Investigation of Desiccant Air-Conditioning and Evaporative Cooling Systems for Agricultural Storage Applications

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INVESTIGATION OF DESICCANT AIR-CONDITIONING AND EVAPORATIVE COOLING SYSTEMS FOR AGRICULTURAL STORAGE APPLICATIONS

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June 2017

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JAPAN

INVESTIGATION OF DESICCANT AIR-CONDITIONING AND EVAPORATIVE COOLING SYSTEMS FOR AGRICULTURAL STORAGE APPLICATIONS

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE AWARD OF THE DEGREE OF

DOCTOR OF ENGINEERING (Dr.Eng.)

By

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DECLARATION

I hereby declare that the work, which is being presented in the thesis entitled *"Investigation of Desiccant Air-Conditioning and Evaporative Cooling Systems for Agricultural Storage Applications"* submitted in partial fulfillment of the requirements for the award of the degree of Doctor of Engineering (Dr.Eng.), Interdisciplinary Graduate School of Engineering Sciences, Kyushu University, Japan is an authentic record of my own research work.

The research work presented in the thesis has not been submitted by me for the award of any other degree in this or any other university.

Muhammad Hamid Mahmood

June 2017

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ALLAH the Almighty never spoils any effort. Every piece of work is rewarded according to the nature of devotion. I invoke ALLAH's blessings, peace for His Holy Prophet "MUHAMMAD" (Peace Be Upon Him), the messenger of ALLAH, the most perfect and exalted among and of even born.

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Muhammad Hamid Mahmood

CONTENTS

SUMMARY

Energy efficient and low-cost agricultural storage is a burning issue of 21st century where postharvest losses (PHL) are ranging from 25-50% of net agricultural produce. Lots of storage techniques have been adopted worldwide in this regard but these are either expensive or inefficient. Therefore, Desiccant Air-Conditioning (DAC) and Evaporative Cooling (EC) systems are investigated in this study in order to establish low-cost, energy-efficient and environmentally benign system. The study focuses on theory of agricultural storage and consequently performs series of experiments for the performance evaluation of desiccant unit. Simplified mathematical models of evaporative cooling are employed in order to opt-in these technologies into the proposed cooling system. The key features of the study are highlighted as follows:

Theory of agricultural storage as well as insights of storage are explained in the very first chapter keeping in view the complex mechanism of respiration, transpiration and fermentation. The PHL in existing agricultural/food production system are elaborated. Factors affecting the PHL in terms of products quality, quantity, storage life and mal/nutrition are correlated. Studied technologies are motivated vis-à-vis typical heating, ventilation and air-conditioning (HVAC) systems, and thermodynamics limitations are discussed. In the second chapter extensive literature has been reviewed for existing and advanced storage practices. In addition to the proposed storage, following storage techniques are reviewed: drying, ice, ventilation, refrigeration, HVAC, hypobaric, hyperbaric, heat treatment, ultraviolet irradiation, and controlled and modified atmosphere.

The chapter three is devoted for evaporative cooling technologies. Direct and indirect evaporative cooling (DEC/IEC) technologies are explained and compared. As all the conventional evaporative cooling is based on wet-bulb cooling conception therefore the present study emphasis on dew-point based advanced indirect evaporative cooling (named as Maisotsenko cycle or M-cycle cooling) for agricultural storage application. In addition to subjected application, the applicability of M-Cycle has also been studied and analyzed for various applications which include cooling tower, condenser, heat recovery from turbines using various cycles (see appendices). It has been found that direct evaporative cooling is only applicable for dry regions or climates where the prime objective is to control the temperature irrespective of air enthalpy. On the other hand indirect evaporative cooling including M-Cycle cooling can be utilized sensibly in order to reduce the air enthalpy and dry-bulb temperature simultaneously. It is worth mentioning that the benefits of M-Cycle integration are obvious due to its dew-point cooling approach.

In contrary to evaporative cooling systems, desiccant air-conditioning is investigated in chapter four for humid climates/regions where the prime objective is to reduce the humidity. An open-cycle experimental apparatus was setup for the performance evaluation of hydrophilic polymer based desiccant blocks. Series of experiments are conducted for various: ambient air conditions, regeneration conditions, cycle time, and switching time. Generalized root sum of squares method is used to calculate the experimental uncertainty, and experimental data can be reproduced within ±2-3% error. It is examined that when humid ambient air passes through the

desiccant unit deep dehumidification occurs. It shows huge potential of air dehumidification at low regeneration temperature (50-60ºC). It is determined that under the relatively dry ambient air conditions equivalent heat of adsorption (q_{eq}) profile approaches to net value equals zero. The average effective dehumidification slightly increases under humid ambient air conditions by changing the switching time ratio from 1:1 to 2:3 due to higher process air relative humidity. However, it keeps decreasing with increase in switching time ratio for relatively dry ambient conditions. It has been found that the switching time depends on dehumidification amount, nature of application and operating conditions. From the bunch of experiments, it has been concluded that the switching time ratio of 1:2, 2:3 and 1:1 can be selected for the operation of DAC system for high, medium and low humidity operating conditions, respectively. The desiccant dehumidification process should follow isenthalpic line on psychrometric chart in an ideal scenario when there is no adsorption heat. The slope of dehumidification line on psychrometric chart is therefore modified for the realization of real desiccant dehumidification process based on experimental data. Consequently a simplified correlation is developed by which real desiccant dehumidification process can be predicted on psychrometric chart for polymeric desiccant. In addition the study also highlights the applicability of few polymer- and carbon- based nano and micro sorbents for agricultural storage (see appendices). The correlation leads toward the stead-state analysis of DAC systems.

In chapter five, six kinds of desiccant air-conditioning systems are proposed and analyzed for the summer conditions of Fukuoka (Japan). In each system latent load of AC was accomplished by desiccant unit whereas sensible load of AC was accomplished by M-Cycle evaporative cooling. Storage compatibility of agricultural products is crucial to maintain the quantitative and qualitative attributes. In this regard, three different compatible groups of postharvest agricultural products (fruits and vegetables) according to their temperature and relative humidity (RH) are established for the psychrometric presentation of ideal storage zones. In case of dried fruits the ideal temperature and RH zones are established using Guggenheim, Anderson and De-Boer adsorption model. The effects of temperature on the storage life of dried fruits are realized by which it is found that shelf life reduces 50% by increase of 10ºC temperature. It has been found that the system without pre-evaporative cooling on regeneration side, require less regeneration heat due to the provision of regeneration air stream to the heat exchanger (HX) at higher dry-bulb and lower dew-point temperature. It also provides higher dehumidification and wet-bulb/dew-point effectiveness. Similarly system pre-cooling does not give positive outcomes even when HX is not integrated in to the systems. On the other hand, system with pre-evaporative cooling on regeneration side in the absence of MEC enables higher cooling capacity, supply air relative humidity and COP.

The study concludes that the evaporative cooling (preferably M-Cycle) systems should be considered on top priority for agricultural storage applications wherever these are thermodynamically and meteorologically applicable. When these are not applicable, thermally driven DAC systems could yield advance agricultural storage system which can control temperature and humidity distinctly irrespective to conventional compressor based AC systems. Therefore, integration of evaporative cooling unit(s) into DAC system will lead towards energy-efficient and reliable low-cost AC systems for various applications. However optimum operational conditions will need to be determined and regulated for particular application. Herein it has been concluded that one or other DAC system could be efficiently utilized for the storage of agricultural products.

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CHAPTER 1

BACKGROUND OF THE STUDY

CHAPTER 1

BACKGROUND OF THE STUDY

This chapter describes the fundamentals background about the necessity of agricultural products storage and utilization of desiccant air-conditioning system for the purpose. The key factors responsible for postharvest losses (PHL) are determined. The role of temperature, relative humidity for storage of agricultural products for extended period is highlighted. The complex mechanism of respiration, transpiration and fermentation of agricultural products is explained in order to maintain the maximum quantity and quality of agricultural products and to avoid malnutrition. Motivation towards the use of desiccant air-conditioning and evaporative cooling systems for product storage instead of conventional vapor compression systems is explained from thermodynamics and product quality point of view.

1.1 Introduction

World population has crossed 7.3 billion by 2015 (FAOSTAT, 2015), which will reach to 9.1 billion by 2050 (FAO, 2009). In this scenario, additional agricultural produce has to be grown to combat food shortage (Hodges et al., 2011). In contrary, there are huge postharvest losses (PHL) in the existing agricultural/food production system. According to a study (Lal Basediya et al., 2013), higher moisture contents (60-95%) are usually the key reason for PHL which ultimately shorten the shelf life. The PHL are the losses of quality and quantity of the agricultural products. It is important to mention that the level of postharvest losses is different in different countries. However, 20-30% can be set an average as reported by Atanda et al., 2011 and El-Ramady et al., 2015. In case of developing countries (particularly along the tropical belt)

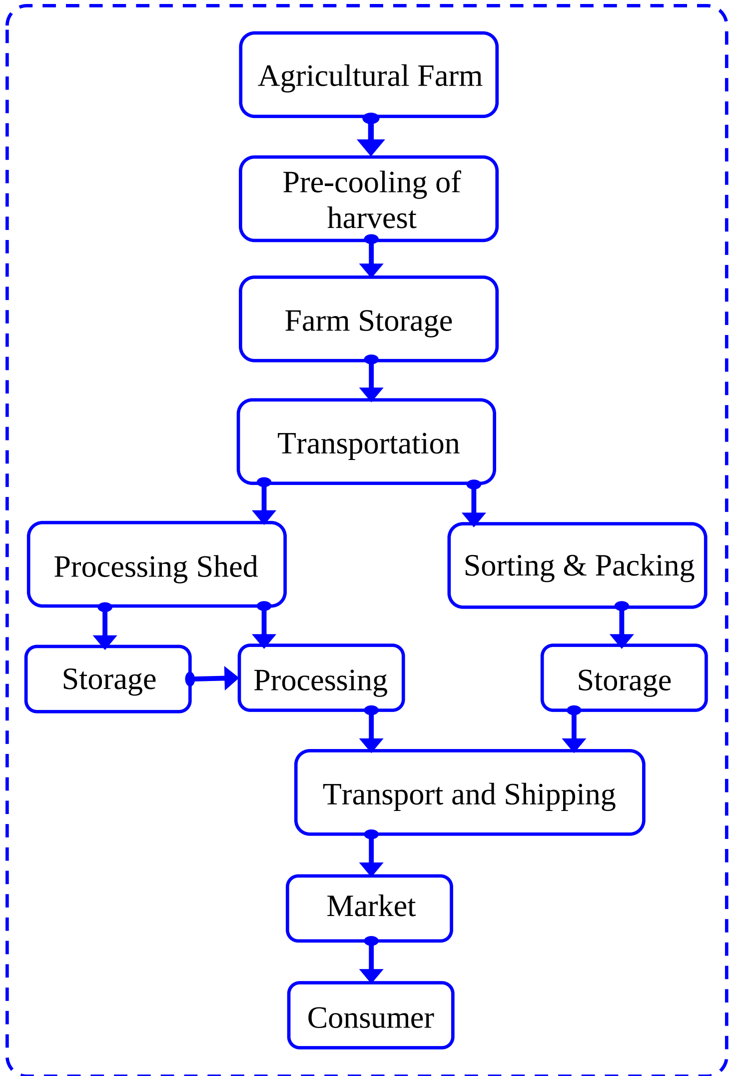

Figure 1.1 Typical postharvest chain of agricultural products (reproduced from Kader and Rolle, 2004; Mishra and Gamage, 2007).

these losses may exceed from half of the actual produce (Atanda et al., 2011; Burden and Wills, 1989; El-Ramady et al., 2015; Sanzani et al., 2016). Such losses may occur at any stage throughout the postharvest chain such as shown in Figure 1.1 (Kader and Rolle, 2004; Mishra and Gamage, 2007). However, the PHL can be minimized by cooling and storing the agricultural products on-farm and/or ex-farm right after their harvest that ultimately will increase their shelf/storage life. In this context, it has been reported that the storage/shelf life of the products can be enhanced by avoiding the postharvest cooling delays such as the shelf life of lettuce can be extended to 12 days by keeping it under optimal environmental conditions within one hour after the harvest (Agüero et al., 2014). Contrary, it has also been reported in the literature that the cooling delay of 3 hours at 37°C or 6 hours at 24°C after harvesting lowers the marketable quality and weight of the Japanese eggplants (Cantwell et al., 2012). Likewise, the cooling delay of just 3 hours after the harvest reduces the shelf life of the broccoli because of floret openings which ultimately affect its visual appearance (Brennan and Shewfelt, 1989) and cooling delay of 48 hours at 20°C shortened the storability of European plum due to increased internal breakdown (Guerra and Casquero, 2009).

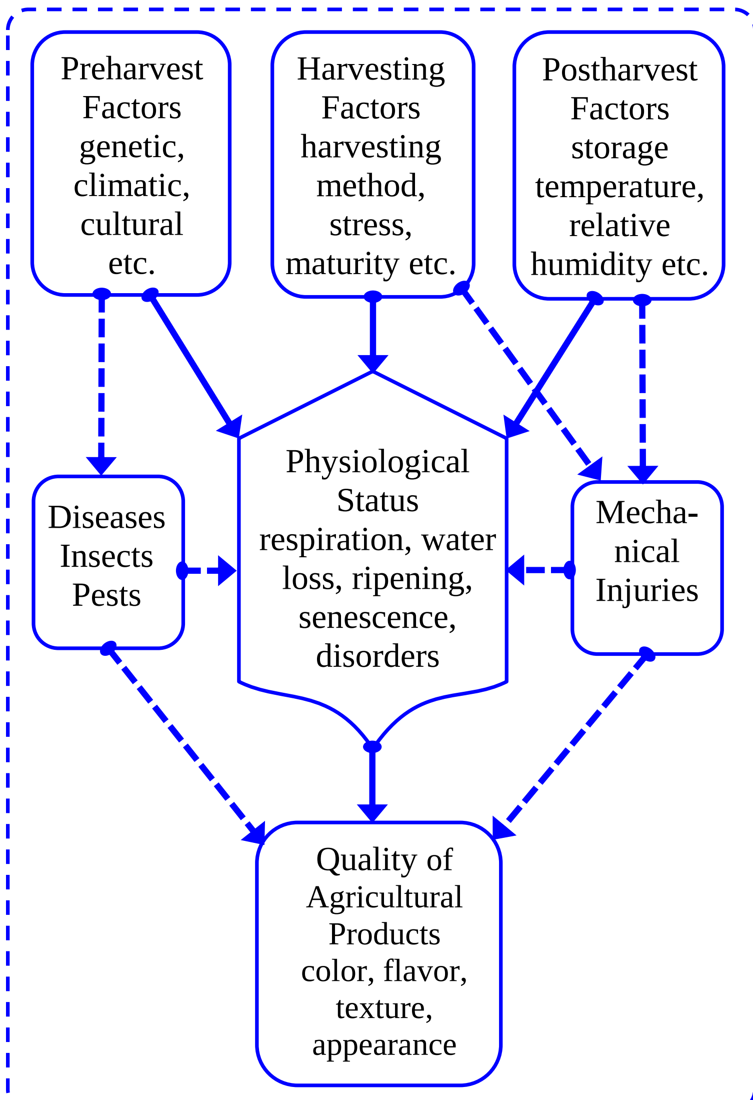

Figure 1.2 Key factors affecting the product quality (reproduced from Mishra and Gamage, 2007).

From the above prospective, the key factors responsible for postharvest losses are determined as shown in Figure 1.2 (El-Ramady et al., 2015; Mishra and Gamage, 2007). It is important to mention that the preharvest and harvest factors cannot be avoided/minimized after the harvesting, however, the postharvest factors can be controlled to slow down the decay process in the agricultural products.

The management of the postharvest factors is crucial for extending the shelf/storage life of the harvested products with maximum quantity and quality. Otherwise, the inadequate consumption and/or availability of fruits and vegetables are causing about 2.7 million deaths per year globally, and contribute 1.8 % worldwide diseases (Lock et al., 2005). The rationale about insights of agricultural product storage is described in detail in heading 1.2.

1.2 Insights of Storage

Postharvest losses mainly depend on temperature and relative humidity (Mahmood et al., 2016). The postharvest agricultural products perform respiration and require certain level of oxygen as described by Eq. (1.1) (ASHRAE, 2010).

$$C_6H_{12}O_6 + 6O_2 \rightarrow 6CO_2 + 6H_2O + respiration\ heat \qquad (1.1)$$

During this process ambient air oxygen reacts with the reserve sugar/starch of the harvested products and breakdown it into carbon dioxide (CO_2), water and heat energy (2667 kJ) is released during this reaction. The water produced during this reaction (Eq. (1.1)) remained within the product tissue, however the CO_2 releases and resulted in 3-5% weight loss of the product (Rao, 2015). The ratio between the volume of carbon dioxide liberated to the volume of oxygen absorbed is known as respiratory quotient (R_q). The value of the respiratory quotient for aerobic respiration varies from 0.7 to 1.3 depending upon the type of substrate being oxidized (e.g. in case of carbohydrates R_q = 1) (Rao, 2015). Moreover, the rapid removal of respiration heat is always required from the package and/or cold storage. It is worthy to mention that every agricultural product enables specific respiration rate which can be calculated from the following relationship (ASHRAE, 2010).

$$R = \frac{10.7\,\theta_1}{36 * 10^5}\,(1.8\,T + 32)^{\theta_2} \tag{1.2}$$

where R and T are respiratory heat generation rate [W/g] and product temperature [°C], respectively. The parameters θ_1, θ_2 are respiratory coefficients which varies product to product (ASHRAE, 2010; Becker and Fricke, 1996). It can be notice from Eq. (1.2) that respiration rate mainly depends on the temperature which influences the decay/aging process of the products. In simple words, higher the respiration rate, shorter will be the product shelf life, and lower the respiration rate, longer will be the product shelf life. The postharvest respiration rate in almost all fresh vegetables (instead of root crops) remained high initially for 1-2 days. Later, it quickly lowers and achieves the equilibrium rate. On the other hand, the fresh fruits which do not ripen during storage (e.g. citrus fruits, grapes etc.) have fairly constant respiration rate. However, the ripening of fruits (e.g. apples, pears etc.) during storage increases the respiration rate. It is important to mention that when fruits are stored at relatively higher temperature (about 10-15°C) the respiration rate first increases due to ripening and later decreases, contrary, at low storage temperatures (0°C) no ripening takes place and accordingly almost no respiration heat produces by the products (ASHRAE, 2010). So, the temperature is most important factor in extending the postharvest shelf life of the agricultural products as the respiration rate is the function of the temperature.

It is important to mention that good supply of fresh air (with oxygen concentration about 20%) is crucial for normal respiration process in the agricultural products (Burden and Wills, 1989). The oxygen concentration (<10%) may control the respiration rate and slow down the aging (Rao, 2015) but the sufficient level of the oxygen (>2%) is always required to avoid the fermentation (Burden and Wills, 1989; Mishra and Gamage, 2007; Rao, 2015). Fermentation is an anaerobic respiration (R_q > 1.3) process in which sugar from agricultural product breaks down into ethanol, CO_2, and heat energy (92 kJ) is liberated during this reaction (Eq. 1.3). The poor ventilation of the cold storage due to restricted fresh supply air may promote the fermentation. Fermentation causes the unpleasant flavor, decay and early aging in agricultural products (Mishra and Gamage, 2007; Rao, 2015). The excessive accumulation of the CO_2 around the agricultural products (due to poor ventilation) also causes the other physiological changes/disorders in them (Burden and Wills, 1989; Mishra and Gamage, 2007; Rao, 2015). In this regard, black heart potato disease due to higher carbon dioxide and lower oxygen concentration during storage/shipment is well known in the literature (Davis, 1926; Boyd, 1951).

$$C_6H_{12}O_6 \rightarrow 2C_2H_5OH + 2CO_2 + fermentation\ heat \quad (1.3)$$

In order to maintain the quality of agricultural products, the relative humidity is also an important parameter during transpiration of the postharvest products. The transpiration is loss of water from the postharvest agricultural products. It involves the transport of moisture through the skin, evaporation, and convective mass transport of moisture to the surroundings (Becker et al., 1996). The endothermic evaporation at product surface cools its surface, lowers the vapor pressure and thus resulting in reducing the transpiration. On the other hand, the respiration within the agricultural product increases the surface vapor pressure due to raising product temperature and consequently the transpiration increases (Gaffney et al., 1985). The loss of moisture affects the products' physical appearance (wilting, shriveling), flavor, texture (softening, juiciness, limpness, flaccidity, crispness etc.), net weight and ultimately nutritional value (Lal Basediya et al., 2013; Mishra and Gamage, 2007). Five to ten percent reduction in fresh weight of agricultural product due to excessive transpiration (moisture loss) damages the product quality (Burden and Wills, 1989). The high relative humidity cannot inhibit the moisture loss if the product temperature is not close to the air temperature (Paull, 1999). The net transpiration rate varies from product to product and can be influenced by air temperature, relative humidity, flow rate, surface air to volume ratio, nature of surface coating, atmospheric pressure, mechanical damage etc. (Mishra and Gamage, 2007; Rao, 2015). However, it varies mainly with relative humidity, air flow rate and skin mass transfer coefficient. In other words, water vapor pressure deficit (VPD) between the product and surrounding is the primary driving force for transpiration (Sultan et al., 2016). The transpiration rate (q) [ng/kg·sec] is a linear function of VPD and can be calculated by Eq. (1.4) (ASHRAE, 2010).

$$q = k_t\ (P_{vs} - P_a) \quad (1.4)$$

where the parameters P_{vs} [Pa], P_a [Pa] and k_t [ng/kg·sec·Pa] represents the saturated water vapor pressure, actual water vapor pressure and transpiration coefficient, respectively. The transpiration coefficient (k_t) is considered constant for particular product to simplify the calculations. The pictorial representation of the mechanism involved in photosynthesis and transpiration in the growing plant, and the respiration and transpirations processes in the harvested product is shown in Figure 1.3 (Burden and Wills, 1989).

In case of dried agricultural products the respiration rate is very low as compare to fresh agricultural products. The dried products are more sensitive to the surrounding moisture conditions in the cold storage. If the moisture contents in the storage environment increases the product will absorb more moisture and becomes highly prone to the deterioration. Contrary, if the moisture contents decrease the product will lose moisture and its weight will be reduced. The reduction in product weight will result in economic loss (Pahlevanzadeh and Yazdani, 2005). In this regard, it is crucial to know about the equilibrium moisture content (MC_e) of the product in order to identify the required storage conditions (T and RH). The MC_e depends upon the temperature, water activity and the product type. The water activity (a_w) is a temperature dependent intrinsic property of the product. It represents the decimal form of the equilibrium relative humidity (RH_e) at given temperature and moisture content. The RH_e is the property of the atmosphere/storage environment (i.e. extrinsic property) in equilibrium with the product. The water activity becomes equal to the decimal form of the relative humidity at equilibrium moisture content condition (Wilhelm et al., 2004). In short, the main controlling factor towards the preservation of dried agricultural products is water activity. In this context, the dried fruits become unacceptably hard upon losing their moisture below 0.5-0.7 a_w (Kochhar and Rossell, 1982; Taoukis et. al., 1997). The self-explanatory stability map of agricultural products as a function of water activity is shown in Figure 1.4 (Labuza, 1975; Taoukis et. al., 1997). It can be seen from Figure 1.4 that the typical products (like starch based snack food, cake mixes, crackers etc.) at 0.3 a_w are more stable against the deterioration/spoilage due to lower rate of lipid oxidation, nonenzymatic browning, enzymatic activity and consequently no growth of molds, yeasts and bacteria. The further increase in the water activity results in higher probability of the food deterioration due to growth of certain microorganisms. The dried fruits in the range of 0.6-0.7 a_w can be affected by the microorganisms (some molds and yeasts). The fresh agricultural products possess higher water activity ($a_w \geq 0.95$) and their quality is mostly affected by the bacteria (Wilhelm et al., 2004).

From the above detail discussion, it is generally concluded that the temperature and relative humidity are the key controlling factors for the stable storage of fresh and dried agricultural products for extended period.

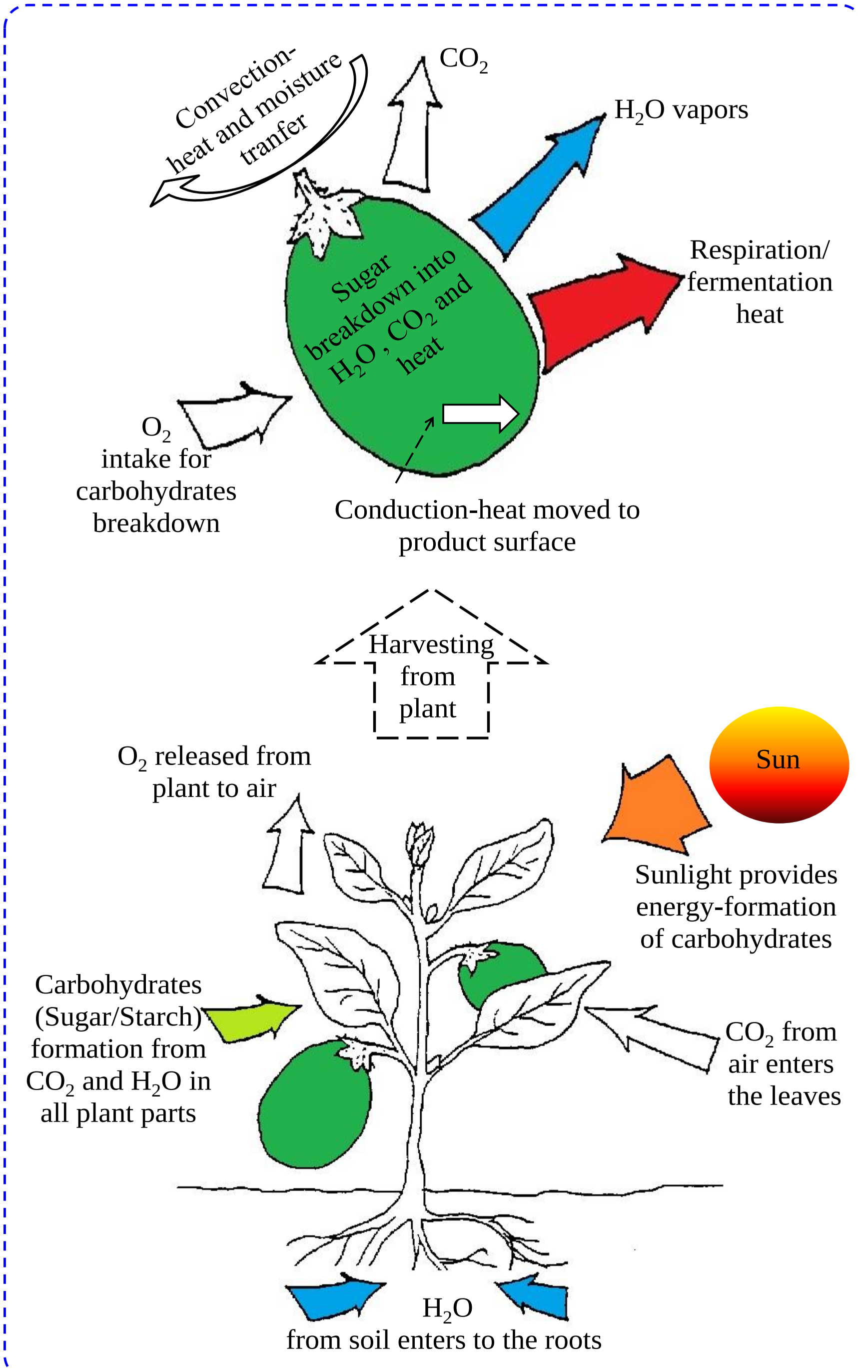

Figure 1.3 The pictorial representation of mechanism of photosynthesis, transpiration and respiration (reproduced from Burden and Wills, 1989).

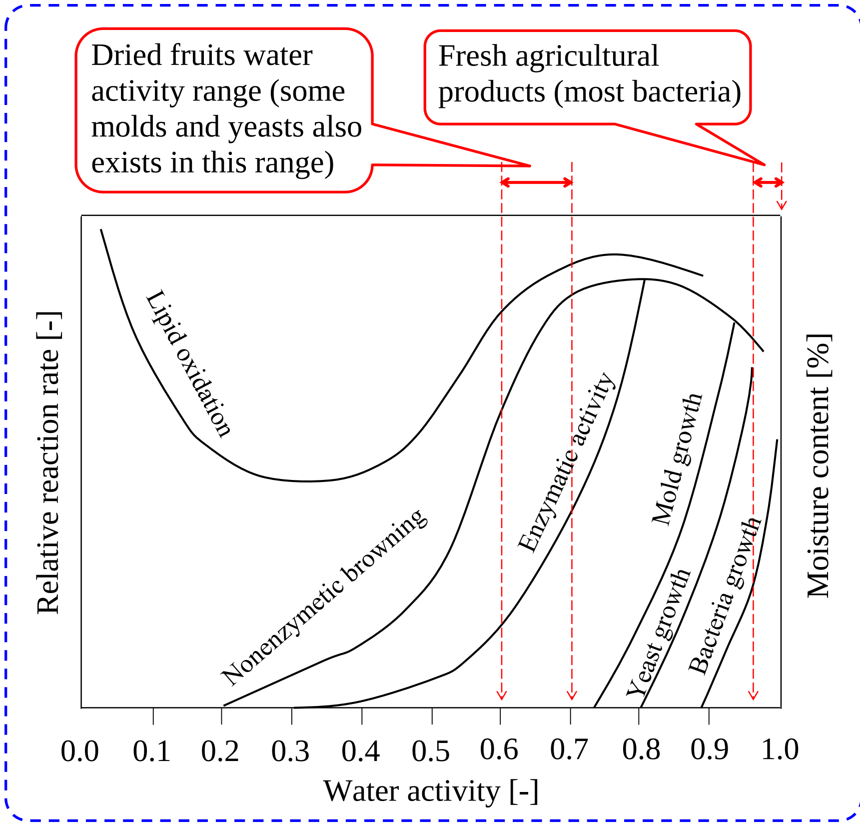

Figure 1.4 Generalized overview of stability map of agricultural products (reproduced from Labuza, 1975; Taoukis et. al., 1997).

1.3 Motivation of the Study

The conventional vapor compression refrigeration (VCR) and/or air-conditioning (VAC) systems are usually being used for storage of agricultural products. Such systems can regulate temperature precisely whereas humidity is controlled indirectly (Paull, 1999; Sharkey and Peggie, 1984). Moreover, anaerobic respiration may be prompted due to unavailability of the oxygen as discussed in heading 1.2. Therefore, conventional systems are difficult to provide optimum conditions for the storage of agricultural products. It is mainly due to the uneven control of humidity and/or ventilation. These systems can also cause chilling injury, off-flavor and discoloration in agricultural products e.g. mango, banana, tomato, leafy vegetables etc. (Ndukwu and Manuwa, 2015; Olosunde et al., 2016). Chilling injury can cause internal browning, surface pitting and also give rise to decay vulnerability in tropical root and tuber crops

(Kitinoja and Kader, 2002). Apart from other quality losses, the loss of "Vitamin C" from the product become greater due to higher temperature, lower relative humidity and chilling injury during storage (Lee and Kader, 2000; Paull, 1999). Moreover, ninety percent of "Vitamin C" in human diet is supplied through agricultural products (Lee and Kader, 2000). It is crucial for prevention of scurvy (low red blood cells, gum diseases, skin bleeding), arteriosclerosis (restriction of blood flow to organs and tissues), cardiovascular diseases, some forms of cancer and maintenance of healthy skin, gums and blood vessels (Harris, 1996; Lee and Kader, 2000). In this regards, conventional vapor compression air-conditioning (VAC) might be considered in order to maintain the storage conditions. However, VAC systems consume primary energy and have certain thermodynamic limitations which include: limitation of fresh air, poor ventilation (excessive CO_2) etc. In addition the VAC system cannot control the temperature and relative humidity distinctly as depicted on Figure 1.5 (Sultan et al., 2015; Sultan et al., 2016). In this perspective, low-cost environmental friendly evaporative cooling technologies like direct evaporative cooling (DEC), indirect evaporative cooling (IEC)/M-Cycle evaporative cooling (MEC) have shown potential to provide storage conditions for particular agricultural products. However, these technologies cannot be used effectively for the storage of agricultural products under largely varying ambient conditions (particularly humid) due to limited cooling performance. The scope of DEC and IEC/MEC (for detail see chapter 2 and chapter 3) for the storage of wide range of agricultural products under varying environmental conditions can be extended by the integration of desiccant dehumidification. The desiccant air-conditioning (DAC) comprises of desiccant dehumidification cum evaporative cooling has ability to deal the latent and sensible load of air-conditioning distinctly. Furthermore, the DAC system can regulate relative humidity by means of desiccant whereas in case of VAC system it is only possible by cooling below dew-point temperature. Furthermore, DAC system is an environmental friendly technology and enables zero global warming and ozone depletion potential. It is operated on thermal energy preferably solar heat and/or biogas/mass available at the farm. Its design possesses better ventilation and consequently may help to control respiration and transpiration. The existing DAC system have shown the worth for various applications e.g. buildings (Enteria et al., 2009; Enteria et al., 2010); supermarkets, schools, hotels, ice arenas, cold warehouses, hospitals, theaters (Dabrowski, 1998); greenhouses (Longo and Gasparella, 2015; Sultan et al. 2014); automobiles (Nagaya et al., 2006); marine ships (Guojie et al., 2012; Zhu and Chen, 2014), museums (Ascione et al., 2009; Ascione et al., 2013); wet markets (Lee and Lee, 2013) product storage and preservation (Sultan et al. 2016). Therefore, the present study investigates the DAC and Evaporative Cooling systems for the storage of agricultural products.

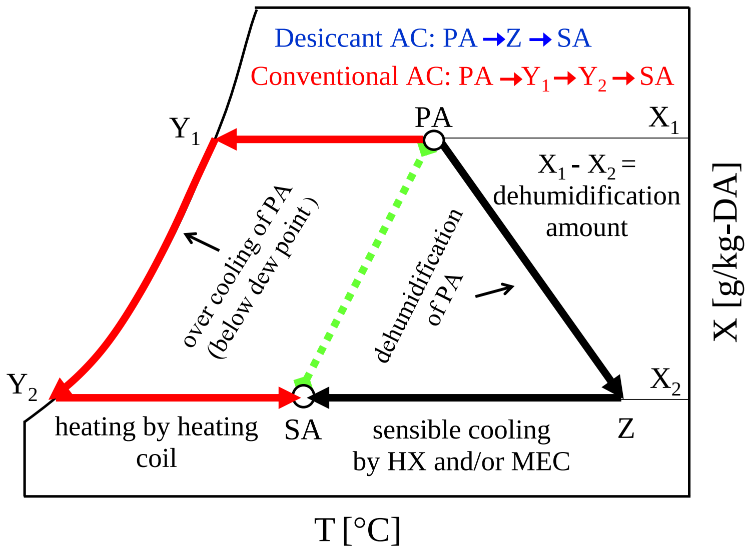

Figure 1.5 Psychrometric comparison between working principle of VAC and DAC.

1.4 Scope and Objectives

The scope and objectives of the work presented in this thesis are as follows

- To investigate the practical feasibility of Maisotsenko Cycle based evaporative cooling systems for farm air-conditioning applications based on following conceptions:
    - Wet-bulb and dew-point evaporative cooling
    - Sensible load control limited to dry conditions
- To develop simplified methodology for performance evaluation of desiccant unit.
    - Various ambient and regeneration air conditions (five cases)
    - Desiccant response analysis and correlation formulation
- To investigate various desiccant air-conditioning systems' configurations for humid areas for agricultural applications.
    - Six DAC systems' configuration and analyses
    - Selection of regeneration temperature and system COP
    - Overall systems' applicability for different applications

1.5 Thesis Outline

This thesis comprises of seven chapters and four supporting appendices in order to achieve the scope and objectives of the presented work. The summary of each chapter is as follows

Chapter 1 describes the fundamentals background about the necessity of agricultural products storage and utilization of desiccant air-conditioning system for the purpose. The key factors responsible for postharvest losses (PHL) are determined. Importance of reducing the PHL on the extension of products storage/shelf life is highlighted and typical postharvest chain of agricultural products right from agricultural farm to the end user is presented. The complex mechanism of respiration, transpiration and fermentation of agricultural products is explained in order to maintain the maximum quantity and quality of agricultural products and to avoid malnutrition. The motivation towards the use of DAC for product storage instead of conventional vapor compression refrigeration machines is explained from thermodynamic and product quality point of view.

Chapter 2 presents the extensive literature review about the agricultural product storage techniques. First part of the chapter covers the conventional and advanced storage techniques. The comparative analysis is made between different types of environmentally benign solar dryers. The methodology involved in natural, artificial, iced and ventilated storage is briefly reviewed along with merit/demerits of each technique. Salient features of advanced storage techniques are also explained along with their limitations. Second part of chapter includes the insight review about the vapor compression refrigeration and/or air-conditioning, evaporative cooling based AC, thermally driven desiccant based AC systems for agricultural products storage. The working principle and features of these systems are elaborated in this regards. Moreover, the solar energy operated single-stage DAC, multi-stage DAC and hybrid systems are also reviewed in order to extend the scope of DAC system for product storage in multi-climatic regions.

Chapter 3 briefly explains the principle and features of advanced dew-point Maisotsenko Cycle evaporative cooling (MEC) in comparison with conventional evaporative cooling. The potential applicability of low cost environmental friendly evaporative cooling technologies like conventional direct and indirect evaporative cooling and M-Cycle evaporative cooling for agricultural products storage is discussed in conjunction with Appendix B. The broad spectrum applications of M-Cycle are also presented in this chapter. Various modifications in MEC design are discussed in order to investigate its applicability in humid regions. It is found that desiccant based MEC systems enable huge energy saving potential to achieve the sensible and latent load of AC in humid regions.

Chapter 4 presents the setup of an open-cycle experimental apparatus for desiccant AC applications. Honeycomb like cubical shaped desiccant blocks are used for experimental dehumidification analyses. The orientation of the each unit of the experimental setup is explained briefly along with pictorial representation. The operation of each unit of the experimental setup is also explained distinctly. The efficacy of the supplementary heating unit to the increase regeneration air stream temperature is highlighted. The procedure adopted during regeneration and process air experimentation is described and experimental data recoding sheet is also presented in this chapter. The generalized root of sum of squares method is used to calculate the experimental uncertainty. It is carried out for measured and/or calculated variables in order to determine the accuracy of the experimental data. The experiments are performed under different ambient conditions, and various regeneration air temperatures and switching time ratios (five cases). The net and average effective dehumidification performances of desiccant unit are determined and presented. Influence of regeneration temperature on equivalent heat of adsorption is ascertained. The temporal variations in wet-bulb and dew-point temperature differences are presented. The optimized switching time ratio between regeneration and dehumidification of desiccant unit is suggested. Various scenarios of steady-state adsorption are explored and possible options are figure out for the development of simplified correlation for desiccant performance evaluation. In this regard a novel correlation is developed on the conception of modification of isenthalpic slope of dehumidification line on psychrometric chart. The experimental validation of slope of dehumidification under varying regeneration temperature is also investigated.

Chapter 5 presents the experimental investigation of the combined effect of desiccant dehumidification and sensible cooling (via MEC or HX) for the storage of agricultural products. The ideal storage zones of agricultural products, dried fruits, dried foods & feeds in comparison with greenhouse growth and humans thermal comfort zones are established on the psychrometric chart. Six different configurations of DAC systems are proposed and their performance evaluation is made under the ambient conditions of Fukuoka-Japan. A simplified methodology is developed for performance evaluation of proposed DAC systems under varying regeneration temperatures. The parametric and thermodynamic analysis of all the system configurations (S-I to S-VI) is made to investigate that which configurations could yield better system performance. The psychrometric evaluation of three optimized DAC systems is performed for the storage of agricultural products, dried fruits, dried foods & feeds in comparison with greenhouse and humans AC.

Chapter 6 includes the general conclusions of this study and suggested future work.

1.6 Nomenclature

a_w	water activity [-]
DAC	desiccant air-conditioning
DEC	direct evaporative cooling
IEC	indirect evaporative cooling
k_t	transpiration coefficient [ng/kg·sec·Pa]
MC_e	equilibrium moisture content [%]
MEC	Maisotsenko cycle evaporative cooling
P_a	actual water vapor pressure [Pa]
PHL	postharvest losses
P_{vs}	saturated water vapor pressure [Pa]
q	transpiration rate [ng/kg·sec]
R	respiratory heat generation rate [J/sec/g]
RH	relative humidity [%]
R_q	respiratory quotient [-]
T	temperature [°C or K]
VAC	vapor compression air-conditioning
VCR	vapor compression refrigeration

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CHAPTER 2

REVIEW OF LITERATURE

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REVIEW OF LITERATURE

This chapter presents the extensive literature review about the agricultural product storage techniques, conventional vapor compression based AC, evaporative cooling based AC, thermally driven desiccant based AC and hybrid AC systems. The salient features of advanced storage techniques are described in depth along with their limitations. The insights review about various air-conditioning systems for agricultural products storage is made. Moreover, the solar energy operated single-stage DAC, multi-stage DAC and hybrid systems are also reviewed in order to extend the scope of DAC system for product storage in multi-climatic regions.

2.1 Introduction

The agricultural products (fruits and vegetables) are perishable in nature and possess short shelf life under ambient conditions. These products are cultivated and harvested during specific period/duration in a year. However, their market demand mostly remains throughout the year. In this scenario it is required to store the excess production in order to meet the supply and demand gap. The storage is also made to get higher/reasonable returns in off-season. Storage period can be short, mid or long term depending on product type, usage, perishability, internal characteristics, physiology etc. (Camelo, 2004; Rao, 2015). The highly perishable agricultural products like raspberries and other cultivars of berries can only be stored for short period. The products (e.g. mangoes, bananas, cabbages, brinjals, tomatoes etc.) can be stored for midterm period (few weeks) depending on the produce quality, type and the market demand. However, the long term storage of onions, potatoes, sweet-potatoes, garlic, squash, pumpkins, oranges, apples can be made in order to attain the good market price. Moreover, the remaining surplus products, after fulfilling storage requirements, can be preserved by drying. The shelf life of dried agricultural products can be extended from six months to one year or even more by providing optimal storage conditions (temperature and relative humidity). Such controlled conditions are essential because dried agricultural products require certain percentage of storage moisture for technical, practical; and commercial point of view (Dauthy, 1995). The details about these storage/preservation techniques are explained under subsequent headings.

2.2 Storage Techniques

2.2.1 Drying

The drying is considered as an oldest technique for preservation of agricultural products (Dauthy, 1995). The simultaneous heat and mass transfer during the drying process result in removal of moisture from the product (El-Sebaii, 2012). The dried agricultural products possess longer shelf life and can be transported easily due to lighter weight. These products also required less space for storage (Dauthy, 1995; Ertekin, 2004).

The surplus amounts of postharvest fresh agricultural products are traditionally preserved by exposing directly to the sun. It is widely used common method of preservation in the developing countries. The products are spread over the floors and exposed to the natural solar energy directly for certain period of time depending on the product type, climatic conditions etc. (Greig and Reeves, 1985; Proctor, 1994). However, the adverse climatic conditions, attack of birds, insects/pests can deteriorate the quality and quantity of the naturally dried agricultural products (El-Sebaii, 2012). In this regards, advancements have been made in the solar drying methods in parallel to the traditional natural solar drying in order to minimize the losses (qualitative, quantitative, nutritive etc.). These drying methods can be broadly classified in to four different types (as follow) depending on the mode of energy supplied to the agricultural products for the removal of moisture (El-Sebaii, 2012; Furlan et al., 1983; Kumar et al. 2016).

- Natural Solar Dryer
- Direct Solar Dryer
- Indirect Solar Dryer
- Mixed/Hybrid Type Solar Dryer

In natural solar drying the agricultural products are placed on the ground or above ground surface. The plastic/tarpaulin sheets are usually placed beneath the products (Greig and Reeves, 1985). The products are exposed directly to ambient environmental conditions (solar energy, wind, rain) and infestation of birds, insects/pests etc. This method of drying is quite cheap but it causes higher loss of product quality and quantity (El-Sebaii, 2012; Fudholi et al., 2010). However, in direct solar dryer the products are placed in small wooden/metallic chambers/cabinets. The drying chamber is covered with transparent (polyethylene) sheet/cover. It is used to avoid the products from adverse environmental conditions, infestation of birds, insects/pests etc. (El-Sebaii, 2012; Fudholi et al., 2010). The products which are going to dry are placed inside the dyer chamber. The solar radiations pass though the sheet (polyethylene) and absorbed by products. Thus, the absorption of solar energy by the products results in removal of moisture from products and moved away due to air circulation. Though such dryers are cheaper and simple in construction but cannot be used widely due to limited product drying capacity and damaging its quality due to overheating (El-Sebaii, 2012; Kumar et al. 2016).

