INVESTIGATION OF A SOLAR-ASSISTED THERMALLY ACTIVATED AIR CONDITIONING SYSTEM

BY

Muhammad Mujahid Rafique

A Thesis Presented to the

DEANSHIP OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

November 2015

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN- 31261, SAUDI ARABIA

DEANSHIP OF GRADUATE STUDIES

This thesis, written by **Muhammad Mujahid Rafique** under the direction of his thesis advisor and approved by his thesis committee, has been presented and accepted by the Dean of Graduate Studies, in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE IN MECHANICAL ENGINEERING.**

Dr. Palanichamy Gandhidasan
(Advisor)

Dr. Zuhair Mattoug Gasem
Department Chairman

Dr. Luai M. Al-Hadhrami
(Member)

Dr. Salam A. Zummo
Dean of Graduate Studies

25/11/15
Date

Dr. Haitham M. S. Bahaidarah
(Member)

© Muhammad Mujahid Rafique

2015

ALL RIGHTS RESERVED

Dedicated to my beloved parents, brothers, sisters, cousins and friends, whose constant prayers, sacrifice and inspiration led to this wonderful accomplishment

ACKNOWLEDGMENT

"In the name of Allah, The Most Gracious and Most Merciful"

All praises to Allah Almighty, for giving me patience and courage to complete my thesis work. My due acknowledgements to King Fahd University of Petroleum & Minerals for providing me this opportunity to accomplish my M.S. With due respect, I would like to thank my advisor Dr. Palanichamy Gandhidasan for his continuous guidance and help through all the difficult stages. I would like to acknowledge the help from my committee members, Dr. Haitham M.S. Bahaidarah and Dr. Luai M. Al-Hadhrami.

I would like to extend my thanks to the Heat Engine laboratory engineer Mr. Karam for his help in experimental work. I thank all my fellow graduate students at KFUPM for making my stay at KFUPM a memorable and enjoyable one. Special thanks to my parents, brothers and sisters for their encouragement, support and constant prayers for my success.

TABLE OF CONTENTS

ACKNOWLEDGMENT ..... **V**

TABLE OF CONTENTS ..... **VI**

LIST OF TABLES ..... **X**

LIST OF FIGURES ..... **XI**

NOMENCLATURE ..... **XV**

ABSTRACT ..... **XVIII**

ملخص الرسالة ..... **XX**

CHAPTER 1 INTRODUCTION **1**

1.1 Energy and environmental issues **4**

1.2 Human comfort and building indoor air quality **6**

1.3 Alternative cooling systems and dehumidification **6**

1.4 Background **7**

CHAPTER 2 DESICCANT BASED EVAPORATIVE COOLING **10**

2.1 Introduction **10**

2.2 Basic principle and types of evaporative cooler **11**

2.3 Desiccant dehumidifier **17**

2.4 Desiccant based evaporative cooling **19**

2.4.1 General idea **19**

2.4.2 System description **22**

2.4.3 Literature survey **23**

2.5 Advantages of desiccant-aided evaporative cooling **31**

2.6 Major applications of desiccant based evaporative cooling 32

2.7 Performance Index 32

2.8 Modified evaporative cooler 34

2.9 Developments in evaporative cooling research 40

2.10 Conclusions 43

CHAPTER 3 LITERATURE REVIEW OF LIQUID DESICCANT MATERIALS AND DEHUMIDIFIER 46

3.1 Introduction 46

3.2 Liquid desiccant cooling 47

3.3 Performance parameters 48

3.4 Advantages of using liquid desiccants 49

3.5 Liquid desiccant materials 52

3.6 Composite desiccant materials 56

3.7 Liquid desiccant dehumidifiers 59

3.8 Adiabatic dehumidifiers 60

3.9 Dehumidifier with inner cooling 67

3.10 Developments of liquid desiccant dehumidifier 76

3.11 Dehumidifier packing material selection 81

3.12 Flow pattern inside the dehumidifier 83

3.13 Conclusions 85

3.14 Objectives of the present work 87

CHAPTER 4 MATHEMATICAL MODELING OF LIQUID DESICCANT WHEEL 88

4.1 Introduction 89

4.2 Mathematical modeling 93

4.2.1 System description 93

4.2.2 Modeling approach 94
4.2.3 Governing equations 95
4.3 Boundary and initial conditions 97
4.4 Performance index 98
4.5 Results and discussions 99
4.5.1 Feasibility analysis 100
4.5.2 Parametric analysis 104
4.6 Conclusions 113

CHAPTER 5 EXPERIMENTAL INVESTIGATION OF LIQUID DESICCANT COOLING CYCLE 116

5.1 Introduction 116
5.2 Test facility 117
5.3 Desiccant preparation 122
5.4 Instrumentations 122
5.5 Test procedure and conditions 123
5.6 Uncertainty analysis 124
5.7 Performance parameters 126
5.8 Results and discussion 128
5.8.1 Effect of ambient air humidity ratio 128
5.8.2 Effect of ambient air temperature 132
5.8.3 Effect of process air mass flux 135
5.8.4 Effect of regeneration air mass flux 138
5.8.5 Effect of regeneration temperature 140
5.8.6 Effect of desiccant concentration 143

5.9 Comparison of heat and mass transfer model of liquid desiccant dehumidifier with experimental data 145

5.10 Conclusions 152

CHAPTER 6 FEASIBILITY ANALYSIS OF SOLID DESICCANT COOLING SYSTEM UNDER CLIMATIC ZONE OF DHAHRAN, SAUDI ARABIA 154

6.1 Introduction 155

6.2 Description of solid desiccant cooling cycles 156

6.3 Simulation Model 158

6.4 Cycle analysis 160

6.5 Performance Index 161

6.6 Results and discussion 161

6.7 Conclusions 171

CHAPTER 7 CONCLUSIONS AND FUTURE WORK 173

7.1 Conclusions 173

7.2 Recommendations for future work 175

REFERENCES 177

VITAE 196

LIST OF TABLES

Table 2.1: Types and characteristics of evaporative cooling systems. 17
Table 2.2: Summary of different studies related to evaporative cooling systems. 41
Table 2.3: Needed R & D for desiccant based evaporative cooling technology [121]..... 43
Table 3.1: Performance parameters used for liquid desiccant cooling systems. 49
Table 3.2: Comparison between liquid-desiccant cooling system and conventional air-conditioning system. 51
Table 3.3: Weighing factors and figures of merit for desiccant selection [154]..... 59
Table 3.4: Details of Adiabatic/internally cooled dehumidifiers. 85
Table 4.1: Overview of thermal cooling techniques [228]. 91
Table 4.2: Liquid vs. solid desiccant air conditioning [230]. 91
Table 4.3: Operating and geometric parameters. 100
Table 5.1: Specifications of sensors..... 123
Table 5.2: Uncertainties of measuring instruments used in experiment. 125
Table 5.3: Operating parameters. 128
Table 6.1: Meteorological Data of Dhahran, Saudi Arabia for the month of April (15th April). 162
Table 6.2: Meteorological Data of Dhahran, Saudi Arabia for the month of July (17th July). 163
Table 6.3: Meteorological Data of Dhahran, Saudi Arabia for the month of October (15th October). 163
Table 6.4: Operating parameters. 164

LIST OF FIGURES

Figure 1.1: Thermal comfort zone according ASHRAE 55 [1]..... 1
Figure 1.2: The functions of air conditioning [2]. 2
Figure 1.3: Interrelation between energy, environment and technologies issues to the nature and humanity survival [7]..... 4
Figure 1.4: Long-lived greenhouse gases concentrations in atmospheric [12]..... 5
Figure 1.5: Methods for the conversion of solar energy into air conditioning or cooling process [21]..... 7
Figure 1.6: Graphical representation of the world energy consumption [23]..... 8
Figure 1.7: HVAC equipment demand and annual growth [27]..... 9
Figure 2.1: Cooling process representation of indirect evaporative cooler on psychrometric chart. 13
Figure 2.2: The schematic of the indirect evaporative cooler [37]. 14
Figure 2.3: Flow arrangement inside indirect cooler [38]. 14
Figure 2.4: Direct evaporative cooler (a) schematic diagram (a) psychrometric process [40]. 15
Figure 2.5: Psychrometric Chart showing the Enthalpy Change used to determine Capacity [46]..... 16
Figure 2.6: Dehumidification and regeneration process of a desiccant dehumidifier [52]. 18
Figure 2.7: Different configurations for various desiccant system [53]. 18
Figure 2.8: Principle of the thermally activated evaporative desiccant cooling systems [59]. 20
Figure 2.9: Desiccant evaporative cooling system operating on ventilation cycle [64]. 21
Figure 2.10: Desiccant evaporative cooling system operating on recirculation cycle [64]. 21
Figure 2.11: Desiccant evaporative cooling system operating on Dunkle Cycle [65]..... 22
Figure 2.12: A simple schematic of experimental desiccant cooling system in ventilation mode, and its psychrometric chart representation for a typical operation [66]. 23
Figure 2.13: Schematic of compact cross flow type plate heat exchanger [76]. 26
Figure 2.14: Schematic diagram of the standard desiccant cooling system with pre-cooling and DEC [91]..... 30
Figure 2.15: Schematic diagram of the standard desiccant cooling system with pre-cooling [91]. 30
Figure 2.16: The schematic of the two-stage evaporative cooler [92]..... 35
Figure 2.17: The psychrometric process representation of the two-stage evaporative cooler. 36
Figure 2.18: Modern Evaporative Cooling Media (Source: Munter). 38
Figure 2.19: Schematic diagram of modified indirect evaporative cooler [112]..... 39

Figure 2.20: Schematic of indirect evaporative cooler to achieve dew point temperature of the incoming air [39]. 39
Figure 3.1: Heat and mass exchanger configurations for various desiccant cooling technologies [133]. 51
Figure 3.2: Different desiccant materials isotherms for water adsorption [150]. 57
Figure 3.3: A liquid desiccant dehumidification system with liquid-liquid heat exchanger [163]. 62
Figure 3.4: Desiccant bed details [177]. 65
Figure 3.5: (a) Overall view of the system; (b) Cross-sectional view; (c) Side view [178]..... 66
Figure 3.6: Schematic diagram of dehumidifier with inner cooling [180]. 68
Figure 3.7: Spaced parallel plate packing material dehumidifier with inner cooling [183]. 69
Figure 3.8: Inner cooled dehumidifier with finned tube [184]. 70
Figure 3.9: Inner cooled dehumidifier (cross flow): (a) directions of air flow (b) fluid sprays contacting with different air streams [176]..... 72
Figure 3.10: A new type of internally cooled/heated dehumidifier/regenerator [189]. 74
Figure 3.11: Schematic of internally cooled/heated liquid desiccant dehumidification [191]. 75
Figure 3.12: Schematic of spray type internally cooled dehumidifier [192]. 76
Figure 3.13: Multi-stage liquid desiccant dehumidifier [135]. 77
Figure 3.14: Dehumidifier/regenerator with auxiliary cooling/heating module [193]. 77
Figure 3.15: Two-stage liquid desiccant dehumidification system [195]. 78
Figure 3.16: Structure of a parallel-plate membrane module [199]..... 79
Figure 3.17: Internally cooled vertical flat plate dehumidifier [95]. (a) Side view; (b) Inner view. 80
Figure 3.18: Internally cooled dehumidifier with corrugated plates [201]. 80
Figure 3.19: Schematic diagram of hollow fiber membrane module type multistage dehumidifier [193]. 81
Figure 4.1: Schematic of rotary liquid desiccant dehumidifier..... 94
Figure 4.2: Pattern of flow in a slot. 95
Figure 4.3: Monthly variations of DCOP, latent effectiveness, moisture removal capacity, and SER. 103
Figure 4.4: Hourly variations of DCOP, latent effectiveness, moisture removal capacity, and SER..... 103
Figure 4.5: Effect of process air mass flux on heat and mass transfer coefficients. 105
Figure 4.6: Effect of process air mass flux on outlet air humidity ratio. 106
Figure 4.7: Effect of process air flux on the latent effectiveness and DCOP. 107
Figure 4.8: Effect of inlet air humidity ratio on the latent effectiveness and DCOP..... 108

Figure 4.9: Effect of rotational speed on DCOP for different regeneration temperatures. 109
Figure 4.10: Effect of rotational speed on DCOP for different mass flow rate ratios. 110
Figure 4.11: Effect of rotational speed on dehumidifier moisture removal capacity for different process air mass flow rate. 110
Figure 4.12: Effect of inlet air conditions on sensible energy ratio for different ambient air temperatures. 112
Figure 4.13: Effect of mass flow rates ratio on sensible energy ratio for different regeneration temperatures. 113
Figure 5.1: Schematic of experimental setup. 118
Figure 5.2: Photographic view of experimental setup. 119
Figure 5.3: (a) Desiccant wheel; (b) direct evaporative cooler; (c) indirect evaporative cooler. 121
Figure 5.4: Psychrometric representation of desiccant cooling cycle. 122
Figure 5.5: Effect of ambient air humidity ratio on: (a) COP; (b) ECOP; (c) TCOP; (d) moisture removal rate; (e) DCOP; (f) sensible energy ratio. 132
Figure 5.6: Effect of ambient temperature on: (a) COP; (b) ECOP; (c) TCOP; (d) moisture removal rate; (e) DCOP; (f) sensible energy ratio. 135
Figure 5.7: Effect of process air mass flux on: (a) COP; (b) ECOP; (c) TCOP; (d) moisture removal rate; (e) DCOP; (f) sensible energy ratio. 138
Figure 5.8: Effect of regeneration air mass flux on: (a) COP; (b) TCOP; (c) DCOP; (d) sensible energy ratio 140
Figure 5.9: Effect of regeneration temperature on: (a) COP; (b) TCOP; (c) moisture removal rate; (d) DCOP; (e) sensible energy ratio. 143
Figure 5.10: Effect of desiccant concentration on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio. 144
Figure 5.11: Comparative results for effects of ambient air humidity ratio on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio. 147
Figure 5.12: Comparative results for effects of ambient air temperature on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio. 148
Figure 5.13: Comparative results for effects of process air mass flux on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio. 150
Figure 5.14: Comparative results for effects of regeneration temperature on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio. 151
Figure 6.1: Microbial growth as a function of relative humidity [241]. 156
Figure 6.2: (a) Systematic solid desiccant cooling system with indirect evaporative cooler (b) psychrometric processes. 159
Figure 6.3: Variations of moisture removal from the process air during the day. 165
Figure 6.4: Variations of wheel effectiveness on process side during the day. 165
Figure 6.5: Variations of wheel effectiveness on regeneration side during the day. 166

Figure 6.6: Variations of cooling effect produced during the day. 168
Figure 6.7: Variations of required regeneration load during the day. 168
Figure 6.8: Variations of COP during the day. 169
Figure 6.9: Variations of COP for different ratios of mass flow rates. 169
Figure 6.10: Variations of required regeneration load for different ratios of mass flow rates and regeneration temperature. 170
Figure 6.11: Variations of cooling effect for different process air flow rates. 170
Figure 6.12: Variations of moisture removal capacity for various process air mass flow rates. 171

NOMENCLATURE

A	channel area (m^2)
C_p	specific heat ($kJkg^{-1}\ ^oC^{-1}$)
h_t	heat transfer coefficient ($kWm^{-2}\ K^{-1}$)
h	specific enthalpy ($kJkg^{-1}$)
h_{fg}	latent heat of vaporization ($kJkg^{-1}$).
k	mass transfer coefficient ($kgm^{-2}\ s^{-1}$)
L	dehumidifier width (m)
$\dot{m}$	mass flow rate (kgs^{-1})
$\dot{m_{v1}}$	mass flow rate of water vapor at inlet of evaporative cooler (kgs^{-1})
$\dot{m_{v2}}$	mass flow rate of water vapor at outlet of evaporative cooler (kgs^{-1})
$\dot{m_w}$	evaporation rate of water (kgs^{-1})
M	moisture removal rate (kgs^{-1})
N_D	number of data points (-)
N	number of cells in each slot (-)
p_v	vapor pressure (kPa)
$\dot{Q}$	sensible cooling capacity (kW)
$\dot{Q}_t$	total cooling capacity (kW)

$\dot{Q}_c$	cooling load (kW)
$\dot{Q}_r$	regeneration heat (kW)
S	student test at a 95% confident interval (-)
t	time (s)
T	dry bulb temperature (^oC)
T_w	wet bulb temperatures (^oC)
T_{dp}	dew point temperature (^oC)
U	uncertainty error (-)
u	face velocity (ms^{-1})
V_H	air specific volume (m^3kg^{-1})
$\dot{V}$	volume flow rate of delivered air (m^3s^{-1})
W_{in}	energy input (kW).
w	desiccant moisture content ($kg_v\ kg_a^{-1}$)
x	axial distance (m)

Greek Symbols

Ɛ	effectiveness (-)
ω	humidity ratio ($kg_vkg_a^{-1}$)
ρ	density (kg/m^3)

ϑ	standard deviation (-)
μ	degree of freedom (-)

Subscripts

1, 2, 3…	state points
a	air
c	cross sectional
d	desiccant material
f	wall material
h	internal surface
i	inlet
o	outlet
p	process
r	regeneration
s	desiccant surface
v	vapor
w	liquid water

ABSTRACT

Full Name : Muhammad Mujahid Rafique

Thesis Title : Investigation of a solar assisted thermally activated air conditioning system

Major Field : Mechanical Engineering

Date of Degree : November 2015

In the Kingdom of Saudi Arabia, a vast amount of energy is used for air conditioning. One of the objectives of this work is to develop solar assisted energy-efficient air conditioning system using desiccant technology with compact components, suitable for households since the demand for domestic production capacity of window type unitary unit is about 74%. On the east and west coastal regions of the Kingdom, the temperature and humidity are high during the summer months. Dehumidification of air in hot and humid conditions is as important as cooling and removal of moisture from the air is much easier to achieve than cooling the air. Thus air is used as the energy transfer fluid in the proposed system.

The proposed technology is based on chemical dehumidification of air followed by evaporative cooling. The major energy required for the proposed system is low grade thermal energy such as solar energy for the regeneration of the desiccant.

In order to make the system compact and to be used as the window type unit, rotor disc is used for the dehumidifier. The proposed system works at atmospheric conditions and hence the construction is simpler. The proposed system offers an environmental friendly air conditioning system. It provides 100% fresh air without the application of chlorofluorocarbon and other similar refrigerants. A laboratory scale experimental facility

has been fabricated to evaluate the performance of a desiccant based air conditioning system and to validate the proposed theory.

ملخص الرسالة

الاسم الكامل: محمد مجاهد رفيق

عنوان الرسالة: استكشاف نظام تكييف هواء جديد يعتمد على الطاقة الشمسية ويشغل بالحث الحراري.

التخصص: الهندسة الميكانيكية

تاريخ الدرجة العلمية: نوفمبر 2015

تستهلك كمية هائلة من الطاقة في المملكة العربية السعودية لتكييف الهواء. وأحد أهداف هذا الاقتراح هو تطوير نظام تكييف للهواء بمساعدة الطاقة الشمسية يكون موفرا للطاقة ومستخدما التكنولوجيا المجففة ذات المكونات المدمجة ومناسبا للاستخدام المنزلي ، حيث أن الطلب على الإنتاج المحلي من وحدات تكييف النوافذ يبلغ حوالي 74٪. و بالنسبة للمباني التي تقع على الجانب الشرقي والمناطق الساحلية الغربية من المملكة، تكون درجة الحرارة والرطوبة مرتفعتان خلال أشهر الصيف. وإزالة الرطوبة من الهواء في الأجواء الساخنة والرطبة لها من الأهمية كالتبريد حيث إزالة الرطوبة من الجو أسهل بكثير من تبريد الهواء. ومن ثم يتم استخدام الهواء كسائل ناقل للطاقة في النظام المقترح.

وتستند هذه التقنية على التجفيف الكيميائي للهواء باستخدام السائل المجفف ثم بالتبريد التبخيري ثابت الحرارة. و الطاقة الرئيسية اللازمة للنظام المقترح هي أي طاقة حرارية منخفضة الدرجة مثل الطاقة الشمسية من أجل إعادة إنتاج المادة المجففة.

ومن أجل جعل النظام متوافقا وقابلا للاستخدام كوحدة نافذة، سيتم استخدام أقراص الدوران بوحدة التكييف، والنظام المقترح يعمل عند الضغط الجوي ومن ثم فإن تركيبه يكون أسهل. وهذا النظام فيكون من أنظمة التكييف الصديقة للبيئة حيث أنه يزودنا بهواء منعش بنسبة 100 % بدون أية تطبيقات أخرى من الكلوروفلوروكربون أو ما شابهها من المبردات. ثم أنه أيضا تم تنفيذ تجارب في المختبر لتقييم أداء أنظمة التكييف المعتمدة على المجفف لاثبات النظرية المقترحة.

CHAPTER 1

INTRODUCTION

In order to provide the human comfort indoor conditions, the cooling requirements should not be mentioned in terms of sensible cooling capacity (temperature control) only but latent cooling (control of humidity) should also be included especially for hot and humid outdoor conditions. The range of human comfort conditions and basic requirements for the human comfort need to be provided by the air conditioning system are illustrated in Fig. 1.1 [1] and Fig. 1.2 [2], respectively.

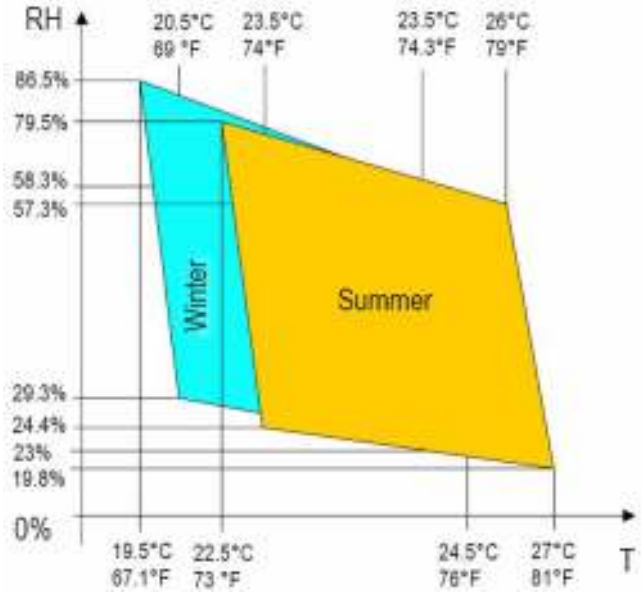


Figure 1.1: Thermal comfort zone according ASHRAE 55 [1].

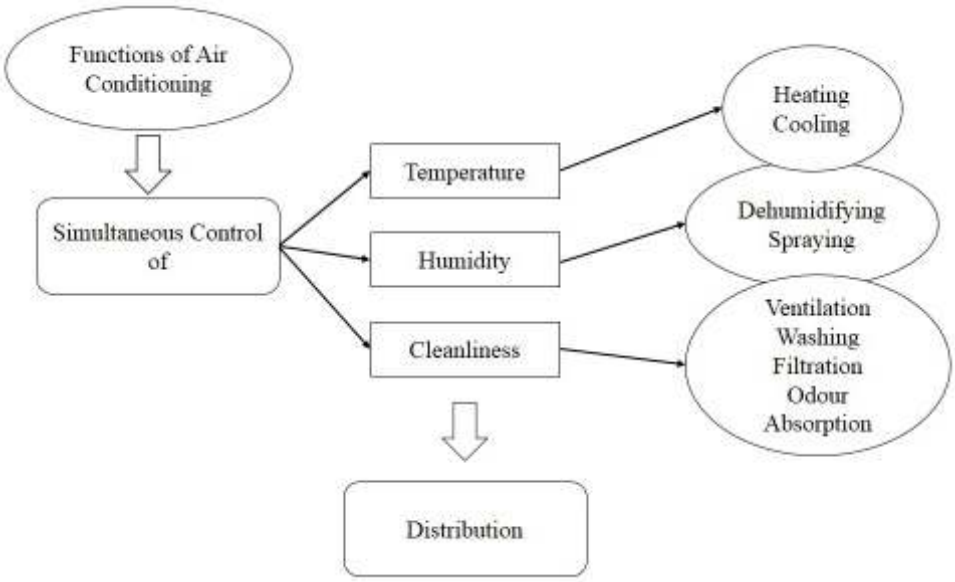


Figure 1.2: The functions of air conditioning [2].

The two components of the load are described by the sensible heat ratio which is the ratio between sensible load to the total load, that is, (sensible + latent). Smaller the value of sensible heat ratio larger the value of latent cooling load.

$$\text{Sensible heat ratio} = \frac{\text{Sensible heat}}{\text{Sensible heat+Latent heat}} \tag{1.1}$$

The value of sensible heat ratio is about 0.75 for the commonly used conventional vapor compression air-conditioning systems which means that 75% capacity of the system is used to control the sensible load and the remaining 25% for the latent load. So, the conventional systems can provide the comfort conditions only when sensible heat ratio is greater than 0.75 [3]. The value of designed sensible heat ratio can be significantly less than 0.75 for the hot and humid climates and which cannot be achieved using a conventional air conditioning system and hence thermal comfort conditions cannot be achieved. Secondly, the condensate coming out due to the overcooling, can evaporate back to the conditioned

space which may result in increased humidity level in the comfort zone [4]. These problems of conventional air conditioning systems can be addressed using a technology called desiccant cooling. This technology is a combination of a desiccant dehumidifier and evaporative cooler. The only energy used in this system is to drive the fans, water pump and to regenerate the desiccant dehumidifier during the regeneration process. This energy can be provided from any low grade thermal energy source such as solar, waste heat, etc. The sensible and latent loads can be controlled separately in this system using a humidistat and thermostat for the control of wet and dry bulb temperatures, respectively. This system can operate on wide range of sensible heat ratios because of the decoupling of sensible and latent cooling loads.

The simple evaporative cooler is not useful in hot and humid climates. Under such conditions, an evaporative cooling system can be used in conjunction with other dehumidification systems which can extract the water vapor from the air. This deficiency of the evaporative cooling system can be overcome by using it in combination with desiccant dehumidifier to dry the air. The application of adsorbent based dehumidification will allow the effective use of direct as well as indirect evaporative coolers in hot and humid climates [5,6].

1.1 Energy and environmental issues

The fast depletion of conventional energy resources and increasing demand of human comfort conditions because of increase in world population is becoming a major global environmental issue. These issues of energy, environment and technology are interrelated and for clean and greener environment these issues must be treated simultaneously and with interconnectivity as shown in Fig. 1.3 [7].

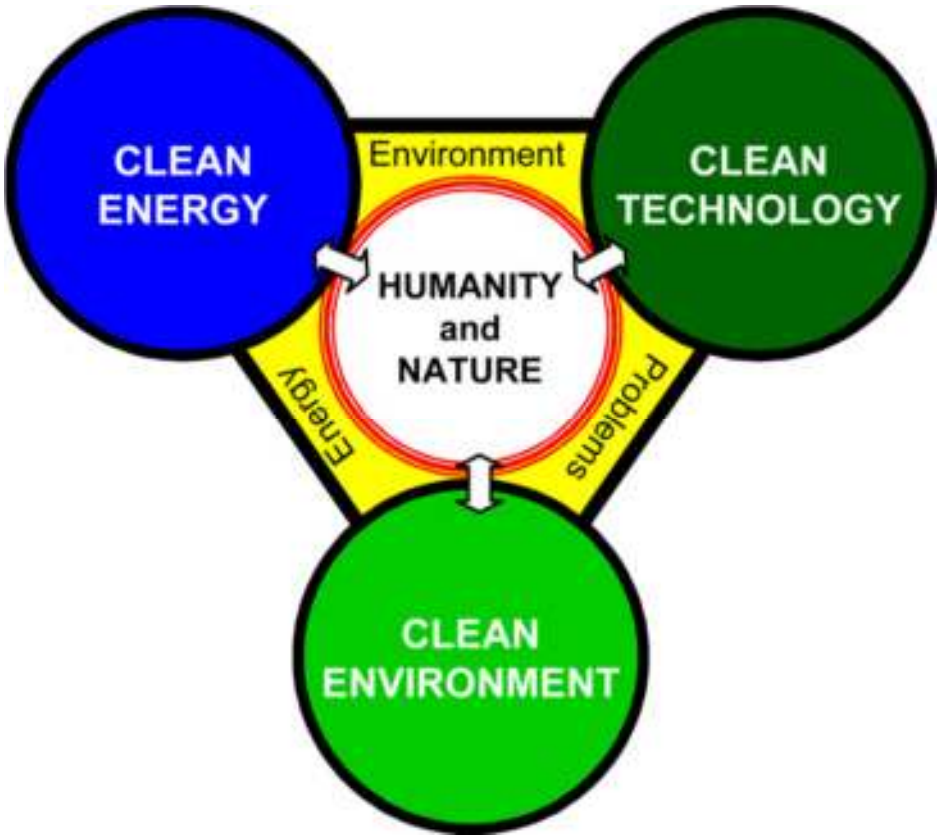


Figure 1.3: Interrelation between energy, environment and technologies issues to the nature and humanity survival [7].

A large amount of greenhouse gases are emitted due to the burning of conventional energy resources. The emission of CO_2 is increasing every year mostly because of economic development and fast increase in population in developing countries of the world. The use of air conditioning and heating devices plays a major role in the emission of these gases which causes depletion of ozone layer and other environmental issues [8-10]. All of these cause an increase in global temperature which causes many climatic and weather disturbance, that is, cyclones, flooding, etc. [11]. Figure 1.4 represents the increase in greenhouse gases because of human activities which affects the human health and sustainability of civilization [12,13].

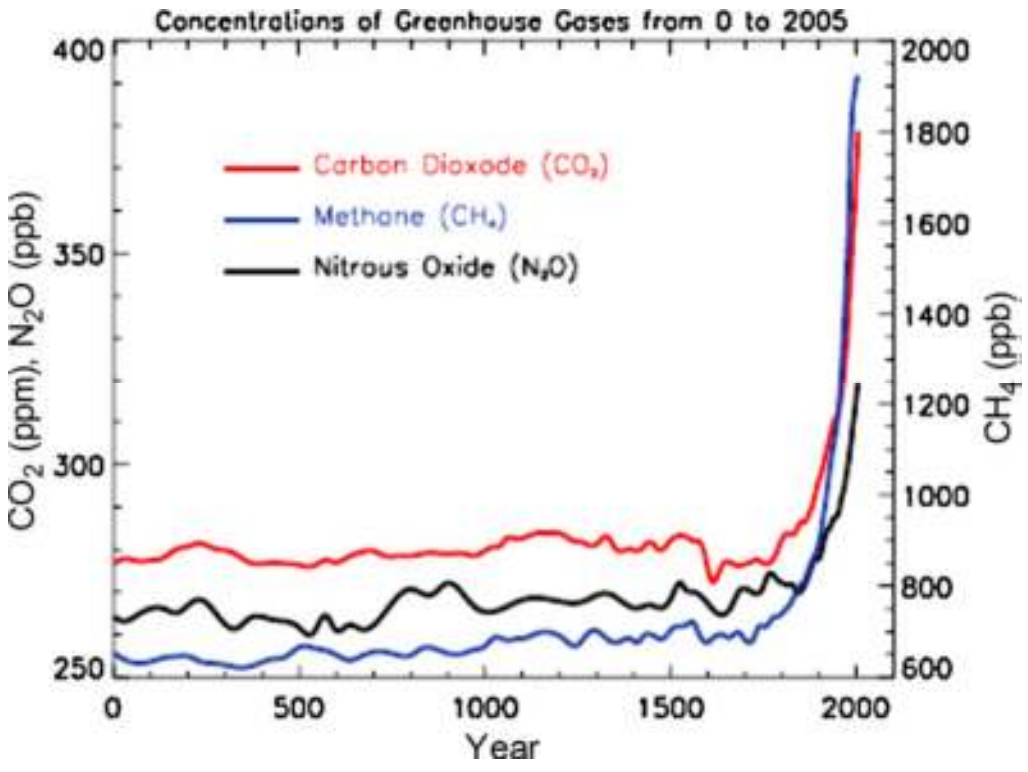


Figure 1.4: Long-lived greenhouse gases concentrations in atmospheric [12].

1.2 Human comfort and building indoor air quality

Because of the fact that indoor environment is mostly used for human activities so, residential as well as commercial buildings uses a large amount of primary energy for its maintenance and to support its occupant's activities [14]. This demand of energy is expected to increase because of increase in population and higher standards of living which directly affects the energy consumption. It is expected that the consumption of electrical energy for indoor environment will increase 120% from 2002 to 2030 [15]. The consumption of electrical energy for the agricultural sector increases to 56.7% in 2006 from 44.2% in 1973. The energy required to provide human thermal comfort conditions (control of temperature and relative humidity) is about 50% of building total energy consumption which in most cases is in the form of electrical energy [16].

1.3 Alternative cooling systems and dehumidification

In order to reduce the consumption of energy and to reduce the greenhouse gases emissions some alternative methods are required to provide the thermal comfort conditions and better indoor air quality [17]. These alternative systems reduces the building energy consumption largely through the utilization of some alternative resources of energy, for example, solar, biomass etc. [18]. Some alternative thermal cooling systems which directly utilizes the thermal energy are desiccant cooling, absorption cooling and jet cooling [19]. Solar energy can be effectively used for these systems because of variation in cooling load in phase with the solar radiation during the day [20]. Figure 1.5 illustrates some methods to convert

the solar energy for the purpose of air conditioning [21, 22]. In Fig. 1.5 market available solar assisted systems are represented as dark grey and technologies available as pilot projects are marked as light grey. The thermal cooling system utilizes thermal energy to provide the cooling effect.

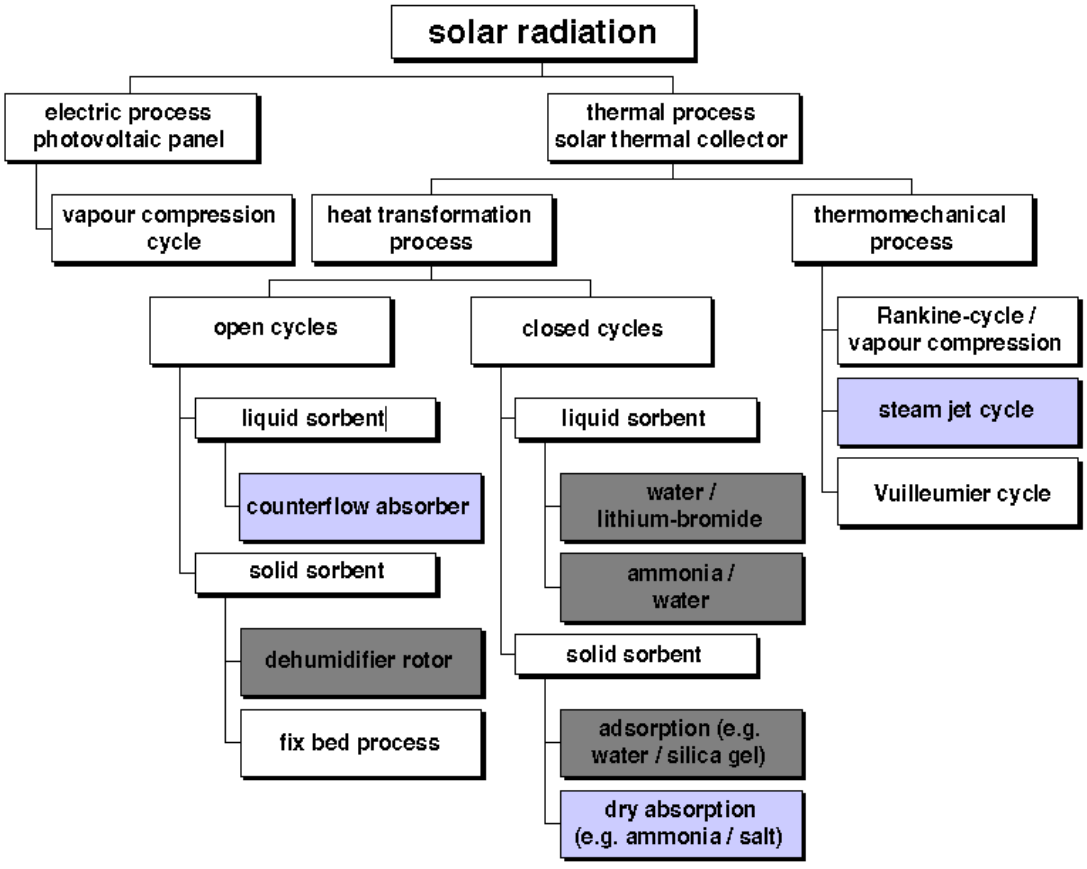


Figure 1.5: Methods for the conversion of solar energy into air conditioning or cooling process [21].

1.4 Background

The primary sources of energy, that is, oil, natural gas (NG), and coal are being consumed largely as compared to the renewable and environment friendly energy resources such as

solar, wind, geothermal, biomass, and hydro, as illustrated in Fig. 1.6 [23]. Apart from the fast depletion of primary energy resources, burning of these fuels causes CO_2 emission. This emission of CO_2 is increasing yearly mostly in developing countries. The increased emission of CO_2 and other greenhouse gases will cause many climatic disturbances because of increase in global temperature [24-26]. The heating, ventilating and air conditioning (HVAC) load of the world is estimated to rise 6.2% per year as shown in Fig. 1.7 [27].

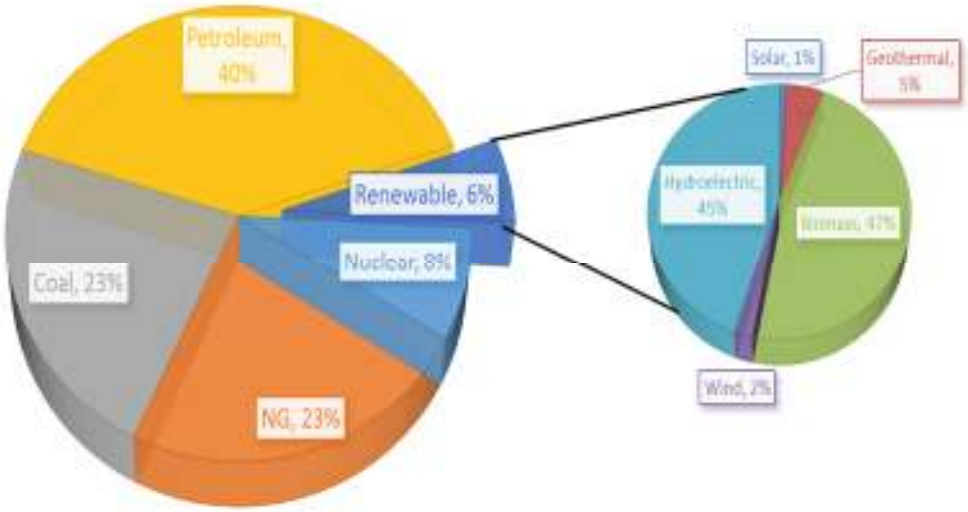


Figure 1.6: Graphical representation of the world energy consumption [23].

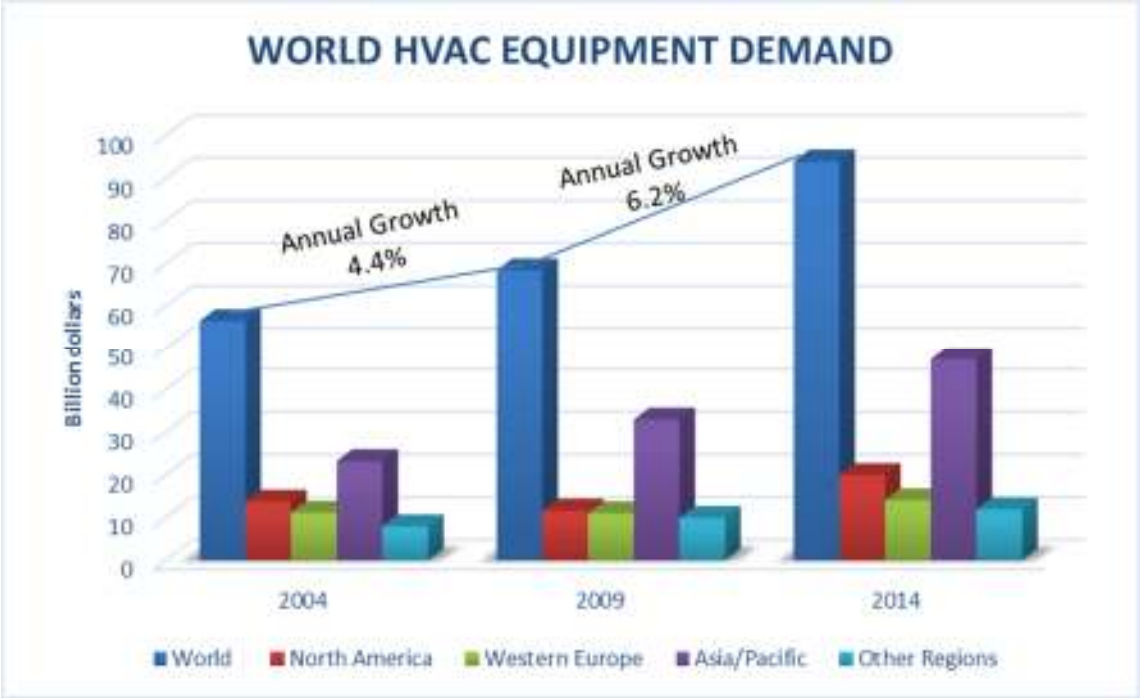


Figure 1.7: HVAC equipment demand and annual growth [27].

The renewable energy which was not utilized so well in the past is gaining attention now because of competitive cost, commercial acceptance, ease of maintenance and operation, and its environment friendly nature. Major part of the primary energy consumed in the building is accounted for cooling or heating. So in order to reduce the emission of CO_2 and CFCs to the environment the need of the alternative/s to the conventional cooling systems is needed which could make use of renewable energies in a better way.

The proposed research is to design, develop and operate the solar assisted energy-efficient air conditioning system using desiccant technology. The system is constructed in the laboratory at King Fahd University of Petroleum and Minerals with monitoring system to measure the performance of the developed system.

CHAPTER 2

DESICCANT BASED EVAPORATIVE COOLING

2.1 Introduction

The air conditioner should control the building sensible and latent load properly in order to provide the indoor comfort conditions. The conventional mechanical vapor compression system usually controls the latent load by the process of condensation of water vapor in which air is cooled below its dew point temperature and then reheated again up to the required supply conditions. The conditions where latent load is dominant these two processes i.e. overcooling and then reheating again will increase the consumption of electrical energy and emission of CO_2 remarkably. To avoid this wastage of primary energy and emission of harmful gases, desiccant based evaporative cooling system is a good alternative to traditional air conditioning system which is cost effective as well as environment friendly. It can be driven by thermal energy which makes a good use of solar energy which is free as well as clean.

The evaporative coolers appeared around 2500 B.C., when porous clay jars containing water were used by the ancient Egyptians for air cooling purpose. This evaporative cooling mechanism was applied to cool the ancient Egypt buildings and then spread across the hot region of Middle East. Similar types of mechanism to produce the cooling effect in the building also appeared at that time such as pools, water ponds, porous water pots, and thin water chutes. The evaporative coolers of modern type were started in USA. Several air conditioning devices which included indirect as well as direct evaporative coolers were

also invented in Arizona and California [28]. Many residential and commercial spaces were equipped with water dripping air coolers in Southwest, in late 1930s. In early 1950s, these air coolers were developed and available in wide range of market places including Canada, USA, and Australia.

In this chapter, a review of desiccant based evaporative cooling systems has been presented. The present study is undertaken from variety of aspects including background and need of alternative cooling systems, concept of conventional and desiccant based evaporative coolers, system configurations, operational modes, as well as current status of the desiccant based evaporative cooling technology. The review work indicated that the technology of desiccant based evaporative cooler has a great potential of providing human thermal comfort conditions in hot and humid climatic conditions at the expense of less primary resources of energy as compared to conventional cooling systems. Some modified and modern evaporative coolers have also been introduced in this chapter.

2.2 Basic principle and types of evaporative cooler

As mentioned previously, a well suitable alternative of mechanical vapor compression system is evaporative cooling system which can be efficiently used for air conditioning applications with less power requirements i.e. one fourth of the mechanical vapor-compression. It is an energy saving, cost effective, simple, and environment friendly air conditioning technique. Many researchers have investigated different types of evaporative coolers such as direct, indirect and modified coolers [29, 30].

Evaporative cooling systems are suitable for dry and high temperature climatic conditions [31]. These units can be used as direct contact evaporative cooling unit [32,33], indirect contact evaporative cooling [34,35] or as combination of both. Technologies exist which makes use of evaporative cooler to cool down the air without adding moisture content to it is commonly known as indirect evaporative cooling [36]. In the indirect evaporative system the process air stream does not interact directly with the cooling fluid stream rather it is cooled sensibly. The cooling process inside an indirect evaporative cooler is represented on psychrometric chart shown in Fig. 2.1. The temperature of air is lowered using some type of heat exchange arrangement in which primary air is cooled sensibly using a secondary air stream. The secondary air is cooled using water. In the indirect evaporative cooling system, both dry as well as wet bulb temperature of the air are lowered. The indirect evaporative cooling has an efficiency of 60-70%. The schematic and flow arrangements inside the indirect evaporative cooler is shown in Fig. 2.2 [37] and Fig. 2.3 [38], respectively.

In direct evaporative system moisture is also added to the cooled air stream because process air stream comes in direct contact with the cooling water. In direct evaporative cooling system, the temperature of the process air is lowered because of the high moisture content in the air so it is an adiabatic process which is only suitable for hot and dry climates and for hot and humid climates indirect evaporative cooler is preferred. In the direct evaporative cooling dry bulb temperature of the air is lowered and wet bulb temperature remains unchanged. The wet bulb temperature is an important parameter for the performance of direct evaporative cooler. The effectiveness of a well-made direct

evaporative cooler reaches an effectiveness of approximately 85% [39]. Both the schematic and psychrometric process of the direct evaporative cooler is shown in Fig. 2.4 [40]. The ambient air comes in direct contact with the spayed water which decreases the temperature of the supply air and adds moisture content to it as shown on the psychrometric chart. The transfer of heat and mass between process hot air and cooling water for a direct evaporative cooler is expressed as [41],

$$(\dot{m}_1 h_1 + \dot{m}_{v1} h_{v1}) - \dot{m}_w h_{fg} = (\dot{m}_1 h_2 + \dot{m}_{v2} h_{v2}) \quad \textbf{(2.1)}$$

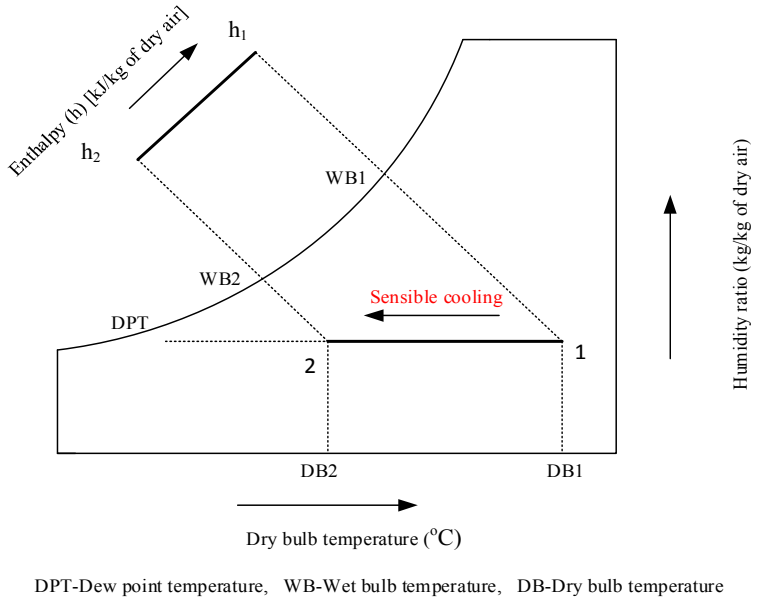


Figure 2.1: Cooling process representation of indirect evaporative cooler on psychrometric chart.

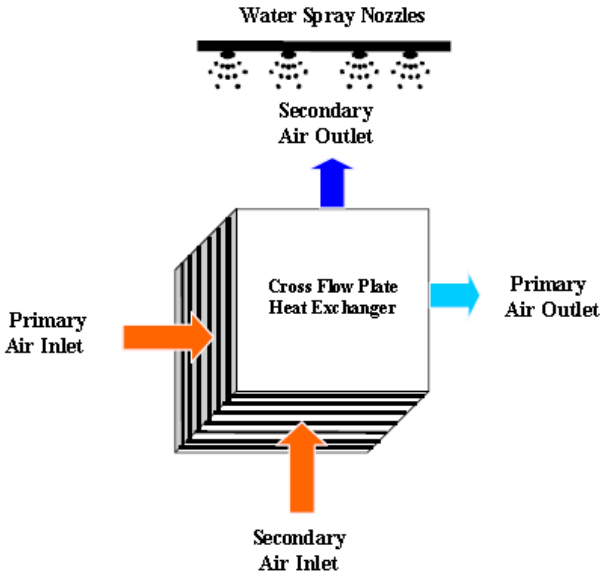


Figure 2.2: The schematic of the indirect evaporative cooler [37].

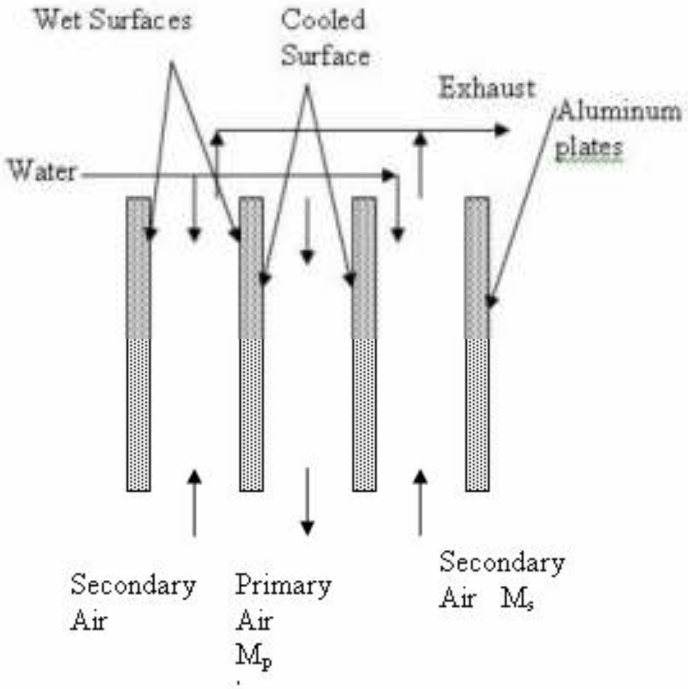


Figure 2.3: Flow arrangement inside indirect cooler [38].

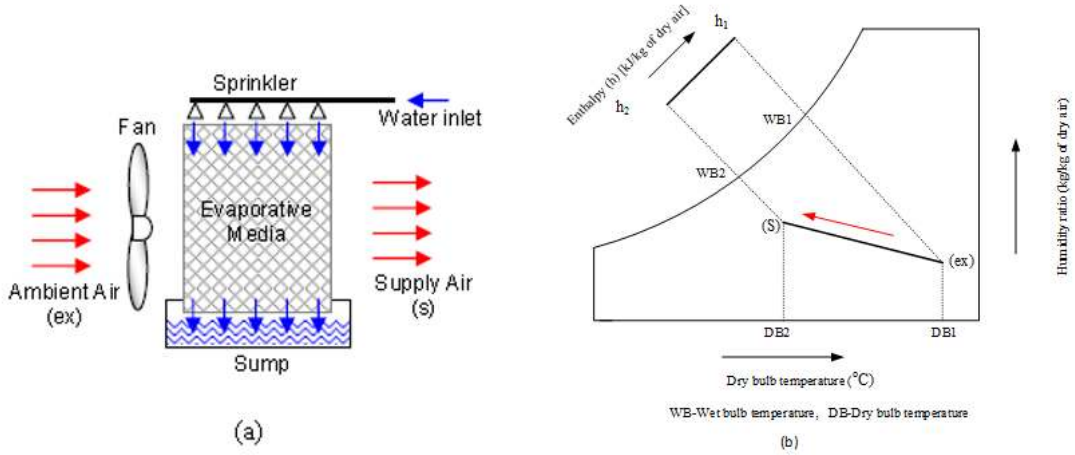


Figure 2.4: Direct evaporative cooler (a) schematic diagram (a) psychrometric process [40].

The process of indirect evaporative cooling needs input energy only for the water pump and fan that is why this system has high coefficient of performance. Camargo et al. [42] presented the evaporative cooling principles to achieve the human thermal comfort for indoor conditions. The principle of porous ceramics indirect evaporative cooler was presented by Riffat and Zhu [43]. The results showed that under dry and windy conditions, a high cooling capacity of the system can be achieved. It was also concluded in [43] that the better performance of the cooler can be achieved with reasonable velocity of the indoor air (0.6 ms^{-1}) and by increasing the thermal conductivity of heat pipe condenser. The air flow passage dimensions, air ratio (process to intake), and velocity of air are main parameters which defines the effectiveness and energy efficiency of a counter flow heat and mass exchanger while temperature of the feed water has less effect on effectiveness [44]. Similarly, the heat and mass transfer is less affected by the thermal properties (thermal conductivity and porosity) of the material used for heat and mass transfer in an indirect

evaporative cooler while cost, shape formation, holding ability, durability, compatibility with water-proof coating become the main factors of concern [45].

The difference of cooling processes of mechanical cooling system and direct evaporative cooler on psychrometric chart is illustrated in Fig. 2.5 [46]. The chart shows that a greater value of mass flow rate is required for evaporative cooler as compared to conventional air conditioner in order to control the same cooling load because of smaller enthalpy difference. Figure 2.5 also illustrates that under the same operating conditions the resulting humidity of the supply air will be much higher with the evaporative cooler as compared to the conventional air conditioners. Performance comparison and characteristics of different types of evaporative cooling system is presented in Table 2.1.

