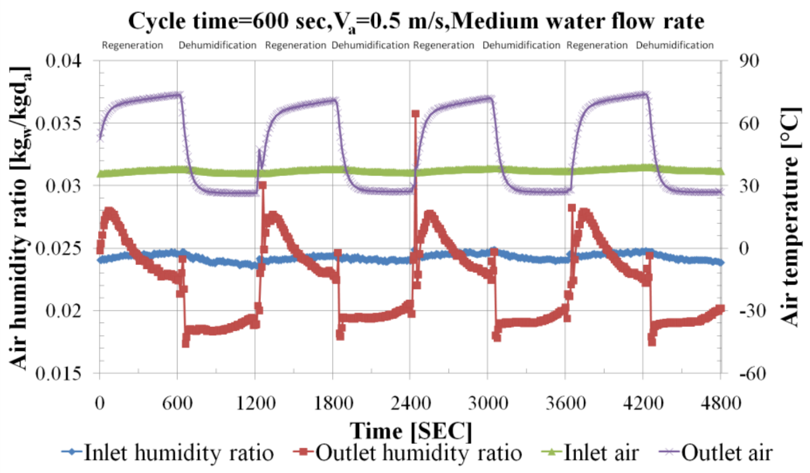
d) Medium water flow rate and velocity of air 0.5 m/s