The indirect solar dryer overcome the demerits associated with the direct solar dryer. It consists of solar collector, drying unit, air ducting, and, blower (in case of active indirect solar dryer) (Esper and Mühlbauer, 1998). The solar collector and drying units are separately placed and connected with each other through ducting unit. The air heated by the solar collector is moved to the drying unit/chamber through air ducting unit. The better operating conditions (higher temperature, air mass flow rate etc.) can be achieved by this type of dryer.

In mixed/hybrid type solar dryer the products receive direct solar energy (as in direct solar dryer) and pre-heated air is also supplied to the product. The supply of pre-heated air is made by solar/electric or bio-mass/bio-gas heater etc. (Kumar et al. 2016). The rapid drying of agricultural products can be made by such type of dryers. In this regard, Prasad et al., 2006, developed a hybrid solar cum biomass energy operated dryer for the drying of turmeric. This hybrid system dried the turmeric rhizomes in only 1.5 days, whereas traditional natural solar drying took 11 days for such drying.

2.2.2 Natural and Artificial Farm Storage

In natural storage method the crops belong to roots and tubers group are left underground for several weeks/months. Some of root and tuber crops include carrot, sweet-potato, cassava, yam, garlic and potato etc. (Rao, 2015). These crops are harvested before the start of rainy season and/or when the demand arising from the markets in order to get the good market value (price) of the produce (Camelo, 2004). This storage technique is very cheap because it does not require the separate storage structure/building. So it does not involve any fixed and/or variable cost of operation. However, the products quality is affected by insects/pests and disease attack.

In the artificial storage method the products (e.g. onion, potato, sugar beet, sweet-potato, carrot turnip etc.), unlike the natural farm storage, are first harvested from plant and then store on farm. These products are stored by making heap/mound like structure and/or by digging a trench/pit. The products are protected from field moisture and weather conditions by covering with straw, plastic-sheets, tarpaulin (polyethylene), and soil (Rao, 2015). The extra available bin/barrel filled with products can also be buried in the soil for the purpose (Kitinoja and Kader, 2002). Another alternative of this method is to place the products container/bins on top of each other in the field (Camelo, 2004). The topmost container is finally protected from weather. This alternative facilitates the mechanized handling of the containers.

2.2.3 Iced and Ventilated Storage

In iced storage method, the ice was used (in the warehouse) for the preservation of meat, fish and perishable agricultural products. The ice was harvested in winter from the frozen lakes/ponds. It was stored in the ice storehouses and utilized as refrigerant in the product’s cold storages when needed. The ice starts melting by absorbing product heat. The water discharge due to ice melting is a major drawback of this storage technique. About 325 kJ heat is absorbed due to melting of 1 kg ice (Rao, 2015). The small size ice box, now mostly used for transporting food/medicine at domestic and/or commercial level, is an advance form of iced storage.

The ventilation of the stored products is utmost important for the removal of the heat and humidity produced due to respiration/transpiration. In this regard natural or forced air ventilation can be employed in the storage structure. The storage structure can be constructed on ground and underground. Cellar is an example of underground ventilated storage structure. Cellar provides good storage conditions in the cold and snowy regions. It is mostly built for storage of products like potato, onion, turnip, garlic, sugar beet etc. It can be constructed into hillside with shingle roofs covered by sods and soil (Bubel and Bubel, 1979; Hobson, 1981; Rao, 2015).

2.2.4 Refrigerated Cold Storage

Cold storages are the structures (buildings) equipped with refrigeration system along with necessary facilities and gadgets. The conventional vapor compression refrigeration (VCR) systems are mostly used in cold storages facilities. The main purpose of cold storage is to provide the storage conditions (temperature, relative humidity, ventilation) to the agricultural products for specific period. Usall et al., 2015 stated that storing the agricultural products at low temperature and high relative humidity in the cold storage is not an antifungal treatment; however, it can directly and indirectly influence the host and pathogen. According to Usall et al., 2015, the age delay is mainly due to: (i) loss of pathogens growth and corresponding enzymatic functions (straightforward influence) and (ii) metabolism reduction in the product (indirect influence). This fact keeps the resistance against infections particularly the caused by fungus. It is the most adopted physical method for the storage of agricultural products (Eckert and Sommer, 1967).

The cold storage building can be constructed using different materials like concrete, metal, wood etc. The walls, floors and ceilings should be well insulated. The insulation materials (e.g. polyurethane, cork etc.) can be used for the purpose. It is generally common that 75-80% of the total surface area of the building is used for products storage (Camelo, 2004). Rest of the surface area is usually occupied by walkways and aisles etc. The height of the product chamber depends on product type and stacking pattern. It varies from 3 meters to 6 meters for hand stacking and forklift stacking, respectively.

2.2.5 Controlled and Modified Atmosphere Storage

The separate storage chambers are provided in the conventional cold storage for controlled atmosphere provisions to the produce. In controlled atmosphere storage (CAS) the low temperature air with varying gas composition (O_2 and CO_2) is supplied to the fruits and vegetables. The low (1-3%) and high (4-5%) concentration levels of oxygen and carbon dioxide are usually provided around the products, respectively. The levels of these air gases mostly depend upon the product type and its variety too. CAS facilities are equipped with specialized controlled equipment like gas generators and carbon dioxide scrubbers in order to provide the different ratios of O_2 and CO_2 such as shown in Figure 2.1 (Hoehn et al., 2009). Rao, 2015 stated that the higher concentration of carbon dioxide is considered as antiseptic against the vital activity of phytopathogenic microorganism. The antiseptic properties of CO_2 appear at very high concentration in the air. However, the concentration of CO_2 higher than permissible level may result in browning of product tissue. It has been reported that CAS provides better storage conditions as compare to conventional cold storage particularly to apple fruit (Nilsson et al., 1956). It is due to reduction in postharvest decay (Nilsson et al., 1956), and considerably less existence of blue molds (Sams and Conway, 1987).

The modified atmosphere storage/packaging (MAP) is used to extend the storage life of the fresh and minimally processed agricultural products. In this technique the produce is packed in pouches/bags and containers made by polymeric material. The air composition inside the bag around the product is changed to another composition depending upon the product type, storage temperature and packaging material (Rao, 2015). The higher level of CO_2 and lower level of O_2 in the modified atmosphere package decreases the product respiration rate, consequently the product with extended shelf life can be transported easily for long distances (Barkai-Golan, 2001). In this regard, the export of sweet cherries (from turkey) through this storage technique has been reported in the literature (Barkai-Golan, 1990). Such storage technique preserve the fruit quality (color, firmness) and quantity (water loss, shriveling) (Karabulut et al., 2010). The products packed in modified atmosphere packaging are stored in conventional cold storage. Therefore it does not involve any extra cost for the construction of cold storage building (Rao, 2015).

The controlled and modified atmosphere storage techniques both can extend the particular products shelf life by delaying the postharvest decay. The modified atmosphere packaging can be used for the storage of variety of products. However, the main limiting factors in the adoptability of controlled atmosphere storage are the requirement of skilled labors/technicians and specialized facilities (equipments). Finally the cost benefit ratio plays the crucial role in deciding the storage technique (Usall et al., 2015). Such factors limits it applicability for few fruits like apples, kiwifruit, persimmon, and pomegranate etc. (Tonutti, 2015).

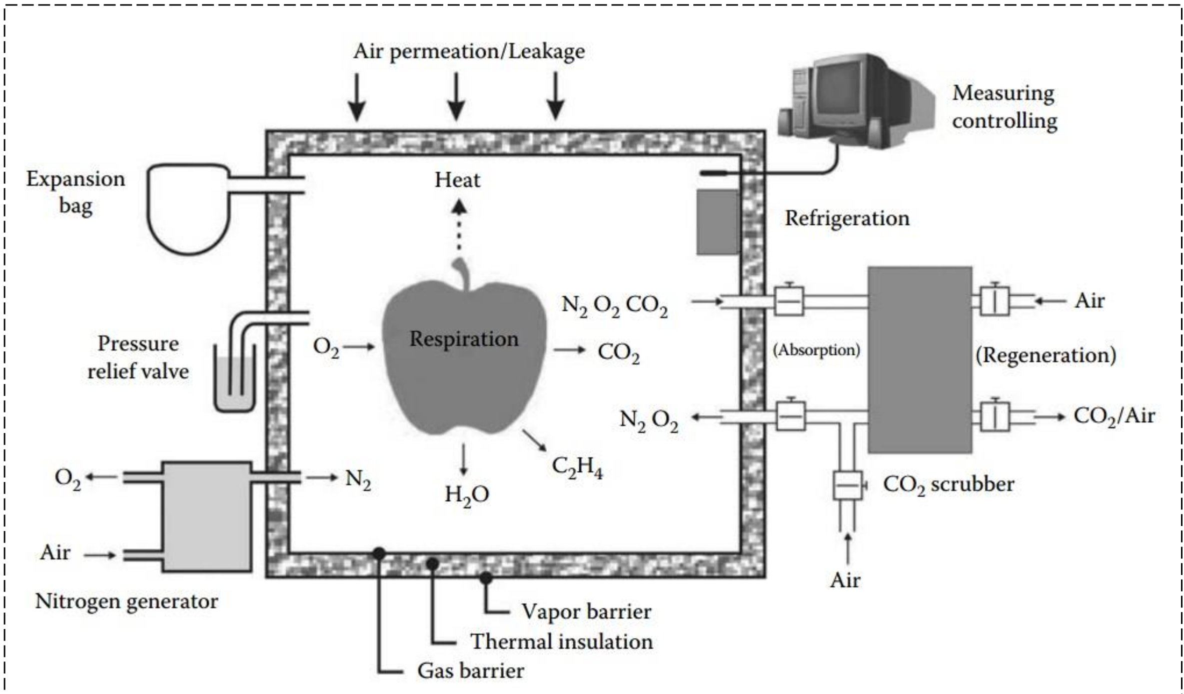

Figure 2.1 Self-explanatory explanation of controlled atmosphere storage representing the stored product respiration and controlling gadgets (Hoehn et al., 2009).

2.2.6 Hypobaric and Hyperbaric Storage

Hypobaric storage is also termed as low pressure or sub-atmospheric storage in the literature (Thompson, 2015). The hypobaric pressure (0-100 kPa absolute) is applied to the fruits and vegetable for short period of time (Goyette et al., 2007; Romanazzi et al., 2001). Such pressure application induces the natural resistance in agricultural products against the postharvest decay. Workman et al., (1957) reported that the respiration rate of tomatoes when stored at 20°C and 10.67 kPa was reduced as compared to tomatoes stored at 20°C under atmospheric pressure. He is probably the first who records the hypobaric conditions which were applied to the agricultural products (Thompson, 2015). The hypobaric chambers, used for products storage, always required proper ventilation control in order to maintain the permissible oxygen level. It is because of continues respiration and ethylene production by fruit and/or vegetables. Otherwise, rising levels of ethylene can rapidly deteriorate the quality of stored products. Therefore, the hypobaric chambers are equipped vacuum pump, which continuously evacuate the chamber air and replenish it with outside fresh air. Thompson, 2015 stated that in order to apply the hypobaric storage technology, the store structure must be constructed strong enough (e.g. with steel sheet) so that it can withstand the range of low pressure without any collapse. Furthermore, the air supply to the store must be saturated (relative humidity about 100%) otherwise application of low pressure may result in dehydration (rapid loss of moisture) of agricultural products.

Chen et al., 2013, experimentally investigated the effects of hypobaric storage on the quality of bamboo shoots. In this regard, the bamboo shoots were placed in the storage chamber of hypobaric storage system for 35 days at pressure and temperature of 50 kPa and 2(±1)°C, respectively. Chen et al., 2013, reported that the hypobaric pressure and low temperature storage of bamboo shoots not only maintain their quality till the end of storage period (35 days), nonetheless, some attributes were also improved by this storage technique.

In hyperbaric storage, unlike the hypobaric pressure, higher pressure than atmospheric pressure is applied to the stored agricultural products. The provision of pressure slightly higher than atmospheric pressure is not much explored yet, however, its higher range application (about 405 to 1215 MPa) has been broadly studied for agricultural products (Thompson, 2015). The hyperbaric pressure is also applied for short period of time to the agricultural products. Romanazzi et al., 2008 investigated that the postharvest storage/shelf life of sweet cherries and table grapes can be extended by exposing them to pressure of 152 kPa for 4 and 24 hours. The hyperbaric chambers are mainly made for medical and diving industries; however, they are also used for extending shelf life of agricultural products (Thompson, 2015).

2.2.7 Heat Treatment Options

The heat treatment is leading non-damaging physical treatment to avoid the postharvest decay in agricultural products due to insects, pests, and diseases (Ji et al., 2012; Lurie and Pedreschi, 2014). The heat treatments can extend the shelf life of agricultural products due to its effects on product ripening, senescence, and chilling injury (Sisquella et al., 2014). Lurie, 2008 stated that the heat treatments can inhibit the ethylene production in climacteric fruits which consequently delay the ripening, softening and discoloration. It can also reduce the softening rate and colour development in case of climacteric fruits (ripening does not depend on ethylene production). The yellowing of the green vegetables due to senescence (aging) can be reduced by heat treatments because of its effect on the physiology of product. Moreover, the heat treatments induce the resistance in subtropical agricultural products against the susceptibility to chilling injury. Thus the subtropical agricultural products after such treatments can be stored at low temperature, and transported through ships.

The heat treatments can be performed by hot air (vapor heat and forced air), hot water (hot water dipping and brushing), and dielectric heating (radio frequency and microwave) (Lurie, 2008; Usall et al., 2015). In case of hot air treatment, air at temperature and relative humidity higher than 30°C and 90% respectively, is usually provided to the fruits (Usall et al., 2015). The hot air treatment also reported in the literature for curing citrus fruit against rind/peel wound and green mold (Ben-Yehoshua and Porat, 2005; Hopkins, and Loucks, 1948). However, the hot water treatment is considered advantageous over hot air treatment due to the provision of efficient heat transfer with less time. It is also cheaper than other options of the heat treatments (Jacobi et al. 2001; Sivakumar, and Fallik, 2013). The water at temperature higher than 40°C is used in hot water treatment for certain period of time depending on several factors (Usall et al., 2015). Small scale hot water treatment plant was proposed under trade related technical assistance programme (TRTA II) in Pakistan for the treatment of fruits fly infestation of mangoes (www.trtapakistan.org).

The dielectric heating is also used for rapid heat transfer to the agricultural products, though the limited literature is available in this regard (Casals et al., 2010).

2.2.8 Ultraviolet-C Irradiation Treatments

The ultraviolet-C (UV-C) irradiation of wavelength ranging from 190 nm to 280 nm has shown promising results against the postharvest decay in agricultural products (Romanazzie et al., 2006; Usall et al., 2015). It is naturally present in the sunlight, but absorbed completely by ozone layer and earth atmosphere (Washington State University, 2015). It induces the resistances in the host against the pathogens. However, such induced resistance starts decreasing with the ripening of fruits (D'Hallewin et al., 1999). This treatment has been experimentally tested in order to inhibit the fungal growth and/or aging in different types of fruits and vegetables from last few decades (Usall et al., 2015). The application of low dose of UV-C irradiation has resulted in effective control over postharvest diseases in sweet potatoes (Stevens et al., 1990), peaches and apples (Lu et al., 1991), carrots (Mercier et al., 1993), grapefruits (Droby et al., 1993), strawberries (Nigro et al., 2000), and tomatoes (Liu et al., 2012) etc. as explained in the respective literature (Usall et al., 2015). Moreover, the delayed aging of peach fruit treaded with this treatment has also been reported (Yang et al., 2014).

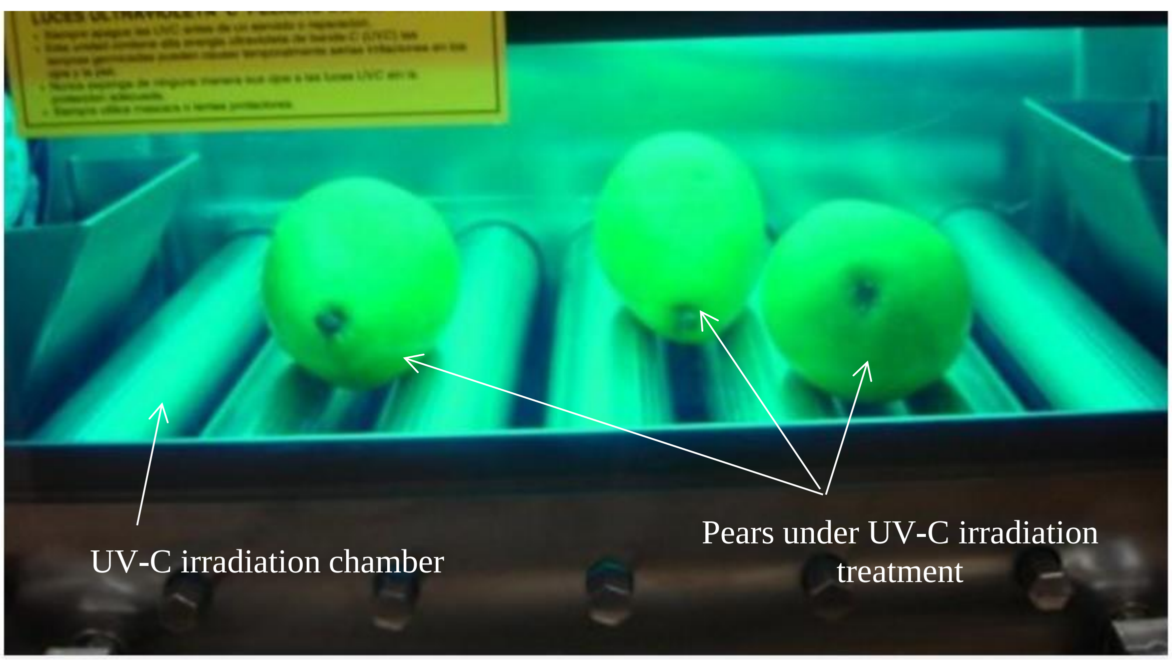

Figure 2.2 Pictorial view of UV-C irradiation treatment for smooth surfaced pear fruit (Washington State University, 2015).

Adhikari et al., 2015 experimentally investigated the effectiveness of UV-C irradiation treatment on the surface of various fresh fruits against the foodborne pathogens. In this regard, the smooth surface (apples, pears) and rough surface (strawberries, red raspberries, cantaloupe melons) fresh fruits were exposed to the UV-C irradiation for certain period of time (10 sec-840 sec) depending on different factors (e.g. fruit type, UV-C dose, etc.). The study concludes that UV-C irradiation treatment is highly effective against the food borne pathogens in case of smooth surface fruits. However, this treatment is found less effective for rough surface fruits because these fruits provide more hiding places to the pathogens. The pictorial view of UV-C irradiation treatment on pears is shown in Figure 2.2 (Washington State University, 2015).

2.3 Vapor Compression based Air-Conditioning

The importance and necessity of the conventional vapor compression refrigeration and/or air-conditioning for various applications/processes (at industrial, institution commercial and domestic etc. levels) is well known in the literature. The air-conditioning meanly deals with the provisions of conditioned air for human thermal comfort. However, its broad spectrum application also includes the conditioning of air for industrial processes, food processing, storage of agricultural products and materials (Khurmi, Gupta, 2008), and livestock thermal comfort. The conditioning of air means its temperature, relative humidity, purity and circulation.

The conventional vapor compression refrigeration (VCR) system mainly consists on an evaporator, condenser, compressor, expansion valve and fan (blower). In case of agricultural product storage, the evaporator coils and blowers are integrated into single unit and placed inside the storage room in order to provide the cold forced air circulation throughout the storage facility (Rao, 2015). The air becomes hot (due to heat of respiration) as it passes through the stored products. This air is then returns and passes over the evaporator to exchange the heat. The refrigerant (e.g. ammonia, freon etc.) inside the evaporator coils absorb this heat and cold air is again deliver to the stored produce. The phase changed refrigerant (liquid to vapor) then passes through the compressor. The compressor compresses the refrigerant (vapor), increased it pressure and temperature and transported it to the outside condenser in order to release the product heat to the atmosphere. The refrigerant (vapor to liquid) at high pressure enters to expansion valve. Now the refrigerant (vapor) is throttled down at low pressure to the evaporator in order to complete the refrigeration cycle. During this cyclic process the expansion valve controls the evaporation and flow of refrigerant (Camelo, 2004).

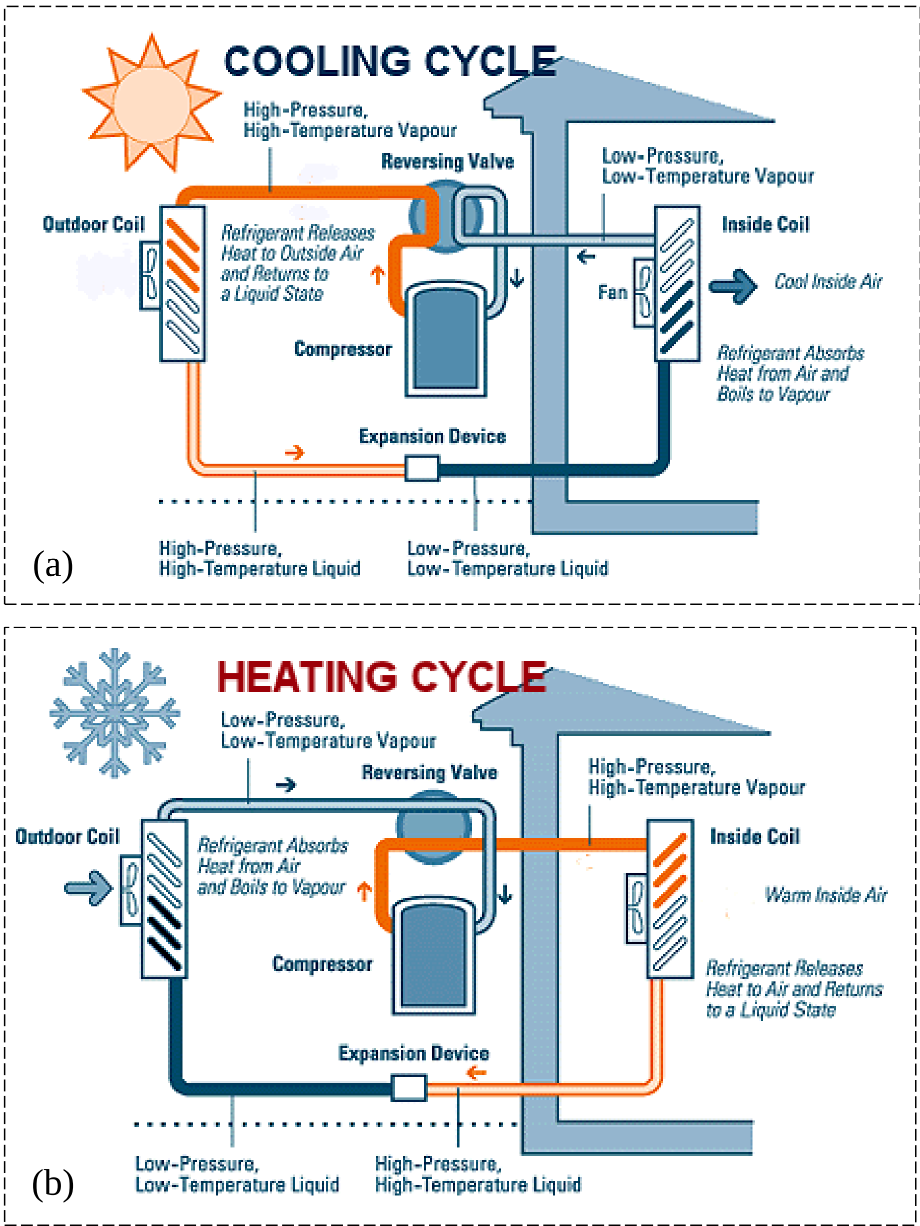

Figure 2.3 Schematics of heat pump cycles for: (a) cooling mode; and (b) heating mode (www.kendallcountyair.com).

The conventional vapor compression air-conditioning (VAC) systems might be considered in order to maintain the storage conditions for agricultural products. However, these systems consume primary energy and have certain thermodynamic limitations which include: limitation of fresh air, poor ventilation (excessive CO_2) etc. Moreover, such systems also cannot provide the optimal storage conditions (temperature and relative humidity) particularly to the tropical agricultural products and dry fruits (Mahmood et al., 2016a, Mahmood et al., 2016b). In addition the conventional vapor compression air-conditioning cannot control the temperature and relative humidity distinctly as compare to desiccant air-conditioning (Sultan et al., 2015; Sultan et al., 2016).

As far the as the thermal comfort of human and livestock is concerned, it is uneconomical to have two separate systems for cooling (air conditioners) and heating (heat pumps). The heat pumps and air conditioner have same components; therefore one system can be used for both cooling (in summer) and heating (in winter) (Cengel and Boles, 2006). The heating and cooling modes of a heat pump system are shown in self-explanatory Figure 2.3(a) and 2.3(b), respectively (www.kendallcountyair.com). Cengal and Boles, 2006, stated that heat pumps are more economical/competitive for areas of high cooling and less heating loads and vice versa.

2.4 Evaporative Cooling based Air-Conditioning

Evaporative cooling is low cost environmental friendly air-conditioning system (Jha and Chopra, 2006; Zahra and John, 1996), which works on the basis of induced processes of heat and mass transfer (Camargo, 2007). In this system water and air are used as working fluids (Camargo, 2007). The details about advances in evaporative cooling systems are explained in chapter 3. Evaporative cooling systems have been practically studied for extending the shelf/storage life of fruits and vegetables (Dadhich et al., 2008; Jha and Chopra, 2008; Odesola and Onyebuchi, 2009; Zhao et al., 2008). Moreover, it has also been proved successful for on-farm short term storage of agricultural products in hot and dry areas/locations (Jha and Chopra, 2008). Rao, 2015, explained two types of evaporative cooling systems for the storage of agricultural products. The main difference between the Fan & Pad type and Mist/Spray type evaporative cooling systems is the provision of water supply to the incoming hot air. In Fan and Pad type evaporative cooling system water is supplied to the pad (made of plastic coir/ wood fibers) and the hot ambient air is passed through wetted pad. On the other hand, water is sprinkled on the hot incoming air in case of Mist/Spray type evaporative cooling system. In such systems water evaporates by absorbing heat of vaporization from incoming hot air, therefore the amount of heat removed from the air remains equal to the amount of heat absorbed by evaporated water (ASHRAE, 2007).

Rao, 2015, proposed the commercial type evaporative cooler for storage of agricultural products that works on the principle of Fan and Pad type evaporative cooling system as explained above. In order to achieve the efficient and uniform cooling, the containers of agricultural products in cold store/room must be stacked in a way that supply air can move all around the product's containers at minimum velocity.

The different types of evaporative cooling systems depending on various types of construction materials (bricks, sands, ceramics, woods, metals, fibers, zeolites etc.) and energy sources (solar, electric etc.) have been experimentally studied for the storage of agricultural products in literature (Habibunnisa et al., 1988; Roy, 1984; Roy and Khurdiya, 1982). In this regard, Chouksey, 1985, has studied the design aspects and performance of a 20 tonne capacity solar cum wind aspirator evaporative cooling system for the storage of potatoes and other semi perishable agricultural products. This system maintained the storage/supply air temperature between 21-25°C (80-90% RH) when ambient air temperature was 40-42°C (30-35% RH) with the ventilation rate of $24m^3$/min. Moreover, the potatoes stored in evaporative cooling storage are more suitable for making potato chips due to less physiological losses, and lower reduction in sugar contents as compare to refrigerated cold storage (Lal Basediya et al., 2013). However, the requirement of continuous water supply, big system size, and limited cooling performance in high humidity conditions are the major drawbacks of the evaporative cooling systems.

2.5 Desiccant based Air-Conditioning

The desiccant air-conditioning (DAC) deals the latent and sensible load of air-conditioning distinctly. Such distinct control of sensible and latent load of AC makes it more versatile and flexible for wide range of applications. Desiccant materials (e.g. silica gel, activated alumina, molecular sieve, composite materials, liquid and polymer desiccants, bio-desiccant and activated carbons, bentonite clay etc.) as hygroscopic in nature possess water loving ability (Sultan et al., 2015). The main driving force for moisture adsorption by desiccant material is the difference in vapor pressure between air and desiccant surface.

The water loving ability of desiccants makes them able to be used in applications where the control over relative humidity is crucial. The general applications of the DAC include buildings (Enteria et al., 2009; Enteria et al., 2010), supermarkets, schools, ice arenas, cold warehouse, hotels, theaters, hospitals (Dabrowski, 1998), automobiles (Nagaya et al., 2006), wet markets (Lee and Lee, 2013), marine ships (Guojie et al., 2012; Zhu and Chen, 2014), museums (Ascione et al., 2009; Ascione et al., 2013) etc. However, its agricultural based applications may include greenhouses (Longo and Gasparella, 2015; Sultan et al. 2014, Sultan et al., 2016), products storage and preservation (Mahmood et al., 2015, Mahmood et al., 2016, Sultan et al. 2016), and Livestock cooling etc. In DAC, latent load is achieved through desiccant dehumidification while the sensible load is achieved by sensible heat exchanging through heat exchanger and/or evaporative cooler.

The thermally driven DAC systems can be broadly classified into solid DAC system, liquid DAC system and hybrid DAC system. However, the selection and performance of typical solid DAC system depends on the ambient condition (e.g. temperature, relative humidity, sensible to latent heat ratio), demand conditions (e.g. temperature, relative humidity, flow/ventilation rate) and the type of available regeneration heat source (e.g. electric, gas, solar, waste heat etc.) (Sultan et al., 2015).

The working principle of typical solid DAC system can be explained by Figure 2.4 (Miyazaki, et. al., 2010). The outdoor ambient/process air when pass through desiccant rotor becomes dehumidified and its temperature increases due to the heat of adsorption. The dehumidified air then enters to the heat recovery devices (e.g. heat exchanger) for heat exchanging to the regeneration air. This warm dehumidified process air can be supplied for particular applications. However, in most of cases cooling of the process air is required in order to meet the desired levels of temperature and relative humidity. Therefore, cooling of the process air is accomplished through cooling devices (direct/indirect evaporative cooler) for supplying cool air. Moreover, the regeneration of the desiccant dehumidifier is always required for its continuous cyclic operation. It can be done by supplying hot regeneration air to the desiccant dehumidifier. The heat energy can be supplied to regeneration air through renewable/non-conventional and conventional energy sources (Enteria and Mizutani, 2011).

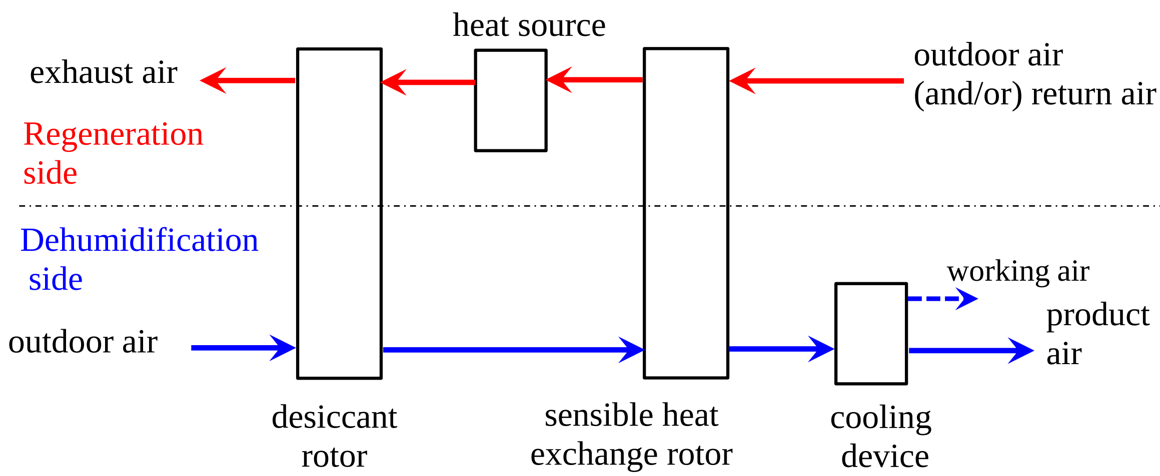

Figure 2.4 Schematic of typical desiccant air conditioning system (reproduced from Miyazaki, et. al., 2010).

2.5.1 Single-Stage Desiccant Air-Conditioning

A typical DAC system is mainly consists of desiccant, cooling and regeneration units. In this regard, La et al., 2010, reviewed the different type of single-stage rotary desiccant air-conditioning systems, presented their system configurations and psychrometric representation. These systems include, Pennington/ventilation cycle, Modified ventilation cycle, Recirculation cycle, Dunkle cycle, SENS cycle, REVERS cycle, DINC cycle and Staged regeneration cycle. It is important to mention that the most of the existing DAC cycles are mainly originated from three basic cycle configurations: Ventilation, Recirculation, and Dunkle cycles (La et al. 2010). Therefore, only the basic cycles are briefly discussed here; however the details about all listed cycles can be found from cited reference. Pennington, 1955, patented the first rotary DAC cycle. The working principle of typical Pennington/Ventilation cycle can be explained as: the ambient air when passes through the desiccant wheel (DW) it becomes dehumidified. The temperature of dehumidified air increases due to heat of adsorption. The dehumidified air is then sensibly cooled through heat exchanger. The further cooling is achieved by direct evaporative cooler (DEC) in order to supply the air to conditioned space. On the regeneration side, room return air is passed through the DEC. The DEC cooled and humidified the regeneration air. The regeneration air is then passed through the heat exchanger for sensible precooling of process air and heating itself. The regeneration air is further heated by supplying heat through heat source. This hot regeneration air is used to regenerate the desiccant wheel and finally exhausted to the environment. The recirculation cycle as a variation of ventilation cycle is developed for increasing the cooling capacity. In recirculation cycle the room returns air is used as process air and ambient air is used as regeneration air (La et al. 2010). The rest of system operation is similar to ventilation cycle. The thermal COP of recirculation cycle is not usually more than 0.8 (Waugaman et al., 1993).

The ventilation and recirculation cycles are suitable for different particular applications. The ventilation cycle is suitable for applications/spaces which required more fresh air whereas recirculation cycle enables less supply of fresh air. The Dunkle cycle combines the merits of both the ventilation cycle and regeneration cycle. However, its design configuration requires two heat exchangers (La et al. 2010). The applicability of the Dunkle cycle is also restricted due to supply of less fresh air like recirculation cycle (La et al., 2010).

2.5.2 Multi-Stage Desiccant Air-Conditioning

In multi-stage desiccant air-conditioning system lower regeneration is required as compare to single stage desiccant air-conditioning system (Sultan et al., 2015). The multi-stage dehumidification promotes the isothermal dehumidification which ultimately results in the requirement of less regeneration temperature. Lower regeneration temperature benefitted to economic system design and operation. It promotes the utilization of low grade waste heat and solar energy etc. (Huan, and Jianlei, 1999). A comparison between single-stage and multi-stage dehumidification is shown in Figure 2.5. The regeneration temperature required for multi-stage process air dehumidification (1-2a-3a-4a-5a) is T_1 and T_2, whereas it is T_3 in case of single-stage process air dehumidification (1-6a-7a). Therefore, regeneration temperature (T_1, T_2) of multi-stage dehumidification is much lower than regeneration temperature (T_3) of single-stage dehumidification as shown in Figure 2.5.

Two-stage desiccant unit was developed for fast-food restaurants by Gershon Meckler Associates, P.C. (Mei et al., 1992). The thermal COP achieved by this system was 0.89. However, the annual energy cost and electricity use by the system was reported as 40% and 60% less, respectively, when compare with vapor compression air-conditioning.

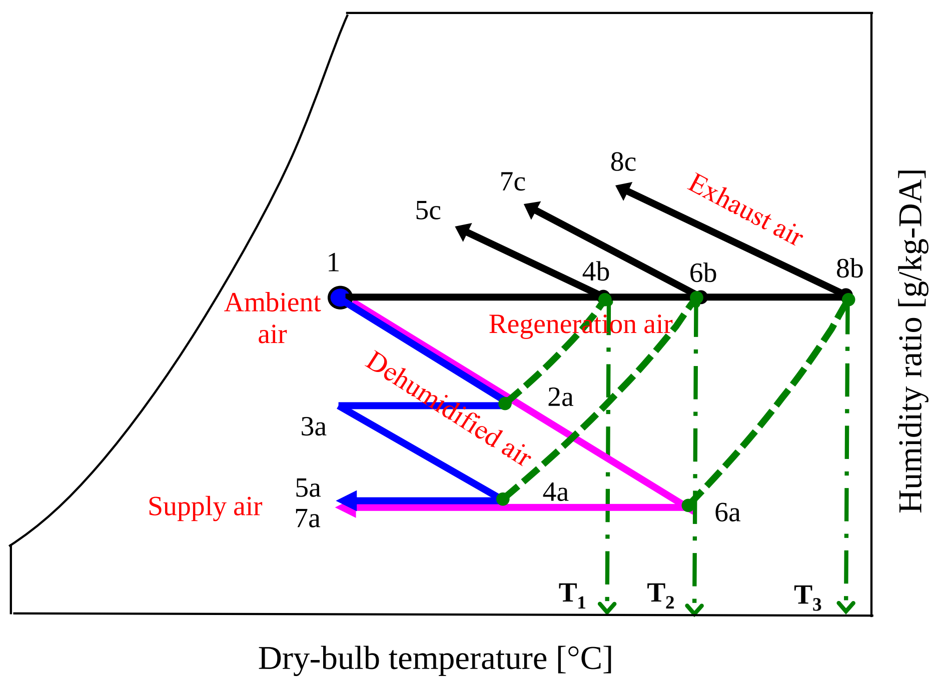

Figure 2.5 Psychrometric comparison between single-stage & multi-stage desiccant dehumidification and air conditioning.

Two types of two-stage rotary desiccant cooling systems have been experimentally tested by authors (Ge et al., 2008) and (Ge et al., 2009). These systems named as two-stage rotary desiccant cooling system (TSDCS) and one-rotor two-stage rotary desiccant cooling system (OTSDCS). The main difference between theses system was in the divisions of cross section of desiccant wheel. In TSDCS the desiccant wheel was divided into two parts: one for process and other for regeneration air, while in OTSDCS desiccant wheel was divided in four parts: two for process and two for regeneration air (Ge et al., 2008). Silica-gel lithium chloride based composite was used as desiccant material. It was concluded in these studies that both systems can achieve thermal COP greater than 1.0 and can be driven by heat sources of above 50°C. Moreover, design configuration of one-rotor two-stage rotary cooling desiccant reduces the system size about half of two-stage rotary desiccant cooling (Ge et al., 2008; Ge et al., 2009; La et al., 2010).

2.5.3 Solar Energy Operated Desiccant Air-Conditioning

A solar powered desiccant air-conditioning system is installed in the building of Chamber of Trade and Commerce, Freiburg/Germany for the air-conditioning of seminar room and cafeteria (Henning, 2007). System consists of desiccant/sorption wheel (silica-gel), heat recovery wheel (heat exchanger, solar collector and humidifiers etc.) as shown in Figure 2.6 (Henning, 2007). Solar air collectors of 100 m^2 were used for the regeneration of desiccant wheel. Desiccant wheel handles the air flow rate of 10,200 m^3/h. The installed system maintains the indoor conditions properly, though its COP was less than expected design value (Henning, 2007).

Enteria et al., 2009, developed a solar thermal and electric energy operated desiccant air-conditioning system at Tohoku University, Japan. The experimental system consists of two subsystems: thermal energy subsystem and desiccant cooling subsystem as shown in Figure 2.7 (Enteria et al., 2009). The objective of thermal energy subsystem is to collect the thermal energy for desiccant regeneration either from solar radiation or electric heater. The thermal energy subsystem was consists of five collector panels, an electric heater for night-time thermal energy storage and for lower solar radiation period, and a thermal storage tank with water as working fluid. In order to avoid the freezing of water in solar collector during winter operation, an automatic compressed air mechanism is attached to drain the water as soon the solar collector surface temperature approaches to water freezing point. The desiccant cooling subsystem was mainly consists of a silica-gel desiccant wheel, two cross-flow heat exchangers, and an evaporative cooler. The system design allows the thermal energy storage during night-time in order to use it for early daytime operation. The reason to store the energy at nighttime is to make the system cost effective by using relatively lower price of electricity. The system was experimentally tested for standard: outdoor air conditions (T: 30ºC, RH: 60%) and indoor return air conditions (T: 26ºC, RH: 55%) in Japan. The outdoor and return indoor air flow rates were maintained as 200 m^3/h and 100 m^3/h, respectively. The calculated total COP of the system was quite low (0.25) due to utilization of energy by auxiliary devices (i.e. control systems, instrumentations, etc.). However, system can work in both day and night time (Enteria et al., 2009). The study was further extended by the same authors (Enteria et al., 2010) in order to check the improvement in system performance and COP by increasing the regeneration temperature. The desiccant dehumidification performance increases by increasing the regeneration temperature from 60°C to 75°C but COP decreases accordingly from 0.44 to 0.35. Moreover, the supply air temperature also increases by increasing the regeneration temperature. Therefore, it can be concluded that thermal energy supplied to the desiccant air-conditioning system is not proportional the produced cooling. It was found in further analyses of the system that maximum energy and exergy loss occur in solar collectors (Enteria et al., 2013).

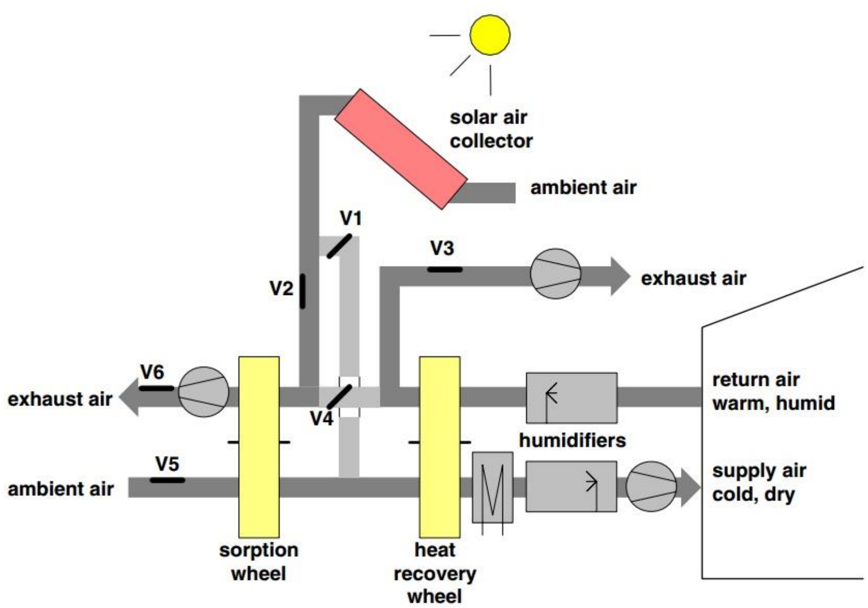

Figure 2.6 Schematic of solar powered desiccant air-conditioning system installed at Freiburg, Germany (Henning, 2007).

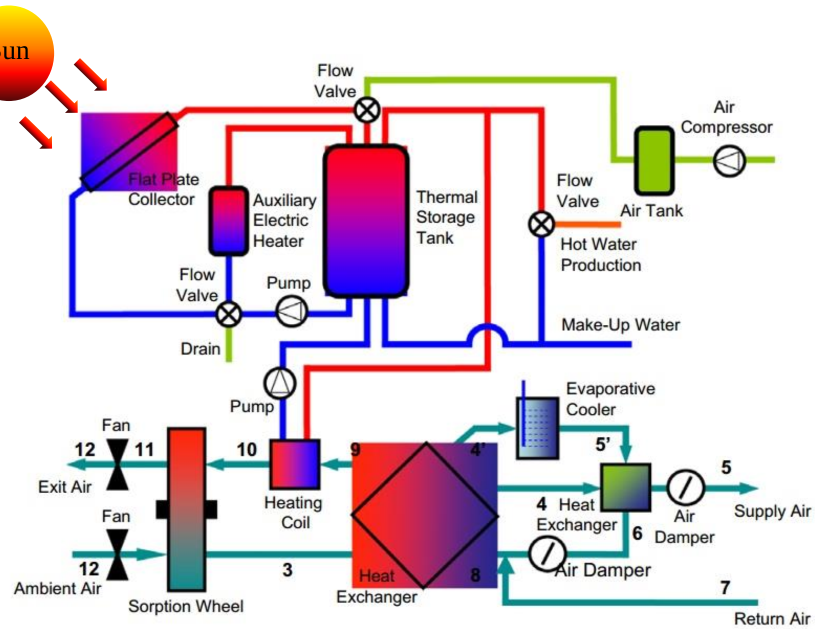

Figure 2.7 Schematic of solar thermal and electric energy operated desiccant air-conditioning system installed at Tohoku University, Japan (Enteria et al., 2009).