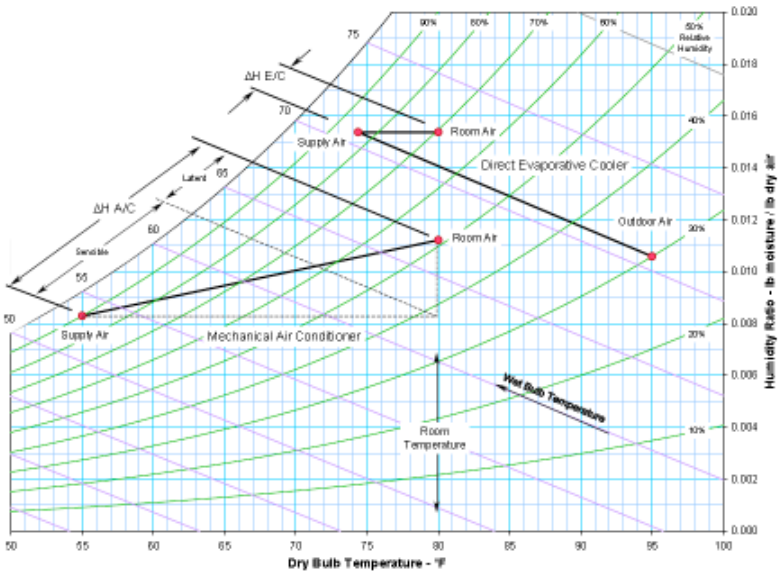


Figure 2.5: Psychrometric Chart showing the Enthalpy Change used to determine Capacity [46].

Table 2.1: Types and characteristics of evaporative cooling systems.

Type	Technology status	Comfort impact	Energy savings potential	Effectiveness
Direct	Mature	Humidity Increase	70%	80-90%
Indirect	Early production	No humidity increase	50%	About 85%
Indirect-Direct	Early production	Slight humidity increase	80%	110%

2.3 Desiccant dehumidifier

The desiccant dehumidifier composed of a desiccant material which removes moisture from the air by the process of dehumidification. Different desiccant materials attracts the moisture from the air at different capacities [47]. The desiccant materials can be solid as well as liquid. Silica gel, calcium chloride, lithium bromide, lithium chloride, activated ammonia and natural zeolite are the most commonly used desiccants. The desiccant dehumidifier is regenerated using thermal energy. Several researchers used solar energy as the input source for regeneration of desiccant dehumidifier [48-50]. The standard method which is mostly utilized is rotating desiccant wheel embedded with silica gel or lithium chloride [51]. The process of dehumidification and regeneration of a desiccant dehumidifier is illustrated in Fig. 2.6 [52]. The desiccant removes moisture from the air (1-2) and desiccant is regenerated by removing moisture from it using hot air (2-3). During process (3-1) desiccant is cooled down again. Solar energy can be used effectively to provide heat for regeneration. Ahmed et al. [53] stated the different possibilities to use solar

energy for regeneration in solid desiccant systems. The desiccant dehumidifier can be of different configurations which are illustrated in Fig. 2.7.

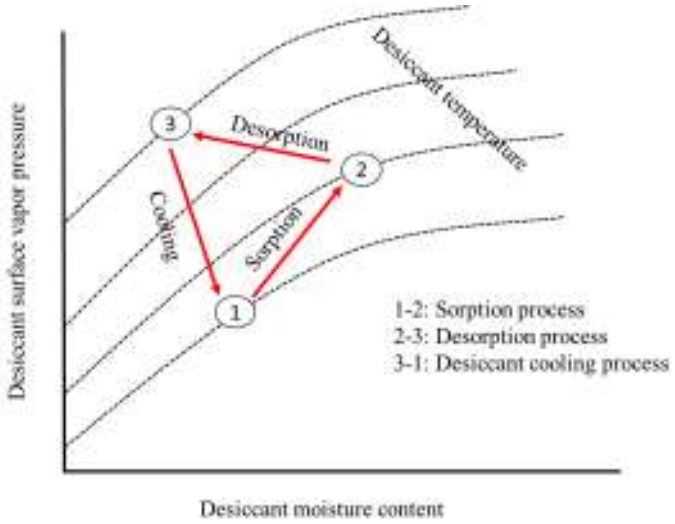


Figure 2.6: Dehumidification and regeneration process of a desiccant dehumidifier [52].

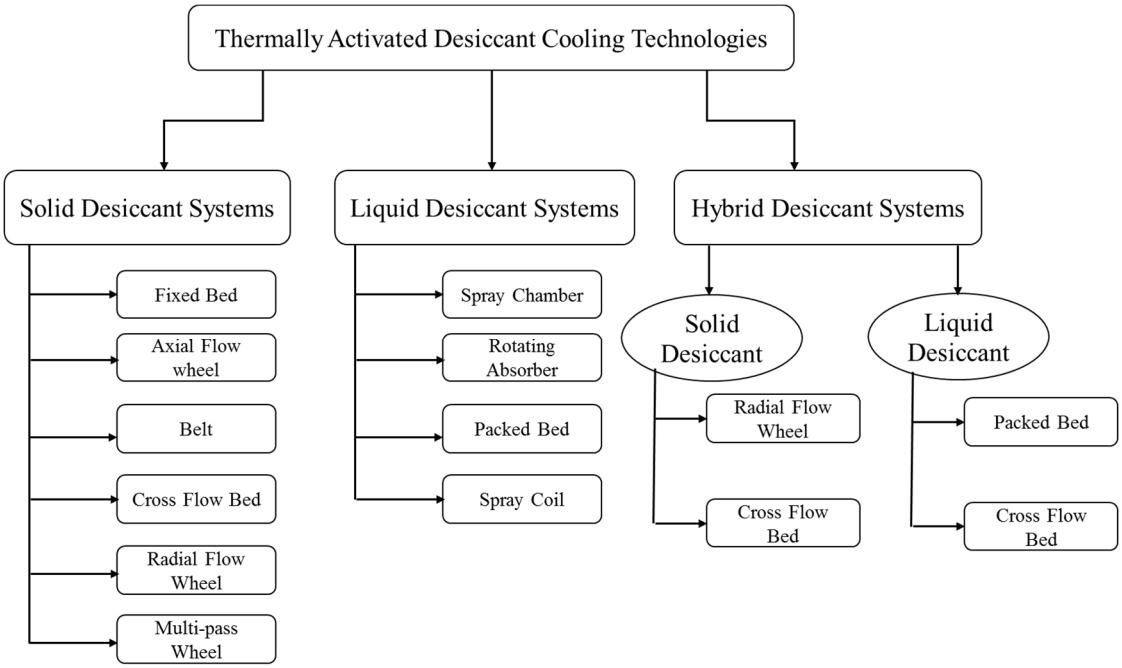


Figure 2.7: Different configurations for various desiccant system [53].

2.4 Desiccant based evaporative cooling

2.4.1 General idea

In general, evaporative cooling systems are applied when the wet bulb temperature does not exceed much beyond 25°C frequently [54]. The evaporative cooling units can operate with a high coefficient of performance (COP) in dry climatic conditions [55]. But because of the air saturation of the surrounding air in humid climates, the effectiveness of these cooling units drops remarkably. That is why evaporative cooler is best suited in conjunction with the desiccant dehumidifier, which removes the moisture from the process air and thus these cooling units can function effectively. The desiccant dehumidifier composed of some desiccant material (silica gel, lithium chloride, lithium bromide etc.) which is used to remove the moisture from the moist air. A desiccant material is one which absorbs or adsorbs and hold water vapor from the humid air by the process of absorption or adsorption [56,57]. The evaporative desiccant cooling system compromises of a desiccant dehumidifier, a regenerator, and a cooling unit [58]. The basic working principle of a thermally activated evaporative desiccant cooling system is illustrated in Fig. 2.8 [59]. The air is dehumidified using desiccant dehumidifier and its temperature is lowered using evaporative cooler or some other cooling device. For continuous operation of the system the desiccant dehumidifier is regenerated by using thermal energy provided by solar collectors or some other source of energy as shown in Fig. 2.8. Some heat recovery units are also utilized to make the system more efficient.

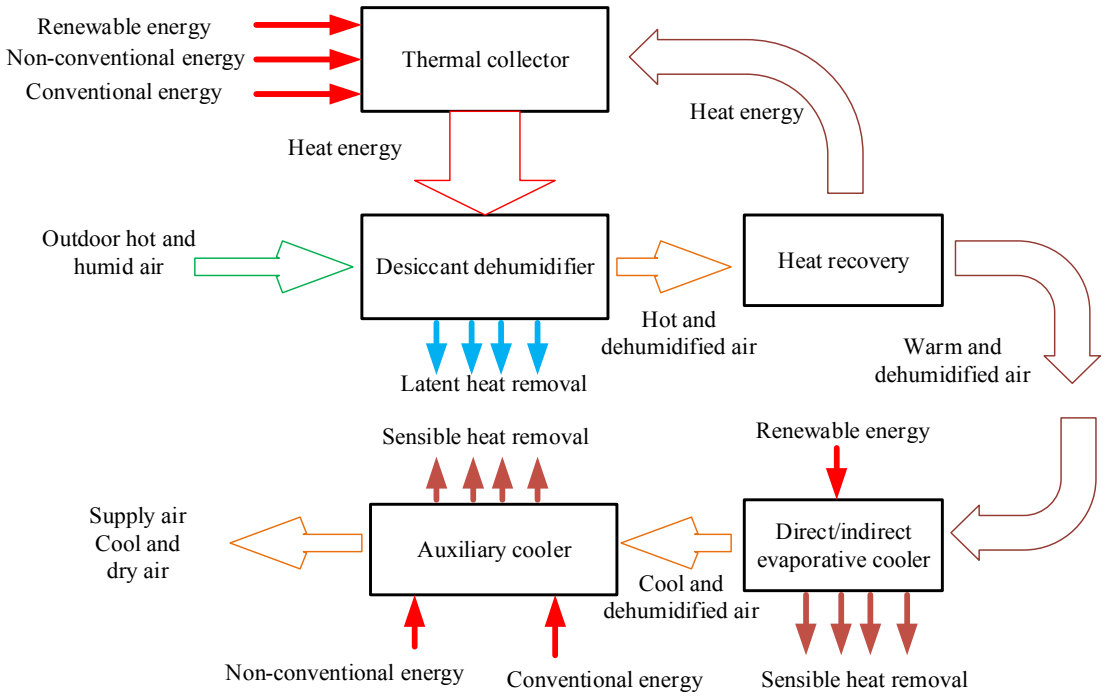


Figure 2.8: Principle of the thermally activated evaporative desiccant cooling systems [59].

In desiccant based evaporative cooling technique, latent and sensible loads are separately removed using desiccant dehumidification system and cooling unit, respectively. The type of cooling units used to reduce the temperature of dehumidified air, mainly defines the type of hybrid desiccant cooling system. The selection of the cooling unit depends on operating conditions, that is, humidity and temperature of the air. The most commonly used cycles for desiccant based evaporative cooling systems are recirculation [60], ventilation [61], Dunkle and wet surface heat exchangers [62]. Dezfouli et al. [63] compared the performance of solar desiccant cooling system operating on ventilation and recirculation mode under the climatic conditions of Malaysia. They concluded that the system has a coefficient of performance 0.8 and 1.6 under ventilation and recirculation mode,

respectively. The schematic of basic cycle for desiccant evaporative cooling system working on ventilation, recirculation and Dunkle mode are shown in Fig. 2.9, Fig. 2.10 [64] and Fig. 2.11 [65], respectively. In ventilation mode the returned indoor air passes through an evaporative cooler and then utilized for regeneration process as shown in Fig. 2.9, while in recirculation mode the 100% indoor air is recirculated again as the process air as shown in Fig. 2.10. Dunkle cycle has an additional heat exchanger with 100% recirculation of air through the desiccant dehumidifier on the process side while on the regeneration side, outdoor air is utilized as shown in Fig. 2.11.

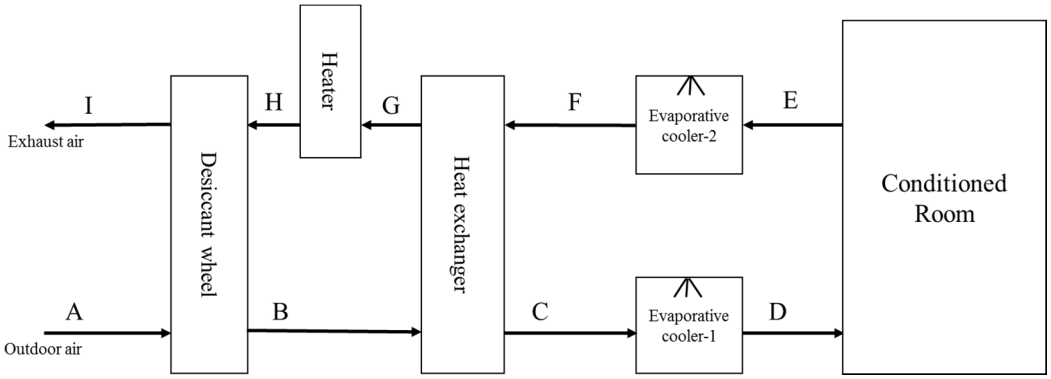


Figure 2.9: Desiccant evaporative cooling system operating on ventilation cycle [64].

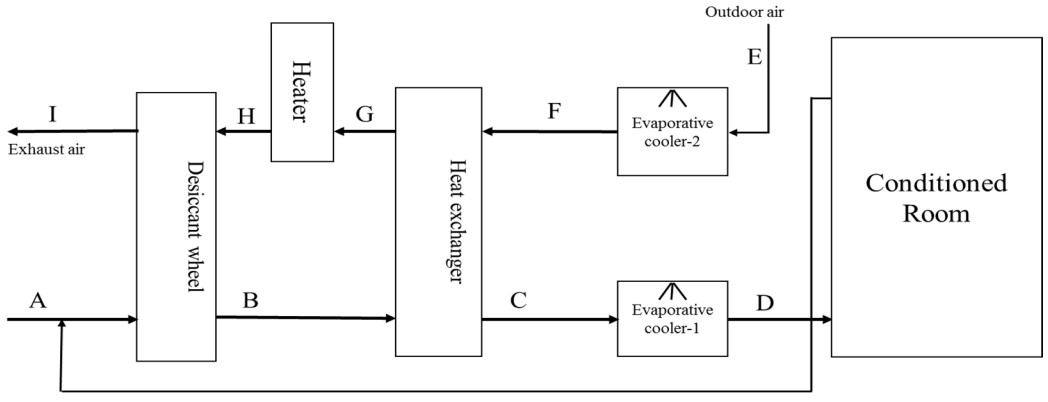


Figure 2.10: Desiccant evaporative cooling system operating on recirculation cycle [64].

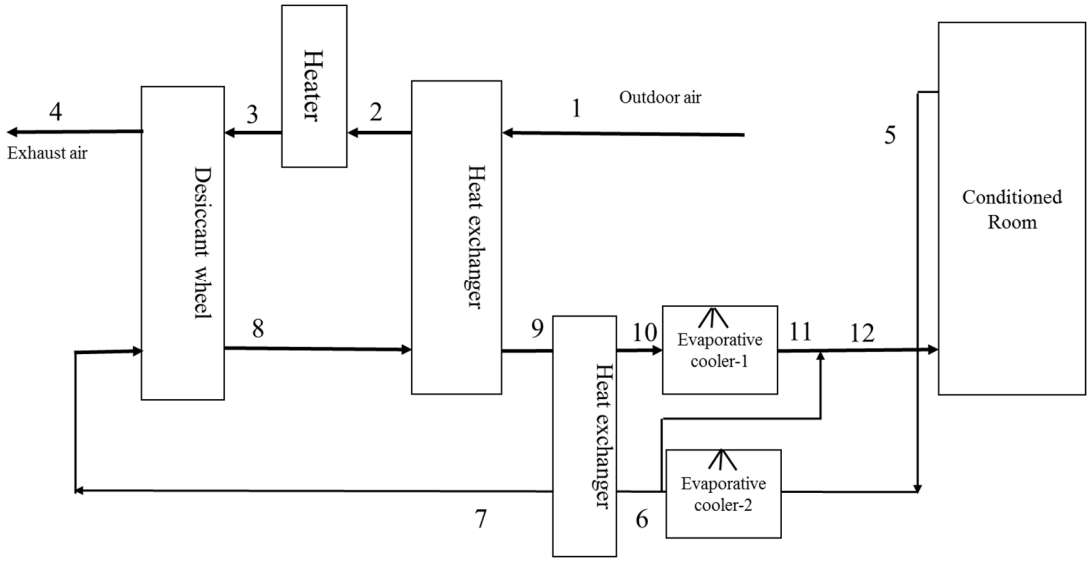
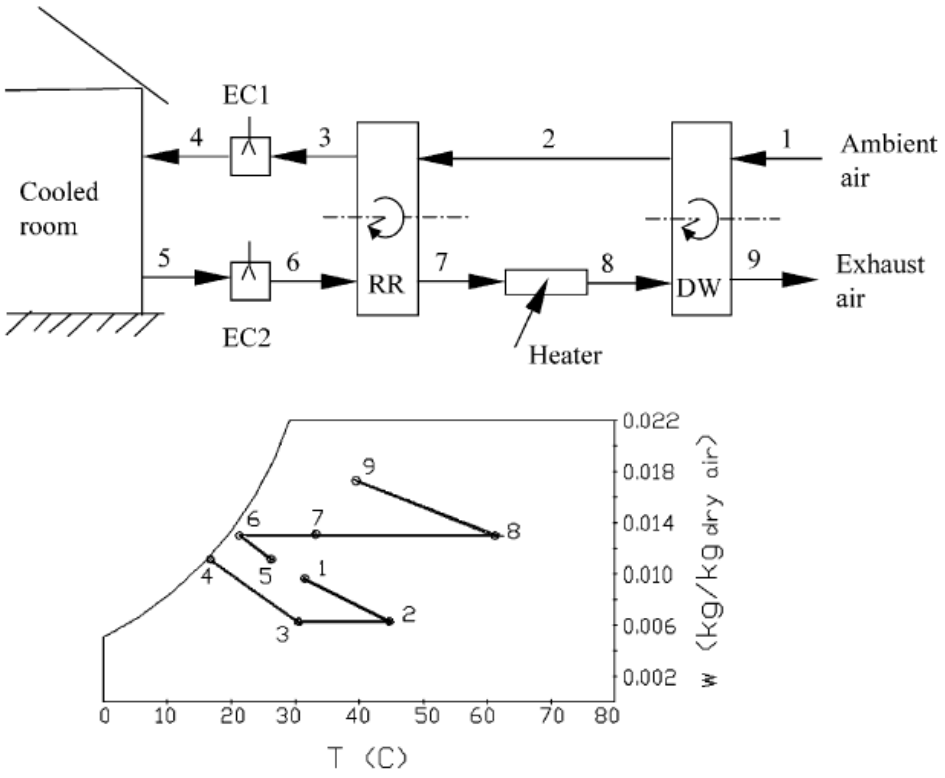


Figure 2.11: Desiccant evaporative cooling system operating on Dunkle Cycle [65].

2.4.2 System description

The schematic of a typical desiccant dehumidification system in conjunction with evaporative cooler, shown in Fig. 2.12, is operated on ventilation mode. The psychrometric representation of its processes is also shown in Fig. 2.12 [66]. As shown in Fig. 2.12, the hot, humid air is passed through the desiccant dehumidifier (state 1), which absorbs moisture from the air and hot, dry air leaves the desiccant dehumidifier at state 2. The hot air is cooled from state 2 to state 3 in the rotary regenerator that is a heat exchanger. Then evaporative cooler (EC1) is used to cool the process air from state 3 to state 4 according to the supply conditions. The return air from the conditioned space is passed through second evaporative cooler (EC2) to cool down the regeneration air from state 5 to state 6, for the effective exchange of heat between hot process air and cold regeneration air in the rotary regenerator (state 7 to state 8). At state 8, air is heated up to the required regeneration

temperature using a heat source to regenerate the desiccant material in desiccant wheel. Then the air is exhausted to the atmosphere at state 9 and the cooling cycle is completed.



DW: desiccant wheel, RR: rotary regenerator, EC1: evaporative cooler 1, EC2: evaporative cooler 2

Figure 2.12: A simple schematic of experimental desiccant cooling system in ventilation mode, and its psychrometric chart representation for a typical operation [66].

2.4.3 Literature survey

The desiccant evaporative cooling systems lead a remarkable reduction in electrical energy consumption as compared to conventional units and it also reduces the number of discomfort hours inside the conditioned space [67]. Uçkan et al. [68] presented the first experimental results of a desiccant based evaporative cooling system for hot and humid

climatic conditions. The evaporative cooler used by Uçkan et al. in their experiment is of indirect type (IEC). The results showed that the ambient air can be cooled down to 19˚C from 31˚C and a continuous supply of air at 25˚C can be maintained to a conditioned space. Riangvilaikul and Kumar [69] proposed and tested an indirect evaporative cooling unit to reduce the temperature of the air leaving the desiccant dehumidifier without disturbing the humidity of air. The results showed a good performance of the system for both humid and dry outdoor conditions. The system resulted in a dew point effectiveness of 0.58-0.84 and wet-bulb effectiveness of 0.9-1.14, [69]. Instead of the evaporative cooler, a heat exchanger was used by Katejanekarn and Kumar [70] to lower the temperature of dry air coming from the desiccant dehumidifier without adding moisture to the air.

Parmar and Hindoliya [71] studied the potential of a solid desiccant based direct evaporative cooler for the climatic conditions of five cities of India and compared the performance of the system for all the cities. The 50% return air from the conditioned space was mixed with the regeneration air and the remaining 50% with the process air. The results showed that system COP for different cities varies between 0.14 and 0.21. The best performance of the system was observed under the conditions of coastal city (Mumbai). It was also concluded that the inlet humidity of the air and effectiveness of the direct evaporative cooler have a strong effect on the system performance. The system effectiveness increases 30-50% with the 15% increase in effectiveness of evaporative cooler. A cooling system using liquid desiccant dehumidifier using lithium bromide as liquid desiccant in conjunction with direct evaporative cooler without the circulation of air was proposed by Oliveira et al. [72]. The simulation results showed that this alternative

novel air conditioning system has a great potential to replace conventional air conditioning systems with initial cost lower than the conventional system.

Kessling et al. [73] experimentally tested liquid desiccant system in conjunction with the indirect evaporative cooler using a desiccant solution of lithium chloride which can be regenerated at low temperature of about less than 80°C. Wurtz et al. [74] studied the cooling potential of desiccant based evaporative cooling system for the climatic conditions of France using simulation model and validated the results experimentally. The results showed that system is suitable for the regions with moderate humidity ratio. Ouazia et al. [75] developed a prototype model of a desiccant evaporative cooling system for the residential buildings and observed its performance theoretically as well as experimentally. The obtained results showed that desiccant evaporative cooling is a suitable option to replace the conventional air conditioning systems for better control of both temperature and humidity especially for the areas with high latent load.

Saman and Alizadeh [76,77] proposed a liquid desiccant cooling system in conjunction with an indirect evaporative cooling system as shown in Fig. 2.13. It is a plate type heat exchanger (PHE) of counter-flow configuration having a number of air-water and desiccant solution passages separated from each other using plastic thin plates. The thin plates also provide contact surface for heat and mass transfer between air-water and desiccant solution. Secondary air which is cooled by direct evaporative cooler is brought in contact with the water on one side of the separating plate. The cooled secondary air removes heat from the primary air on the other side of the plate thus making use of indirect evaporative cooling.

This primary air is dehumidified concurrently using desiccant solution sprayed on cross flow contact area. The performance of the proposed model was observed theoretically and then experimentally under climatic conditions of Brisbane, Australia. It was found that the effectiveness of the evaporative cooler could reach 75% for the exchanger angle (angle made by the direction of desiccant spray with the horizontal) of 45°.

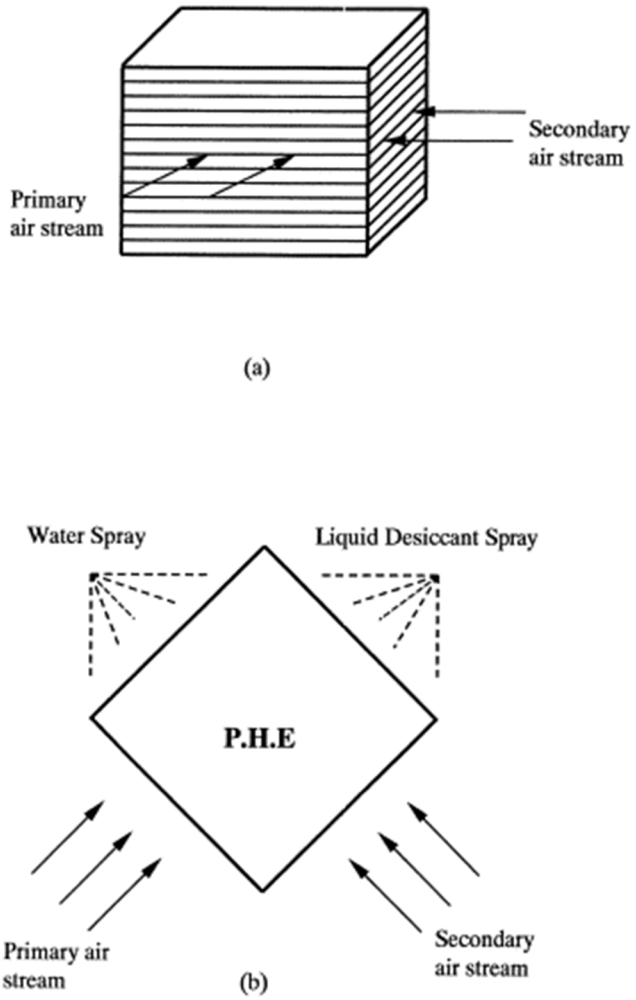


Figure 2.13: Schematic of compact cross flow type plate heat exchanger [76].

Nelson et al. [78] developed and simulated two different models of desiccant evaporative cooling system working on recirculation and ventilation cycles for the conditions of Miami, Florida. The results showed that 95% of energy can be obtained from the sun for the regeneration of desiccant wheel using a collector of $45m^2$ for the system working on ventilation cycle. Smith et al. [79] developed and studied the desiccant cooling operating on ventilation cycle in conjunction with direct evaporative coolers for three different locations Pittsburgh (Pennsylvania), Macon (Georgia), and Albuquerque (New Mexico). The obtained results showed that desiccant based evaporative cooling system had good performance and can meet the cooling load demands for all three locations. The maximum solar energy fraction of the regeneration energy requirements is for the Pittsburgh (about 75%). Al-Sulaiman et al. [80] analyzed a multistage evaporative cooling system using liquid desiccant dehumidifier between the stages. The thermal (line heater) as well as mechanical energy (Reverse Osmosis process) were used for the regeneration of the desiccant solution. The results showed that the energy consumption was 25% more while using mechanical source for regeneration as compared to the thermal source to increase the temperature of desiccant solution by 22°C.

Many researchers have studied the heat and mass transfer of desiccant evaporative cooling systems applying partial differential equations of heat and mass for individual components that is desiccant wheels or indirect evaporative coolers without analyzing the performance of overall system [81, 82]. Radhwan et al. [83] mathematically modeled a solar assisted liquid desiccant (calcium chloride) evaporative cooling system and observed the system performance for long term operation under conditions of Jeddah, Saudi Arabia. System

thermal ratio (STR), desiccant replacement factor (DRF) and solar utilization factor (SUF) were defined to observe the system performance for different conditions of weather. The results showed that system has a good performance for hot and humid climates. Bellemo et al. [84] numerically studied dew point evaporative cooler, also called as regenerative indirect evaporative cooler, which is a part of desiccant cooling system and analyzed its performance for different flow rates of air, inlet air conditions and recirculation fractions. The results showed that cooling capacity of dew point evaporative cooler was maximized for recirculation fraction of about 0.3 and supply conditions were mostly affected by the inlet air humidity ratio. Because of the regenerative arrangement dew point evaporative cooler did not require secondary air stream as required in indirect evaporative cooler. In regenerative arrangement, the wet bulb temperature of the inlet air can be achieved [85].

Goldsworthy and White [86] studied a solid desiccant evaporative cooling system design using an indirect evaporative cooler. It was concluded that the proposed system has a great potential to save energy and reduce the emission of greenhouse gases. The results also showed that a high value of electrical coefficient of performance for indirect evaporative cooler could be achieved with a regeneration temperature of 70°C, ratio of process to regeneration air flow rate of 0.67, and ratio of secondary to primary air flow rate of 0.3. Pescod [87] described a cross flow indirect evaporative cooler with a flow of water in the wet channel opposite to the flow of secondary air. The spacing of channel, protrusion details, length of the channel, flow velocities and flow rates of water were the key design parameters used in the study. A thermo economic analysis of direct evaporative cooler coupled with desiccant dehumidifier was carried by Camargo et al. [88] based on first and

second law of thermodynamics. Chen et al. [89] developed a heat and mass transfer model for the calculation of thermal and hydraulic performance of indirect evaporative cooler. They concluded that the system had much higher performance when room air was used as secondary air. Subramanyam et al. [90] studied the effect of different parameters, that is, air flow rate, speed of wheel etc. on the performance of the solid desiccant wheel for low humidity conditioning. The optimum wheel speed was found to be 17.5 rpm for better system performance and moisture removal capacity of the desiccant wheel.

Khoukhi [91] simulated solid desiccant cooling system with direct and indirect evaporative cooler using measured data sets to observe the system performance under hot and humid climatic conditions. Figure 2.14 shows the temperatures and humidity at each point of the cycle for a standard desiccant cooling cycle with precooling and direct evaporative cooler while Fig. 2.15 shows the results for the desiccant system just with pre-cooling only. The results showed that for the ambient dry bulb temperature and relative humidity of 36°C and 70%, respectively, the solid desiccant cooling system in conjunction with direct or indirect evaporative cooler can achieve the acceptable range of temperature and humidity for human comfort (29°C and 59%). Suryawanshi et al. [92] concluded that, as compared to conventional systems, two stage evaporative cooler is 4.5 times more efficient but only in hot and dry climatic conditions. For the hot and humid climatic conditions it can be combined with desiccant dehumidifiers. Mohammad et al. [93] found from the experimental results that a direct evaporative cooler can be used effectively in conjunction with the desiccant dehumidifier to provide the comfort conditions for the climatic conditions of Kuala Lumpur, Malaysia.

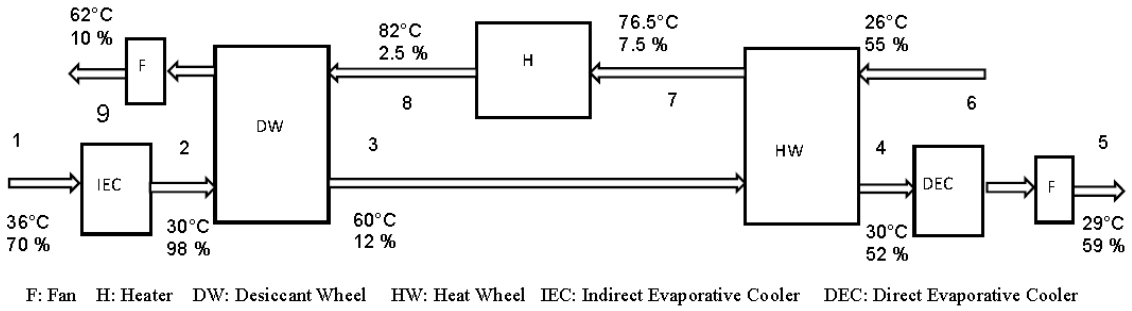


Figure 2.14: Schematic diagram of the standard desiccant cooling system with pre-cooling and DEC [91].

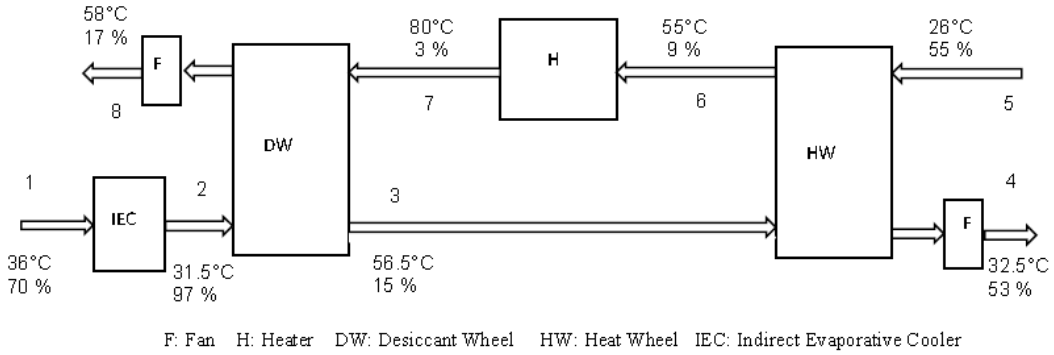


Figure 2.15: Schematic diagram of the standard desiccant cooling system with pre-cooling [91].

Jain et al. [94] evaluated the performance of solid desiccant based direct evaporative coolers operating on different cycles for 16 cities in India. The results showed that system operating on Dunkle cycle gives the best performance among ventilation, recirculation and Dunkle cycles. Kim and Jeong [95] investigated the solid desiccant and evaporative cooling system based on 100% outdoor air to observe the thermal and energy performance of the system. Both indirect evaporative cooler (IEC) and direct evaporative cooler (DEC) are utilized in their work. The results showed that, this system could save about 74~77% of total system operating energy as compared to the conventional systems. Glav [96]

introduced staged regeneration for solid desiccant dehumidifier in his patent for better performance of the system. Worek et al. [97] reported that by using a desiccant of Type 1M which can be regenerated at 165°C and with staged regeneration fraction of 16%, a ventilation cycle can operate with high performance. A rotary two stage desiccant cooling system using a composite desiccant material was developed by Ge et al. [98]. The experimental results showed that high performance of the system can be achieved with lower regeneration temperature which makes the use of low grade energy feasible.

2.5 Advantages of desiccant-aided evaporative cooling

Some advantages of the desiccant cooling technology in conjunction with the evaporative cooler are:

- It can be used for hot and humid climates because evaporative cooling alone is not feasible for such conditions.
- A lot of energy is saved as compared to vapor compression cycle because of no preheating is required.
- Environment friendly system because of no use of refrigerant which affects the ozone layer.
- Separate and better control of sensible and latent loads. The desiccant wheel controlling the latent part and the evaporative cooler controlling the sensible one.
- The overall system has low maintenance cost because it operates at almost atmospheric conditions.

- Low grade energy such as solar, biomass, etc. can be effectively used to drive the system.

2.6 Major applications of desiccant based evaporative cooling

In some spaces, better control of both temperature and humidity used to be required in order to avoid the growth of fungi and bacteria which affects the human health. Some of the main applications of desiccant based evaporative cooling are listed below [99]:

- Supermarkets
- Theatres
- Hospitals
- Hotels
- Office buildings
- Indoor swimming pools
- Pharmaceutical manufacturing plants

2.7 Performance Index

The cooling capacity ($\dot{Q}$) (sensible load) of an evaporative cooler is given as,

$$\dot{Q} = \dot{V} \times \rho_a \times C_{pa} \times \left(T_i - T_o\right) \quad \textbf{(2.2)}$$

The total cooling load (sensible and latent) of the desiccant evaporative cooling system is given as,

$$\dot{Q}_t = \dot{V} \times \rho_a \times (h_1 - h_2) \quad \textbf{(2.3)}$$

The evaporative cooler effectiveness can be given as,

$$\text{Effectiveness} = \frac{\text{Actual temperature drop achieved}}{\text{Maximum possible temperature drop}} \quad \textbf{(2.4)}$$

For wet bulb effectiveness, actual temperature drop is the difference between the dry bulb temperature of inlet process air and the temperature of process air at outlet of the evaporative cooler. The maximum possible temperature drop can be obtained from the difference between the dry bulb temperature and wet bulb temperature [42,100]. In dew point effectiveness the maximum possible achievable temperature drop is obtained from the difference between dry bulb and dew point temperature of inlet air [101].

$$\varepsilon_{wet} = \frac{T_i - T_o}{T_i - T_w} \quad \textbf{(2.5)}$$

$$\varepsilon_{dew} = \frac{T_i - T_o}{T_i - T_{dp}} \quad \textbf{(2.6)}$$

It is to be noted that counter flow indirect evaporative coolers can have a dew point effectiveness of above 100% because it can cool the air below its dew point temperature.

The Energy Efficiency Ratio (EER) of the evaporative cooler is the ratio between the cooling produced to the input energy to the cooler.

$$EER = \frac{\dot{Q}}{W_{in}} \quad \textbf{(2.7)}$$

The direct evaporative cooling systems have temperature effectiveness of about 70–95% [102]. Stoitchkov and Dimitrov [103] reported the indirect evaporative cooling system (IEC) are more attractive than the direct evaporative system but IEC has lower cooling effectiveness (about 40% to 60%). Camargo et al. [104] featured an effectiveness of 70-80% and 90% for indirect and direct evaporative coolers, respectively.

The wet bulb effectiveness of the evaporative coolers used in the desiccant based evaporator cooling system, shown in Fig. 2.12, can be evaluated as

$$\varepsilon_{EC1} = \frac{T_3 - T_4}{T_3 - T_{w3}} \tag{2.8}$$

$$\varepsilon_{EC2} = \frac{T_5 - T_6}{T_5 - T_{w5}} \tag{2.9}$$

The rate of moisture added by EC1 and EC2 to the process and regeneration air can be given as

$$\dot{m}_{w,\,EC1} = \dot{m}_1(\omega_4 - \omega_3) \tag{2.10}$$

$$\dot{m}_{w,\,EC2} = \dot{m}_2(\omega_6 - \omega_5) \tag{2.11}$$

2.8 Modified evaporative cooler

Eskra [105] introduced the concept of two stage evaporative cooling for higher efficiency of the system. In two-stage evaporative cooling both direct and indirect processes are combined. Both, the schematic diagram and psychometric process of the two-stage

evaporative cooler is shown in Fig. 2.16 and Fig. 2.17, respectively. The air is pre-cooled in the first stage using a heat exchanger by evaporation on the outside. In the second stage, air from the first stage is cooled and moisture is added to it as passes through the soaked pads. As air temperature is lowered in the first stage so less moisture is added in the second stage which in turns leads to better thermal comfort conditions. Two-stage evaporative cooler uses 100% fresh outside air and as compared to conventional systems it reduces the energy consumption of about 60 to 75%.

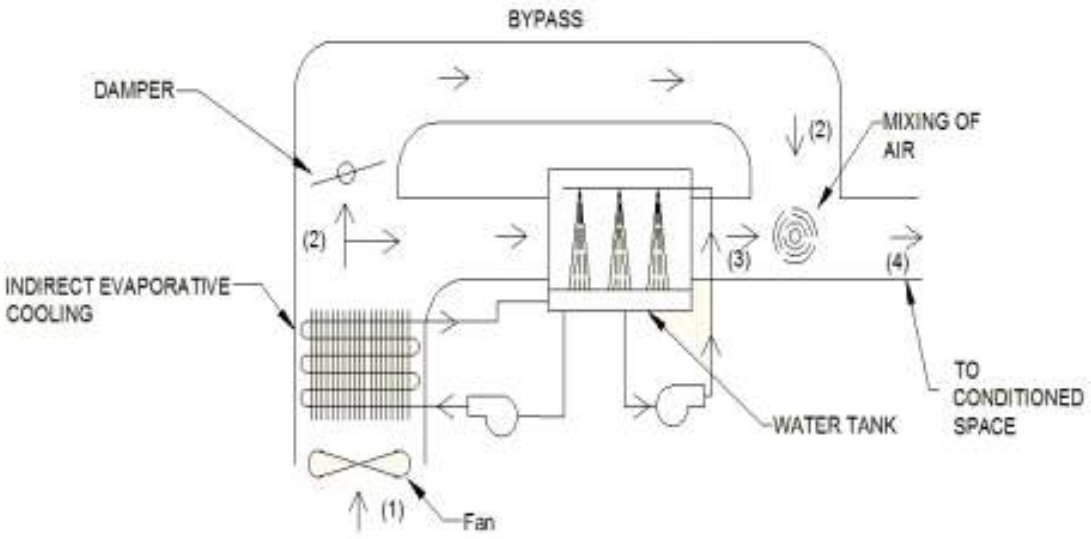


Figure 2.16: The schematic of the two-stage evaporative cooler [92].

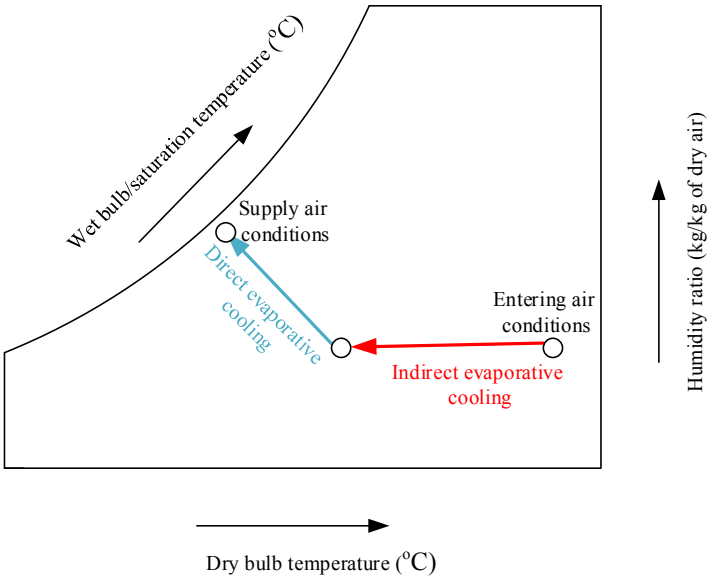


Figure 2.17: The psychrometric process representation of the two-stage evaporative cooler.

Kulkarni and Rajput [106] theoretically analyzed the performance of two stage evaporative cooler for the climatic conditions of Bhopal, India. The results showed that, for the air flow rate of primary air 0.3 to 1.25 kgs^{-1} the effectiveness of indirect evaporative cooler varied from 0.95 to 0.82. For the two stages combined operation, saturation efficiency varied between 121% and 107% and cooling load capacity from 5.06 to 20.50 kW as compared to saturation efficiency of 89% to 64% and cooling capacity of 3.18 to 16.6 kW for single stage direct evaporative cooling. The temperature obtained at the outlet of the cooler lies between 22.5°C and 24.6°C for ambient dry bulb temperature and relative humidity of 39.9°C and 32.8%, respectively. Watt [107] first analyzed different types of evaporative cooling systems. Watt also discussed about the history of different types (direct, indirect, two stage) of evaporative coolers and their operating principles. Maclaine-cross and Banks [108] developed a model for wet surface heat exchangers which can be used to predict the

performance of different types of evaporative coolers. The results showed that by using such a heat exchanger, the proposed regenerative evaporative cooling unit has excellent overall performance. Yellott and Gamero [109] studied different types of indirect evaporative coolers by introducing psychrometric analysis related to indirect evaporative coolers. Evaporative coolers can be utilized in almost all climatic conditions. The wet-bulb temperature of the entering air stream and secondary air stream limits the use of direct and indirect evaporative cooler respectively.

Eskra [110] studied a two stage indirect direct evaporative cooling system. The results showed that, for the ambient wet bulb temperature below 25^{o}C, the system would produce cooling of 5 to 15 times per Watt of energy input. In order to maximize the moisture transfer, some modern evaporative coolers have been introduced which combines the high performance media with low velocity air as shown in Fig. 2.18. These evaporative coolers with well design and thickness of media of about 10 to 12 inches (25.4 to 30.4cm) or more can have an effectiveness of around 93% as compared to maximum effectiveness of 80% of typical systems. In modified evaporative coolers, the air can be cooled to the temperature lower than the temperature achieved by indirect or direct evaporative cooler without disturbing the humidity of the air. Theoretically, the dew point temperature of the inlet process air can be achieved using modified evaporative cooler. But the modified evaporative cooler requires more fan power as compared to indirect or direct evaporative cooler because of splitting of air. It consists of a plate type sensible heat exchanger and a direct evaporative cooler [111].

Figure 2.18: Modern Evaporative Cooling Media (Source: Munter).

Bisoniya et al. [112] presented a model for modified indirect evaporative cooler and compared results of theoretical and experimental thermal analysis. The schematic is shown in Fig. 2.19 which is a cross flow heat exchanger with one fluid mixed and other unmixed. The results showed that theoretical model presented can be used to predict the performance of the modified indirect evaporative cooler. The results also showed that, the evaporative cooler had best performance in hot and dry climatic conditions. Bruno [39] observed the performance of indirect evaporative cooler installed at Roxby Downs, Australia for cooling a residential building. The schematic of the system is shown in Fig. 2.20 in which part of the cooled air is returned along the wet channel to achieve the dew point temperature of the incoming air. The obtained results showed the cooling power of 10.8 kW over the average day can be achieved with this indirect evaporative cooler for dry and hot conditions. The average value of outlet temperature of the system achieved can reach 14.9^{o}C at an ambient temperature of 34.7^{o}C. The system has average wet bulb and dew point effectiveness of 124% and 74%, respectively.

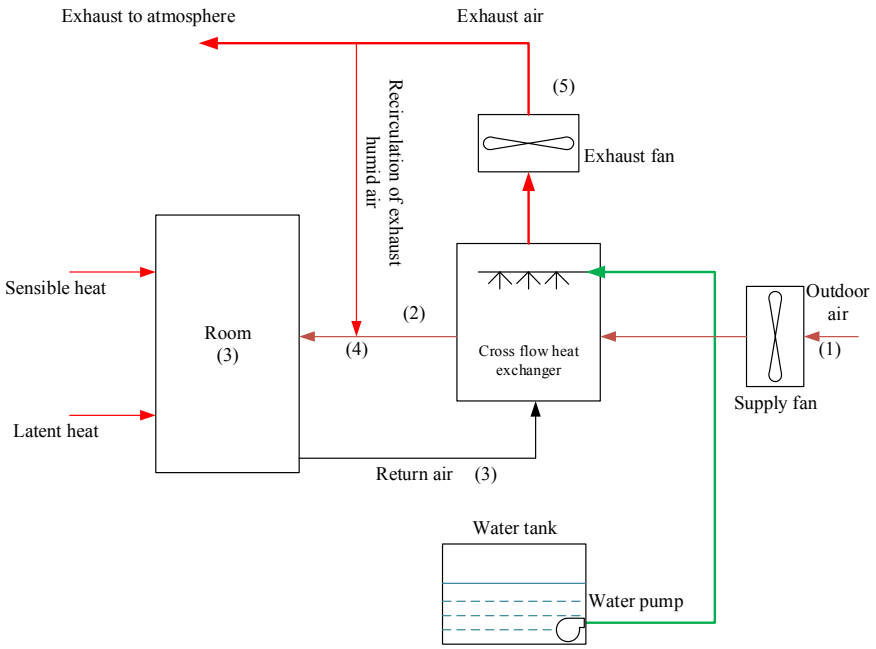


Figure 2.19: Schematic diagram of modified indirect evaporative cooler [112].

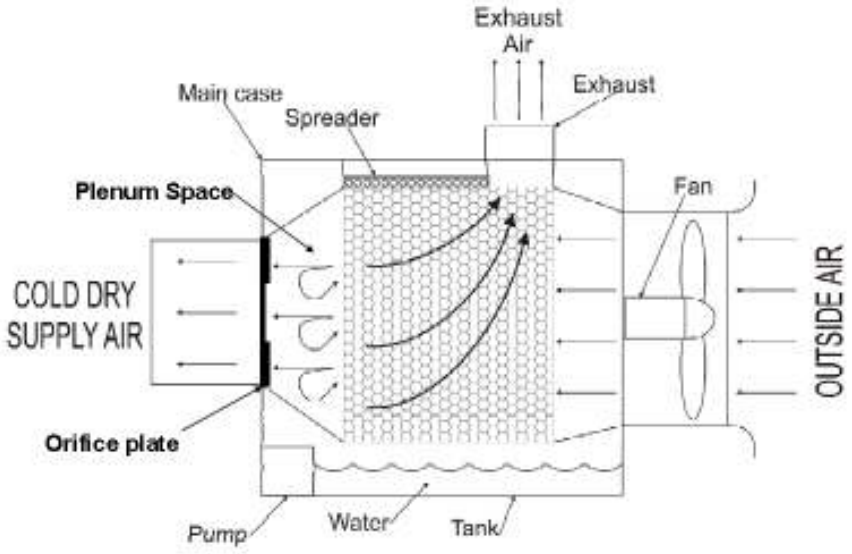


Figure 2.20: Schematic of indirect evaporative cooler to achieve dew point temperature of the incoming air [39].

2.9 Developments in evaporative cooling research

Because of the increasing interest and significant potential of the technology, various methods of evaporative cooling have been investigated by many researchers. Recently evaporative based air conditioning units are not commonly used as small household units. Al-Sulaiman [113] evaluated the local fibres performance in evaporative cooling using wetted pads of date palm fibres, jute and Luffa in the evaporative cooler. The obtained results showed that jute has highest efficiency of 62.2% while Luffa, commercial pads and date fibres has 55.1%, 49.9% and 38.9%, respectively. Alonso et al. [114] developed a universal heat and mass transfer model for design and optimization of indirect evaporative coolers. The results of detailed numerical study showed that transfer of heat from gas-liquid interface in determined by the latent heat transfer.

The 50% of the building heating load comes from the roof. Al-Nimr et al. [115] observed the performance of evaporative cooler by using different techniques to reduce the roof heating load like painting the roof with white, by insulating the roof from inside etc. the results showed an increase in the performance of the evaporative cooler. Ibrahim et al. [116] studied porous ceramic evaporators for building cooling applications using direct as well as indirect evaporative coolers. The results showed that ceramic materials with high porosity have better performance. Jain and Hindoliya [117] tested a regenerative evaporative cooler. It was developed by adding a water to air heat exchanger in the path of outgoing air stream from the direct evaporative cooler. It was found that COP and efficiency of the unit increases 20-25% because of the higher cooling capacity of regenerative evaporative cooler. The results of different studies showed that indirect

evaporative coolers are mostly preferred over the direct evaporative coolers especially when high humidity is not desired [99]. Different parametric and comparative studies have also been carried out by many researchers to analyze and compare the performance of evaporative coolers [118-120]. Different studies related to evaporative cooling systems has been summarized in Table 2.2. The evaporative cooling technology is under process of development. Some important steps needs to be taken for development of this technology have been summarized in Table 2.3 [121].

Table 2.2: Summary of different studies related to evaporative cooling systems.

Study	System description	Method	System performance/Outcomes
Zhao et al. [44]	Counter-flow indirect evaporative cooler (IEC) made from plate fin heat exchanger	Simulation	Wet bulb effectiveness (54-130%) Dew point effectiveness (36-82%)
Halliday et al. [48]	Solid desiccant based evaporative cooling system driven by solar energy under climatic conditions of UK	Analytical	About 72% of the thermal energy can be provided by solar energy for New Mexico
Enteria et al. [50]	Solid desiccant cooling system in conjunction with evaporative cooler	Experiment	Solar energy can be efficiently used for desiccant cooling system to minimize the use of electric energy
Archibald [55]	Solid desiccant based evaporative cooling	Experiment	The system can handle about 87% of total building cooling and heating load
Haddad et al. [67]	Desiccant-evaporative cooling system	Experiment and simulation	The system offers a promising alternative to vapor compression system

Riangvilaikul and Kumar [69,85]	Counter-flow IEC made from flat sheets, stacked structure heat exchanger	Experiment and simulation	Wet bulb effectiveness (92-114%) Dew point effectiveness (58-84%)
Parmar and Hindoliya [71]	Solid desiccant based direct evaporative cooler (DEC)	Analytical	COP (0.14-0.21)
Saman and Alizadeh [76,77]	Liquid desiccant based IEC made from plate type heat exchanger	Experiment and Analytical	Wet bulb effectiveness about 75%
Radhwan et al. [83]	Liquid desiccant evaporative cooling system	Analytical	-
Goldsworthy and White [86]	solid desiccant based IEC	Analytical	$COP_{electrical} > 20$
Khoukhi [91]	solid desiccant cooling system with direct and indirect evaporative cooler	Simulation	-
Bruno [101]	Counter-flow plate type exchanger based IEC	Experiment	Wet bulb effectiveness (106-124%) Dew point effectiveness (65-106%)
Camargo et al. [104]	Comparison of DEC and IEC	Analytical	-
Eskra [105]	Two stage evaporative cooling	Simulation	Reduction of energy consumption (60-75%)
Kulkarni and Rajput [106]	Two stage evaporative cooler	Analytical	Saturation efficiency (64-89%)
Eskra [110]	Two stage evaporative cooler	Analytical	Wet bulb effectiveness (93%)
Guo [119]	IEC made from plate fin heat exchanger	Analytical	Wet bulb effectiveness (78-95%)

Table 2.3: Needed R & D for desiccant based evaporative cooling technology [121].

Topic	Actions to be taken
Technology development	• Development of different tools and software for modeling. • Collaborative partnership between desiccant based evaporative cooling manufacturers, the HVAC industry, and researchers need to be established.
Field testing and performance mapping	• Additional field trials to be conducted. • Performance maps development for different climatic conditions and operating parameters.
Tools and software	• Simulation software tools should incorporate evaporative cooling technologies.

2.10 Conclusions

The desiccant based evaporative cooling system is relatively a new technology and is a good alternative for conventional mechanical vapor compression air conditioning system especially in hot and humid climatic conditions but less familiar as compared to conventional cooling system. In order to familiarize the desiccant based evaporative cooling systems, designed activities such as workshop, seminars, onsite visit, lectures, exhibition, and publicizing the research results in dedicated ways are required. Standardizations, legislations, public awareness, and regulations are the main issues needs to be focused for such cooling systems. In this paper, the work done by different researchers on desiccant based evaporative coolers has been summarized. The main observations are summarized as follows:

- The conventional air conditioning systems are not suitable for hot and humid climatic conditions because of its high sensible heat ratio. These systems can provides the comfort conditions only when sensible heat ratio is greater than 0.75.
- The evaporative cooling system is a good alternative to conventional air conditioning systems which not only saves energy but also is environmental friendly.
- In the indirect evaporative system the process air stream not directly interact with the cooling fluid stream and is only cooled sensibly while in direct evaporative system moisture is also added to the cooled air stream because process air streams comes in direct contact with the cooling water.
- The evaporative cooling system becomes less efficient in hot and humid climatic conditions. Under these conditions, evaporative cooling system can be operated in conjunction with desiccant dehumidifier with high efficiency.
- The heat and mass transfer is less affected by the thermal properties (thermal conductivity and porosity) of the packing material used for heat and mass transfer in an indirect evaporative cooler while main factors to be concerned are cost, shape formation, holding ability, durability, compatibility with water-proof coating.
- In modified evaporative cooler, the air can be cooled to the temperature lower than the temperature achieved by indirect or direct evaporative cooler without disturbing the humidity of the air.
- Low grade thermal energy such as solar, biomass etc. can be effectively used for the continuous operation of the desiccant based evaporative cooling system.