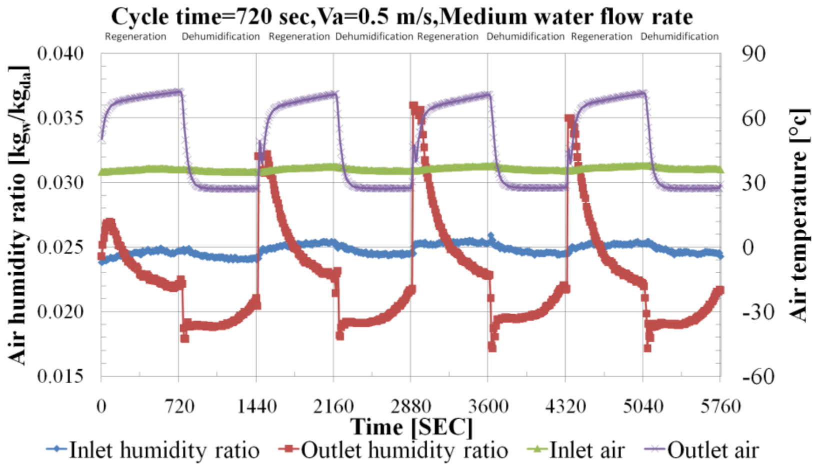
(a) Cycle time 360 seconds



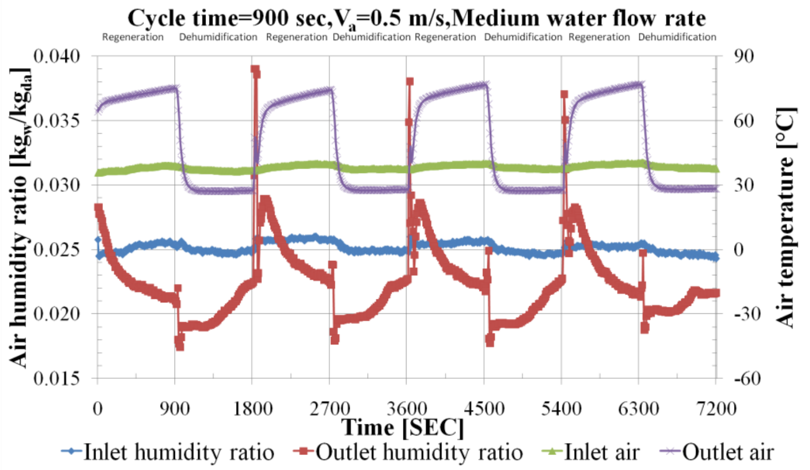
(b) Cycle time 480 seconds



(c) Cycle time 600 seconds



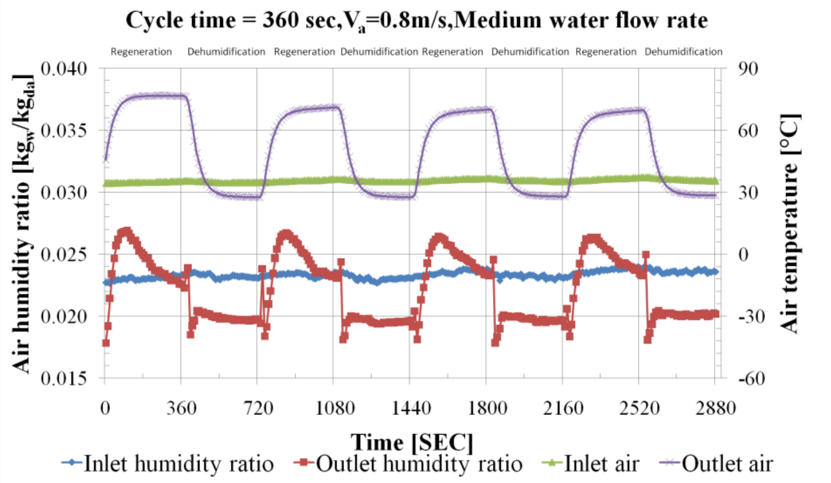
(d) Cycle time 720 seconds



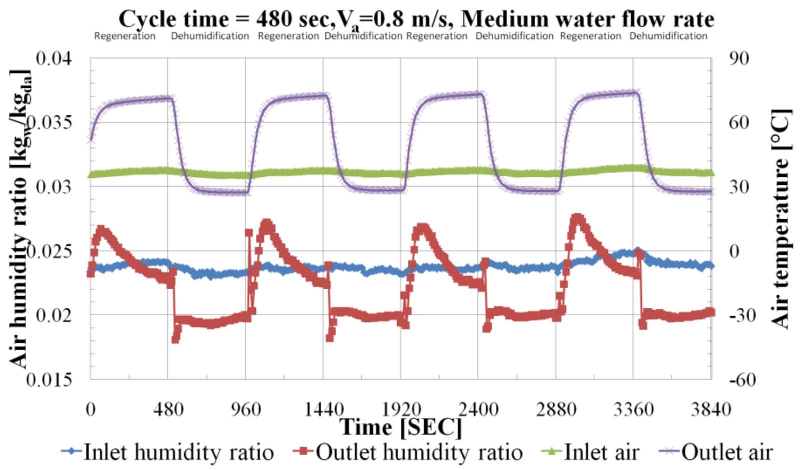
(e) Cycle time 900 seconds

Figure A.4 The result of medium water flow rate and velocity of air 0.5 m/s

e) Medium water flow rate and velocity of air 0.8 m/s



(a) Cycle time 360 seconds



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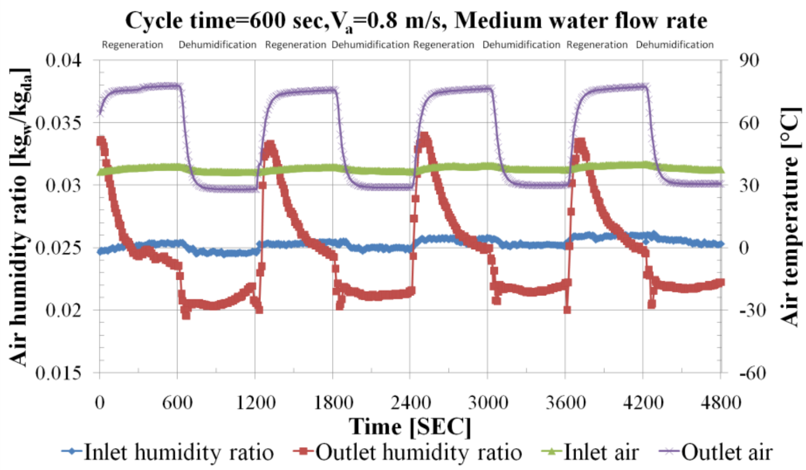

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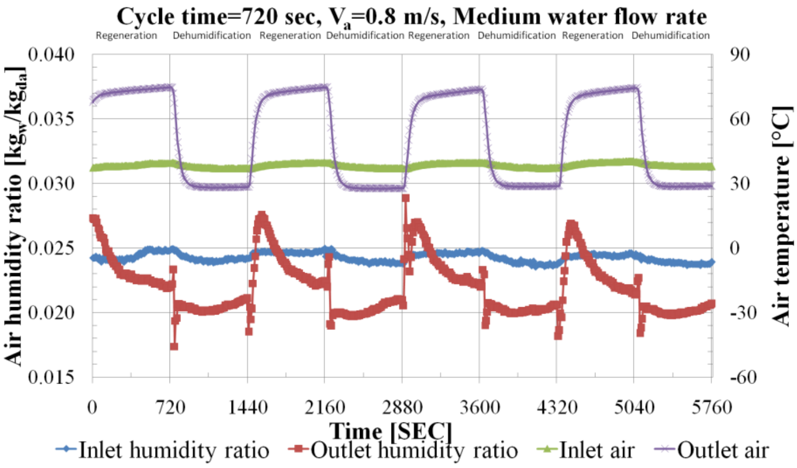