2.6 Hybrid System

Hybrid system combines the features of coupled air-conditioning/cooling systems and mitigates the problems associated with their air-conditioning/cooling processes (Kojok et al., 2016). The properly selected hybrid systems for particular applications may reduce the energy consumption and improve the system COP.

2.6.1 Evaporative Cooling based Hybrid System

Cengal and Boles, 1998, stated that one degree centigrade (1°C) increase in condenser temperature of conventional vapor compression air-conditioning (VAC) system results in 2-4% decrease in system performance. In this regard, evaporative coolers particularly in hot and dry climatic conditions can be used to pre-cool the ambient air before it passes over the condenser coil of VAC system.

Delfani et al., 2010, constructed a hybrid system consists of indirect evaporative cooler (IEC), conventional vapor compression air conditioning system and other necessary instrumentation. The purpose of IEC was to supply the pre-cool air to the VAC unit. The system performance was experimentally investigated for different climatic conditions of Iran. It was concluded that the coupling of IEC with VAC unit results in saving of 55% electrical energy consumption.

Hajidavalloo and Eghtedari, 2010, conducted an experimental study by connecting the direct evaporative cooler (DEC) to the air-cooled condenser of existing split air-conditioner. The performance of retrofitted DEC condenser and air-cooled condenser of existing air-conditioner are compared. It was concluded that system performance can be improved about 50%, and power consumption can be reduced about 20%, when employing retrofitted DEC condenser as compared to air-cooled condenser.

2.6.2 Desiccant based Hybrid System

The concept of hybrid desiccant air-conditioning using DAC system with assistance of VAC got so much attention because of its high performance, feasibility, and reliability as compared to standalone DAC system. Usually when the standalone DAC system could not achieve the demand conditions fully, it is suggested to switch on hybrid system. The hybrid DAC system can work effectively in almost all climatic conditions without any interruption (day and night) for different applications.

Burns et al., 1980, studied the three different configurations of hybrid desiccant air-conditioning system for high latent load applications (i.e. supermarkets). These are one of the earliest major studies done for solid desiccant based hybrid air-conditioning system. In these studies comparison was made between hybrid DAC and VAC systems. The hybrid DAC systems, for specified design conditions (ambient and indoor air conditions), resulted in total air-conditioning energy savings of about 57-66% (Burns et al., 1980).

Baniyounes et al., 2013, installed a single-stage solar hybrid desiccant air-conditioning system at Central Queensland University, Australia. The system mainly consists of three units: solar thermal unit, DAC and VAC units as shown in Figure 2.8 (Baniyounes et al., 2013). The subcomponents of the solar thermal unit incudes flat plate solar collectors (10 m^2), hot water buffer tank, two backup heaters (4.8 kW and 9 kW) and valves etc. However, the DAC and VAC units had typical subcomponents. The system performance was measured both by experimentation and simulation. The experimentally measured annual energy savings achieved by hybrid DAC system was 18% as compared to standalone conventional VAC system. However, its simulated value was found as 23%. The COP of the system was measured as high as 0.83 (simulated: 0.728) in the month of December. The detail month wise trends of COP and energy savings can be seen from cited reference. In another study, El-Agouz and Kabeel, 2014, simulated that the COP of single-stage hybrid DAC system can be achieved maximum up to 1.03.

La et al., 2011, configured and experimentally investigated the two-stage solar hybrid DAC system in Jiangsu, China. The performance of the system was theoretically analyzed. The detail system components and schematics can be found from cited reference. It is determined that system can achieve the designed cooling load (i.e. about 10 kW) under particular local conditions with thermal and electric COP of 1.0 and 10, respectively. Moreover, the system is feasible for its successful operation in wide range of climatic conditions like extremely humid (e.g. Hong Kong), humid (e.g. Shanghai), and temperate (Beijing) climatic conditions

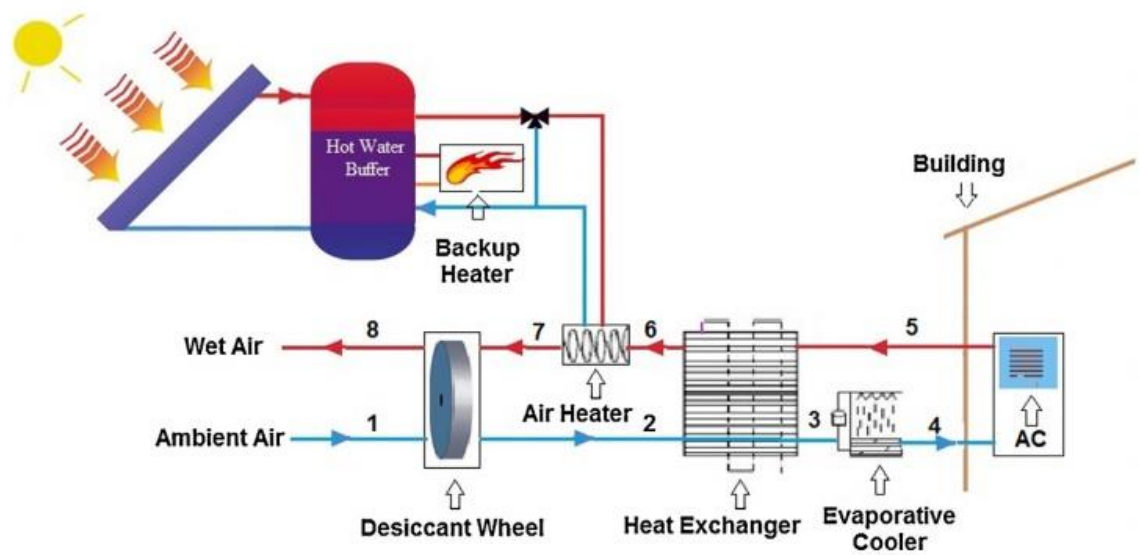

Figure 2.8 Schematics of single-stage solar hybrid desiccant air-conditioning system installed at Central Queensland University, Australia (Baniyounes et al., 2013).

2.7 Nomenclature

AC	air-conditioning
CAS	controlled atmosphere storage
COP	coefficient of performance
DAC	desiccant air-conditioning
DEC	direct evaporative cooler
DW	desiccant wheel
IEC	indirect evaporative cooler
MAP	modified atmosphere packaging
UV-C	Ultraviolet-C
VAC	vapor compression air-conditioning

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CHAPTER 3

ADVANCES IN EVAPORATIVE COOLING SYSTEMS

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ADVANCES IN EVAPORATIVE COOLING SYSTEMS

This chapter briefly explains the principle and features of advanced dew-point Maisotsenko Cycle evaporative cooling (MEC) in comparison with conventional evaporative cooling. The potential applicability of low cost environmental friendly evaporative cooling technologies like conventional direct and indirect evaporative cooling and M-Cycle evaporative cooling for agricultural products storage is discussed. Various modifications in MEC design are discussed in order to investigate its applicability in humid regions.

3.1 Introduction

The most of the cold storages available today are equipped with conventional vapor compression refrigeration and/or air-conditioning. Such systems are not only degrading the environment and consuming high primary energy but also difficult to provide the optimal storage conditions (temperature and relative humidity) particularly to the tropical agricultural products (Mahmood et al., 2015; Mahmood et al., 2016a, Ndukwu and Manuwa, 2015; Olosunde et al., 2015). It is because of diversified nature of agricultural products, chilling injury, and complex mechanism of transpiration, respiration and fermentation etc. In this perspective, low cost environmental friendly evaporative cooling technologies like conventional direct and indirect evaporative cooling and M-Cycle evaporative cooling have shown potential to provide storage conditions for particular agricultural products (Mahmood et al., 2016b). Therefore, the principle and features of conventional direct and indirect evaporative cooling are shortly described. On the other hand, advance M-Cycle evaporative cooling due to its potential of dew point cooling (theoretically) is discussed in detail.

3.2 Wet-Bulb Evaporative Cooling

3.2.1 Direct Evaporative Cooling

The conventional direct evaporative cooling (DEC) works on the principle of isenthalpic cooling. The principle operation of DEC is shown in Figure 3.1(a). It can be seen that when ambient air (1) passes through the wet channel it becomes cool (2). The cooling process in the wet channel is considered as adiabatic. During this process sensible heat of the ambient air is converted in to latent heat. The cooling performance of the DEC is obtained by commonly known wet-bulb effectiveness (ε_{wb}) as given in Eq. (3.1) (Anisimov et al., 2014a; Bruno, 2011).

$$\varepsilon_{wb} = \frac{T_1 - T_2}{T_1 - T_{1,wb}} \tag{3.1}$$

The parameters of Eq. (3.1) are similar as explained on Figure 3.1. The wet-bulb effectiveness achieved by DEC is usually ranges between 0.75-0.95 (Sultan, 2015b). However, it can be ideally equal to 1 (Mahmood et al., 2016b).

3.2.2 Indirect Evaporative Cooling

Conventional indirect evaporative cooling (IEC) works on the principle of sensible cooling that means the humidity ratio of ambient and supply air remained same. The working principle of IEC is shown in Figure 3.1(b). It consists of two channels namely dry-channel and wet-channel. The wet-channel remains wetted with moisture/water. The product/process air flows in the dry-channel, and working air flows oppositely in the wet-channel. In this regards, it can be seen (from Figure 3.1b) that when ambient air (1) is flow through the dry-channel it becomes sensibly cooled (2) at constant humidity ratio by transferring its heat to the working air flowing in the wet-channel. The cooling performance of the IEC is also obtained by commonly known wet-bulb effectiveness (Anisimov et al., 2014a; Bruno, 2011) as given in Eq. (3.1). The wet-bulb effectiveness achieved by IEC is usually ranges between 0.50-0.65 (Sultan, 2015b). However, it can be ideally equal to 1 (Mahmood et al., 2016b).

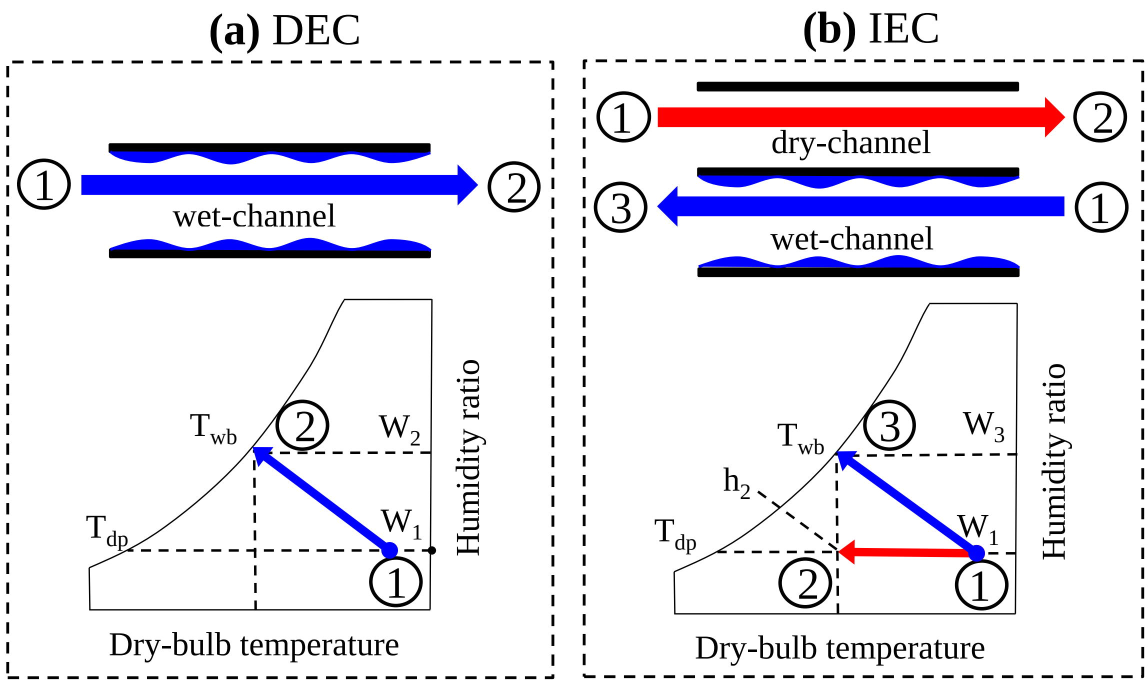

Figure 3.1 Principle operation of conventional evaporative cooling techniques showing their cooling limit for: (a) DEC, and (b) IEC.

3.3 Dew-Point Evaporative Cooling

The Maisotsenko Cycle (M-Cycle) is a thermodynamic process which captures energy from the air by utilizing the psychrometric renewable energy available from the latent heat of water evaporating into the air (Anisimov et al., 2014b; Anisimov and Pandelidis, 2014; Bruno, 2011; Cui et al., 2015). It combines thermodynamic processes of heat transfer and evaporative cooling to facilitate product temperature to reach the dew-point temperature of the ambient air. In other words, it is also an advance indirect evaporative cooling (IEC) by which the air can be cooled to the dew-point temperature rather than wet-bulb temperature (Miyazaki et al., 2011a). However, apart from the cooled air the M-Cycle produces saturated hot air which is required by many applications. Thus, the M-Cycle is a heat recovery process (Alsharif et al., 2011; Buyadgie et al., 2015; Maisotsenko et al., 2004; Maisotsenko et al., 2005b; Maisotsenko et al., 2006) by which the system efficiency can be increased tremendously for various applications (see heading 3.4 and appendices).

The basic principle and features of the M-Cycle can be explained from Figure 3.2(a) and 3.2(b) representing the old and modified M-Cycle, respectively. The psychrometric representation of old and modified M-Cycle is shown in Figure 3.2(c). It consists of two kinds of primary channels named as wet and dry channels. The product as well as working channels are devoted for air flow in case of old M-Cycle (Figure 3.2(a)), whereas modified M-Cycle (Figure 3.2(b)) gives the freedom to recover the heat from any fluid/gas by using an additional dry-channel. For cooling and AC applications, the product/process air flows into the dry-channel whereas working air flows into the wet-channel. For example, ambient air (1) is flowed into the dry-channel where it is sensibly cooled at constant humidity to cycle point (2) by transferring the heat to the wet-channel. The operational principle of M-Cycle is based on diverting the cooled air (2) to the wet-channel in order to use as working air. It results in subsequently decrement of effective dry-bulb (1 ➔ 2a; 2b; 2c; 2) and wet-bulb (1w ➔ 2a,w; 2b,w; 2c,w; 2dp) temperatures of the working air in the wet-channel as shown in Figure 3.2(d). Sequential decrement of dry-bulb temperature in the wet-channel brings the effective wet-bulb temperature to be ideally equal to the dew-point temperature. Hence for an ideal heat transfer surface, the product air can be sensible cooled to the dew-point temperature of the ambient air. Moreover, saturated hot air (3) is rejected from the wet-channel equivalent to the evaporated water and recovered heat. Depending upon the nature of M-Cycle application, the product and working channels can be interchanged in order to utilize the saturated hot air (see appendix D).

Using the air inlet and outlet conditions, the M-Cycle performance is usually estimated by dew-point effectiveness. It is the ratio of inlet and outlet dry-bulb temperature difference to the temperature difference between inlet dry-bulb and the corresponding dew-point temperature (Anisimov et al., 2015b; Hasan, 2012; Riangvilaikul and Kumar et al., 2010a). The dew-point effectiveness (ε_{dp}) is given by Eq. (3.2) as follows

$$\varepsilon_{dp} = \frac{T_1 - T_2}{T_1 - T_{1,dp}} \tag{3.2}$$

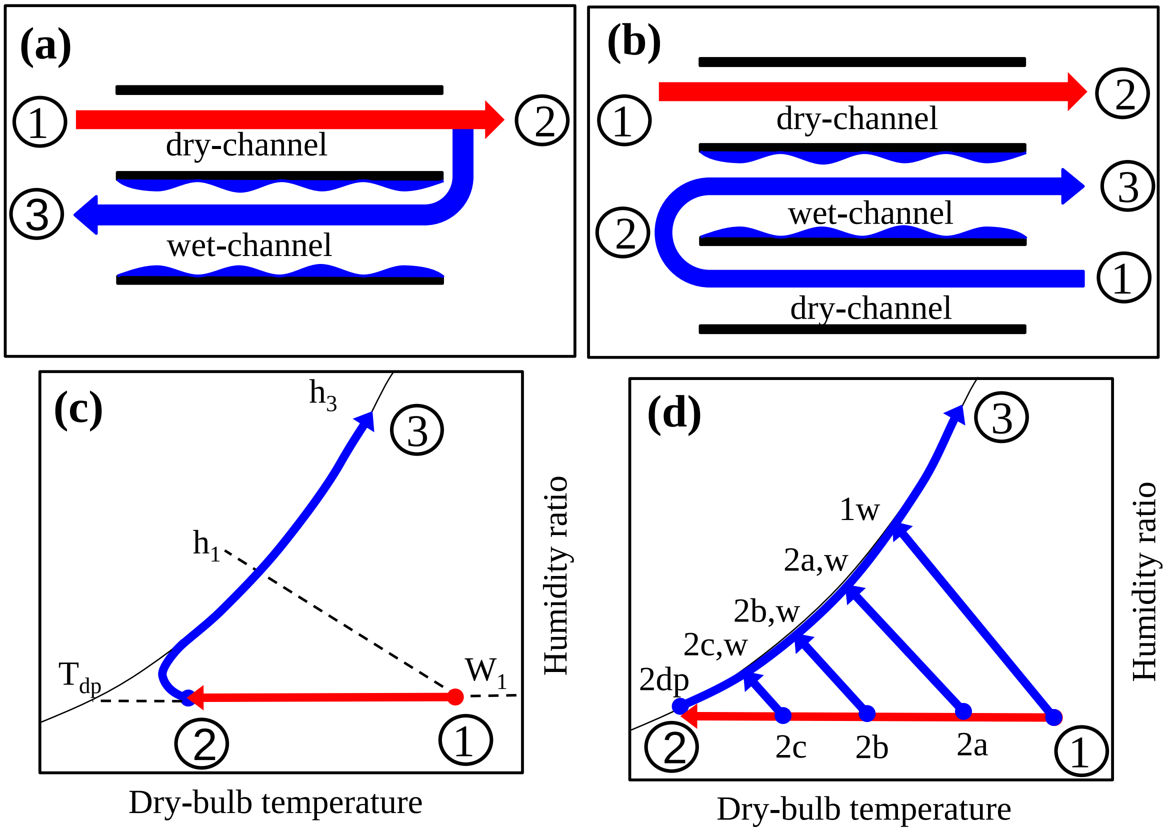

Figure 3.2 Schematic diagram of Maisotsenko Cycle for: (a) old M-Cycle, (b) modified M-Cycle, (c) psychrometric representation, and (d) sequential temperature decrement in wet-channel.

The parameters of Eq. (3.2) are similar as explained on Figure 3.2. The conventional DEC (i.e. isenthalpic cooling) and IEC (i.e. sensible cooling) processes are presented on Figure 3.1(a) and 3.1(b), respectively in order to compare with M-Cycle versatility. It can be noticed that the theoretical limit of DEC and IEC processes is wet-bulb temperature whereas it is dew-point temperature in case of M-Cycle (Anisimov et al., 2014b; Anisimov et al., 2014a; Cui et al., 2014a) as elaborated in Figure 3.2(c). The wet-bulb effectiveness achieved by the M-Cycle is up to 1.80 (Anisimov et al., 2014a; Anisimov et al., 2014b; Anisimov et al., 2015b; Anisimov and Pandelidis, 2014; Gillan, 2008; Zhan et al., 2011a; Zhan et al., 2011b; Zhao et al., 2008) whereas it can be ideally equal to 1.0 for DEC/IEC techniques. Unlike conventional evaporative cooling, the M-Cycle efficiency increases with the increase in temperature at $X_{inlet} \leq 11.2$ g/kg-DA

(Anisimov et al., 2014a; Khalatov et al., 2011), because of its dependency on dew-point temperature rather than wet-bulb temperature. In other words, at certain absolute humidity the wet-bulb temperature increases with the increase in dry-bulb temperature whereas the dew-point temperature remains constant.

The versatile applicability of the M-Cycle has been studied in the literature for various energy recovery applications. It can be categorized into three main sections based HVAC, cooling (Appendix C) and gas turbine power cycles (Appendix D). The coming headings will discuss M-Cycle HVAC. A brief overview of M-Cycle applications is presented by Figure 3.3 which shows its significance in thermal engineering sciences. Moreover, broad spectrum applications of M-Cycle can be found from Mahmood et al., 2016b.

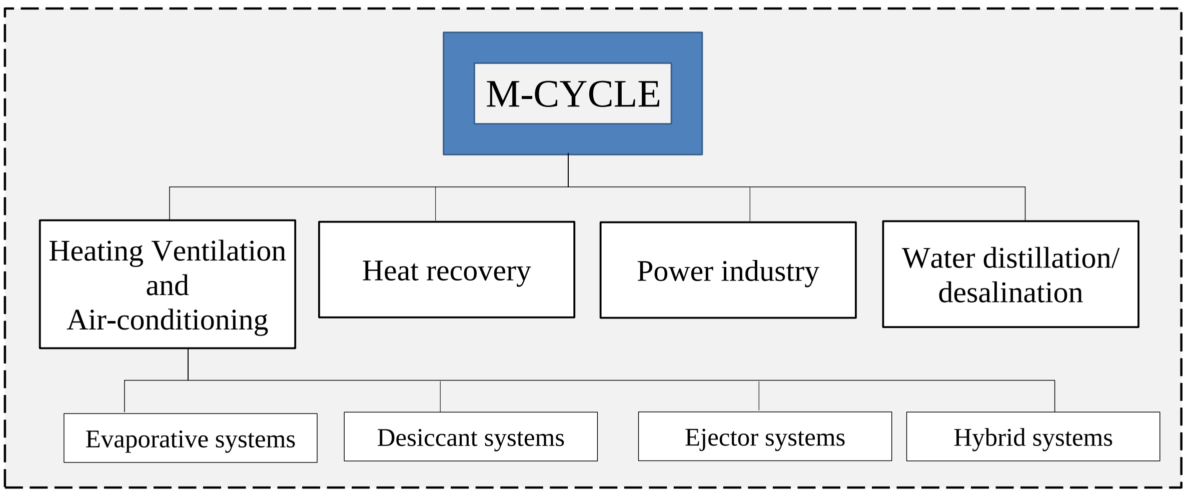

Figure 3.3 Overview of the Maisotsenko Cycle applications.

3.4 Maisotsenko Cycle based HVAC Systems

The M-Cycle based HVAC technologies are getting much attention by the researchers because of the dew-point evaporative cooling potential. Many HVAC system designs are investigated in the literature in order to achieve the sensible and latent load of AC for various applications which include: conventional AC for residential and office buildings (Caliskan et al., 2011a; Caliskan et al., 2012b; Chua et al., 2013; Cui et al., 2014b; Pandelidis et al., 2015a; Pandelidis and Anisimov, 2015b; Riangvilaikul and Kumar

2010b; Rogdakis et al., 2014; Zhan et al., 2011a); chilled ceiling and/or displacement ventilation (Bruno, 2011; Itani et al., 2015); data center cooling (Weerts, 2011; Weerts, 2012; Weerts et al., 2012; FEMP report, 2012); gas turbine inlet air cooling (Saghafifar and Gadalla, 2015); greenhouse AC (Sultan, 2015b); electronic cooling (Maisotsenko and Reyzin, 2005); automobile batteries cooling (Khazhmuradov et al., 2011); manufacturing and storage processes (Miyazaki et al., 2010b); frost formation for energy recovery (Anisimov et al., 2015a) etc. The additional potential applications are highlighted in Figure 3.3. The coming sections will discuss the M-Cycle HVAC system designs in detail for many applications.

3.4.1 Standalone Maisotsenko Cycle Air-Conditioning

The standalone M-Cycle AC (MAC) (see Figure 3.2) unit provides two types of air flows simultaneously i.e. (1) sensibly cooled air and (2) saturated hot air, which can be used for cooling in summer and humidification cum heating in winter seasons (Anisimov et al., 2014a; Anisimov et al., 2014d; Lee and Lee, 2011; Miyazaki, 2010b, Miyazaki et al., 2011b; Sultan et al., 2015a). The basic working principle of the system is elaborated in section 3.2; however the detail design varies according to the cooling capacity and ambient conditions. The advantages of evaporative cooling over conventional vapor compression AC (VAC) are well known in the literature (Caliskan et al., 2011a; Duan et al., 2012; Jaber and Ajib, 2011; Khalatov et al. 2011). In particular, the standalone MAC possesses huge energy saving potential and consumes 10 times less primary energy as compared to typical VAC system (Anisimov and Pandelidis, 2011a; Khalatov et al., 2011; Maisotsenko and Treyger, 2011; Tertipis et al., 2015; Weerts, 2011). It provides the conditioned air to inhabitant without recirculating the indoor air (Anisimov et al., 2014b; Anisimov and Pandelidis, 2011b; Chua et al., 2013; Reznikov, 2011). On the other hand, typical VAC system recirculates about 85% of the indoor air in order to make the system cost effective (Khalatov et al., 2011). The fresh air intake is very obligatory for some of the AC applications e.g. hospitals/clinics where the patients are sensitive to indoor air quality (Sultan et al., 2015a), and greenhouses when the plants require sufficient amount of CO_2 for effective photosynthesis (Sultan et al., 2014). In this regard, the standalone MAC systems have been successfully studied for different kinds of applications (Alklaibi, 2015; Caliskan et al., 2011a; Caliskan et al., 2012b; Chua et al., 2013; Heidarinejad and Moshari, 2015; Itani et al., 2015; Lee and Lee, 2013; Miyazaki et al., 2011a; Pandelidis et al., 2015a; Pandelidis and Anisimov, 2015b; Weerts, 2011; Weerts, 2012; Weerts et al., 2012; Xuan et al., 2012; Zhan et al., 2011a).

A standalone MAC system has been experimentally investigated in detail for building AC (Anisimov et al., 2014a; Anisimov et al., 2014b; Anisimov et al., 2015b; Anisimov and Pandelidis, 2014; Anisimov and Pandelidis, 2015; Pandelidis et al., 2015a; Pandelidis et al., 2015b; Pandelidis and Anisimov, 2015a; Pandelidis and Anisimov, 2015b). In the cited literature the authors established energy and mass conservation balance equations, and performed mathematical simulation for heat and mass transfer of the standalone M-Cycle unit. The mathematical model was validated against the experimental result which gives the accurate agreement. It is the most accurate approximations from any available M-Cycle models as far as our understanding is concern. Effect of inlet air conditions on the M-Cycle supply air temperature and dew-point effectiveness is shown in Figure 3.4(a) and 3.4(b), respectively. It can be noticed from Figure 3.4(a) that the system delivers highly cooled air at relatively lower humidity ratio and higher temperature of inlet air, because of the corresponding lower dew-point temperature. Figure 3.4(b) gives more comprehensive outlook in order to optimize the ideal inlet air conditions for the system. The dew-point effectiveness of the system increases with the increase in inlet air temperature at X_{inlet} = 11.2 g/kg-DA. However, it starts decreasing when the humidity ratio exceeds from 11.2 g/kg-DA. It can be concluded that the standalone MAC is efficient when the ambient air humidity ratio is ≤ 11.2 g/kg-DA.

It is therefore concluded that the system can provide desired thermal comfort in residential and commercial buildings when inlet temperature and humidity ratio are up to 45°C and 11.2 g/kgDA, respectively (Anisimov et al., 2014b; Pandelidis et al., 2015a; Pandelidis and Anisimov, 2015a; Pandelidis and Anisimov, 2015b). On the other hand, the system cooling capacity increases at higher temperatures (Anisimov et al., 2014a) which also motivate its applicability in various industrial applications (Khalatov et al., 2011). Furthermore, the system is practically investigated for National Snow and Ice Data Center (NSIDC) in a project funded by National Science Foundation and NASA (Weerts, 2011; Weerts 2012; Weerts et al., 2012) through the complete retrofit of a conventional AC system, the cooling energy has been reduced up to 70% in summer and 90% in winter by

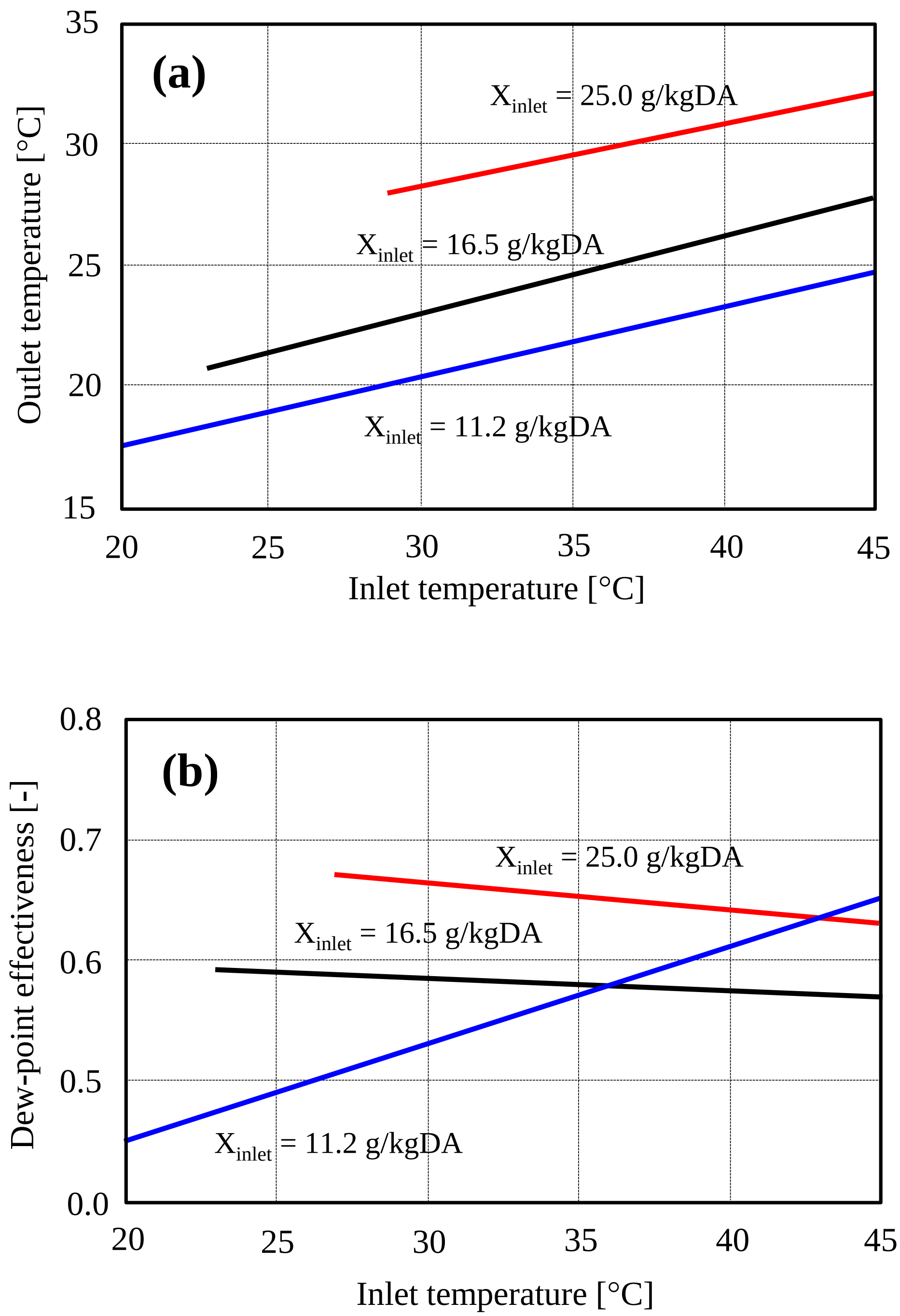

Figure 3.4 Effect of inlet air conditions on the performance of standalone M-Cycle AC unit for: (a) supply air temperature (reproduced from Anisimov et al., 2014a; Anisimov et al., 2014b; Pandelidis and Anisimov, 2015a; Pandelidis et al., 2015a); and (b) dew-point effectiveness (reproduced from Anisimov et al., 2014a).

means of M-Cycle unit. In another study (Caliskan et al., 2011a; Caliskan, 2011b; Caliskan et al., 2012b) the standalone MAC system is found more sustainable in comparison with three kinds of IEC system because of the high exergy efficiency at dead state temperature ≥ 23°C. Dead state is a reference temperature at which thermodynamic system is in equilibrium with the environment. Therefore, it works more efficiently in hot and dry climatic conditions because higher inlet temperature results in more sensible heat transfer between dry and wet channels (Gillan and Maisotsenko, 2008; Maisotsenko and Treyger, 2011).

In addition to above mention scenarios, the standalone MAC can also be used as passive cooling technology for displacement ventilation as well as cooling load reduction. The coming sub-heading discusses the applicability of MAC as passive cooling technology.

3.4.1.1 Chilled Ceiling and Displacement Ventilation

The importance of passive cooling, chilled ceiling and displacement/natural ventilation is well-known in the literature (Ghaddar et al., 2013; Hao et al., 2007; Novoselac and Srebric, 2002; Rees and Haves, 2013; Taki et al., 2011). It helps in cooling load reduction and improves in indoor air quality by possessing high ventilation. The M-Cycle being an advance IEC system has been successfully considered for chilled ceiling displacement ventilation (Itani et al., 2015; Miyazaki et al., 2010a; Miyazaki et al., 2011a). The concept was originally recognized by (Miyazaki et al., 2010a; Miyazaki et al., 2011a) in which the authors performed the system dynamic simulation while using the solar chimney as a system driving force. The schematic diagram of the proposed system is shown in Figure 3.5 (Miyazaki et al., 2010a; Miyazaki et al., 2011a). The study concludes that the system is feasible as a solar energy driven cooling system because the sufficient air flow to the M-Cycle evaporative cooling channel could be induced by the solar chimney. The system can also achieve the radiative load of 40-50 W m^{-2} without increasing the ceiling temperature. Furthermore, the system reduces 10% of the maximum cooling load by replacing quarter of the ceiling area with the proposed system. Similarly, another study (Itani et al., 2015) on M-Cycle based displacement ventilation system showed the improvement in sensible load removal of 18% to 72% for the supply air RH ranging from 90% to 10%, respectively.

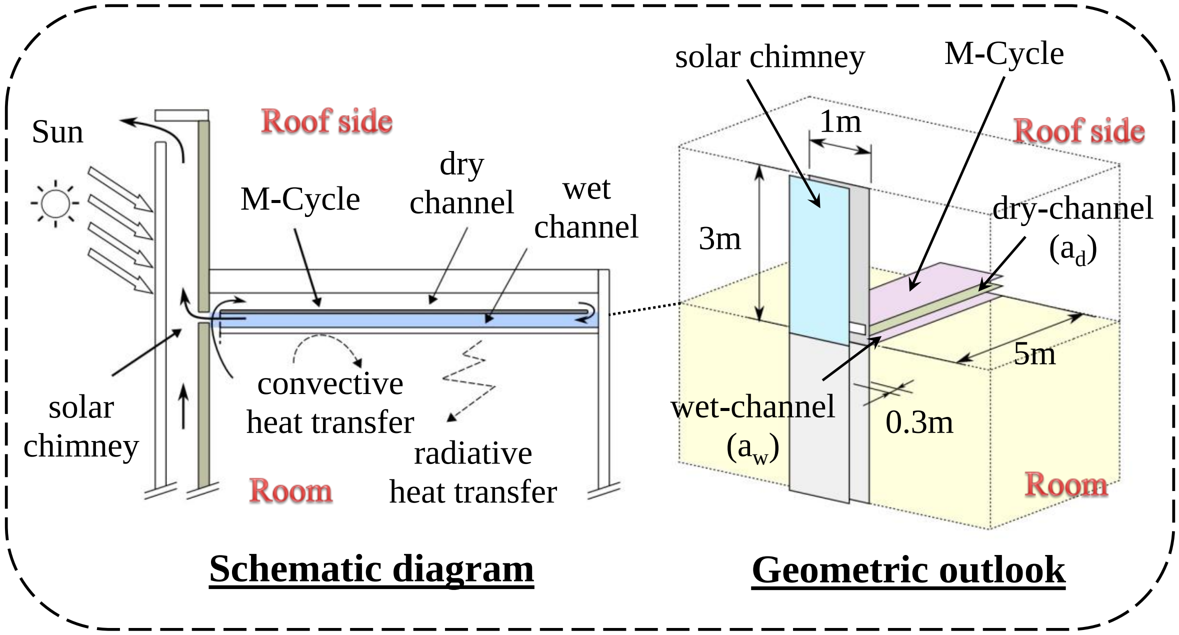

Figure 3.5 Schematic and geometric representation of solar chimney and M-Cycle in a passive cooling system (reproduced from Miyazaki et al., 2010a; Miyazaki et al., 2011a).

From the above prospective it has been concluded that the standalone MAC can achieve the AC load of an active or passive AC system irrespective of temperature only when the humidity is not so high. On the other hand the system design can be modified by many ways in order to achieve the AC loads in humid regions. The details about such modifications are discussed in the coming sections.

3.4.2 Hybrid Maisotsenko Cycle Air-Conditioning

The hybrid M-Cycle AC (H-MAC) combines the features of vapor compression AC (VAC) and M-Cycle AC (MAC), though the operational scheme could be different depending upon the system design. A simple schematic of the system is shown in Figure 3.6 (Anderson et al., 2011; Duan, 2011; Kozubal and Slayzak, 2010). It can be seen that the MAC unit cools the air sensibly to a certain temperature whereas the VAC unit achieves the remaining sensible and latent load of AC by cooling below the dew-point. Furthermore, the return air from the conditioned space can be passed through the M-Cycle wet-channel followed by the condenser's vicinity in order to improve the overall system performance.

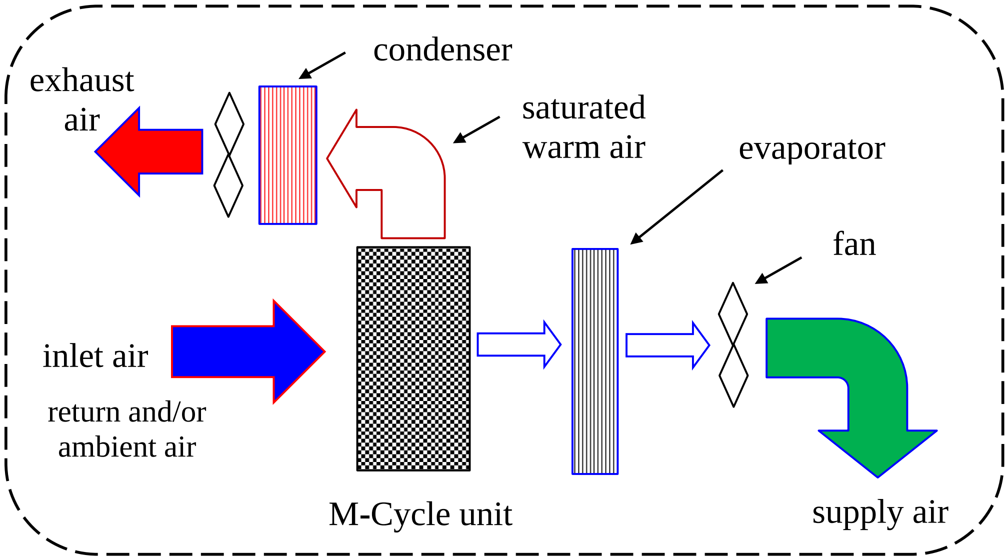

Figure 3.6 Schematic diagram of the hybrid M-Cycle AC (H-MAC) system (Anderson et al., 2011; Duan, 2011; Kozubal and Slayzak, 2010).

The H-MAC system was experimentally investigated by National Renewable Energy Laboratory (NREL) in 2009 (Kozubal and Slayzak, 2010). It has been reported that the system enables energy saving potential of 80% as compared to conventional VAC system (Anderson et al., 2011; Anisimov and Pandelidis, 2011a; Khalatov et al., 2011; Kozubal and Slayzak, 2010; Zube and Gillan, 2011). In another study the H-MAC system has been simulated for hot and humid climates (Cui et al., 2015). The authors proposed a particular H-MAC design and developed the numerical simulation model to investigate the overall system performance. The system cooled the ambient air below the dew-point, and in this regard condensation occurred in the dry-channels due to the lower temperature in wet-channel (working air) as compared to dry-channel dew-point temperature. The lower working air temperature was achieved by mixing the return air from the conditioned space. Results showed that the MAC unit successfully accomplished 40% to 47% of total cooling load depending upon the ambient conditions. Moreover, the system enables higher latent heat transfer rate at various relative humidity when compared with conventional IEC. It has been concluded that the MAC unit as a pre-cooling unit under humid climates will lead to huge energy saving with improved efficiency and reduced vapor compression cooling capacities (Cui et al., 2015). In a similar simulation based study (Dirkes II, 2011) the annual cooling energy saving of 11% to 35% was obtained by H-MAC system when operated in different climatic cities. Furthermore, 25% to 50% less compressor size was obtained when compared with conventional vapor compression system.

3.4.3 Ejector Maisotsenko Cycle Air-Conditioning

The ejector M-Cycle AC (E-MAC) combines the features of ejector AC (EAC) and M-Cycle AC (MAC). In other words, it replaces the compressor of H-MAC with the ejector and the associated assembly. The simplified schematic diagram of the E-MAC system is shown in Figure 3.7 (Buyadgie et al., 2011). The operational mechanism of E-MAC is similar to the H-MAC, however the ejector in the E-MAC system is applied as a jet compressor which is operated by thermal heat most preferably solar energy or low grade waste heat as shown in Figure 3.7. The details of the ejector working principle can be found from the references (Abdulateef et al., 2009; Chen et al., 2013; Chen et al., 2014; Sarkar, 2012). The conventional EAC system enables very low COP as compared to the VAC or other AC systems (Al-Zubaydi, 2011; Chen et al., 2014). Therefore, the combination of EAC with other AC systems (e.g. absorption, VAC, MAC etc.) have been studied with the aim to increase the overall system performance (Aphornratana and Eames, 1998; Buyadgie et al., 2011; Sun, 1997). However, the literature on E-MAC system is limited.

Buyadgie et al., 2011, conducted a study on E-MAC system in order to analyze the system performance for various climatic conditions. The proposed system uses the binary fluid because of the higher COP than the single fluid (Artemenko et al., 2013; Buyadgie et al., 2012a; Buyadgie et al., 2012b). The single and binary fluid based EAC are identical in terms of evaporator, refrigerant condenser, vapor generator, thermal pump, expansion valve and ejector, however the binary fluid based EAC possesses an additional fractionating condenser. The purpose of the fractionating condenser is to separate the mixed fluid (coming from the ejector) into the working and refrigerant fluids. The proposed system is similar to the one shown in Figure 3.7. It is important to mention that the refrigerant and fractionating condensers are presented as single unit on Figure 3.7 in order to avoid the complexity in understating the system operation. The performance of E-MAC and conventional EAC system has been compared at different ambient conditions as shown in Figure 3.8 (Buyadgie et al., 2011). It can be noticed that the E-MAC system possesses

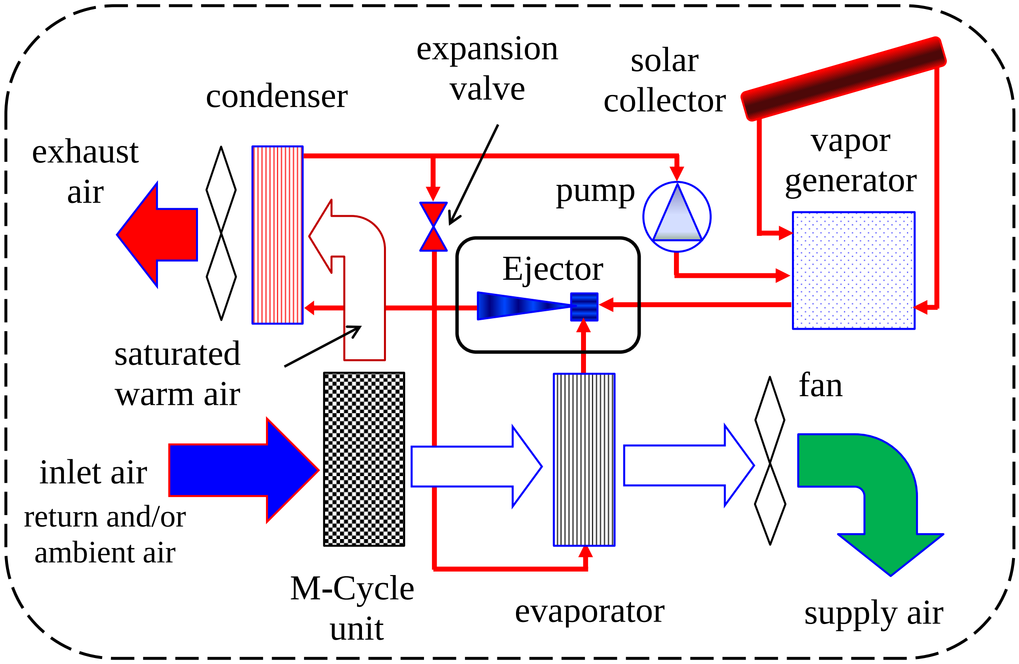

Figure 3.7 Schematic diagram of the ejector M-Cycle AC system (E-MAC) (reproduced from Buyadgie et al., 2011).