This technology is relatively immature in terms of production scale as compared to the conventional cooling systems. Furthermore, the environmental, economic, and social impacts which are related to this technology must be investigated and a clear understanding and knowledge of these issues needs to be addressed.

The desiccant based evaporative cooling technology can first be targeted for commercial buildings rather than residential buildings because of high energy and financial payback in hot and humid climatic areas. The combination of recent advances in desiccant materials and the evaporative cooling technology must be used to design cheaper, reliable and compact cooling systems. Finally, the technology should be converted into products which are market attractive.

CHAPTER 3

LITERATURE REVIEW OF LIQUID DESICCANT MATERIALS AND DEHUMIDIFIER

3.1 Introduction

It is important to decrease the energy cost of heating, ventilating and air conditioning (HVAC) systems without compromising indoor air quality and comfort conditions due to the rising cost of fossil fuels and other environmental concerns. Liquid desiccant cooling systems are one of the alternatives in this regard which is not only environmental friendly (no use of any refrigerant which deplete ozone layer) but at the same time make a good use of alternative sources of energy like, solar, biomass etc. This chapter presents different commercially available liquid desiccants and their composites which combines the properties of two or more desiccant materials for better performance. A good desiccant should have better moisture absorption capability and lower temperature of regeneration. This paper also includes the different configurations of liquid desiccant dehumidifiers and their advantages as well as drawbacks. Some new configurations of liquid desiccant dehumidifiers have been introduced which greatly improves the system overall performance. Moreover, a summary of the performance parameters has been made to analyze the system performance. This review is meaningful for the research and technical development process of liquid desiccant technology.

3.2 Liquid desiccant cooling

The control of indoor air temperature and relative humidity for the human comfort divides the building's cooling load into sensible and latent load, respectively. The conventional compressor based air conditioners can control the building sensible load effectively but these systems are less efficient to take the building latent loads. In these systems a lot of energy is wasted to overcool the air below its dew point to remove the moisture from the air by the process of condensation and then again reheating the air to the required supply temperature. Secondly, this process of overcooling provides the conditions for the growth of molds and bacteria because of surface wetting which can cause health issues by affecting indoor air quality [122]. Some alternative is required to avoid this wastage of energy because of overcooling and reheating [123]. The latent load is more dominant in hot and humid climates and there is a need for air conditioning system to effectively handle the latent loads [124]. The liquid desiccant cooling units seem to be a feasible and cost effective alternative to provide the human comfort conditions in hot and humid climates. It absorbs and desorbs the moisture from and to the air during absorption and regeneration processes, respectively [125].

In the liquid desiccant cooling system simultaneous transfer of heat and mass occurs. The latent load is controlled by the transfer of mass (moisture) from the humid air to the liquid desiccant surface. For the process of mass transfer, three theories namely penetration, film and surface renewable are mostly used. The film theory is mostly used for the mass transfer phenomenon. It was proposed by Nernset [126] by assuming the presence of resistance to

mass transfer within small region close to interference of two streams. Two film theory was developed by Whiteman [127].

The liquid desiccant systems are more energy efficient and provides an effective control of indoor air humidity [128]. Most of the liquid desiccant units are direct contact type in which air comes in direct contact with the desiccant solution [129]. The problem of carryover of desiccant solution is a major problem in these direct contact type units. The problem of carryover will affect the indoor air quality and will also increase the cost of maintenance because of corrosion [130]. Abdel-Salam et al. [131] proposed a new type of liquid air membrane energy exchanger as the dehumidifier to avoid the problem of desiccant solution carryover. The results under fully developed conditions give system coefficient of performance of 0.6 and humidifier sensible heat ratio of 0.3 - 0.5. Bichowsky [132] utilized the lithium chloride (LiCl) solutions for the purpose of drying the air. A field test was conducted for residential scale system but at that time this was not introduced to the market.

3.3 Performance parameters

The commonly used performance parameters which are utilized to investigate the performance of the system are listed in Table 3.1.

Table 3.1: Performance parameters used for liquid desiccant cooling systems.

Moisture removal rate (kg/s)	It represents the removed water vapor from the humid process air by the liquid desiccant cooling unit.
Humidity or moisture effectiveness	It is ratio between actual changes in humidity ratio of the air across the dehumidifier to the maximum possible change in humidity ratio of air.
Enthalpy effectiveness	It is ratio between actual changes in enthalpy of the air across the dehumidifier to the maximum possible change in enthalpy of air.
Sensible heat ratio	It is the ratio between the sensible energy to the total energy removed from the process air by the dehumidifier.
Cooling capacity (kW)	It is the overall cooling effect produced by the system. It is a difference between the enthalpies of inlet and process air.
Coefficient of performance	It represents the ratio between the cooling capacity to the total consumed energy (electrical and thermal) by the liquid desiccant cooling system.
Electrical coefficient of performance	It is a ratio between the system cooling capacity to the consumed electrical energy by the liquid desiccant cooling system.
Thermal coefficient of performance	It is a ratio between the unit cooling capacity to the consumed thermal energy in the liquid desiccant cooling system.

3.4 Advantages of using liquid desiccants

To control the latent load for air conditioning solid as well as liquid desiccant systems can be used in different configurations as shown in Fig. 3.1 [133] . The liquid desiccant system has following advantages:

- Lower drop in the pressure of process air stream.
- Low grade energy sources can be utilized for regeneration purposes because liquid desiccant systems have lower regeneration temperature (about 65 - 80°C) with good dehumidification performance.

- For large capacity cooling system multiple liquid desiccant cooling systems can be connected by pumping the liquid desiccant in between them.
- The dehumidification and regeneration of liquid desiccant simultaneously, is eliminated because weak liquid desiccant can be stored until regeneration heat is not available.
- The entire weak desiccant solution need not to be circulated for regeneration. A small amount of desiccant solution can be over-concentrated and mixed with weak solution.
- It is highly beneficial factor that liquid desiccants have the ability to filter different bacteria, microbial contamination, viruses, and molds which can affect the human health from process air stream.

The difference of surface vapor pressure between liquid desiccant layer and humid air act as the driving force for mass/moisture transfer; therefore, in order to increase the regeneration efficiency of the system, the surface vapor pressure of desiccant must be increased by preheating the desiccant solution leaving the dehumidification unit. Similarly, in order to achieve higher efficiency for the dehumidification process, pre-cooling of liquid desiccant at the inlet is needed to lower its surface vapor pressure. Heat exchangers are utilized in between the two streams to effectively preheat the weak solution leaving the dehumidification unit and to pre-cool the entering strong solution. In order to avoid the crystallization, an auxiliary heater can be equipped with the storage tank. The normal operation of the system can be affected by this auxiliary heater because clogging in the pipe [134]. Jiang et al. [135] compared liquid desiccant cooling system and conventional air conditioning systems which are listed in Table 3.2.

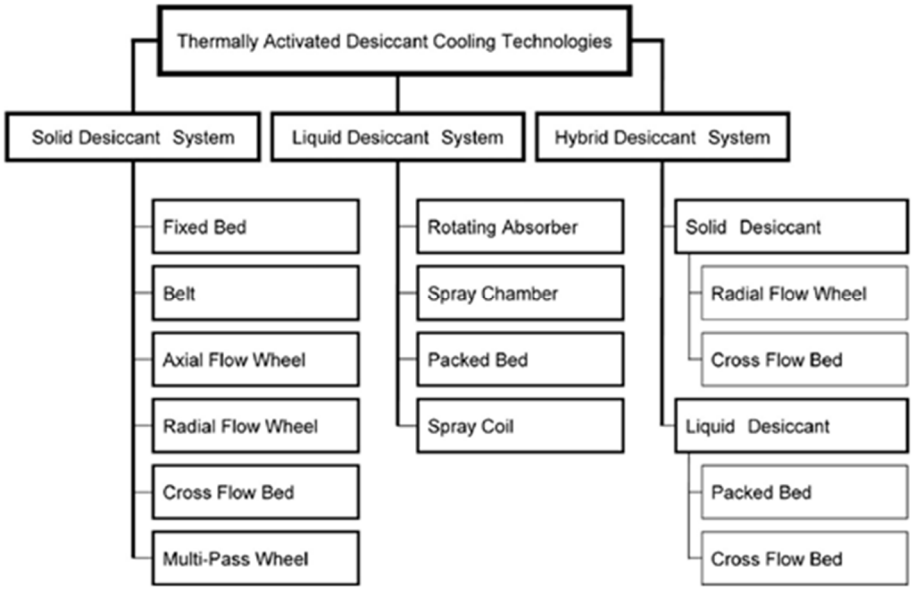


Figure 3.1: Heat and mass exchanger configurations for various desiccant cooling technologies [133].

Table 3.2: Comparison between liquid-desiccant cooling system and conventional air-conditioning system.

Parameter	Central air-conditioning system (CACS)	liquid-desiccant dehumidification system (LDDS)
Cost of operation	High	Save about 40%
Driving source of energy	Electricity, Natural gas, Vapor	Low grade energy e.g. solar energy, waste heat etc.
Humidity control	Average	Accurate
Quality of indoor air	Average	Good
System installment	Average	Slightly complicate

3.5 Liquid desiccant materials

A desiccant material attracts the water vapor towards itself and these materials are used where air of low dew point is needed. The strength of a liquid desiccant can be measured by its equilibrium vapor pressure, which is water vapor pressure that is in equilibrium with liquid desiccant material. The vapor pressure exponentially increases with the temperature of the water or desiccant and also increases as the water is absorbed by the desiccant, that is, equilibrium vapor pressure will be higher for a dilute liquid desiccant than a concentrated one [136].

Some other parameters which indicate desiccant materials performance are:

- Energy storage density
- Temperature for regeneration
- Boiling point elevation (BPE)
- Availability
- Cost

A good desiccant should have the following properties:

- Large saturation absorption capacity
- Low regeneration temperature
- Low Viscosity
- High heat transfer
- Non-volatile

- Non-corrosive
- Odorless
- Non-toxic
- Non-flammable
- Stable
- Inexpensive

Surface tension plays important role in static hold up and surface wetting of heat and mass exchanger of liquid desiccant system.

Commonly used liquid desiccants in the industrial dehumidifiers are glycols and solutions of halide salts which include lithium chloride (LiCl), lithium bromide (LiBr), calcium chloride ($CaCl_2$), triethylene glycol and mixture of salts etc. The selection of a desiccant material will have a direct effect on the desiccant dehumidifier design.

The earliest used liquid desiccant is triethylene glycol but its use is limited because it has high viscosity which causes system operation unstable because of liquid residence. They are low toxic and compatible with most of the metals but all glycols are volatile because of very low surface vapor pressure, some of it evaporates with air into the conditioned space which sometimes makes them unacceptable for air conditioning applications [137]. The dew point temperature of air by using a solution of triethylene glycol (96%) and water (4%) can be achieved with LiCl solution concentration of 42% by weight. But at equilibrium point, the molar concentration of the glycol in the air will be in the order of

1% that of the water vapor which means annual loss of triethylene glycol in air conditioning applications will be very high.

Salts of Halides such as LiCl and LiBr can dry air to 15% and 6% relative humidity, respectively but these salts are naturally corrosive. LiCl is a very good desiccant because at ambient conditions it will not vaporize and it has low viscosity which reduces the required pumping power but its mixing with the process air must be avoided. Halide salts are relatively expensive in nature [138].

Another alternate of low corrosive and non-volatile desiccants are salts of weak organic acids such as, potassium or sodium formate and acetate. Potassium acetate is less expensive it can dry air up to 25% relative humidity but its viscosity becomes very high. At a temperature of 27ºC and concentration of 70% by weight of potassium acetate has a viscosity of about 28 cp (0.028 Pa/s) but LiCL solution concentration of 43.l % by weight at the same temperature, has almost half of that viscosity.

Weak organic acid salts like sodium formate or potassium and acetate are less corrosive and less volatile as compared to halide salt. Formate salts are less viscous than acetate salts at concentrations with equivalent equilibrium relative humidity. Although potassium formate is a relatively weaker desiccant as compared to LiBr or LiCl but it has the ability to dry the air below 30% relative humidity and it can be a good alternative desiccant for many applications.

Zuber et al. [139] and Ahmed et al. [140] provided thermodynamic properties of single desiccants. Among different desiccants LiCl has lowest vapor pressure and found to be

most stable but this desiccant is very expensive as compared to other liquid desiccants. Calcium chloride ($CaCl_2$) is readily available and the cheapest desiccant. But its stability depends upon conditions of inlet air and concentration of desiccant in the solution which makes its use limited. Morillon et al. [141] measured the water vapor pressure above salt solutions such as, $CaCl_2$, LiCl, and LiBr.

Sun et al. [142] thermodynamically analyzed different liquid desiccants and calculated the vapor pressure of these desiccants. Thermodynamic properties of LiBr aqueous solution are provided by McNeely [143]. While an equation is developed by Kaita [144] for thermodynamic properties of aqueous LiBr solution at high temperatures. This equation is valid for temperatures of 40–210^{o}C and concentration of 40–65% by weight. de Lucas et al. [145] provided the properties of the water, LiBr and potassium acetate mixture and also for the mixture of water, LiBr and sodium lactate. Park et al. [146] made an attempt by adding four 8-C alcohol additives (2-octanol, n-octanol and 3-octanol) to liquid desiccant to lower its surface vapor pressure.

Ertas et al [147] presented the properties of LiCl and $CaCl_2$ mixture at a temperature range of 26.5 to 65.4^{o}C with a purity of 99.3% and 90%, respectively. The results of investigation showed that viscosity of the mixture is low and it is highly soluble. As compared to pure $CaCl_2$ solution the mixture has lower vapor pressure.

The term brine bulb temperature is used to represent the temperature which is achieved at the interface of air and liquid desiccant. This temperature depends on temperature and humidity of air as well as concentration of desiccant. A solution of air and lithium chloride

(43%) at a dry and wet bulb temperature of 30.0 and 25.6°C, respectively will have a brine bulb temperature of 47.8°C. [136].

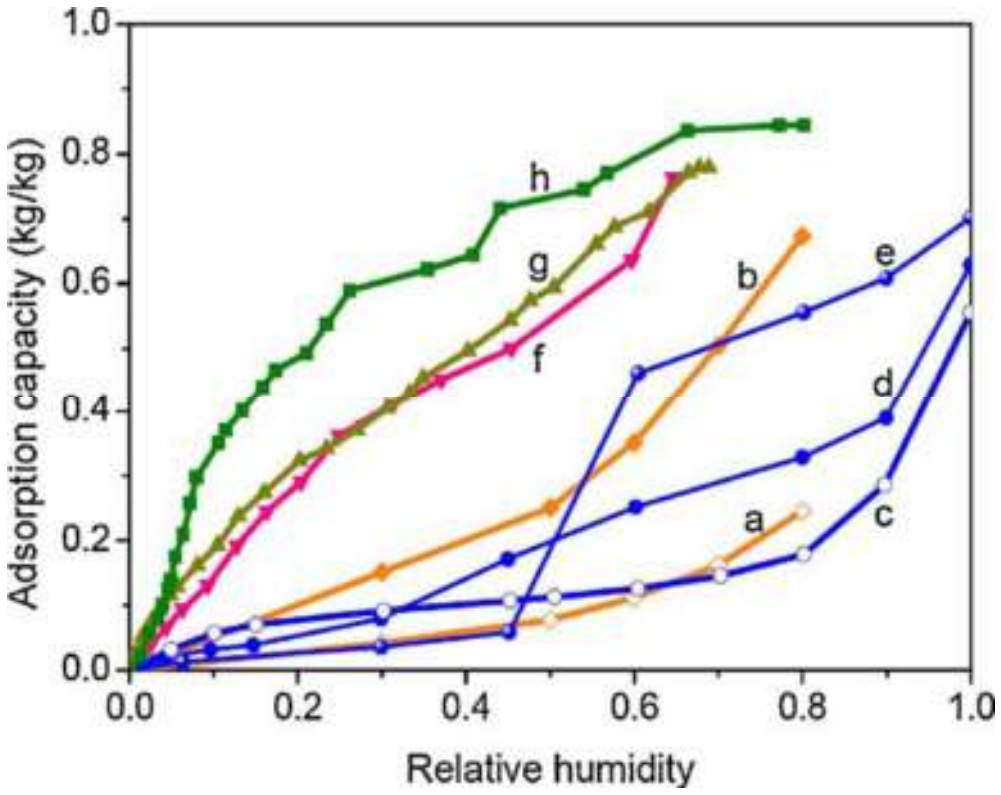
3.6 Composite desiccant materials

A number of composite desiccant materials have been developed in the past few years to improve their performance. Aristov et al. [148] have developed composite desiccant materials compromising of silica gel/SiO_2 and inorganic salt ($CaCl_2$, LiBr, $SrCl_2$, and $NaSO_4$). The results showed that these composite materials have lower desorption temperature. About 80% of adsorbed water could be desorbed from these composite desiccant materials at a temperature between 80 and 90^oC.

Jia et al. [149] have developed composite desiccant of silica gel impregnated with $CaCl_2$ and LiCl. After experimental results for regeneration temperature of 120^oC under humid and hot climates, moisture removal capacity of silica gel-calcium chloride wheel was found to be 20% better than the conventional silica-gel wheel. Adsorption capacity was improved by 67–145% by using composite of silica gel-LiCl instead of traditional silica gel and 13X molecular sieve under temperatures of 25^oC, 35^oC and 40^oC. The adsorption capacity of different composite materials are shown in Fig. 3.2 [150].

Liu and Wang [151] developed a composite material compromising of macro-porous silica gel and $CaCl_2$ (SiO2.x-xH2O.yCaCl2). They studied the impact of porous structure. The adsorption capacity for this composite at intake conditions of 25^oC and 40% relative humidity (RH) was found to be 0.4 g of water/g of dry adsorbent, which was 5.7, 2.1, 1.9, 6.8 times more than that of macro-porous silica gel, micro porous silica gel, synthetic

zeolite 13X and activated carbon, respectively. Moreover, this desiccant material has low regeneration temperature of about 60–80ºC.



(a) silica-gel (25ºC); (b) silica gel-Lithium chloride (25ºC); (c) sepiolite (23ºC); (d) sepiolite-carbont (steam activated) (23ºC); (e) sepiolite-carbon (chemically activated with KOH) (23ºC); (f) $CaCl_2$–SiO_2 sol–gel (25ºC); (g) $CaCl_2$-MCM-41 (20ºC); (h) silica gel-LiBr at 20ºC.

Figure 3.2: Different desiccant materials isotherms for water adsorption [150].

Gonzalez et al. [152] introduced a new desiccant which is a combination of sepiolite and activated carbon or sepiolite and $CaCl_2$. This new desiccant combines the fibrous and hydrophilic nature of sepiolite and hydrophobic nature of activated carbon. The combination of these properties makes sepiolite applicable for different applications. Adsorption capacity of sepiolite is enhanced by using a composite of sepiolite- $CaCl_2$.

A good and cost effective alternative to LiCl is $CaCl_2$ but it is a weak desiccant as compared to LiCl. $CaCl_2$ solution of 42% concentration by weight (which can be used without encountering crystallization) will dry air to about 35% RH while a 43% LiCl solution can dry air to a 15% RH. Ertas et al. [147] proposed a low cost composite desiccant as an alternative to high cost LiCl which is a mixture of LiCl and $CaCl_2$. The $CaCl_2$ has 20 times lower cost as compared to LiCl. They showed that 43% solution of the 50/50 mixture of these two desiccants will behave like 40% of pure LiCl. It may be noted that $CaCl_2$ solution of 43% concentration by weight will have the same properties as the solution of 34% LiCl.

The liquid desiccant must exhibit low equilibrium water vapor pressures at the available heat rejection temperature level to achieve low air dew point temperatures and thus a strong air dehumidification with comparably low driving temperatures [153]. From the various liquid desiccants some figures of merit can be developed for the selection of suitable desiccant. These figures of merits are listed in Table 3.3 [154]. The term heat of mixing or differential heat of solution represents the amount of heat liberated by absorption of water into a solution of desiccant at a fixed composition. A relatively low heat of mixing is desirable. Parasitic power losses represent the pumping power required for the circulation of desiccant solution.

Table 3.3: Weighing factors and figures of merit for desiccant selection [154].

Characteristics	Weighting factor	Figure of merit
Safety	1.0	Lethal dose (LD50)
Corrosion	0.8	Corrosion rate
Mass transfer potential	0.8	Equilibrium vapor pressure
Heat of Mixing	0.6	Energy/ kg of absorbed water
Cost of desiccant	0.5	Cost/ 100kg of solution
Heat transfer potential	0.5	Thermal conductivity
Parasitic power losses	0.3	Viscosity

3.7 Liquid desiccant dehumidifiers

The application of a desiccant dehumidification unit is for transfer of mass and heat between process air and the desiccant solution. For a high performance, liquid desiccant dehumidifiers should have the following characteristics [133]:

- High rates of mass and heat transfer.
- Low pressure drop of the process air stream while passing through the dehumidifier.
- Small liquid side resistance to moisture diffusion.
- Large surface contact area per unit volume.
- In order to avoid corrosion the dehumidifier should be made of a material compatible with the liquid desiccant. The material should be inexpensive too.
- There should be no liquid desiccant carryover with the process air.

Liquid desiccant dehumidifiers are classified as adiabatic and internally cooled [155]. The air directly contact with the desiccant solution in the adiabatic dehumidifiers while in internally cooled dehumidifiers apart from the contact between two streams, desiccant solution is cooled down using some cooling medium to increase the system performance.

3.8 Adiabatic dehumidifiers

One of the adiabatic dehumidifiers namely packed bed dehumidifier is the main focus of most of the researchers. Fumo and Goswami [156] modeled a packed bed dehumidifier for heat and mass transfer using packing of polypropylene having a volumetric surface area of 210 m^2/m^3 and experimentally measured its performance with LiCl solution. They concluded that because of the high surface tension, solution of LiCl did not wet the packing uniformly. An empirical formula for estimation of fraction of bed wetted surface area was developed and the performance results of the theoretical model were found in good agreement with the measured results.

Khoukhi et al [157] experimentally studied the twin rotor desiccant cooling system and Enteria et al. [158] made a parametric investigation for a modeled twin rotor desiccant cooling. Enteria et al. [159] made a comprehensive investigation of different thermally activated desiccant cooling systems and experimentally showed the system dependence upon climatic conditions and different components performance.

Liu et al. [160] used structured packing with a solution of LiCl for estimation of a cross-flow desiccant dehumidifier performance. They presented the experimental results in term of rate of moisture removal and the dehumidifier performance for various values of

desiccant flow rates, air flow rates, desiccant and air inlet temperature, desiccant inlet concentration, and specific humidity of inlet air.

Liu et al. [161] and Liu and Jiang [162] presented an analytical solution for the coupled heat and mass transfer for a packed-bed system by using the assumptions of minimal change in desiccant concentration and Lewis number of one. The results of analytical solutions are closely in agreement with exact numerical solutions and with experimental data.

Jain and Bansal [163] used empirical relations to predict the performance of packed bed dehumidifiers compromising of three different liquid desiccant materials ($CaCl_2$, LiCl, and triethylene glycol). It was found that, the value of dehumidifier effectiveness varies from 10% to 50% or more. The results also showed that, more comprehensive empirical models are required for better estimation of liquid desiccant dehumidifier performance. The performance of liquid desiccant dehumidifier can be improved by using a liquid-to-liquid heat exchanger between the absorber and regenerator as shown in Fig. 3.3.

Katejaneken et al. [164] experimentally studied a liquid desiccant (LiCl) ventilation preconditioning system. The results showed that system can reduce the relative humidity by about 11% while the temperature remains almost equal to the ambient air temperature. It has also been found that evaporation rate remains always greater than the absorbed moisture by the liquid dehumidifier which makes the system well feasible for continuous operation.

Lof [165] by using triethylene glycol as the desiccant material, suggested and experimentally tested the earliest liquid desiccant system and found the system well feasible for hot and humid climatic conditions. Patnaik et al. [166] developed a liquid desiccant dehumidification system which compromises of packed bed dehumidifier which utilizes aqueous LiBr to dry the air. The results showed that cooling capacities of about 3.5–14.0 kW can be achieved with such systems.

Nagaya [167] made an investigation for the desiccant dehumidifier performance for automobile air conditioner and the results obtained showed that this system is more efficient as compared to conventional cooling system. Nóbrega and Brum [168] investigated the desiccant cooling unit for equatorial and tropical climates of different cities and investigated the system performance for different operating parameters.

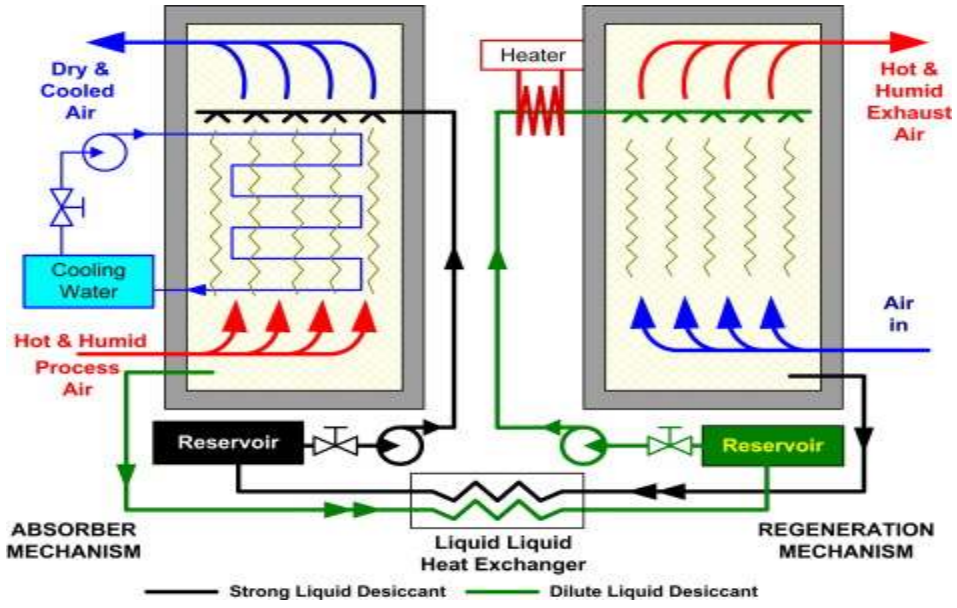


Figure 3.3: A liquid desiccant dehumidification system with liquid-liquid heat exchanger [163].

Babakhani [169] developed an analytical solution for simultaneous transfer of heat and mass in an adiabatic liquid desiccant dehumidifier. The results showed that, a better prediction of dehumidifier performance can be made by using Lewis number (Le) of 1.1 instead of 1. Similarly, Diaz [170] developed a transient heat and mass transfer model for a liquid desiccant dehumidifier. The liquid desiccant falls down the walls of dehumidifier as a thin film. The transient distribution of water vapor and instantaneous average Sherwood and Nusselt numbers along desiccant-air interface were studied.

Seenivasan et al. [171] investigated the effect of different parameters like desiccant concentration, desiccant solution temperature, desiccant and air flow rates, and relative humidity on the effectiveness of liquid desiccant dehumidifier using $CaCl_2$ as the desiccant material. The optimum values of these parameters were found for high effectiveness of dehumidifier under the following operating conditions: desiccant temperature 25^{o}C, concentration 40%, desiccant flow rate 2.25 kg/m^2.s, air flow rate 1 kg/m^2.s, and RH 85%.

Gommed et al. [172] experimentally measured the performance of packed-bed heat and mass exchangers using LiCl solution. The copper and polypropylene tubes were used as contact surfaces in the dehumidifier. The results showed that the copper tubes are more active to corrosion while in contact with the desiccant solution and on the other hand wetting is much difficult while using polypropylene tubes. Then the researchers employed adiabatic packed beds of 285 m^2/ m^3. The results showed that dripping of LiCl solution over the packing with suitable drop size to avoid carryover of the desiccant is better than spraying.

Gommed and Grossman [173] made a field test of 16 kW capacity liquid desiccant cooling system using LiCl solution flowing through the dehumidifier to absorb moisture from the air. The results showed the system can achieve the coefficient of performance of 0.8.

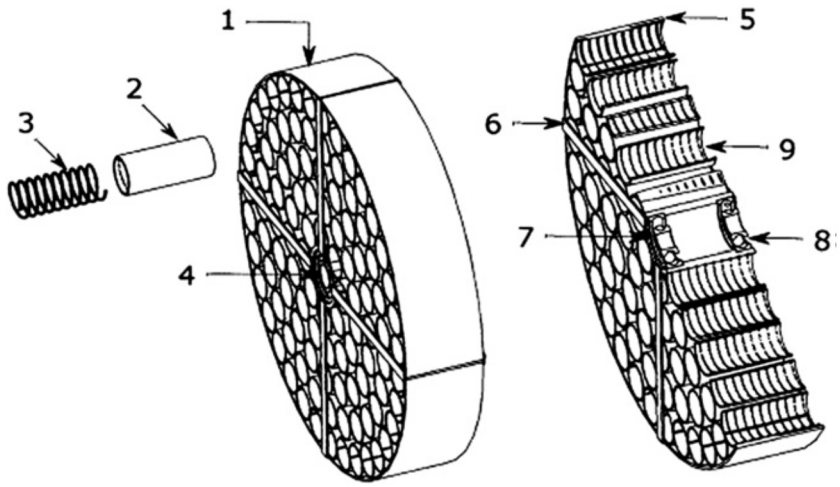
Chen et al. [174] used four cross flow packed bed dehumidifiers with 42-48% (by weight) LiBr solution to conduct a field test for liquid desiccant cooling system. The air flows in series through all the four beds. The flow of air and the desiccant was in counter flow so that the air can flow in direction of increasing desiccant concentration through the bed. Hot water at about 69°C and 73°C was used as regeneration sources for the desiccant. The obtained results showed that the unit had an average coefficient of performance of 1.5.

Kumar [175] developed new cycles for the improvement of the overall system performance and experimentally studied the influence of different parameters on dehumidifier and regenerator individual performance. To avoid the desiccant carry over with the air, mist eliminators (droplet filters) are utilized. These mist eliminators can decrease carryover of desiccant to parts per billion of process air flow, but air side pressure drop is increased due to these eliminators and more maintenance is required.

Saman and Alizadeh [176] tested a liquid desiccant air conditioner with evaporative cooler by using a plastic plates of 0.2 mm thickness and of 600 m^2 as contact surface area inside the dehumidifier. A solution of 40% $CaCl_2$ was used as the desiccant which was sprayed on the plates. For the value of maximum process air flow, the mass flow rates ratio of desiccant to air was 0.1. The results showed that with 0.3 kg/s of process air flow rate,

effectiveness (process air enthalpy change/ maximum theoretical limit) of 0.75 can be achieved.

Hamed et al. [177] proposed and experimentally tested a new liquid dehumidifier having a rotary shape of 50 cm diameter and thickness of 10 cm. The rotor of the wheel consists of multiple narrow slots as shown in Fig. 3.4. These narrow slots are filled with a thick cloth impregnated with LiCl solution. The experimental results showed that 95 g water per hour can be absorbed in the absorption cycle and at regeneration temperature of 85^{o}C, the absorbed and desorbed amount of water becomes equal.



1, Rotating desiccant bed; 2, spring with cloth layer; 3, spring; 4, rotating shaft hole; 5, desiccant bed cover; 6, ball bearing support; 7, ball bearing holder; 8, ball bearing; 9, slot.

Figure 3.4: Desiccant bed details [177].

So'Brien and Satcunanathan [178] proposed a liquid desiccant cooling system with simultaneous dehumidification of air and regeneration of desiccant solution without using any mechanical circulation as shown in Fig. 3.5. The proposed system consists of two chambers made of stainless steel. One chamber is dehumidifying chamber and the other

one is regenerating chamber. The chambers are interconnected by two passages. The upper passage is filled with porous material and lower one is unfilled. In order to avoid the mixing between the air flowing over the surface of the solution in the dehumidifying chamber and that in the regenerating chamber, the both chambers are filled with liquid desiccant to the level where the porous material is completely submerged. The COP of the system was found in the order of 0.3 even at lower regeneration temperature of 50^{o}C which makes the use of solar energy feasible for the continuous operation of the system.

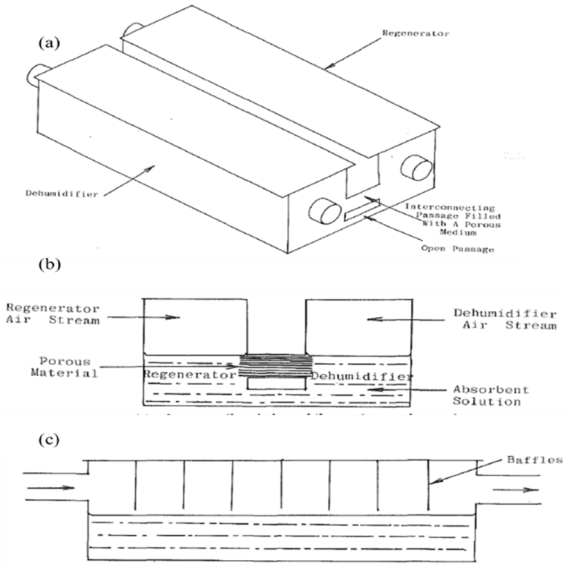


Figure 3.5: (a) Overall view of the system; (b) Cross-sectional view; (c) Side view [178].

Koronaki et al. [179] developed an analytical model for coupled heat and mass transfer performance of the liquid desiccant dehumidifier and whole cooling unit under Mediterranean climatic conditions and investigated the effect of ambient temperature and humidity ratio on thermal COP of the system, load coverage and dehumidification rate.

3.9 Dehumidifier with inner cooling

The adiabatic dehumidifiers are widely used for commercial as well as residential cooling applications because it allows large contacting area between air and desiccant with simple configuration. The heat and mass transfer efficiency of these dehumidifiers is very high but it causes a large pressure drop on process air side while flowing through the packing material. Another drawback of these dehumidifiers is the desiccant temperature increase during removal of moisture from the process air which decreases the dehumidifier performance and in turn air temperature and humidity cannot be controlled efficiently. Also, for complete wetting of packing material and to avoid desiccant solution temperature to rise, the flow rates should be high in packed bed dehumidifiers. The condition of complete wetting can be fulfilled at low desiccant flow rates by the addition of surfactants to the desiccant or by increasing surface energy of the packing by different treatments. However, again in order to keep the desiccant at low temperature, a high flow rate will always be required.

In order to avoid these problems and to remove heat generated inside the dehumidifier, dehumidifier with inner cooling is gaining attention, as shown in Fig. 3.6. These are good alternative in which high wetting of surface as well as lower temperature of desiccant solution can be achieved without high flow rates and desiccant droplets carryover as well as the higher pressure drop can be avoided [180].

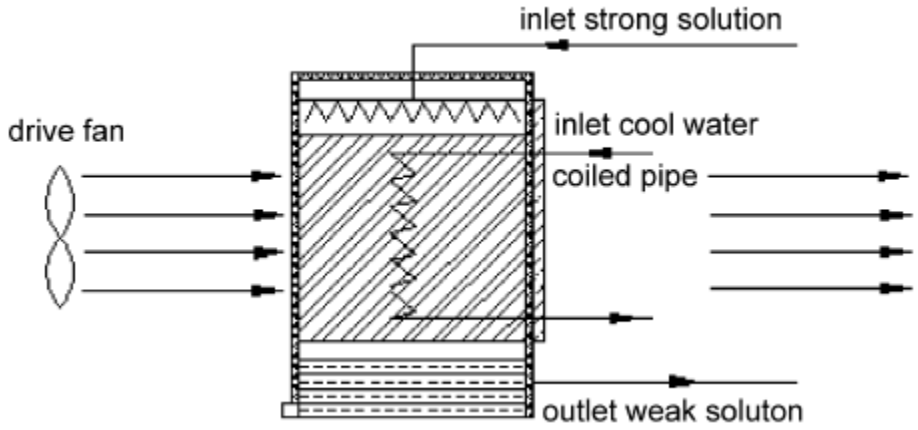


Figure 3.6: Schematic diagram of dehumidifier with inner cooling [180].

Scalabrin and Scaltriti [181] simulated a dehumidifier with internal cooling and heated regenerator of an open processed summer air conditioning system. At a low temperature of liquid desiccant more moisture can be absorbed because of low vapor pressure. Gommed [182] experimentally analyzed the performance of packed bed heat and mass transfer exchanger with a solution of LiCl. The dehumidifier (conditioner) and regenerator used were internally cooled units with contact surfaces made of copper tubes or polypropylene tube. But main drawback of using copper and polypropylene tubes is that copper can be easily corroded and wetting is very difficult in the case of polypropylene tubes.

During the dehumidification, the embedded cooling coils in the packing material removes heat from the dehumidifier and the outer insulation layer serves to prevent transfer of heat from outside air to the dehumidification unit. A cross flow dehumidifier unit, as shown in Fig. 3.6 [180], requires relatively large flow rate of desiccant solution for the better control of humidity. This configuration can be replaced by counter or parallel-flow.

The embedded cooling coils in the consecutive plates of the packing layers are another possible configuration for inner cooled dehumidifier to remove the heat during moisture removal process. In order to increase the desiccant and air contacting area corrugated plates can be used instead of consecutive plates. Yoon et al. [183] proposed dehumidifier with different channels with one for the flow of air and desiccant solution and the other for the flow of cooling water or cooling air, as shown in Fig. 3.7.

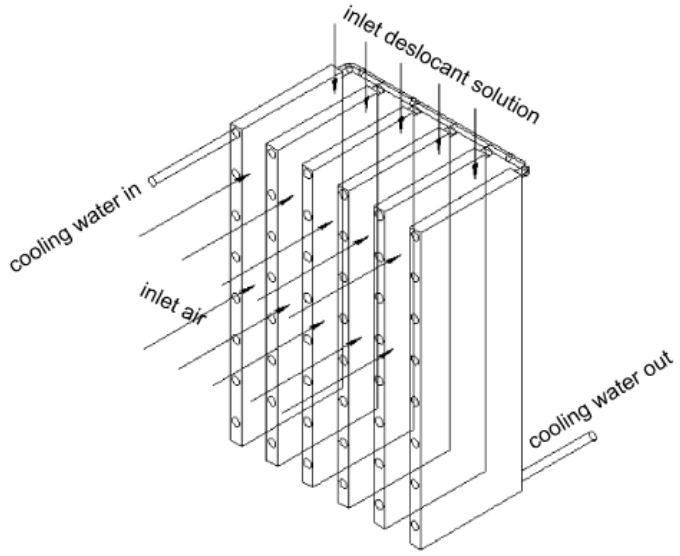


Figure 3.7: Spaced parallel plate packing material dehumidifier with inner cooling [183].

In another configuration for inner cooled dehumidifier, the packing material can be replaced by cooling coil which serves as mass and heat transfer contact area between desiccant and air. To increase the contacting area, cooling coils are usually of finned types. The horizontal and vertical distance between the tubes are important parameter for optimum design of mass and heat transfer. Because of the corrosive nature of desiccant solution, the material of the finned tube should be non-corrosive nature or it can be metal of high thermal conductivity with coating of anti-corrosion layer. Khan [184] developed

an internally cooled dehumidifier model as shown in Fig. 3.8 and numerically studied the effect of different parameters on the system performance.

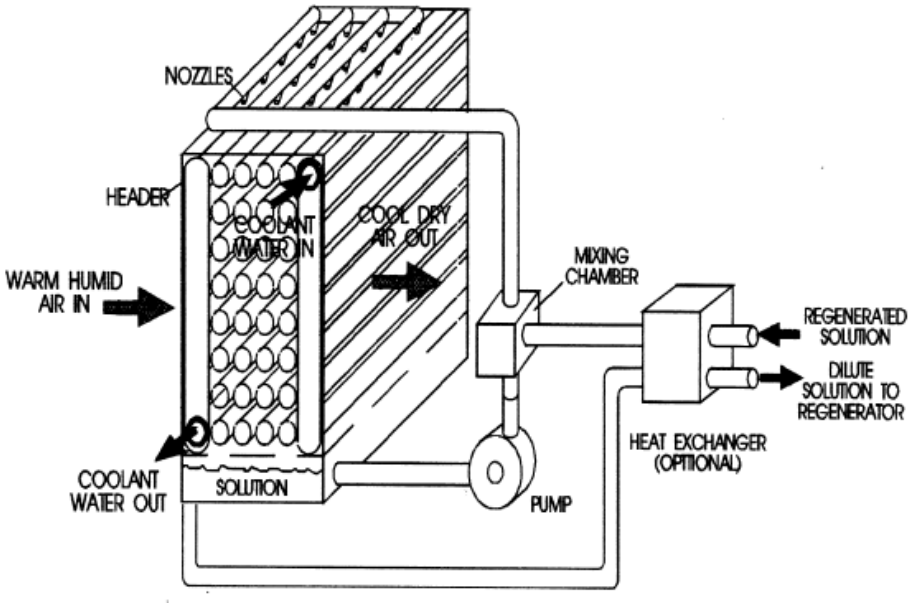


Figure 3.8: Inner cooled dehumidifier with finned tube [184].

The water-cooled banks of metal tubes are used in many industrial dehumidifiers as the contact area between the process air and the desiccant but this type of internally cooled dehumidifier can be expensive especially with halide salts. Saman and Alizadeh [185] developed and tested cross-flow model for the evaporative cooled conditioner. They performed heat and mass balance on the incremental control volume to calculate the liquid film thickness by using Nusselt theory for laminar falling films. The measured performance of the system was found to be within 15% of the numerically predicted value.

Saman and Alizadeh [176] presented a new dehumidifier which is a derivative of the dehumidifier with inner cooling which makes use of direct evaporative cooling unit. This configuration of the system is shown in Fig. 3.9. In this unit a stream of secondary air is

brought in contact with the spray of water in order to cool down the desiccant solution flowing in another channel.

The droplet carryover from liquid desiccant dehumidifiers in which halide salts are used is a dangerous and discouraging issue for air conditioning applications. This problem can be eliminated by using internally cooled dehumidifiers which can work at relatively very low flow rates. Laevemann et al. [186] and Lowenstein et al. [138] proposed desiccant dehumidifiers in which desiccant is directly delivered onto the conditioner cooled plates. The velocity of air at the gap is less than 4.5 m/s in this system which avoids the desiccant carry over with the air.

Packed-bed is most commonly used for the contact surface between air and desiccant solution and in some conditioners inner cooling is also utilized. The contact surface between the air and the desiccant is a bundle of tubes and the material of these tubes depends upon the nature of the desiccant material. For example, for halide salts copper tubes are used to limit the corrosion problems. The desiccant is usually sprayed or dripped onto these tubes and air is drawn past them. An internally cooled conditioner does not require high thermal capacitance of the desiccant to lower the desiccant temperature during absorption but for complete wetting of the packing, still high flow rate is required. Coil or tube-type dehumidifiers are rarely used with halide salts desiccants because of corrosion and high cost.

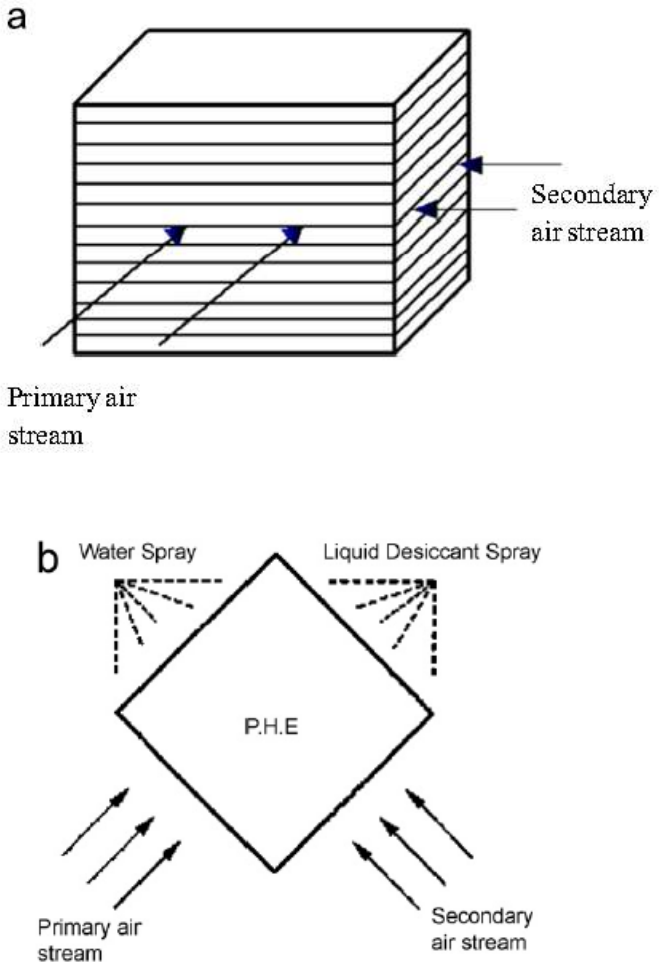


Figure 3.9: Inner cooled dehumidifier (cross flow): (a) directions of air flow (b) fluid sprays contacting with different air streams [176].

Mesquita et al. [187] developed three models of laminar flow and water-cooled liquid desiccant conditioner. Using published data with simplest model, mass and energy balances on a discrete control volume, heat and mass transfer coefficient between the air and the desiccant was calculated. For concentration gradient and the temperature across the desiccant and air films, the most sophisticated model was used in order to calculate the thickness of the film. Queiroz et al. [188] developed a model for the calculation of operating conditions for a non-adiabatic liquid desiccant dehumidifier. Triethylene glycol

and air flows counter currently outside the falling film staggered copper tubes. Water flows inside the tubes to increase the mass transfer potential.

Yin et al. [189] designed an internally cooled/heated dehumidifier/regenerator based on plate-fin heat exchanger (PFHE) as shown in Fig. 3.10. The schematic of PFHE is shown in Fig. 3.10(a). The designed unit consists of six PFHEs stacking up along the vertical direction as illustrated in Fig. 3.10(b). The effects of the air flow rate, cooling water temperature, and desiccant solution temperature on cooling efficiency and the performance of internally cooled dehumidifier were observed. The results showed that cooling efficiency decreases with the increase in cooling water temperature and at lower desiccant solution temperature, system has better dehumidification performance. It was also concluded that internally cooled dehumidifier has better thermal performance as compared to the adiabatic dehumidifier.

Khan and Martinez [190] studied the performance of internally cooled dehumidifier with LiCl as the desiccant. The parameters like humidity and enthalpy effectiveness were used to predict the thermal performance of the unit. The results showed that humidity and enthalpy effectiveness are greatly affected by number of mass transfer units.

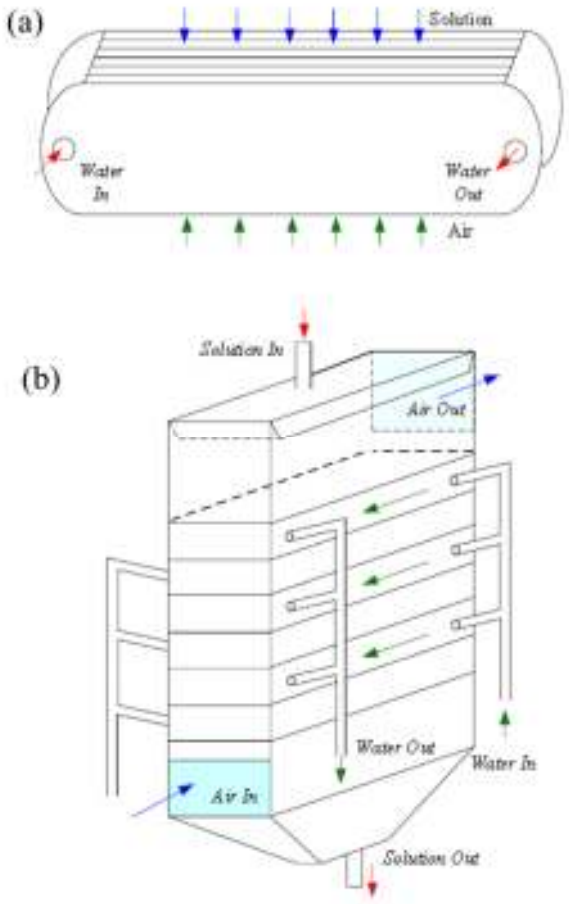


Figure 3.10: A new type of internally cooled/heated dehumidifier/regenerator [189].

Figure 3.11 shows an internally cooled/heated liquid desiccant dehumidification system [191]. The system uses LiCl as desiccant material and water as cooling fluid. It was concluded that temperature of the desiccant solution plays a major role in the system performance. Humidity of air, flow rate of air, solution concentration, and flow rate of solution also significantly influence the performance of the system.

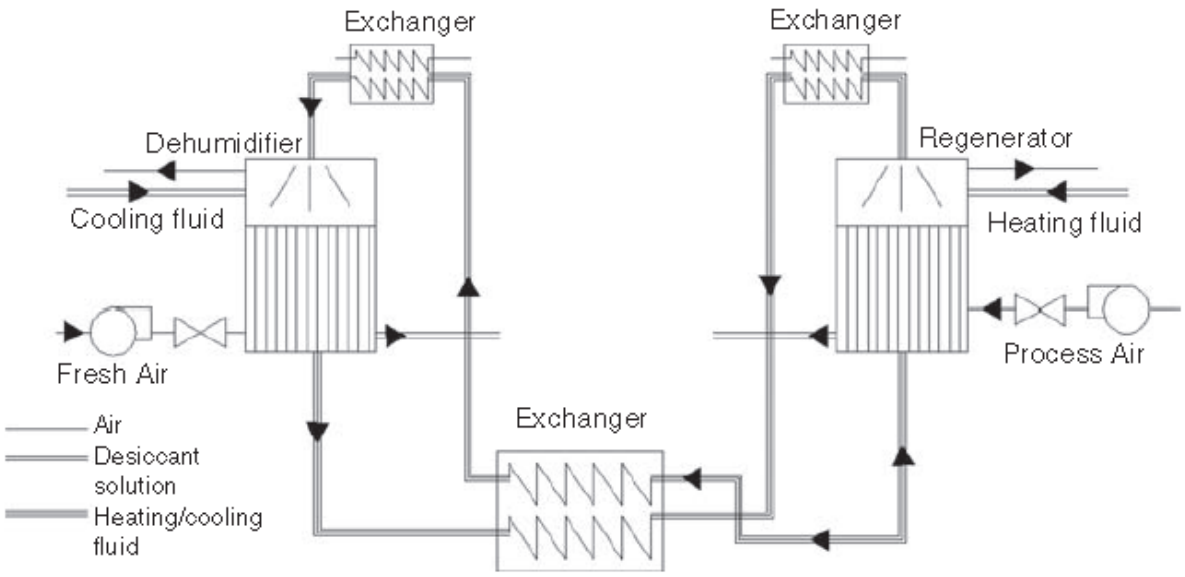


Figure 3.11: Schematic of internally cooled/heated liquid desiccant dehumidification [191].

Khan and Sulsona [192] presented an internally cooled desiccant dehumidifier in which solution of LiCl is uniformly sprayed over evaporator coil bundle as shown in Fig. 3.12. The desiccant and air flows in counter flow arrangement. The cooling and dehumidification of outside air is achieved by direct contact of air with the desiccant solution on the coil surface. The weak desiccant solution is collected in a sump underneath the tube bundle. The effect of desiccant temperature, process air temperature on the system performance was analyzed in this study.

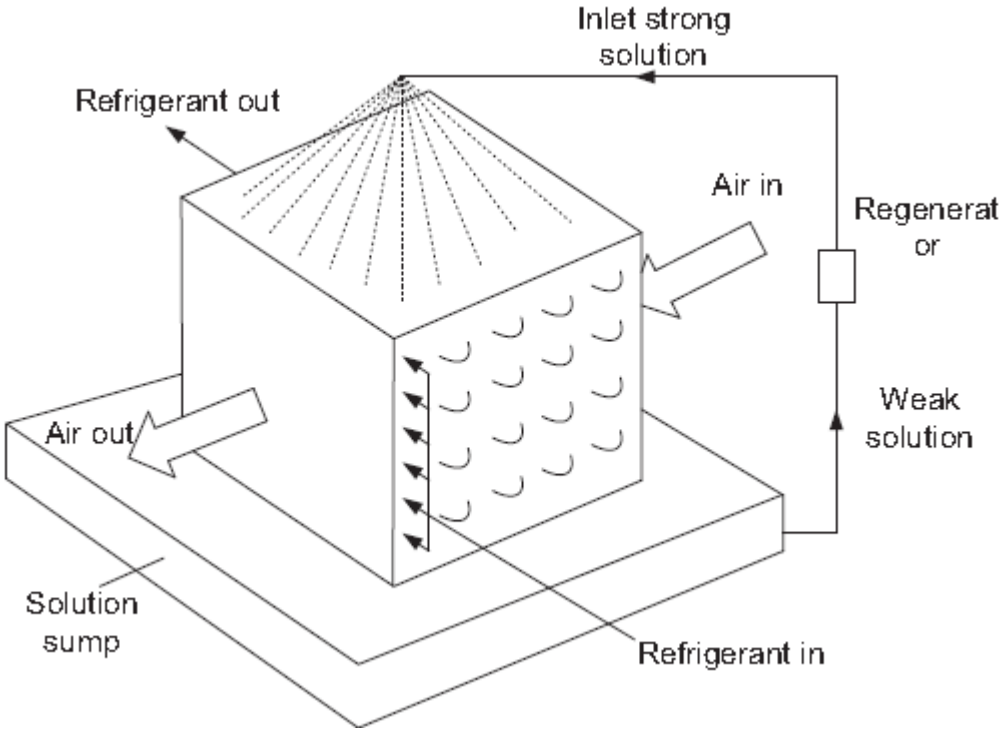


Figure 3.12: Schematic of spray type internally cooled dehumidifier [192].