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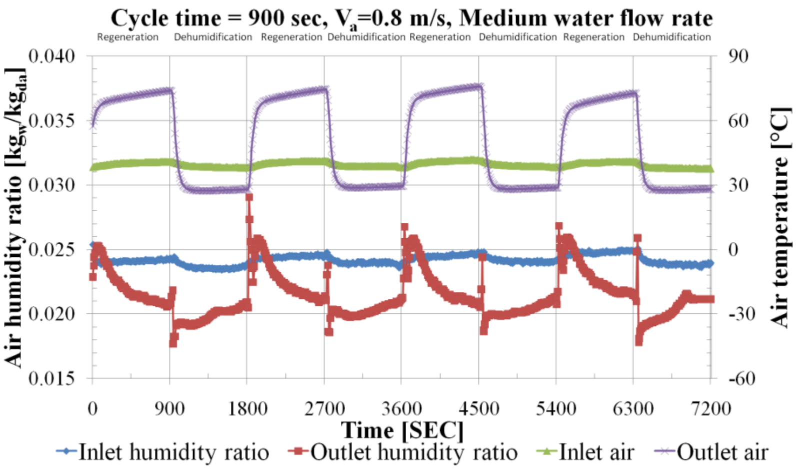
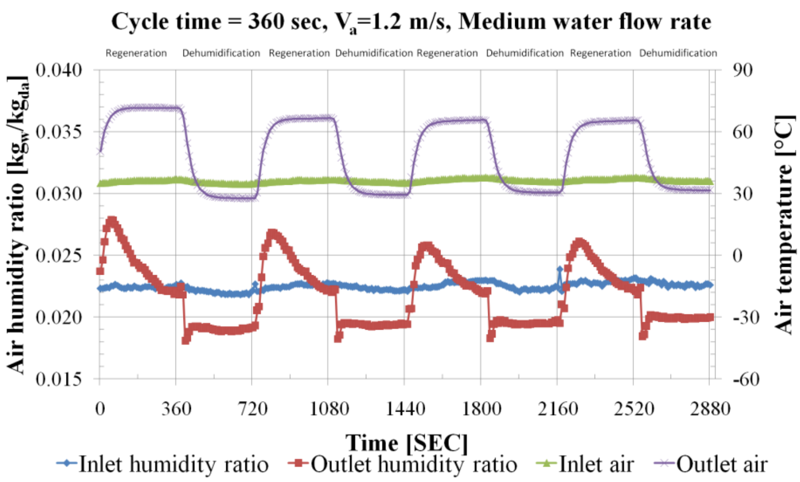