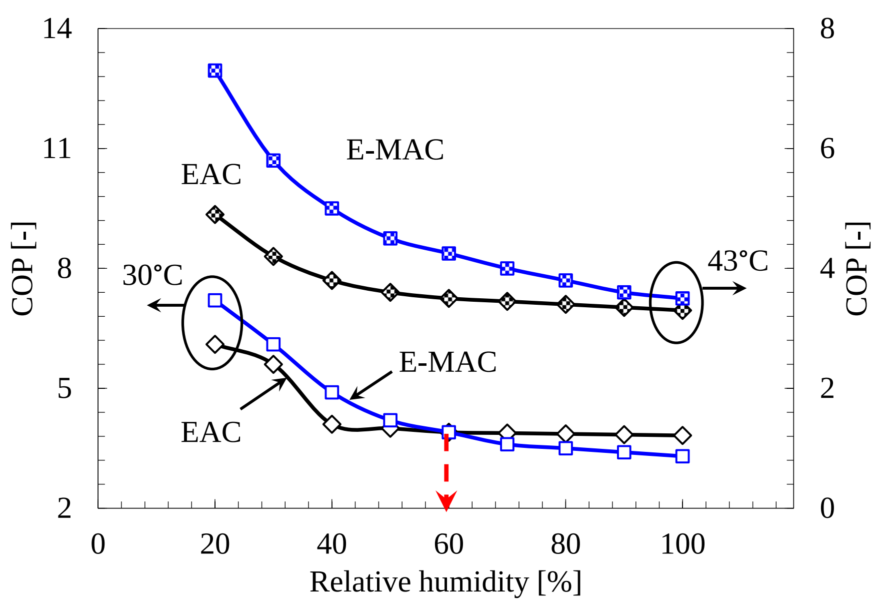

Figure 3.8 Performance comparison between EAC and E-MAC systems at different ambient conditions (reproduced from Buyadgie et al., 2011).

higher COP than the conventional EAC system at RH less than 60% when the ambient air temperature is 30°C. On the other hand the E- MAC delivers higher COP throughout the RH range when the ambient air temperature is 43°C. It was concluded that the E-MAC in humid areas should be considered only if the ambient air temperature is more than 40°C. Furthermore, the system yields the minimum energy consumption for processing of unit air flow rate.

3.4.4 Desiccant Maisotsenko Cycle Air-Conditioning

The desiccant M-Cycle AC (D-MAC) system combines the features of desiccant AC (DAC) and M-Cycle AC (MAC). It has been recently studied by many researchers in order to establish an efficient AC system for humid climates (Amer et al., 2015; Gao et al., 2015b; Miyazaki et al., 2010b; Miyazaki et al., 2011b; Saghafifar and Gadalla, 2015c; Sultan et al., 2015a; Worek et al., 2012). The system can be supplement of standalone MAC in order to be feasible for all kinds of climates. It achieves the latent load of AC by desiccant dehumidification whereas sensible load is accomplished by evaporative cooling (Daou et al., 2006; La et al., 2010; Mei and Dai, 2008; Sultan et al., 2015a). In principle, M-Cycle gives better evaporative cooling as compared to conventional DEC/IEC techniques (Anisimov et al., 2015b), as explained in heading 3.3. It also helps to reduce the system cost by eliminating heat exchanger which costs about 45% of the conventional DAC system (Worek et al., 2012). Moreover, the concept of desiccant integrated MAC (as a single unit) protected by the US patent (Maisotsenko et al., 2002) sound more feasible in order to develop a compact system. The concept covers the variety of solid and liquid desiccants however the literature in this field is limited. The coming headings briefly discuss the potential of M-Cycle in solid and liquid based DAC applications.

3.4.4.1 Solid Desiccant System

The conventional solid DAC system has shown the potential for various kinds of AC applications e.g. residential and office buildings (Baniyounes et al., 2013; Enteria et al., 2009; Enteria et al., 2010, La et al., 2011); automobiles (Nagaya et al., 2006); wet markets (Lee and Lee, 2013); drying grains (Ismail et al., 1991); greenhouses (Sultan et al., 2014); marine ships (Guojie et al., 2012; Zhu and Chen, 2014); museums (Ascione et al., 2009; Ascione et al., 2013); hospitals; product storage and preservation etc. The M-Cycle being an advance IEC can improve the existing DAC systems for different applications covering all kinds of climates. Moreover, the solid D-MAC system has the ability to utilize the exhaust waste heat from the M-Cycle wet-channel for desiccant regeneration, which will increase the system COP. Recent studies have shown its applicability for multiple applications e.g. thermal comfort (Anisimov and Pandelidis, 2011a; Gao et al., 2015a; Miyazaki et al., 2011b; Worek et al., 2012); turbines inlet air cooling (Saghafifar and Gadalla., 2015); greenhouses (Sultan, 2015b); manufacturing and storage processes (Miyazaki et al., 2010a).

Miyazaki et al., 2011b, investigated the potential of solid D-MAC system by means of theoretical and experimental analysis. The simple schematic of the proposed system is shown in Figure 3.9 (Miyazaki et al., 2011b). In addition to M-Cycle unit, the system was consisting of two desiccant beds, a heat exchanger, and a thermal heat unit. A numerical simulation model was developed by the author to simulate the system dynamic performance, and reasonable agreement was obtained between the experimental and simulation results. The dew-point effectiveness (ε_{dp}) by the M-Cycle unit (as given by Eq. 3.2) was found about 0.60-0.75 for supply air flow ratio of $\leq$ 0.75, respectively. The instantaneous variation of final cooling effect of M-Cycle on dehumidified air was determined as shown in Figure 3.10. Results showed that the M-Cycle successfully cooled the dehumidified air from 50-80 °C to the supply air conditions of ~18°C.

According to an experimental study (Enteria et al., 2010), the coefficient of performance (COP) by the standalone DAC system is limited to 0.35-0.44 for regeneration temperature of 60-75°C. However, the theoretical COP by the solid D-MAC system is ranging from 0.60 to 1.10 for regeneration temperature of 70°C to 90°C, respectively (Miyazaki et al., 2011b). Another comparative study (Saghafifar and Gadalla 2015) on four different cooling system showed that the solid D-MAC is the most economically justified inlet cooling technology for a 50 MWe gas turbine power plant in UAE. It possesses life savings of 31.882 MUS$ and life span of 25 years. Furthermore, a numerical simulation based analysis showed the potential of solid D-MAC for reduction in electricity consumption of various applications (Miyazaki et al., 2010b). A payback period of less than 5 years is obtained when the load hours was more than or equal to 4000.

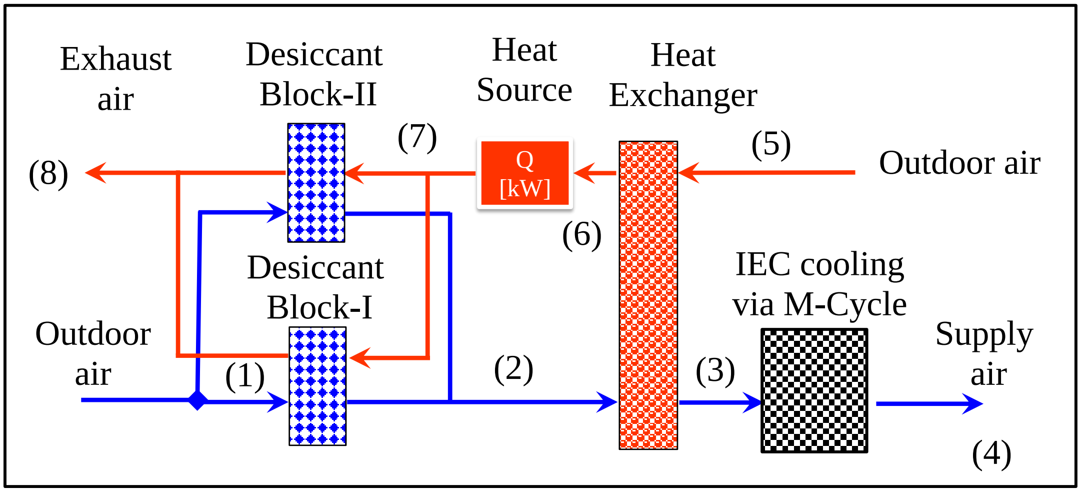

Figure 3.9 Schematic diagram of solid desiccant M-Cycle AC (D-MAC) system (reproduced from Miyazaki et al., 2011b).

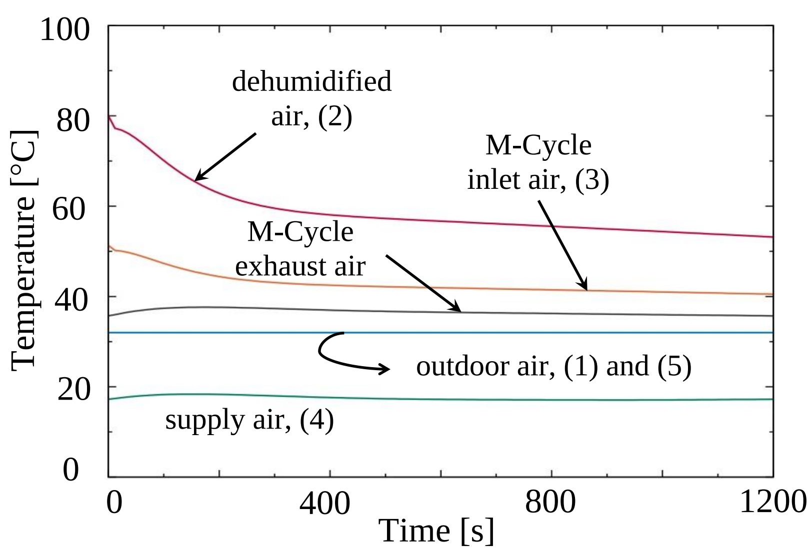

Figure 3.10 Cooling performance of solid D-MAC system (reproduced from Miyazaki et al., 2011b).

3.4.4.2 Liquid Desiccant System

Liquid DAC systems have been widely studied for AC in humid areas (Mohammad et al., 2013a; Mohammad et al., 2013b; Qi et al., 2014). Solar operated liquid DAC systems have shown a huge energy saving potential as compared to conventional AC system (Buker and Riffat, 2015; Mohammad et al., 2013c; Qi and Lu, 2014). However, the system performance is dependent on the sensible AC loads. The MAC can be a good supplement to achieve the sensible AC loads distinctly. Furthermore, the thermal energy from the M-Cycle exhaust air can be used to regenerate the desiccant economically. In this regard, a liquid D-MAC system was experimentally investigated for humid climates (Gao et al., 2015b). The system prototype was developed which uses lithium bromide as desiccant solution. The system top view for one channel pair is presented in Figure 3.11 which shows the working principle of M-Cycle in the liquid D-MAC system (Gao et al., 2015b). Effects of operating parameters on system performance are analyzed.

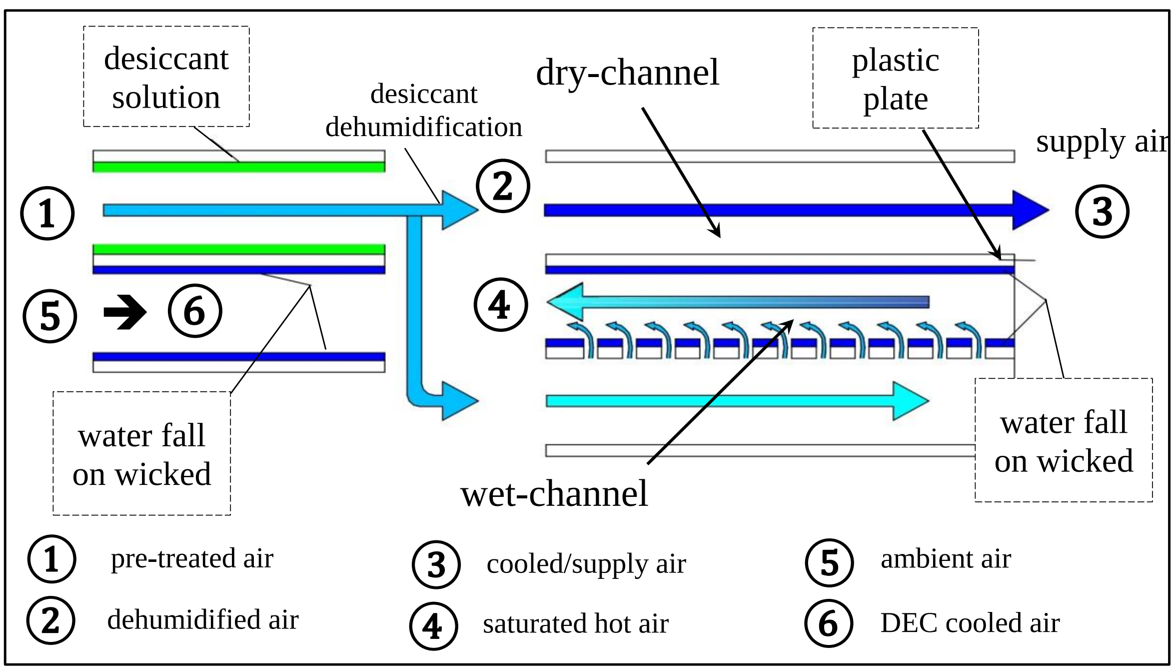

Figure 3.11 Schematic diagram of liquid desiccant M-Cycle AC (D-MAC) system (reproduced from Gao et al., 2015b).

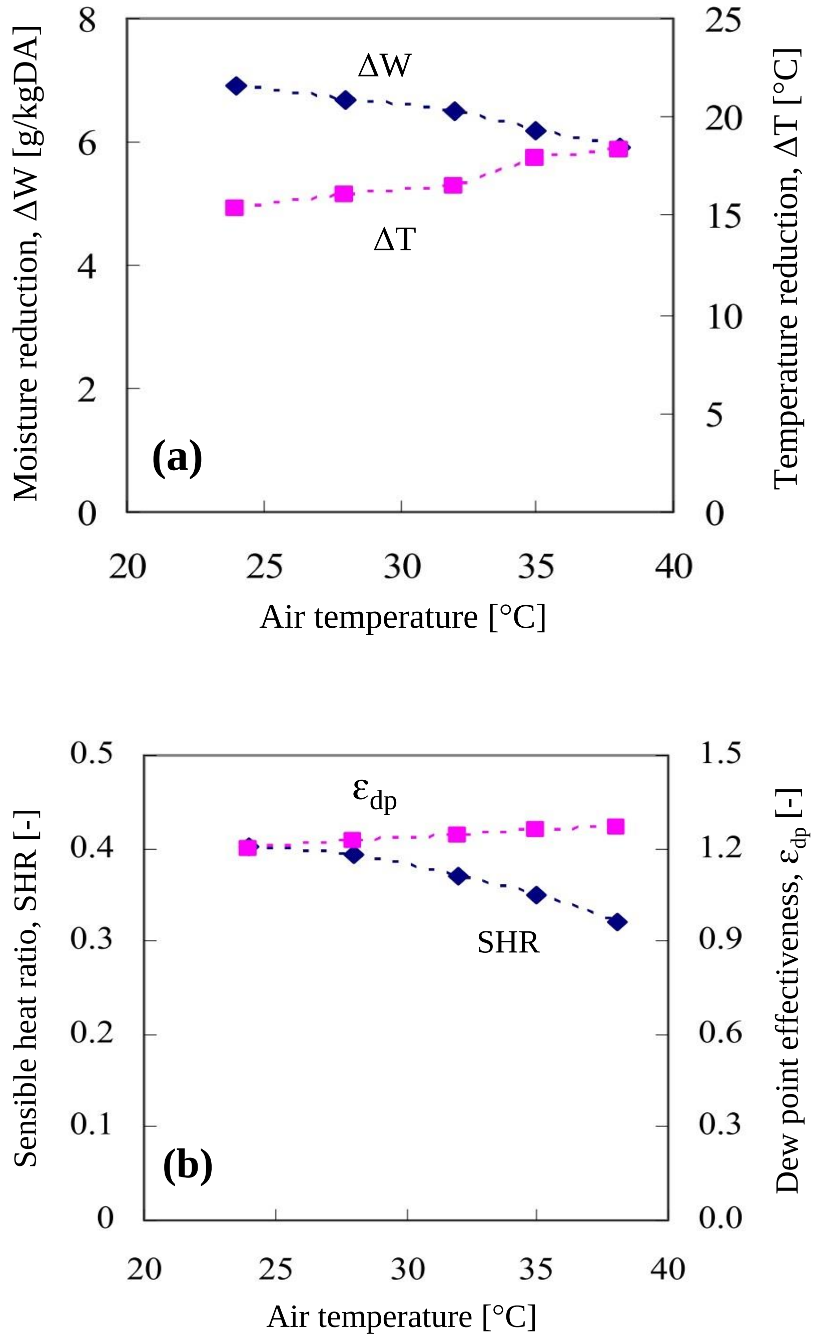

Figure 3.12 Effect of air temperature on the performance of the liquid D-MAC system (reproduced from Gao et al., 2015b).

According to the results, the dew-point effectiveness of about 1.20 is obtained when inlet air humidity ratio was ranging from 9 to 20 g/kg-DA. It is worth mentioning that the dew-point effectiveness exceeds from unity because the dew-point temperature by the dehumidified air is well below than the inlet air. It can be seen from Figure 3.12(a) and (b) that the cooling ability (ΔT) and dew-point effectiveness (ε_{dp}) increases with the increase in air inlet temperature. It is because of M-Cycle dependency on dew-point temperature (Khalatov et al., 2011) as explained in section 3.2. Therefore, the liquid D-MAC has a potential to achieve the sensible AC loads (after desiccant dehumidification) for various humid climates. The study concludes that the moisture removal capacity (in dehumidifier) and sensible heat (in M-Cycle unit) can be improved significantly at the same time by increasing the liquid desiccant flow rate or inlet concentration. Furthermore, a new conception of membrane AC based on liquid D-MAC is proposed recently (Burch et al., 2012; Kozubal et al., 2011; Kozubal et al., 2012; Woods and Kozubal, 2012; Woods and Kozubal, 2013a). Significance of membrane AC is well-known because of the isothermal dehumidification (El-Dessouky et al., 2000; Yang et al., 2015), though the efficiency is debatable. In the cited references (Burch et al., 2012; Kozubal et al., 2011; Kozubal et al., 2012; Woods and Kozubal, 2012; Woods and Kozubal, 2013a), the authors developed numerical model of the proposed membrane/desiccant-MAC system on the basis on their previous studies (Woods et al., 2009; Woods et al., 2011; Woods and Kozubal, 2013b), and successfully validated the model with the experimental data within ±10% error. It has been concluded that there is an inherent design tradeoff between COP and system size (Woods and Kozubal, 2012). According to the analysis, the system enables the energy saving potential as compared to conventional AC technologies. The detail energy saving potential will be demonstrated in the future after field installation of the system as reported by the authors (Woods and Kozubal, 2013b).

3.5 Nomenclature

AC	air-conditioning
COP	coefficient of performance [-]
DAC	desiccant air-conditioning
DEC	direct evaporative cooling
D-MAC	desiccant M-Cycle air-conditioning
EAC	ejector air-conditioning
E-MAC	ejector M-Cycle air-conditioning
ε	effectiveness [-]
H-MAC	hybrid M-Cycle air-conditioning
HVAC	heating, ventilation, and air-conditioning
IEC	indirect evaporative cooling
MAC	M-Cycle air-conditioning
M-Cycle	Maisotsenko Cycle
T	temperature [°C or K]
VAC	vapor compression air-conditioning
X	humidity ratio [g/kg-DA]

Subscript

dp	dew-point
wb	wet-bulb

3.6 References

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CHAPTER 4

EXPERIMENTAL PERFORMANCE EVALUATION OF DESICCANT UNIT

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This chapter presents the setup of an open-cycle experimental apparatus for desiccant dehumidification analysis. Honeycomb like cubical shaped polymer desiccant blocks are used for the purpose. The experiments are performed under different ambient conditions, and various regeneration air temperatures and switching time ratios (five cases). The generalized root of sum of squares method is used to calculate the experimental uncertainty. The net and average effective dehumidification performances of desiccant unit are determined and presented. Influence of regeneration temperature on equivalent heat of adsorption is ascertained. The optimized switching time ratio between regeneration and dehumidification of desiccant unit is suggested. A novel correlation is developed on the conception of modification of isenthalpic slope of dehumidification line on psychrometric chart. The experimental validation of slope of dehumidification under varying regeneration temperature is also investigated. The correlation leads toward the steady-state analysis of desiccant air-conditioning systems.

4.1 Introduction

Low-cost energy-efficient air-conditioning systems for human and nonhuman applications are the dire need of the day due to increasing threats of global warming and climate change. The evaporative cooling technologies are considered low-cost options but cannot be used as standalone under varying climatic (humid) conditions (see appendices). In this regard, the desiccant dehumidification plays a pivotal role to enhance the scope of evaporative cooling in the form of desiccant air-conditioning. The experimental investigation of dehumidification performance of desiccant unit is always required for the designing of optimized DAC system for certain application. Moreover, the experimentally validated models/correlations need to be developed for the particular adsorbent. Such correlations ease the dehumidification analyses for certain applications under varying climates/regions. Therefore, in this chapter an open-cycle experimental apparatus was setup for the performance evaluation of hydrophilic polymer based desiccant blocks. Series of experiments are conducted for various: ambient air conditions, regeneration conditions, cycle time, and switching time. Generalized root sum of squares method is used to calculate the experimental uncertainty. A simplified correlation is developed, on the basis of bunch of experiments, by which real desiccant dehumidification process can be predicted on psychrometric chart for polymeric desiccants. The correlation leads toward the stead-state analysis of desiccant air-conditioning systems. The analyses conducted in the next chapter (chapter 5) are based on this novel and simplified correlation for various applications.

4.2 Experimental Section

4.2.1 Adsorbent/Desiccant

Honeycomb like cubical shaped desiccant blocks were used for desiccant AC application. Study utilizes eight desiccant blocks (20 x 20 x 20 cm^3 each) for the experimental analysis. The desiccant blocks were composed of hydrophilic organic polymer based sorbent (HPBS) (Mahmood et al., 2016, Sultan et al., 2016a) provided by Showa Manufacturing Co., Ltd., Japan. The pictorial and magnifying view of a desiccant block is shown in Figure 4.1. The parametric description of the desiccant block is given in Table 4.1 (Yoshida, 2014). Moreover, the close and open cycle kinetics parametric values of desiccant blocks are described in Table 4.2 (Sultan et al., 2016a). It can be seen (Figure 4.1) that each block contains series of honeycomb like channels. During the experimentation air at designated flow rate was passed through these successive channels.

The cubical shape of desiccant block facilitates easy integration in the system and may also provide ease in batch processing during regeneration and dehumidification.

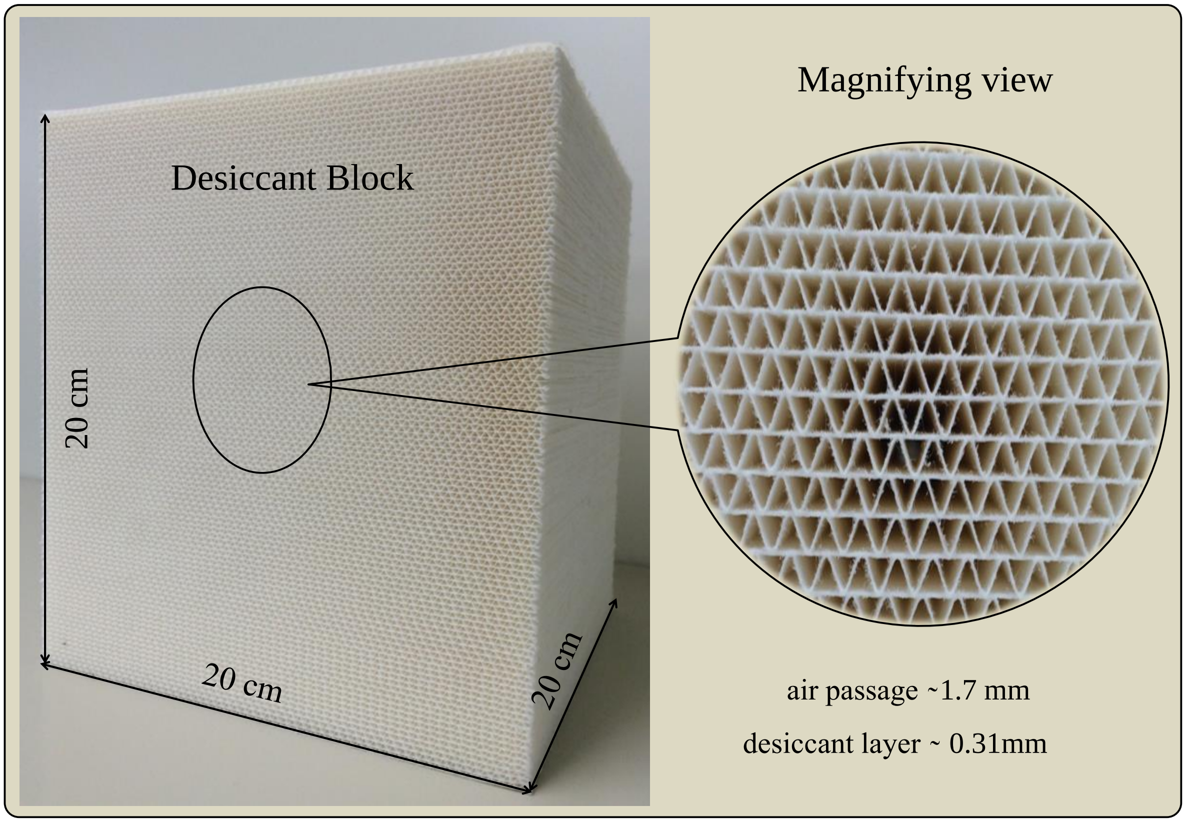


Figure 4.1 Pictorial and magnifying view of the desiccant block.

Table 4.1 Parametric description of desiccant block (Yoshida, 2014).

Parameters description	Values
Dimensions of block [cm]	20 x 20 x 20
Half thickness of air layer [mm]	1.7
Half thickness of sorbent [mm]	0.31
Sorbent density [kg-sorbent/m^3]	1500
Sorption heat [kJ/kg-water]	2.5 x 10^3
Effective diffusion coefficient [m^2/sec]	7.0 x 10^{-4}

Table 4.2 Close and open cycle kinetics parameters values of desiccant block (Sultan et al., 2014a).

Average T_{ads} [K]	$F_o \frac{D_s}{R^2{}_p}$ [value x 10^{-4}]	Average $F_o \frac{D_s}{R^2{}_p}$ [value x 10^{-4}]	Average T_{ads} [K]	$F_o \frac{D_s}{R^2{}_p}$ [value x 10^{-4}]	Average $F_o \frac{D_s}{R^2{}_p}$ [value x 10^{-4}]
Open-cycle adsorption kinetics			Close-cycle adsorption kinetics		
296.24	2.51	1.90	296.70	3.26	2.92
296.22	2.47		296.66	2.23	
295.75	2.36		296.67	2.51	
295.68	2.06		295.15	2.33	
294.90	2.06		295.15	1.31	
294.30	1.17		295.10	5.89	

4.2.2 Experimental Setup

An open-cycle experimental apparatus was setup in this study. It consists of desiccant blocks, air filter, air flow valves, air blowers, water to air heat exchanger, temperature control water bath, water circulator, temperature and relative humidity sensors, differential pressure transmitter, data logger. On the basis of these components the experimental setup is divided into five units such as desiccant block unit (DBU), regeneration air unit (RAU), supplementary heating unit (SHU), process air unit (PAU), and flow rate control unit (FCU). As far DBU is concerned it consists of eight desiccant blocks (HPBS), air filter, inlet and outlet temperature and relative humidity sensors (VAISALA: HMT 333, RH_{ac} = ±1-1.7%; T_{ac} = ±0.2-0.3°C). The orientation of desiccant block unit in experimental setup is shown in Figure 4.2. The desiccant blocks are placed in the experimental setup in such a way (two rows and columns) that it has dimensions of (40 x 40 x 40 cm^3). A netted filter was used in order to avoid the entry of any foreign material to the HPBS. The air temperature and relative humidity at the inlet and outlet of the HPBS is measured by sensors (VAISALA: HMT 333). The detail about other units is described in the relevant subsequent headings.

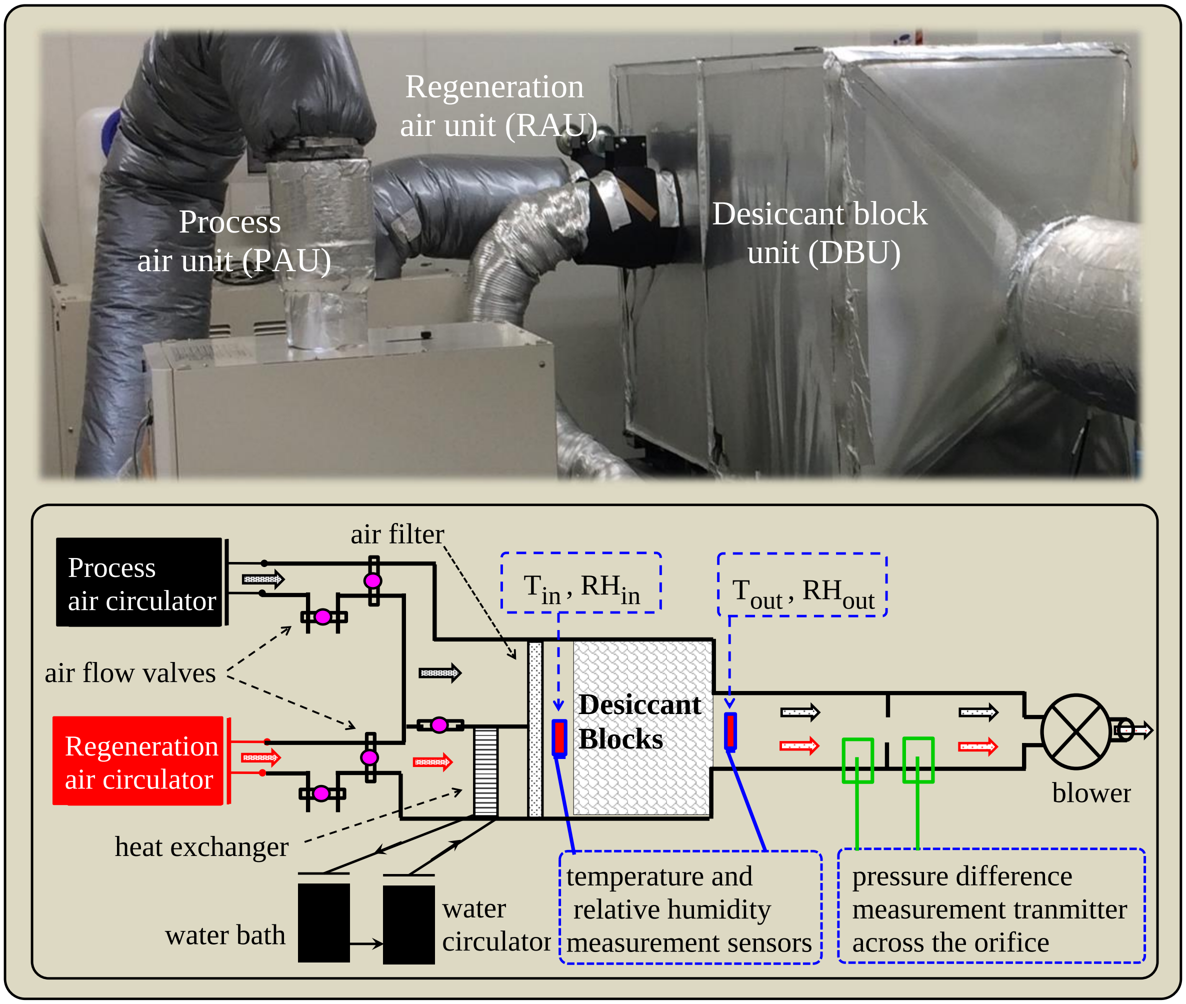


Figure 4.2 Detailed illustration of experimental setup.

4.2.3 Development of Equilibrium Conditions

The experimental setup was placed in an environmentally controlled room. The room was therefore equipped with central air-conditioning (AC) unit which maintains its temperature and relative humidity. The required room temperature and relative humidity are regulated through control panel located outside of the experimental room. The main interface of the control panel has the options to start/stop the system, stop buzzing alarm, reset alarm and menu. It also continuously displays the preset, and achieved value of temperature and relative humidity. The temperature and relative humidity values can also be changed. It must always be confirmed before starting the AC unit that water supply valve connected to the humidifier is open. However, in case of any unusual operation/mishandling the unit starts buzzing alarm and indicate the reason of alarming. The buzzing of alarm can be due to the abnormality in: blower, heater, humidifier, air conditioner, upper/lower limit of temperature and relative humidity, and due to overheat. The indoor environmentally equilibrium conditions were achieved before start of each experiment as explained above. Such conditions permit to analyze the repeatable performance of the desiccant blocks.

4.2.4 Adjustment of Operating Conditions

4.2.4.1 Regeneration Air Flow

The desiccant blocks are substantially regenerated before the dehumidification process. The hot regeneration air (RA) is supplied for the purpose through RAU. The RAU consists of regeneration air circulator with temperature and relative humidity sensor (APSITE: PAU-H3200-6KHC, T_{ac} = ±0.5°C; RH_{ac} = ±2%), supply and exhaust air duct, water source, and air flow valves. The orientation of RAU in experimental setup is shown in Figure 4.2. It can supply the regeneration air at temperature and relative humidity ranging from 0-55°C, and 0-85% respectively. The supply of water to the RAU must always be ensured in order to have active control over relative humidity. Moreover, the temperature and relative humidity sensors of RAU must be placed properly in its supply air duct or within the experimental setup. Its front panel contains temperature and relative humidity controller, RUN/STOP and Drain ON/OFF buttons/switches, and light indications about power, alarm, fan, heater, humidifier and water. The adjustments of regeneration air temperature and relative humidity was made by regulating the RAU as follows

i. ensure that water supply valve is open and there is no water lock in supply line
ii. ensure that drain switch is in OFF position
iii. check that all the front control panel light indications turned green except alarm indication (which remains off) during normal working operation
iv. adjust the temperature and relative humidity as set values (SV)

v. preset values (PV) of temperature and relative humidity start increasing/decreasing as per required SV
vi. wait until the PV matches to SV of temperature and relative humidity, during this period air was exhausted outside of experimental room through exhaust channel
vii. regeneration air temperature and relative humidity as reached to desire set value, close the exhaust valve in order to pass the hot regeneration air through the desiccant blocks

It is important to mention that the RAU increases the temperature of the air up to 55°C, however further increase in air temperature was made by employing SHU. The SHU consists of air to water heat exchanger (HX), temperature control water bath (ADVANTEC: TBN402DA, T_{ac} = ±0.1°C), water circulator (EYELA CTP-3000, T_{ac} = ±0.1°C) and accessories. The temperature control water bath can raise the water temperature up to 90°C. The hot water from temperature control water bath is supplied to the HX through water circulator. Moreover, the heating mode provision of the water circulator enable to control any fluctuation in the temperature of water supplied to the HX. The hot water circulating in the HX exchanges heat with air passing over it. The warm water coming out from the HX is flowed to the temperature control water bath in order to maintain the hot water supply cycle.

4.2.4.2 Process Air Flow

The air supplied to the desiccant blocks during dehumidification mode is termed as process air (PA). It was supplied to the desiccant blocks through process air unit (PAU). The PAU consists of process air circulator with temperature sensor (APSITE: PAU-AZ1800SE, T_{ac} = ±0.05-0.1°C), supply and exhaust air duct, and air flow valves. The orientation of PAU in experimental setup is shown in Figure 4.2. It can supply the air at temperature ranging from 8°C to 47°C (when damper fully opened), however the relative humidity is automatically adjusted as per the room ambient conditions. The temperature sensor needs to be place inside the outlet air duct of the PAC or inside the experimental setup. The process air temperature during experimentation was adjusted using SEL and

Up/Down push buttons. The desired set value (SV) of the temperature was achieved in couple of minutes depending upon the preset value and ambient environmental conditions. The air was exhausted outside of the experimental room till it achieves the temperature set value. It is important to mention that the experimental setup also has option to use the process air circulator as regeneration air circulator. The temperature of the air thus can be increased by passing it through HX of SHU in order to use it as regeneration air. However, when both the PAU and RAU were employed, the process air by-passed the heat exchanger.

4.2.4.3 Air Flow Rate

The regeneration and process air flow rate during experiments is regulated by FCU. It consist of circular orifice, differential pressure transmitter (TESTO: 6349, P_{ac} = 0.35Pa + 0.6% of full scale) and variable speed blower (SHOWA: EC-100T-R313) with accessories as shown in Figure 4.3. The flow rate of PA and RA is regulated by changing the speed of blower and measuring the pressure difference (ΔP) across the circular orifice. The ΔP across the circular orifice was measured by differential pressure transmitter. In this regard, its positive and negative pressure hose are connected with the inlet and outlet hose of the orifice, respectively. Therefore, changes in the pressure across the orifice create movement in its membrane spring which oscillates between two inductive transmitters to indicate the ΔP. The flow rate of PA and RA are then calculated using particular orifice Eq. (4.1) (ASHRAE, 2009).

$$m = C_d\, A\, \sqrt{2\, \rho\, \Delta P} \tag{4.1}$$

where m is air mass flow rate [kg/sec], C_d is the discharge coefficient [-], A is the area of the orifice [m^2], ΔP is air pressure difference between inlet and outlet of the orifice [Pa] and ρ is the density of the air [kg/m^3]. The diameter of the orifice and discharge coefficient is taken as 8.0 cm and 0.6, respectively.

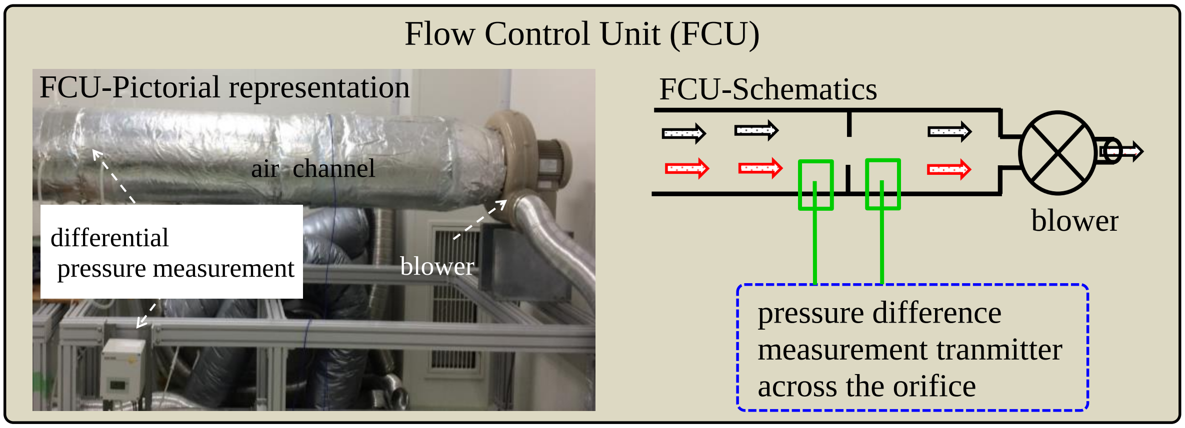


Figure 4.3 Illustration of flow control unit of experimental setup.

4.2.5 Experimental Procedure

The ambient conditions of the experimental room are always maintained before start of experiments as explained in heading 4.2.3. Afterwards, experiments are started by performing initialization of desiccant blocks. During initialization desiccant blocks are first regenerated and then dehumidified in order to provide the thermodynamically equilibrium conditions for subsequent regeneration and dehumidification processes. It is important to mention that the experimental data taken after the initialization is used in the analyses. Referring to Figure 4.2 the regeneration air is supplied to the desiccant blocks through RAU. The regeneration was performed at varying temperature (40-60°C). The regeneration at low temperatures was selected in order to simulate the conditions that can be available through the use of low grade waste heat, solar energy (preferably solar thermal) and biogas. As RAU increases the regeneration air temperature up to 55°C, therefore further increase in regeneration air temperature (when required) was made using SHU. Such adjustments (T and RH) in regeneration air were made as explained in heading 4.2.4.1. After the regeneration of the desiccant blocks, the process air is supplied to the desiccant block through PAU. The varying process air conditions (T_{in} and RH_{in}) were provided to the desiccants block in order to analyze the dehumidification performance of the desiccant blocks. The switching time ratios between regeneration and dehumidification modes were also varied (as 1:1, 2:3, 1:2) during the experiments. The time ratio (1:1) means that after performing 60 minutes regeneration the desiccant blocks are switched for 60 minutes dehumidification. Moreover, the flow rate of PA and RA were regulated by FCU as explained in heading 4.2.4.3. The instantaneous values of temperature, relative humidity and differential pressure during experiments were recorded by data logger (DAQMASTER: MX 100, accuracy = ±0.01%) connected with personal computer. It was used to record the data for every 10 second interval. The representation of experimental data recoding sheet is shown in Table 4.3.

Table 4.3 Data recording sheet.

Time (sec)	ΔP (Pa)	T_{in} (°C)	T_{out} (°C)	RH_{in} (%)	RH_{out} (%)
-	-	-	-	-	-
-	-	-	-	-	-
-	-	-	-	-	-
-	-	-	-	-	-

4.2.6 Uncertainty Analysis

The uncertainty analysis of the measured and/or calculated variables was carried out to determine the accuracy of the experimental data. The uncertainty in experimental data depends on the instrument type, instrument calibration, ambient conditions, observations, readings, data recordings, experiment planning and other associated factors (Hepbasli and Akdemir, 2004). The uncertainty in the resultant function (σ_R) of the independent variables can be determined by the generalized root of sum of squares method (Esen et al. 2006; Holman, 2012) as described in Eq. (4.2).

$$\sigma_R = \sqrt{\left(\frac{\partial R}{\partial N_1}\sigma_1\right)^2 + \left(\frac{\partial R}{\partial N_2}\sigma_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial N_n}\sigma_n\right)^2} \tag{4.2}$$

where σ_1, σ_2 ,…, σ_n represents the uncertainties in the independent variables N_1, N_2,…, N_n. Whereas, the result R is given function of independent variables. The total uncertainties (σ_t) in RH, T, and ΔP measurements are however calculated by Eq. (4.3) as cited in the literature (Hepbasli and Akdemir, 2004; Uçkan et al. 2013).

$$\sigma_t = \sqrt{\sigma_{sr}^2 + \sigma_{cp}^2 + \sigma_{du}^2} \tag{4.3}$$

where σ_{sr}, σ_{cp} and σ_{du} describe the uncertainties of sensor, calibration process, and data acquisition unit, respectively. The values of these uncertainties are provided by the particular instrument's manufacturer. Thus, the total uncertainties in the measurements of relative humidity, temperature and pressure are calculated as ±1.97%, ±1.10% and ±2.47%, respectively. However, the percent uncertainty in the mass flow rate ($\frac{\sigma_m}{m}$) (Hepbasli and Akdemir, 2004; Holman, 2012) is calculated using the Eq. (4.5). The Eq. (4.5) is obtained by the algebraic manipulation of Eqs. (4.1), (4.2) and (4.4).

$$\sigma_m = \sqrt{\left(\frac{\partial m}{\partial \Delta P}\right)^2 \sigma_{\Delta p}^2 + \left(\frac{\partial m}{\partial \rho}\right)^2 \sigma_\rho^2} \tag{4.4}$$

$$\frac{\sigma_m}{m} = \sqrt{0.25 \left(\frac{\sigma_{\Delta P}}{\Delta P}\right)^2 + 0.25 \left(\frac{\sigma_\rho}{\rho}\right)^2} \tag{4.5}$$

The uncertainty in calculating the mass flow rate is found as ±1.81%. It is important to mention that the uncertainty in the mass flow rate is mainly due to the uncertainty in the measurement of pressure difference across the orifice.