3.10 Developments of liquid desiccant dehumidifier

The desiccant solution must be cooled before it is pumped to the desiccant dehumidifier but temperature increase of liquid desiccant solution as it absorbs moisture will reduce the heat and mass transfer potential between air and desiccant solution. This problem may leads to the lower overall performance of the single stage dehumidification system. To overcome this adverse temperature increase, the idea of multistage liquid desiccant dehumidifier was presented by Jiang et al. [135]. In this system, several desiccant dehumidifiers are connected in series as shown in Fig. 3.13. The desiccant solution is cooled in every stage of the dehumidifier separately, to increase the overall system performance. On the other hand, instead of cooling/heating the solution internally in each

stage of dehumidifier, the system performance can be improved by cooling/heating the desiccant solution using some external auxiliary cooling/heating medium as shown in Fig. 3.14 [193]. Li [194] studied and compared irreversible losses for multistage dehumidifier and conventional single stage dehumidifier.

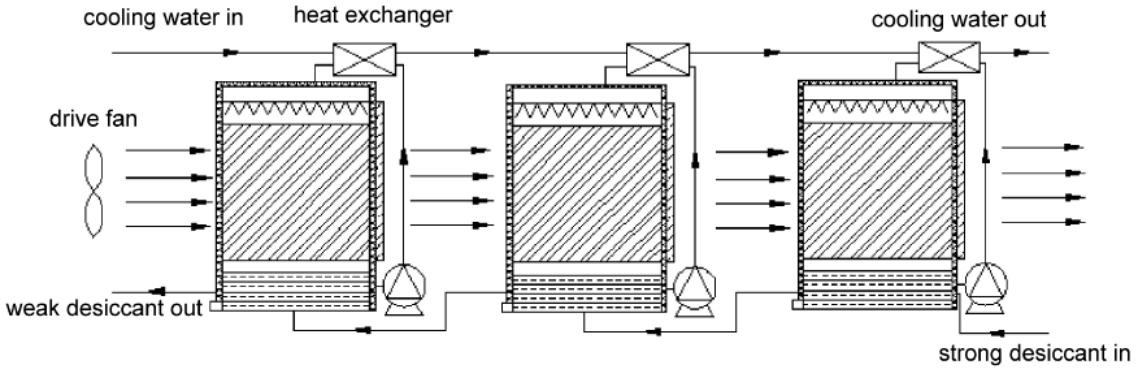


Figure 3.13: Multi-stage liquid desiccant dehumidifier [135].

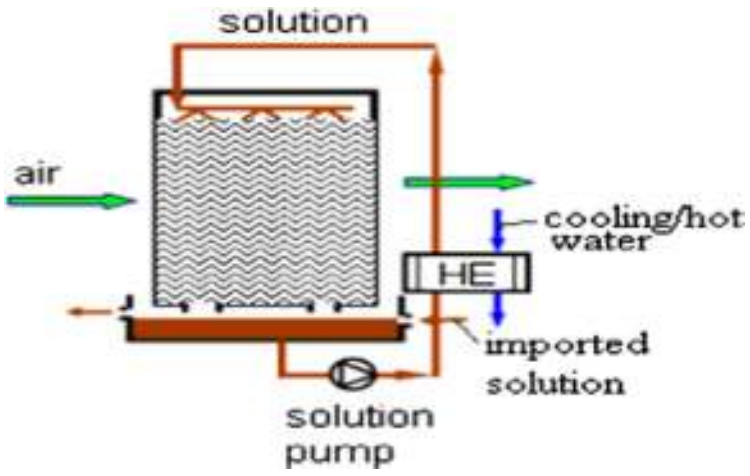


Figure 3.14: Dehumidifier/regenerator with auxiliary cooling/heating module [193].

Xiong et al. [195] observed an increase in the system COP from 0.23 to 0.72 by using a two-stage novel liquid desiccant cooling system which is shown in Fig. 3.15. On the process side, the air is first dehumidified in the first dehumidifier (1st DEH) using solution

of $CaCl_2$ and then, in the second dehumidifier (2^{nd} DEH) air is further dehumidified by solution of LiCl up to the desired humidity of air. Kumar et al. [196] carried out the performance analysis of falling film liquid desiccant dehumidifier operating on two new liquid desiccant cycles (Multi-absorber cycles) and found that performance of the system is improved significantly by using these cycles.

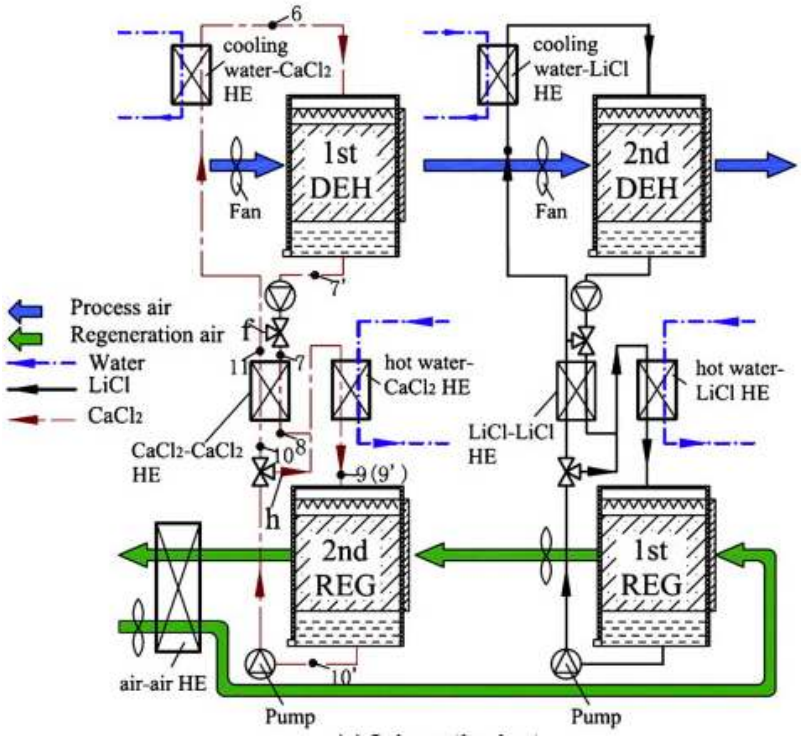


Figure 3.15: Two-stage liquid desiccant dehumidification system [195].

Xiong et al. [197] studied a two stage liquid desiccant dehumidification system using two desiccant materials (LiCl and $CaCl_2$). In this system, the air is pre-dehumidified in the first stage using $CaCl_2$ and then further dehumidified using LiCl. The results showed that investment cost can be reduced by about 53% as compared to single stage dehumidification using LiCl because $CaCl_2$ is much cheaper than LiCl.

The parallel-plates membrane modules can be used as air to air heat and mass exchanger [198]. Figure 3.16 [199] shows parallel-plates membrane module used as liquid desiccant dehumidifier. The flow channels were created by keeping the equal spacing between the membrane modules. The desiccant solution and air flows through the flow channels in counter-flow arrangement. The desiccant solution absorbs the moisture from the air through the membrane, eliminating the problem of desiccant carry over.

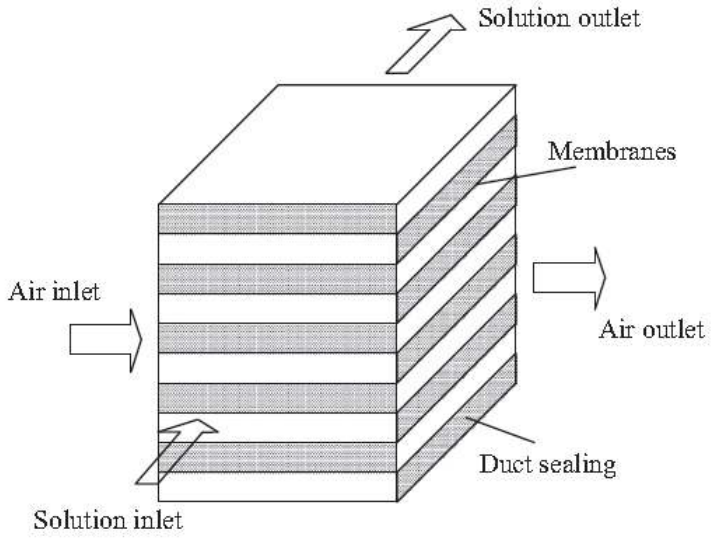


Figure 3.16: Structure of a parallel-plate membrane module [199].

A flat plate internally cooled dehumidifier using water as the cooling fluid is shown in Fig. 3.17 [200]. Cooling water is circulated through the polypropylene double plates. The air flows upward and desiccant solution flows downward across the polypropylene plates. For uniform flow of the desiccant solution, a desiccant distributer is provided on top of each plate. The corrugated plate internally cooled dehumidifier is shown in Fig. 3.18 [201]. Results showed that corrugated plate dehumidifier has better performance as compared to

the flat plate dehumidifier (shown in Fig. 3.9) because of better cooling performance and air turbulence.

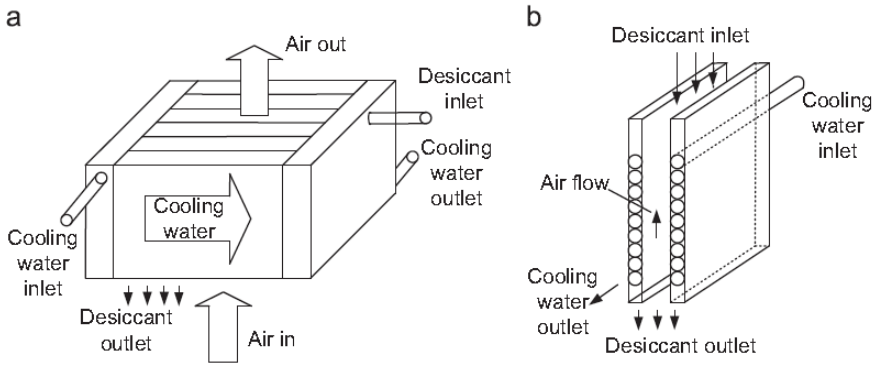


Figure 3.17: Internally cooled vertical flat plate dehumidifier [95]. (a) Side view; (b) Inner view.

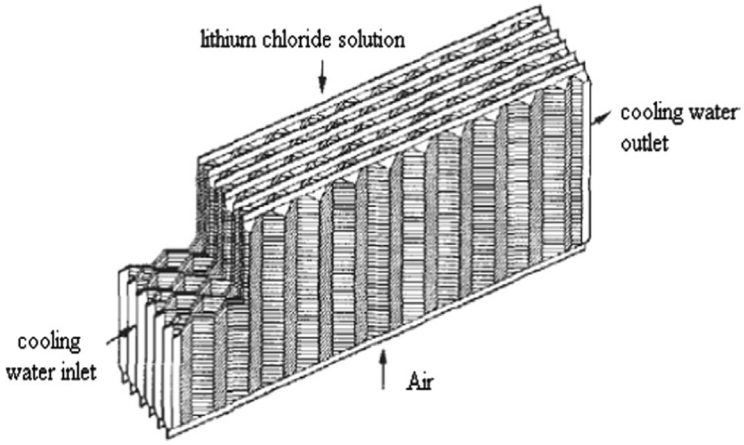


Figure 3.18: Internally cooled dehumidifier with corrugated plates [201].

Cheng [202] developed a liquid desiccant dehumidifier using fin-plastic tube heat exchanger. The developed system has great potential for corrosion prevention and can be used as internally cooled/heated dehumidifier/regenerator. Considering the advantages of multistage liquid desiccant dehumidifier, the membrane based liquid desiccant dehumidifier can also be extended to multistage configuration, where packed bed columns are replaced by membrane modules as shown in Fig. 3.19 [193].

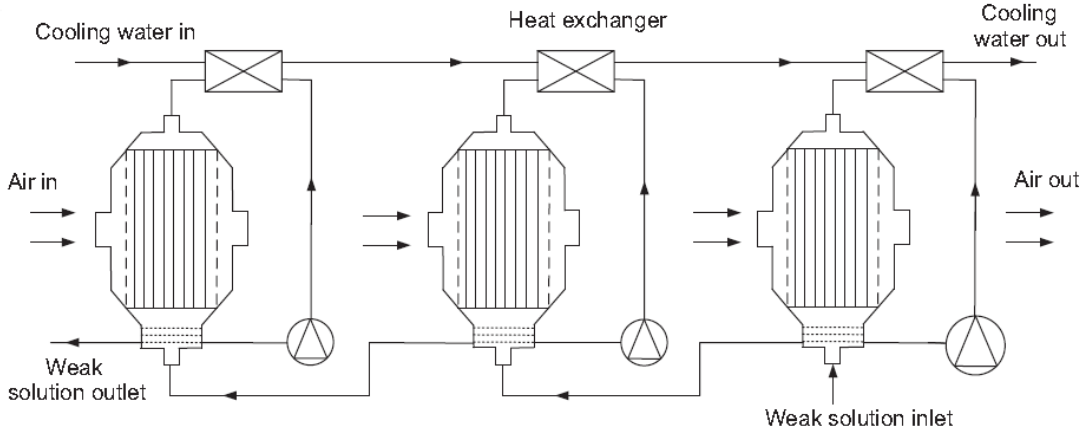


Figure 3.19: Schematic diagram of hollow fiber membrane module type multistage dehumidifier [193].

3.11 Dehumidifier packing material selection

Packing materials are used to provide contact surface for desiccant and process air interaction in the liquid desiccant dehumidifier. The selection of this packing (material as well as arrangement) affects the performance of the desiccant dehumidifier. These materials can be categorized as random and structured packing materials. The random packing materials like rosette ring, Pall ring, ladder ring, etc., are those materials which are randomly placed in the packing film and do not have a regular geometric forms. While, structured packing materials have a fixed geometric form.

While choosing a packing material the pressure drop during irrigation is one of the main criteria for liquid desiccant cooling systems because more pressure drop will require the higher fan power for air. Gandhidasan [203] developed the model for random as well as for structured packing to predict the irrigated pressure drop. The results of this study

showed that the structured packing is better in term of pressure drop as compared to the random packing for liquid desiccant cooling systems.

To predict the performances of both packing materials comparative experiments were conducted on random as well as structured packing materials by Chung et al. [204]. Bravo et al. [205] concluded that structured packing has higher efficiency and more capacity for mass and heat transfer materials with lower pressure drop.

The factors such as, volumetric area of packing material, void volume, and spacing intervals between the layers of packing materials are used to evaluate the contacting surface area between air and the desiccant material. The resistance to air flow can be measured by void ratio, which relates inversely with the spacing interval. An optimum value of spacing interval between layers of packing will cause less inlet air resistance and also increase the coverage ratio of the sprinkled desiccant on the packing material. Normally the spacing interval lies between 6–8 mm.

Another important parameter to predict the heat and mass transfer within the packing material is the droplets diameter. Al-Farayadhi et al. [206] defined an equivalent diameter of structured packing materials by taking the arithmetic average of hydraulic diameter of different flow sections using flow channel of different cross sections.

The wetting ratio or effective interfacial area of packing materials is another essential factor to predict the packing material heat and mass transfer performance. The wetting ratio mainly depends on cross sections, flow channels, and surface characteristics of the desiccant material. A correlation for mass transfer coefficient was developed by Shi and

Mersmann [207] on the bases of the packing column principles of effective mass transfer. Gandhidasan [208] compared the three available models used to calculate the effective interfacial area of packing materials.

Height, length and quality of packing are also very important along with the selections of packing materials. The quality of packing means there should be no distortion if the packing is socked in a liquid desiccant for a long period of time and the layers of the packing material should be strong enough to withstand the required inlet air velocity.

3.12 Flow pattern inside the dehumidifier

Generally parallel, counter and cross flow configurations are used for the liquid desiccant dehumidifiers. For the analysis of dehumidification process finite difference model [209-211], ε -NTU model [212,213], and fitted algebraic equations model [214,215], are commonly used. From these three models the most accurate performance analysis with basic equations is provided by the finite difference model.

Rahamah et al. [216] theoretically analyzed the parallel flow desiccant dehumidification system using control volume approach. The results of the analysis showed that better dehumidification and cooling can be obtained by using low air flow rate (mass transfer will be increased because of the more air-desiccant contacting time) and large channel height (air-desiccant contact area will be enlarged).

Rahamah et al. [217] made an investigation of different performance controlling parameters of liquid desiccant system for a fin-tube channel with a parallel flow configuration. Elsayed et al. [218] numerically analyzed the characteristics for transfer of

mass and heat between air and solution of $CaCl_2$ using counter flow configuration in a packed bed dehumidifier and developed a chart to predict the exit air temperature and humidity.

Lazzarin [219] developed a heat and mass transfer model for packed bed liquid desiccant dehumidifier and carried out a parametric study using solution of H_2O/LiBr and $H_2O/CaCl_2$, to determine the optimum operating conditions for the packed bed dehumidifier. Dai and Zhang [220] developed the mathematical model for liquid desiccant dehumidification system in a cross flow configuration by employing packing material of honey comb paper to determine the Sherwood and Nusselt numbers at the interface of air and liquid desiccant.

Ali et al. [221] studied the effects on transfer of heat and mass between streams of air and liquid desiccant in a cross flow dehumidifier by adding Cu-ultra fine particles. It was observed that Cu-ultra fine particles addition stabilize the desiccant solution and increases the transfer rate of heat and mass between two streams.

Liu et al. [222] investigated a liquid desiccant dehumidifier consists of honeycomb paper packing in a cross flow arrangement and provided the distribution of temperature and humidity of air within the packing materials. The desiccant solution concentration in the layers of packing material was also analyzed by them. For the internally cooled dehumidifiers, water, air, ammonia, and R22 [223] are commonly used as the cooling fluid with cooling fluid-desiccant flow pattern either cross [224] or counter [190]. The air-

desiccant flow pattern in adiabatic dehumidifier can either be counter [225, 226] or cross. Some of the presented models are summarized in Table 3.4.

Table 3.4: Details of Adiabatic/internally cooled dehumidifiers.

Model	Desiccant	Adiabatic/internally cooled	Air-desiccant flow pattern	Desiccant-cooling fluid flow pattern	Cooling fluid
Mesquita et al. [187]	$CaCl_2$	Internally cooled	Cross	Counter	Water
Diaz [170]	-	Internally cooled	Counter	-	-
Khan et al. [190]	LiCl	Internally cooled	Concurrent	Counter	Water and air
Saman et al. [185]	$CaCl_2$	Internally cooled	Counter	Cross	Water and air
Liu et al. [200]	LiCl	Internally cooled	Six configurations		Water
Khan et al. [213]	LiCl	Internally cooled	Cross	Cross	Ammonia
Park et al. [223]	Triethylene glycol	Internally cooled	Cross	Cross	R22
Hueffed et al. [224]	LiCl	Internally cooled	Cross	-	-
Khan et al. [214]	-	Adiabatic	Counter	-	-
Liu et al. [225]	LiBr	Adiabatic	Counter or cross	-	-
Oberg and Goswami [211]	Triethylene glycol	Adiabatic	Counter	-	-
Fumo et al. [156]	LiCl	Adiabatic	Counter	-	-
Liu et al. [226]	LiBr	Adiabatic	Counter	-	-

3.13 Conclusions

Liquid desiccant cooling system could help to solve the major issues now facing in heating, ventilating and air conditioning (HVAC) industry such as, peak electrical load demand created by conventional air conditioning systems, high indoor humidity and poor indoor air quality. The liquid desiccant materials and dehumidifiers discussed in this chapter can be summarized as:

- The liquid desiccant should be non-corrosive and non-volatile for a HVAC application. The most commonly used industrial liquid desiccants are LiCl, LiBr, $CaCl_2$, and triethylene glycol.
- LiCl has lowest vapor pressure and is most stable liquid desiccant but have high cost. The economic issue of LiCl can be resolved by using it in combination with $CaCl_2$, which is readily available and the cheapest desiccant.
- To improve the characteristics of a single desiccant material such as to improve its absorption capability and to lower its regeneration temperature, a number of composite desiccant materials have been developed.
- Desiccant dehumidifier is the heart of a liquid desiccant cooling system which is mostly of adiabatic type or inner cooled. The adiabatic type dehumidifier requires high flow rates for complete wetting of the surface which causes desiccant carry over. This problem of required high flow rates and carry over can be avoided by using inner cooled dehumidifier.
- A rotary type of liquid desiccant dehumidifier is proposed for lower pressure drop and to avoid desiccant carry over in which a porous medium such as, thick cloth can be used to carry the desiccant solution inside a rotor disc.
- The corrosive nature of a desiccant can be dealt with very low flow rate of the desiccant both in the dehumidifier and the regenerator. To operate the system at these low flow rates continuous cooling of the desiccant in the dehumidifier is required which have been done by many researchers by proposing different configurations of inner cooled dehumidifiers.

- To overcome the problem of droplet carryover a new type of liquid-air membrane heat and mass exchanger in counter flow arrangement as dehumidifier has been proposed and they provide the better performance. Further, multistage membrane based liquid desiccant dehumidifier would be the future development of liquid desiccant dehumidification systems.
- The selection of packing material, its arrangement and flow pattern inside the desiccant dehumidifier are important for its performance.
- More field work is required to design and specify liquid desiccant cooling systems which can comfortably replace the conventional air conditioners.

3.14 Objectives of the present work

Some primary objectives of the present study are:

- Literature review of desiccant cooling cycles.
- Literature review of desiccant based evaporative cooling systems.
- Literature review of liquid desiccant materials.
- Feasibility analysis of solid desiccant cooling system in conjunction with evaporative coolers under climatic zone of Dhahran, Saudi Arabia.
- Modeling and performance investigation of a rotary liquid desiccant dehumidifier.
- Design and manufacturing of a compact liquid desiccant dehumidifier with electric heater as regeneration source.
- Testing the manufactured dehumidifier.

CHAPTER 4
MATHEMATICAL MODELING OF LIQUID DESICCANT WHEEL

The liquid desiccant cooling system is found to be a good alternative of conventional air conditioning system for better control of both latent and sensible loads. The major component of a liquid desiccant cooling system is desiccant dehumidifier which controls the latent cooling load. In this chapter a mathematical model for a newly proposed rotary type liquid desiccant dehumidifier has been presented. The dehumidifier has a cylindrical shape with a number of identical narrow circular slots, distributed uniformly over the rotor cross section. This type of dehumidifier is commonly known as desiccant wheel. The energy and exergetic performance of the dehumidification system is discussed in this chapter for different operating conditions. The wheel performance curves which help to optimize the performance of the dehumidifier are drawn for a wide range of wheel rotational speeds (0.5 – 7 rpm), wheel width (0.1 – 0.6 m), air mass flux (1 – 8 $kg/m^2.s$), and regeneration temperature (50 – 90^oC). Furthermore, a feasibility analysis of the proposed dehumidification system has been carried out for climatic conditions of Dhahran, Saudi Arabia. The computed results show that better supply air conditions can be obtained to provide human comfort in the hot and humid climate with effectiveness of the system largely dependent on air flow rate, wheel width and humidity ratio of process air.

4.1 Introduction

Air conditioning (AC) load includes sensible and latent loads. Although conventional air conditioning units are effective for removing sensible cooling loads but not much efficient to remove the building latent loads. To control the indoor air humidity using conventional system the moisture from the air is removed by the process of condensation. In the process of condensation, cooling coil temperature should be lower than the supply or process air dew point temperature. This decreased temperature can wet the surfaces of cooling coil which provide the environment for the growth of bacteria and molds which directly affect the health and indoor air quality (IAQ) [227]. Another drawback of the conventional AC systems is that, air is usually overcooled to remove the moisture and this air is reheated again before supplying to the conditioned space. This process of overcooling and reheating reduces the system coefficient of performance (COP). The large consumption of power by conventional air conditioning system arises the need of low energy consuming devices which can handle latent loads effectively.

Some alternative low energy consuming cooling systems which directly utilizes the thermal energy and also reduces the emission of greenhouse gases are desiccant and absorption cooling. Table 4.1 summarizes the technical overview of these two thermally driven cooling techniques [228]. Among these systems, desiccant cooling is the main focus of many researcher for past few years. System with rotary dehumidifier commonly known as desiccant wheel, are most widely used and studied. Although, absorption chillers are used in European market as solar cooling systems but trend is now changing towards desiccant cooling technology [229]. Different advancements have been made to the basic desiccant

cooling cycle to improve the system efficiency. The basic advantages of desiccant cooling can be summarized as:

- The energy cost is significantly reduced.
- Free, low grade thermal energy sources such as solar, waste heat, etc. can be used to provide regeneration heat.
- Environmental friendly because of no emission of greenhouse gases since no chlorofluorocarbon based refrigerant is used.
- The sensible and latent cooling loads are independently and efficiently controlled.

The desiccant cooling system can either be solid or liquid depending upon the type of desiccant material used. Liquid desiccants have advantage over the solid desiccants that these only requires low temperature heat source (60–80^{o}C) to drive the system. This makes the use of some renewable energy resources like solar, biomass, etc. more feasible and efficient. A comparison between liquid and solid desiccant cooling system have been presented in Table 4.2 [230]. Liquid desiccant cooling system (LDCS) compromises of a liquid dehumidification cycle accompanied by cooling process that is, evaporative cooling. The dehumidification cycle includes the dehumidification of process air and regeneration of liquid desiccant and this cycle controls most of the building latent cooling load. The analysis of liquid desiccant cooling processes help to understand the drawbacks and advantages of the system.

Table 4.1: Overview of thermal cooling techniques [228].

Method	Closed cycle		Open cycle	
Type of refrigerant cycle	Closed refrigerant cycle		Refrigerant is in direct contact with air	
Principle	Chilled water		Dehumidification of air and evaporative cooling	
Phase of sorbent	Solid	Liquid	Solid	Liquid
Typical material pair	Water - silica gel	Water - lithium bromide, ammonia- water	Water - silica gel / Water - lithium chloride	Water - calcium chloride, Water - lithium chloride
Available technology	Adsorption chiller	Absorption chiller	Solid desiccant cooling	Liquid desiccant cooling
Typical coefficient of performance	0.5 - 0.7	0.6 - 0.75 (single effect)	0.5 - > 1	> 1
Typical driving temperature	65 - 90 °C	80 - 110°C	80 - 120°C	60 - 80°C

Table 4.2: Liquid vs. solid desiccant air conditioning [230].

Factor	Liquid desiccant	Solid desiccant
Dehumidification	Highly-effective method of dehumidification	Highly-effective method of dehumidification
Humidity control	Highly precise response	Less precise response
Energy	Less energy required than solid desiccant	High energy consumption for regeneration and post cooling
Performance degradation	No	Over time, some performance degradation occurs
Indoor air quality	Naturally disinfects and cleans air	No air cleaning benefits.
Maintenance	Minimal maintenance and infrequent replacement of parts	High maintenance cost

Most of the studies carried out for LDCSs are direct contact type in which there is a direct contact between liquid desiccant and air streams. Direct contact type system has a drawback that supply air can carry some liquid desiccant droplets with it, which can cause corrosion of ducting and other equipment. This will increase the maintenance costs and will decrease the equipment life. Secondly, desiccant carryover can affect the indoor air quality which will directly affect the human health within the conditioned space. Direct contact type system also requires high fan and pumping power because of the higher pressure drop and for continuous supply of desiccant solution. To overcome this problem, a porous medium can be used to carry the liquid desiccant solution in a rotary wheel. These wheels have lower pressure drops as compared to packed bed dehumidifiers and at the same time providing large surface area for contact between desiccant and humid air. In the past, most of the investigations had been carried out for solid desiccant rotary dehumidifier and very few work have been carried out on the rotary liquid desiccant dehumidifier [231]. Some general advantages of using rotary desiccant wheel are:

- These systems are more convenient to install.
- Dehumidification and regeneration are carried out synchronously in these systems and the system runs continuously.
- These systems occupy a smaller space but offering more contact surface area with air than the packed bed dehumidifiers.
- Dehumidification/regeneration capacity of the dehumidifier can be efficiently controlled by changing the rotational speed of the desiccant wheel.

The main objective of this research is to provide the research community with practical and reliable liquid desiccant rotary dehumidification system. As desiccant dehumidifier is the most crucial component of the desiccant cooling system, mathematical modeling of desiccant dehumidifier has a significant effect on overall system performance. The wheel dimensions and speed, and operating parameters such as ambient conditions, regeneration temperature, and air flow rate had already been optimized for better performance of the solid desiccant system. However, effective work needs to be conducted for absorption type rotary dehumidifiers.

In this chapter a mathematical model is developed to study and discuss the performance of a newly proposed rotary type liquid desiccant dehumidifier in terms of its dehumidification performance, effectiveness, and sensible energy ratio considering different parameters such as the air flow rates, regeneration temperature, and wheel speed. Also, the effect of ambient conditions on dehumidifier performance has been examined. Furthermore, a feasibility analysis of the proposed system have been carried out for the climatic conditions of Dhahran, Saudi Arabia keeping in view of a vast amount of energy is used for air conditioning in the Kingdom.

4.2 Mathematical modeling

4.2.1 System description

The proposed liquid desiccant dehumidifier is a rotary wheel of radius R and width L. The rotor consists of number of identical narrow slots uniformly distributed over its cross-section as shown in Fig. 4.1. The slots are covered with porous media impregnated with

solution of liquid desiccant. The wheel has separate sections for the flow of process air and regeneration air. These two streams of air flows in counter arrangement.

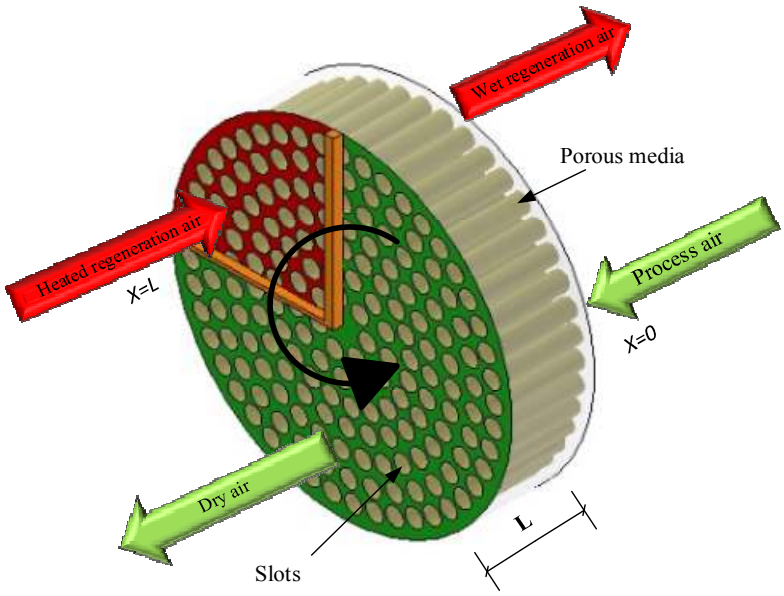


Figure 4.1: Schematic of rotary liquid desiccant dehumidifier.

4.2.2 Modeling approach

For modeling purpose, one slot is divided into *N* number of nodes (i=1, 2, 3…N) as illustrated in Fig. 4.2. One dimensional mass and energy balance equations are developed for air stream and desiccant surface, to determine the outlet conditions of air with the following assumptions:

- The flow of air through each slot is considered as plug flow in which model variations exists only in axial direction.

- The air is assumed to be distributed uniformly across dehumidification and regeneration sections.
- The axial heat conduction and mass diffusion are neglected.
- It is assumed that there is no carry-over of the desiccant solution.
- The thermodynamic properties are assumed as constant.
- Each slot is assumed to be adiabatic.

The process air directly contact with the desiccant surface. Water vapor are absorbed by the desiccant from the process humid air because of the higher partial pressure of water vapor in air as compared to the surface of desiccant. The released vaporization heat causes an increase in desiccant temperature which increases the desiccant surface vapor pressure and causes a decrease in dehumidification process.

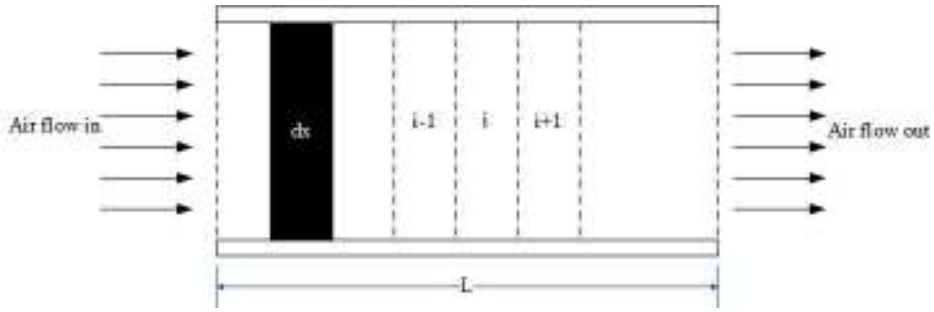


Figure 4.2: Pattern of flow in a slot.

4.2.3 Governing equations

The difference of vapor pressure between air and desiccant surface acts as driving force for mass transfer. For a control volume, air stream and desiccant surface moisture balance can

be written in terms of moisture variations and convective mass transfer between humid air and desiccant.

Mass balance on humid air stream:

$$A_c \rho_a L\left(\frac{\partial \omega_a}{\partial t} + u\frac{\partial \omega_a}{\partial x}\right) = kA_h(\omega_s - \omega_a) \quad \textbf{(4.1)}$$

Mass balance on desiccant surface:

$$\rho_d A_c L\left(\frac{\partial \omega_s}{\partial t} + \frac{\partial w}{\partial t}\right) = k A_h(\omega_s - \omega_a) \quad \textbf{(4.2)}$$

The sensible and latent heat is transferred between air and desiccant due to convection and adsorption phenomenon. Therefore, the energy balance for air stream and desiccant surface can be written as:

Energy balance on the air stream:

$$A_h C_{pa} \rho_a L\left(\frac{\partial T_a}{\partial t} + u\frac{\partial T_a}{\partial x}\right) = h_t A_c(T_s - T_a) + C_{pv} k A_c(\omega_s - \omega_a)(T_s - T_a) \quad \textbf{(4.3)}$$

Energy balance on the desiccant surface:

$$C_{pd} \rho_d A_h L\left(\frac{\partial T_s}{\partial t}\right) = h_t A_c(T_a - T_s) + k A_c(\omega_a - \omega_s)h_{fg} + C_{pv} A_c(\omega_a - \omega_s)(T_a - T_s) \quad \textbf{(4.4)}$$

The humidity on the desiccant surface can be evaluated in terms of saturated water vapor pressure on the surface of desiccant:

$$\omega_s = \frac{0.0622 p_v}{1.0133\times10^5 - p_v} \quad \textbf{(4.5)}$$

Saturated water vapor pressure on the desiccant surface is given as [232]:

$$p_v = e^{\left(23.196 - \frac{3816.44}{Ts - 46.14}\right)} \quad \textbf{(4.6)}$$

Coefficient of heat and mass transfer for rotary type liquid desiccant dehumidifiers can be expressed as [233]:

$$k = 0.704.\dot{m}_a.\mathrm{Re}^{-0.51} \quad \textbf{(4.7)}$$

$$h_t = 0.671.\dot{m}_a.\mathrm{Re}^{-0.51}.C_{pa} \quad \textbf{(4.8)}$$

4.3 Boundary and initial conditions

The flow of process air and regeneration air are in counter-flow arrangements. The boundary conditions for temperature and humidity ratio can be written as:

Process air enters the channel at axial distance L = 0:

$$T_a(0,t) = T_{p,i} \quad \textbf{(4.9)}$$

$$\omega_a(0,t) = \omega_{p,i} \quad \textbf{(4.10)}$$

Regeneration air enters the channel at axial distance L:

$$T_a(L,t) = T_{r,i} \quad \textbf{(4.11)}$$

$$\omega_a(L,t) = \omega_{r,i} \quad \textbf{(4.12)}$$

The initial conditions for temperature and humidity ratio are [234]:

$$T_a(\mathrm{x},0) = T_{p,i} \tag{4.13}$$

$$T_s(\mathrm{x},0) = T_{r,i} \tag{4.14}$$

$$\omega_a(\mathrm{x},0) = \omega_{p,i} \tag{4.15}$$

$$\mathrm{w}(\mathrm{x},0) = \mathrm{w}_i \tag{4.16}$$

4.4 Performance index

The primary purpose of the desiccant dehumidifier is to dehumidify the air. The dehumidification capacity is represented by moisture removal which is the amount of water desorbed from the process air by the desiccant. Moisture removal capacity of the desiccant dehumidifier can be written as:

$$M = \dot{m}_p \times (\omega_{a,i} - \omega_{a,o}) \tag{4.17}$$

Latent effectiveness of the dehumidifier is given as:

$$\mathcal{E} = \frac{(\omega_{a,i} - \omega_{a,o})}{(\omega_{a,i} - \omega_{a,o\ ideal})} \tag{4.18}$$

Where, $\omega_{a,out\ ideal}$ represents the ideal specific humidity ratio. Assuming that at this point air is completely dehumidified, $\omega_{a,o\ ideal}$=0.

Dehumidification coefficient of performance (DCOP) is another parameter used to represent the desiccant dehumidifier capacity to dehumidify the air. For the different mass flow rates of process and regeneration air, DCOP is given as:

$$DCOP = \frac{\dot{m}_p \times h_{fg} \times (\omega_{a,i} - \omega_{a,o})}{\dot{m}_r \times (h_r - h_{amb})} \quad \textbf{(4.19)}$$

A higher DCOP indicates a better system performance because the energy input to the regeneration air is utilized in a better way or less heat is being used to heat up the desiccant.

The sensible energy ratio (SER) is also used when evaluating the desiccant dehumidifier. The SER can quantify additional sensible load to be handled by some additional cooling mechanism at the downstream such as an evaporative cooling or a vapor compression cycle. The SER is given by:

$$SER = \frac{\dot{m}_p \times (T_{a,o} - T_{a,i})}{\dot{m}_r \times (T_r - T_{amb})} \quad \textbf{(4.20)}$$

For better dehumidification performance of the desiccant wheel the value of SER should be lower. The higher value of SER means more cooling load on the cooling device because of higher temperature of process air at exist of desiccant wheel. Lower values of SER mean that the desiccant dehumidifier is creating a lower sensible cooling load, which indicates better performance of the system.

4.5 Results and discussions

A rotary type liquid desiccant dehumidifier is proposed in this paper to have lower regeneration temperature, no carryover of desiccant solution and better supply air conditions. A code has been developed in Engineering Equation solver (EES) using mathematical model and performance of the system is investigated under geometric and

operating parameters shown in Table 4.3. Additionally, the properties of calcium chloride and moist air are taken from the built in function of EES interface. Firstly the system performance have been observed for hot and humid climatic conditions of Dhahran, Saudi Arabia and then parametric analysis have been performed. Only one parameter is varied in each case, keeping all other parameters constant at the base values.

Table 4.3: Operating and geometric parameters.

Parameter	Value range	Base value
Process air inlet temperature (^oC)	30 – 45	40
Regeneration air inlet temperature (^oC)	50 – 90	65
Regeneration air inlet humidity ratio (kg_v/kg_a)	0.005 – 0.025	0.020
Process air inlet humidity ratio (kg_v/kg_a)	0.005 – 0.025	0.020
Process air flow rate ($kg/s.m^2$)	0.5 – 4	3
Regeneration air flow rate ($kg/s.m^2$)	0.5 – 4	1
Width of dehumidifier (m)	0.1 – 0.6	0.1
Desiccant material	$CaCl_2$	-

4.5.1 Feasibility analysis

One of the greatest needs for comfort in hot and humid climate countries such as Saudi Arabia, is air conditioning to control both temperature and humidity of air. In this section

the monthly and daily variations of different performance parameters for the proposed liquid desiccant dehumidifier operating under climatic conditions of Dhahran, Saudi Arabia (26.2° North, 50.6° East) have been presented. Figures 4.3 and 4.4 show the results for different outdoor conditions of Dhahran. These results were obtained for an indoor temperature of 23°C. The performance of the system varies with ambient temperature and humidity. The behavior of dehumidification performance of the dehumidifier for the whole year is illustrated in Fig. 4.3. The major factor which determines the dehumidification performance of the system is the ambient air absolute humidity. This happens because, the moisture transfer potential from air to desiccant increases with the higher specific humidity, which further augments the enthalpy difference between inlet and supply air. When the water content of the ambient air increases, the DCOP of the liquid desiccant dehumidifier also increases. The average value of DCOP for the year is found to be 0.55 which shows that system can control the latent load efficiently throughout the whole year. The maximum value of DCOP (about 0.70) was obtained for May and minimum (about 0.19) for January. Figure 4.4 shows DCOP results varying from 0.2 to 0.7 for one typical day in July. An average of 0.48 was obtained for the period.

Figure 4.3 also represent the monthly variations of latent effectiveness. Both maximum and actual possible change in humidity ratio increases but the rate of increase for maximum possible dominates the actual one. Thus the effectiveness increases with the increase in specific humidity of ambient air. This is mainly because increasing the air humidity will result into an increase in the air partial vapor pressure, and thus providing larger vapor pressure difference between the process air and the liquid desiccant surface. This increase

in the vapor pressure difference is accompanied by an increase in the desiccant moisture absorption capacity and dehumidifier effectiveness. The maximum values of wheel effectiveness achieved was 0.76 for the month of May and September, while a minimum effectiveness value of 0.50 was obtained for the month of January. The hourly variations of effectiveness are not significant as it can be observed from Fig. 4.4. Effectiveness results varies from 0.60 to 0.64 for one typical day in July with an average of value of 0.63.

The monthly moisture removal capacity of desiccant dehumidifier from the process air is presented in Fig. 4.3. The removed moisture depends on different parameters such as inlet air humidity ratio, air flow rate and the desiccant wheel material. The removed moisture is following the same trend as the humidity of inlet air for all months. The change in specific humidity and moisture removal capacity increase with the increase in inlet air specific humidity. The maximum amount of moisture removed from the process air is 0.026 kg/s for the month of September. The moisture removal capacity varies from 0.008 to 0.020 kg/s for one typical day in July as illustrated in Fig. 4.4.

Figure 4.3 also shows the monthly variations of sensible energy ratio (SER). The value of SER depends on ambient conditions that is, temperature and humidity ratio. The maximum and minimum value of sensible energy ratio are 0.85 and 0.09 for the month of July and January, respectively. Figure 4.4 shows SER results varying from 0.60 to 0.88 for one typical day in July. An average of 0.73 was obtained for the period.

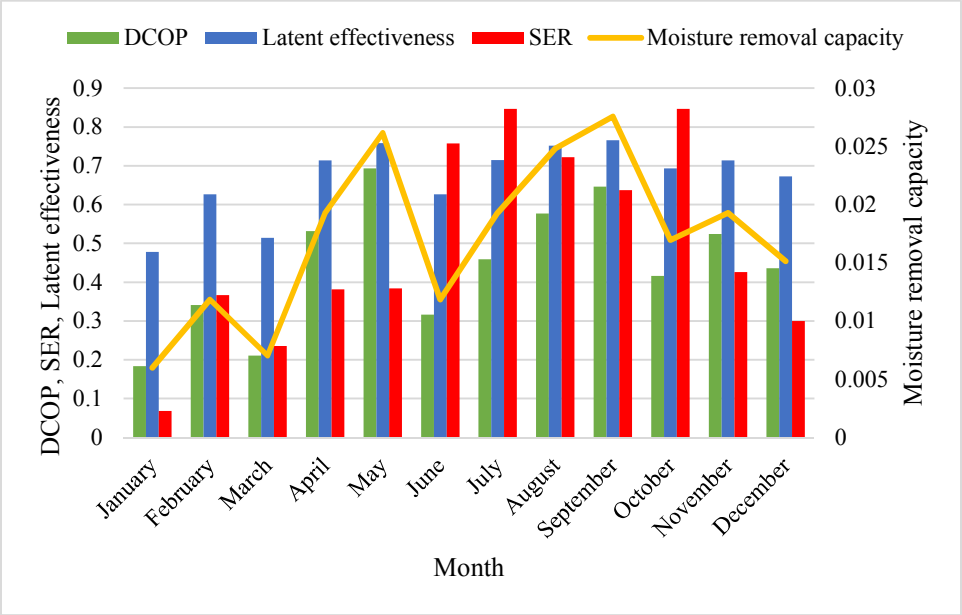


Figure 4.3: Monthly variations of DCOP, latent effectiveness, moisture removal capacity, and SER.

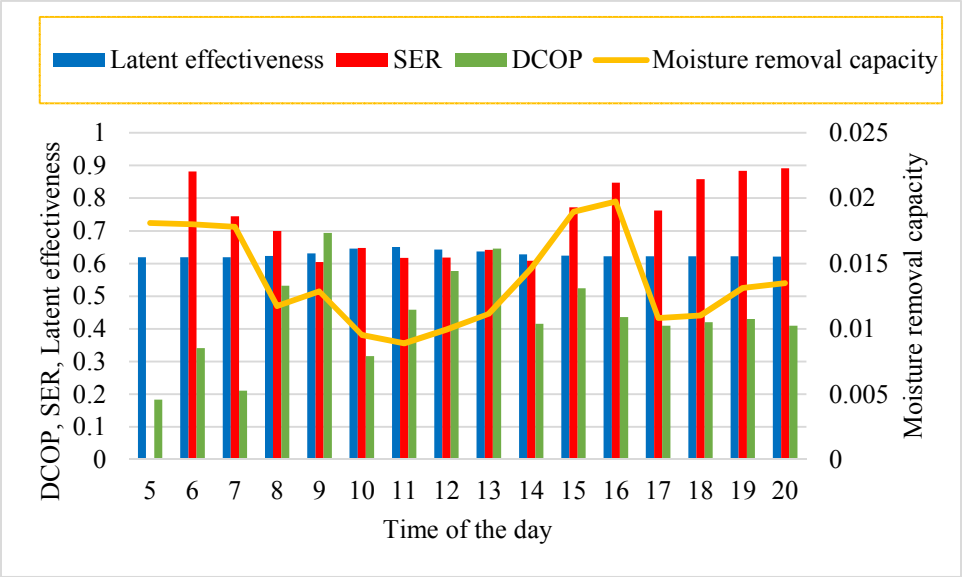


Figure 4.4: Hourly variations of DCOP, latent effectiveness, moisture removal capacity, and SER.

4.5.2 Parametric analysis

The difference of temperature and vapor pressure between air and desiccant streams acts as driving force for transfer of heat and mass, respectively. The effect of process air mass flux on heat and mass transfer coefficients is shown in Fig. 4.5. Both coefficients increase as process air mass flux increases because of increase in driving force for heat and mass transfer. As the air mass flux increases from 0.01 to 4 kg/m^2.s, an increase from 1 to 13 W/m^2.K and 0.002 to 0.04 kg/m^2.s is observed in the heat and mass transfer coefficients, respectively. The increase in mass transfer coefficient causes a better transfer of moisture from humid air to the desiccant which increases the dehumidification capacity of the desiccant dehumidifier.

The effect of process air mass flux on exit air humidity ratio for different width of the rotary dehumidifier is illustrated in Fig. 4.6. The exit air humidity ratio increases with the increase in air mass flux which shows a decrease in air dehumidification with the increase in mass flux of air. This is due to the fact that at high flow rates, the air has less contact time with the desiccant surface and therefore, mass transfer is smaller than that at low flow rates. The system performance for dehumidification as well as for cooling process are largely affected by the width of the wheel. Figure 4.6 also shows for any given value of air mass flux, exit humidity ratio decreases with the increase of wheel width. This decrease in humidity ratio is due to increase in surface contact area and residual time between air and desiccant. It can also be observed that, after a certain value of width the decrease in humidity ratio becomes almost constant and a negligible decrease occurs with further increase in width. The reduction of about 30% in humidity ratio is observed with an increase in the wheel width

from 0.1 to 0.2 m. While the width increases from 0.2 to 0.6 m, only 18% reduction in humidity ratio is observed. This happens because, as the air flows across the dehumidifier, the moisture difference between humid air and desiccant surface decreases which causes a decrease in driving force for moisture transfer. This means for any value of air mass flux there should be an optimum value of wheel width for required exit conditions of air. The optimization of wheel width is favorable for the purpose of air conditioning system design because it reduces the system overall manufacturing cost.

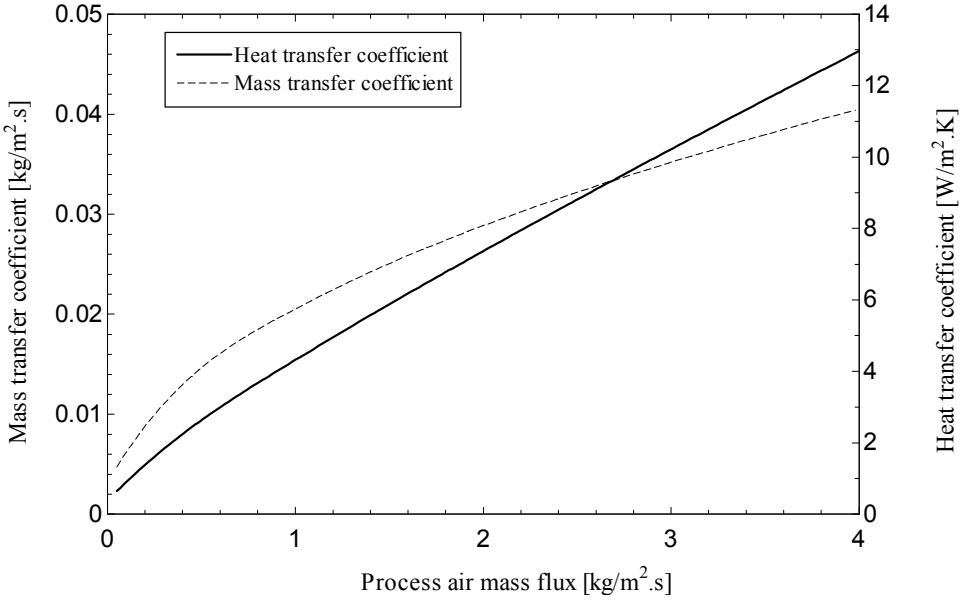


Figure 4.5: Effect of process air mass flux on heat and mass transfer coefficients.

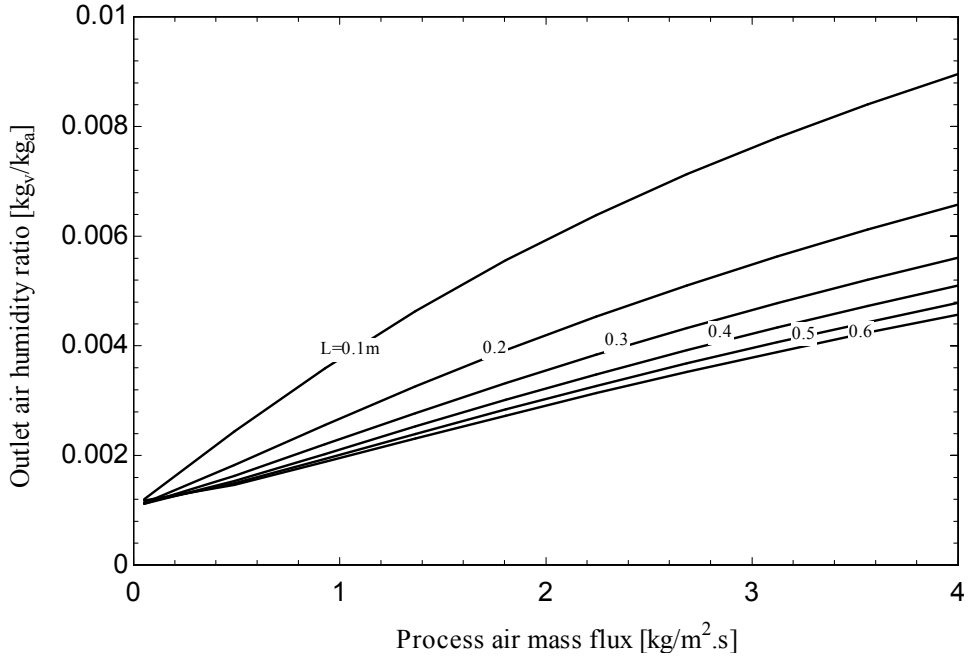


Figure 4.6: Effect of process air mass flux on outlet air humidity ratio.

Figure 4.7 shows the air flux effect on the latent effectiveness and dehumidification performance of the dehumidifier. Both latent effectiveness and DCOP decreases with the increase in air mass flux. Latent effectiveness is related to the dehumidification capacity of the dehumidifier; drier the air at the exit of dehumidifier, higher will be the latent effectiveness of dehumidifier. At higher flow rate the exit humidity ratio of the process air increases which causes decrease in latent effectiveness.

The variation of dehumidifier effectiveness and dehumidification performance with humidity ratio is illustrated in the Fig. 4.8. As the humidity ratio of process air at the inlet increases the dehumidifier effectiveness and DCOP increase because of better mass transfer between two streams. It can be observed from the Fig. 4.8 that as the inlet humidity ratio of the air increases from 0.005 to 0.020 kg$_v$/kg$_a$, the effectiveness increases from 0.2

to 0.7 but after 0.020 kg_v/kg_a, there is very small change in effectiveness, which means the increasing of absolute moisture removal is not as significant as the increasing of $\omega_{a,in}$ under the simulated condition. In fact, increasing the air inlet humidity ratio causes an increase in the driving force and hence increases the mass transfer potential within the dehumidifier. Consequently, the desiccant dehumidifier can obtain a better effectiveness and high DCOP with higher $\omega_{a,in}$. The partial vapor pressure is the governing factor for the mass transfer between process air and desiccant surface. As the inlet air humidity ratio increases, the partial pressure of water vapor in the air also increases which in turn enhances the difference between the partial vapor pressure in the inlet air stream and on the desiccant surface resulting an increase in the moisture absorbing capacity of desiccant.