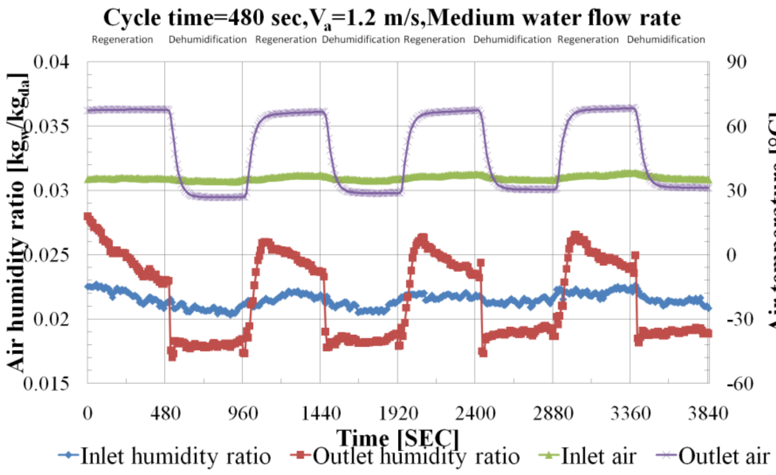
(e) Cycle time 900 seconds

Figure A.5 The result of medium water flow rate and velocity of air 0.8 m/s

f) Medium water flow rate and velocity of air 1.2 m/s



(a) Cycle time 360 seconds



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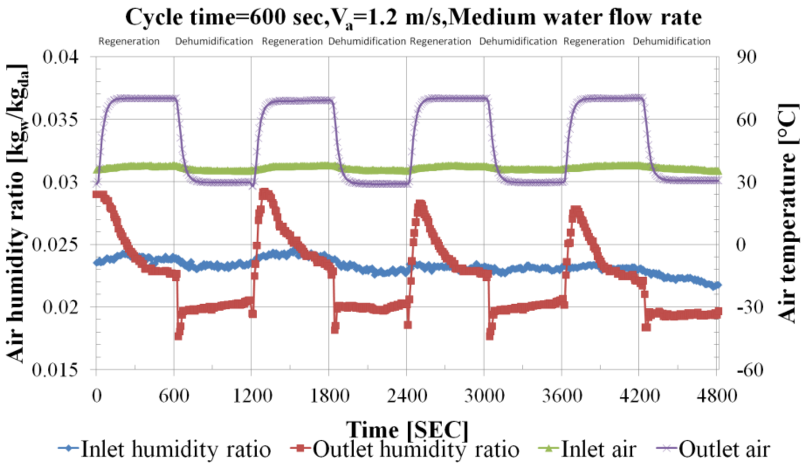
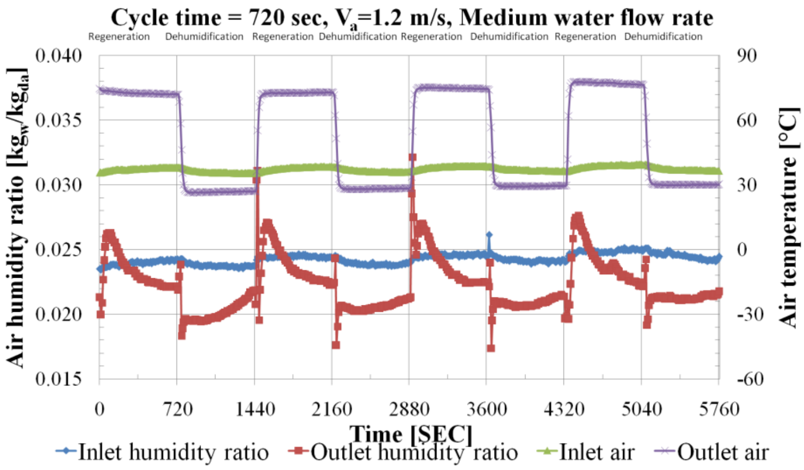
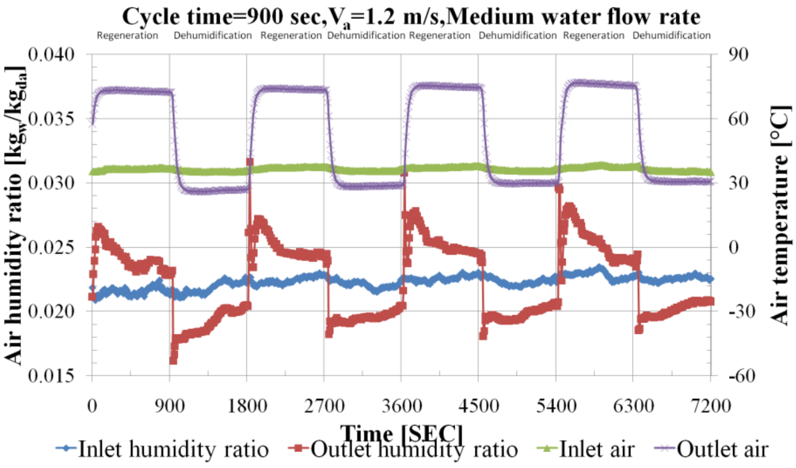