4.3 Theory and Methods

Performance evaluation of desiccant blocks is experimentally investigated using an open-cycle experimental setup (Figure 4.2) for various process and regeneration air conditions. The experiments were initially conducted for two reference inlet air conditions case-A ($T_{in} \approx 31°C$, $X_{in} \approx 6$ g/kg-DA) and case-B ($T_{in} \approx 13°C$, $X_{in} \approx 6$ g/kg-DA) as briefly elaborated in Table 4.4. The inlet and outlet air conditions of desiccant unit are measured experimentally whereas rests of the conditions are calculated by Eqs. (4.6)-(4.14) (ASHRAE, 2009; Hyland and Wexler 1983; Stull, 2011). The development of experimental setup and its operating procedure is however discussed in heading 4.2.

$$X = 0.621945 \frac{P_v}{P_{atm} - P_v} \tag{4.6}$$

$$P_v = P_{vs}\, RH \tag{4.7}$$

$$lnP_{vs} = \left[\frac{\zeta_1}{T} + \zeta_2 + \zeta_3\, T + \zeta_4\, T^2 + \zeta_5\, T^3 + \zeta_6\, ln\, T\right] / 1000 \tag{4.8}$$

$$T_{dp>0} = \zeta_7 + \zeta_8\, \alpha + \zeta_9\, \alpha^2 + \zeta_{10}\, \alpha^3 + \zeta_{11}\, (P_v)^{0.1984} \tag{4.9}$$

$$T_{dp<0} = 6.09 + 12.608\, \alpha + 0.4959\, \alpha^2 \tag{4.10}$$

$$\alpha = ln\, P_v \tag{4.11}$$

$\zeta_1 = -5.8002206E + 03$

$\zeta_2 = 1.3914993E + 00$

$\zeta_3 = -4.8640239E - 02$

$\zeta_4 = 4.1764768E - 05$

$\zeta_5 = -1.4452093E - 08$

$\zeta_6 = 6.5459673E + 00$

$\zeta_7 = 6.54$

$\zeta_8 = 14.526$

$\zeta_9 = 0.7389$

$\zeta_{10} = 0.09486$

$\zeta_{11} = 0.4569$

$$\rho = \frac{P}{R_{DA}\, T} \tag{4.12}$$

where X is humidity ratio [kg/kg-DA], P_v is partial pressure of water vapor [kPa], P_{vs} is saturation pressure of water vapor [kPa], P_{atm} is atmospheric air pressure [kPa], P is air pressure [Pa], ρ is air density [kg/m^3], R_{DA} is specific gas constant of dry air [J/kg·K], RH is relative humidity [-], T is absolute temperature [K], T_{dp} is dew-point temperature [°C].

The mass flow rate of process air (m_{PA}) and regeneration air (m_{RA}) was determined (Eq. 4.1) as 0.1 kg/sec during the desiccant block dehumidification and/or regeneration open-cycle experiments. Therefore, the process to regeneration air flows ratio was 1:1 during experiments and analyses. The air enthalpy [ASHRAE, 2009] and wet-bulb temperature [Stull, 2011] are however calculated by Eq. (4.13) and Eq. (4.14), respectively.

$$h = 1.006\, T + X\,(2501 + 1.86\, T) \tag{4.13}$$

$$\begin{aligned} \mathrm{T_{wb}} = \; & T\, tan^{-1}\left(0.151977 * \sqrt{RH + 8.313659}\right) + tan^{-1}\,(T + RH) \\ & - tan^{-1}\,(RH - 1.676331) + 0.00391838\,(RH)^{3/2}\, tan^{-1}(0.023101\, RH) \\ & - 4.686035 \end{aligned} \tag{4.14}$$

where h is air enthalpy [kJ/kg-DA], $\mathrm{T_{wb}}$ is wet-bulb temperature [°C], T is air dry bulb temperature [°C], RH in case of calculating wet-bulb temperature is in percentage [%].

The net water vapors adsorption/desorption amount (ΔX) is calculated by Eq. (4.15). The average dehumidification ($\overline{\Delta X}$) is calculated by Eq. (4.16). Whereas the effective average dehumidification ($\overline{\overline{\Delta X}}$) is calculated by Eq. (4.17).

$$\Delta X = X_{in} - X_{out} \tag{4.15}$$

$$\overline{\Delta X} = \frac{1}{\bar{n}} \sum_{i=1}^{\bar{n}} (X_{in} - X_{out})_i \tag{4.16}$$

$$\overline{\overline{\Delta X}} = \frac{1}{\bar{\bar{n}}} \sum_{i=1}^{\bar{\bar{n}}} (X_{in} - X_{out})_i \tag{4.17}$$

where $\bar{n}$ represents number of experimental entries during dehumidification cycle and $\bar{\bar{n}}$ represents the number of experimental entries during dehumidification and regeneration cycle. In other words, $\overline{\Delta X}$ deals the dehumidification cycle whereas $\overline{\overline{\Delta X}}$ presents the combined impression of dehumidification and regeneration.

4.4 Results and Discussion

4.4.1 Performance Evaluation of Desiccant Unit

An open-cycle desiccant block experimental setup was developed for the performance evaluation of desiccant blocks at different process and regeneration air conditions (Table 4.4 and Table 4.5). The study initially focuses two reference inlet air conditions i.e. case-A ($T_{in} \approx 31$°C, $X_{in} \approx 6$ g/kg-DA) and case-B ($T_{in} \approx 13$°C, $X_{in} \approx 6$ g/kg-DA).

The regeneration of desiccant blocks was made at low regeneration temperatures (Table 4.4) for both the cases (A and B) in order to simulate the conditions that can be available through the use of low grade waste heat, solar energy, bio-gas etc. The air mass flow rate for all experiments was used as 0.1 kg/sec. Whereas, the regeneration to process air flow ratio was maintained at 1:1. The experimental profiles of dynamic behavior of process and regeneration air streams of case-A and case-B at regeneration temperature (T_{reg}) of 60°C are shown by Figure 4.4(a) and 4.5(b), respectively. Similarly, the experimental temperature and relative humidity profiles at T_{reg} = 40°C for case-A and case-B are however shown in Figure 4.5(a) and 4.6(b), respectively. Referring to dehumidification cycle in Figure 4.4(b), ambient air enters in to desiccant block at RH~65-70% and exits at RH~12-25%. It shows huge potential of air dehumidification which can be

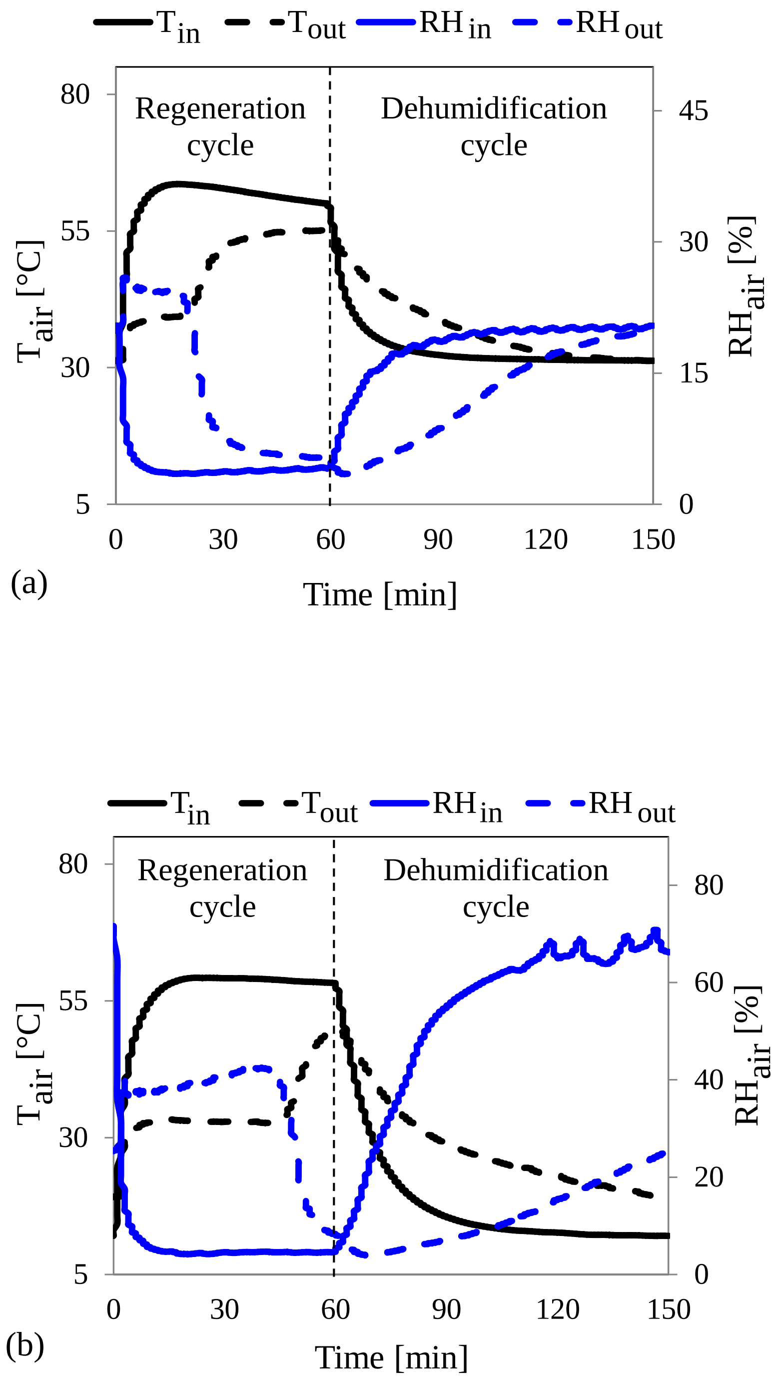
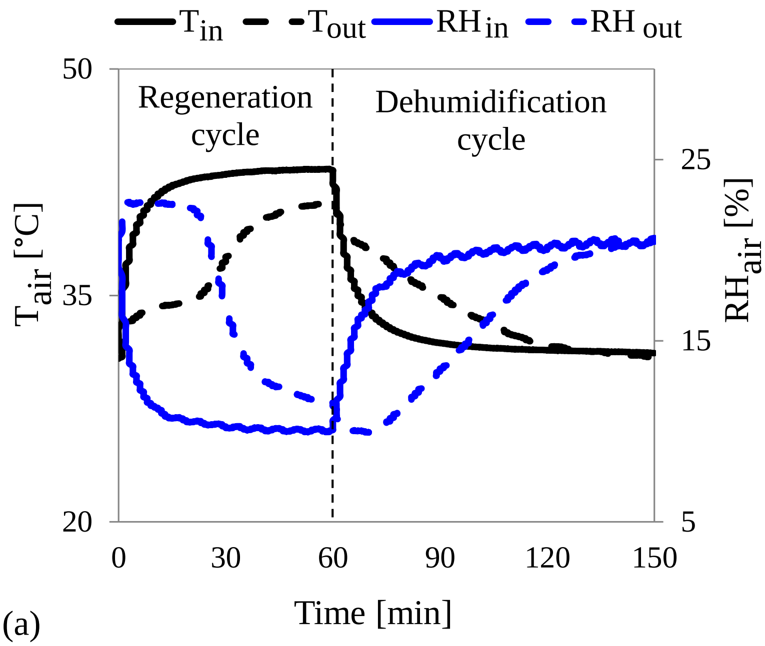
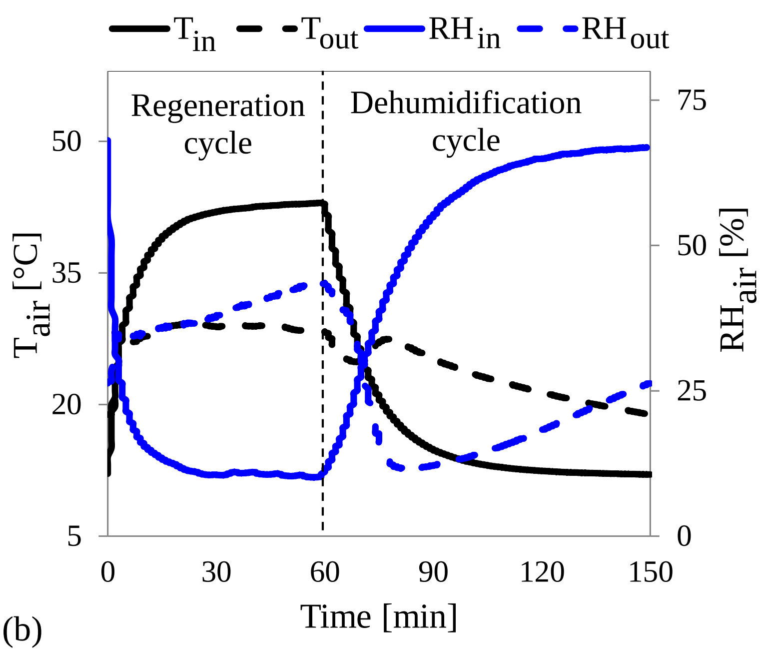


Figure 4.4 Experimental profiles of ambient air conditions on desiccant dehumidification performance at 60°C regeneration temperature: (a) case-A, and (b) case-B.



(a)



(b)

Figure 4.5 Experimental profiles of ambient air conditions on desiccant dehumidification performance at 40°C regeneration temperature: (a) case-A, and (b) case-B.

Table 4.4 Experiments under varying regeneration temperatures and constant switching time ratio.

Case	Process air		Regeneration air		Switching time (Reg:Deh) [min]
	T_{in} [°C]	X_{in} [g/kg-DA]	T_{in} [°C]	X_{in} [g/kg-DA]	
A	31	6	40-60	6	60:90
B	13				

utilized for various DAC applications. For example, return air from postharvest storage zone I to III (or ambient air) (see chapter 5) can be successively dehumidified to lower relative humidity in order to provide optimum storage conditions. Consequently, required level of respiration, transpiration and fermentation can be maintained by means of exposure of dehumidified return/fresh air. Similarly, desiccant based systems can successfully achieved required conditions of temperature and relative humidity for various applications e.g. agricultural greenhouses (Sultan et al., 2016b), animal AC (Sultan et al., 2017), wet markets (Lee and Lee, 2013). It is important to mention that desiccant dehumidifies the air at one side, and heated the air on the other side due to heat of adsorption/condensation. The present study presents the similar behaviour of desiccant dehumidification (Figure 4.4 and Figure 4.5) in which the dehumidified air is heated up more than the heat of condensation due to the isosteric heat of adsorption. Most of the DAC systems possess the desiccant unit followed by the heat exchanger (HX) in order to recover the sensible energy, consequently similar conception has also been considered in the proposed DAC systems (chapter 5). Hence adsorption heat generates can be somehow managed by HX unit.

From the above prospective, Figure 4.6 shows two identical experiments at regeneration temperature of 40°C and 60°C in order to investigate adsorption heat. In the present study effect of adsorption heat has been realized by equivalent heat of adsorption (q_{eq}) as expressed by Eq. (4.18).

$$q_{eq} = m\,(h_{out} - h_{in}) \tag{4.18}$$

where m is air mass flow rate [kg/sec] and h is air enthalpy [kJ/kg]. The numerical values of q_{eq} in both cases is increasing with the regeneration temperature, however decreases temporarily at all regeneration temperatures. This phenomena of adsorption heat (Q_{st}) is obvious and can be understood from Clausius–Clapeyron relationship as expressed by Eq. (4.19) (Sultan et al., 2015).

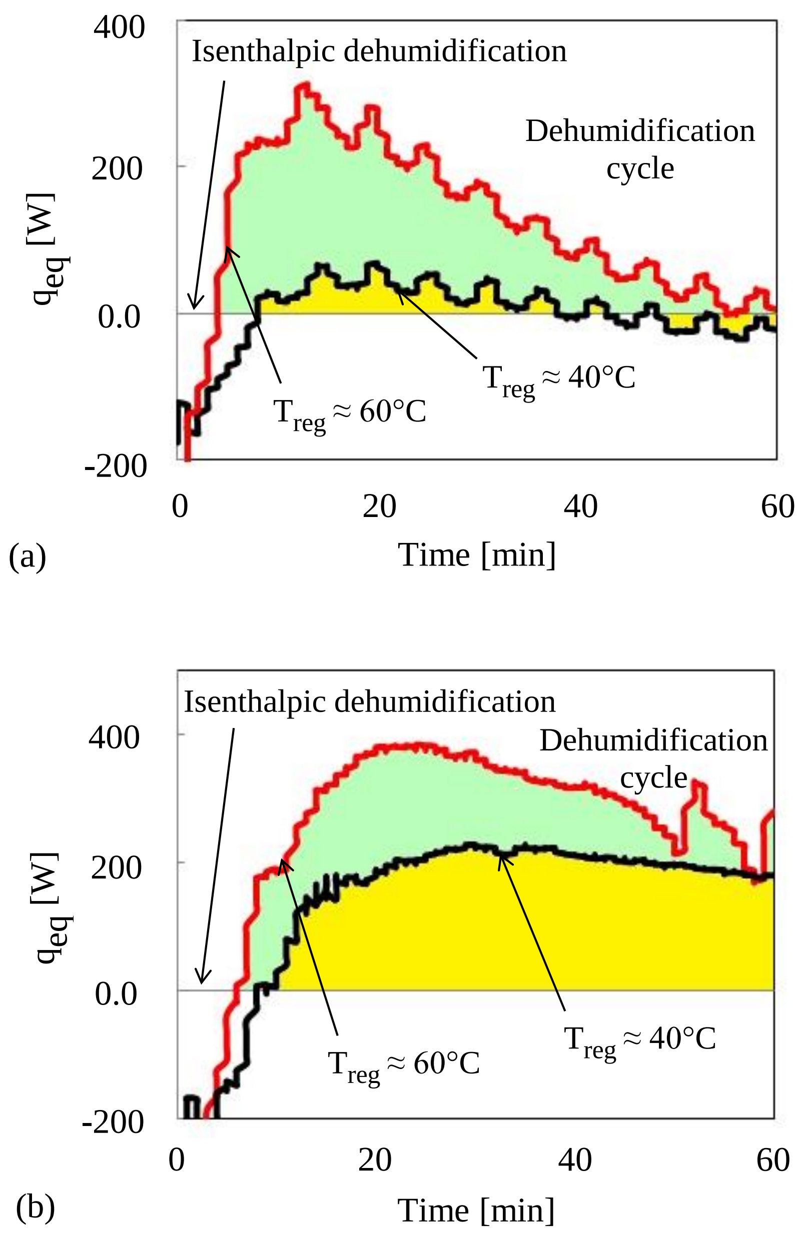


Figure 4.6 Influence of regeneration temperature on equivalent adsorption heat for: (a) case-A, and (b) case-B.

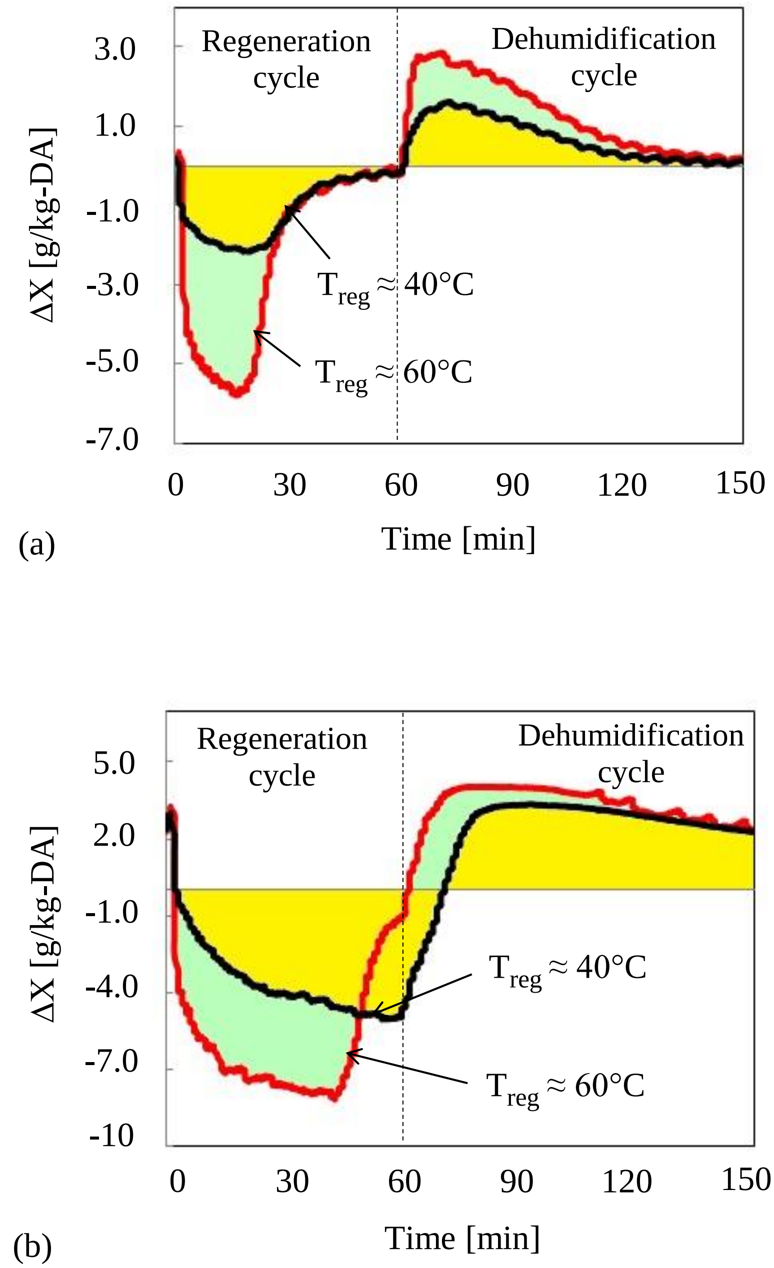


Figure 4.7 Effect of regeneration temperature on net dehumidification and regeneration performance for: (a) case-A, and (b) case-B. Negative and positive values indicate regeneration and dehumidification cycle, respectively.

$$\frac{Q_{st}}{R_v} = -\left[\frac{\partial \ln P_v}{\partial \left(1/T_{db}\right)}\right] \tag{4.19}$$

where P_v, R_v , and T_{db} represent the partial water vapor pressure [kPa], water vapor gas constant [kJ/kg·K], and dry bulb temperature [K], respectively. Referring to Clausius–Clapeyron equation the adsorption heat is the function of adsorbed water vapors, therefore variation in q_{eq} in both cases and different regeneration temperatures (Figure 4.6) is due to the amount of adsorbed water vapors. The q_{eq} line approaches to net value equals zero indicates the isenthalpic adsorption or zero adsorption phenomena. This representation is handy in order to quantify the desiccant unit switching time (t). For example in Figure 4.6(b) process air condition at t = 60 min indicates that the desiccant still has ability to dehumidify the air, therefore dehumidification cycle time can be increased accordingly at both regeneration temperatures. On the other hand in Figure 4.6(a), the q_{eq} of process air (t = 60 min) is approaches to zero which shows that the desiccant probably has been saturated and therefore should be switched for regeneration. Similarly in another study (Sultan et al., 2017) effect of adsorption heat on desiccant performance has been reported.

Effect of regeneration temperatures on net water vapors adsorption/desorption amount per kg of dry air has been investigated using Eq. (4.15) at T_{reg} of 40°C and 60°C as shown in Figure 4.7(a) and 4.7(b) for case-A and case-B, respectively. According to the thermodynamics laws total water vapor adsorption (dehumidification cycle) is equal to total water vapor desorption amount (regeneration cycle) at certain regeneration temperature. For example the area under the line (Figure 47) should be equal in dehumidification and regeneration cycle. The fact is quite clear for both cases and multiple regeneration temperatures. However, for both cases it is interesting to note that the total water vapor desorption amount per minute (regeneration cycle) is significantly higher than the total adsorption per minute (dehumidification cycle). Therefore, the dehumidification cycle must be longer than the regeneration cycle (particularly in case-B) in order to establish a cost effective DAC system. It can also be concluded that the regeneration cycle can be even shrinked by employing higher regeneration temperature. The reason of this behaviour is due to typical adsorption kinetics phenomena of polymer based sorbents as reported by Sultan et al., 2016c. It can be simply explained that as the water vapor sorption kinetics is higher at higher regeneration temperatures resulting the lower cycle time for regeneration as compare to the dehumidification. This is also one of the fact which motivate the block based DAC as compare to rotary DAC, which provides the provision of variable dehumidification and regeneration cycle time. However in case of rotary DAC regeneration and dehumidification are made simultaneously (Sultan et al., 2016b). Similarly the proposed systems (see chapter 5) are useful for dehumidification at night time and regeneration at day time preferably by solar radiation. In addition, the amount of dehumidification presented by Figure 4.7 is useful for designing the process/return air flow rates and ultimately the regeneration temperature for particular agricultural product storage (see Table 5.1).

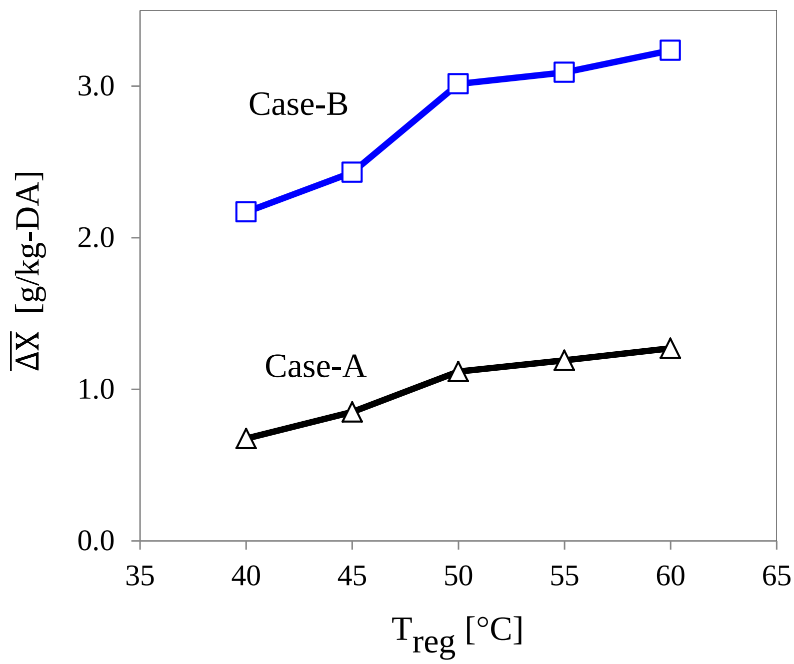


Figure 4.8 Effect of regeneration temperature on average dehumidification of desiccant unit.

From the discussion point of view, the average dehumidification ($\overline{\Delta X}$) performed by desiccant blocks under case-A and case-B at T_{reg} = 40°C-60°C is compared and presented by Figure 4.8. It can be seen (Figure 4.8) that higher $\overline{\Delta X}$ is resulted by case-B at all the regeneration temperatures as desiccant blocks accommodates huge amount of water vapor adsorption due to higher RH (case-B). It is obvious higher regeneration temperatures bring deep dehumidification though it might not be effective in terms of net COP of DAC system. Moreover, $\overline{\Delta X}$ increases with the increase in the regeneration temperature (40°C-60°C). On the other hand relative dehumidification difference decreases with the increase in the regeneration temperature from

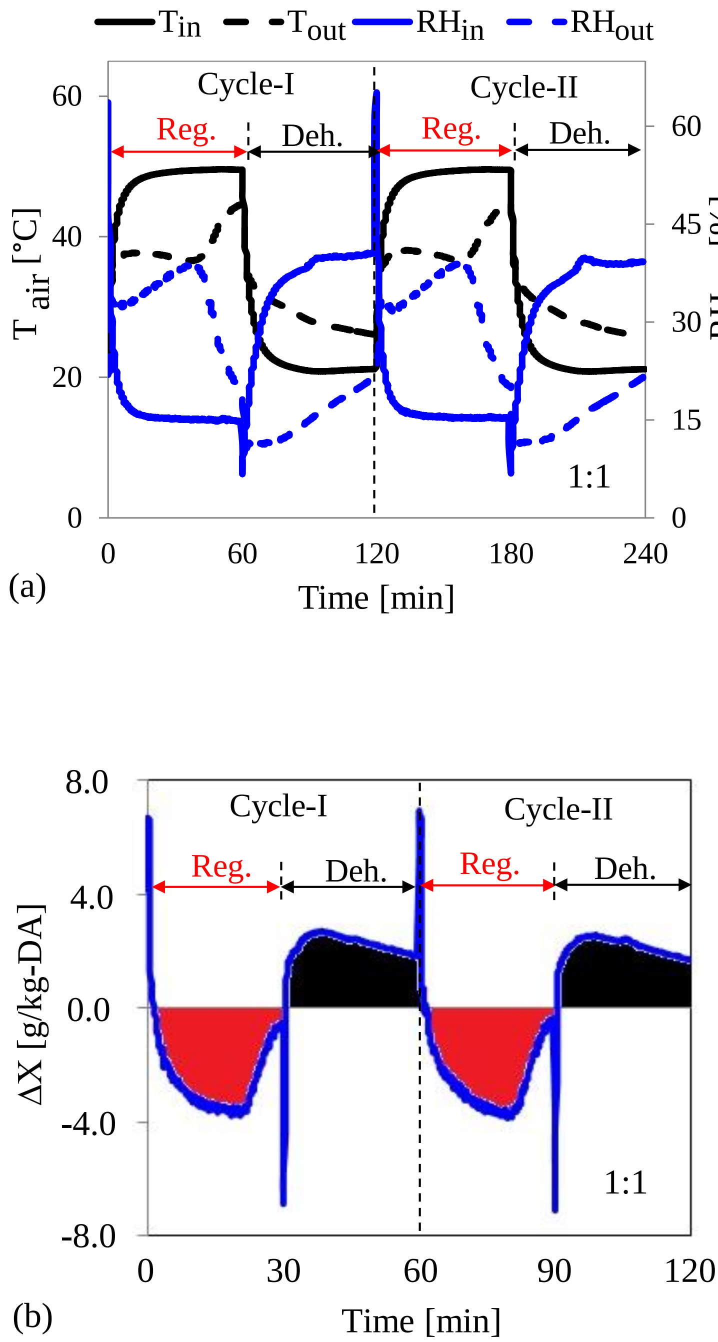
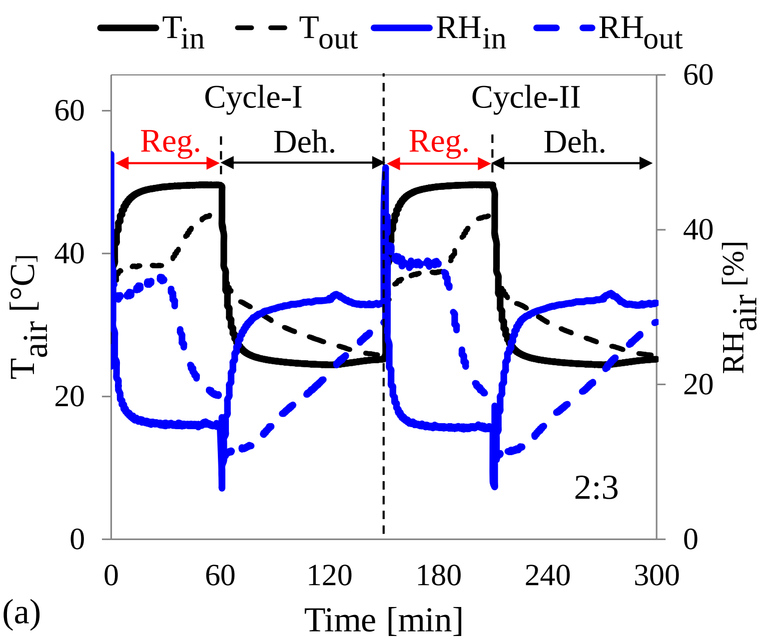
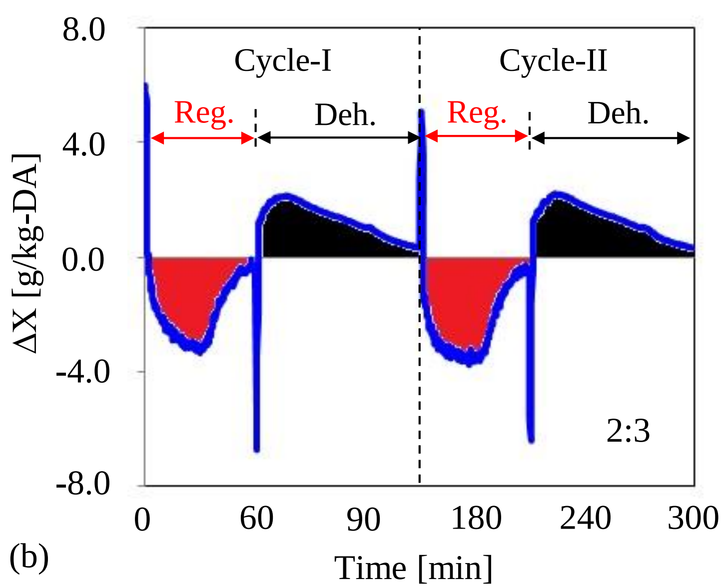


Figure 4.9 Experimental performance evaluation of desiccant unit under case-C: (a) temperature and relative humidity profiles (a) resulted desorption and adsorption profiles.

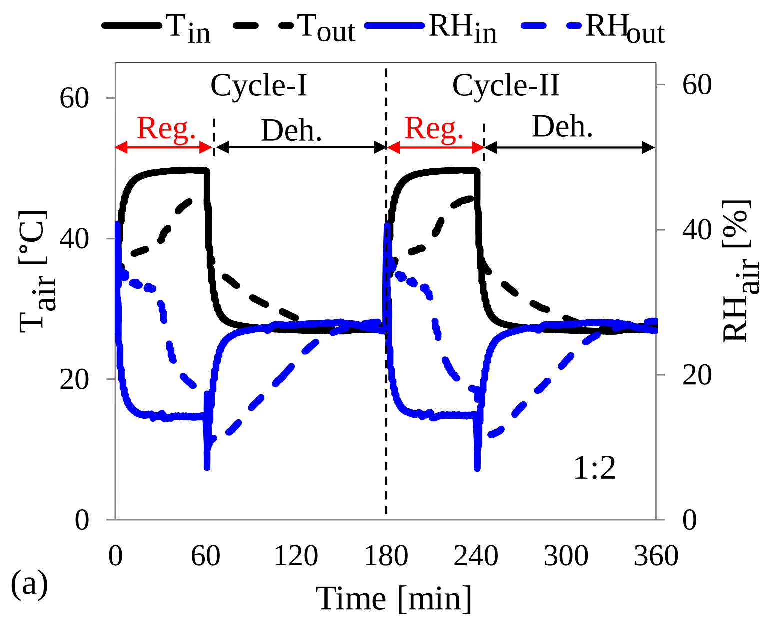


(a)



(b)

Figure 4.10 Experimental performance evaluation of desiccant unit under case-D: (a) temperature and relative humidity profiles (a) resulted desorption and adsorption profiles.



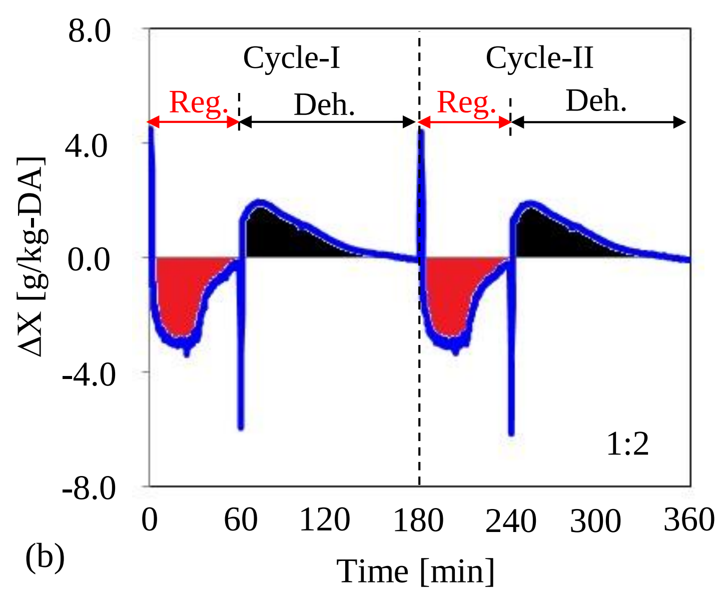


Figure 4.11 Experimental performance evaluation of desiccant unit under case-E: (a) temperature and relative humidity profiles (a) resulted desorption and adsorption profiles.

50°C - 60°C for both cases. This leads towards the optimum dehumidification performance of the desiccant blocks at low regeneration temperature (≈50°C). Therefore, the study is further extended by keeping the regeneration temperature constant (50°C) and varying the switching time ratios in order to optimize the cycle time as explained in coming paragraphs. The details of switching time are given in Table 4.5.

The experiments are further expanded for three different process air conditions (case-C, D, E) and for three switching time ratio (1:1, 2:3, 1:2) in order to optimize the switching time as explained in Table 4.5. The regeneration of desiccant blocks was made at 50°C and X_{in} = 11 g/kg-DA, respectively. The experimental results are repeated for two cycles (cycle-I and cycle-II) in order to analyze the experimental repeatability. Such repeatable performance of desiccant blocks is verified for all cases as shown by Figures 4.9-4.11. The corresponding dehumidification performance of desiccant unit is evaluated (Eq. 4.15) as shown in Figures 4.9(b)-4.11(b). Referring to Figures 4.9-4.11, it seems that desiccant blocks performed almost same dehumidification during both cycles (I and II). In addition, average dehumidification ($\overline{\Delta X}$) is calculated by Eq. (4.16) for both cycles and all cases. For example in case-C, the $\overline{\Delta X}$ of cycle-I and cycle-II is found as 2.114 g/kg-DA and 2.023 g/kg-DA, respectively (Figure 4.9). Cyclic performance is found similar or even better in other operating conditions as shown by Figure 4.10 and Figure 4.11. Therefore, it has been concluded that experimental setup successfully reproduces 95-98% of the numerical values in terms of air temperature and humidity.

Table 4.5 Experiments under varying switching time ratios and constant regeneration temperature.

Case	Process air		Regeneration air		Switching time [min]		
	T_{in} [°C]	X_{in} [g/kg-DA]	T_{in} [°C]	X_{in} [g/kg-DA]	Reg. = 60 Deh. = 60 (1:1)	Reg. = 60 Deh. = 90 (2:3)	Reg.= 60 Deh.= 120 (1:2)
C	20	6	50	11	○	○	○
D	25				○	○	○
E	28				○	○	○

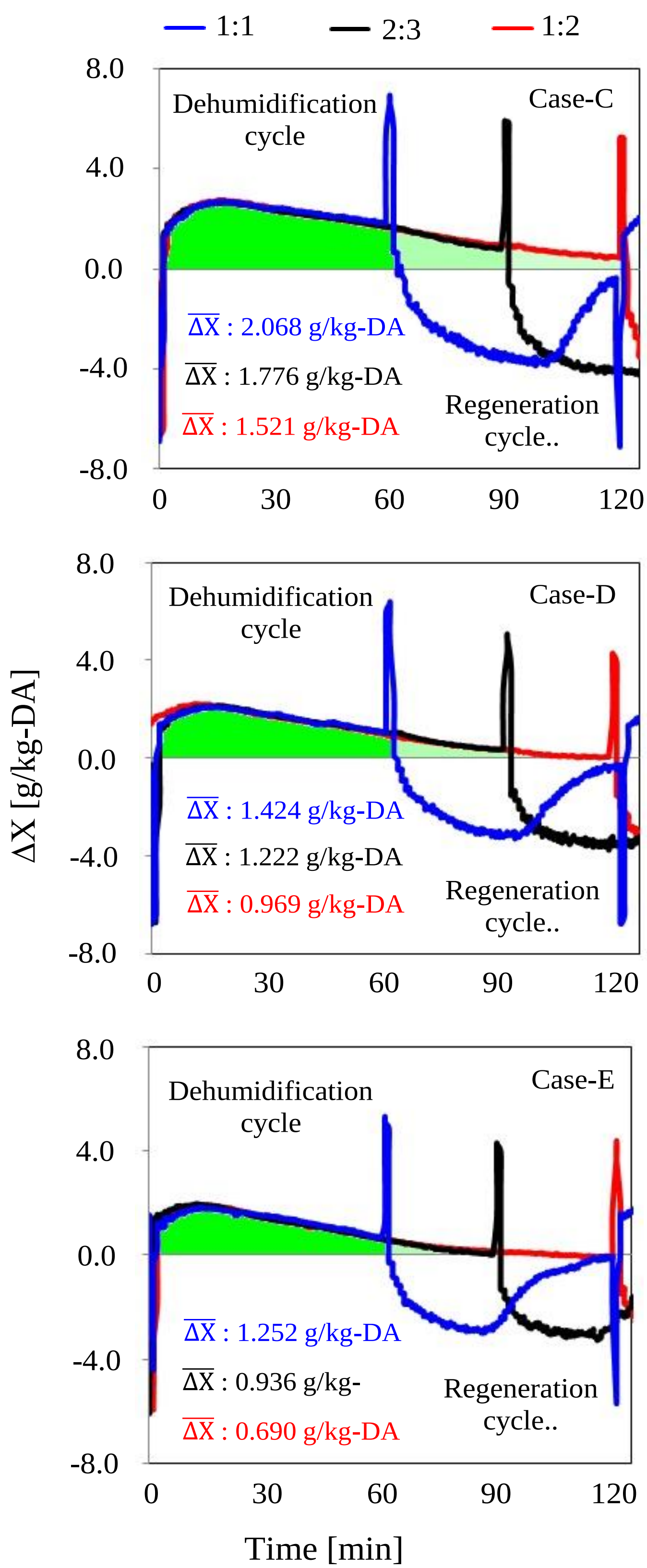


Figure 4.12 Effect of switching time ratios on net dehumidification.

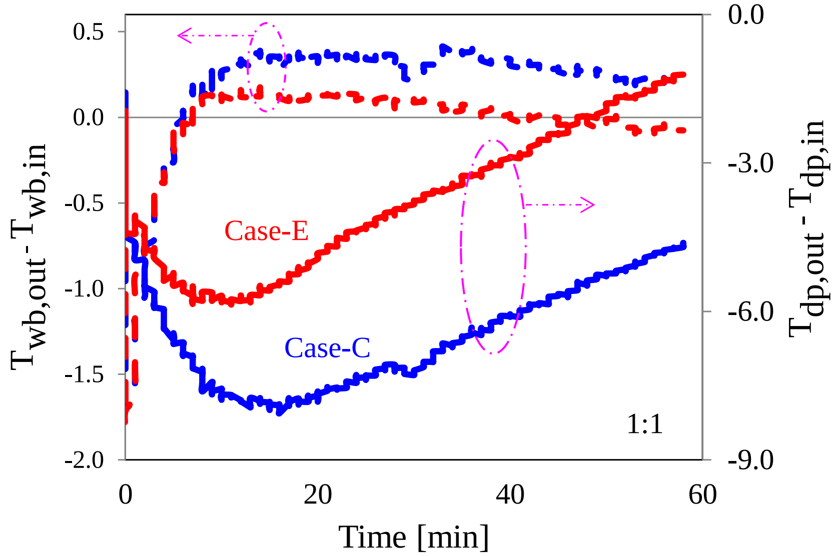


Figure 4.13 Temporal variation in wet-bulb and dew-point temperature depression.

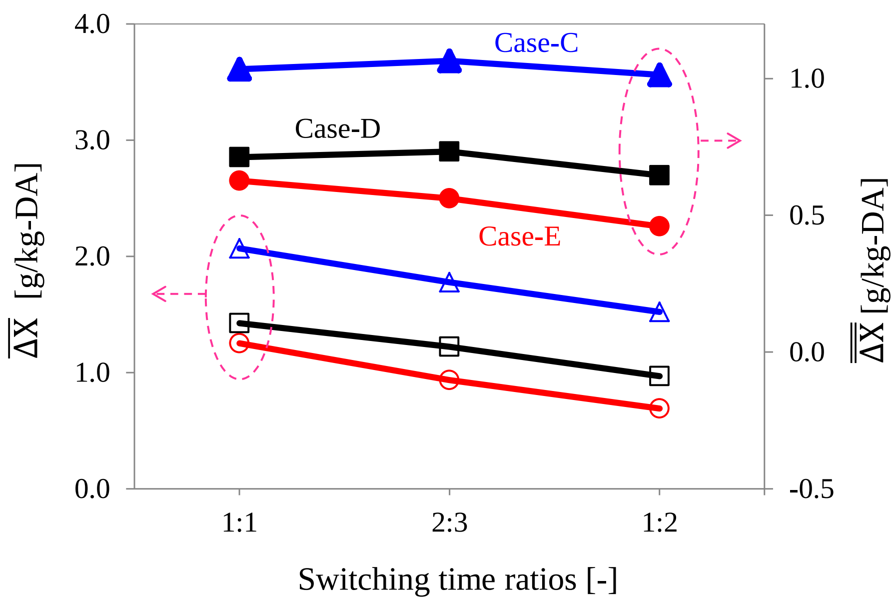


Figure 4.14 The resulted dehumidification comparison at different switching time ratios.