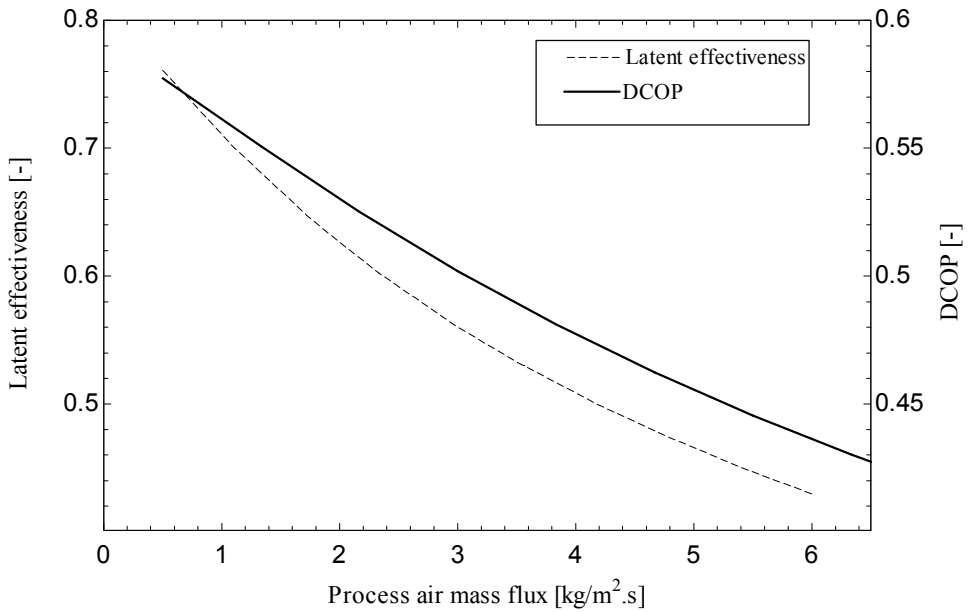


Figure 4.7: Effect of process air flux on the latent effectiveness and DCOP.

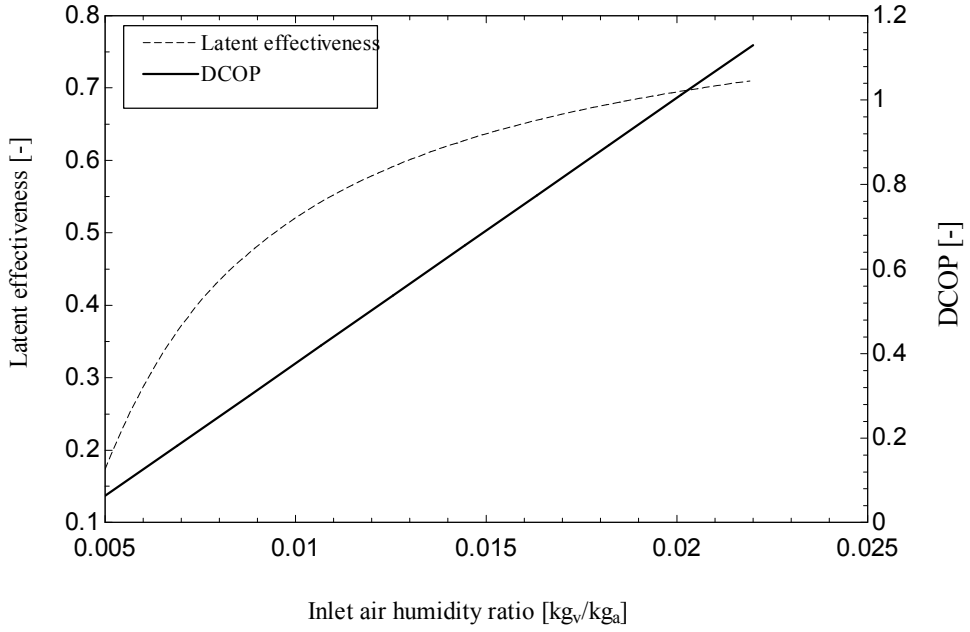


Figure 4.8: Effect of inlet air humidity ratio on the latent effectiveness and DCOP.

The variation of DCOP with rotational speed of the wheel at different regeneration temperatures is presented in Fig. 4.9. The DCOP decreases with the increase in rotational speed because of less contact time between humid air and desiccant surface. The occurrence of the maximum DCOP at a low rotational speed is expected. If the rotational speed is too high the desiccant cannot fully absorb or desorb water molecules because of the less process time. Since the main purpose of the desiccant dehumidifier is to remove moisture rather than exchanging heat, the rotational speed that yields the maximum DCOP is recommended. Figure 4.9 also shows the strong dependence of DCOP on the regeneration temperature. As the regeneration temperature increases, DCOP for latent cooling decreases. Although the higher value of regeneration temperature will make the moisture removal process faster but at the same time this higher value of temperature will dry-up the desiccant wheel before the completion of regeneration period. Hence, some

added energy will be wasted and will not be utilized during absorption period for moisture removal. Secondly, the higher value of regeneration temperature will increase the input energy which decreases the DCOP of the cycle. A low regeneration temperature enhances the potential of these systems to use in solar application.

The other parameter which affect the DCOP of the system is ratio of regeneration to process air flow rate. The DCOP increases as this ratio decreases as shown in Fig. 4.10. The increase in DCOP with decrease in this ratio of air flow rates occurs because of decrease in required regeneration heat. As the ratio of air flow rates or flow rate of regeneration air decreases, required heat of regeneration decreases. The removal of moisture from the air decreases with the increase in speed of the wheel as shown in Fig 4.11. As the wheel speed increases the contact time between air and desiccant decreases, the removal of moisture from the air decreases.

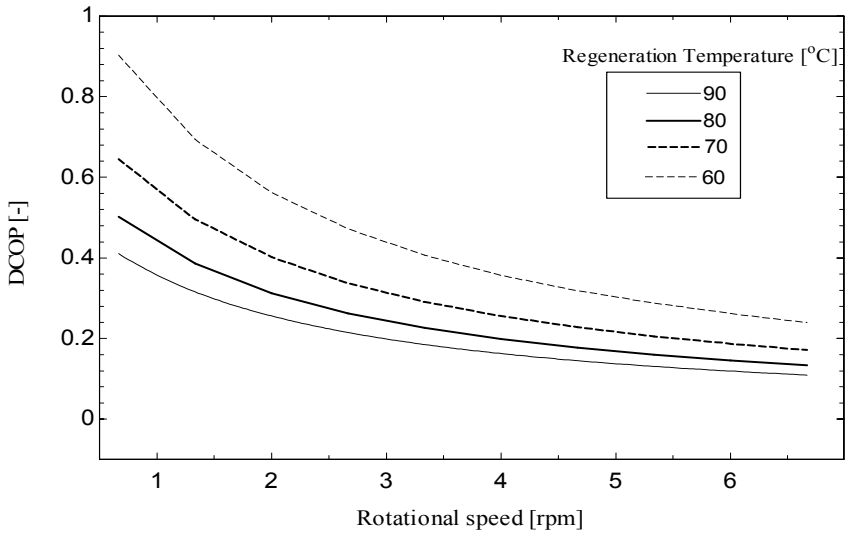


Figure 4.9: Effect of rotational speed on DCOP for different regeneration temperatures.

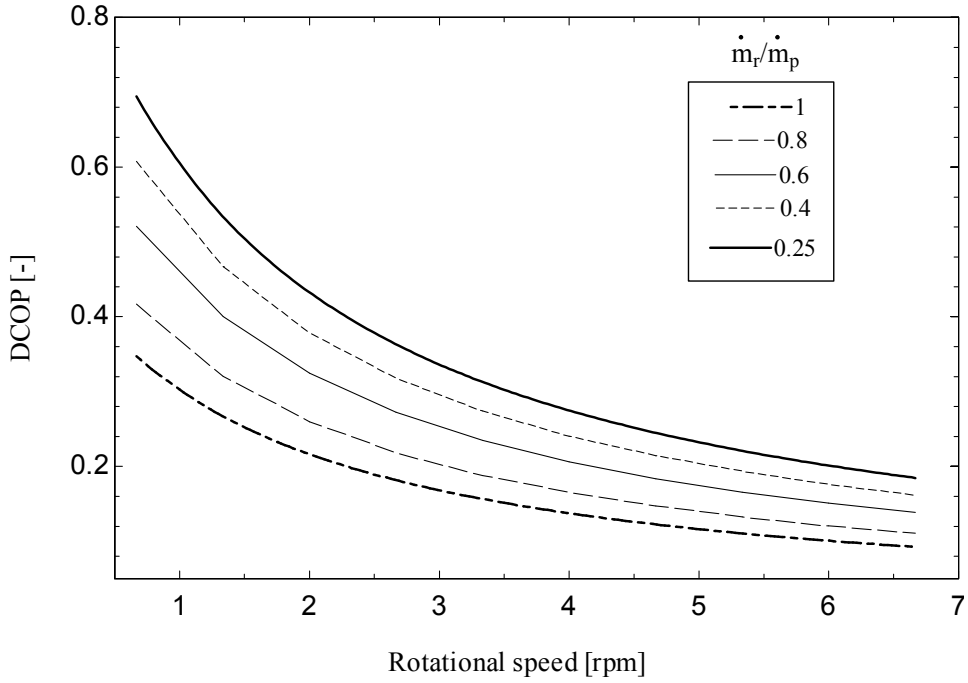


Figure 4.10: Effect of rotational speed on DCOP for different mass flow rate ratios.

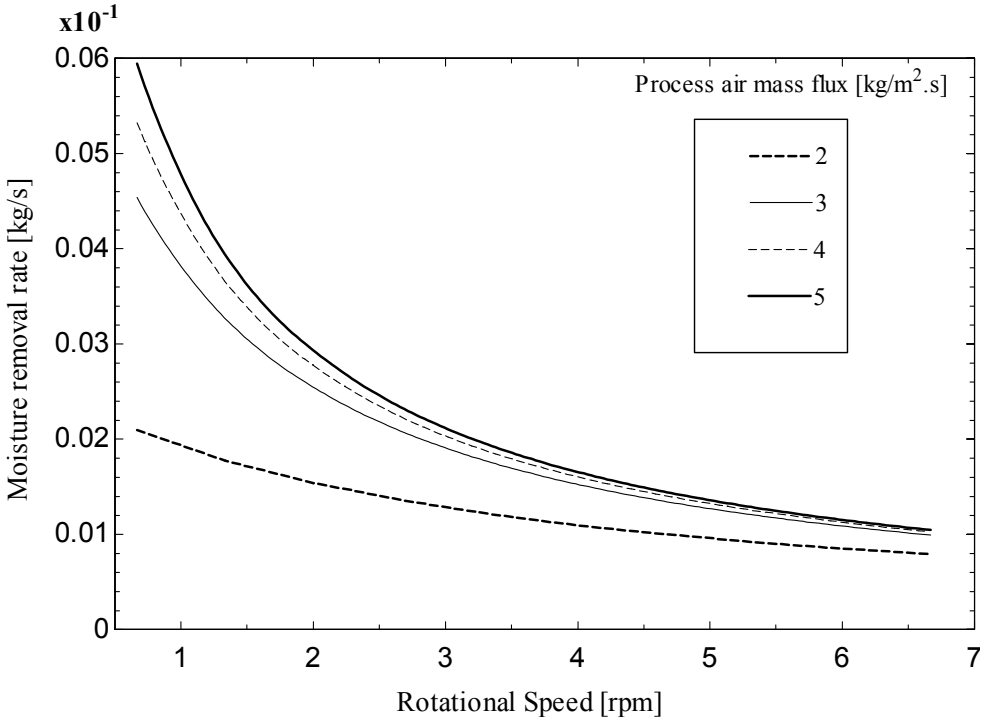


Figure 4.11: Effect of rotational speed on dehumidifier moisture removal capacity for different process air mass flow rate.

The sensible energy ratio is another important performance parameter which defines the additional cooling required to provide the thermal comfort conditions. The variations of sensible energy ratio for various inlet air conditions is shown in Fig. 4.12. As air humidity ratio increases, a greater amount of water vapor is absorbed by the desiccant dehumidifier which increases the released absorption heat. This increase in absorption heat causes an increase in air temperature at the exit of the desiccant wheel. At a constant value ambient and regeneration air temperature, the increase in air temperature at the exit of the dehumidifier cause a decrease in sensible heat ratio. Thus, with the increase in inlet humidity ratio sensible heat ratio increases. The increase in inlet air temperature will reduces the water removal from the process air, slightly. This will decrease the heat of absorption and air temperature at the exit of the dehumidifier. Thus, the increase in inlet air temperature causes a decrease in the sensible energy ratio as it can be observed from Fig. 4.12. But the variations of sensible energy ratio due to inlet air temperature are not much significant because of small variations in absorption heat and air temperature at exit of dehumidifier.

The variations of sensible energy ratio for different ratios of air mass flow rates (regeneration to process air) and regeneration temperature are shown in Fig. 4.13. As mentioned above, the increase in ratio of regeneration to process air flow rate causes a decerese in DCOP for desiccant desiccant dehumifidier because of increse in required regeneration heat. Also the increase in this ratio will cause an increase in water absorption rate and released heat of absorption. This increased heat of absorption will cause an increase in temperature at the exit of the dehumidifier which in turns will decrease the

sensible heat ratio. Thus, with the increase in ratio between regeneration and process air flow rates, a reduction in sensible energy ratio occurs. The increase in regeneration temperature will also cause a decrease in sensible energy ratio because of increase in required heat input.

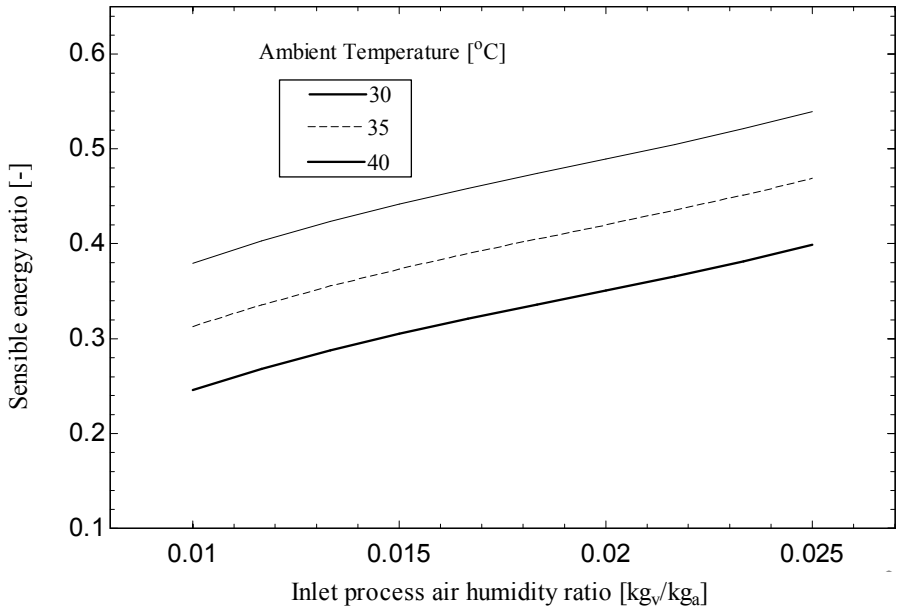


Figure 4.12: Effect of inlet air conditions on sensible energy ratio for different ambient air temperatures.

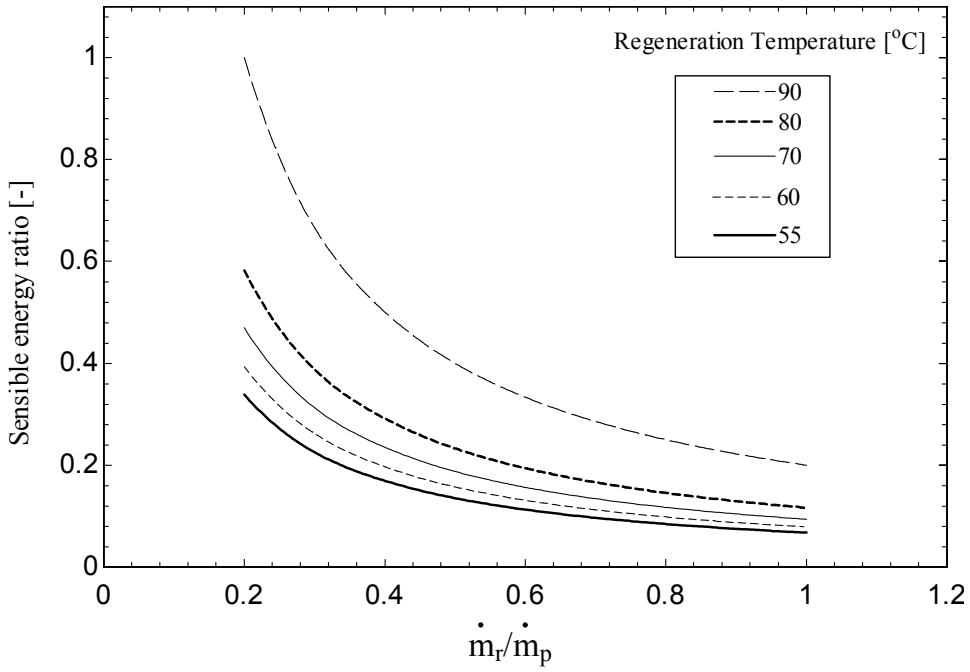


Figure 4.13: Effect of mass flow rates ratio on sensible energy ratio for different regeneration temperatures.

4.6 Conclusions

A theoretical model for the liquid desiccant dehumidifier is developed to observe the heat and mass transfer phenomenon for absorption and desorption process under different operating parameters like process air flow rate, humidity ratio of process air, and width of the dehumidifier. Effect of these parameters also been observed on dehumidifier effectiveness, moisture removal rate and DCOP. The computed results show that supply air condition can be obtained to provide human comfort in the hot and humid climate with effectiveness of the system largely dependent on air flow rate, dehumidifier width and humidity ratio of process air. For better performance of the desiccant cooling system the regeneration heat required should be as minimum as possible. The regeneration heat depends on the regeneration air flow rate, an optimum and feasible value of the

regeneration air flow rate can be obtained by the parametric analysis in the required zone of operation.

Different performance curves of the desiccant dehumidifier have been drawn in order to determine optimum values of different parameters. The result shows that there is an optimum value of each design parameter under each operating condition and above that value, there is very little change in the performance of desiccant dehumidifier. The main conclusions emerging from the present work can be summarized as:

- The dehumidification performance of the desiccant wheel increase with the increase in process air inlet humidity ratio because of better mass transfer between two streams.
- Increasing the wheel width leads to better dehumidification of process air; however, the dehumidification rate is decreased because of reduction in moisture difference between humid air and desiccant surface. The reduction of about 30% in humidity ratio is observed with an increase in the wheel width from 0.1 to 0.2 m. While the width increases from 0.2 to 0.6 m, only 18% reduction in humidity ratio is observed.
- The COP of the system is high at lower rotational speed because of the more process time for absorption and desorption of moisture.
- As the regeneration temperature or flow rate of regeneration air increases, COP for latent cooling decreases due to high required heat input.

- The exergy efficiency of the system increases with the increase in ambient air temperature and decreases with the increase in regeneration temperature, ambient air humidity ratio, and ratio of mass flow rates.
- The feasibility analysis shows that the system can operate with a good dehumidification capacity through the whole year. The average value of DCOP for the year is found to be 0.55 which shows that system can control the latent load efficiently throughout the whole year. The maximum value of DCOP of about 0.7 was obtained for May and September and minimum of about 0.2 for January.

For better coefficient of the performance of the overall system the desiccant material should have lower temperature of regeneration which is a key factor for future research.

CHAPTER 5

EXPERIMENTAL INVESTIGATION OF LIQUID DESICCANT COOLING CYCLE

5.1 Introduction

The simultaneous control of temperature and humidity of air is required to provide the human comfort conditions. The conventional air conditioning system cools the air below its dew point temperature to remove the moisture from air. In hot and humid regions, considerable amount of load is used to remove the moisture from the air using these conventional vapor compression refrigeration systems. Because of the high energy cost of these conventional systems and poor control of latent load in hot and humid conditions, need arises for some alternative cooling system. Desiccant cooling is found to be an alternative to vapor compression refrigeration for applications of space cooling. It is beneficial for environment and also cost effective since no refrigerant is required which can deplete the ozone layer. Low temperature heat sources, like waste heat from engine or solar heat, can be used to operate the system.

Desiccant cooling is a thermal technique based on air dehumidification followed by evaporative cooling. This technology is a good alternative for conventional vapor compression systems to meet new economic, environmental, and regulatory challenges. The major component of a desiccant cooling system is desiccant dehumidifier which is used to control the latent load by the dehumidification of the humid air. The desiccant

dehumidifier can be of different configurations emended with a desiccant material. The desiccant cooling system can either be solid or liquid depending upon the type of desiccant material used.

The liquid desiccant wheel compromises of two sections. One section is for process air and the other for the regeneration air. In the process side water vapor from the humid air is absorbed in the matrix of the desiccant and then this absorbed water vapor is removed from the desiccant by hot regeneration air, as the wheel rotates to the regeneration side.

In this present work, an experimental unit is built to study and discuss the performance of a liquid desiccant wheel operating in conjunction with indirect ant direct evaporative coolers. The computed results show that better supply air conditions can be obtained to provide human comfort in the hot and humid climate conditions.

5.2 Test facility

The schematic of the experimental setup is shown in Fig. 5.1. The test facility mainly consists of following components:

- A rotary liquid desiccant dehumidifier (30 cm diameter).
- A direct evaporative cooler (30×60 cm^2).
- An indirect evaporative cooler (30×60 cm^2).
- A water spraying system (0.37 kW pump).
- A variable capacity electric heater (0 – 4 kW).
- Two air blower

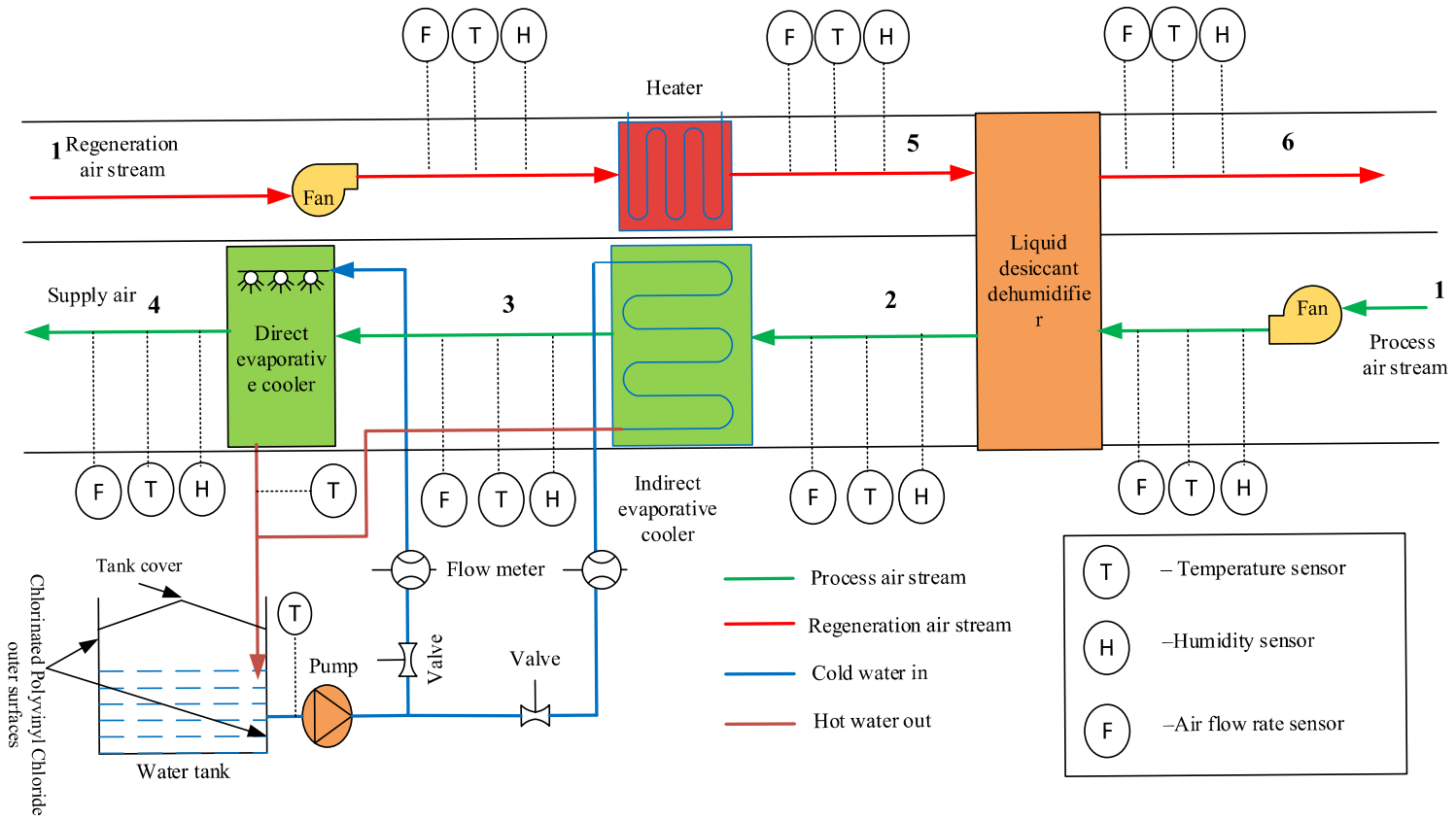
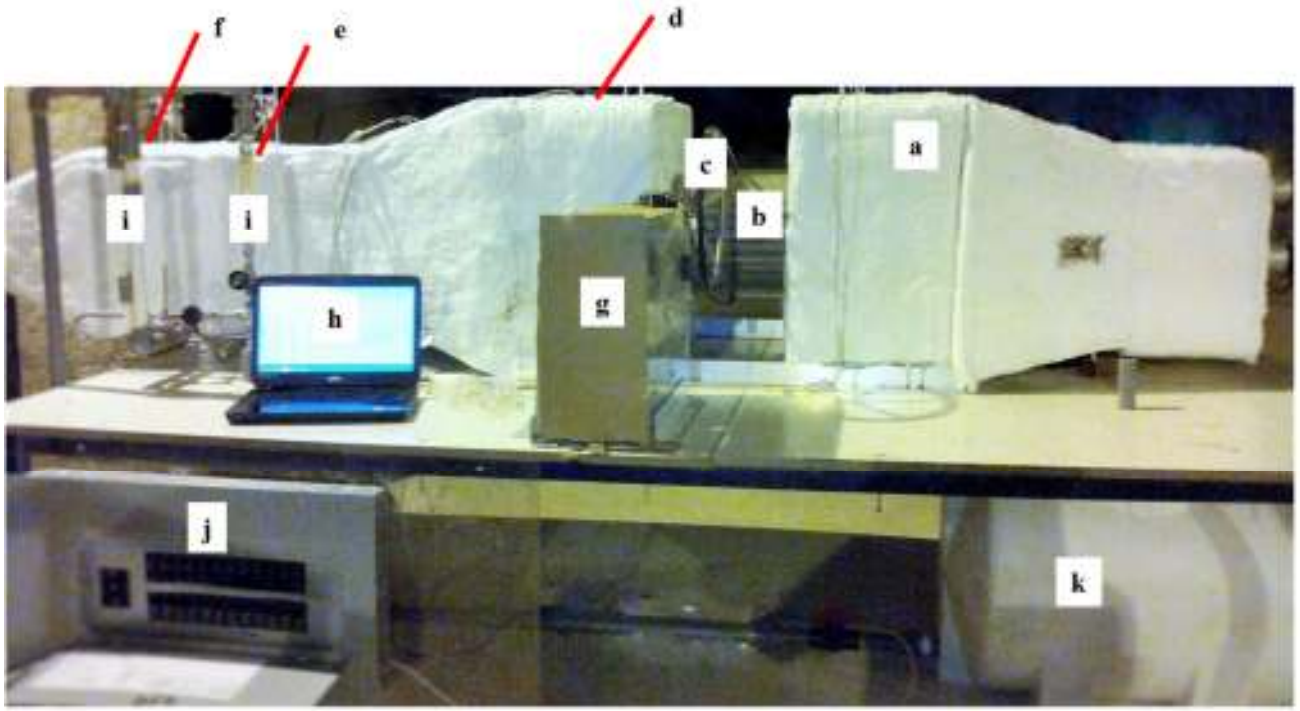


Figure 5.1: Schematic of experimental setup.



(a)- Ducting with insulation, (b)- Desiccant dehumidifier (wheel), (c)- Pulley and belt, (d)- Heater (inside), (e)- Indirect evaporative cooler (inside), (f)- Direct evaporative cooler (inside), (g)- Variable speed motor, (h)- Data acquisition system, (i)- Flow meter, (j)- Electrical panel, (k)- Water tank

Figure 5.2: Photographic view of experimental setup.

The details of all other supporting and operation facilities are shown in Fig. 5.2. The rotating desiccant wheel shown in Fig. 5.3(a) has a cylindrical shape of 30 cm diameter is fabricated from flexi glass. Flow area is created using pipes of Chlorinated polyvinyl chloride of diameter 21 mm with minimum spacing between two pipes. These pipes are uniformly distributed over the cross section of the wheel. To form the absorbing surface, a thick cloth layer impregnated with the calcium chloride solution is placed inside the pipes using springs. The variable speed motor is used to vary the rotational speed of the desiccant wheel. The regeneration heat is provided using a variable capacity electric heater. In the process air stream, after the desiccant dehumidifier direct and indirect evaporative coolers are used for cooling purpose. With direct evaporative cooling, outside air is blown through a water saturated medium and cooled by evaporation. The packing material for the used direct evaporative cooler is composed of cellulose pad. The size of the pad is 15 cm deep 30 cm wide and 60 cm high. The water is distributed using spray nozzles at the top of cooling pads. While in indirect evaporative cooler, cooling water circulates inside the coils and air cools as it passes across the tubes. The indirect evaporative cooler coils are constructed from copper tubes with bonded aluminum fins. Fins are attached to the outside of tubes to enhance the surface area for heat transfer. The coils have an anti-corrosion coating such as blue fin coating for extra protection and longer life. The detailed working principle for both types of evaporative coolers have already been discussed in Chapter 2. The photographic views of direct and indirect evaporative cooler used in this work have been shown in Fig 5.3(b and c), respectively.

Two air dampers are used to vary the flow rates of process and regeneration air. The 25% face area of the wheel is for the regeneration stream of the air and remaining 75% area is for the process air stream.

The two streams of air flow in the counter flow direction as shown in Fig. 5.1. For the process air stream, humid air passes through the desiccant wheel and moisture from the air is absorbed by the liquid desiccant (1 – 2). This dehumidified air is first passed through an indirect evaporative cooler (2 – 3) and then direct evaporative cooler (3 – 4) to lower its temperature according to supply demands. The ambient air on the regeneration side is passed through the electric heater to bring it to the desired temperature for reactivation of desiccant wheel (1 – 5). The high temperature air then enters the wheel and moisture is removed from the desiccant (5 – 6). The complete desiccant cooling cycle is represented on psychrometric chart in Fig. 5.4.

(a) (b) (c)

Figure 5.3: (a) Desiccant wheel; (b) direct evaporative cooler; (c) indirect evaporative cooler.

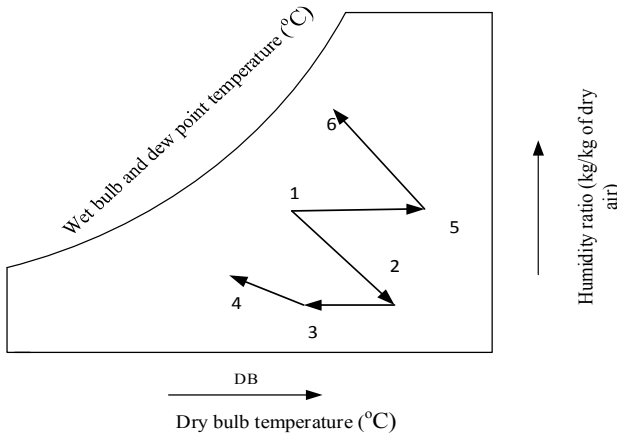


Figure 5.4: Psychrometric representation of desiccant cooling cycle.

5.3 Desiccant preparation

The concentration of calcium chloride was obtained experimentally using a simple procedure which can be described as follow: A solution was prepared in a close container using a known mass of water and $CaCl_2$ flakes. The solution was covered and placed in an environment with temperature around 25 – 30°C for 48 hours. The solution was filtered to remove any impurities. Then, the specific gravity and temperature of the solution were measured simultaneously and desiccant concentration was determined using $CaCl_2$ handbook [235].

5.4 Instrumentations

Temperature of the process and regeneration air at all the state points are measured using k-type thermocouples as shown in Fig. 5.1. A digital hygrometer and digital anemometer were used to measure humidity and velocity of the air, respectively at different points. All

properties obtained such as temperatures and RHs were used to evaluate the psychrometric properties including humidity ratio and enthalpy. The power consumption of the electric heater was measured with digital watt meters. The details of all the sensors are provided in Table 5.1 and they were calibrated under testing conditions.

Table 5.1: Specifications of sensors.

Parameters	**Devices**	**Model no.**	**Accuracy**
Air temperature	K-type thermocouple	KK-K-30	±0.2°C @23±5°C
Air relative humidity	Hygrometer	RHXL3SD	**For ≥ 70% RH** ±(3%+ 1% RH) **For < 70% RH** ± 3% RH
Air flow rate	Anemometer	HHF1001A	1.5% Full scale accuracy
Heater capacity	Watt meter	GH-019D	± (0.2% reading + 0.05% FS)
Desiccant density	Hydrometer	-	1 kg/m^3

5.5 Test procedure and conditions

The data acquisition system was established which control the relative humidity, temperature, and mass flow can rate of all state points shown in Fig. 5.1. Both blowers and desiccant wheel motor were turned on. Once the desired air conditions were reached, the electric heater was turned on to obtain the desired regeneration temperature. After steady state conditions had been achieved, data were recorded with a specified time interval.

5.6 Uncertainty analysis

The experimental error can be classified into two broad categories: systematic error and random error, which typically accounts for the uncertainty of the acquired data [236]. The systematic error is associated with the various instrumentation precisions, manifested in temperature, pressure, and relative humidity measurements. The random error pertains to the acquired data deviation from the mean value. Performance indices, or dependent variables, are correlated with the error propagation of the individual measurement uncertainties. The general error propagation is given by Eq. (5.1). Here u is a function of x, y and z. Δu is the uncertainty of u, similar to Δx, Δy and Δz.

$$\Delta u = \sqrt{\left(\frac{\partial u}{\partial x}\right)^2 \times (\Delta x)^2 + \left(\frac{\partial u}{\partial y}\right)^2 \times (\Delta y)^2 + \left(\frac{\partial u}{\partial z}\right)^2 \times (\Delta z)^2} \quad \textbf{(5.1)}$$

An example for the propagated error, pertaining to moisture removal capacity (MRC) is given below by Eq. (5.2):

$$\Delta MRC = \sqrt{\begin{matrix}\left(\Delta \dot{m}_p\right)^2 \times \left(\omega_{p,in} - \omega_{p,out}\right)^2 + \left(\dot{m}_p \times \omega_{p,out}\right)^2 \times \left(\Delta \omega_{p,in}\right)^2 \\ +(\dot{m}_p \times \omega_{p,in})^2 \times (\Delta \omega_{p,out})^2\end{matrix}} \quad \textbf{(5.2)}$$

The combined uncertainty U is determined by the calibration of all sensors:

$$U = \sqrt{(U_{instrument})^2 + (U_{caliberation})^2 + (U_{random})^2} \quad \textbf{(5.3)}$$

Calibration error is the error of the reference device. Random or precision error which is defined in accordance with Deutsche Kallibriendienst (DKD) as

$$U_{random} = \frac{S_{\mu,95\%}\vartheta}{\sqrt{N_D}} \quad \textbf{(5.4)}$$

Where,

$$\mu = \text{N-1} \quad \textbf{(5.5)}$$

Illustration for calculation: A standard platinum resistance thermometer (Type PT25) with an uncertainty of 2.0-4.0 mK in the temperature range of 0-120°C, is used for the calibration of K-type thermocouple. Both the thermocouple and PT25 are placed in thermostat with an immersed copper block calibrator for calibration. An uncertainty in the range of 0.10 - 0.25 K is realized for all thermocouples used in this work. This range is equivalent to 0.5-1.0% for the temperature range of 20-70°C; where 20°C is the average room temperature during the measurement and 85°C is the maximum regeneration temperature used in this work. The calculated uncertainties for different instrumentations are listed in Table 5.2.

Table 5.2: Uncertainties of measuring instruments used in experiment.

Instrument	Parameter	Uncertainty
K-type thermocouple	Temperature	± 0.1°C
Anemometer	Velocity of air	± 0.1m/s
Density meter (hydrometer)	Desiccant density	± 1 kg/m^3
Electronic weighing machine	Desiccant mass	± 0.001 kg
Rota meter	Water flow rate	± 0.5 L/min

5.7 Performance parameters

The rate of moisture removal from the process air by the liquid desiccant cooling system is defined as moisture removal rate (M_r).

$$M_r = \dot{m}_p(\omega_1 - \omega_2) \quad \textbf{(5.6)}$$

Note that all the state points refers to Fig. 5.1.

Dehumidification coefficient of performance (DCOP) is another parameter used to represent the desiccant dehumidifier capacity to dehumidify the air. For the different mass flow rates of process and regeneration air, DCOP is given as:

$$DCOP = \frac{\dot{m}_p \times h_{fg} \times (\omega_1 - \omega_2)}{\dot{m}_r \times (h_7 - h_6)} \quad \textbf{(5.7)}$$

A higher DCOP indicates a better system performance because the energy input to the regeneration air is utilized in a better way or less heat is being used to heat up the desiccant.

The sensible energy ratio (SER) is also used when evaluating the desiccant dehumidifier. The SER can quantify additional sensible load to be handled by some additional cooling mechanism at the downstream such as an evaporative cooler. The SER is given by:

$$SER = \frac{\dot{m}_p \times (h_2 - h_1)}{\dot{m}_r \times (h_7 - h_6)} \quad \textbf{(5.8)}$$

For better dehumidification performance of the desiccant wheel the value of SER should be lower. The higher value of SER means more cooling load on the cooling device because of higher temperature of process air at the outlet of desiccant wheel. Lower values of SER

mean that the desiccant dehumidifier is creating a lower sensible cooling load, which indicates better performance of the system.

The overall cooling provided by the desiccant system is defined as cooling capacity. The difference of enthalpy between outdoor and supply air is used to represent overall cooling capacity because it includes both sensible and latent loads.

$$Cooling\ capacity\ (CC) = \dot{m}_p\ (h_1 - h_4) \quad \textbf{(5.9)}$$

The overall performance of the system is represented by coefficient of performance (COP).

$$COP = \frac{CC}{E_{thermal} + \frac{(E_{cool} + E_{pumping})}{\beta}} \quad \textbf{(5.10)}$$

where, β equivalent conversion coefficient of electric power and thermal energy and its value is taken as 0.3 [131].

The ratio between the CC and electrical energy consumption is given by electrical coefficient of performance (ECOP) while ratio between CC to consumption of thermal energy is represented by thermal coefficient of performance (TCOP).

$$\mathrm{E}COP = \frac{CC}{E_{cool} + E_{pumping}} \quad \textbf{(5.11)}$$

$$\mathrm{T}COP = \frac{CC}{E_{thermal}} \quad \textbf{(5.12)}$$

5.8 Results and discussion

Experiments were carried out with calcium chloride as desiccant by varying the inlet operating conditions of humid air. The concentration of desiccant solution was checked before placing inside the flow channels and adjusted to the specific value. Experiments were carried out with the range of operating parameters shown in Table 5.3. Generally, high values of the determination coefficient or *R*-squared (R^2) have been obtained in all cases with a linear fitted regression line. R^2 is a statistical measure of how close the data are to the fitted regression line. The high value of R^2 indicates that data is uniformly distributed along the regression line.

Table 5.3: Operating parameters.

Parameter	Value range	Base value
Ambient air temperature (°C)	25 – 40	35
Ambient air humidity ratio (kg_v/kg_a)	0.010 – 0.025	0.020
Process air mass flux ($kg/s.m^2$)	1 – 6.5	3
Regeneration air flow rate ($kg/s.m^2$)	1 – 4	1
Regeneration temperature (°C)	55 - 90	70
Desiccant concentration (%)	40 - 44	42

5.8.1 Effect of ambient air humidity ratio

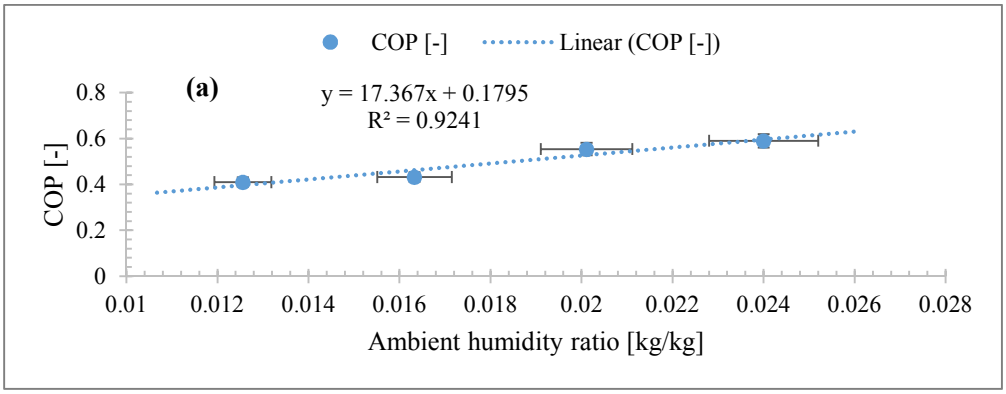
The partial vapor pressure is the governing factor for the mass transfer between process air and desiccant surface. As the inlet air humidity ratio increases, the partial pressure of water vapor in the air also increases which in turn enhances the difference between the partial

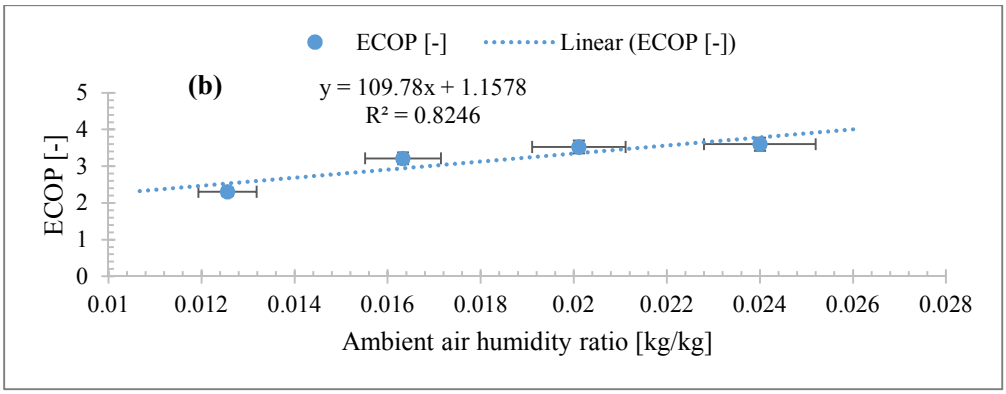
vapor pressure in the inlet air stream and on the desiccant surface resulting an increase in the moisture absorbing capacity of desiccant. The effect of the process air inlet humidity was investigated as shown in Fig. 5.5. The overall performance (COP), electrical coefficient of performance (ECOP), and thermal coefficient of performance (TCOP) of the system increases with the increase in inlet air humidity ratio as shown in Fig. 5.5(a, b, and c). The increase in humidity ratio of the ambient air results in the increased potential for mass transfer. Therefore, this increase in mass transfer enhances the capacity of the system to remove the latent load from the conditioned space. Although, with the increase in ambient air humidity required input heat will also increase for desorption of moisture but this increase is not as high as capacity of the system to remove latent load. Thus, higher the ambient air humidity, better will be the performance (COP, TCOP, and ECOP) of the system.

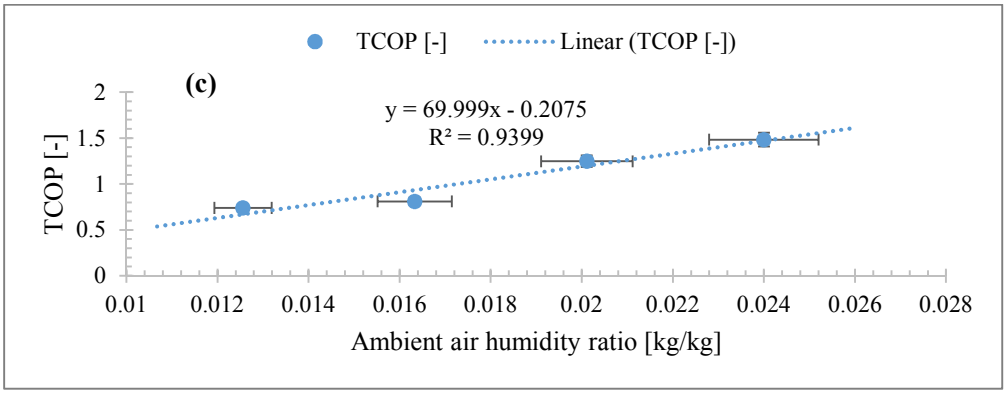
Fig. 5.5(d) shows that moisture removal rate increases with the increase in inlet air humidity ratio due to the increase in mass transfer potential. In fact, increasing the air inlet humidity ratio causes an increase in the driving force and hence increases the mass transfer potential within the dehumidifier. Increased humidity ratio indicates higher partial pressure of water vapor, which means higher tendency for water to condense and be absorbed by the desiccant material, thus larger driving force for absorption. Similar to moisture removal rate, a higher dehumidification coefficient of performance (DCOP) was achieved at higher inlet humidity ratio because of an increase in the moisture absorbing capacity of desiccant dehumidifier as shown in Fig. 5.5(e). For instance, the moisture removal rate and DCOP

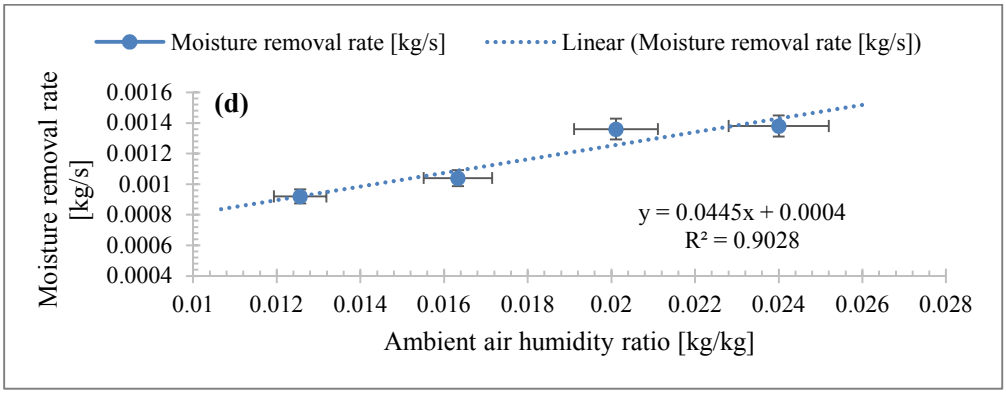
increased by 48 and 62%, respectively when the humidity ratio is increased from 0.01 to 0.025 kg/kg.

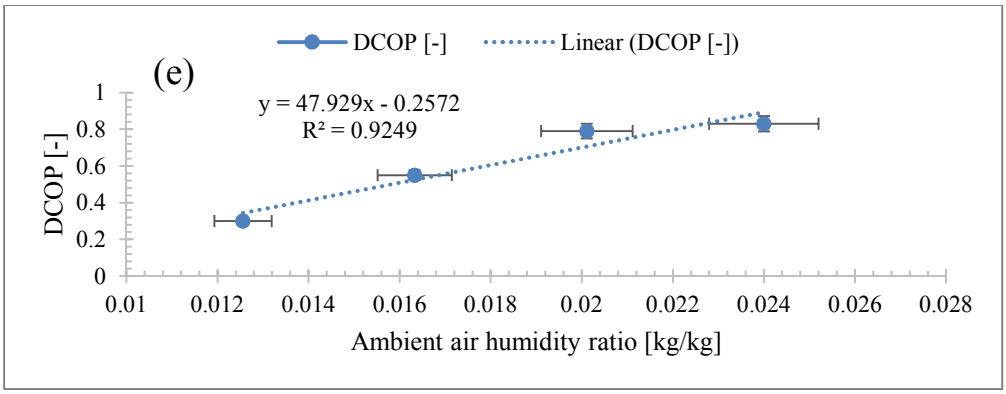
As regards to sensible energy ratio, the temperature at the exit of dehumidifier strongly increases with humidity ratio of inlet air because of larger quantity of water vapor absorbed by the desiccant. Therefore, temperature at state point 2 increases due to the increase in the released heat of absorption. Thus, increase in humidity ratio of inlet air will increase the sensible energy ratio of the system at fixed value of regeneration and supply air temperature as shown in Fig. 5.5(f).






TCOP [-]
Linear (TCOP [-])
(c)
y = 69.999x - 0.2075
R² = 0.9399
TCOP [-]
2
1.5
1
0.5
0
0.01
0.012
0.014
0.016
0.018
0.02
0.022
0.024
0.026
0.028
Ambient air humidity ratio [kg/kg]


Moisture removal rate [kg/s]
Linear (Moisture removal rate [kg/s])
(d)
y = 0.0445x + 0.0004
R² = 0.9028
Moisture removal rate [kg/s]
0.0016
0.0014
0.0012
0.001
0.0008
0.0006
0.0004
0.01
0.012
0.014
0.016
0.018
0.02
0.022
0.024
0.026
0.028
Ambient air humidity ratio [kg/kg]


DCOP [-]
Linear (DCOP [-])
(e)
y = 47.929x - 0.2572
R² = 0.9249
DCOP [-]
1
0.8
0.6
0.4
0.2
0
0.01
0.012
0.014
0.016
0.018
0.02
0.022
0.024
0.026
0.028
Ambient air humidity ratio [kg/kg]

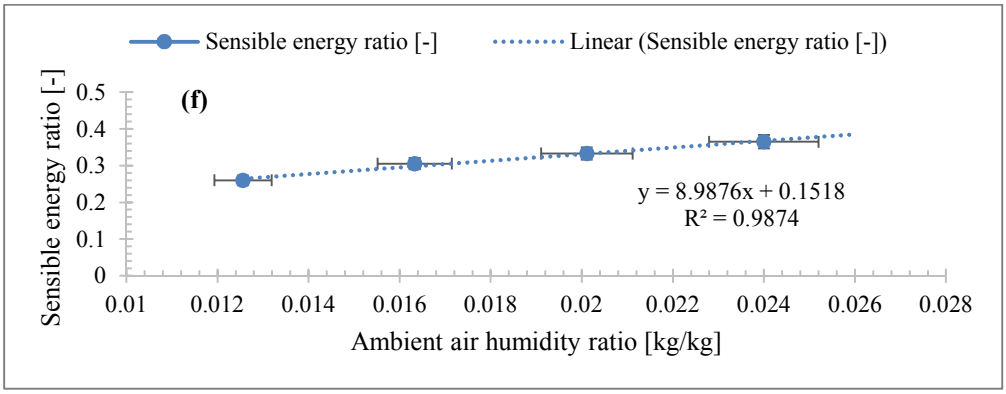


Figure 5.5: Effect of ambient air humidity ratio on: (a) COP; (b) ECOP; (c) TCOP; (d) moisture removal rate; (e) DCOP; (f) sensible energy ratio.

5.8.2 Effect of ambient air temperature

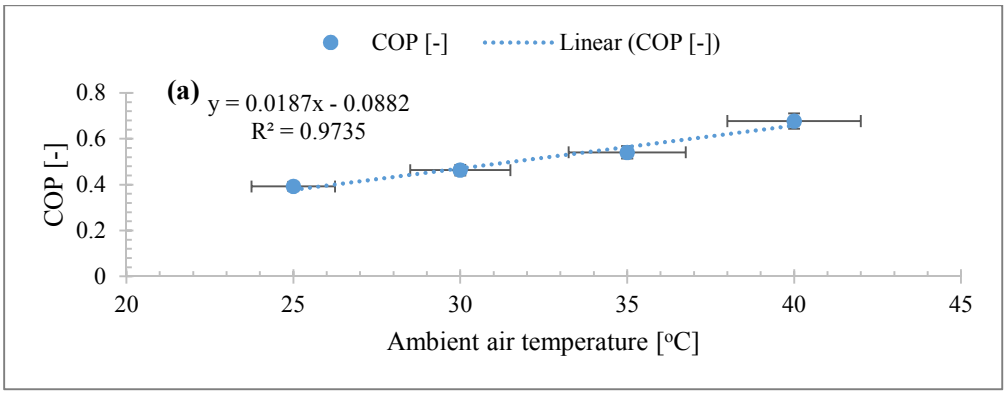
The liquid desiccant cooling system should be equipped with a highly efficient control system in order to have a better control of supply conditions according to outdoor and indoor conditions.

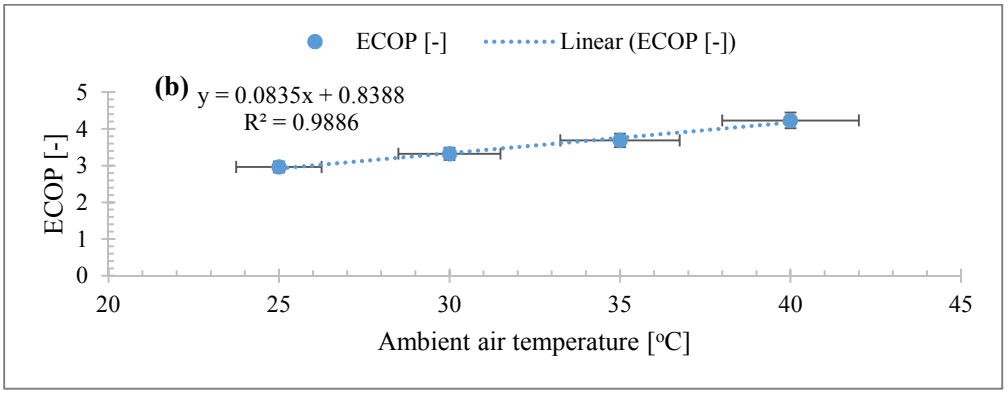
The temperature of air at the exit of dehumidifier (state point 2) in Fig. 5.1 increases as the temperature of the ambient air increases. This increase in difference of temperature, increases the difference in enthalpy of air across the rotary dehumidifier. This increased difference of enthalpy causes an increase in cooling capacity of the system. The achieved improvement in the system performance (COP, ECOP, and TCOP) as shown in Fig. 5.6(a, b, and c) is because of the fact that at higher ambient air temperature, the amount of heat and mass transfer from the air to the desiccant will be enhanced. This enhancement of heat and mass transfer will also increase the amount of heat released from the air to the desiccant during the absorption process.