(c) Cycle time 600 seconds



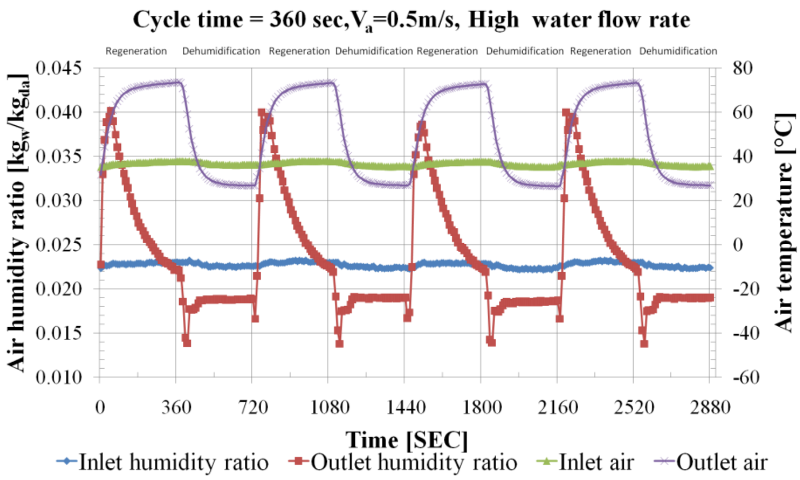
(d) Cycle time 720 seconds



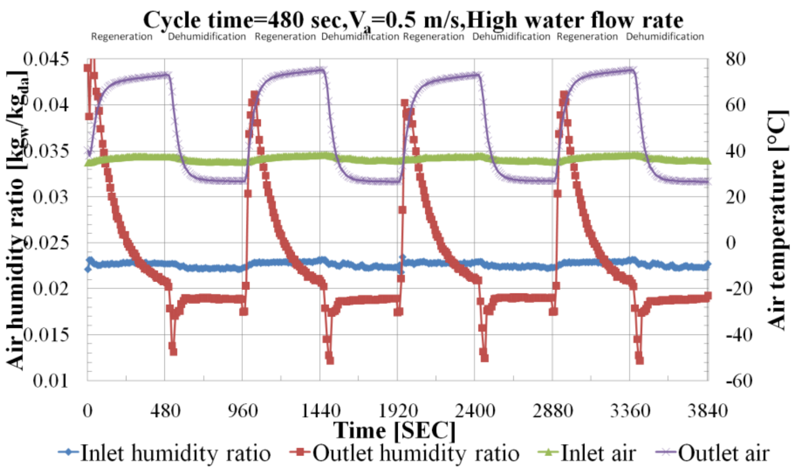
(e) Cycle time 900 seconds

Figure A.6 The result of medium water flow rate and velocity of air at 1.2 m/s

g) High water flow rate and velocity of air 0.5 m/s



(a) Cycle time 360 seconds



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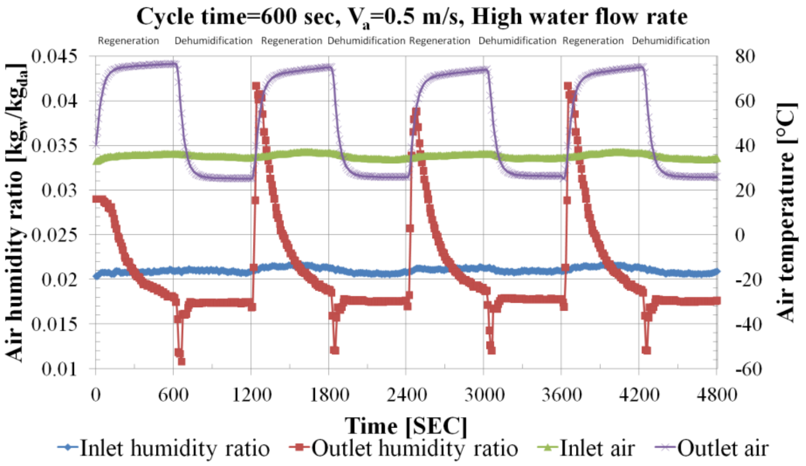
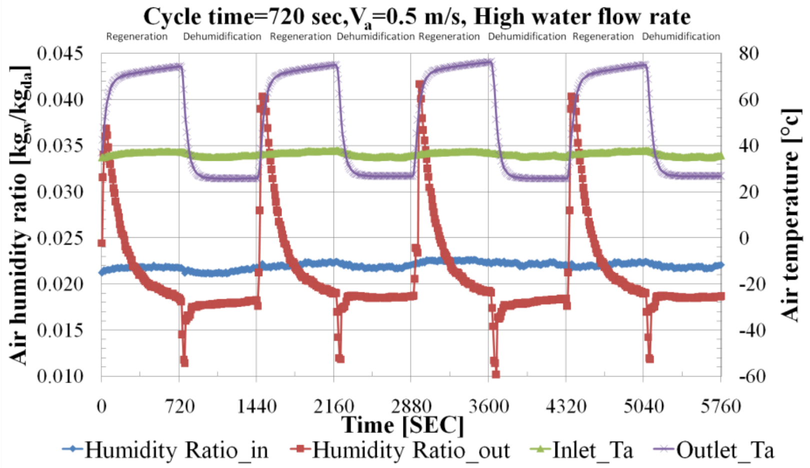