It is determined that the net adsorption (ΔX) per kg of air decreases with elapsed cycle time as shown in Figure 4.12. Higher ΔX was found for process air condition of case-C as compare to case-D and case-E due to possessing higher relative humidity (Figures 4.9 – 4.11). Therefore, higher wet-bulb ($T_{wb,out} - T_{wb,in}$) and dew-point temperature ($T_{dp,out} - T_{dp,in}$) differences were found for case-C throughout the cycle time as compare to case-E as shown in Figure 4.13. The sequential decrement of ΔX remained same for all process air condition and $\overline{\Delta X}$ decreases with the increase in switching time (Figure 4.12). The $\overline{\Delta X}$ decrement for all cases by increasing switching time is obvious. Therefore $\overline{\Delta X}$ is further compared with average effective dehumidification ($\overline{\overline{\Delta X}}$) which is expressed by Eq. (4.17). It is interesting to note that $\overline{\overline{\Delta X}}$ slightly increases for case-C and case-D by changing the switching time ratio from 1:1 to 2:3 and again decreases at 1:2 (Figure 4.14). Moreover, it keeps decreasing with increase in switching time ratio for case-E due to arising of equilibrium conditions. It is envisaged therefore that selection of switching time mainly depends on required level of ambient air dehumidification. In addition, it may also depend on certain AC applications, desiccant material, ambient and operating conditions, and the availability of energy source. Similarly, according to a recent study (Sultan, 2015) of activated carbon based DAC system for agricultural greenhouse AC, the longer switching time presents optimum operating condition. On the other hand, it may not guarantee the higher overall system performance. Therefore, it has been concluded from bunch of experiments that the switching time ratio of 1:2, 2:3 and 1:1 can be selected for the operation of DAC system for high, medium and low humidity operating conditions, respectively.

4.4.2 Development of Correlation

On the basis of above discussion, it is ascertained that dehumidified air conditions are crucial for the performance evaluation of desiccant unit. Once the air is passed through the desiccant unit the absolute amount of water vapors existing in the air reduces due to the water vapor adsorption onto desiccant blocks. Ultimately absolute humidity as well as relative humidity of the air at desiccant exist end decreases. On the other hand, temperature of dehumidified air increases due to water vapor adsorption heat and/or heat of adsorption. In case of ideal dehumidification process all the adsorption heat is produced due to the water vapors from the vapor phase adsorbed onto desiccant in condensed form. So, the net adsorption heat in this case should be equivalent to the heat of condensation per unit mass multiply by mass of water vapor adsorbed. If an adsorbent follows this kind of ideal adsorption phenomena then the dehumidification process line of the adsorbent on psychrometric chart should follow the isenthalpic behavior. On the other hand, it has been practically observed that the net adsorption heat is well above the ideal adsorption heat and consequently it has been reported by various studies (Sultan et al., 2017). The relationship of this kind of behavior can be expressed by the following relationship (Eq. 4.20) as described in the literature (Sultan et al., 2014).

$$\phi_{deh} = \frac{\delta}{Q_{st}} \phi_h \tag{4.20}$$

where ϕ_h and ϕ_{deh} represent the slope of enthalpy line and desiccant dehumidification line on the psychrometric chart, respectively. The δ and Q_{st} represent the heat of water vapor condensation [kJ/kg] and isosteric adsorption heat [kJ/kg], respectively. In ideal dehumidification process the Q_{st} remains zero and consequently ϕ_h follow the isenthalpic trend. However, the desiccant dehumidification process usually does not exhibit isenthalpic behaviour. The difference between ϕ_{deh} and ϕ_h depend upon the adsorbent/adsorbate pairs' interactions and nature of adsorption e.g. monolayer/multilayer etc.

In addition to the complex behavior of adsorption heat discussed above, the desiccant performance is extremely influence by the dynamic/instantaneous water vapor pressure deficit as well as thermal stresses, and it has been reported in various studies (Sultan et al., 2015, Sultan et al., 2017). Therefore, it is really complex to simulate the performance of desiccant due to extreme variation encountered conditions of process and regeneration air streams. Consequently, lot of techniques has been established for the performance evaluation of desiccant unit which can be characterized as steady-state and dynamic operations. For example, in case of steady-state the techniques/models developed by: Beccali et al., 2003; Jurinak, 1982; Koronaki et al., 2012; Panaras et al., 2010; Sultan et al., 2016b are important from subject point of view. In case of dynamics operations desiccant models presented by: Miyazaki et al. 2009, Sultan et al. 2017 are important from the view point of research conducted in our lab. In this regard this thesis mainly considers the steady-state behavior of adsorption for the performance evaluation of desiccant block. The current study explores the various scenarios of steady-state adsorption and figure out the possible options for the development of a novel and simplified correlation for desiccant

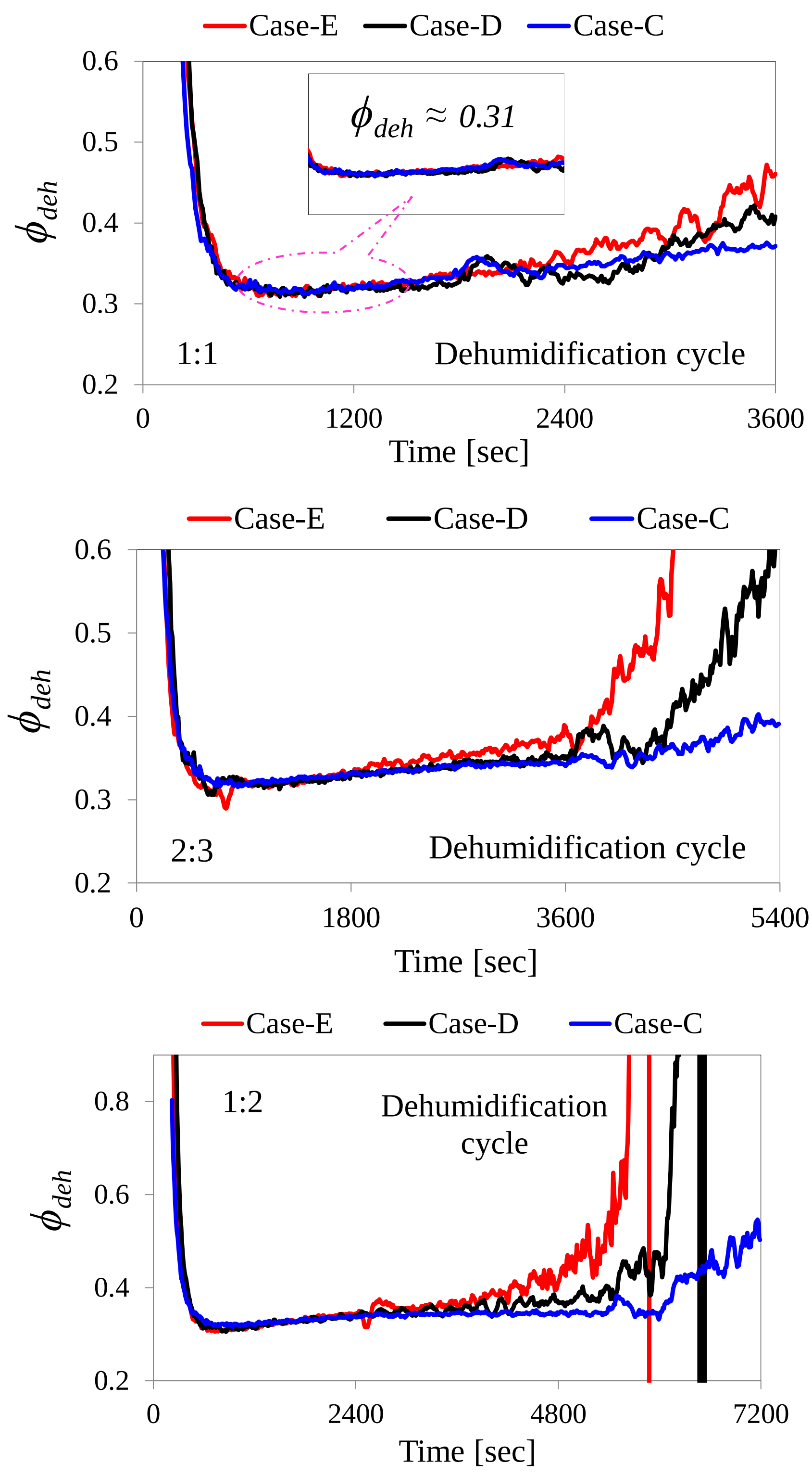


Figure 4.15 Experimental investigation of slope of dehumidification.

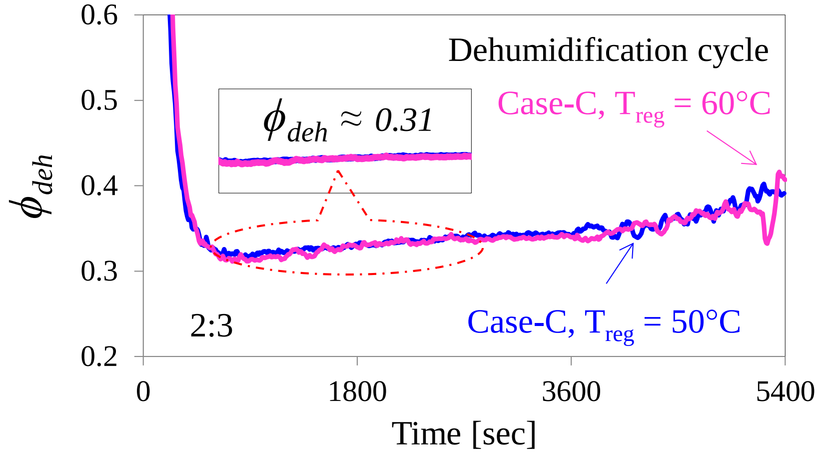


Figure 4.16 Experimental investigation of dehumidification slope at varying T_{reg}.

performance evaluation. It is important to mention here that the analyses conducted in the coming chapter (chapter 5) will be solely based on this novel and simplified correlation. The correlation was basically based on the conception of modification of isenthalpic slope of dehumidification line on psychrometric chart. Use of this conception is limited to polymeric sorbent (4.2.1) irrespective of regeneration temperatures.

On the basis of above explained conception, the slope of dehumidification line is experimentally determined for the studied polymer desiccant blocks. A simplified relationship is developed for determining the ϕ_{deh} as described by Eqs. (4.21)-(4.24)

$$\phi_{deh} = \frac{T_{out} - T_{in}}{X_{in} - X_{out}} \tag{4.21}$$

$$\phi_{deh} = \frac{T_{out} - T_{in}}{\left(\frac{h_{in} - C_p T_{in}}{h_w + C_{pv} T_{in}}\right) - \left(\frac{h_{out} - C_p T_{out}}{h_w + C_{pv} T_{out}}\right)} \tag{4.22}$$

$$\phi_{deh} = \frac{(T_{out} - T_{in})(h_w + C_{pv} T_{in})(h_w + C_{pv} T_{out})}{\left[\left(h_{in} - C_p T_{in}\right)\left(h_w + C_{pv} T_{out}\right) - \left(h_{out} - C_p T_{out}\right)\left(h_w + C_{pv} T_{in}\right)\right]} \tag{4.23}$$

$$\phi_{deh} = \frac{(T_{out} - T_{in})\left[h_w^2 + C_{pv}^2 T_{in} T_{out} + h_w C_{pv}(T_{in} + T_{out})\right]}{\left[\left(h_{in} - C_p T_{in}\right)\left(h_w + C_{pv} T_{out}\right) - \left(h_{out} - C_p T_{out}\right)\left(h_w + C_{pv} T_{in}\right)\right]} \tag{4.24}$$

where ϕ_{deh} represents the slope of real/experimental dehumidification line [-]. T_{in}, T_{out} and X_{in}, X_{out} represent the air temperature [°C] and humidity ratio [kg/kg-DA], respectively at the inlet and outlet of desiccant blocks (see Figure 4.2). h_{in} and h_{out} are the corresponding air enthalpy [kJ/kg-DA]. C_p and C_{pv} denote the specific heat of dry air [kJ/kg·K] and water vapor [kJ/kg·K], respectively whereas h_w represents the evaporation heat of water [kJ/kg].

The experimental results of ϕ_{deh} for varying process air conditions (case-C,D,E) at different switching time ratios are presented in Figure 4.15. Moreover, in order to realize the effects of increasing regeneration temperature on ϕ_{deh} the experiments are further conducted for process air conditions of case-C at T_{reg} = 60°C. The results of ϕ_{deh} are therefore compared at both the regeneration temperature (50°C and 60°C) as shown in Figure 4.16. It is ascertained that the ϕ_{deh} does not vary largely for about 3600 sec. Therefore its experimental value ($\phi_{deh} = 0.31$) can be set as constant for analyzing dehumidification performance of desiccant unit (4.2.1). Moreover, it can be concluded that variation in regeneration temperature does not vary the ϕ_{deh} as evident from Figure 4.16.

4.5 Conclusions

An open-cycle experimental apparatus was setup for the performance evaluation of hydrophilic polymer based desiccant blocks. Series of experiments are conducted for various: ambient air conditions, regeneration conditions, cycle time, and switching time. Generalized root sum of squares method is used to calculate the experimental uncertainty, and experimental data can be reproduced within 2-3% error. It is examined that when humid ambient air passes through the desiccant unit deep dehumidification occurs. It shows huge potential of air dehumidification at low regeneration temperature (50-60ºC). It is determined that under the relatively dry ambient air conditions equivalent heat of adsorption (q_{eq}) profile approaches to net value equals zero. The average effective dehumidification slightly increases under humid ambient air conditions by changing the switching time ratio from 1:1 to 2:3 due to higher process air relative humidity. However, it keeps decreasing with increase in switching time ratio for relatively dry ambient conditions. It has been found that the switching time depends on dehumidification amount, nature of application and operating conditions. From the bunch of experiments, it has been concluded that the switching time ratio of 1:2, 2:3 and 1:1 can be selected for the operation of

DAC system for high, medium and low humidity operating conditions, respectively. The desiccant dehumidification process should follow isenthalpic line on psychrometric chart in an ideal scenario when there is no adsorption heat. The slope of dehumidification line on psychrometric chart is therefore modified for the realization of real desiccant dehumidification process based on experimental data. Consequently a simplified correlation is developed by which real desiccant dehumidification process can be predicted on psychrometric chart for polymeric desiccant.

4.6 Nomenclature

$\overline{\Delta X}$	average dehumidification [g/kg-DA or kg/kg-DA]
$\overline{\overline{\Delta X}}$	effective average dehumidification [g/kg-DA or kg/kg-DA]
P_{atm}	atmospheric air pressure [Pa or kPa]
P_v	partial pressure of water vapor [Pa or kPa]
P_{vs}	saturation pressure of water vapor [Pa or kPa],
Q_{st}	heat of adsorption [kJ/kg]
R_{DA}	specific gas constant of dry air [J/kg·K or kJ/kg·K]
R_v	specific gas constant of water vapors [J/kg·K or kJ/kg·K]
$\bar{n}$	number of experimental entries during dehumidification
$\bar{\bar{n}}$	number of experimental entries of regeneration and dehumidification
q_{eq}	equivalent heat of adsorption [W]
ϕ_h	slope of enthalpy line
ϕ_{deh}	slope of dehumidification line
Δ	difference (in - out)
ΔX	net water vapors adsorption/desorption amount [g/kg-DA or kg/kg-DA]
A	area of orifice [m^2]
AC	air conditioning
C_d	discharge coefficient [-]
COP	coefficient of performance [-]
DA	dry air
DAC	desiccant air-conditioning

DBU	desiccant block unit
FCU	flow control unit
h	air enthalpy [kJ/kg-DA or J/kg-DA]
HPBS	hydrophilic organic polymer based sorbent
HX	heat exchanger
PA	process air
PAU	process air unit
RA	regeneration air
RAU	regeneration air unit
RH	relative humidity [%]
SHU	supplementary heating unit
T	temperature [°C or K]
X	humidity ratio [g/kg-DA or kg/kg-DA]
P	air pressure [Pa or kPa]
RH	relative humidity [% or -]
m	air mass flow rate [kg/sec]
δ	heat of condensation of water [kJ/kg]
ρ	air density [kg/m^3]

Subscript

ac	accuracy
DA	dry air
PA	process air
RA	regeneration air

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CHAPTER 5

OPTIMIZATION OF DESICCANT AIR-CONDITIONING SYSTEMS FOR STORAGE OF FOOD PRODUCTS

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This chapter presents the development of ideal storage zones of agricultural products, dried fruits, dried foods & feeds in comparison with greenhouse growth and humans thermal comfort zones. Six different configurations of DAC systems are proposed and their performance evaluation is made under the ambient conditions of Fukuoka-Japan. A simplified methodology is developed for performance evaluation of proposed DAC systems under varying regeneration temperatures. The parametric and thermodynamic analysis of all the system configurations (S-I to S-VI) is made to investigate that which configurations could yield better system performance. The psychrometric evaluation of three optimized DAC systems is performed for the agricultural storage and parallel applications.

5.1 Introduction

The postharvest shelf/storage life of agricultural products can be extended through cold storage, drying, chemical additives and heat treatments etc. The cold storage is considered most adopted physical treatment to combat postharvest losses and ultimately to extend the product’s storage life (Usall et al., 2016). The most of the cold storages available today are equipped with conventional vapor compression refrigeration machines. Such systems are not only degrading the environment and consuming high primary energy but also cannot provide the optimal storage conditions (temperature and relative humidity) particularly to the tropical agricultural products and dried fruits (Mahmood et al., 2016; Sultan and Miyazaki, 2017). It is because of diversified nature of agricultural products, chilling injury, and complex mechanism of transpiration, respiration, fermentation etc. In this perspective, low cost environmental friendly evaporative cooling technologies like direct evaporative cooling (DEC), indirect evaporative cooling (IEC)/M-Cycle evaporative cooling (MEC) have shown potential to provide storage conditions for particular agricultural products (Lal-Basediya et al., 2013; Sultan and Miyazaki, 2017). However, these technologies cannot be used effectively for the storage of agricultural products under largely varying ambient conditions (particularly humid) due to limited cooling performance (Lal-Basediya et al., 2013; Sultan and Miyazaki, 2017). The scope of DEC and IEC/MEC for the storage of wide range of agricultural products under varying environmental condition can be extended by the integration of desiccant dehumidification. The desiccant air-conditioning (DAC) comprises of desiccant dehumidification cum evaporative cooling has ability to deal the latent and sensible load of air-conditioning distinctly. Such ability of the DAC system makes it promising to provide optimal conditions for the storage of agricultural products without affecting their quantitative, qualitative and nutritive attributes. The approach adopted for storage of food products through DAC system can be described by self-explanatory Figure 5.1.

The present chapter experimentally investigates the combined effect of desiccant dehumidification and sensible heat exchange (via heat exchanger and/or M-Cycle evaporative cooling) for the storage of agricultural products. Six different configurations of DAC systems are proposed and their performance evaluation is made under the ambient conditions of Fukuoka-Japan. The ideal storage zones of agricultural products, dried fruits, dried foods & feeds are developed and compared with greenhouse growth and humans thermal comfort zones. Finally the psychrometric evaluation of three optimized DAC systems is made for food products storage and parallel applications.

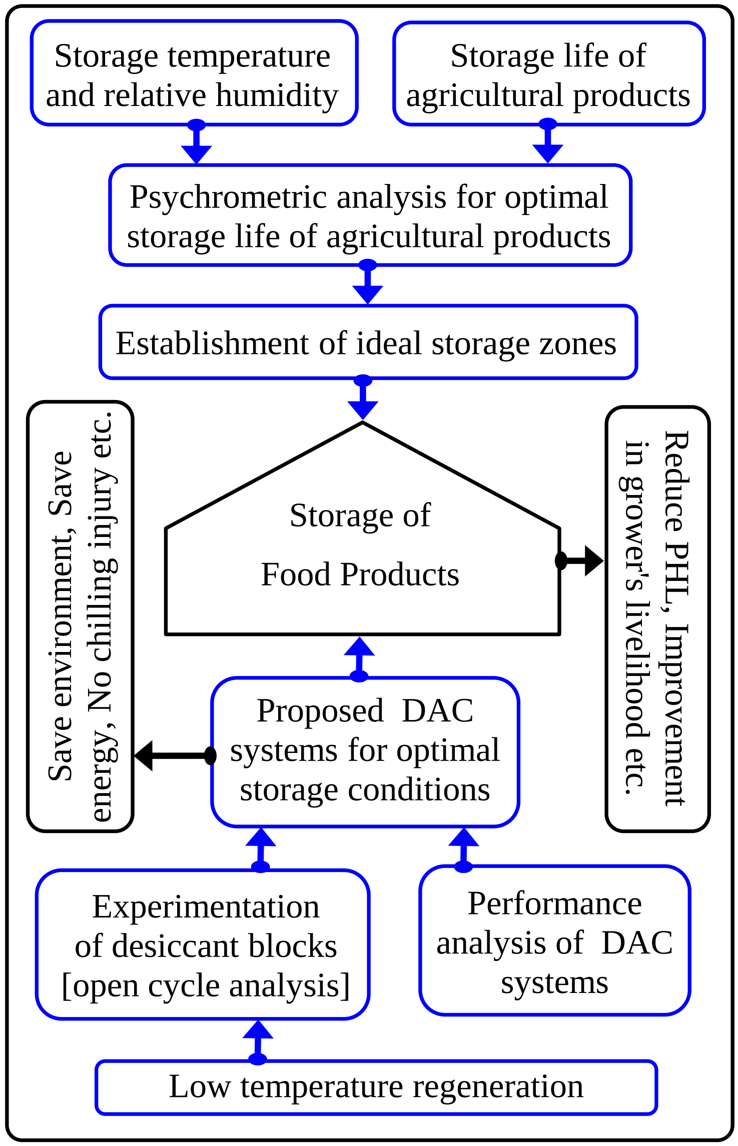

Figure 5.1 Simplified scheme of adopted approach for storage of food products.

5.2 Development of Ideal Storage Zones

It is worthy to mention that more than one crop should not be stored in the same storage room except only for short period (< 1 week) or during transportation (Thompson, et al., 1996). Camelo, 2004, stated that sharing of same storage space/room can cause differences in storage temperature and relative humidity, ethylene and chilling sensitivity, odor contamination and promotion of other quality and shelf life effecting factors.

Moreover, the highly incompatible products should not be stored in the same storage room for more than one or two days. However, compatible products storage groups have been suggested in the literature in order to optimize the storage space. These groups are developed with the assumption that the ambient ethylene concentration does not exceed 1 ppm (Thompson et al., 1996). In this regard, Thompson et al., 1996, developed three different temperature (T) and relative humidity (RH) compatible groups of fruits and vegetable for short term (7 days) storage. Tan, 1996, however recommends five different compatible groups for mixed storage of agricultural products. McGregor, 1989, on the other hand, suggested the seven different T and RH compatible groups of fruits and vegetables.

Table 5.1 Compatible storage groups (I, II and III) of the agricultural products (ASHRAE, 2010; Kitinoja and Kader, 2002).

Groups*	T [°C]	RH [%]	Products**
I	10	85-90	Cucumber, Eggplant, Okra (ladyfinger), Olive, Pepper, Potatoes-storage, Squash-summer
II	13-15	85-90	Avocado, Banana, Coconut, Ginger root, Grapefruit, Guava, Jackfruit, Lemon, Lime, Mango, Melon, Papaya, Potato-new, Pineapple, Pumpkin, Rambutan, Squash-winter, Tomatoes-ripe
III	18-21	85-90	Jicama, Sweetpotatoes, Tomatoes-mature green, Watermelon, White sapote, Yams

*Psychrometric representation of groups is given on Figure 5.2.
**For detail, refer to ASHRAE (2010) and Kitinoja and Kader (2002)

It is ascertained on the basis of above discussion that the storage compatibility of agricultural products is crucial to maintain the quantitative and qualitative attributes. In this regard, three different compatible groups of postharvest agricultural products (fruits and vegetables) according to their T and RH are established for the development of ideal storage zones on psychrometric chart as shown in Figure 5.2. The details of groups including product type, temperature and relative humidity are shown in Table 5.1 (ASHRAE, 2010; Kitinoja and Kader, 2002). Only the few products are presented in Table

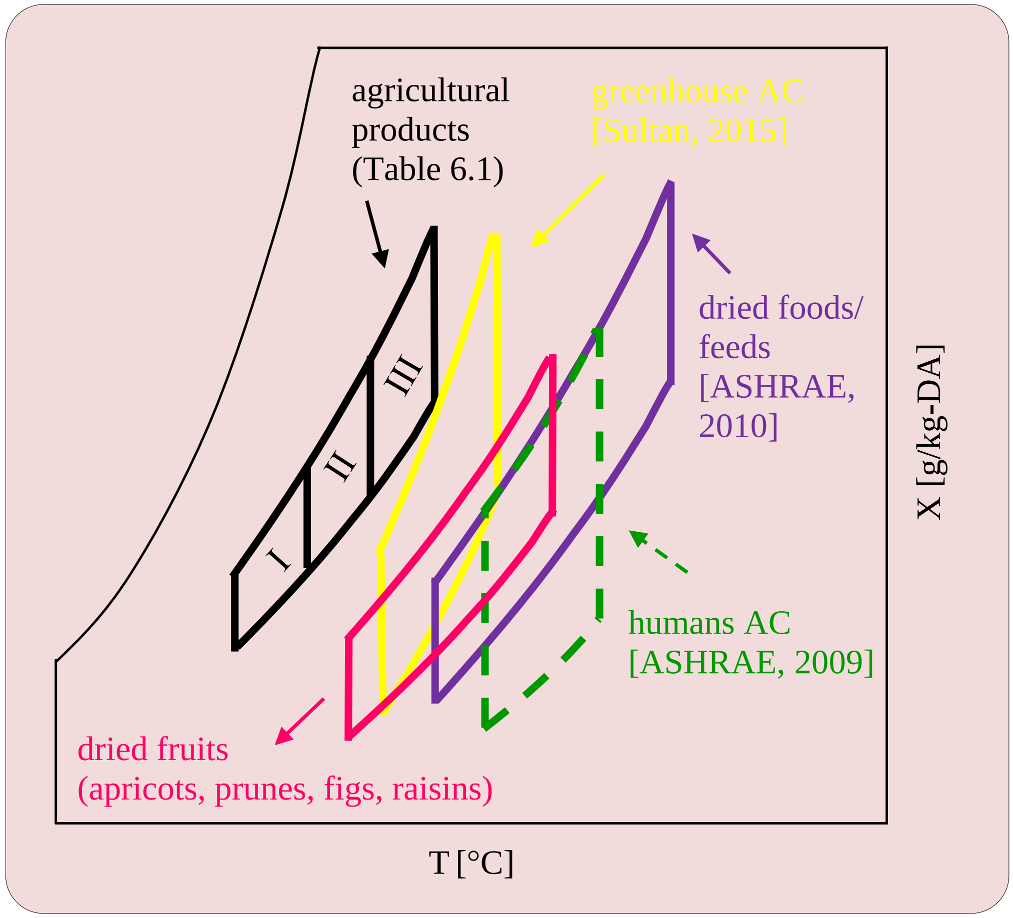

Figure 5.2 Psychrometric comparison among agricultural products, dried fruits, dried foods/feed storage, greenhouse AC and humans' AC.

Table 5.2 Recommended moisture contents of dried fruits for safe storage, shipping and buying/consumption (Dauthy, 1995).

Dried fruits	Dried forms	Moisture [%]
Apricots	Caps	17-20
Prunes	Whole	18-20
Figs	Whole	21-25
Raisins	Whole	15-19

5.1, however the details about other agricultural product and storage life can be found from the cited references. The group I and II are selected because of their sensitivity to the chilling injury, whereas group III is selected due to the requirement of slightly warmth conditions.

As far as the storage of dried agricultural products is concerned, the recommended moisture contents contained by the dried fruits (apricots, prunes, figs and raisins) with additions of ±2% in figs and raisins are given in Table 5.2 (Dauthy, 1995). Such moisture content percentage is needed for their safe storage, shipping and buying/consumption by the end users (Dauthy, 1995). In case the moisture contents of the dried fruits increases than the recommended value then chances of fungi attack will be higher. On the other hand, if the moisture contents decreases then the weight of the dried fruits will decrease tremendously and also the products will become so hard which ultimately affect the taste of the product (Pahlevanzadeh and Yazdani, 2005). In order to maintain the equilibrium moisture content of the dried fruits, the ideal T and RH zone is determined using reference (Maroulis et al., 1988) along with the Guggenheim (1966), Anderson (1946) and De-Boer (1953) (GAB) model. The GAB model can be simply represented by Eq. (5.1). It has been well described in the literature for various applications e.g. food products storage (Akanbi et al., 2006; Wolf et al., 1984) and polymer based DAC (Sultan et al., 2015) etc. It is mostly employed for analysis of moisture sorption in fruits, vegetables and meats (Andrade et al., 2011). The GAB model as improved form of Brunauer-Emmett-Teller (BET) theory (Brunauer et al., 1938) which possesses an additional constant K. All of the constants (M_{m}, C, K) of GAB model are function of adsorption temperature and have physical meaning as well. The constants along with other parameters are explained by Eqs. 5.2-5.6.

$$M = \frac{M_m a_w C K}{(1 - K a_w)(1 - K a_w + C K a_w)} \tag{5.1}$$

$$M_m = M_{\mathrm{o}}\, exp\left(\frac{l_m}{R\,T}\right) \tag{5.2}$$

$$C = C_{\mathrm{o}}\, exp\left(\frac{\Delta H_c}{R\,T}\right) \tag{5.3}$$

$$K = K_{\mathrm{o}}\, exp\left(\frac{\Delta H_k}{R\,T}\right) \tag{5.4}$$

$$\Delta H_c = H_{\mathrm{m}} - H_{\mathrm{n}} \tag{5.5}$$

$$\Delta H_k = \omega_{\mathrm{w}} - H_{\mathrm{n}} \tag{5.6}$$

where M is the moisture content of the material on dry basis [kgH_2O/kg-dry solids]; M_{m} is moisture contents on dry basis corresponding to an absorbed monolayer [kg/kg]; a_w is the water activity; C and K are constants related to the temperature effect, M_{o}, l_{m}, C_{o}, and K_{o}, are adjusted constants for the temperature effect, ΔH_{C} and ΔH_{k} are functions of the heat of sorption of water, H_{m} and H_{n} represent the heat of sorption of monolayer and multilayer respectively, ω_{w} is heat of condensation water vapor [kJ/kg], T is absolute temperature [K] and R is gas constant [kJ/kg·K] (Maroulis et al., 1988; Sultan et al., 2015).

It is important to mention that GAB equation becomes BET equation when constant K equals to unity (i.e. $C_{BET} = C_{\mathrm{GAB}}\, K$) (Sultan et al., 2015). The direct and indirect methods are used in order to estimate the constants of GAB model. In indirect method the three constants of GAB model (M_{m}, C, K) are preliminary estimated at each temperature and afterwards all the other constants (M_{o},l_{m}, C_{o}, K_{o}, H_{C} and ΔH_{k}) are estimated. Such indirect method of estimating the constants is not preferred in literature. It is because the reliability of this method is based on confidence limit and regions of preliminary estimated constant (M_{m}, C, K) and are interconnected with each other (Maroulis et al., 1988, Sultan et al., 2015). On the other hand, in direct method all GAB constant are estimated by the algebraic manipulation of Eqs. 5.1-5.6.

Though the relative humidity for storage of dried fruits at temperature 15-25°C varies for each product (apricots, prunes, figs, raisins), however it falls between 50-70%. Therefore, the representative ideal storage zone for dried fruits is developed at temperature and relative humidity of 15-25°C and 50-70%, respectively, as shown in Figure 5.2. On the other hand, as per recommendations laid in ASHRAE, 2010, the bulk storage of dried foods and feeds (like dehydrated milk and alfalfa meal) require temperature and relative humidity of about 21-32°C and 40-60%, respectively, as shown in Figure 5.2. Their (dried foods/feeds) storage life varies between three months to one year depending upon the provision of temperature and relative humidity conditions (ASHRAE, 2010). However, storage of dried fruits at 21°C enables one year life. Moreover, the further decrease in temperature results in higher storage life. But, lowering of temperature must be managed that it will not affect the recommended moisture contents of dried fruits. Otherwise, increasing and decreasing moisture contents can result in loss of quality (fungi attack) and quantity (weight), respectively. In this regard, the approximate shelf life (Γ) of food at fluctuating temperatures can be evaluated using Eq. (5.7) and Eq. (5.8) (Meurant, 2012; Toledo, 2007; Singh, 1994).

$$\Gamma_2 = \Gamma_1 \, exp\left(-\theta(T_2 - T_1)\right) \tag{5.7}$$

$$\theta = \frac{ln\, Q_{10}}{10} \tag{5.8}$$

where Γ_1 represents product known shelf life at reference temperature (T_1) and Γ_2 represents the expected product shelf life at target temperature (T_2). The factor Q_{10} is temperature coefficient. In case of reactions like nonenzymatic browning (Samaniego-Esguerra, 1989) the value of Q_{10} can be taken as 2 (Toledo, 2007), therefore same value is considered for shelf life evaluation of dried fruits. The equations (5.7 and 5.8) are solved by taking the dried fruits shelf life as 12 months at reference temperature of 21°C (Dauthy, 1995). It was found that Γ increases with decreasing storage temperature as shown in Figure 5.3. It becomes double (24 months) with 10°C reduction in temperature (i.e. 11°C).

The ideal storage zones of agricultural products (Table 5.1), dried fruits (apricots, prunes, figs, raisins), dried foods & feeds (ASHRAE, 2010) are also compared with greenhouse growth (Sultan, 2015, Sultan et al., 2016) and humans thermal comfort zones (ASHRAE, 2009) as shown in Figure 5.2. It can be noticed that low temperature and high relative humidity is required for storage of agricultural products as compare to humans thermal comfort zone.

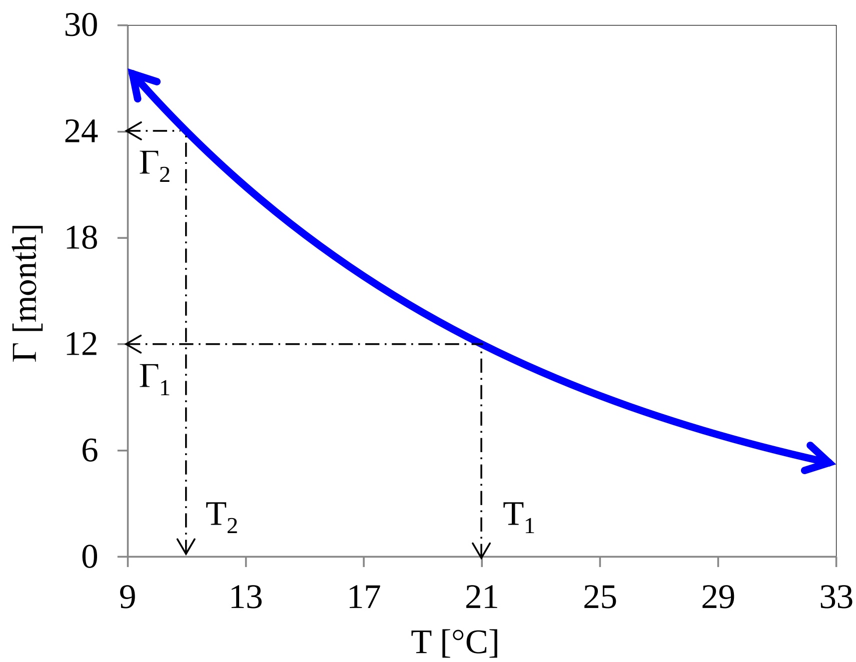

Figure 5.3 Effects of temperature on storage life of dried fruits.

5.3 Description of Proposed Systems

On the basis of desiccant block dehumidification performance analysis (chapter 4) the study is further extended by considering the incorporation of these blocks into desiccant air-conditioning (DAC) system. In this regard six different configurations of DAC systems are proposed. These system configurations are named here as System-I (S-I), System-II (S-II), System-III (S-III), System-IV (S-IV), System-V (S-V) and System-VI (S-VI). Their schematics along with psychrometric representation are shown in Figures 5.4(a)-(f).

The S-I consists of two desiccant unit, a heat exchanger (HX), a M-Cycle evaporative cooler (MEC), a direct evaporative cooler (DEC) and heat source. It is important to mention that each proposed system configuration commonly contains heat source and two desiccant units. Two units of desiccant blocks (DB) are used to enable their switching during regeneration and dehumidification. However, the other components are used alternatively as shown in the respective system configurations (Figure 5.4). It is worthy to mention that system configurations (S-I to S-IV) employ the MEC to achieve the

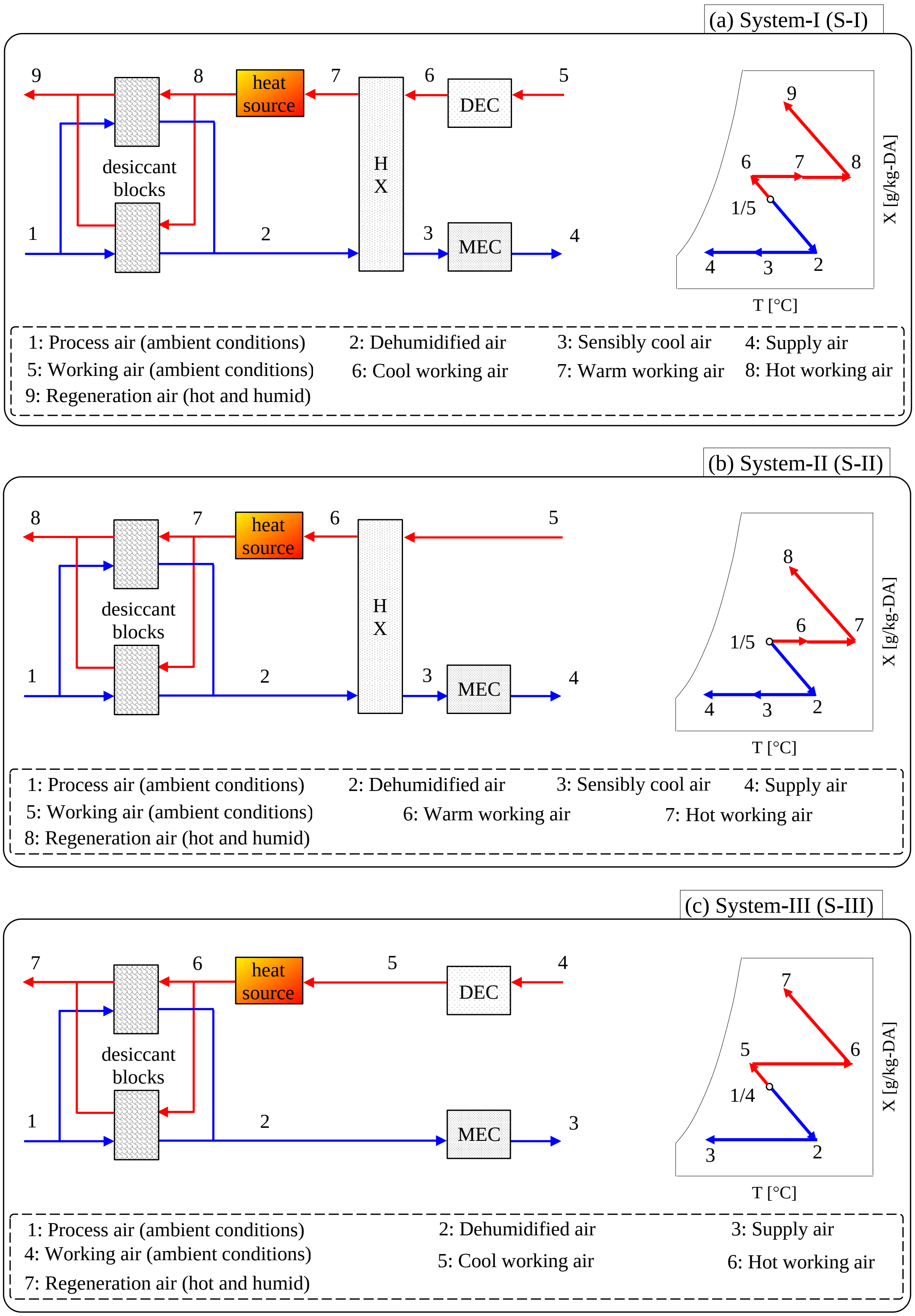

Figure 5.4 Schematics of proposed DAC systems along with psychrometric representation (a)-(f) represents system (I)-(VI), respectively.

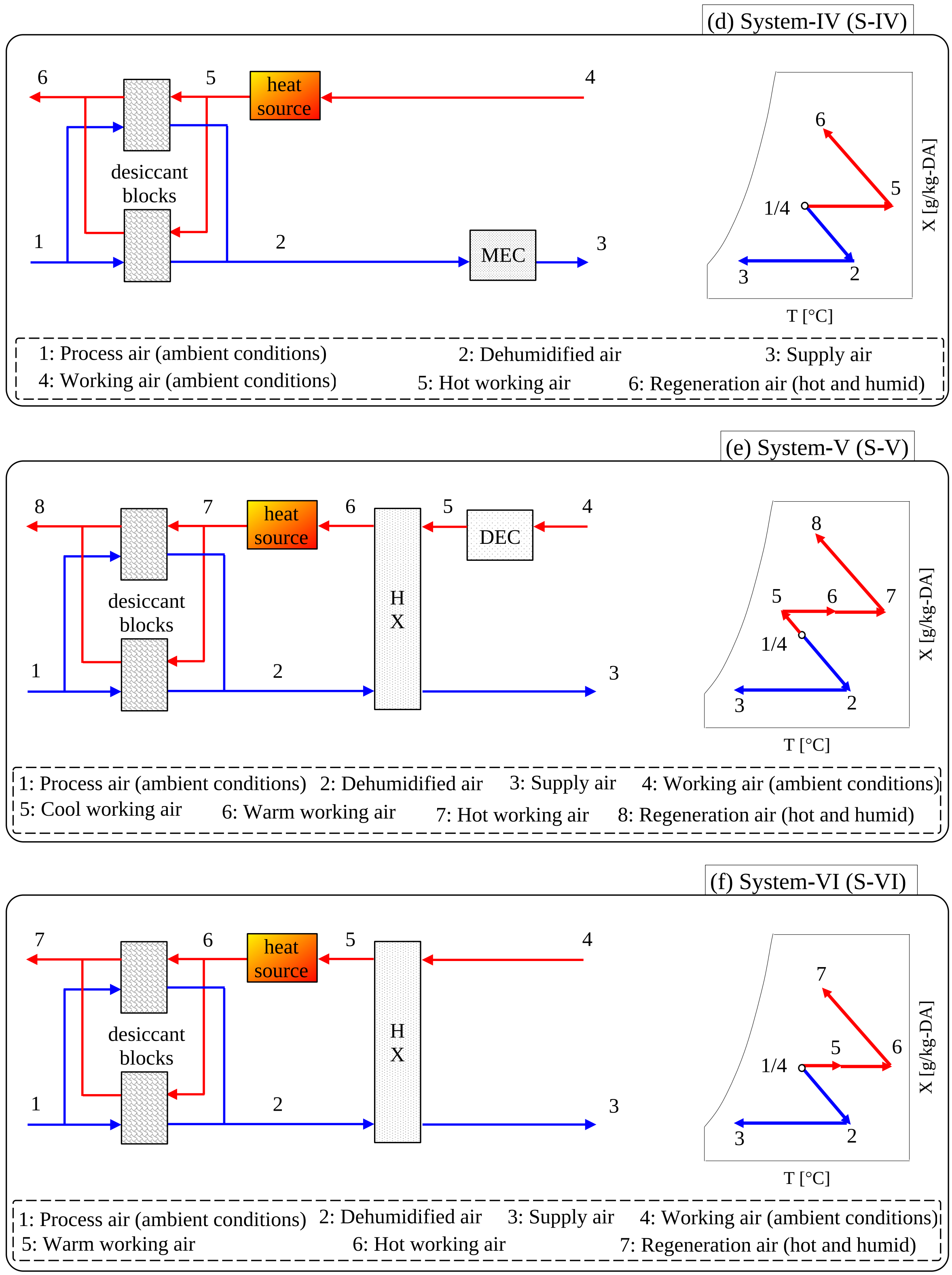

Figure 5.4 Schematics of proposed DAC systems along with psychrometric representation (a)-(f) represents system (I)-(VI), respectively.

sensible load of AC. In contrast with conventional indirect evaporative cooling methods, the MEC utilizes novel dew-point evaporative cooling conception via M-Cycle (Mahmood et al., 2016). Therefore, MEC has an ability to cool the air below its wet-bulb temperature and approaches to the corresponding dew-point. The fundamental details and principles of M-Cycle operations are explained in chapter 3. Referring to Figure 5.4(a), the working principle of the proposed system is as follow: ambient/process air (1) is dehumidified by desiccant blocks that exits at condition (2). The dehumidified air (2) is initially cooled by HX followed by MEC which exits at condition (3) and (4) respectively. The supply air (SA) at condition (4) can be used for the relevant application. For the cyclic usage the desiccant blocks are required to be regenerated. Therefore, the working air at condition (5/1) initially passes through the DEC (6) followed by HX (7) in order to recovers the heat of adsorption of the dehumidified process air. The warm working air at condition (7) is further heated by the heat source. The hot working air (8) at regeneration temperature (T_{reg}) 50-80°C passes through the desiccant blocks for their regeneration. Finally, the regeneration air (9) is exhausted to the atmosphere. The methods used for the analyses of all the system configurations are explained in next heading.