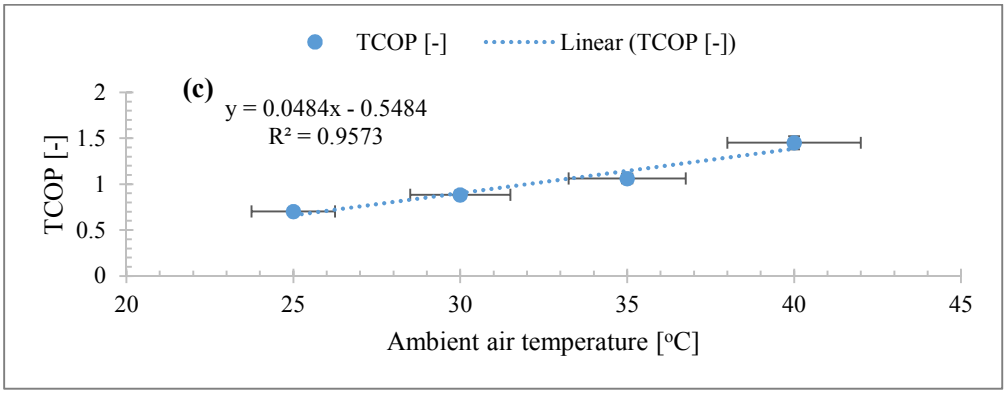
Figure 5.6(d) shows the impact of process air inlet temperature on the moisture removal rate. There is a slight decrease in the moisture removal rate when the inlet air temperature

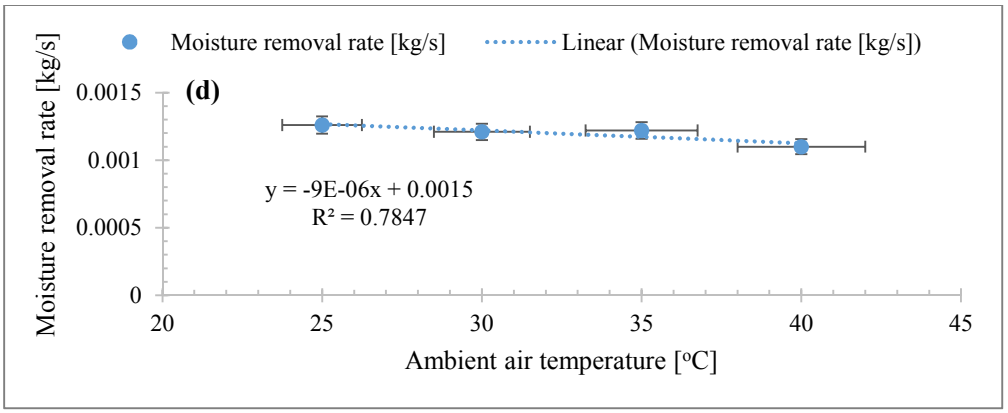
is increased. This is due to the fact that the desiccant can be cooled adequately at the lower temperature and hence leading to a high absorption capability. This may be explained as follows: as the process air inlet temperature is increased, the temperature of the desiccant inside the dehumidifier is increased along with the partial vapor pressure on the desiccant surface. When the desiccant surface vapor pressure increases, the potential of the absorption process is decreased causing air to become more humid and in turn low moisture removal rate. When temperature of ambient air is increased from 25 to 45°C, the moisture removal rate is decreased by about 15%.

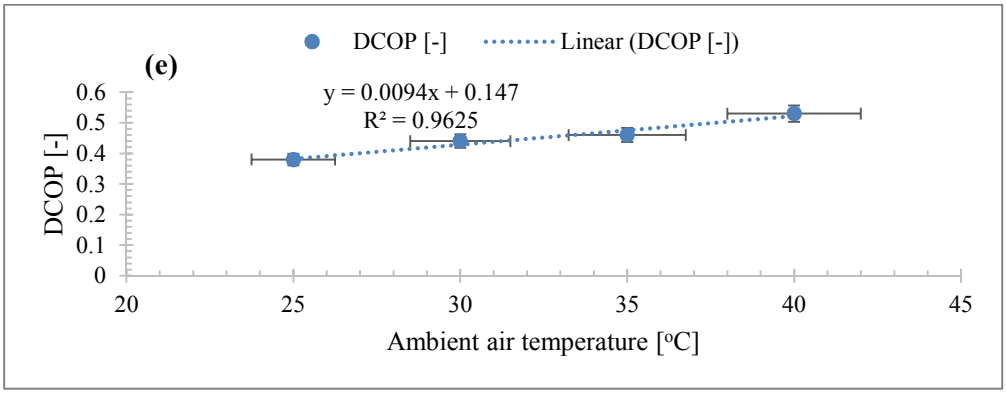
DCOP as a function of outdoor air temperature has been shown in Fig. 5.6(e). As regards to DCOP, the increase in process air inlet temperature causes a slight reduction of moisture removal capacity, but there is an increase in inlet enthalpy; therefore the heat needed to achieve a fixed regeneration temperature significantly reduces with inlet temperature, and DCOP rises. As regards to sensible energy ratio, the Fig. 5.6(f) shows similar trends as the increase in process air inlet temperature does not cause a significant change in SER.




ECOP [-]
Linear (ECOP [-])
(b)
y = 0.0835x + 0.8388
R² = 0.9886
ECOP [-]
0
1
2
3
4
5
20
25
30
35
40
45
Ambient air temperature [°C]


TCOP [-]
Linear (TCOP [-])
(c)
y = 0.0484x - 0.5484
R² = 0.9573
TCOP [-]
0
0.5
1
1.5
2
20
25
30
35
40
45
Ambient air temperature [°C]


Moisture removal rate [kg/s]
Linear (Moisture removal rate [kg/s])
(d)
y = -9E-06x + 0.0015
R² = 0.7847
Moisture removal rate [kg/s]
0
0.0005
0.001
0.0015
20
25
30
35
40
45
Ambient air temperature [°C]



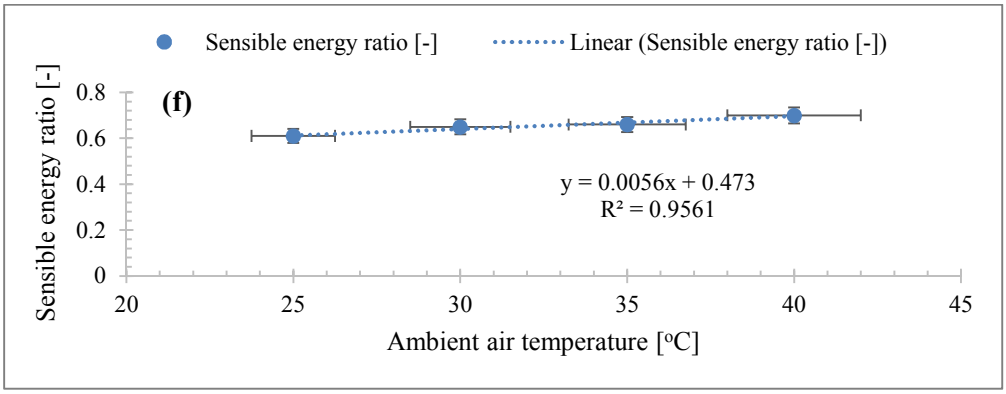


Figure 5.6: Effect of ambient temperature on: (a) COP; (b) ECOP; (c) TCOP; (d) moisture removal rate; (e) DCOP; (f) sensible energy ratio.

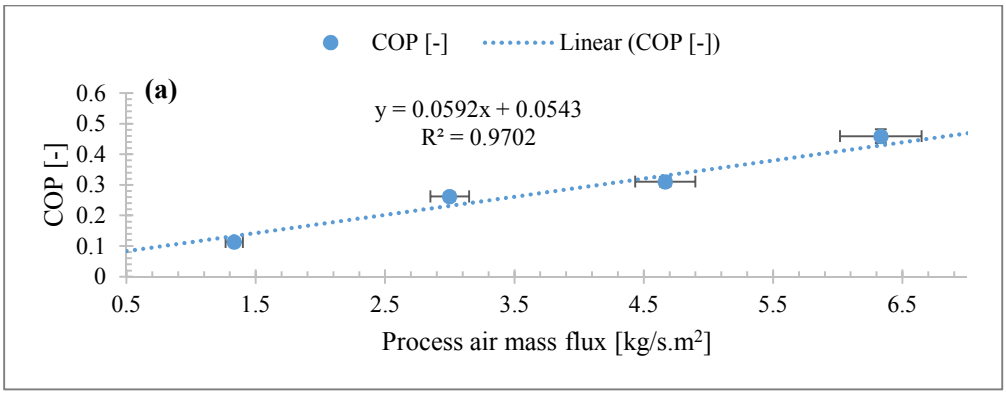
5.8.3 Effect of process air mass flux

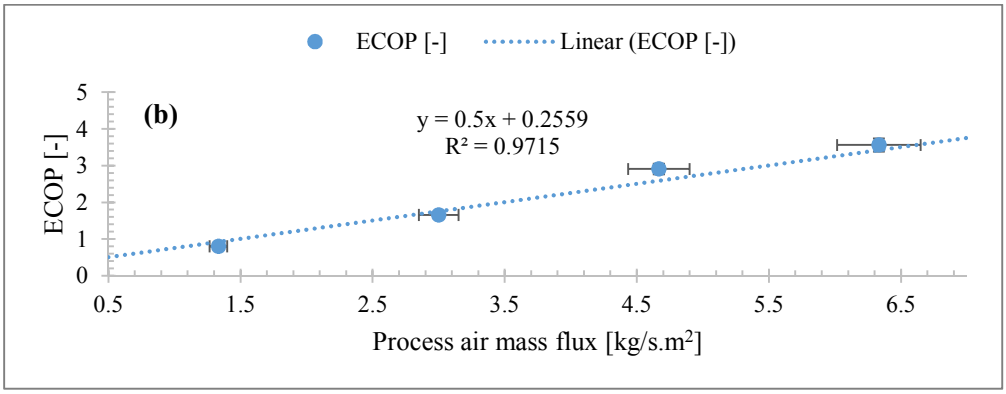
Figure 5.7 presents the effect of the process air mass flux on the overall performance of the system. As shown in Figure 5.7(a, b, and c) increasing the mass flux of air leads to an increase in system coefficient of performance (COP, ECOP, and TCOP). This is because of the increase in cooling capacity of the system while keeping the input energy constant. It is to be noted that the mass flux of regeneration air have been kept constant and hence, the required thermal load has not been increased. The optimum values of process and regeneration air flux are very important for the best performance of the system. Due to high mass flow rate of process air and low mass flow rate of regeneration air, the absorbed

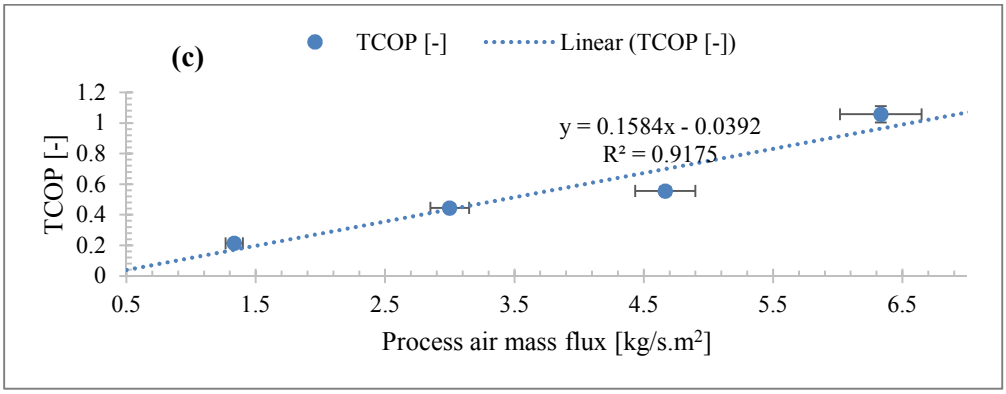
moisture will not be efficiently removed from the dehumidifier. Similarly, at low flow rate of process air and high flow rate of regeneration air, the required heat input will be increased to high level as compared to cooling capacity and it can also dry the desiccant before the regeneration process is being completed.

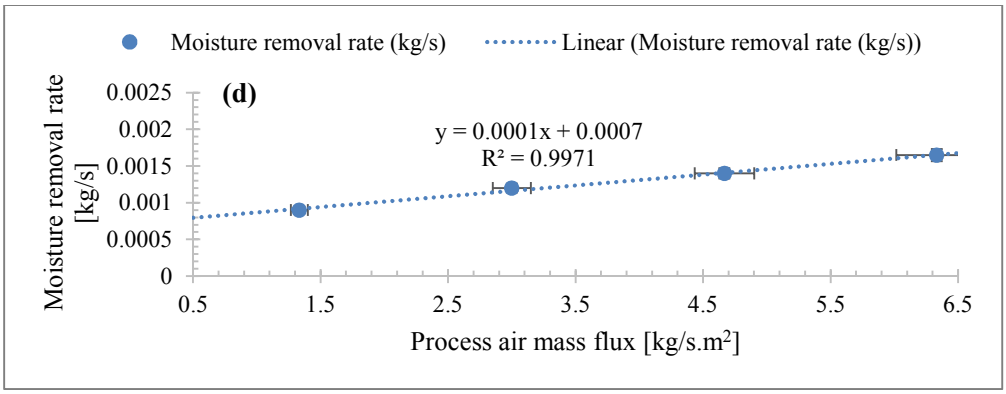
Variations of the moisture removal rate and DCOP with the process air mass flux are plotted in Fig. 5.7(d and e). The moisture removal rate increases with the increase in process air mass flux because of the increase in driving force for moisture transfer. However, the total moisture removal capacity decreases due less residence time at higher flow rates. Thus, DCOP decreases with mass flux of process air as illustrated in Fig. 5.7(e).

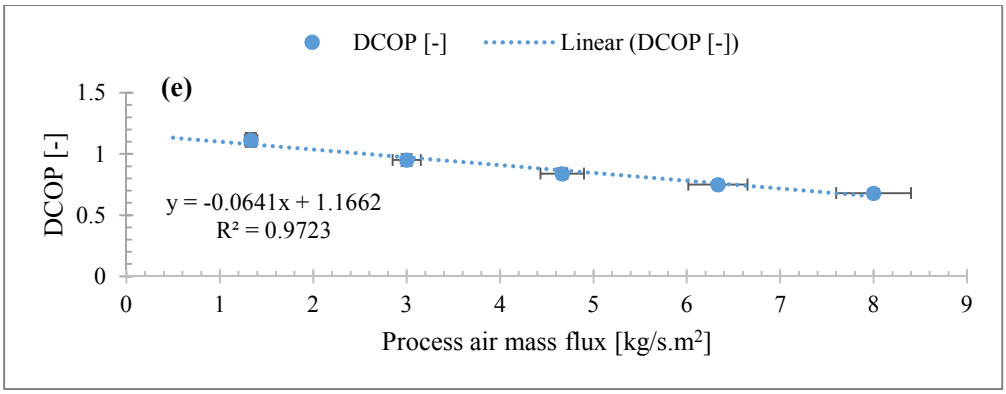
As increasing the inlet flow rate of process air leads to an increase in both the supply air humidity ratio and temperature inside the dehumidifier. Thus, keeping the regeneration temperature constant, sensible energy ratio will increase with inlet process air mass flux as shown in Fig. 5.7(f). This is basically due to the larger amount of air introduced to the dehumidification unit and thus the capability of the dehumidifier to reduce the humidity of the air decreases.




ECOP [-]
Linear (ECOP [-])
(b)
y = 0.5x + 0.2559
R² = 0.9715
ECOP [-]
Process air mass flux [kg/s.m²]


TCOP [-]
Linear (TCOP [-])
(c)
y = 0.1584x - 0.0392
R² = 0.9175
TCOP [-]
Process air mass flux [kg/s.m²]


Moisture removal rate (kg/s)
Linear (Moisture removal rate (kg/s))
(d)
y = 0.0001x + 0.0007
R² = 0.9971
Moisture removal rate [kg/s]
Process air mass flux [kg/s.m²]


DCOP [-]
Linear (DCOP [-])
(e)
y = -0.0641x + 1.1662
R² = 0.9723
DCOP [-]
Process air mass flux [kg/s.m²]

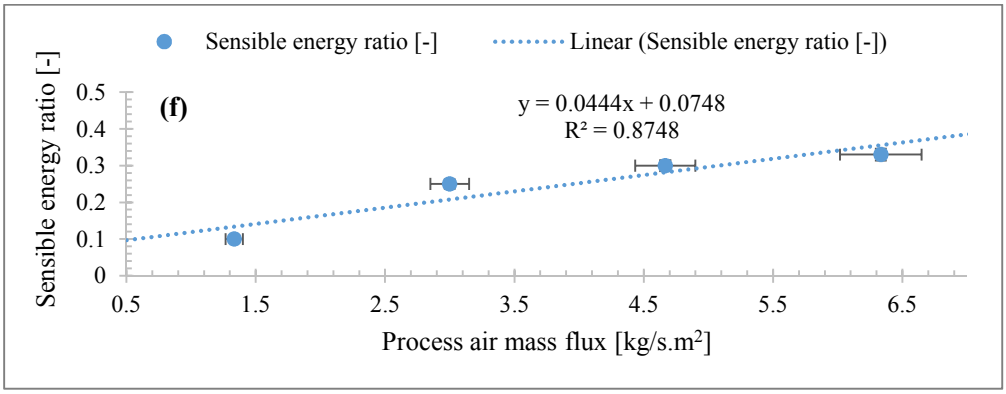


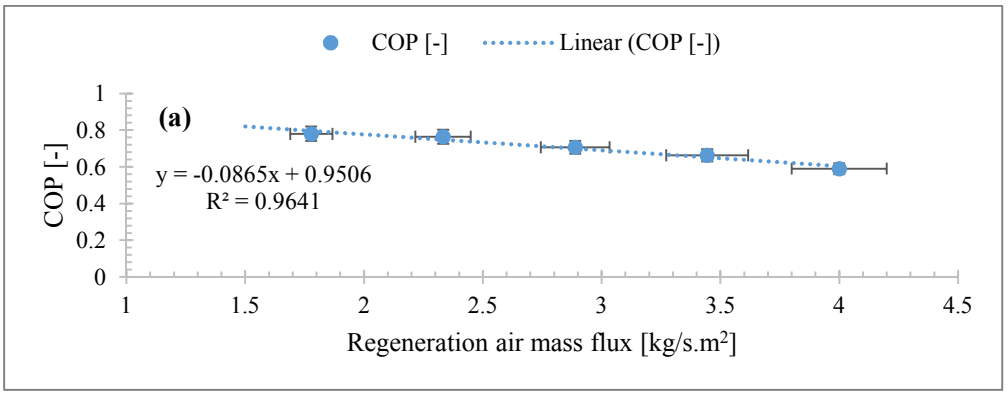
Figure 5.7: Effect of process air mass flux on: (a) COP; (b) ECOP; (c) TCOP; (d) moisture removal rate; (e) DCOP; (f) sensible energy ratio.

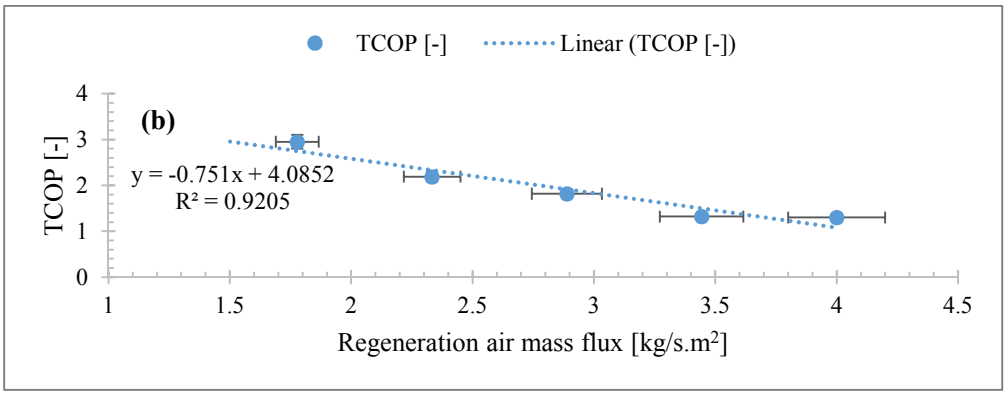
5.8.4 Effect of regeneration air mass flux

The flow rate of regeneration air and regeneration temperature have a strong impact on the performance of desiccant cooling system. The performance improvement of the system is expected at lower regeneration temperature and mass flow rate of regeneration air since both of these factors have direct effect on required input thermal energy.

The influence of regeneration air flow rates on performance of the system is shown in Fig. 5.8. In this case the value of process air flow rate is kept constant at the base value shown in Table 5.3 while mass flow rate of regeneration air is varied. The COP and TCOP decreases by 25 and 60%, respectively with the regeneration air flow rate increasing from 1.5 to 5 kg/m^2.s as shown in Fig. 5.8(a and b). This may be explained as follows: when flow rate of regeneration air increases, the input energy for regeneration will increase which causes a decrease in the performance of the system. Consequently, more thermal energy is consumed to condition the regeneration air to the specified temperature before entering the regeneration portion. In Fig. 5.8(c and d), DCOP and SER as a function

regeneration air mass flow rate are shown. The increase in mass flow rate of regeneration air causes a reduction in DCOP and it is due to a proportional rise in energy required for regeneration. As regards to SER, the increase in regeneration air flow rate determines a rise in absorbed moisture and heat of absorption. This causes an increase in process air outlet temperature at the exit of desiccant wheel. Thus, SER decrease with the increase in mass flow rate of regeneration air as shown in Fig. 5.8(d).





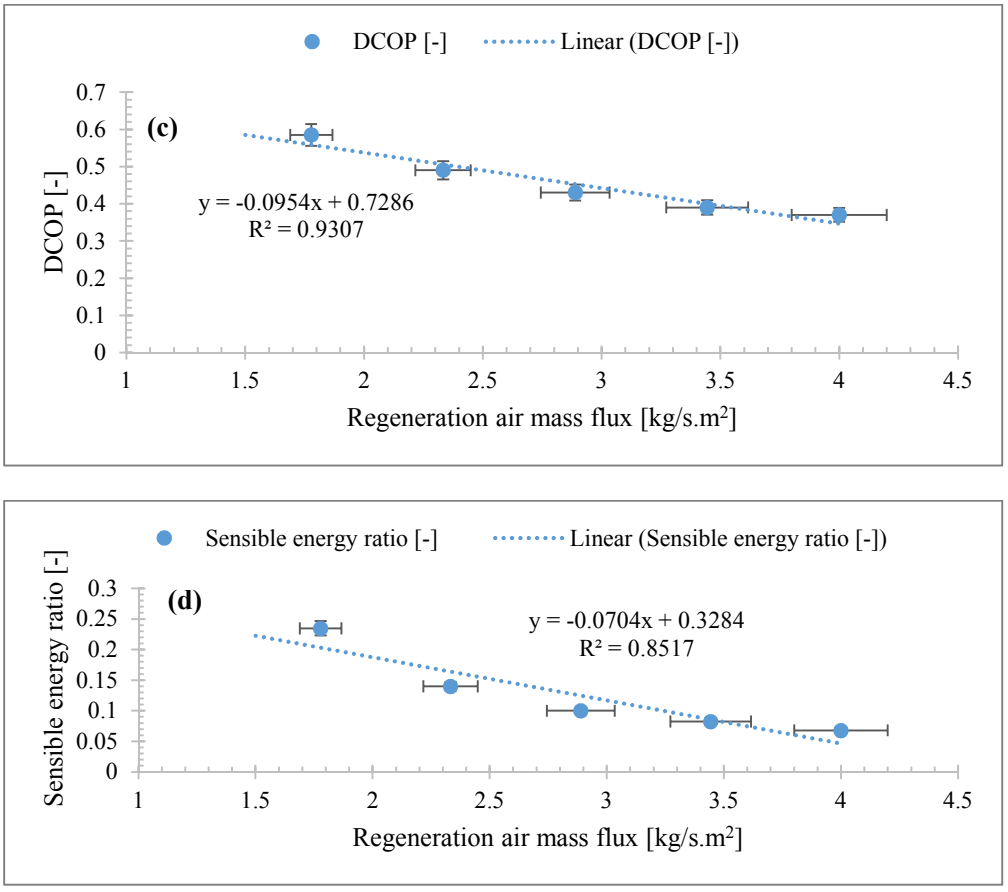


Figure 5.8: Effect of regeneration air mass flux on: (a) COP; (b) TCOP; (c) DCOP; (d) sensible energy ratio.

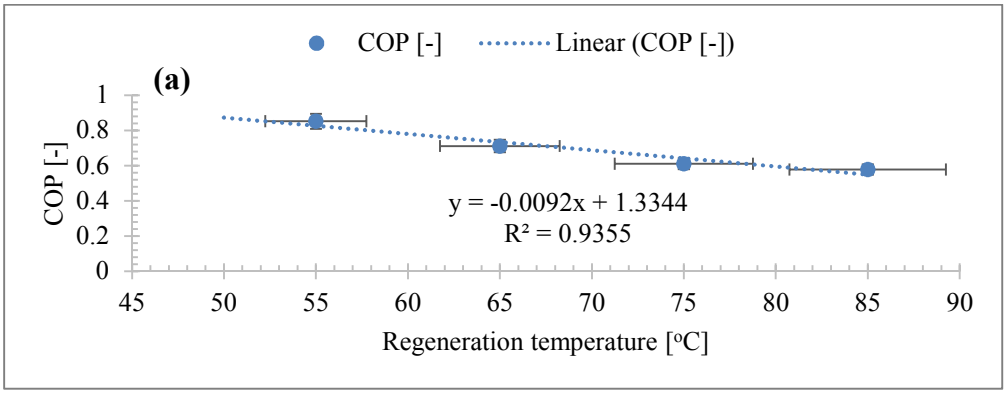
5.8.5 Effect of regeneration temperature

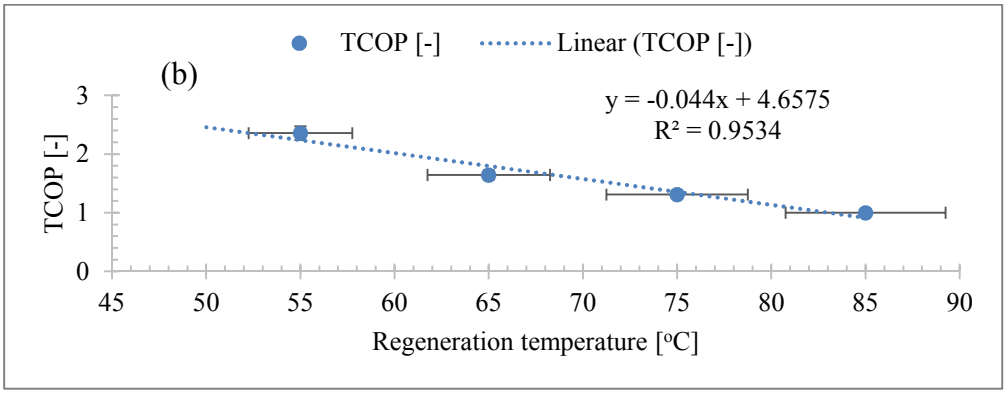
As mentioned earlier, the regeneration temperature is an important parameter for liquid desiccant cooling system because it affects the performance of the overall system significantly. The thermal energy input increases with the increase in regeneration temperature which will decrease both overall COP and TCOP of the system as shown in Fig. 5.9(a and b). The regeneration air inlet temperature is varied while the temperature and humidity ratio of inlet air remained constant. Although the higher value of regeneration temperature will make the moisture removal process faster, this higher value of temperature

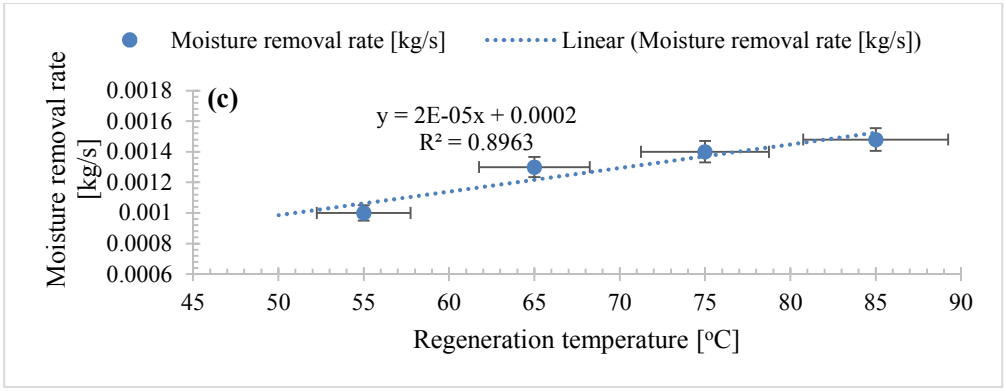
will dry up the desiccant inside the dehumidifier before the completion of regeneration period. Hence, some added energy will be wasted and will not be utilized during absorption period for moisture removal. Secondly, the higher value of regeneration temperature will increase the input energy which decreases the COP of the cycle. The COP and TCOP of the system decreases from 0.91 to 0.58 and 3.4 to 1, respectively, when regeneration temperature increases from 50 to 85^{o}C.

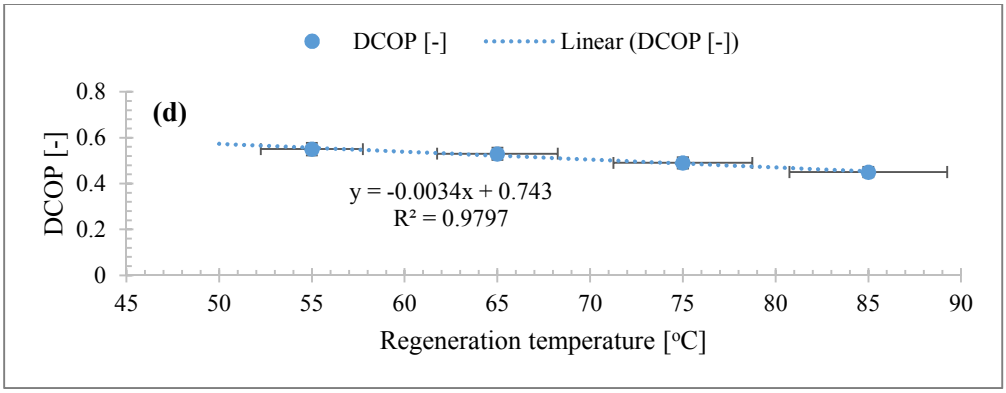
The moisture removal rate as well as DCOP of the desiccant dehumidifier has been plotted as a function of regeneration temperature in Fig. 5.9(c and d), which shows that the moisture removal capacity increases with regeneration temperature, whereas DCOP decreases with an increase in regeneration temperature. In terms of DCOP, the increase in moisture removal at higher regeneration temperature is not enough to balance the rise in the input thermal energy per unitary mass flow rate. Hence, DCOP decreases with an increasing regeneration temperature.

The sensible energy ratio is decreased with the increase in regeneration temperature as shown in Fig. 5.9(e). This decrease of SER is because of increase in difference between regeneration and process air inlet temperatures. In terms of thermal performances of the wheel, the rise in process air temperature at outlet of dehumidifier with regeneration temperature is because of the increased heating of the desiccant material in the regeneration section and consequently in the process section when the dehumidifier rotates from the former to the latter; but the increase in regeneration temperature itself causes a reduction of SER, as expected from Eq. (5.8).


COP [-]
Linear (COP [-])
(a)
COP [-]
y = -0.0092x + 1.3344
R² = 0.9355
Regeneration temperature [°C]


TCOP [-]
Linear (TCOP [-])
(b)
TCOP [-]
y = -0.044x + 4.6575
R² = 0.9534
Regeneration temperature [°C]


Moisture removal rate [kg/s]
Linear (Moisture removal rate [kg/s])
(c)
Moisture removal rate [kg/s]
y = 2E-05x + 0.0002
R² = 0.8963
Regeneration temperature [°C]


DCOP [-]
Linear (DCOP [-])
(d)
DCOP [-]
y = -0.0034x + 0.743
R² = 0.9797
Regeneration temperature [°C]

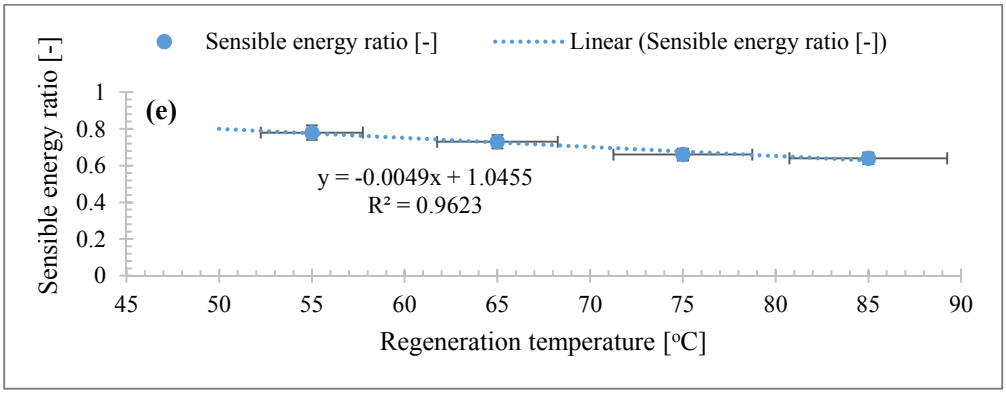


Figure 5.9: Effect of regeneration temperature on: (a) COP; (b) TCOP; (c) moisture removal rate; (d) DCOP; (e) sensible energy ratio.

5.8.6 Effect of desiccant concentration

Figure 5.10 shows the effect of desiccant concentration on the moisture removal rate, DCOP and sensible energy ratio of the dehumidifier. When initial concentration of desiccant is increased, moisture removal rate and DCOP are increased as shown in Fig. 5.10(a, b). When desiccant concentration increases, vapor pressure at the desiccant surface is reduced leading to low humidity ratio of air at the exit of desiccant dehumidifier which in turn increases moisture removal rate as shown in Fig. 5.10(a). Figure 5.10(b) shows the effect of desiccant concentration on the system DCOP. As concentration increases, DCOP is increased. Increasing concentration will increase the affinity of desiccant to absorb moisture, leading to an increase in DCOP. This increase in DCOP is because of the more moisture absorbed and latent load controlled by the desiccant wheel at higher concentration of desiccant. An increase of concentration from 40 to 45% will increase the DCOP by about 25%.

The temperature at the exit of desiccant dehumidifier increases with the increase in desiccant concentration because of more released heat of absorption. Thus, at constant

value of regeneration temperature, sensible energy ratio increase with the increase in desiccant concentration as shown in Fig. 5.10(c).

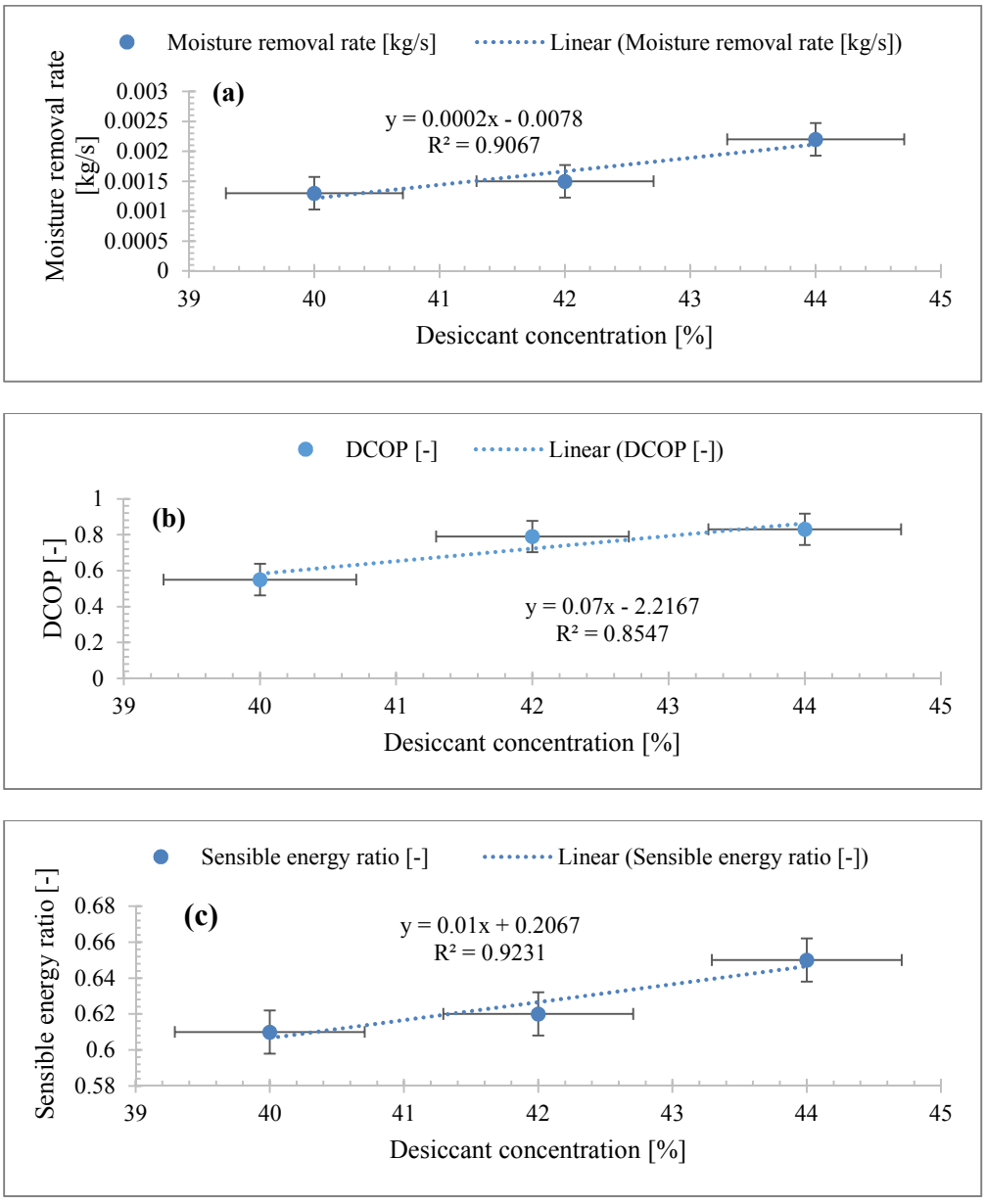
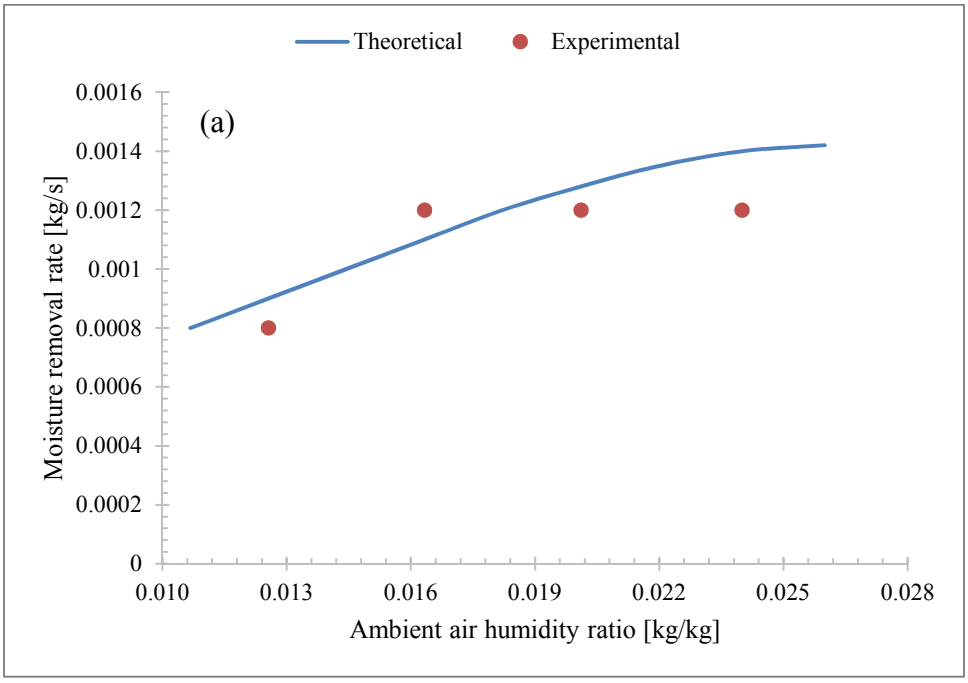


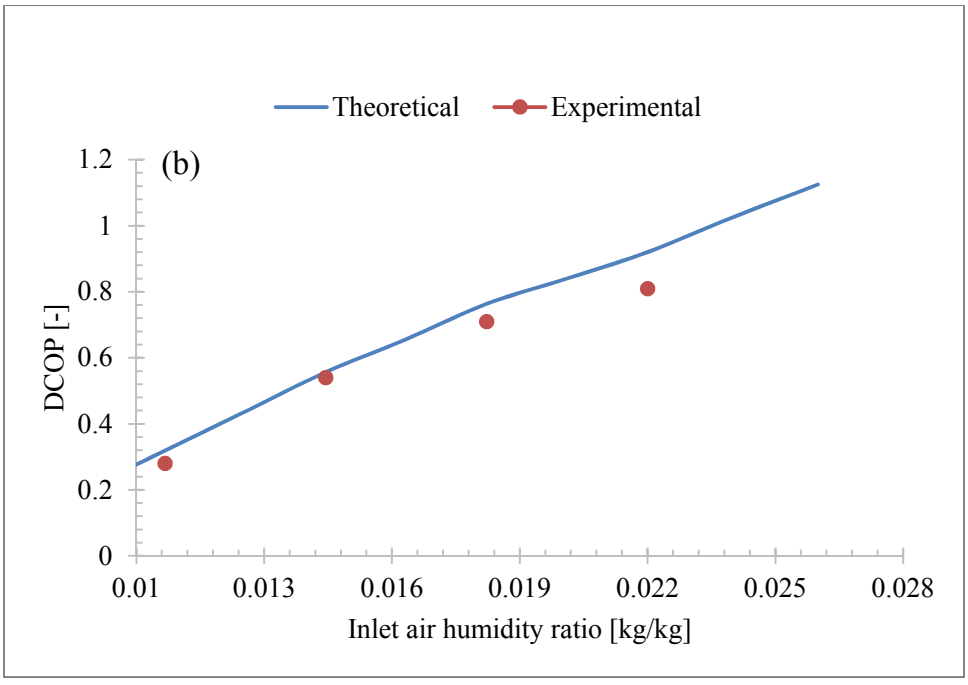
Figure 5.10: Effect of desiccant concentration on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio.

5.9 Comparison of heat and mass transfer model of liquid desiccant dehumidifier with experimental data

Figures 5.11 to 5.14 show the comparison between the experimental and modeling results for the heat and mass transfer performance of desiccant dehumidifier. The comparative results for the effect of ambient air humidity ratio on moisture removal rate, DCOP, and sensible energy ratio are shown in Fig. 5.11. The maximum difference between modeling and experimental results are 10, 10, and 8% for moisture removal rate, DCOP, and sensible energy ratio, respectively. The modeling results for effect of ambient temperature on different performance parameters also shows good agreement with the experimental data as shown in Fig. 5.12. A maximum difference of 5% for moisture removal rate, 5% for DCOP, and 4% for sensible energy ratio is observed between modeling and experimental results for the effect of ambient temperature.

According to the results shown in Figs. 5.13 and 5.14, the theoretical results are in reasonably good agreement with the experimental data for effect of process air mass flux and regeneration temperature, respectively. The maximum difference between modeling and experimental results for the effect of process air mass flux on moisture removal rate, DCOP, and sensible energy ratio are 7, 9, and 10%, respectively. For the effect of regeneration temperature, the maximum difference between modeling and experimental results are 5, 8, and 6% for moisture removal rate, DCOP, and sensible energy ratio, respectively. Based on this comparison, the mathematical model can give good predictions and can be used to estimate the dehumidification capacity of the system.


Theoretical
Experimental
(a)
Moisture removal rate [kg/s]
0.0016
0.0014
0.0012
0.001
0.0008
0.0006
0.0004
0.0002
0
0.010
0.013
0.016
0.019
0.022
0.025
0.028
Ambient air humidity ratio [kg/kg]


Theoretical
Experimental
(b)
DCOP [-]
1.2
1
0.8
0.6
0.4
0.2
0
0.01
0.013
0.016
0.019
0.022
0.025
0.028
Inlet air humidity ratio [kg/kg]

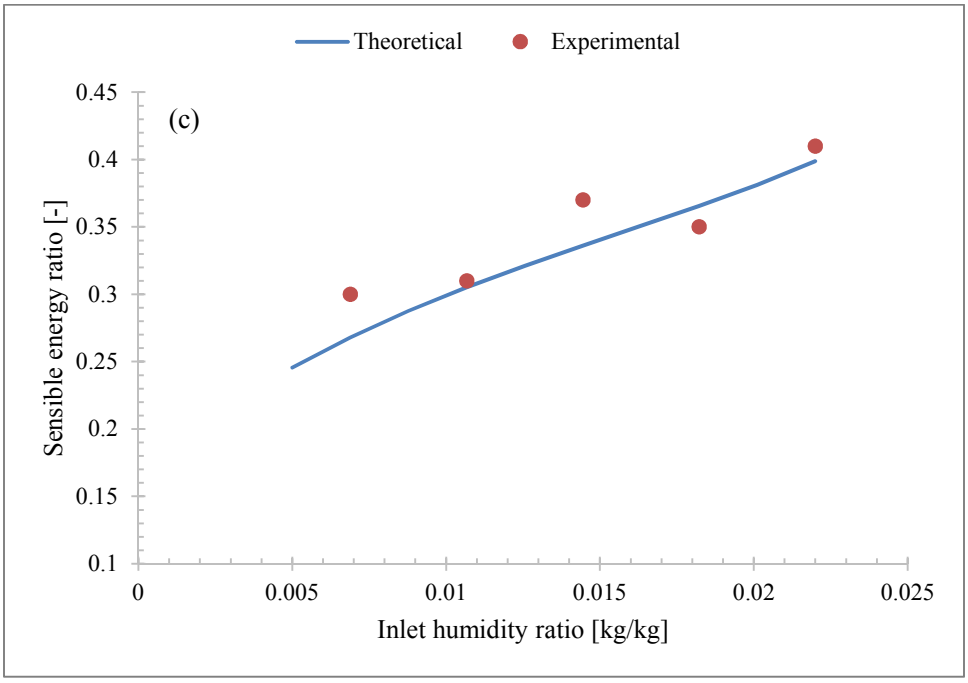
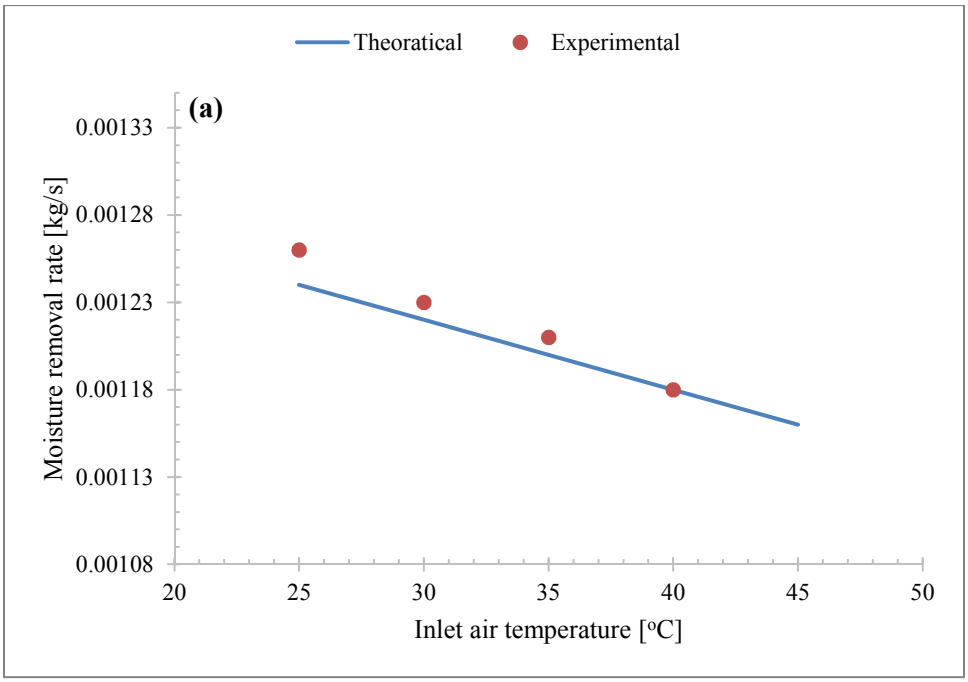
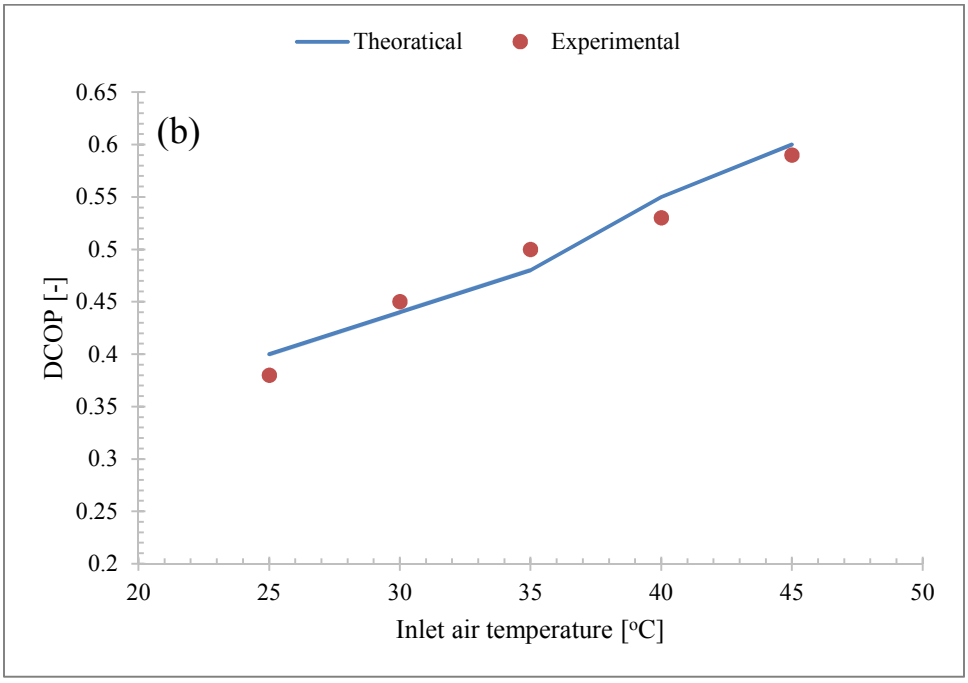


Figure 5.11: Comparative results for effects of ambient air humidity ratio on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio.