(c) Cycle time 600 seconds



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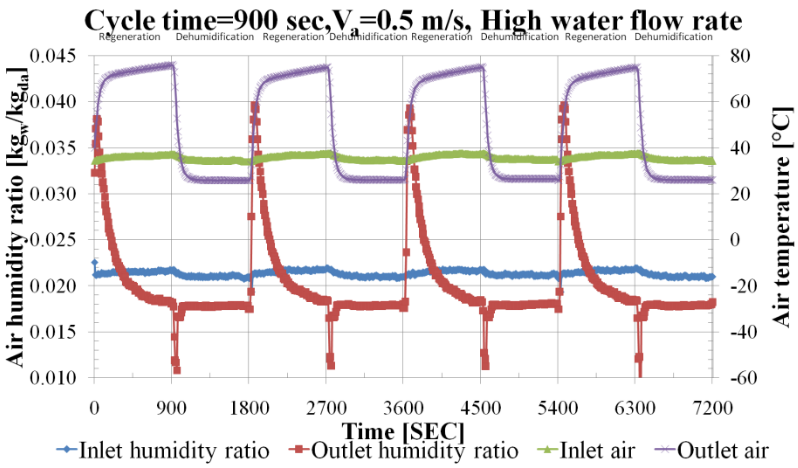
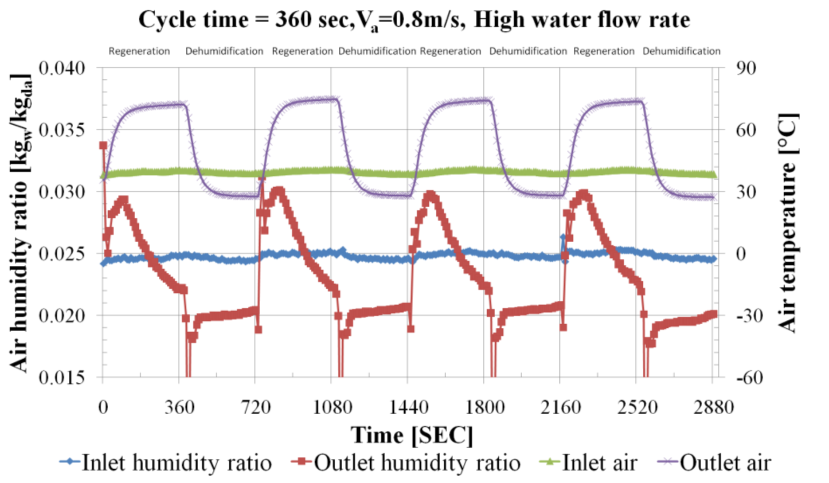


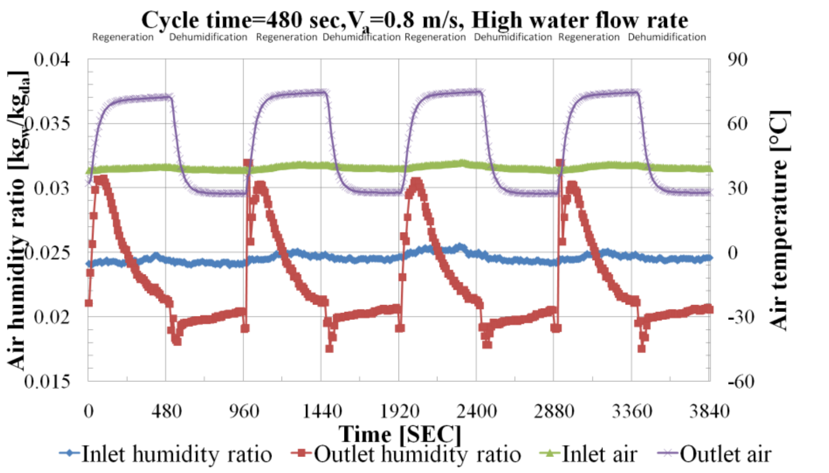
(e) Cycle time 900 seconds

Figure A.7 The result of high water flow rate and velocity of air at 0.5 m/s

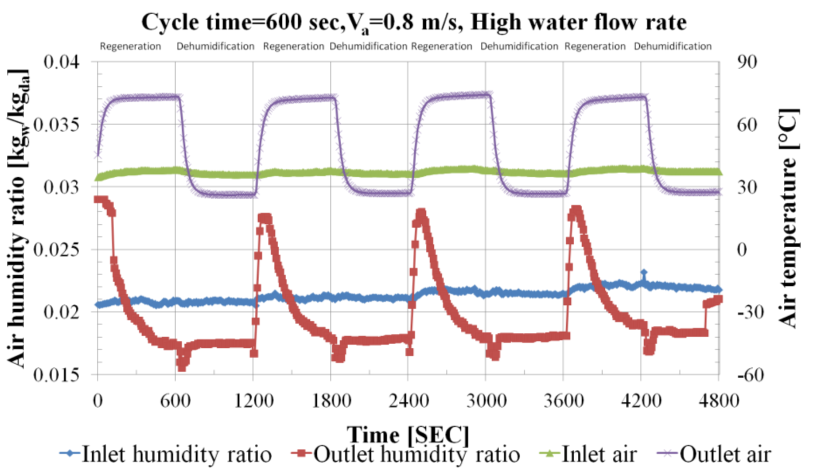
h) High water flow rate and velocity of air at 0.8 m/s