5.4 Methods and Analyses

The phenomena of desiccant block dehumidification have been explained in detail in chapter 4 wherein a simple correlation is developed, on the basis of series of experiments, in order to find out slope of dehumidification line (ϕ_{deh}) (Eqs. 4.21-4.23). The detailed methodology adopted for the analysis of proposed systems is shown in Figure 5.5. When the ambient and/or return air pass through the desiccant unit it become dehumidified consequently its temperature increases and relative humidity decreases. In this regard, it is worthy to mention that dehumidification (latent load) performance of desiccant blocks in all system configurations is evaluated by the control strategy utilizing experimentally validated ϕ_{deh} (chapter 4). Moreover, the low temperature regeneration (50-80°C) of desiccant blocks was made as shown in Figure 5.5. On the other hand, in order to achieve the sensible load of air-conditioning the dehumidified air at optimized conditions (T and RH) is passed either through HX and/or MEC depending upon the system configuration (Figure 5.4). The conditioned supply air (SA) temperature by MEC is determined through simplified correlation (Eq. 5.9) validated by published experimental data (Sultan and Miyazaki, 2017). However, the rest of air conditions are determined by fundamental thermodynamic expressions (Eqs. 5.10-5.14) according to of each system configuration. Therefore, it is worthy to mention that the system process air analyses are determined by experimentally validated correlations however the regeneration/working air analyses are made by fundamental thermodynamic expressions.

$$T_{SA/(MEC,o)PA} = 6.70 + 0.2630\, T_{(MEC,i)PA} + 0.5298\, X_{(MEC,i)PA} \tag{5.9}$$

$$T_{(HX,o)PA} = T_{(DB,o)PA} - \varepsilon_{HX}\left[T_{(DB,o)PA} - T_{(HX,i)RA}\right] \tag{5.10}$$

$$T_{(DEC,o)RA} = T_{(DEC,i)RA} - \varepsilon_{DEC}\left[T_{(DEC,i)RA} - T_{(DEC,i,wb)RA}\right] \tag{5.11}$$

$$T_{(HX,o)RA} = T_{(HX,i)RA} + \varepsilon_{HX}\left[T_{(DB,o)PA} - T_{(HX,i)RA}\right] \tag{5.12}$$

$$Q_{reg} = m_{RA}C_p\left[T_{(HS,o)RA} - T_{(HS,i)RA}\right] \tag{5.13}$$

$$COP = \left[\frac{h_{(DB,i)PA} - h_{(MEC,o)PA}}{h_{(HS,o)RA} - h_{(HS,i)RA}}\right] \tag{5.14}$$

where COP, T and X represent the thermal coefficient of performance [-], air temperature [°C] and humidity ratio [g/kg-DA], respectively. Subscript i , *o* represent process (PA) and regeneration air (*RA*) conditions at the inlet (i)/outlet (o) of desiccant block (*DB*), direct evaporative cooler (*DEC*), heat source (*HS*), heat exchanger (*HX*), M-Cycle evaporative cooler (*MEC*). ε , *h* and subscript *wb* represent the effectiveness [-], enthalpy [J/kg-DA] and wet-bulb, respectively. Q_{reg} represents the heat input [kW] through heat sources for regeneration of desiccant blocks. *m* and C_p represent the air mass flow rate [kg/sec] and specific heat capacity [kJ/kg·K], respectively.

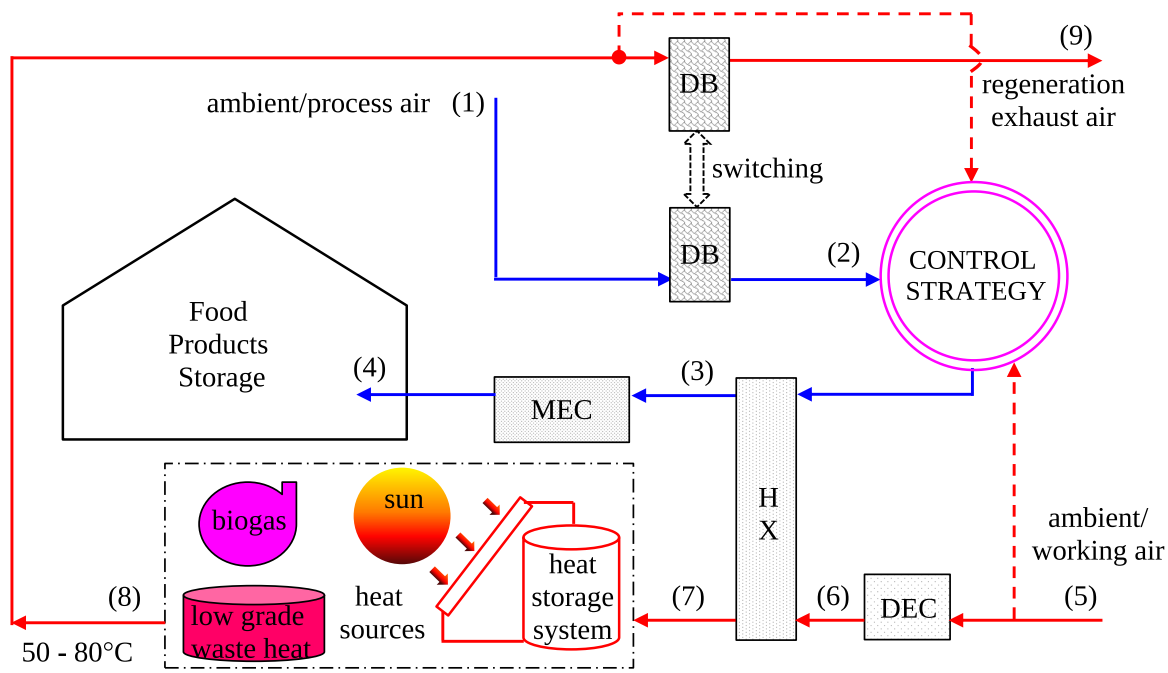

Figure 5.5 Schematics elaborated the methodology adopted for the analysis of proposed systems.

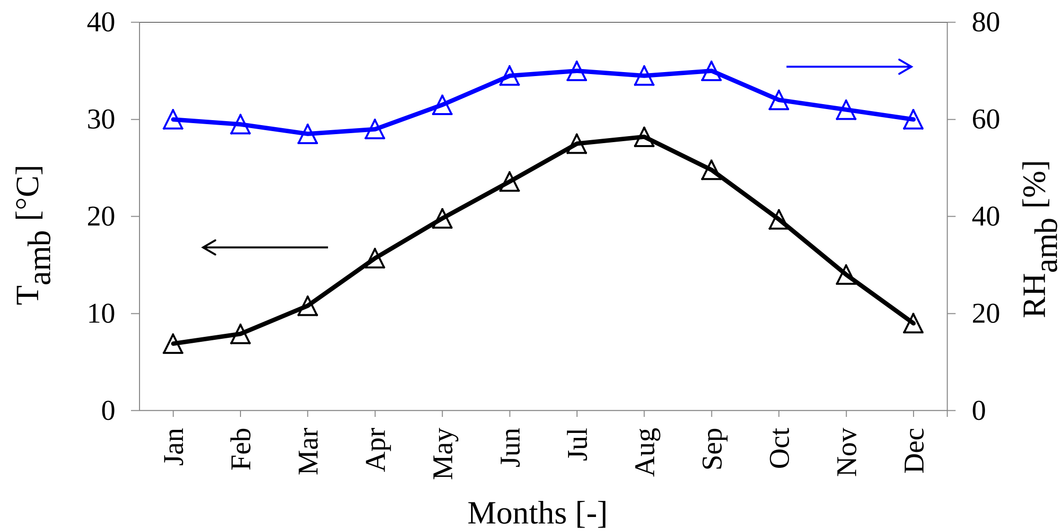

Figure 5.6 Ambient temperature and relative humidity profiles of Fukuoka, Japan.

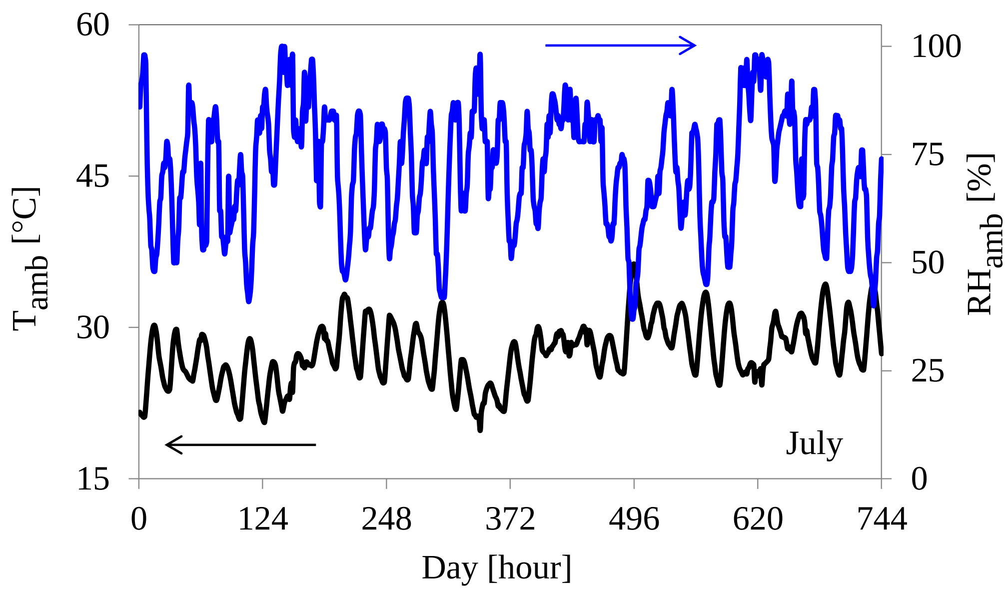

Figure 5.7 Profiles of ambient temperature and relative humidity for the month of July.

The performance evaluation of the proposed systems is carried out for the ambient conditions of Fukuoka, Japan as explained above. In this regard the data of ambient conditions (T, RH, Solar radiation) for the month of July is obtained from "Meteonorm 7"; licensed version software developed by Meteotest, Switzerland. The month of July is selected because it is one of hot and humid month throughout the year as shown in Figure 5.6. However, the daily T and RH profile for the month of July are shown in Figure 5.7. The data of T and RH (Figure 5.6 and Figure 5.7) is based on ten years (2000-2009) records; however the solar radiation data is based on twenty years (1991-2010) records.

The supply air temperature and relative humidity are determined as explained above. The average supply air temperature (T_{SA}) and relative humidity (RH_{SA}) are then determined for the month of July. The net and average dehumidification (ΔX and $\overline{\Delta X}$) is determined by Eq. 4.15 and 4.16, respectively as explained in chapter 4. However, in calculating the $\overline{\Delta X}$ the parameter $\bar{n}$ here represents the number of ambient conditions entries for the month of July. Similarly, the thermal COP of all DAC systems is calculated by the respective equation as depicted on their schematics (Figure 5.4). Moreover, wet-bulb

(ε_{wb}) and dew-point (ε_{dp}) effectiveness of MEC integrated in DAC system is determined by Eq. 3.1 and Eq. 3.2 respectively, as elaborated in chapter 3. The heat input through heat sources for regeneration of desiccant blocks (Q_{reg}) during the month of July is calculated by fundamental thermodynamic expression (Eq. 5.13) (Mahmood et al. 2016). The results and discussion about the performance evaluation of proposed systems is explained in next heading.

5.5 Results and Discussion

5.5.1 Performance Evaluation of System-I and II

In system configurations (S-I and S-II), the supply air temperature decreases with increasing regeneration temperature. It is determined that the T_{SA} of S-I and S-II decreases from 20.01-17.93°C and 20.74-18.68°C, respectively by increasing the T_{reg} from 50°C to 80°C. The corresponding RH_{SA} also decreases as shown in Figure 5.8. Hence the cooling capacity of S-I is relatively slightly higher than S-II. It is due to the supply of cool working air, through DEC, to the HX of S-I during the regeneration of desiccant block. On the other hand, it is found that the desiccant blocks in system configuration of S-II enable comparatively higher net dehumidification than S-I at identical regeneration temperature as shown in Figure 5.9. It is because of configuration of S-II which permits the supply of working air at lower humidity ratio than S-I. Therefore, the average dehumidification of S-I and S-II at T_{reg} of 50-80°C is determined as 4.26-8.94 g/kg-DA and 4.58-9.20 g/kg-DA, respectively. It is worthy to mention that S-II performs higher dehumidification but required less heat input for regeneration (Q_{reg}) desiccant blocks as compare to S-I (Figure 5.10). The Q_{reg} for S-II was determined as 688 kW and 1934 kW at regeneration temperature of 50°C and 80°C respectively. Whereas, it increases correspondingly for S-I as 785 kW and 2018 kW.

The thermal COP of S-II remained higher than S-I at T_{reg} = 50-80°C. However, it is found that the relative difference between thermal COP of S-I and S-II decreases with increasing regeneration temperature. Similarly, the corresponding Q_{reg} tends to decrease as shown in Figure 5.10.

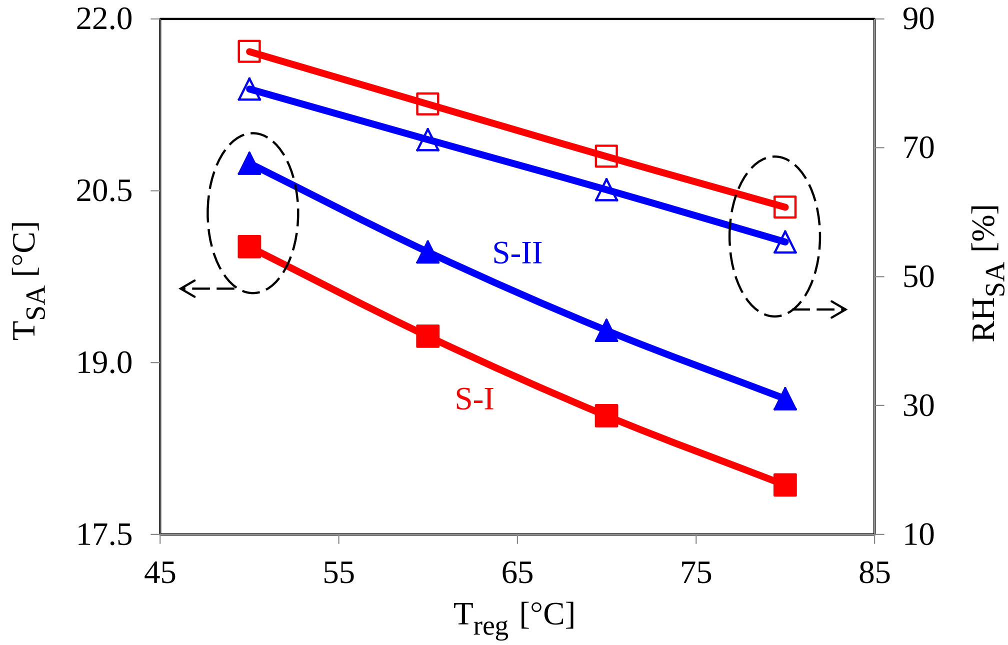

Figure 5.8 Effect of regeneration temperature on supply air temperature and relative humidity of S-I and S-II.

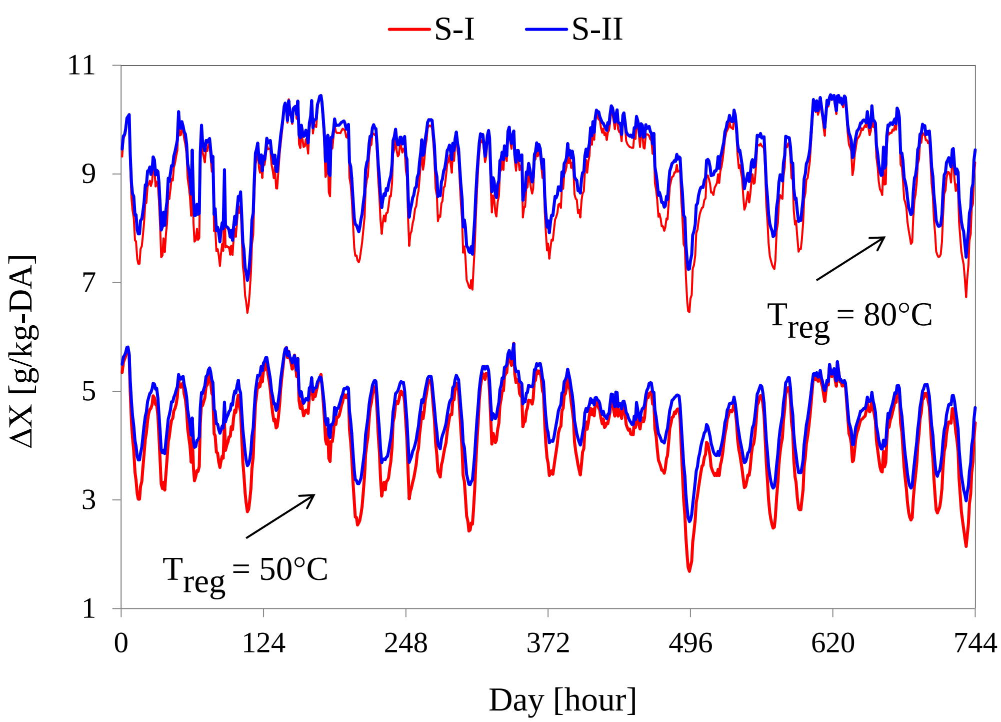

Figure 5.9 Net dehumidification performance profiles of S-I and S-II under varying ambient conditions.

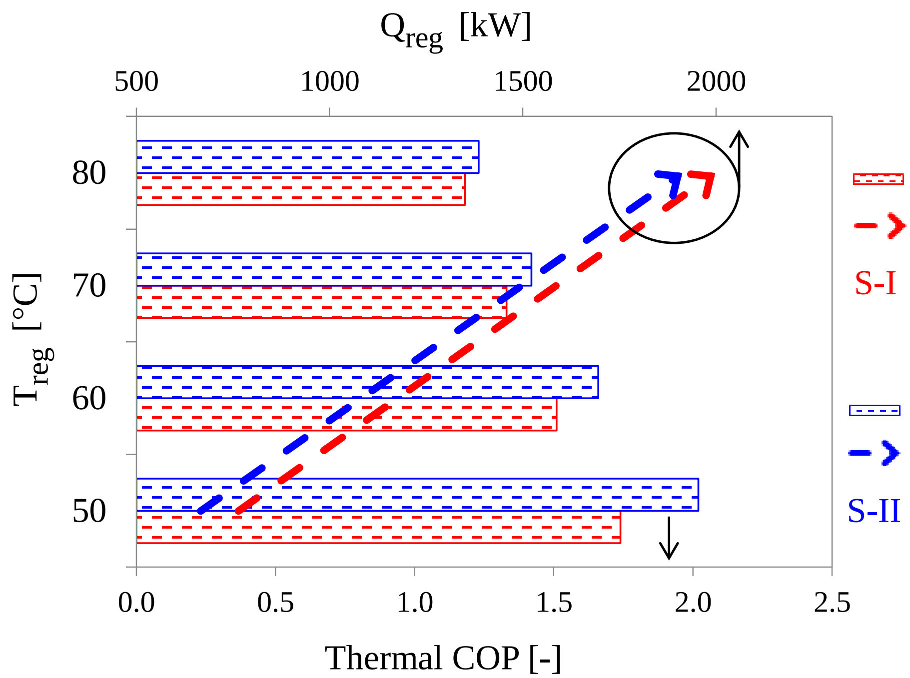

Figure 5.10 Effect of regeneration temperature on system's (S-I, S-II) thermal COP and heat input.

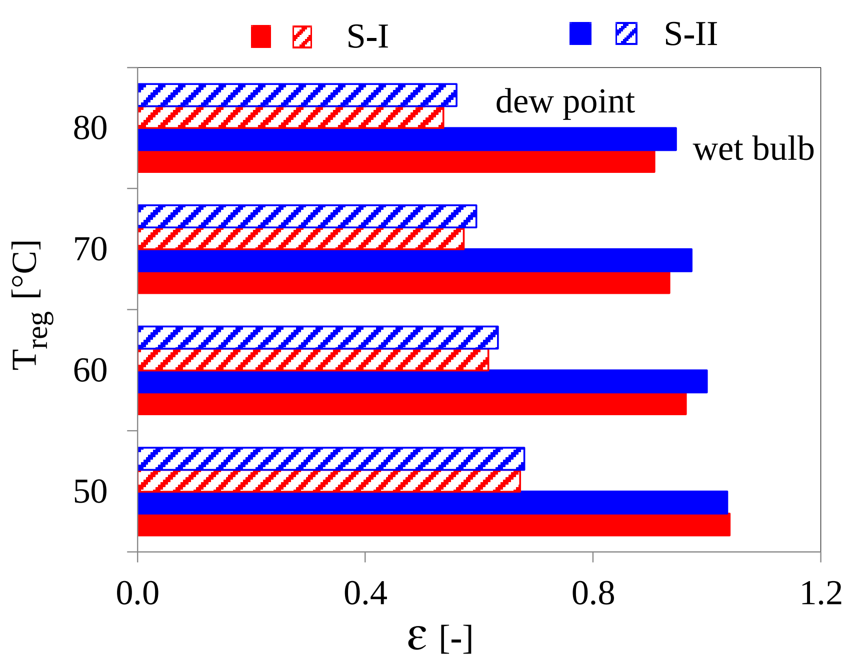

Figure 5.11 Dew-point and wet-bulb effectiveness comparison of MEC integrated in S-I and S-II.

The wet-bulb effectiveness (ε_{wb}) of MEC integrated in S-I and S-II was found almost same at T_{reg} = 50°C. Accordingly, the corresponding dew-point effectiveness (ε_{dp}) also remains same. However, both the effectiveness starts decreasing by increasing the regeneration temperature from 60°C to 80°C. Moreover, such decrement remained higher in S-I than S-II that means the ε_{wb} and ε_{dp} of S-II becomes higher than S-I at identical elevated T_{reg} (60-80°C) as shown in Figure 5.11.

On the basis of above prospective, it can be concluded that the system configuration (S-II) is more suitable due to the requirement of less regeneration heat input, higher dehumidification, higher wet-bulb and dew-point effectiveness, and almost same cooling capacity as of S-I. The S-II also does not involve the additional cost of DEC. Moreover, the supply of higher regeneration heat input in S-I incur variable cost.

5.5.2 Performance Evaluation of System-III and IV

In contrast to S-I and S-II, the supply air temperature increases with increasing regeneration temperature in S-III and S-IV. However, it is determined that T_{SA} (also RH_{SA}) remains same for both S-III and S-IV at identical T_{reg} as shown in Figure 5.12. It means that cooling capacity of both systems remains almost same. Therefore, such system configurations can be potentially operated at low regeneration temperature to achieve the less sensible load of AC. However, as far the latent load is concerned the dehumidification of desiccant blocks of S-III and S-IV increases with increasing regeneration temperature as shown in Figure 5.13. The dehumidification performance of S-III and S-IV was found consistent with S-I and S-II, respectively. It is due to systems' design which facilitates supply of working air at identical humidity ratio during the regeneration of desiccant blocks. However, the heat energy required for regeneration of desiccant blocks of S-III and S-IV is determined about two times greater than of S-I and S-II in order to get the same level of dehumidification. Such higher regeneration heat input is due to the elimination of heat exchanger in system configurations of S-III and S-IV. Therefore, the Q_{reg} for S-IV was calculated as 1683 kW and 3929 kW at regeneration temperature of 50°C and 80°C respectively. Whereas, it increases correspondingly for S-III as 1959 kW and 4205 kW. It is important to note that despite of utilizing less Q_{reg} (Figure 5.14), the S-IV enables higher dehumidification than S-III at same regeneration temperature (Figure 5.13).

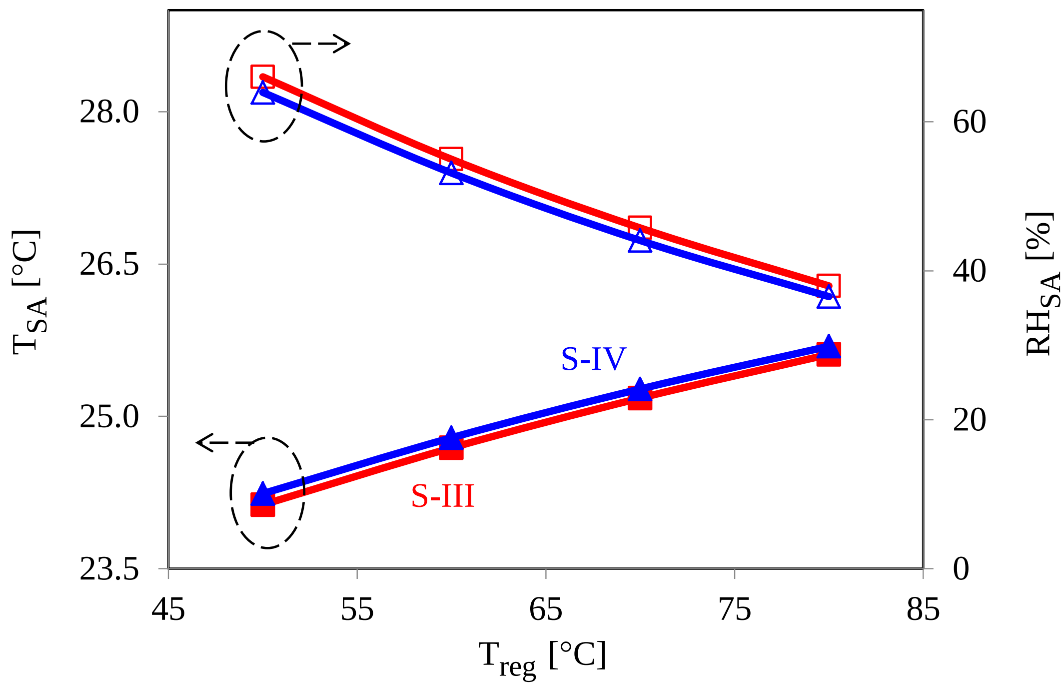

Figure 5.12 Effect of regeneration temperature on supply air temperature and relative humidity of S-III and S-IV.

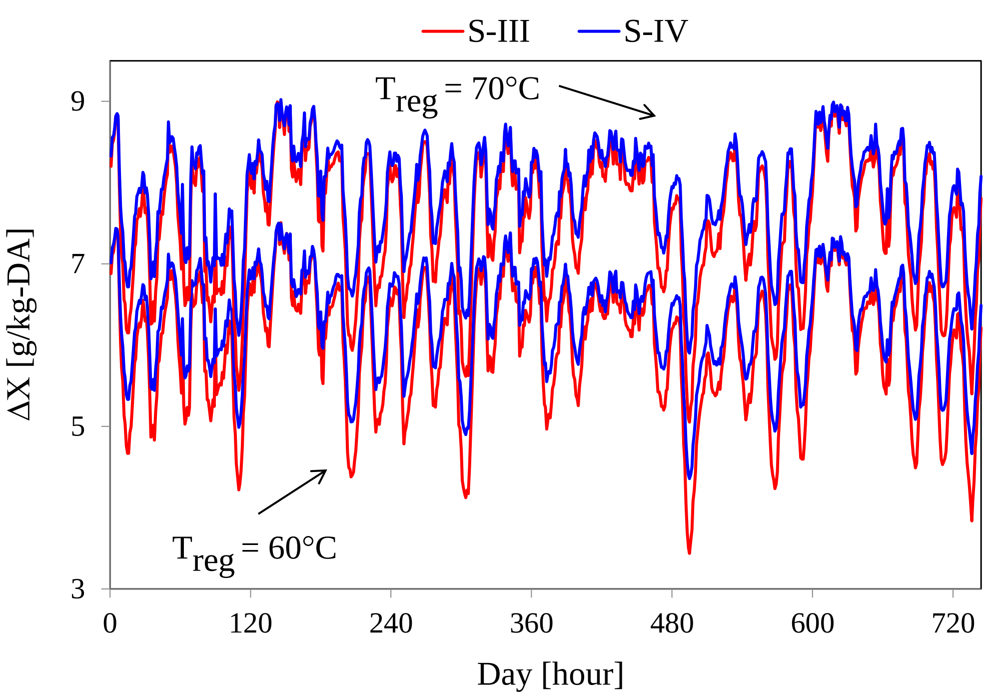

Figure 5.13 Net dehumidification performance profiles of S-III and S-IV under varying ambient conditions.

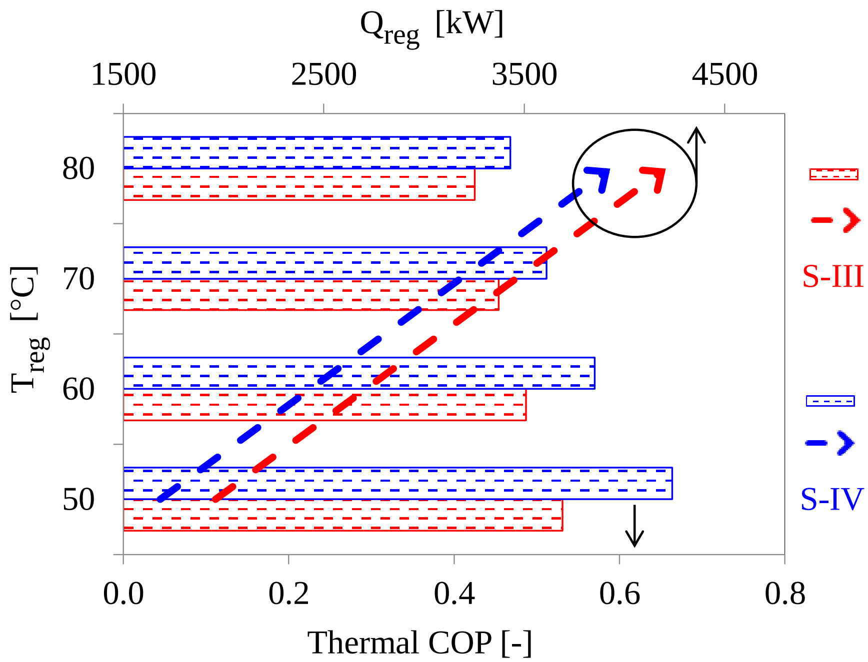

Figure 5.14 Effect of regeneration temperature on system's thermal COP and heat input.

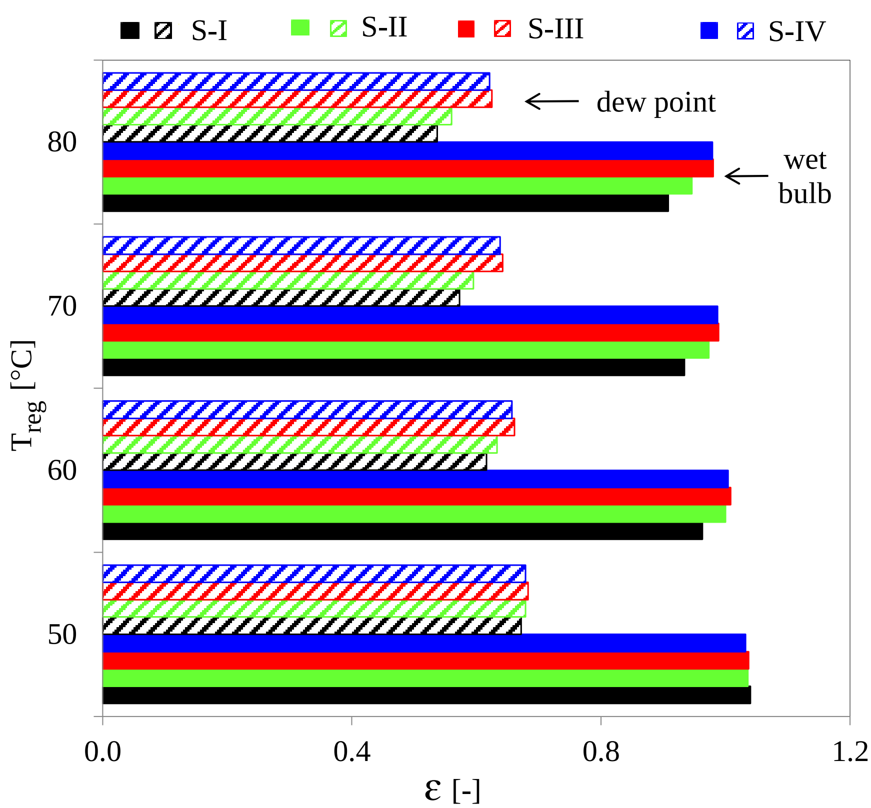

Figure 5.15 Dew-point and wet-bulb effectiveness comparison of MEC integrated in S-I, S-II, S-III and S-IV.

Alike in S-I and S-II, the thermal COP of S-III and S-IV decreases with increasing regeneration temperature. However, the COP of S-III and S-IV was found much lesser than S-I and S-II due to higher Q_{reg}. Moreover, the COP of S-III remained higher than S-IV even at higher regeneration temperature (80°C) as shown in Figure. 5.14.

The wet-bulb effectiveness (ε_{wb}) of MEC integrated in S-III and S-IV was found almost same at the corresponding regeneration temperatures (50-80°C). Accordingly, the corresponding dew-point effectiveness (ε_{dp}) also remains same. However, both the effectiveness was found comparatively higher than S-I and S-II at elevated T_{reg} (60-80°C) as shown in Figure 5.15. Thus ε_{wb} and ε_{dp} of both systems (S-III and S-IV) is determined about 0.98 and 0.63, respectively at T_{reg} = 80°C.

It is ascertained from the above discussion that the system configuration (S-IV) is more suitable due to the requirement of less regeneration heat energy, higher dehumidification, almost same cooling capacity, and wet-bulb and dew-point effectiveness as of S-III. Alike in S-II, the S-IV also does not involve the additional cost of DEC. Moreover, the supply of higher regeneration heat in S-III incurs variable cost.

5.5.3 Performance Evaluation of System-V and VI

Alike in S-III and S-IV, the supply air temperature increases with increasing regeneration temperature in S-V and S-VI. However, in contrast to S-I and S-II, it is determined that T_{SA} of S-V and S-VI increases as 25.57-27.07°C and 29.0-30.47°C, respectively at T_{reg} = 50-80°C as shown in Figure 5.16. Though the cooling capacity of S-V is higher than S-VI but much lesser than rest of the studied systems. Similar to other system configurations (S-I, S-II and S-III, S-IV) the S-VI performs relatively higher dehumidification than S-V. The average dehumidification ($\overline{\Delta X}$) of all the system configurations is determined at varying regeneration temperature in order to compare their dehumidification performance. It is ascertained that S-I, S-III and S-V enable similar $\overline{\Delta X}$ but higher than the S-II, S-IV and S-VI at corresponding regeneration temperature as shown in Figure 5.17. Moreover, $\overline{\Delta X}$ of all systems configuration also increases with increasing regeneration temperature.

The Q_{reg} required for regeneration of desiccant blocks of S-V and S-VI is determined about two times lesser than of S-III and S-IV in order to get the same level of

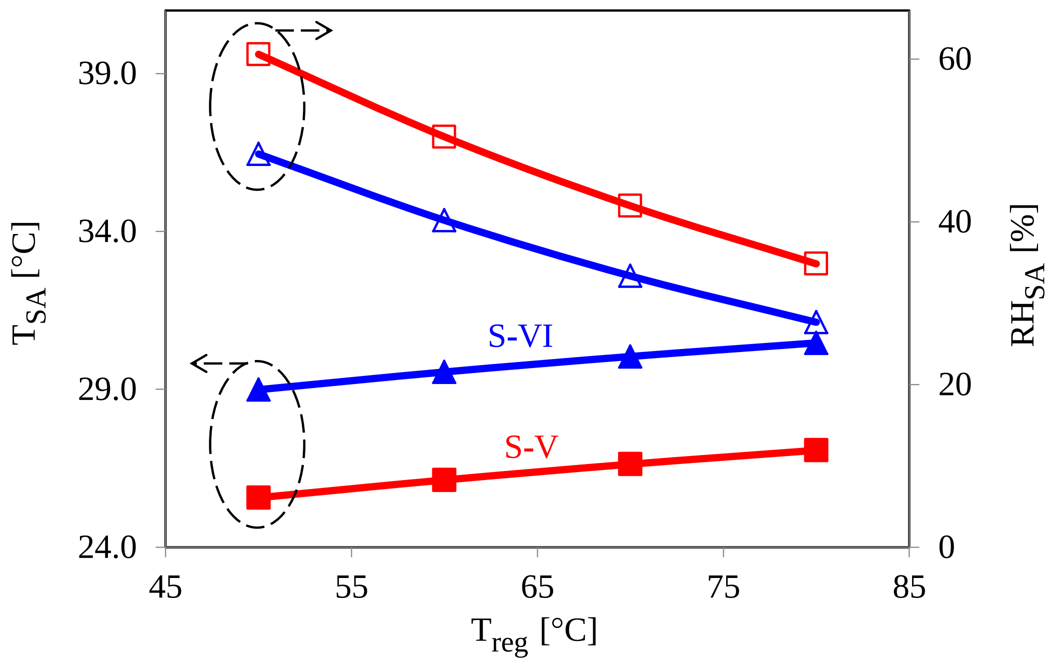

Figure 5.16 Effect of regeneration temperature on supply air temperature and relative humidity of S-V and S-VI.

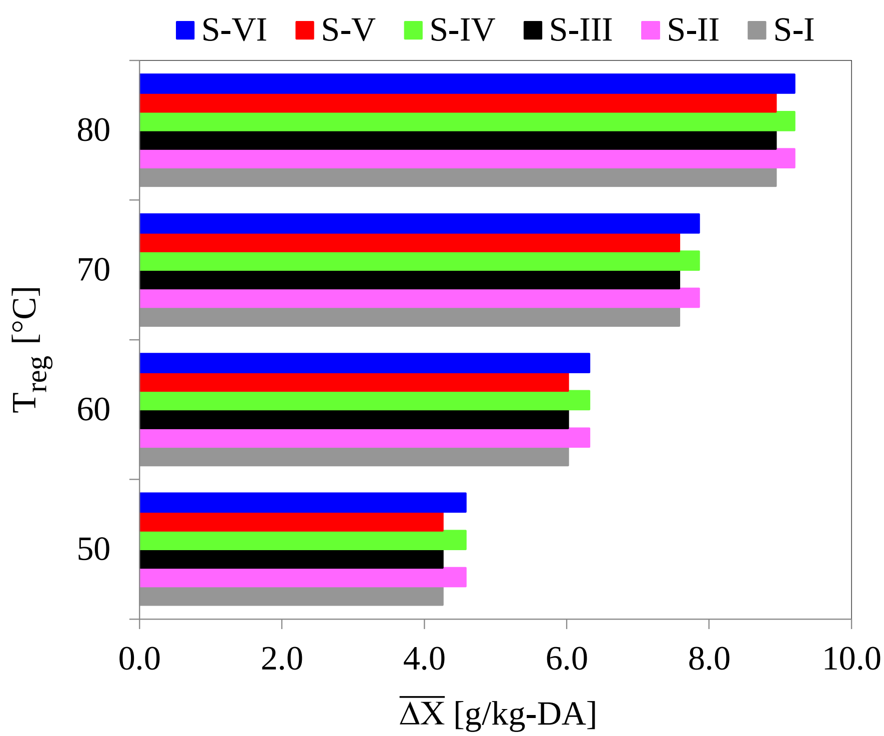

Figure 5.17 Comparison of net dehumidification performance of all proposed systems.

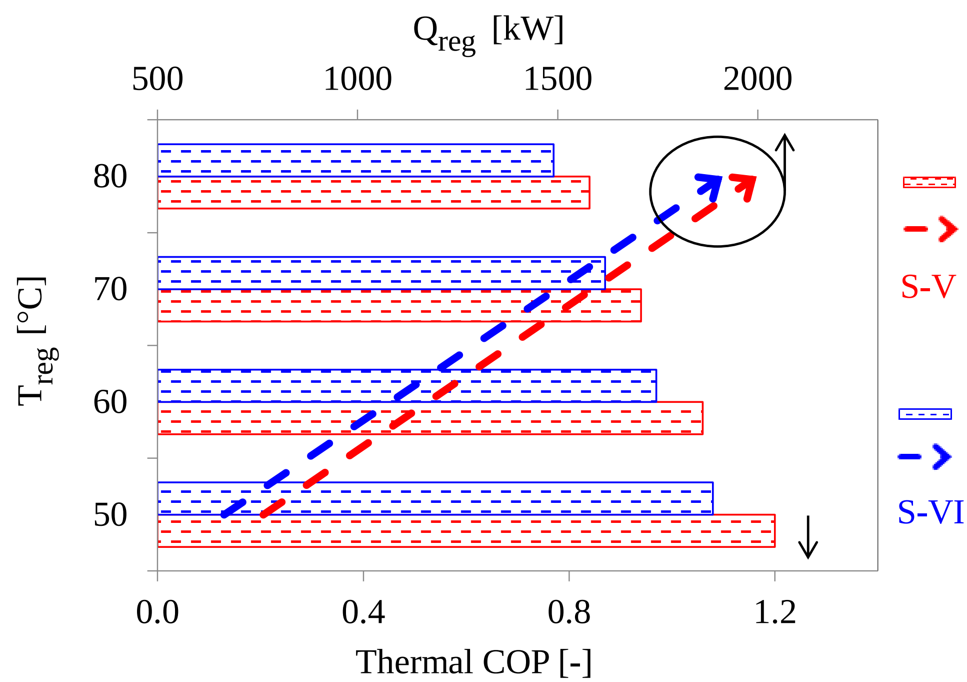

Figure 5.18 Effect of regeneration temperature on system's (S-V, S-VI) thermal COP and heat input.

dehumidification. It means these systems (S-V and S-VI) required the same regeneration heat input as of S-I and S-II. The thermal COP of S-V and S-VI was found lower than S-I and S-II at same Q_{reg} and regeneration temperature (Figure 5.18). However, both the systems (S-V and S-VI) enable about two times higher COP than S-III and S-IV by utilizing two times lower Q_{reg} (see Figure 5.14 and Figure 5.18). Moreover, the COP of S-V in comparison to S-VI remained high even by increasing the regeneration temperature from 50°C to 80°C. It is ascertained that S-V can be better system configuration because of comparatively higher cooling capacity, RH_{SA} and COP than S-IV though it (S-V) require relatively higher Q_{reg}.

It is established on the basis of parametric and thermodynamic analysis of all the system configurations (S-I to S-VI) that S-II, S-IV and S-V could yield better system performance. Therefore, these system configurations are only considered for further psychrometric analysis in order to find out the appropriate system/technology for the storage of agricultural products and other parallel applications as explained in next heading.

5.5.4 Storage Systems

The performance evaluation of studied system configurations (S-I to S-VI) revealed that S-II, S-IV, and S-V could yield better system performance. In this regard supply air conditions of these systems against the ambient conditions of Fukuoka, Japan are determined at regeneration temperature of 50-80°C as explained in heading 5.4. Accordingly, psychrometric evaluation of proposed systems (S-II, S-IV, and S-V) is made for insights about the storage of agricultural products (Table 5.1), dried fruits (apricots, prunes, figs, raisins) and dried foods & feeds (ASHRAE, 2010) in comparison with zone of greenhouse growth (Sultan, 2015; Sultan et al., 2016) and human thermal comfort (ASHRAE, 2009). The development of such zones is explained in heading 5.2.