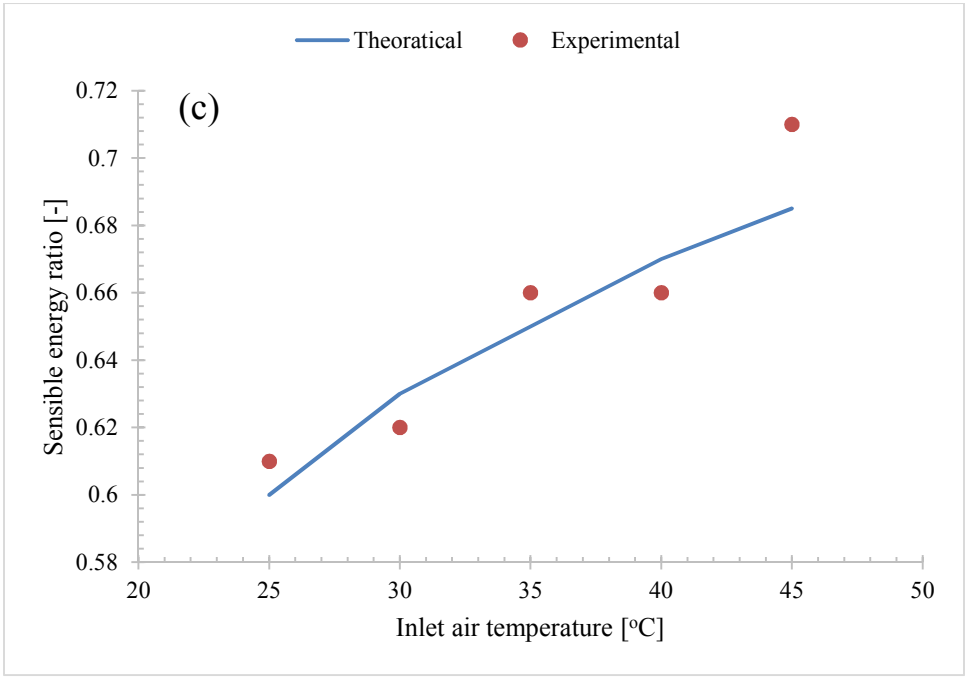
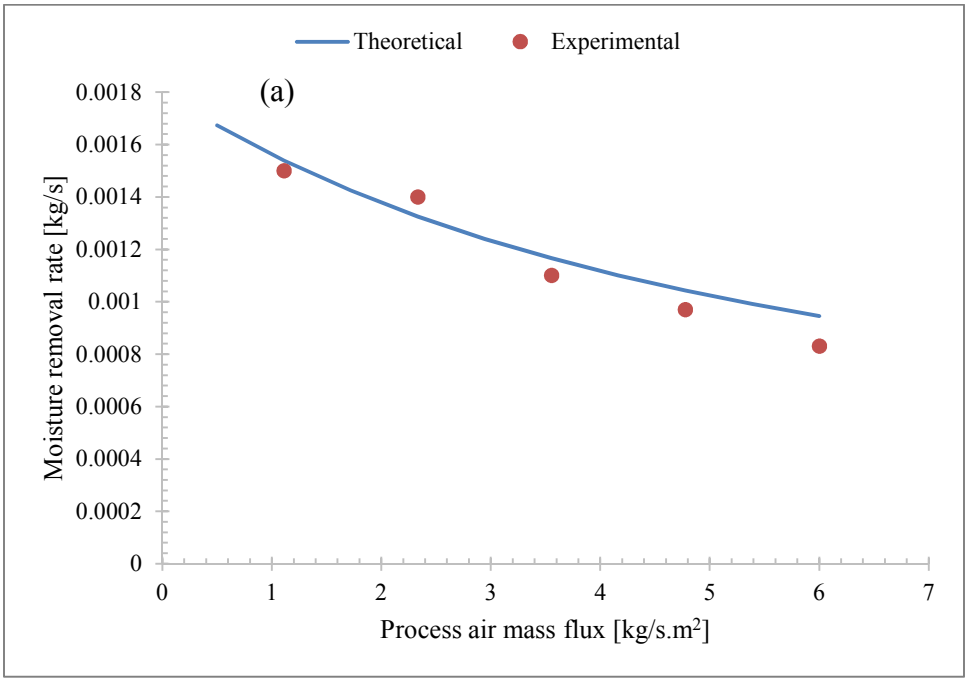
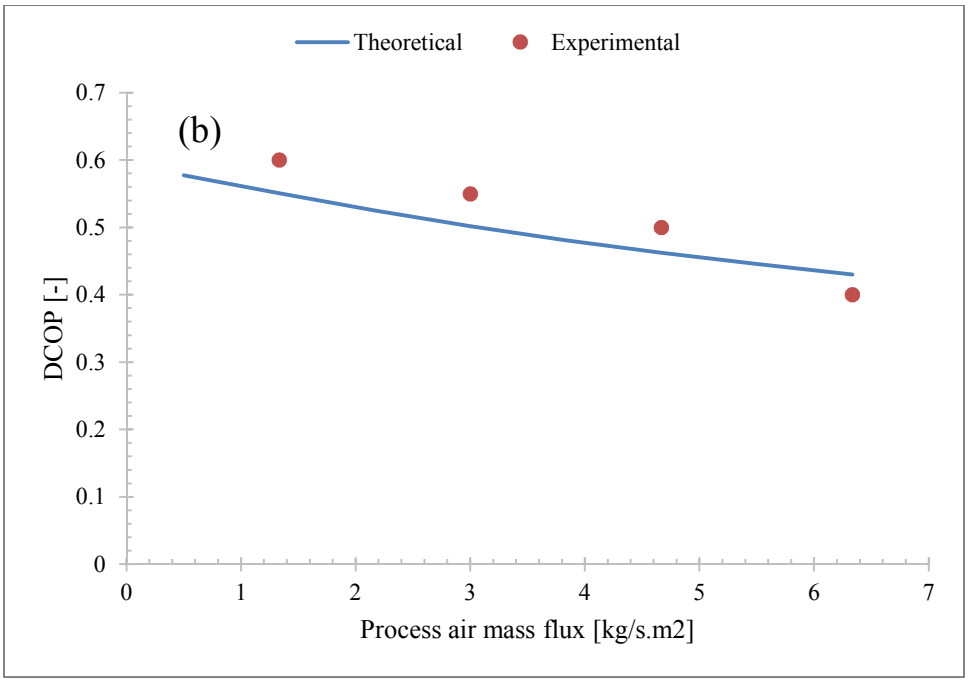


Figure 5.12: Comparative results for effects of ambient air temperature on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio.


Theoretical
Experimental
(a)
Moisture removal rate [kg/s]
0.0018
0.0016
0.0014
0.0012
0.001
0.0008
0.0006
0.0004
0.0002
0
0
1
2
3
4
5
6
7
Process air mass flux [kg/s.m²]


Theoretical
Experimental
(b)
DCOP [-]
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0
1
2
3
4
5
6
7
Process air mass flux [kg/s.m2]

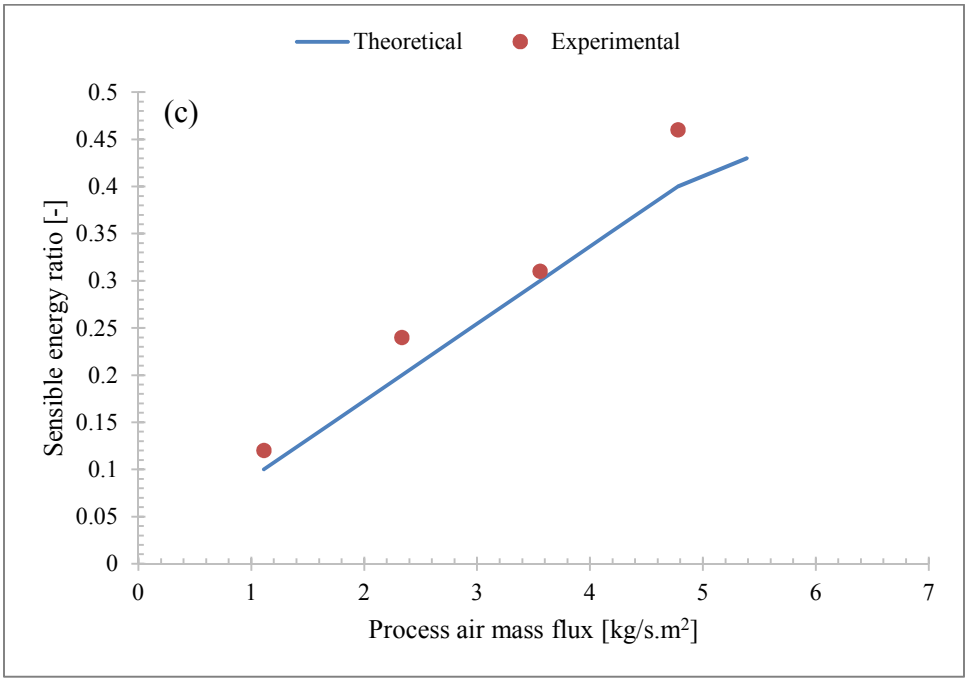
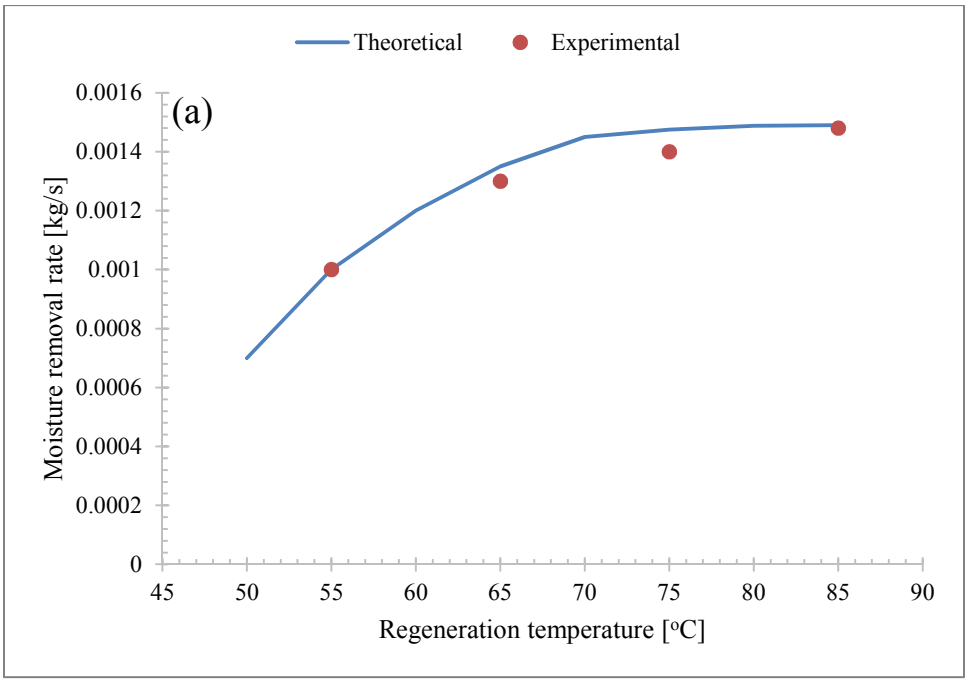
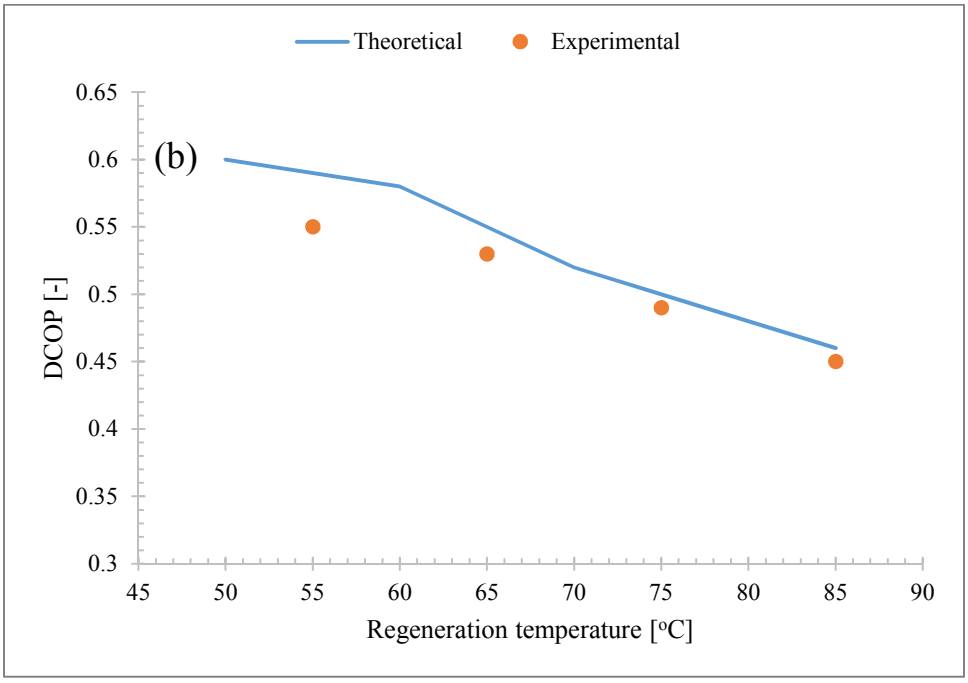


Figure 5.13: Comparative results for effects of process air mass flux on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio.





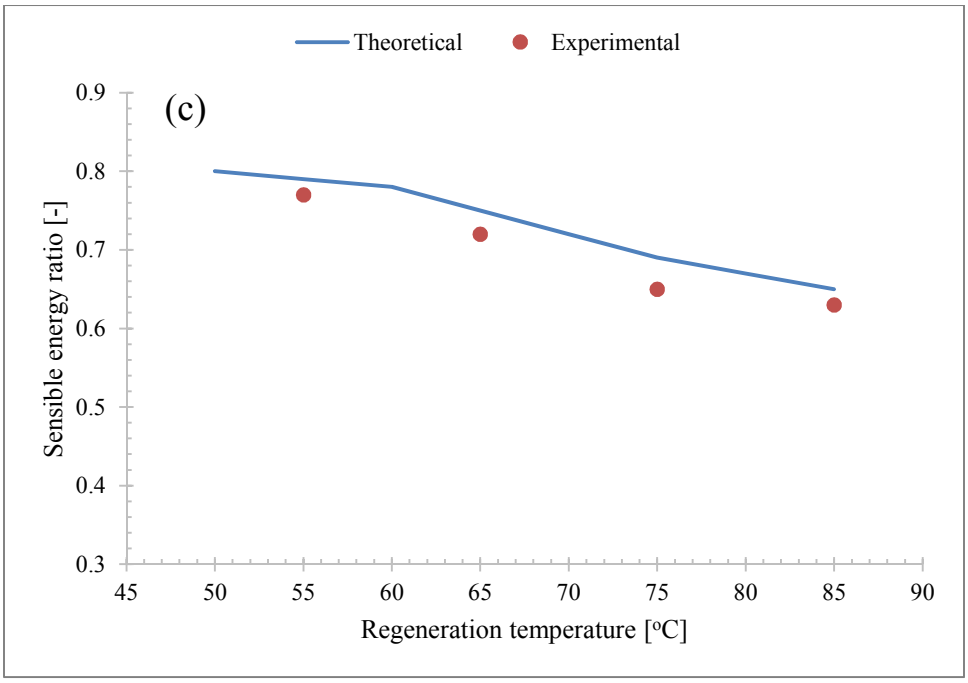


Figure 5.14: Comparative results for effects of regeneration temperature on: (a) moisture removal rate; (b) DCOP; (c) sensible energy ratio.

5.10 Conclusions

The performance of rotary liquid desiccant cooling system was investigated experimentally and is presented in this chapter. The effects of different performance parameters such as ambient air humidity ratio and temperature, process and regeneration air mass flux, regeneration temperature, and desiccant concentration have been investigated. The obtained results show that sensible energy ratio and dehumidification performance (DCOP) of the system increases with increase in ambient air humidity ratio. This is because of the increase in driving force for mass transfer between air and desiccant surface.

When the regeneration temperature increases, the DCOP decreased due to increase in input energy at higher regeneration temperature. The SER was lower at higher regeneration temperatures because of the significant differences between the regeneration and the process inlet temperatures. A higher regeneration temperature would bring a reduced marginal benefit, with respect to the latent performance. The COP, ECOP, and TCOP are largely affected by ambient air conditions, mass flow rates, regeneration temperature, and desiccant concentration. The other major findings can be summarized as:

- It has been found that the effective control of dehumidification capacity can be achieved by regulating the temperature and humidity of the air entering the dehumidifier.
- The higher the ambient air humidity, the higher the cooling capacity of the system and the lower the thermal energy demand for the system. Therefore, the COP, ECOP and TCOP increase with ambient air humidity ratio.

- The lower value of regeneration temperature is beneficial for the performance of the system, as it defines the required input energy.
- The increase in process air inlet temperature causes an increase in DCOP because the heat needed to achieve a fixed regeneration temperature significantly reduces with inlet temperature.
- The moisture removal rate increases with the increase in process air mass flux because of increase in driving force for moisture transfer. However, the total moisture removal capacity decrease because of less residence time at higher flow rate which causes a decrease in DCOP of the system.
- When concentration of desiccant is increased, moisture removal rate and DCOP are increased, but the sensible energy ratio is decreased.

CHAPTER 6

FEASIBILITY ANALYSIS OF SOLID DESICCANT COOLING SYSTEM UNDER CLIMATIC ZONE OF DHAHRAN, SAUDI ARABIA

The control of air conditioning latent and sensible load separately by using a desiccant dehumidifier operating in conjunction with evaporative cooler can reduce the air conditioning power requirement effectively. This system can also utilize the alternative resources of energy that is, solar, waste heat, and natural gas. In this chapter a solar powered solid desiccant cooling system composed of silica gel desiccant wheel is studied theoretically and performance of the system is analyzed for its feasibility analysis under the climatic conditions of Dhahran, Saudi Arabia for three months (April, July, and October) based on constant effectiveness of heat exchanger and evaporative cooler. The performance of the system is also studied under different values of process air flow rates (1 - 3 kg/s), ratio of regeneration to process air flow rates (0.33 - 1), and regeneration temperatures (120 -200^{o}C). Performance curves have been drawn to get optimum values of parameters under different operating conditions. The results showed that the proposed system is feasible for the climatic conditions of Dhahran, Saudi Arabia. It has also been observed that ratio of regeneration to process air flow rate affects the system performance remarkably, because it directly affects the regeneration load and COP of the system. The maximum value of required regeneration load was 118, 59, 38 kW at mass flow rate ratios

of 1, 0.5, 0.33, respectively. In order to lower the manufacturing cost of the system only one evaporative cooler was utilized to cool both process and regeneration air.

6.1 Introduction

Two types of loads (latent and sensible) need to be controlled in any air conditioning system to achieve comfort temperatures of 20-26^{o}C and relative humidity of 30-60% for human comfort. The conventional air conditioning systems can control the sensible load effectively but in order to control the latent load, some alternative cooling methods are required. Desiccant cooling system (DCS) is a suitable and cost effective alternate, especially in climates with high latent loads [237, 238]. The desiccant cooling technology can be utilized as standalone or it can be combined with conventional air conditioning system.

In a conventional system large amount of energy is consumed to cool down the air below its dew point temperature to remove the moisture from the air and then to reheat the air up to the desired supply temperature. While in a hybrid desiccant cooling system, adsorbent with high affinity for water vapor (silica gel, titanium dioxide, etc.) is used to extract the water from the air and excessive energy consumed to cool and reheat the air can be saved which greatly enhances the overall energy efficiency of the system [239].

Desiccant cooling system limits the use of primary energy by utilizing alternative resources of low grade thermal energy such as, solar energy, waste heat, etc. Some other advantages of desiccant cooling technology are [240]; (i) low consumption of electrical energy; (ii) eliminates the use of Chlorofluorocarbons which causes depletion of ozone layer; (iii)

better control of humidity because latent and sensible loads are controlled separately; (iv) better indoor air quality by controlling the growth of harmful fungi and bacteria. Arundel et al. [241] graphically represented a suitable range of relative humidity for the growth of different fungi and bacteria as shown in Fig. 6.1.

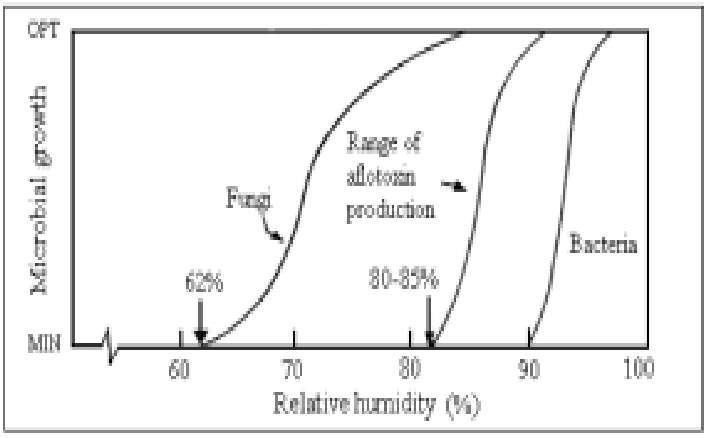


Figure 6.1: Microbial growth as a function of relative humidity [241].

6.2 Description of solid desiccant cooling cycles

Commonly used solid desiccant cooling cycles are:

- Ventilation cycle
- Recirculation cycle
- Dunkle cycle
- Recirculation ventilated cycle
- Wet-surface heat exchanger (WSHE)
- Three mixed-mode cycles

The recirculation cycle utilizes 100% recirculation of air, which means the returned air from the room is mixed with the process air at the inlet of desiccant wheel. Dunkle

recirculation cycle [60] has an additional heat exchanger with 100% recirculation of air through the dehumidifier while for regeneration of desiccant wheel, outdoor air is used. The recirculation and Dunkle cycles are 100% recirculation which is practically not much effective and desirable. However, in both cycles return air can be mixed with ventilation air according to the requirement. Both recirculation and Dunkle cycles were also analyzed with 10% ventilation air mixed with the return air, this cycle is called as recirculation ventilated and Dunkle ventilated cycle.

Maclaine-Cross and Banks [62] developed an idea of wet-surface heat exchanger (WSHE), in which water indirectly cools the process air. The extraction which is the ratio of circulated air and return air is kept about 50%. The wet surface heat exchanger cycle is actually an ideal cycle with two wet surface heat exchangers. In advanced WSHE cycle along with two wet surface heat exchangers, two sensible heat exchangers are used. Three mixed-mode cycles reported by Maclaine-Cross [242] and Kant and Maclaine-Cross [64] in which evaporative coolers are replaced by regenerative/wet surface heat exchangers. The purpose of regenerative/wet surface heat exchanger is to decrease the dry bulb temperature without much increase in humidity of the air. According to Jain et al. [94], the cycle utilizing WSHE can operate with higher coefficient of performance compared to other cycles. The advanced WSHE cycle has maximum COP of 2.6. Dunkle cycle has better performance than ventilation and recirculation cycles.

6.3 Simulation Model

Desiccant cooling technology removes moisture from air by a process known as sorption (adsorption or absorption). Different desiccant materials are used for this process. Due to vapor pressure difference adsorption material adsorbs the water vapor from the air and for the continuous operation; moisture from the desiccant wheel is removed using thermal energy. Desiccant cooling system operating under ventilation cycle is shown in Fig. 6.2. Description of the process is as follows:

- Process 1-2: The moisture from the air is adsorbed by the silica gel desiccant wheel (DW).
- Process 2-3: The hot and dehumidified air is cooled down by exchanging heat in the heat recovery wheel with cold air flowing on the regeneration side.
- Process 3-4: Air is passed through the indirect evaporative cooler where air is further cooled down according to the room supply demand. In the indirect evaporative cooler air passes through the tubes so its humidity remains constant while temperature decreases. Air is circulated to the room to provide comfortable environment.
- Process 4-5: The dehumidified and cooled air is supplied to the room to provide the comfort conditions.
- Process 5-6: The returned room air is passed through the direct evaporative cooler to lower its temperature so that it can cool down the process air in the heat recovery wheel.

- Process 6-7: Heat is exchanged between cooled regeneration air and hot process air in heat recovery wheel (HRW). Note that, the mass flow rate of regeneration air entering at state 5 is equal to the mass flow rate of process air at state 1 for better exchange of heat between process and regeneration air.
- Process 7-8: At state 7 about 50-75% of process air is bypassed the heater so that heating load can be decreased. The rest of the air is heated up to regeneration temperature by solar collector.
- Process 8-9: Desiccant is regenerated using hot air. During process of regeneration the adsorbed moisture is desorbed to repeat the cycle again.

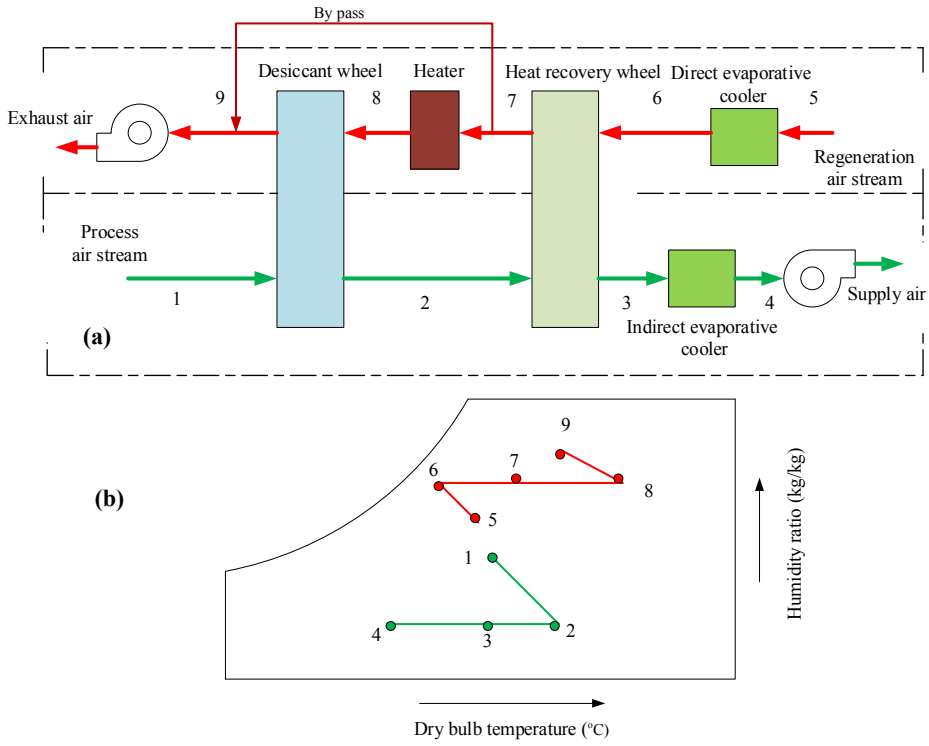


Figure 6.2: (a) Systematic solid desiccant cooling system with indirect evaporative cooler (b) psychrometric processes.

6.4 Cycle analysis

The following assumptions are made in order to perform the theoretical analysis of desiccant cooling system:

(1) Silica gel desiccant wheel has regeneration temperature of about 120-200ºC.

(2) Indirect evaporative cooler has effectiveness of about 85-90%

(3) The heat recovery wheel has effectiveness of about 80%.

(4) The flow rate of process and return air are equal but about 50 to 75% regeneration air is bypassed at state 7 to reduce regeneration heat required as shown in Fig. 6.2.

Temperature and moisture conditions at the exit from the silica gel desiccant wheel can be found from silica gel performance curves [243]. The effectiveness of the heat recovery wheel can be expresses as

$$\varepsilon_{HRW} = \frac{\dot{m}_p \times (T_2 - T_3)}{\dot{m}_r \times (T_2 - T_6)} \tag{6.1}$$

The effectiveness of both evaporative coolers is given by

$$\varepsilon_{IEC,p} = \frac{(T_3 - T_4)}{(T_3 - T_{w3})} \tag{6.2}$$

$$\varepsilon_{DEC,r} = \frac{(T_5 - T_6)}{(T_5 - T_{w5})} \tag{6.3}$$

Here T and T_w represents the dry and wet bulb temperatures, respectively. Also, from the energy and mass balances it can be written

$$T_7 = T_6 + \frac{(T_2 - T_3)}{\dot{m}_r/\dot{m}_p} \tag{6.4}$$

$$T_9 = T_8 + \frac{(T_2 - T_1)}{\dot{m}_r/\dot{m}_p} \tag{6.5}$$

$$\omega_9 = \omega_8 + \frac{(\omega_2 - \omega_1)}{\dot{m}_r/\dot{m}_p} \tag{6.6}$$

6.5 Performance Index

The system coefficient of performance (COP) is given as the ratio of the system cooling load and regeneration heat required

$$COP = \frac{\dot{Q}_c}{\dot{Q}_r} \tag{6.7}$$

The cooling load and regeneration heat is given as

$$\dot{Q}_c = \dot{m}_p \times (h_1 - h_4) \tag{6.8}$$

$$\dot{Q}_r = \dot{m}_r \times (h_8 - h_7) \tag{6.9}$$

The total capacity of moisture removal for the system is represented as the moisture absorbed from the air by the desiccant wheel.

$$M_p = \dot{m}_p \times (\omega_1 - \omega_2) \tag{6.10}$$

The desiccant wheel effectiveness on the dehumidification (process) side is given by;

$$\varepsilon_{ab} = \frac{(\omega_1 - \omega_2)}{(\omega_1 - \omega_{2,ideal})} \tag{6.11}$$

Where, $\omega_{2,ideal}$ represents the ideal specific humidity ratio. Assuming that at this point if air is completely dehumidified, we can take $\omega_{2,ideal}$=0. Similarly, on the regeneration side

$$\varepsilon_{reg} = \frac{(\omega_9 - \omega_8)}{\omega_9} \tag{6.12}$$

6.6 Results and discussion

The purpose of this theoretical investigation is to analyze the performance of a solid desiccant cooling system operating on the ventilation cycle as shown in Fig. 6.2. Theoretical results are obtained under the climatic conditions of Dhahran, Saudi Arabia for three different months. The hourly solar radiation, inlet air dry bulb temperature and

humidity ratio are listed in Tables 6.1, 6.2, and 6.3. The unknown conditions of the air at each point of the cycle are determined using equation (6.1 - 6.12) and the psychrometric chart. The built in function of Engineering Equation Solver (EES) is used to obtain the thermodynamic properties of moist and dry air. The range and base values of operating parameters have been shown in Table 6.4.

Table 6.1: Meteorological Data of Dhahran, Saudi Arabia for the month of April (15th April).

Time (hours)	Solar Radiation (W/m^2)	Dry bulb temperature (^oC)	Relative humidity (%)
06:00	33.41	19.44	75.78
07:00	228.7	20.9	56.84
08:00	463.1	21.74	57.35
09:00	685.7	24.03	38.02
10:00	847	26.72	22.94
11:00	943	27.6	17.84
12:00	987	28.05	15.9
13:00	959	28.18	15.11
14:00	814	28.46	16.11
15:00	673.6	28.53	17.87
16:00	447.8	27.38	27.39
17:00	168.9	26.23	35.12
18:00	43.81	24.82	45.98
19:00	0	23.84	51.33
20:00	0	23.85	46.49

Table 6.2: Meteorological Data of Dhahran, Saudi Arabia for the month of July (17th July).

Time (hours)	Solar Radiation (W/m^2)	Dry bulb temperature (^oC)	Relative humidity (%)
06:00	56.07	31.67	69.07
07:00	235.1	32.11	66.25
08:00	448.9	33.74	51.68
09:00	639.3	36.86	29.43
10:00	787.8	38.74	20.13
11:00	862	40.2	17.61
12:00	875	41.48	18.66
13:00	860	41.02	21.02
14:00	797.8	39.22	31.81
15:00	676.1	37.69	44.79
16:00	493.4	35.57	62.88
17:00	292.5	34.54	66.46
18:00	109.2	34.19	65.39
19:00	8.38	33.25	72.52
20:00	0	32.9	74.59

Table 6.3: Meteorological Data of Dhahran, Saudi Arabia for the month of October (15th October).

Time (hours)	Solar Radiation (W/m^2)	Dry bulb temperature (^oC)	Relative humidity (%)
06:00	7.13	25.34	25.82
07:00	145.8	25.45	26.73
08:00	394.7	26.67	29.36
09:00	427.5	28.37	31.8
10:00	676.6	30.74	32.98
11:00	797.8	32.39	27.66
12:00	827	33.26	28.5
13:00	780.6	33.63	26.04
14:00	644.7	33.78	25.62
15:00	475.4	33.91	18.88
16:00	289.4	33.61	16.81
17:00	85.3	32.72	21.59
18:00	1.274	31.61	29.23
19:00	0	30.81	35.33
20:00	0	30.19	41.78

Table 6.4: Operating parameters.

Parameter	Value range	Base value
Process air flow rate (kg/s)	0.5 – 3.5	1.62
Regeneration air flow rate (kg/s)	0.5 – 3.5	0.54
Regeneration air temperature (°C)	120 – 200	140
Desiccant material	Silica gel	-

Figure 6.3 shows the amount of hourly moisture removed from the process air during the day for the three months. The removed moisture depends on different parameters such as inlet air humidity ratio, air flow rate and the desiccant wheel material which is silica gel in this study. The removed moisture is following the same trend as the humidity of inlet air. The maximum amount of moisture removed from the process air is 0.008, 0.017 and 0.006 kg/kg for the month of April, July and October, respectively.

Figures 6.4 and 6.5 represent the variations of wheel effectiveness on process and regeneration side, respectively, during the day. It is observed that the value of effectiveness on process side was greater as compared to the regeneration side for all three months. The maximum values of wheel effectiveness on process side achieved were 0.78, 0.65 and 0.7 for the month of April, July, and October, respectively at a flow rate of 1.62 kg/s (3000CFM). While at the same operating conditions for the regeneration side the achieved effectiveness values were 0.56, 0.48 and 0.53, respectively. The wheel effectiveness is high for months with high humidity ratio of ambient air.

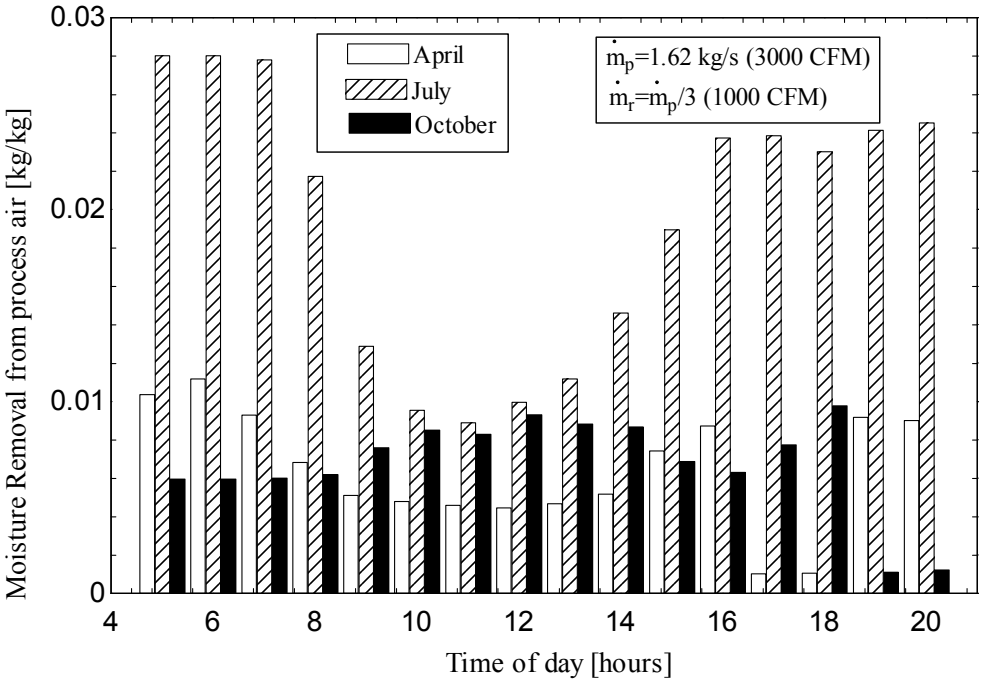


Figure 6.3: Variations of moisture removal from the process air during the day.

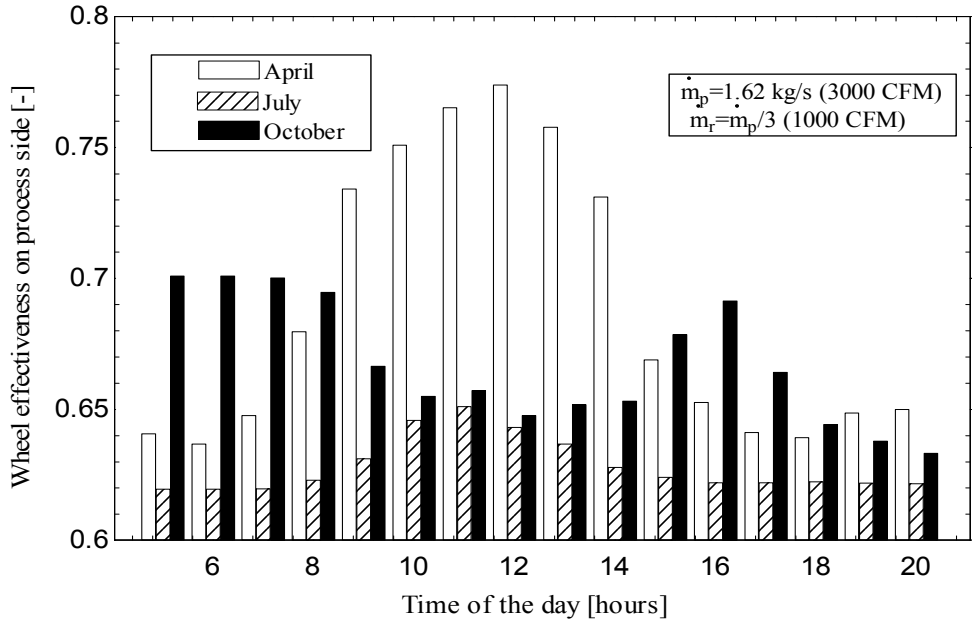


Figure 6.4: Variations of wheel effectiveness on process side during the day.

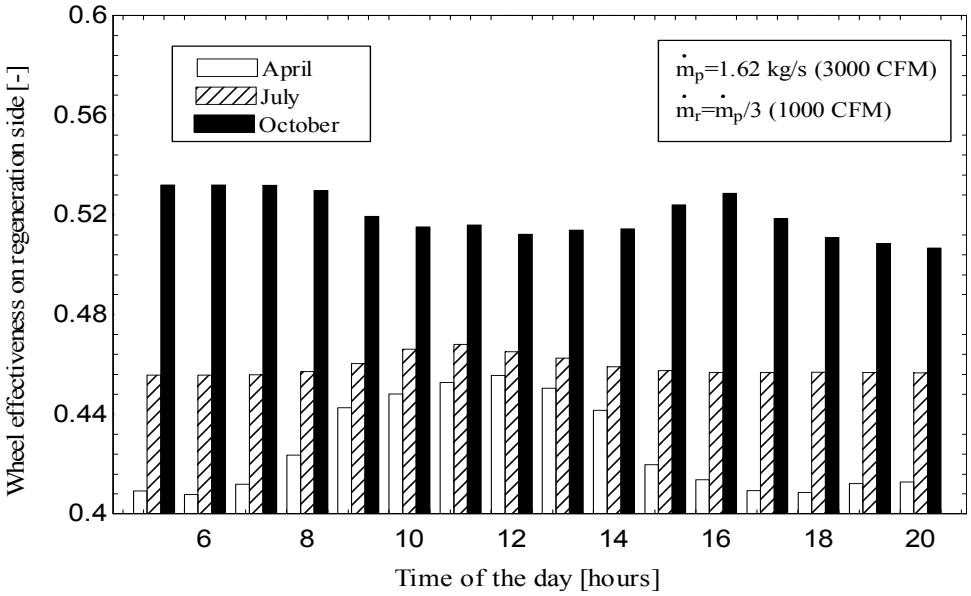


Figure 6.5: Variations of wheel effectiveness on regeneration side during the day.

The behavior of cooling effect produced during the day is illustrated in Fig. 6.6. It can be observed that during the day, the cooling effect increased and then decreased again after reaching the peak at 12:00 hours in July and 15:00 hours in April and October. This is because of high humidity ratio of ambient air which enhances the capability of the system to control the latent load. The maximum cooling effect was obtained in July at a process air flow rate of 1.62 kg/s because of summer conditions. The behavior of required regeneration load for all three months can be observed from Fig. 6.7.

The system COP largely depends on time of the day, mass flow rate ratios, and the required regeneration temperature as illustrated in Figs. 6.8 and 6.9. The maximum value of system COP of 0.41, 0.82 and 0.7 were found for the month of April, July and October, respectively as shown in Fig. 6.8 at a process air flow rate of 1.62 kg/s (3000CFM) and the mass flow rate ratio of 0.33. Figure 6.9 shows the effect of the mass flow rates ratio on COP of the system at different regeneration temperatures. It can be observed that the

system COP decreases with the increase in mass flow rates ratio and required regeneration temperature. This decrease in COP is due to increase in regeneration load with the increase in mass flow rates ratio and required regeneration temperature. Figure 6.10 shows the strong dependence of required regeneration heat on the value of mass flow rate ratio during the day. As, the value of regeneration air flow rate decreases or the ratio decreases, the required regeneration load decreases which in turns increases the system overall performance.

The maximum value of required regeneration load was 118, 59, 38 kW at ratio of mass flow rates of 1, 0.5, 0.33, respectively as it can be observed from Fig. 6.10. The value of this ratio of mass flow rate is controlled by bypassing the air at point 7 as shown in Fig. 6.2. The obtained results for performance of the system at different regeneration temperatures followed the same trend as experimental investigation for the desiccant cooling system carried out by Panaras et al. [244].

Figure 6.11 shows the variations of cooling effect produced for different values of process air mass flow rate. The cooling effect produced strongly depends on the value of process air flow rate. The cooling effect increased from 22 to 78 kW as the mass flow rate increased from 1 to 3 kg/s. It is observed that maximum value of cooling effect produced was 39 kW at a process air flow rate of 1.62 kg/s (3000 CFM) while this value increased to 52 kW as process air flow rate increased to 2.15 kg/s (4000 CFM). Similarly, the moisture removal capacity of the desiccant wheel increased with the increase in process air flow rate as illustrated in Fig. 6.12.

.

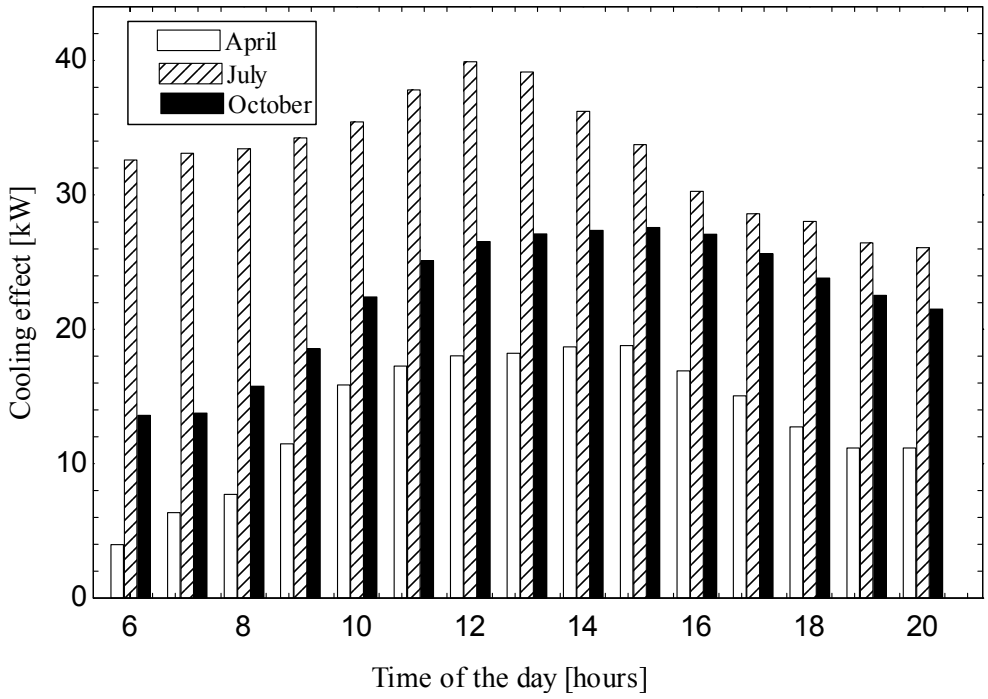


Figure 6.6: Variations of cooling effect produced during the day.

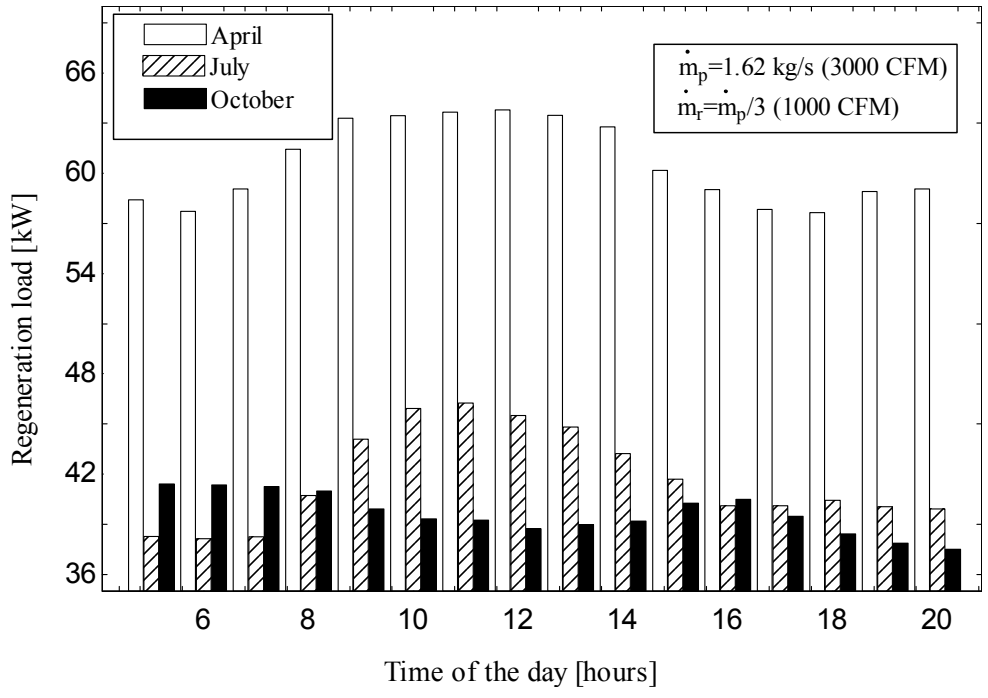


Figure 6.7: Variations of required regeneration load during the day.

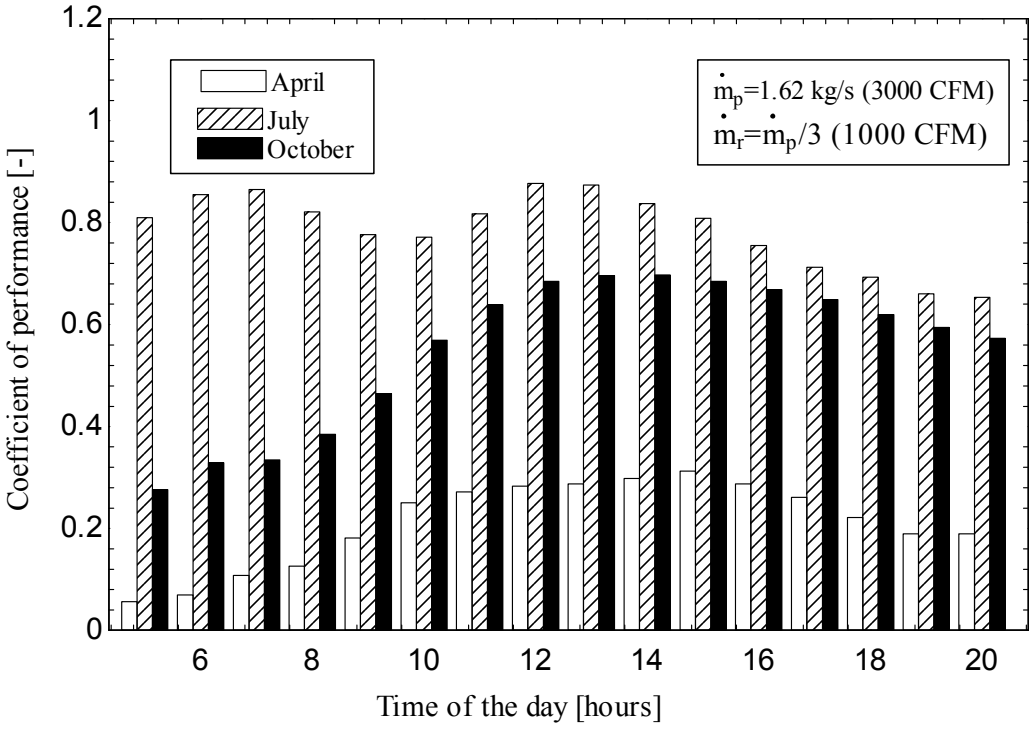


Figure 6.8: Variations of COP during the day.

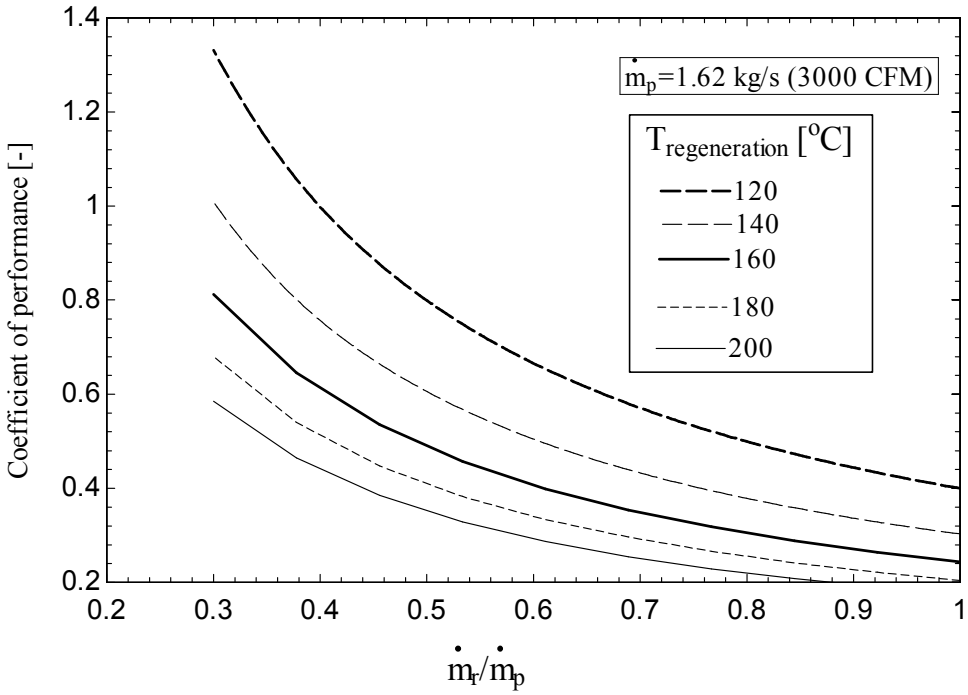


Figure 6.9: Variations of COP for different ratios of mass flow rates.

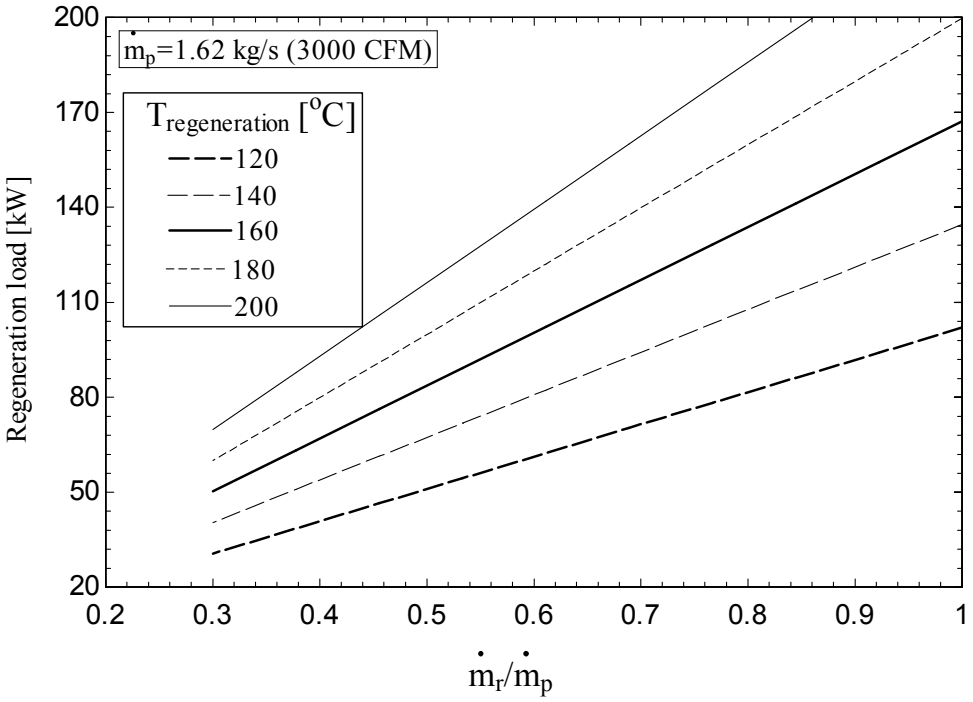


Figure 6.10: Variations of required regeneration load for different ratios of mass flow rates and regeneration temperature.

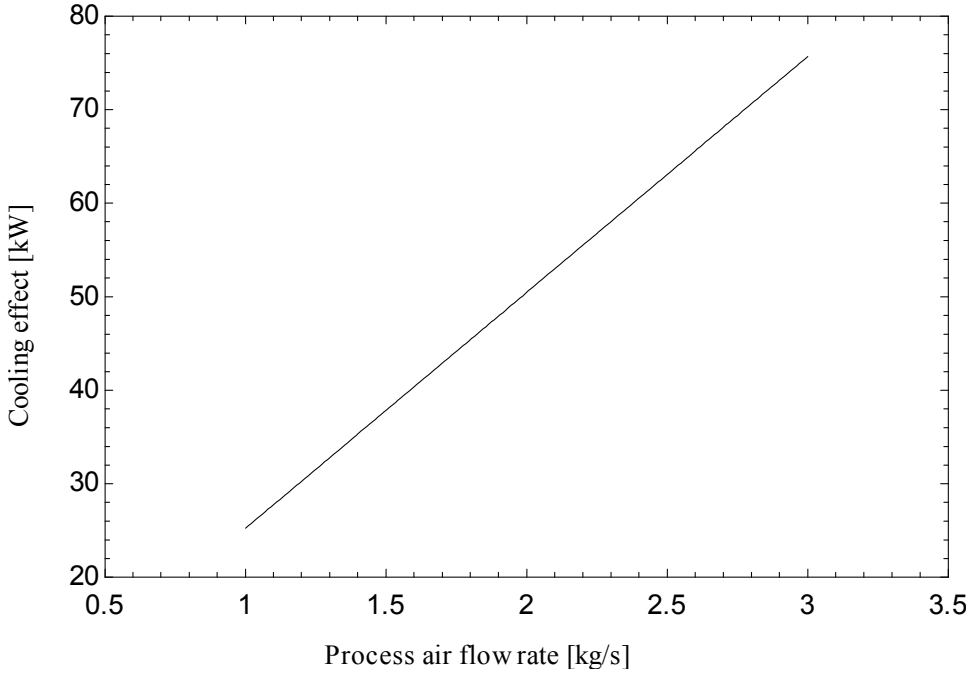


Figure 6.11: Variations of cooling effect for different process air flow rates.