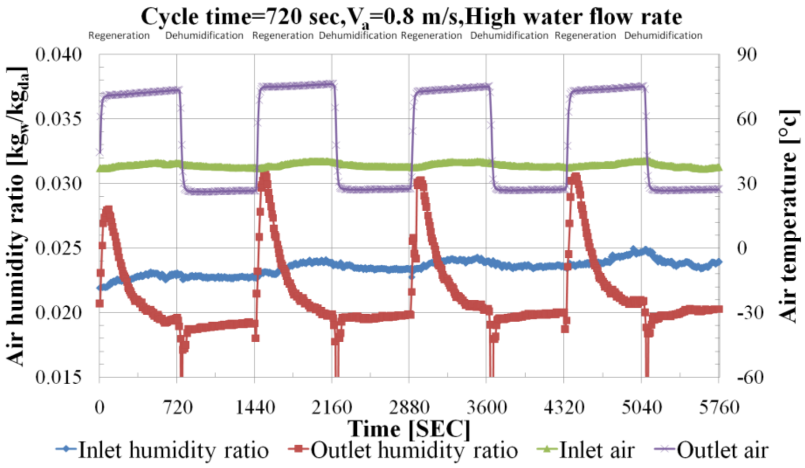
(a) Cycle time 360 seconds



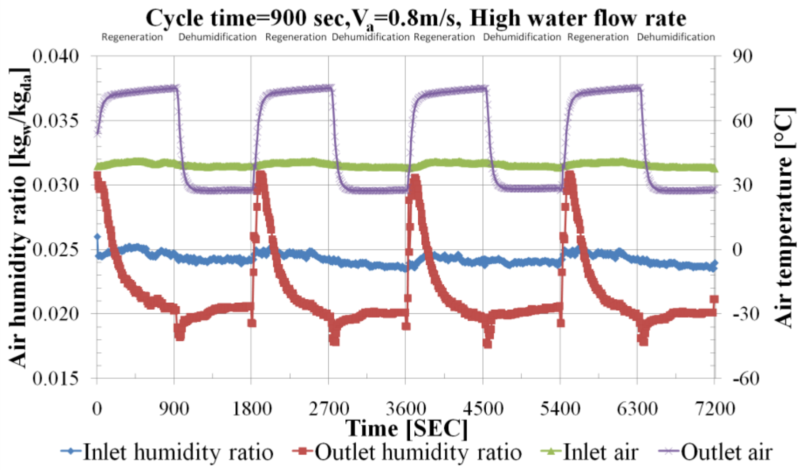
(b) Cycle time 480 seconds



(c) Cycle time 600 seconds



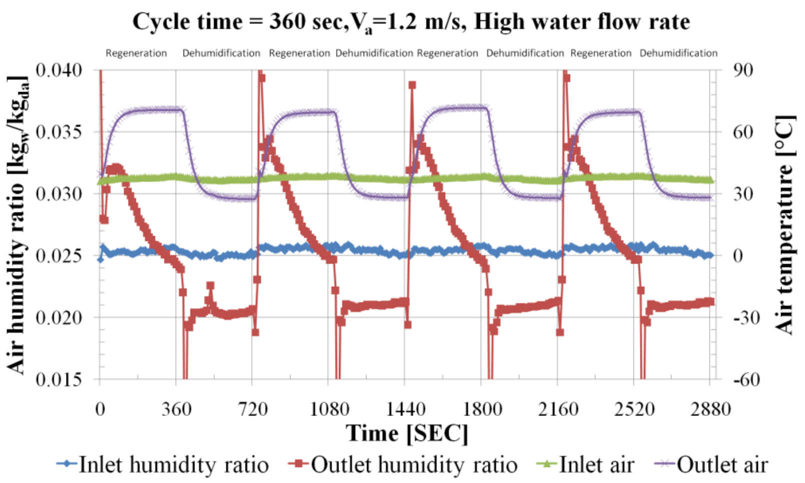
(d) Cycle time 720 seconds



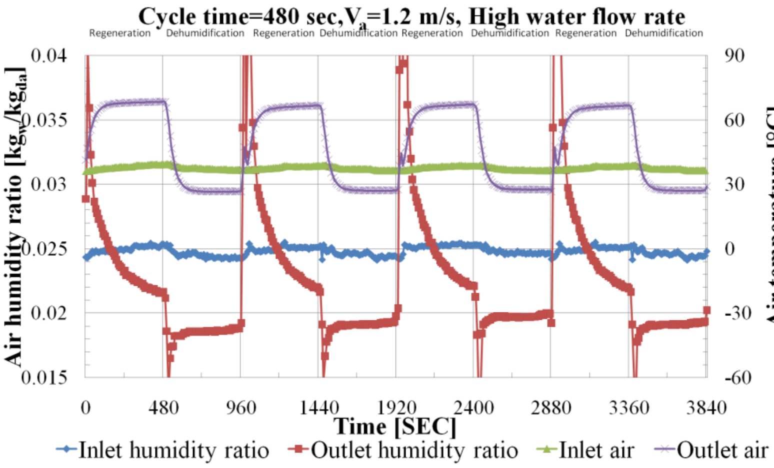
(e) Cycle time 900 seconds

Figure A.8 The result of high water flow rate and velocity of air at 0.8 m/s