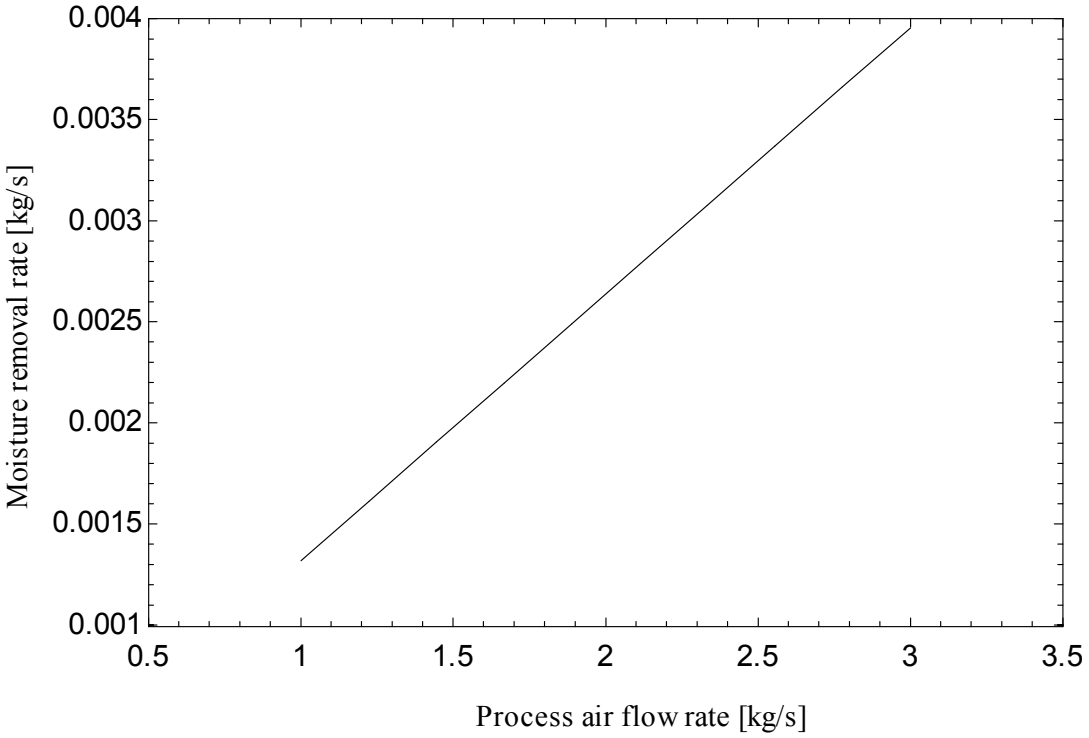


Figure 6.12: Variations of moisture removal capacity for various process air mass flow rates.

6.7 Conclusions

In this chapter a mathematical model for solid desiccant cooling system using a rotary wheel composed of silica gel have been developed to carry out the feasibility analysis of the system operating in conjunction with evaporative cooler under climatic conditions of Dhahran, Saudi Arabia. The system performance was studied for different operating parameters like process and regeneration air flow rate, humidity of process air, and regeneration temperature. The following results can be concluded:

- The effectiveness of the desiccant wheel on process and regeneration side depends on flow rate of air. It approaches 0.65 for the absorption process and 0.48 for the

regeneration process at a flow rate of process air of 1.62 kg/s and flow rate of regeneration air of 0.54 kg/s for the month of July.

- The maximum amount of moisture removed from the process air was 0.017 kg/kg at flow rate of process air 1.62 kg/s in July.
- The maximum value of cooling effect produced is 39 kW at a process air flow rate of 1.62 kg/s (3000CFM) while this value increased to 52 kW as process air flow rate increased to 2.15 kg/s (4000CFM).
- The maximum value of required regeneration load was 118, 59, 38 kW at ratio of mass flow rates of 1, 0.5, 0.33, respectively.

CHAPTER 7

CONCLUSIONS AND FUTURE WORK

7.1 Conclusions

In this thesis, a liquid desiccant cooling system has been theoretically and experimentally investigated. The system is installed at King Fahd University of Petroleum and Minerals (KFUPM). Desiccant cooling systems are a great HVAC alternative to high energy consuming conventional air conditioning units. One of the reasons is that desiccant systems can be powered by low grade thermal energy sources. The goal of this thesis is to evaluate the overall system performance with the main focus on dehumidification capability.

A theoretical model for the liquid desiccant dehumidifier was developed to observe the heat and mass transfer phenomenon for absorption and regeneration process under different operating parameters such as process air flow rate, temperature and humidity ratio of inlet air, and thickness of the dehumidifier. Effect of these parameters have also been studied on dehumidifier effectiveness, moisture removal rate and DCOP.

The necessary temperature of the regeneration air with respect to achieving desired moisture removal was evaluated in the range of 55 - 90˚C. It was shown that the desiccant wheel performance was very dependent on the regeneration temperature, air flow rates, and humidity ratio of the entering process air.

Results from the experiments regarding the overall performance indicated that the desiccant cooling system functioned well under different humidity conditions if the operating conditions were selected properly. However, the system excelled the most optimal performance during high absolute humidity conditions.

The desiccant wheel had a very good performance when acting as the dehumidification unit. When run alone, the desiccant wheel was capable of providing sufficient moisture removal in the range of 0.010 - 0.013 kg_v/kg_a, and had a high dehumidification performance.

Parametric studies were employed to examine the enhancement in heat and mass transfer between the air and the desiccant for different controlling parameters. The mathematical model of desiccant wheel has been verified with the results obtained from experimental rig. The experimental results show a good match with values determined from the mathematical model.

Furthermore, a mathematical model for solid desiccant cooling system using a rotary wheel composed of silica gel was developed to carry out the feasibility analysis of the system operating in conjunction with evaporative cooler under climatic conditions of Dhahran, Saudi Arabia. The system performance was studied for different operating parameters like process and regeneration air flow rates, humidity of process air, and regeneration temperature. The results show that, the cooling effect produced by the system increases from 39 kW to 52kW when flow rate of process air increases from 1.62 kg/s (3000 CFM) to 2.15 kg/s (4000 CFM).

7.2 Recommendations for future work

Although substantial work has been carried out during this research, there are still quite a few problems needing to be solved, and the performance of the system can still be improved to some degree.

Based on the results presented in this thesis a simulation model should be developed where simulated theoretical values and performance indexes can be compared to real experimental measurements and calculations. This will not only help in the performance analysis, but also make it easier to identify optimal operation conditions for further testing.

Economic evaluations should be carried out. This will help in evaluating the tradeoff between the initial cost of purchasing and installing units like pre cooling heat exchangers and the benefits achieved regarding the system performance. Also, a life cycle assessment should be performed to investigate the impact the desiccant system has on the environment.

Further, the possibility of applying processed ventilation air to the conditioned space should be implemented in the system. This can be done by making it possible to divide the process air into two separate air streams before entering the evaporative cooler. One air stream is then directed towards the cooler and the other is directed towards the conditioned space. An air to water heat exchanger should also be installed at a point before the processed air enters the conditioned space, so that the produced chilled water can be utilized to cool the supply air. When supplying ventilation air becomes possible, measurements of the air entering the building should be carried out.

A setup that allows for the desiccant system to be used as a heating cycle during the winter should be investigated. The evacuated tube solar air collectors have high efficiency and could be able to provide heated fresh air to the conditioned space during cold periods. The possibility of using the desiccant wheels to humidify the process air if the ambient air is too dry should also be investigated.

REFERENCES

[1] NSI/ASHRAE55, Thermal Environmental Conditions for Human Occupancy, American Society of Heating: Refrigerating and Air-conditioning Engineers, Atlanta, GA, 1992.

[2] Ameen A. The Challenges of air-conditioning in Tropical and Humid Tropical climates, Proceedings of the International Conference on Mechanical Engineering (ICME2005), Dhaka, Bangladesh, 2005.

[3] Davanagere S, Sherif A, Goswami Y. A feasibility study of a solar desiccant air-conditioning system—Part II: Transient simulation and economics, Int. J. Energy Res. 1999; 23(2):103–116.

[4] Henderson HI, Rengarajan K. Model to predict the latent capacity of air-conditioners and heat pumps at part-load conditions with constant fan operation, ASHRAE Transactions 1996; 102(1): 266-274.

[5] Jain S, Dhar PL, Kaushik SC. Evaluation of liquid dessicant based evaporative cooling cycles for typical hot and humid climates, Heat Recovery Syst. 1994; 14(6):621–632.

[6] Qiu GQ, Riffat SB. Novel design and modelling of an evaporative cooling system for buildings, Int. J. Energy Res. 2006; 30(12):985 – 999.

[7] Enteria N, Yoshino H, Mochida A, Takaki R, Satake A, Yoshie R, Mitamura T, Baba S. Synergization of Clean Energy Utilization, Clean Technology Development and Controlled Clean Environment through Thermally Activated Desiccant Cooling System. 2nd International Conference on Energy Sustainability 2008.

[8] Kahn JR, Franceschi D. Beyond Kyoto: A tax-based system for the global reduction of greenhouse gas emissions. Ecol. Econ. 2006; 58(4):778–787.

[9] York R. Demographic trends and energy consumption in European Union Nations, 1960–2025. Social Science Research 2007; 36(3): 855–872.

[10] Luken R, Grof T. The Montreal Protocol's multilateral fund and sustainable development. Ecol. Econ. 2006; 56(2):241–255.

[11] Barrios S, Bertinelli L, Strobl E. Climatic change and rural–urban migration: The case of sub-Saharan Africa. J. Urban Econ. 2006; 60(3):357-371.

[12] Forster P, Ramaswamy V, Artaxo P, Berntsen T, Betts R, Fahey DW. "Changes in atmospheric constituents and in radiative forcing. In: Solomon S, Qin D, Manning M, Chen Z, Marquis M, Averyt KB, Tignor M, Miller HL, editors. Climate change 2007: the

physical science basis. Contribution of Working Group I to the Fourth Assessment Report of the Intergovernmental Panel on Climate Change. Cambridge, United Kingdom and New York, NY, USA: Cambridge University Press," 2007.

[13] Strum M, Cook R, Thurman J, Ensley D, Pope A, Palma T, Mason R, Michaels H, Shedd S. Projection of hazardous air pollutant emissions to future years. Sci. Total Environ. 2006; 366(2):590–601.

[14] Barkenbus JN. Putting Energy Efficiency in a Sustainability Context: The Cold Facts about Refrigerators. Environ. Sci. Policy Sustain. Dev. 2006; 48(8):10–20.

[15] Radhwan AM, Gari HN, Elsayed MM. Parametric study of a packed bed dehumidifier/regenerator using $CaCl_2$ liquid desiccant. Renewable Energy 1993; 3(1):49–60.

[16] IEA. "Key world energy statistics. Paris, France: International Energy Agency" 2008.

[17] Coiante D, Barra L. Renewable energy capability to save carbon emissions. Sol. Energy 1996; 57(6): 485–491.

[18] Yu BF, Hu ZB, Liu M, Yang HL, Kong QX, Liu YH. Review of research on air-conditioning systems and indoor air quality control for human health. Int. J. Refrig. 2009; 32(1):2009.

[19] Afonso CFA. Recent advances in building air conditioning systems. Appl. Therm. Eng. 2006; 26(16): 1961–1971.

[20] Farber EA. Solar energy, its conversion and utilization. Sol. Energy 1973; 14(3):243–252.

[21] Henning HM. Solar assisted air conditioning of buildings – an overview. Appl. Therm. Eng. 2007; 27(10):1734–1749.

[22] Enteria N, Mizutani K. Solar Energy Sciences and Engineering Application. CRC Press, (2013).

[23] Renewable energy trends, Energy information administration, USA, 2004.

[24] Kahn JR, and Franceschi D. Beyond Kyoto: A tax-based system for the global reduction of greenhouse gas emissions, Ecol. Econ. 2006; 58(4):778–787.

[25] York R. Demographic trends and energy consumption in European Union Nations, 1960–2025, Soc. Sci. Res. 2007; 36(3):855–872.

[26] Barrios S, Bertinelli L, and Strobl E. Climatic change and rural–urban migration: The case of sub-Saharan Africa, J. Urban Econ. 2006; 60(3):357–371.

[27] The Freedonia Group, Inc. "World HVAC equipment demand, Cleveland, Ohio USA", 2012.

[28] Watt JR. Evaporative Air Conditioning Handbook. 3rd edition. Prentice Hall, 1997.

[29] Cerci Y. A new ideal evaporative freezing cycle. Int. J. Heat Mass Transf. 2003; 46(16):2967–2974.

[30] ASHRAE Handbook, Evaporative cooling applications, HVAC Applications, Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers Inc. 2003; 55.1 – 55.3.

[31] Heidarinejad G, Bozorgmehr M, Delfani S, Esmaeelian J. Experimental investigation of two-stage indirect/direct evaporative cooling system in various climatic conditions. Build. Environ. 2009; 44(10):2073–2079.

[32] Joudi KA, Mehdi SM. Application of indirect evaporative cooling to variable domestic cooling load. Energy Convers. Manag. 2000; 41(17):1931–1951.

[33] Saman WY, Alizadeh S. An experimental study of a cross-flow type plate heat exchanger for dehumidification/cooling. Sol. Energy 2002; 73(1):59–71.

[34] Kessling W, Laevemann E, Peltzer M. Energy storage in open cycle liquid desiccant cooling systems. Int. J. Refrig. 1998; 21(2):150–156.

[35] Costelloe B, Finn D. Indirect Evaporative Cooling Potential in Air-water Systems in Temperate Climates. Energy and Buildings 2003; 35:573-591.

[36] Heidarinejad G, Bozorgmehr M, Delfani S, Esmaeelian J. Experimental investigation of two-stage indirect/direct evaporative cooling system in various climatic conditions. Build. Environ. 2009; 44(10):2073–2079.

[37] Shariaty-Niassar M, Gilani N. An Investigation of Indirect Evaporative Coolers, IEC With Respect to Thermal Comfort Criteria. Iran. J. Chem. Eng. 2009; 6(2):14-28.

[38] Kulkarni RK, Rajput SPS. Performance evaluation of two stage indirect/direct evaporative cooler with alternative shapes and cooling media in direct stage. Int. J. Appl. Eng. 2001; 1(4):800.

[39] Bruno F. An indirect evaporative cooler for supplying air near the dew point. 48th AuSES Annual Conference, Canberra, ACT, Australia 2010.

[40] Pimenta JMD, Castro WP. Analysis of different applications of evaporative cooling systems. Proc. 17th Int. Congr. Mech. Engineering, Sao Paulo 2003.

[41] Jones WP. Air Conditioning Engineering. 5th Ed. Routledge 2001.

[42] Camargo JR, Ebinuma CD, Silveira JL. Experimental performance of a direct evaporative cooler operating during summer in a Brazilian city. Int. J. Refrig. 2005; 28(7):1124–1132.

[43] Riffat SB, Zhu J. Mathematical model of indirect evaporative cooler using porous ceramic and heat pipe. Appl. Therm. Eng. 2004; 24(4):457–470.

[44] Zhao X, Li JM, Riffat SB. Numerical study of a novel counter-flow heat and mass exchanger for dew point evaporative cooling. Appl. Therm. Eng. 2008; 28(14-15): 1942–1951.

[45] Zhao X, Liu S, Riffat SB. Comparative study of heat and mass exchanging materials for indirect evaporative cooling systems. Build. Environ. 2008; 43(11):1902–1911.

[46] Davis RA. Evaluation of Advanced Evaporative Cooler Technologies; Pacific Gas and Electric (PG&E) Customer Energy Management Emerging Technologies Program; Report No.: 491-04.7, 2004.

[47] Enteria N, Mizutani K. The role of the thermally activated desiccant cooling technologies in the issue of energy and environment. Renew. Sustain. Energy Rev. 2011; 15(4):2095–2122.

[48] Halliday SP, Beggs CB, Sleigh PA. The use of solar desiccant cooling in the UK: a feasibility study. Appl. Therm. Eng. 2002; 22(12):1327–1338.

[49] Enteria N, Yoshino H, Mochida A, Takaki R, Satake A, Yoshie R, Mitamura T, Baba S. Construction and initial operation of the combined solar thermal and electric desiccant cooling system. Sol. Energy 2009; 83(8):1300–1311.

[50] Enteria N, Yoshino H, Satake A, Mochida A, Takaki R, Yoshie R, Baba S. Development and construction of the novel solar thermal desiccant cooling system incorporating hot water production. Appl. Energy 2010; 87(2):478–486.

[51] Henning HM. Solar-Assisted Air-Conditioning in Buildings: A Handbook for Planners, 2nd edition. Wien ; London: Springer Vienna Architecture 2007.

[52] Nayak SM. Experimental and Theoretical Investigation of Integrated Engine Generator - Liquid Desiccant System. Ph.D., Thesis, College Park, University of Maryland, United States 2005.

[53] Ahmed MH, Kattab NM, Fouad M. Evaluation and optimization of solar desiccant wheel performance. Renewable Energy 2005; 30(3):305-325.

[54] Gandhidasan P. A simplified model for air dehumidification with liquid desiccant. Sol. Energy 2004; 76(4):409–416.

[55] Archibald J. A new desiccant evaporative cooling cycle for solar air conditioning and hot water heating. In forum proceeding, American solar energy society and American institute of architects 2001:237-242.

[56] Stein B. Building Technology: Mechanical and Electrical Systems. 2nd Ed. New York: Wiley, 1997.

[57] Zadpoor AA. Golshan AH. Performance improvement of a gas turbine cycle by using a desiccant-based evaporative cooling system. Energy 2006; 31(14):2652–2664.

[58] Mohammad AT, Mat SB, Sulaiman MY, Sopian K, Al-abidi AA. Historical review of liquid desiccant evaporation cooling technology. Energy Build. 2013; 67:22–33.

[59] Enteria N, Mizutani K. The role of the thermally activated desiccant cooling technologies in the issue of energy and environment. Renew. Sustain. Energy Rev. 2011; 15(4):2095–2122.

[60] Dunkle RV. A method of solar air conditioning Mech Chem En 9 Trans 1965; Inst. Eng. (Australia), vol-1, MC-1, 73-78, 1965.

[61] Pennington. Humidity changer for air conditioning US Patent 2, 700, 537, 1955.

[62] Maclaine-Cross IL, Banks PJ. A general theory of wet surface heat exchangers and its application to regenerative evaporative cooling. Journal of heat transfer 1981; 103(3):579-585.

[63] Dezfouli MMS, Mat S, Sopian K. Comparison Simulation between Ventilation and Recirculation of Solar Desiccant Cooling System by TRNSYS in Hot and Humid Area. Proceedings of the 7th International Conference on Renewable Energy Sources, Kuala Lumpur, Malaysia, ISBN: 978-1-61804-175-3, 2013.

[64] Kang TS, Maclaine-Cross IL. High performance, solid desiccant, open cooling cycles. J. Sol. Energy Eng. 1989; 111(2).

[65] Collier RK, Barlow RS, Arnold FH. An overview of open-cycle desiccant-cooling systems and materials. Journal of Solar Energy Engineering 1982; 104(1):28-34.

[66] Kanoğlu M, Çarpınlıoğlu M, Yıldırım M. Energy and exergy analyses of an experimental open-cycle desiccant cooling system. Appl. Therm. Eng. 2004; 24(5):919–932.

[67] Haddad K, Ouazia B, Barhoun H. Simulation of a desiccant-evaporative cooling system for residential buildings. 3rd Canadian Solar Buildings Conference, Fredericton, N.B., pp. 2008; 1-8.

[68] Uçkan I, Yılmaz T, Büyükalaca O, Hürdoğan E. First Experimental Results of a Desiccant Based Evaporative Cooling System in Adana. ROOMVENT, Norway, 2011.

[69] Riangvilaikul B, Kumar S. An experimental study of a novel dew point evaporative cooling system. Energy Build. 2010; 42(5):637–644.

[70] Katejanekarn T, Kumar S. Performance of a solar-regenerated liquid desiccant ventilation pre-conditioning system. Energy Build. 2008; 40(7):1252–1267.

[71] Parmar H, Hindoliya DA. Performance of a solid desiccant based evaporative cooling system in warm and humid climatic zone of India. International Journal of Engineering Science and Technology 2010; 2(10):5504-5508.

[72] Oliveira AC, Afonso CF, Riffat SB, Doherty PS. Thermal performance of a novel air conditioning system using a liquid desiccant. Appl. Therm. Eng. 2010; 20(13):1213–1223.

[73] Kessling W, Laevemann E, Kapfhammer C. Energy storage for desiccant cooling systems component development. Sol. Energy 1998; 64(4-6):209–221.

[74] Wurtz E, Maalouf C, Mora L, Allard F. Parametric analysis of a solar desiccant cooling system using the SimSPARK Environment. 9th International IBPSA Conference Montréal, Canadas 2005; 15-18.

[75] Ouazia B, Barhoun H, Barhoun H, Haddad K, Armstrong MM, Marchand RG, Szadkowski F. Desiccant-evaporative cooling system for residential buildings. 12th Canadian Conference on Building Science and Technology Montréal, Québec 2009.

[76] Saman WY, Alizadeh S. Modelling and performance analysis of a cross-flow type plate heat exchanger for dehumidification/cooling. Sol. Energy 2001; 70(4):361–372.

[77] Saman WY, Alizadeh S. Modeling and performance of a forced flow solar collector/regenerator using liquid desiccant. Sol. Energy 2002; 72(2): 143-154.

[78] Nelson JS, Beckman WA, Mitchell JW, Close DJ. Simulations of the performance of open cycle desiccant systems using solar energy. Sol. Energy 1978; 21(4):273–278.

[79] Smith RR, Hwang CC, Dougall RS. Modeling of a solar-assisted desiccant air conditioner for a residential building. Energy 1994; 19(6):679-691.

[80] Al-Sulaiman FA, Gandhidasan P, Zubair SM. Liquid desiccant based two-stage evaporative cooling system using reverse osmosis (RO) process for regeneration. Appl. Therm. Eng. 2007; 27(14):2449–2454.

[81] Ge TS, Li Y, Wang RZ, Dai YJ. A review of the mathematical models for predicting rotary desiccant wheel. Renew. Sustain. Energy Rev. 2008; 12(6):1485–1528.

[82] Erens PJ, Dreyer AA. Modelling of indirect evaporative air coolers. Int. J. Heat Mass Transf. 1993; 36(1):17–26.

[83] Radhwan AM, Elsayed MM, Gari HN. Mathematical modeling of solar operated liquid desiccant-evaporative air conditioning system. Engineering Sciences 1999; 11(1):119-141.

[84] Bellemo L, Elmegaard B, Reinholdt LO, Kærn MR. Modeling of a regenerative indirect evaporative cooler for a desiccant cooling system. 4th IIR Conference on Thermophysical Properties and Transfer Processes of Refrigerants, Delft, The Netherlands, 2013.

[85] Riangvilaikul B, Kumar S. Numerical study of a novel dew point evaporative cooling system. Energy Build. 2010; 42(11):2241–2250.

[86] Goldsworthy M, White S. Optimisation of a desiccant cooling system design with indirect evaporative cooler. Int. J. Refrig. 2011; 34(1):148–158.

[87] Pescod D. A heat exchanger for energy saving in an air-conditioning plant. ASHRAE Transactions 1979; 85(2):238-251.

[88] Camargo JR, Ebinuma CD, Cardoso S. A mathematical model for direct and indirect evaporative cooling air conditioning systems. Proceedings of the 9th Brazilian congress of thermal engineering and sciences, ENCIT, UFPB, PB, paper CIT02-0855, 2002.

[89] Chen PL, Qin HM, Huang YJ, Wu HF. A heat and mass transfer model for thermal and hydraulic calculations of indirect evaporative cooler performance. ASHRAE Transactions 1991; 97(2):852-863.

[90] Subramanyam N, Maiya MP, Murthy SS. Application of desiccant wheel to control humidity in air-conditioning systems. Appl. Therm. Eng. 2004; 24(17-18):2777–2788.

[91] Khoukhi M. The Use of Desiccant Cooling System with IEC and DEC in Hot-Humid Climates. Int. J. Energy Eng. 2013); 3(4):107-111.

[92] Suryawanshi SD, Tarup M. Chordia, Nikita Nenwani, Harish Bawaskar, Suraj Yambal, "Efficient Technique of Air-Conditioning.Proceedings of the World Congress on Engineering. Vol III WCE 2011, London, U.K 2011.

[93] Mohammad A, Mat S, Sulaiman MY, Sopian K. Experimental Performance of a Direct Evaporative Cooler Operating in Kuala Lumpur. Int J Therm. Environ. Eng. 2013; 6(1):15-20.

[94] Jain S, Dhar PL, Kaushik SC. Evaluation of solid-desiccant-based evaporative cooling cycles for typical hot and humid climates. Int. J. Refrig. 1995; 18(5):287–296.

[95] Kim MH, Jeong JW. Development of Desiccant and Evaporative Cooling Based 100% Outdoor System. AEI, American Society of Civil Engineers 2013:506–515.

[96] Glav BO. Air Conditioning Apparatus. US Patent 3251402, 1966.

[97] Worek WM, Zheng W, Belding WA, Novosel D. Simulation of advanced gas-fired desiccant systems. ASHRAE Trans 1991; 97(2):609–14.

[98] Ge TS, Li Y, Wang RZ, Dai YJ. Experimental study on a two-stage rotary desiccant cooling system. Int. J. Refrig. 2009; 32(3):498–508.

[99] Khalid A. Experimental Investigation and Mathematical Modeling of a Low Energy Consuming Hybrid Desiccant Cooling System for Hot and Humid Areas of Pakistan. Ph.D., Thesis, NED University of Engineering & Technology, Karachi, 2008.

[100] Camargo JR, Godoy Jr. E, Ebinuma CD. An evaporative and desiccant cooling system for air conditioning in humid climates. J. Braz. Soc. Mech. Sci. & Eng. 2005; 27:243-247.

[101] Bruno F. On-site experimental testing of a novel dew point evaporative cooler. Energy Build. 2011; 43(12):3475–3483.

[102] El-Dessouky H. Enhancement of the thermal performance of a wet cooling tower. Can. J. Chem. Eng. 1996; 74(3):331–338.

[103] Stoitchkov NJ, Dimitrov GI. Effectiveness of crossflow plate heat exchanger for indirect evaporative cooling. Int. J. Refrig. 1998; 21(6):463–471.

[104] Camargo JR, Ebinuma CD, Silveira JL. Thermoeconomic analysis of an evaporative desiccant air conditioning system. Appl. Therm. Eng. 2003; 23(12):1537–1549.

[105] Eskra N. Indirect-direct Evaporative Cooling Systems. ASHRAE J. 1980; 22(5):21-25.

[106] Kulkarni RK, Rajput SPS. Performance evaluation of two stage indirect/direct evaporative cooler with alternative shapes and cooling media in direct stage. International Journal of Applied Engineering Research 2011; 1.4:800-812.

[107] Watt JR. Evaporative Air Conditioning Handbook, 3 Sub edition. Lilburn, GA : Upper Saddle River, NJ: Prentice Hall, 1997.

[108] Maclaine-Cross IL, Banks PJ. A General Theory of Wet Surface Heat Exchangers and its Application to Regenerative Evaporative Cooling. J. Heat Transf.-Trans. Asme - J HEAT Transf. 1981; 103(3):579-585.

[109] Yellott JI, Gamero J. Indirect evaporative air coolers for hot, dry climates. ASHRAE Trans. 1984; 90(1B):139-147.

[110] Eskra E. Indirect/direct evaporative cooling systems. ASHRAE J. 1980; 22(5):21-25.

[111] Crum DR. Open Cycle Air Conditioning Systems. MS, Thesis, Wisconsin – Madison, USA, 1986.

[112] Bisoniya TS, Rajput SPS, Kumar A. Comparative thermal analysis of theoretical and experimental studies of modified indirect evaporative cooler having cross flow heat exchanger with one fluid mixed and the other unmixed. Int. J. Energy Environ. 2011; 2(5):921-932.

[113] Al-Sulaiman F. Evaluation of the performance of local fibers in evaporative cooling. Energy Convers. Manag. 2002; 43(16):2267–2273.

[114] Alonso JFS, Martínez FJR, Gómez EV, Plasencia MA. Simulation model of an indirect evaporative cooler. Energy Build. 1998; 29(1):23–27.

[115] Al-Nimr MA, Nabah BA, Naji M. A novel summer air conditioning system. Energy Convers. Manag. 2002; 43(14):1911–1921.

[116] Ibrahim E, Shao L, Riffat SB. Performance of porous ceramic evaporators for building cooling application. Energy Build. 2003; 35(9):941–949.

[117] Jain JK, Hindoliya DA. Development and Testing of Regenerative Evaporative Cooler. International Journal of Engineering Trends and Technology 2012; 3(6):694-697.

[118] Jose A, San JF, Alvarez-Guerra, MA, and Gomez. Simulation model of an indirect evaporative cooler. Energy and Buildings 1998; 29(1):23–7.

[119] Guo XC, Zhao TS. A parametric study of an indirect evaporative air cooler. International Communications in Heat and Mass Transfer 1998; 25(2):217–226.

[120] Zhan, C., Duan, Z., Zhao, X., Smith, S., Jin, H., and Riffat, S. Comparative study of the performance of the M-cycle counter-flow and cross-flow heat exchangers for indirect evaporative cooling—Paving the path toward sustainable cooling of buildings. Energy 2011; 36(12):6790–805.

[121] SWEEP/WCEC Workshop on modern evaporative cooling technologies, Boulder, Colorado, September 14, 2007.

[122] Pérez-Lombard L, Ortiz J, Pout C. A review on buildings energy consumption information. Energy Build. 2008; 40(3):394–398.

[123] ASHRAE Handbook–HVAC systems and equipment, Inc., Atlanta, Georgia 2008.

[124] Ge G, Xiao F, Niu X. Control strategies for a liquid desiccant air-conditioning system. Energy Build. 2011; 43(6):1499–1507.

[125] Nayak SM. Performance characterization of gas engine generator integrated with a liquid desiccant dehumidification system. Applied Thermal Engineering 2009; 29(2): 479-490.

[126] Nernst W. Theory of reaction velocity in heterogeneous systems. Z Phys Chem. 1904; 47:52–55.

[127] Whitman WG. The two film theory of gas absorption. Int. J. Heat Mass Transf. 1962; 5(5):429–433.

[128] Qi R, Lu L, Yang H. Investigation on air-conditioning load profile and energy consumption of desiccant cooling system for commercial buildings in Hong Kong. Energy Build. 2012; 49:509–518.

[129] Harriman LG, Plager D, Kosar D. Dehumidification and cooling loads from ventilation air. ASHRAE Journal 1997; 96(6): 31-45.

[130] Burns PR, Mitchell JW, Beckman WA. Hybrid desiccant cooling system in supermarket applications. ASHRAE Transactions 1985; 91(1):457-468.

[131] Abdel-Salam AH, Ge G, Simonson GJ. Performance analysis of a membrane liquid desiccant air-conditioning system. Energy Build. 2013; 62:559–569.

[132] Bichowsky, Francis, US Patent 1935; 1 992 177.

[133] Ahmed MH, Kattab NM, Fouad M. Evaluation and optimization of solar desiccant wheel performance. Renewable Energy 2005; 30(3):305-325.

[134] Mei L, Dai YJ. A technical review on use of liquid-desiccant dehumidification for air-conditioning application. Renew. Sustain. Energy Rev. 2008; 12(3):662–689.

[135] Jiang Y, Li Z, Chen XL, Liu XH. Liquid desiccant air conditioning system and its applications. Heat Ventilation Air Conditioning 2004; 34:88–98.

[136] Gandhidasan P, Mohandes MA. Predictions of vapor pressures of aqueous desiccants for cooling applications by using artificial neural networks. Applied Thermal Engineering 2008; 28(2): 126-135.

[137] Elsarrag E. Dehumidification of air by chemical liquid desiccant in a packed column and its heat and mass transfer effectiveness. HVAC&R Research 2006; 12(1):3–16.

[138] Lowenstein A, Slyzak, Kozubul E. A zero carry over liquid desiccant air conditioner for solar applications. ASME International Solar Energy Conference (ISEC 2006) Denver, Colarodo," 2006.

[139] Zuber A, Checoni RF, Mathew R, Santos JPL, Tavares FW, Castier M. Thermodynamic Properties of 1:1 Salt Aqueous Solutions with the Electro lattice Equation of State. Oil Gas Sci. Technol. Rev. IFP Energies nouvelles 2013; 68(2):255–270.

[140] Ahmed SY, Gandhidasan P, Al-Farayedhi AA. Thermodynamic analysis of liquid desiccants. Sol. Energy 1998; 62(1):11–18.

[141] Morillon V, Debeaufort F, Jose J, Tharrault JF, Capelle M, Blond G, Voilley A. Water vapour pressure above saturated salt solutions at low temperatures. Fluid Phase Equilibria 1999; 155(2):297–309.

[142] Sun J, Gong XL, Shi MH. Study on vapor pressure of liquid desiccants solution. J of Refrigeration 2004; 5(1):87–95.

[143] McNelly L. Thermodynamic properties of aqueous solutions of lithium bromide. ASHRAE Trans 1979;85:412–34.

[144] Kaita Y. Thermodynamic properties of lithium bromide–water solutions at high temperatures. Int. J. Refrig. 2001; 24(5):374–390.

[145] de Lucas A, Donate M, Rodríguez JF. Vapour pressures, densities, and viscosities of the (water + lithium bromide + potassium acetate) system and (water + lithium bromide + sodium lactate) system. J. Chem. Thermodyn. 2006; 38(2):123–129.

[146] Park Y, Kim JS, Lee H. Physical properties of the lithium bromide + 1,3-propanediol + water system. Int. J. Refrig. 1997; 20(5):319–325.

[147] Ertas A, Anderson EE, Kiris I. Properties of a new liquid desiccant solution—Lithium chloride and calcium chloride mixture. Sol. Energy 1992; 49(3):205–212.

[148] Aristov YI, Tokarev MM, Cacciola G, Restuccia G. Selective water sorbents for multiple applications, 1. CaCl2 confined in mesopores of silica gel: Sorption properties. React. Kinet. Catal. Lett. 1996; 59(2):325–333.

[149] Jia CX, Dai YJ, Wu JY, Wang RZ Experimental comparison of two honeycombed desiccant wheels fabricated with silica gel and composite desiccant material. Energy Convers. Manag. 2006; 47(15-160:2523–2534.

[150] La D, Dai YJ, Li Y, Wang RZ, Ge TS. Technical development of rotary desiccant dehumidification and air conditioning: A review. Renew. Sustain. Energy Rev. 2010; 14(1):130–147.

[151] Liu Y, Wang R. Pore structure of new composite adsorbent SiO2•xH2O• yCaCl2 with high uptake of water from air. Sci. China Ser. E Technol. Sci. 2003; 46(5):551–559.

[152] González JC, Molina-Sabio M, Rodríguez-Reinoso F. Sepiolite-based adsorbents as humidity controller. Appl. Clay Sci. 2001; 20(3):111–118.

[153] Zegenhagen MT, Ricart C, Meyer T, Kühn R, and Ziegler F. Experimental investigation of a liquid desiccant system for air dehumidification working with ionic liquids. Energy Procedia 70 (2015): 544-551.

[154] Studak JW, Peterson JL. A preliminary evaluation of alternative liquid desiccants for a hybrid desiccant air conditioner. Desiccant cooling and dehumidification, ASHARE, Atlanta, GA 1992.

[155] Kessling W, Laevemann E, Peltzer M. Energy storage in open cycle liquid desiccant cooling systems. International journal of refrigeration 1998; 21(2): 150-156.

[156] Fumo N, Goswami DY. Study of an aqueous lithium chloride desiccant system: air dehumidification and desiccant regeneration. Sol. Energy 2002; 72(4):351–361.

[157] Khoukhi M, Yoshino H, Mochida A, Enteria N, Takaki R, Satake A, Yoshie R, Mitamura T. Study of the conventional twin Rotor desiccant cooling system. Part 1: Experimental set-up, Summaries of Technical Papers of Annual Meeting Architectural Institute of Japan (AIJ) 2006; 1109-1010.

[158] Enteria N, Yoshino H, Mochida A, Khoukhi M, Takaki R, Satake A, Yoshie R, Mitamura T. Study of the conventional twin rotor desiccant cooling system. Part 1: Modeling and parametric investigation. Summaries of Technical Papers of Annual Meeting Architectural Institute of Japan (AIJ) 2006; 1111-1012.

[159] Enteria N, Yoshino H, Takaki R, Satake A, Mochida A, Khoukhi M, Yoshi R, Mitamura T, Baba S. The experimental works and some parametric investigations of thermally activated desiccant cooling system. Proceedings of Clima WellBeing Indoors 2007.

[160] Liu XH, Zhang Y, Qu KY, Jiang Y. Experimental study on mass transfer performances of cross flow dehumidifier using liquid desiccant. Energy Convers. Manag. 2006; 47(15-16):2682–2692.

[161] Liu X, Jiang Y, Xia J, Chang X. Analytical solutions of coupled heat and mass transfer processes in liquid desiccant air dehumidifier/regenerator. Energy Convers. Manag. 2007; 48(7):2221–2232.

[162] Liu XH, Jiang Y. Coupled heat and mass transfer characteristic in packed bed dehumidifier/regenerator using liquid desiccant. Energy Convers. Manag. 2008; 49(6):1357–1366.

[163] Jain S, Bansal PK. Performance analysis of liquid desiccant dehumidification systems. International journal of refrigeration, (2007); 30(5):861-872.

[164] Katejanekarn T, Chirarattananon S, Kumar S. An experimental study of a solar-regenerated liquid desiccant ventilation pre-conditioning system. Sol. Energy 2009; 83(6):920–933.

[165] Lof GOG. Cooling with solar energy. Congress on solar energy, Tucson, Arizona, 1955; 171–189.

[166] Patnaik S, Lenz TG, Lof GOG. Performance studies for an experimental solar open-cycle liquid desiccant air dehumidification system. Solar Energy, 1990; 44(3):123-135.

[167] Nagaya TSK. High energy efficiency desiccant assisted automobile air-conditioner and its temperature and humidity control system. Appl. Therm. Eng. 2006; 26(14):1545–1551.

[168] Nóbrega, CE, Brum. Numerical simulation of desiccant assisted evaporative cooling systems. 20th International Congress of Mechanical Engineering Gramado, RS, Brasil. 2009.

[169] Babakhani D. Developing an application analytical solution of adiabatic heat and mass transfer processes in a liquid desiccant dehumidifier/regenerator. Chemical Engineering & Technology 2009; 32(12):1875-1884.

[170] Diaz G. Numerical investigation of transient heat and mass transfer in a parallel-flow liquid-desiccant absorber. Heat and mass transfer, 2010; 46(11-12):1335-1344.

[171] Seenivasan D, Selladurai V, Senthil P. Optimization of liquid desiccant dehumidifier performance using Taguchi method. Advances in Mechanical Engineering 2014; Article ID 506487:1-6.

[172] Gommed K, Ziegler F, Grossman G. Experimental investigation of a LiCl-water open absorption system for cooling and dehumidification. J. Sol. Energy Eng. 2004; 126(2):710–715.

[173] Gommed K, Grossman G. Experimental investigation of a liquid desiccant system for solar cooling and dehumidification. Sol. Energy 2007; 81(1):131–138.

[174] Chen XY, Jiang Y, Li Z, Qu KY. Field study on independent dehumidification air-conditioning system-I: Performance of liquid desiccant dehumidification system. ASHRAE Trans. 2005; 111(2):271–276.

[175] Kumar R. Stand-alone liquid desiccant air conditioning systems. With emphasis on falling film and spray type absorbers/regenerators. LAP LAMBERT Academic Publishing, 2011.

[176] Saman WY, Alizadeh S. An experimental study of a forced flow solar collector/regenerator using liquid desiccant. Sol. Energy 2002; 73(5): 345-362.

[177] Hamed AM, Khalil A, Kabeel AE, Bassuoni MM, Elzahaby AM. Performance analysis of dehumidification rotating wheel using liquid desiccant. Renew. Energy 2005; 30(11):1689–1712.

[178] So'Brien GC, Satcunanathan S. Performance of a Novel Liquid Desiccant Dehumidifier/Regenerator System. J. Sol. Energy Eng. 1989; 111(4):345–352.

[179] Koronaki IP, Christodoulaki RI, Papaefthimiou VD, Rogdakis ED. Thermodynamic analysis of a liquid desiccant cooling system under mediterranean climatic conditions. International Mechanical Engineering Congress and Exposition. American Society of Mechanical Engineers 2012; pp. 137–146.

[180] Pesaran A, Penney TR, Czanderna AW. Desiccant Cooling: State-of-the-Art Assessment. National Renewable Energy Laboratory, Cole Boulevard 1992.

[181] Scalabrin G, Scaltriti G. A. Liquid sorption-desorption system for air conditioning with heat at lower temperature. J. Sol. Energy Eng. 1990; 112(2): 70–75.

[182] Gommed GGK. Experimental Investigation of a LiCl-Water Open Absorption System for Cooling and Dehumidification. Journal of solar energy engineering 2004; 126(2):710-715.

[183] Yoon JI, Phan TT, Moon CG, Bansal P. Numerical study on heat and mass transfer characteristic of plate absorber. Appl. Therm. Eng. 2005; 25(14):2219–2235.

[184] Khan AY. Cooling and dehumidification performance analysis of internally-cooled liquid desiccant absorbers. Appl. Therm. Eng. 1998; 18(5):265–281.

[185] Saman WY, Alizadeh S. Modelling and performance analysis of a cross-flow type plate heat exchanger for dehumidification/cooling. Sol. Energy 2001; 70(4):361–372.

[186] Laevemann, Hauer A, Peltzer M, Hublitz A, Kroenauer A Raab U. Thermochemical storage for air-conditioning using open cycle liquid desiccant technology. ECOSTOCK 2006, Stockton College of New Jersey, Pomona, NJ, 2006.

[187] Mesquita LCS, Harrison SJ, Thomey D. Modeling of heat and mass transfer in parallel plate liquid-desiccant dehumidifiers. Sol. Energy 2006; 80(11):1475–1482.

[188] Queiroz AG, Orlando AF, Saboya FEM. Performance Analysis of an Air Drier for a Liquid Dehumidifier Solar Air Conditioning System. J. Sol. Energy Eng. 1988; 110(2):120–124.

[189] Yin Y, Zhang X, Wang G, Luo L. Experimental study on a new internally cooled/heated dehumidifier/regenerator of liquid desiccant systems. International Journal of Refrigeration, 2008; 31(5):857-866.

[190] Khan AY, Martinez JL. Modelling and parametric analysis of heat and mass transfer performance of a hybrid liquid desiccant absorber. Energy Conversion Management 1998; 39:1095–1112.

[191] Qi R, Lu L, Yang H. Quick performance prediction for internally cooled/heated liquid desiccant dehumidification system. Building Services Engineering Research and Technology 2012; 1–15.

[192] Khan AY, Sulsona FJ. Modeling and parametric analysis of heat and mass transfer performance of refrigerant and cooled liquid desiccant absorbers. International Journal of Energy Research 1998; 22(9):813–32.

[193] Liu XH, Geng KC, Lin BR, Jiang Y. Combined cogeneration and liquid desiccant system applied in a demonstration building. Energy Build. 2004; 36(9):945–53.

[194] Li Z. Liquid desiccant air conditioning and independent humidity control air conditioning systems. Heat Ventilation Air Condition. 2003; 33:26–31.

[195] Xiong ZQ, Dai YJ, Wang RZ. Development of a novel two-stage liquid desiccant dehumidification system assisted by CaCl2 solution using exergy analysis method. Appl. Energy 2010; 87(5):1495–1504.

[196] Kumar R, Dhar PL, Jain S, Asati AK. Multi absorber stand-alone liquid desiccant air-conditioning systems for higher performance. Solar Energy 2009; 83(5):761-772.

[197] Xiong ZQ, Dai YJ, Wang RZ. Investigation on a two-stage solar liquid-desiccant (LiBr) dehumidification system assisted by CaCl2 solution. Applied Thermal Eng. 2009; 29(5-6):1209–1215.

[198] Zhang LZ. Heat and mass transfer in a cross-flow membrane-based enthalpy exchanger under naturally formed boundary conditions. International Journal of Heat and Mass Transfer 2007; 50(1–2):151–62.

[199] Huang SM, Zhang LZ, Tang K, Peil X. Fluid flow and heat mass transfer in membrane parallel-plates channels used for liquid desiccant air dehumidification. International journal of heat and mass transfer 2012; 55(9–10):2571–80.

[200] Liu XH, Chang XM, Xia JJ, Jiang Y. Performance analysis on the internally cooled dehumidifier using liquid desiccant. Build Environ. 2009; 44(2):299–308.

[201] Zhang XS, Yin YG, Cao YR. The establishment and the experimental research of energy storage type liquid desiccant cooling air conditioning system. J. Eng. Thermo physics 2004; 25(4):546–9.

[202] Cheng L. The performance and the applied research of liquid desiccant fresh air conditioning system. Ph.D. thesis, Tsinghua University, Beijing, China 2009.

[203] Gandhidasan P. Prediction of pressure drop in a packed bed dehumidifier operating with liquid desiccant. Appl. Therm. Eng. 2002; 22(10):1117–1127.

[204] Chung TW, Ghosh TK, Hines AL. Comparison between random and structured packings for dehumidification of air by lithium chloride solutions in a packed column and their heat and mass transfer correlations. Ind. Eng. Chem. Res. 1996; 35(1):192–198.

[205] Bravo JL, Rocha JA, Fair JR. Mass transfer in gauze packings. Hydrocarb. Process. 1985; 64(1): 91–95.

[206] Al-Farayedhi AA, Gandhidasan P, Al-Mutairi MA. Evaluation of heat and mass transfer coefficients in a gauze-type structured packing air dehumidifier operating with liquid desiccant. Int. J. Refrig. 2002; 25(3):330–339.

[207] Shi MG, Mersmann A. Effective interfacial area in packed columns. Journal of Chem. Eng. 2003; 8(2):87–96.

[208] Gandhidasan P. Estimation of the effective interfacial area in packed-bed liquid desiccant contactors. Ind. Eng. Chem. Res. 2003; 42(14):3420–3425.

[209] Gandhidasan P, Ullah MR, Kettleborough CF. Analysis of heat and mass transfer between a desiccant-air system in a packed tower. J. Sol. Energy Eng. 1987; 109(2):89–93.

[210] Factor HM, Grossman G. A packed bed dehumidifier/regenerator for solar air conditioning with liquid desiccants. Sol. Energy 1980; 24(6):541–550.

[211] Öberg V, Goswami DY. Experimental study of the heat and mass transfer in a packed bed liquid desiccant air dehumidifier. J. Sol. Energy Eng. 1998; 120(4):289–297.

[212] Stevens DI, Braun JE, Klein SA. An effectiveness model of liquid-desiccant system heat/mass exchangers. Sol. Energy 1989; 42(6):449–455.

[213] Khan AY, Sulsona FJ. Modelling and parametric analysis of heat and mass transfer performance of refrigerant cooled liquid desiccant absorbers. Int. J. Energy Res. 1998; 22:813–832.

[214]Khan AY, Ball HD. Development of a generalized model for performance evaluation of packed-type liquid sorbent dehumidifiers and regenerators. ASHRAE Trans. 1992; 98: 525-533.

[215] Khan AY. Sensitivity analysis and component modelling of a packed-type liquid desiccant system at partial load operating conditions. Int. J. Energy Res. 2007; 18(7):643 – 655.

[216] Rahamah AS, Elsayed MM, Al-Najem NM. A numerical solution for cooling and dehumidification of air by a falling desiccant film in parallel flow. Renew. Energy 1998; 13(3):305–322.

[217] Rahmah AS, Elsayed MM, Al-Najem NM. A numerical investigation for the heat and mass transfer between parallel flow of air and desiccant falling film in a fin-tube arrangement. HVACR Research 2000; 6(4):307–323.

[218] Elsayed MM, Gari HN, Radhwan AM. Effectiveness of heat and mass transfer in packed beds of liquid desiccant system. Renew. Energy 1993; 3(6-7):661–668.

[219] Lazzarin RM, Gasparella A, Longo GA. Chemical dehumidification by liquid desiccants: theory and experiment. Int. J. Refrig. 1999; 22(4):334–347.

[220] Dai YJ, Zhang HF. Numerical simulation and theoretical analysis of heat and mass transfer in a cross flow liquid desiccant air dehumidifier packed with honeycomb paper. Energy Convers. Manag. 2004; 45(9-10):1343–1356.

[221] Ali A, Vafai K, Khaled ARA. Analysis of heat and mass transfer between air and falling film in a cross flow configuration. Int. J. Heat Mass Transf. 2004; 47(4):743–755.

[222] Liu XH, Jiang Y, Qu KY. A theoretical model for air dehumidification process in a cross-flow dehumidifier using liquid desiccant. Heat Ventil Air Condition. 2005; 35:115–21.

[223] Park MS, Howell JR, Vliet GC, Peterson J. Numerical and experimental results for coupled heat and mass transfer between a desiccant film and air in cross- flow. Int. J Heat Mass Transfer 1994; 37:395–402.

[224] Hueffed AK, Chamra LM, Mago PJ. A simplified model of heat and mass transfer between air and falling-film desiccant in a parallel-plate dehumidifier. J Heat Transfer 2009; 131(5):052001-1–7.

[225] Liu XH, Qu KY, Jiang Y. Empirical correlations to predict the performance of the dehumidifier using liquid desiccant in heat and mass transfer. Renewable Energy 2006; 31:1627–39.

[226] Liu XH, Jiang Y, Qu KY. Heat and mass transfer model of cross flow liquid desiccant air dehumidifier/regenerator. Energy Convers Manage 2007; 48:546–54.

[227] Luis PL, Ortiz J, Pout C. A review on buildings energy consumption information. Energy and Buildings 2008 40 (3):394–398.

[228] Sandanasamy D, Govindarajane S, Sundararajan T. Solar-assisted cooling systems in green buildings: an overview. Indian Journal of Energy, 2012 1(3):1-6.

[229] Sekret R, Turski M. Research on an adsorption cooling system supplied by solar energy. Energy and Buildings, 2012 51:15–20.

[230] Advantix Systems' Liquid Desiccant technology, Florida (http://www.advantixsystems.com/).

[231] Tanthapanichakoon W, Prawarnpit A. New simple mathematical model of a honeycomb rotary absorption-type dehumidifier. Chemical Engineering Journal 2002 86(1):11-15.

[232] Shen WD, Zheng, Jiang. Engineering thermodynamics, High education press (1983).

[233] Hamed AM, Khalil A, Kabeel AE, Bassuoni MM, Elzahaby AM, Performance analysis of dehumidification rotating wheel using liquid desiccant. Renewable energy, 2005 30(11):1689-1712.

[234] Holman JP. Heat transfer 9th ed., McGraw-Hill, Singapore, (2002).

[235] Garrett DE. Handbook of lithium and natural calcium chloride. Academic Press, (2004).

[236] Linberg, V. Uncertainties and Error Propagation Part 1 of a manual on Uncertainties, Graphing, and the Vernier Caliper, 1 July 2000. Available on the web http://www.rit.edu/cos/uphysics/ uncertainties/Uncertaintiespart2.html#addsub.

[237] Sheridan JC, Mitchell JW. A hybrid solar desiccant cooling system. Journal of Solar Energy, 1985 34(2):187-193.

[238] Galiano A. Performance Assessment of a Solar Assisted Desiccant Cooling System. Thermal Science, 2014 18(2):563-576.

[239] Stefano E, Tiberi V. Dimensioning and Efficiency Evaluation of Hybrid Solar Systems for Energy Production. Thermal Science, 2008 12(1):127-138.

[240] Rachman A. Feasibility study and performance analysis of solar assisted desiccant cooling technology in hot and humid climate. Am. J. Environ. Sci., 2011 7(3) (2011): 207-211.

[241] Arundel AV. Indirect health effects of relative humidity in indoor air environments, Desiccant cooling and dehumidification, ASHARE, Atlanta, GA, 1992.

[242] Maclaine-Cross, IL. High performance adiabatic desiccant open cooling cycles. Journal of solar energy engineering, 1985 107(1):102-104.

[243] Parmar H and Hindoliya DA. Performance of solid desiccant-based evaporative cooling system under the climatic zones of India, International Journal of Low-Carbon Technologies, 2013 8: 52–57.

[244] Panaras G. Theoretical and experimental investigation of the performance of a desiccant air-conditioning system, Renew. Energy, 2010 35(7):1368–1375.

Vitae

Name:	Muhammad Mujahid Rafique
Nationality:	Pakistan
Date of Birth:	March 4, 1989
Email:	mujahidrafique89@gmail.com
Address:	Faisalabad, Pakistan

Educational Qualification:

M.S (Mechanical Engineering)
November, 2015
King Fahd University of Petroleum & Minerals,
Dhahran, Saudi Arabia.

B.Sc. (Mechanical Engineering)
August, 2012
University of Engineering and Technology,
Lahore, Pakistan.

Research Contributions

Journal Papers

1) M. Mujahid Rafique, P. Gandhidasan, S. Rehman, LM. Al-Hadhrami, "A review on desiccant based evaporative cooling systems" Renewable and Sustainable Energy Reviews 45 (2015) pp. 145-159.

2) M. Mujahid Rafique, P. Gandhidasan, S. Rehman, LM. Al-Hadhrami, "Energy, exergy and anergy analysis of a solar desiccant cooling system" Journal of Clean Energy Technologies, 4 (1), (2016) pp. 78-83.

3) M. Mujahid Rafique, P. Gandhidasan, H. Bahaidarah, "Liquid desiccant materials and dehumidifiers-A review" Renewable & Sustainable Energy Reviews, (accepted on November 21, 2015).

4) M. Mujahid Rafique, P. Gandhidasan, S. Rehman, LM. Al-Hadhrami, "Performance analysis of solid desiccant cooling system in conjunction with evaporative coolers under climatic zone of Dhahran, Saudi Arabia" Environmental Progress & Sustainable Energy, (submitted).

5) M. Mujahid Rafique, P. Gandhidasan, H. Bahaidarah, "Modeling and performance investigation of rotary liquid desiccant dehumidifier for air conditioning application" Energy & buildings, (submitted).

Conference papers

1) M. Mujahid Rafique, S. Rehman, P. Gandhidasan, LM. Al-Hadhrami, "Performance Comparison of two different configurations of solid desiccant cooling system operating under hot and humid climatic conditions" 14th International Conference on Clean Energy 2015, September 27 to October 3, 2015,Saskatoon, SK, Canada.

2) M. Mujahid Rafique, P. Gandhidasan, S. Rehman, LM. Al-Hadhrami, "Feasibility analysis of solid desiccant cooling system with indirect evaporative cooler under climatic zone of Dhahran, Saudi Arabia" 14th International Conference on Clean Energy 2015, September 27 to October 3, 2015, Saskatoon, SK, Canada.

3) M. Mujahid Rafique, P. Gandhidasan, H. Bahaidarah, "A mathematical model for predicting the performance of liquid desiccant wheel" ISES Solar World Congress 2015, November 8-12, 2015, Daegu, Korea (Accepted).