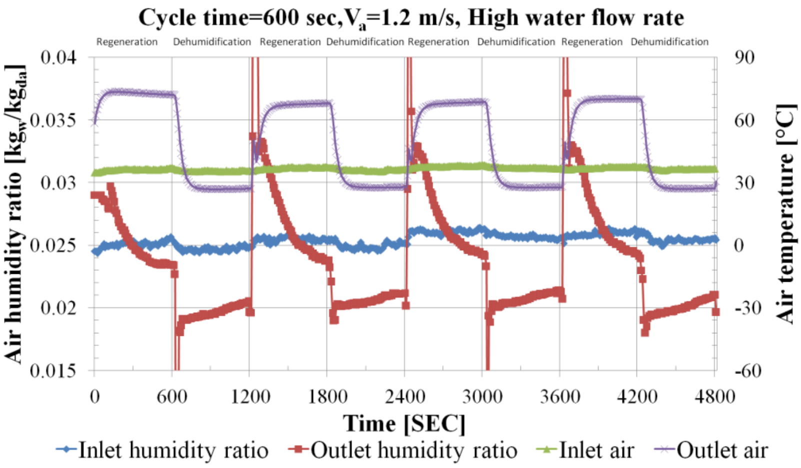
i) High water flow rate and velocity of air at 1.2 m/s



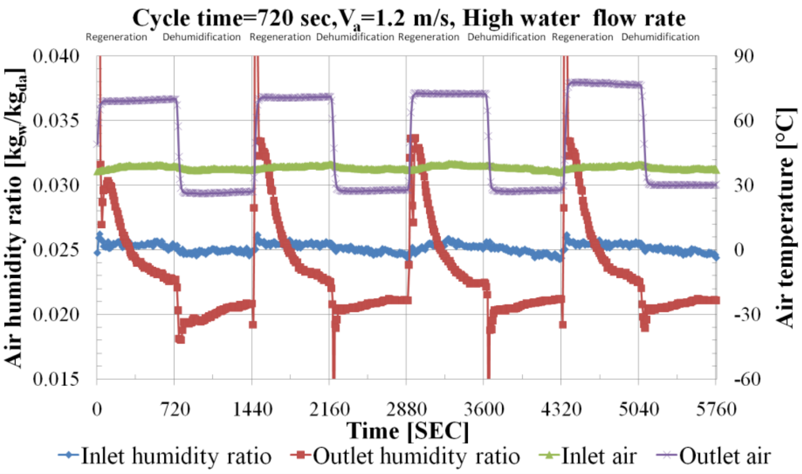
(a) Cycle time 360 seconds



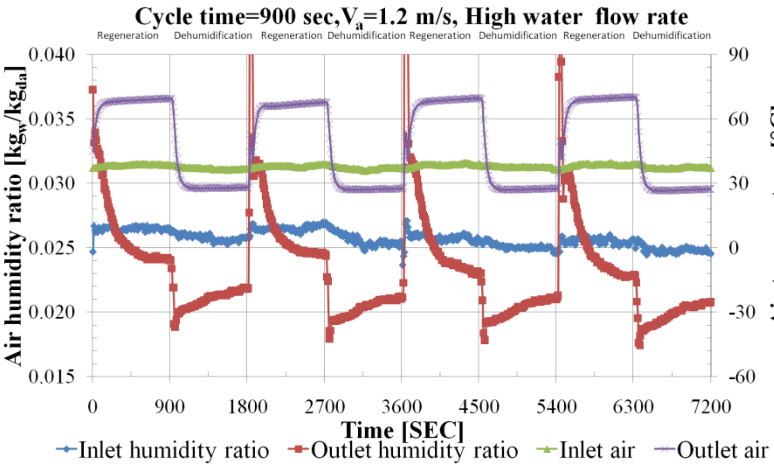
(b) Cycle time 480 seconds



(c) Cycle time 600 seconds



(d) Cycle time 720 seconds



(e) Cycle time 900 seconds

Figure A.9 The result of high water flow rate and velocity of air 1.2 m/s