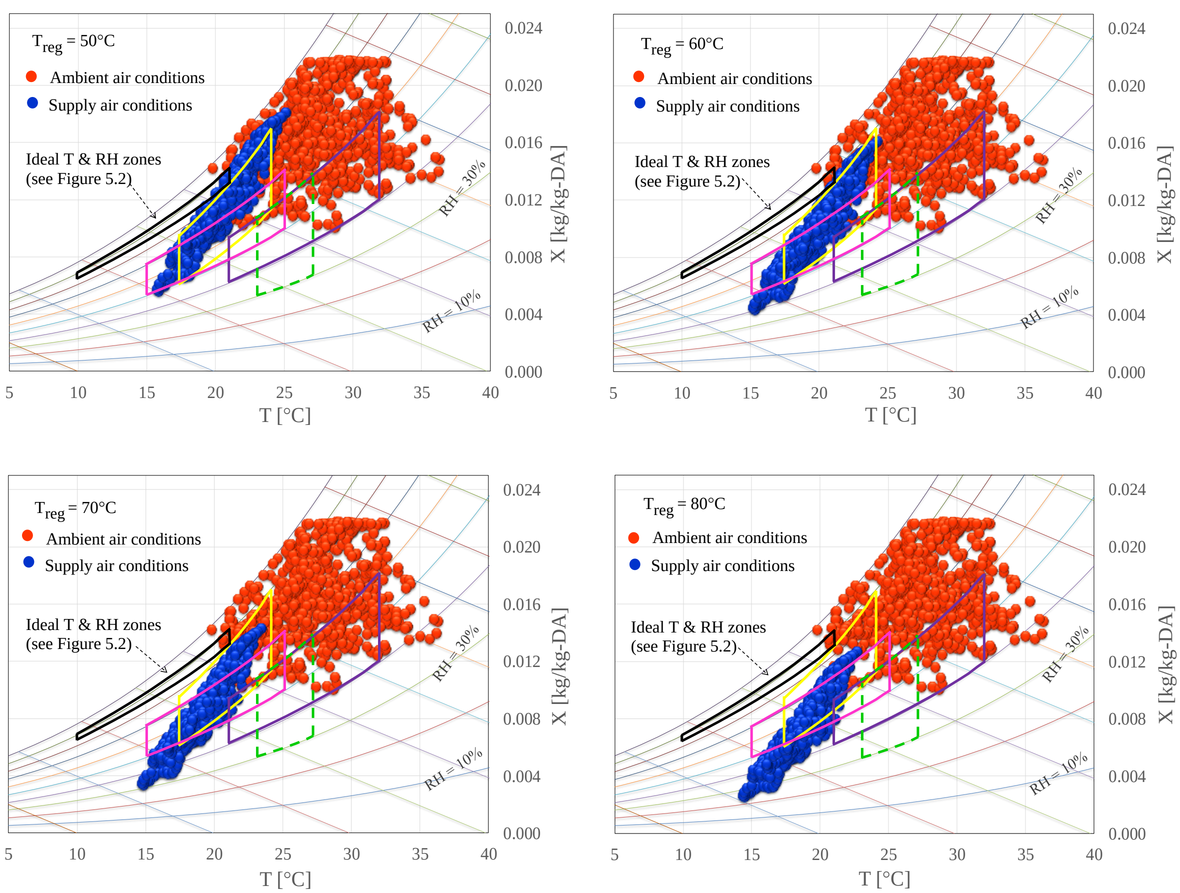

Figure 5.19 Psychrometric evaluation of supply air conditions of S-II for different applications.

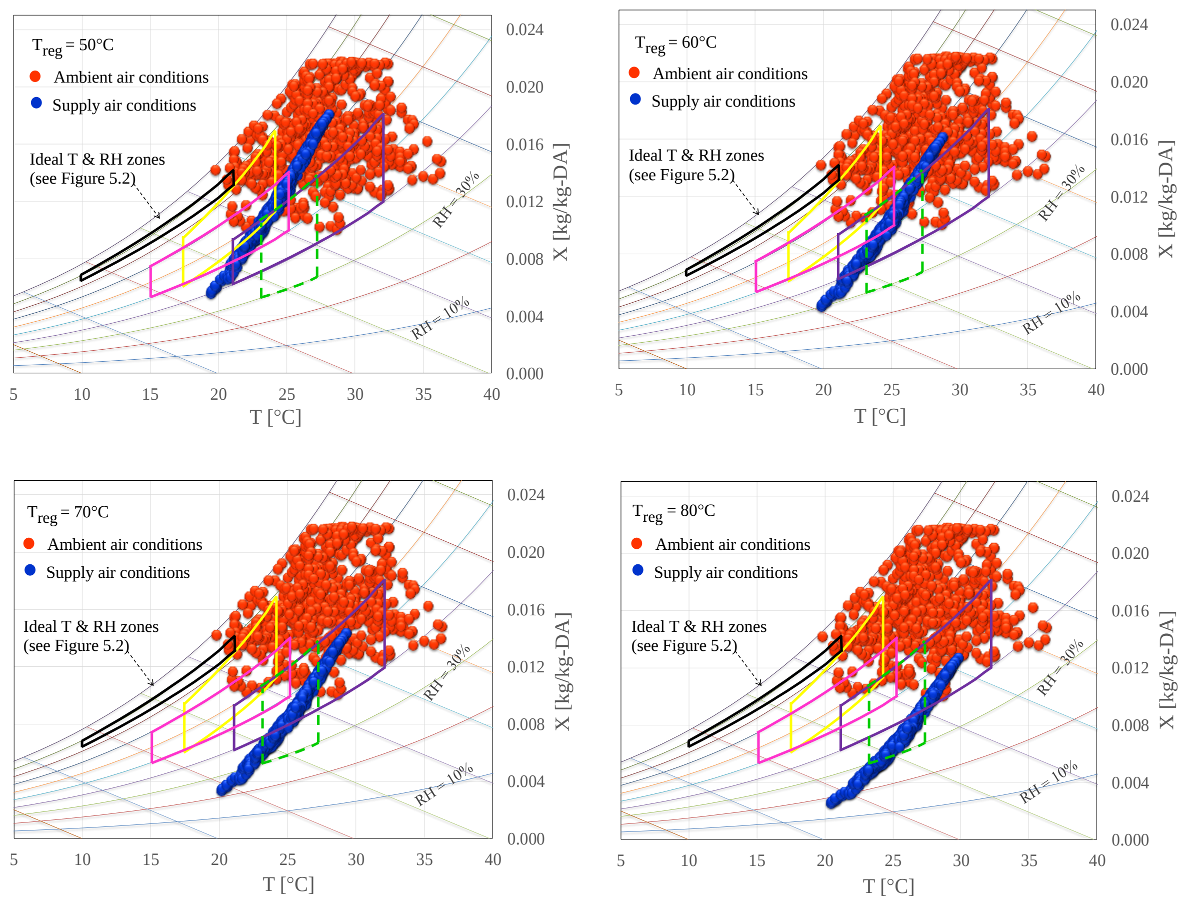

Figure 5.20 Psychrometric evaluation of supply air conditions of S-IV for different applications.

The temporal variation in supply air conditions of S-II at T_{reg} = 50-80°C is presented in Figure 5.19. The psychrometric evaluation revealed that the S-II cannot achieve the latent and sensible load of AC for the storage of agricultural products at T_{reg} = 50-70°C under the varying ambient conditions of Fukuoka. The system (S-II) achieves the latent load at T_{reg} = 80°C (Figure 5.19), however the sensible load can be achieved by further cooling through DEC (i.e. two-stage cooling: MEC+DEC). As the agricultural products required high storage relative humidity and low temperature therefore the integration of DEC after the MEC in S-II configuration can provide the optimum conditions for the storage of agricultural products. Therefore, the configuration of S-II need to be modify as two-stage cooling (MEC+DEC) rather than single-stage cooling (MEC) particularly for the agricultural products storage. But such integration will nullify the cost benefit of S-II over S-I as described above. On the other hand, S-II can provide the storage conditions for the dried fruits at low regeneration temperature (T_{reg} = 70°C) as shown in Figure 5.19. Moreover, the S-II can be effectively used for the provisions of greenhouse growth zone AC even at low regeneration temperature depending upon the type and growth stage of the crop. However, the implication of S-II for rest of the applications can be seen from Figure 5.19.

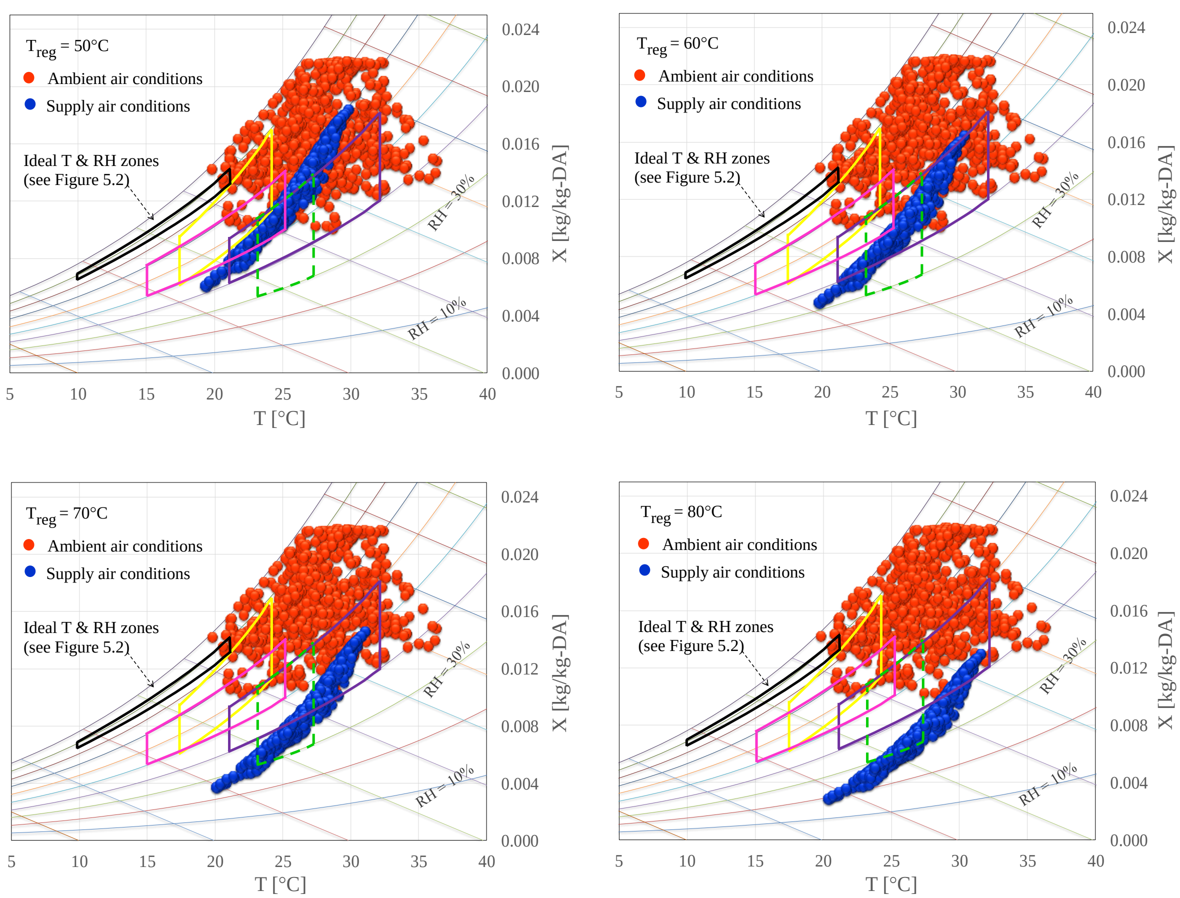

Figure 5.21 Psychrometric evaluation of supply air conditions of S-V for different applications.

As far as psychrometric evaluation of S-IV for the storage of agricultural products is concerned, it is examined that S-IV cannot be effectively used for the storage of agricultural products at T_{reg} = 50-80°C due to non-provision of optimum temperature and relative humidity conditions as shown in Figure 5.20. On the other hand, S-IV almost achieves the latent load at T_{reg} = 70°C but cannot achieves the sensible load of AC for the storage of dried fruits. The sensible load can be achieved by two-stage sensible cooling (MEC + MEC/IEC) in contrast to two-stage cooling (MEC + DEC) of modified S-II. It is because of requirement of less storage relative humidity for dried fruits as compare to agricultural products. Therefore, the configuration of S-IV needs to be modified by integrating one more sensible cooler (MEC/IEC). However, storage of dried food and feeds can be made by S-IV at low regeneration temperature (60°C). Moreover, the S-IV can assist the conventional AC systems for the provision of human thermal comfort.

The supply air conditions of S-V are presented by Figure 5.21. It can be envisaged that T_{SA} and RH_{SA} of S-V are higher and lower respectively in comparison to S-II and S-IV. In this regard, the S-V can be particularly used for the applications which required relatively high temperature and low relative humidity. Therefore it can be potentially employed for the storage of dried food and feeds at low regeneration temperature (60°C). However, the S-V required about two times less regeneration heat input (Q_{reg}) as compare to S-IV.

5.6 Concluding Remarks

The ideal storage zones of agricultural products, dried fruits, dried foods & feeds in comparison with greenhouse growth and humans thermal comfort zones are established on the psychrometric chart. The study experimentally investigates the combined effect of desiccant dehumidification and sensible cooling (via MEC or HX) for the storage of food products. In this regard, six different configurations of DAC systems are proposed and their performance evaluation is made under the ambient conditions of Fukuoka-Japan. It is established on the basis of parametric and thermodynamic analysis of all the system configurations (S-I to S-VI) that S-II, S-IV and S-V could yield better system performance. On the basis of the psychrometric evaluation of three optimized DAC systems (S-II, S-IV and S-V) following itemized concluding remarks can be made:

- S-II cannot achieve the latent and sensible load of AC for the storage of agricultural products at T_{reg} = 50-70°C under the varying ambient conditions of Fukuoka.

- S-II achieves the latent load at T_{reg} = 80°C, however the sensible load can be achieved by two-stage cooling (MEC+DEC). Therefore, the modification of two-stage cooling in is suggested in configuration of S-II.
- S-II can provide the optimum conditions for the storage of dried fruits at low regeneration temperature (T_{reg} = 70°C). Moreover, the S-II can be effectively used for the provisions of greenhouse growth zone AC even at low regeneration temperature depending upon the type and growth stage of the crop.
- S-IV cannot be effectively used for the storage of agricultural products at all (50-80°C) regeneration temperatures.
- S-IV almost achieves the latent load at T_{reg} = 70°C but sensible load for storage of dried fruits can be achieved by two-stage sensible cooling (MEC + MEC/IEC). Therefore two-stage sensible cooling modification is suggested in configuration of S-IV.
- S-IV can assist the conventional AC systems for the provision of human thermal comfort.
- S-V can be potentially employed for the storage of dried food and feeds regeneration temperature (60°C). Moreover, it required about two times less regeneration heat input (Q_{reg}) as compare to S-IV.

Finally it is concluded that thermally driven DAC systems could yield advance agricultural storage system which can control temperature and humidity distinctly irrespective to conventional compressor based AC systems. Therefore, integration of evaporative cooling unit(s) into DAC system will lead towards energy-efficient and reliable low-cost AC systems for various applications. However optimum operational conditions will need to be determined and regulated for particular application. Herein it has been concluded that one or other DAC system could be efficiently utilized for the storage of agricultural products.

5.7 Nomenclature

C_p	specific heat capacity of air [kJ/kg·K]
M_m	moisture contents corresponding to an absorbed monolayer [kg/kg]
Q_{10}	temperature coefficient [-]
Q_{reg}	regeneration heat input [kW]
a_w	water activity [-]
Γ_1	product known shelf life [months]
ω_w	heat of condensation water vapor [kJ/kg]
ϕ_{deh}	slope of dehumidification line
ΔX	net dehumidification [g/kg-DA or kg/kg-DA]
AC	air-conditioning
COP	coefficient of performance [-]
DAC	desiccant air-conditioning
DB	desiccant block
DEC	direct evaporative cooling
ε	effectiveness [-]
h	air enthalpy [kJ/kg-DA or J/kg-DA]
HS	heat source
HX	heat exchanger
IEC	indirect evaporative cooling
M-Cycle	Maisotsenko Cycle
MEC	M-Cycle evaporative cooling
RH	relative humidity [% or -]
T	temperature [°C or K]
X	humidity ratio [g/kg-DA or kg/kg-DA]
M	moisture content of the material on dry basis [kgH_2O/kg-dry solids]
R	gas constant [kJ/kg·K]
Γ	product shelf life [months]

Subscript

dp	dew-point
PA	process air
RA	regeneration air
reg	regeneration
SA	supply air
wb	wet-bulb

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CHAPTER 6

GENERAL CONCLUSIONS

CHAPTER 6

GENERAL CONCLUSIONS

The main objective of the work presented in this thesis was to investigate the applicability of environmentally benign thermally driven desiccant air-conditioning system for agricultural storage applications. In this regards following work is carried out to cope the objectives and scope of the thesis

- The key factors responsible for postharvest losses (PHL) are determined for insights about complex mechanism of respiration, transpiration and fermentation.
- The motivation towards the use of DAC for product storage instead of conventional vapor compression refrigeration and/or air-conditioning systems is ascertained from thermodynamic and product quality point of view.
- Extensive literature review about the existing agricultural product storage techniques and the cooling systems is made to highlight the associated implications. It leads towards the development of low cost energy efficient air-conditioning systems for agricultural storage applications.
- The potential applicability of Maisotsenko Cycle based evaporative cooling systems is analyzed for farm air-conditioning.
- An open-cycle desiccant block experimental apparatus is setup for dehumidification experimentation.
- Uncertainty analysis is made for measure and/or calculated variables in order to determine the efficacy of experimental data.

- Experimental investigation of dehumidification performance of desiccant block for different process/regeneration air conditions and switching time ratio is made (five cases).
- A simplified correlation is developed on the conception of modification of isenthalpic slope of dehumidification line on psychrometric chart.
- The experimental validation of slope of dehumidification line under varying regeneration temperature is also investigated.
- The ideal storage zones of agricultural products, dried fruits, dried foods & feeds in comparison with greenhouse growth and humans thermal comfort zones are established on the psychrometric chart.
- Six different configurations of DAC systems are proposed and their performance evaluation is made under the ambient conditions of Fukuoka-Japan.
- A simplified methodology is developed for performance evaluation of proposed DAC systems under varying regeneration temperatures.
- The psychrometric evaluation of three optimized DAC systems is performed for of agricultural storage and parallel applications.

6.1 General Conclusions

General conclusion from this thesis can be extracted as follows:

- The conventional HVAC systems are difficult to provide optimum conditions for the storage of agricultural products. It is mainly due to the uneven control of humidity and/or ventilation.
- Direct/Indirect evaporative cooling systems yield low cost and environmental friendly AC systems in order to control sensible load of AC.
- M-Cycle based evaporative cooling systems can cool the air below wet-bulb temperature and approaches to the dew-point. Therefore, its applicability has been found suitable for the storage of various agricultural products where main objective is to reduce the temperature sensibly.

- Evaporative cooling systems including direct, indirect, M-Cycle are applicable to dry regions/climate, whereas desiccant air-conditioning can be employed efficiently in humid climates/regions.
- The performance of DAC is mainly based on desiccant unit, therefore experiments have been conducted for the performance evaluation of polymer sorbent based honeycomb like desiccant block. The experimental setup can reproduce the data within 2-3% error.
- The equivalent heat of adsorption (q_{eq}) of case-B at both regeneration temperatures (40°C and 60°C) is found greater than case-A due to higher amount of adsorbed water vapors.
- Average effective dehumidification slightly increases for case-C and case-D by changing the switching time ratio from 1:1 to 2:3 due to higher process air relative humidity. However, it keeps decreasing with increase in switching time ratio for case-E due to saturation of the sorbent.
- It has been found that the switching time depends on dehumidification amount, nature of application and operating conditions. From the bunch of experiments it has been generally concluded that the switching time ratio of 1:2, 2:3 and 1:1 can be selected for the operation of block based DAC system employing the condition of high, medium and low humidity operating conditions, respectively.
- Higher regeneration temperature brings fast and deep dehumidification, though it does not guarantee the higher system COP. An optimized regeneration temperature will be required for each operating conditions in order to make the system cost effective.
- According to the experiments made on open-cycle DAC system, it has been found that the real desiccant dehumidification process does not follow the isenthalpic behavior due to the adsorption heat. In contrary the real desiccant dehumidification process can be imagined on psychrometric charts employing the slope of desiccant dehumidification process line ranging from 0.8-0.31. For dehumidification cycle of longer than 10 minutes the value of slope may be taken as 0.31 for the simple analysis.

- Six DAC system combinations are proposed for various storage applications for Fukuoka climates. It is established on the basis of parametric and thermodynamic analysis of all the system configurations (S-I to S-VI) that S-II, S-IV and S-V could yield better system performance.
- S-II cannot achieve the latent and sensible load of AC for the storage of agricultural products at T_{reg} = 50-70°C under the varying ambient conditions of Fukuoka.
- S-II achieves the latent load at T_{reg} = 80°C, however the sensible load can be achieved by two-stage cooling (MEC+DEC). Therefore, the modification of two-stage cooling in is suggested configuration of S-II.
- S-II can provide the optimum conditions for the storage of dried fruits at low regeneration temperature (T_{reg} = 70°C). Moreover, the S-II can be effectively used for the provisions of greenhouse growth zone AC even at low regeneration temperature depending upon the type and growth stage of the crop.
- S-IV cannot be effectively used for the storage of agricultural products at all (50-80°C) regeneration temperatures.
- S-IV almost achieves the latent load at T_{reg} = 70°C but sensible load for storage of dried fruits can be achieved by two-stage sensible cooling (MEC + MEC/IEC). Therefore two-stage sensible cooling modification is suggested in configuration of S-IV.
- S-IV can assist the conventional AC systems for the provision of human thermal comfort.
- S-V can be particularly used for the storage of dried food and feeds because it required about two times less regeneration heat input as compare to S-IV.

The study concludes that the evaporative cooling (preferably M-Cycle) systems should be considered on top priority for agricultural storage applications wherever these are thermodynamically and meteorologically applicable. When these are not applicable, thermally driven DAC systems could yield advance agricultural storage system which can control temperature and humidity distinctly irrespective to conventional compressor based AC systems. Therefore, integration of evaporative cooling unit(s) into DAC system will lead towards energy-efficient and reliable low-cost AC systems for various applications.

However optimum operational conditions will need to be determined and regulated for particular application. Herein it has been concluded that one or other DAC system could be efficiently utilized for the storage of agricultural products.

6.2 Future Work

- Development and standardization for agricultural products storage zones and validation with the real data.
- Consideration of non-polymeric sorbents for the formation of desiccant blocks to be used for desiccant air-conditioning.
- Dynamic simulation of proposed six systems and validation with experiments for polymeric and non-polymeric materials.
- Size estimation for corresponding DAC systems along with their components for various agricultural storage buildings.

APPENDIX A

SCOPE OF NANO/MICRO POLYMERIC MATERIALS IN FARM AIR CONDITIONING APPLICATIONS

APPENDIX A

SCOPE OF NANO/MICRO POLYMERIC MATERIALS IN FARM AIR CONDITIONING APPLICATIONS

Significance of nano and micro materials is obvious in 21^{st} century for various technological, medical and engineering applications. In this regards, carbon based nano/micro materials have shown huge potential in many adsorption based applications e.g. water and wastewater treatment. In addition, these are successfully utilized for various (methanol-, ethanol-, and ammonia- based) close-cycle adsorption heat pump systems. Significance of nano/micro polymers is also well-known in drug/medical industry, and therefore extensively studied for various aspects of adsorption. Present study focuses its application in open-cycle based dehumidification and air-conditioning for agriculture sector. Study comprises two kinds of carbon and four kinds of polymer materials for potential application in farm air-conditioning. Water vapor adsorption comparison has been made among nano/micro materials and conventional hydrophilic adsorbent i.e. silica-gel. The size of desiccant unit in desiccant air-conditioning system has been determined and compared accordingly. Results showed that 2-3 times smaller system size can be yielded under particular conditions while utilizing nano/micro materials intelligently. It has been found that nano/micro materials facilitate the formation of various conceptual designs (e.g. honeycomb like structure, coating on heat exchanger etc.) of desiccant unit promoting the heat mass transfer between desiccant and air sides. Moreover according to literature review these materials can be regenerated at relatively low regeneration temperature, providing an opportunity to utilize the renewable solar thermal energy, bio-gas and bio-diesel.

A.1 Introduction

The vital need of heating, ventilation and air-conditioning (HVAC) for human thermal comfort is well known in the literature. Many designs of HVAC systems have also been investigated for various applications like conventional vapor compression air-conditioning (VAC) for residential, commercial, institutional and office buildings, data center cooling, electronics cooling, manufacturing and storage processes etc. (Mahmood et al., 2016c). However, the effective farm level applications of these systems have not been much explored in the literature. Its farm level applications may include (i) air-conditioning (AC) for human (farm residents/workers) thermal comfort, (ii) AC for farm livestock thermal comfort, (iii) AC for farm greenhouse, and (iv) AC for pre-cooling and storage of farm produce etc. The conventional vapor compression refrigeration and/or air-conditioning systems are not only responsible for high energy requirements and environmental pollution but also cannot be used to provide optimal storage conditions to the agricultural products (e.g. mango, tomato, leafy vegetables etc.) due to chilling injury, discoloration and off-flavor (Mahmood et al., 2016b; Ndukwu and Manuwa, 2015; Olosunde et al., 2016). Moreover, these systems are not best suited to control the latent and sensible load of greenhouse environment (Sultan, 2015). The complex mechanism of transpiration, respiration, fermentation in stored agricultural product and photosynthesis, evapotranspiration in growing greenhouse plants required active consideration about the provisions of ventilation rate, temperature and relative humidity control (Mahmood et al., 2016b; Sultan, 2015). On the other hand, the much higher ventilation rate is required in animal houses to remove the odors and ammonia. It may go up to maximum 60 air changes per hour in order to maintain the indoor air quality, temperature, and relative humidity (ASHRAE, 2007). The ammonia is most dangerous and chronic contaminant gas produced within the animal house due to the decomposition of manure. The conventional vapor compression refrigeration and/or air-conditioning systems are considered unfeasible and/or uneconomical for animal houses due to the requirements of higher air changes per hour

(ASHRAE, 2007). The high ventilation rate is essential to maintain the indoor air quality by removing odor, ammonia and heat. Therefore, there is dire need of low cost, environmental friendly air-conditioning system/package that can be effectively employed for all farm level applications. In this regard, evaporative cooling technologies like direct evaporative cooling (DEC), indirect evaporative cooling (IEC)/M-Cycle evaporative cooling (MEC) have shown potential for farm level applications (Mahmood et al. 2016a; Sultan, 2015). However, these technologies do not perform efficiently under largely varying ambient conditions (particularly humid) due to limited cooling performance. The scope of DEC and IEC/MEC for various agricultural applications under varying environmental condition can be extended by the integration of desiccant dehumidification. The desiccant air-conditioning (DAC) comprises of desiccant dehumidification cum evaporative cooling has ability to deal the latent and sensible load of air-conditioning distinctly. It also gives opportunity to operate it by low grade waste heat, and renewable energy options like solar energy (particularly solar thermal) bio-gas, bio-diesel etc. Therefore, energy and environmental friendly DAC system represent zero global warming (GWP) and ozone depletion potential (ODP). In this regard, lots of practical air-conditioning systems based on desiccant has been established and working all over the world. Mostly studied desiccant air-conditioning systems are standalone DAC systems with aid of simplified heat exchangers, direct and/or indirect evaporative cooling assisted DAC systems, vapor compression based hybrid DAC systems, solar energy operated DAC systems, bio-gas operated DAC systems, single/multi stage DAC systems, solid/liquid based DAC systems, wheel/block based DAC systems, heat and pressure regeneration based DAC systems and many more. Such characteristics of the DAC system make it promising for the widespread applications like agricultural product storage/preservation, greenhouses, marine ships, automobiles, buildings, museums, hospitals etc. (Ascione et al., 2009; Ascione et al., 2013; Enteria et al., 2009; Enteria et al., 2010; Guojie et al., 2012; Lee and Lee, 2013; Longo and Gasparella, 2015; Nagaya et al., 2006; Sultan et al., 2014a; Zhu and Chen, 2014).

In purview of above discussion, it is ascertain that the DAC system can be a promising solution for all farm level air-conditioning applications. Additionally, it can be operated by harvesting the renewable energy sources (solar energy, bio-gas, bio-diesel etc.)

abundantly available at the farm. The details about the typical system operation can be found from reference (Sultan et al., 2015a). The scopes of nano/micro material in desiccant air-conditioning are explained in heading A.2.

A.2 Scope of the Study

The desiccant air-conditioning despite of energy saving and eco-friendly technology still could not break the barriers for its market penetration. The main barriers are high initial cost, long payback period and system size (Sultan et al., 2015a). The one of the main component in DAC system is desiccant unit by which system size and operational cost are associated. There is dire need of research for the development of small size, low cost, non-toxic and non-corrosive adsorbent bed. The adsorbent should have high adsorption uptake with higher adsorption kinetics. In this regard nano/micro materials can play role towards the optimization of DAC system size. The significance of nano and micro materials is obvious for various technological, medical and engineering applications. Physiochemical and thermodynamic properties of nano/micro materials have been effectively realized for various adsorption based applications e.g. wastewater treatment, adsorption heat pump systems etc. Carbon based nano/micro materials have shown huge potentials in numerous adsorption applications particularly for water and wastewater treatment. It has been also successfully utilized for various (methanol-, ethanol-, and ammonia- based) close cycle adsorption heat pump systems. However, the present study focused towards the open cycle desiccant air-conditioning. The utilization of nano/micro materials in DAC and its further application in agriculture sector is relatively new in the literature. Therefore, .the study comprises two kinds of carbon and four kinds of polymer materials for potential application in farm air-conditioning. Water vapor adsorption comparison has been made among nano/micro materials and conventional hydrophilic adsorbent i.e. silica-gel. The size of desiccant unit in desiccant air-conditioning system has been determined and compared accordingly. The detailed methodology adopted in this regard is explained in heading A.3.

A.3 Materials and Methods

A.3.1 Materials

The study comprises different adsorbents such as polymer based, carbon based, and conventional silica-gel. The polymer based adsorbents include polymer sorbent-I (PS-I), polymer sorbent-II (PS-II) (Sultan et al., 2015b; Sultan et al., 2016a), and polystyrene sulfonic acid, sodium salt - mono sulfonated (PSS-MS) (Toribio et al., 2004; Toribio et al., 2005) and polystyrene sulfonic acid, sodium salt - full sulfonated (PSS-FS) (Toribio et al., 2004; Toribio et al., 2005). The carbon based adsorbents include activated carbon powder (ACP) (Sultan et al., 2016b), and activated carbon fiber (ACF) (Sultan et al., 2016b). The particle size of these adsorbents is given in Table A.1 (Sultan et al., 2015b; Sultan et al., 2016a; Sultan et al., 2016b; Toribio et al., 2004; Toribio et al., 2005). All these adsorbents have specific adsorption characteristics. Silica-gel is mostly used in the designing of DAC systems due to its strong affinity towards water (Sultan et al., 2015a).

Table A.1 The particle diameter of studied adsorbents.

Adsorbents	Particle diameter (μm)	References
PS-I	2.6	Sultan et al., 2015b; Sultan et al., 2016a
PS-II	19.3	Sultan et al., 2015b; Sultan et al., 2016a
PSS-MS	10.0	Toribio et al., 2004; Toribio et al., 2005
PSS-FS	10.0	Toribio et al., 2004; Toribio et al., 2005
ACF	13.0	Sultan, 2015; Sultan et al., 2016b
ACP	105	Sultan, 2015; Sultan et al., 2016b
Silica-gel	1000	Sultan, 2015; Sultan et al., 2016b

The silica-gel has higher bulk density than PS-I but less than PS-II. It has been determined that the PS-II possesses about 1.5 times higher bulk density than PS-I and about 1.16 times higher than silica-gel (Sultan et al., 2015b). The microscopic images of PS-I and PS-II particles are shown in Figure A.1(a) and (b) respectively (Sultan, 2015). The illustration these of materials in to desiccant block, desiccant rotor, desiccant coated heat exchangers is shown in Figure A.1. The other hydrophilic polymer adsorbents (PSS-MS and PSS-FS) were manufactured (by Polysciences, Europe) with the sulfonation of styrene (Toribio et al., 2004). The measured density of mono and full sulfonated PSS at room conditions was 1400 kg/m^3 (Toribio et al., 2004; Toribio et al., 2005). It is important to mention that the density of adsorbent is crucial towards the optimization of system size for different farm level applications. The carbon based adsorbents (ACP, ACF) also have typical properties. The ACP is a highly porous adsorbent, whereas, ACF as a fibrous adsorbent possesses high porosity, and ease in handling [Sultan et al., 2016b]. The detailed information can be found from the cited literature.

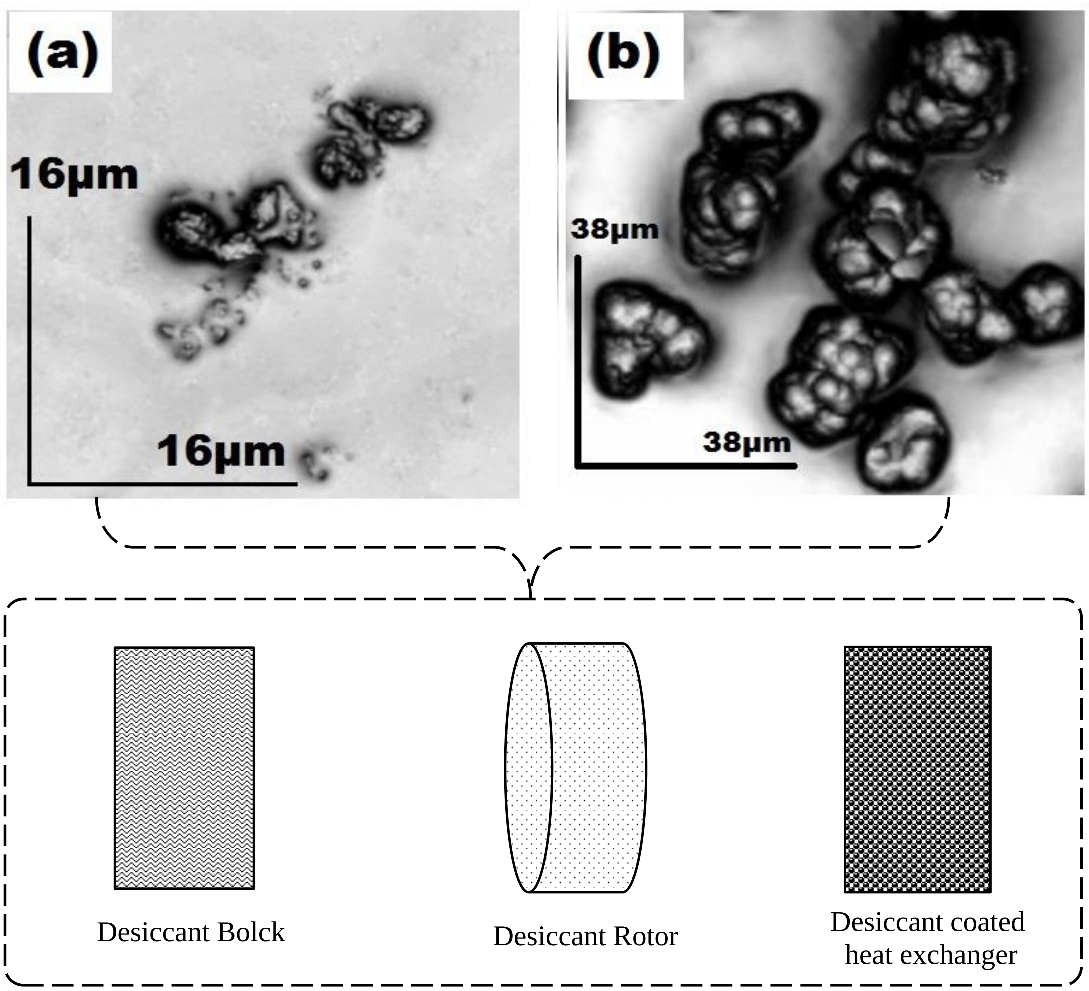

Figure A.1 Few particles microscopic image of (a) PS-I, and (b) PS-II (Sultan et al., 2015b, Sultan et al., 2016a).

A.3.2 Data Analysis and Procedure

The isotherm plots of adsorbents (Table A.1) are drawn and compared in order to determine the adsorption uptake at respective temperature on the basis of data from references (Sultan, 2015; Sultan et al., 2015b; Sultan et al., 2016a; Sultan et al., 2016b; Toribio et al., 2004; Toribio et al., 2005). The isosteric heat of adsorption (Q_{st}) is also compared for PS-I, PS-II and silica-gel (Sultan, 2015; Sultan et al., 2015a; Sultan et al., 2015b). The performance of the adsorption heat pump systems (e.g. DAC system) mainly depends upon the amount of Q_{st}. Moreover, the contribution of the nano/micro polymeric material towards the system size reduction is investigated in comparison to conventional silica-gel. In this regard, the adsorbent to air mass fraction ($MF_{A\text{-}A}$) is calculated by the following relationship as given in the literature [Sultan et al., 2014b; Sultan et al., 2016b].

$$MF_{A-A} = \left(\frac{\Delta W}{\Delta M}\right)_{\Delta RH} = \left(\frac{W_1 - W_2}{M_1 - M_2}\right)_{\Delta RH} \qquad (A.1)$$

where, MF_{A-A} describes the adsorbent to air mass fraction (g_{ads}/kg_{DA}), W_1 and W_2 are the humidity ratios (g_{H2O}/kg_{DA}), M_1 and M_2 are the adsorption uptakes (g_{H2O}/g_{ads}). The $MF_{A\text{-}A}$ of the adsorbents (PS-I, PS-II, silica-gel) is calculated for high, medium and low humidity applications/conditions. The psychrometric representation of these applications is shown in Figure A.2.

The ambient air (T, RH) is considered same for all the adsorbents (PS-I, PS-II, silica-gel) in the respective humidity application. The state (1) in Figure A.3 represents the ambient air for medium humidity applications. Moreover, the states (2,3,4,5) represents the different stages of the simplified typical DAC cycle for different levels (a,b,c) of dehumidification and regeneration. The process/ambient air (1) becomes dehumidified as it passes through the adsorbents. The dehumidified air (2) is further passed through the sensible heat exchanger and/or evaporative coolers to supply the conditioned air. On the regeneration side the same ambient air (1) is used for the purpose. The regeneration air becomes hot by exchanging heat (via heat exchanger) with the process air. This hot regeneration air is then further heated (state 4) through external heat source to pass it though the adsorbent and finally exhaust (state 5) to the environment.

The study towards the application side mainly focuses on the air-conditioning needs at the farm level. The psychrometric evaluation of thermal comfort zones for human and livestock, ideal growth zone for agricultural greenhouse, and ideal storage zones for agricultural products revealed that all these application required certain level of latent and sensible load of air-conditioning. In this regard DAC has shown potential to control the latent and sensible load of AC distinctly (Mahmood et al., 2016a; Mahmood et al., 2016b; Sultan, 2015; Sultan et al., 2015a).

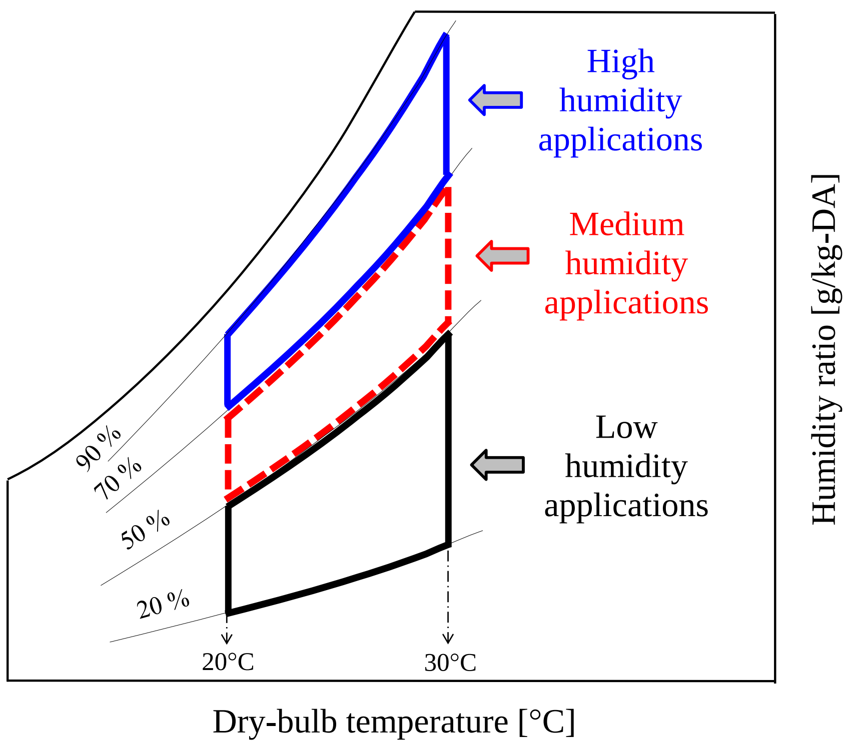

Figure A.2 Psychrometric representation of different humidity applications for determining the adsorbent to air mass fraction.

A.4 Results and Discussion

The water vapor adsorption isotherms are drawn on Figure A.4(a) using the data available in the literature for two carbon based and four polymer based nano/micro materials in comparison with silica-gel (Sultan et al., 2015b; Sultan et al., 2016a; Sultan et al., 2016b; Toribio et al., 2004; Toribio et al., 2005). It can be noticed that carbon based adsorbents i.e. ACP and ACF shows hydrophobic behaviour at low relative pressure range, therefore may be neglected for the further investigation. However it is important to mention that such kind of water vapor adsorption might interesting for various high humidity based DAC applications e.g. agricultural greenhouses, storage of fresh fruits and likewise. This kind of adsorption is based on water vapor condensation using multilayer adsorption phenomena (Sultan et al., 2016a). The water vapor adsorption isotherms of ACP and ACF resemble to the IUPAC type-III and type-V isotherms, respectively. On the other hand, the adsorption isotherms of polymer adsorbents (PSS-MS, PSS-FS) represent the IUPAC type-II isotherm. The lower water vapor adsorption uptake has been observed in both PSS-MS and PSS-FS at low relative pressure. However, the hydrophilic nature of these polymers at higher relative pressure also depends upon the temperature (Toribio et al., 2004; Toribio et al., 2005).

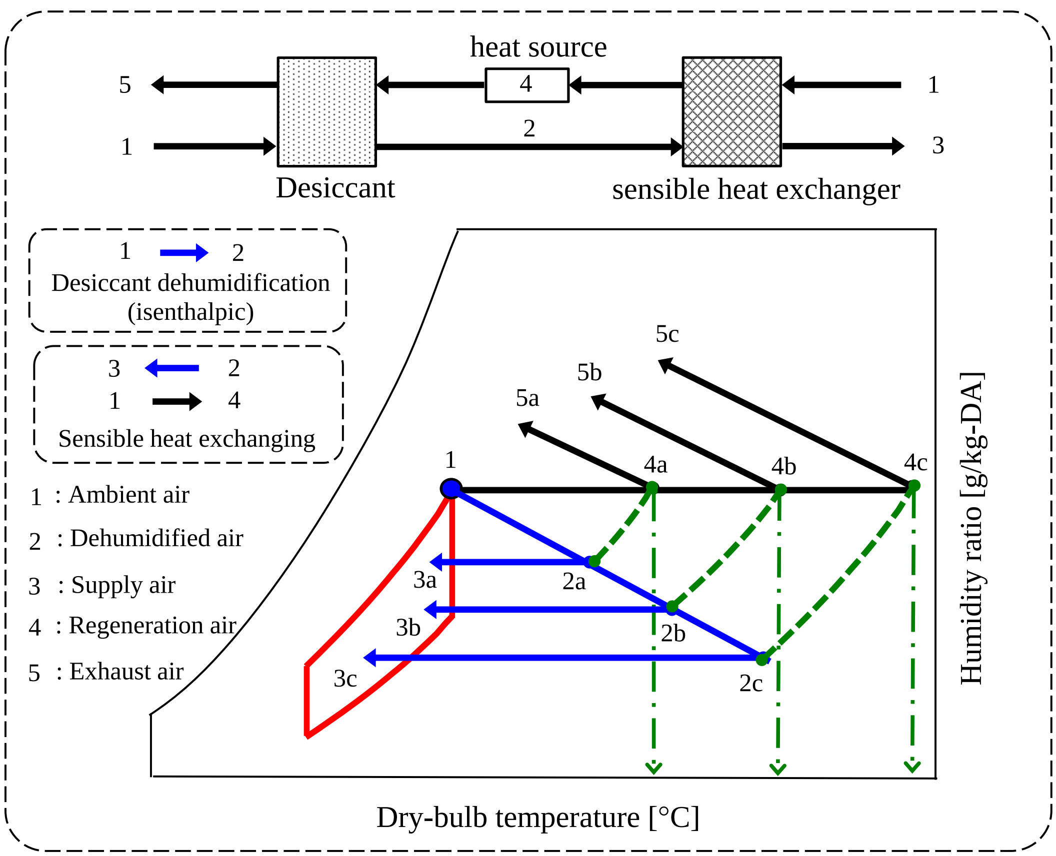

Figure A.3 Schematic of the DAC system with psychrometric representation of ideal DAC cycle for medium humidity applications.

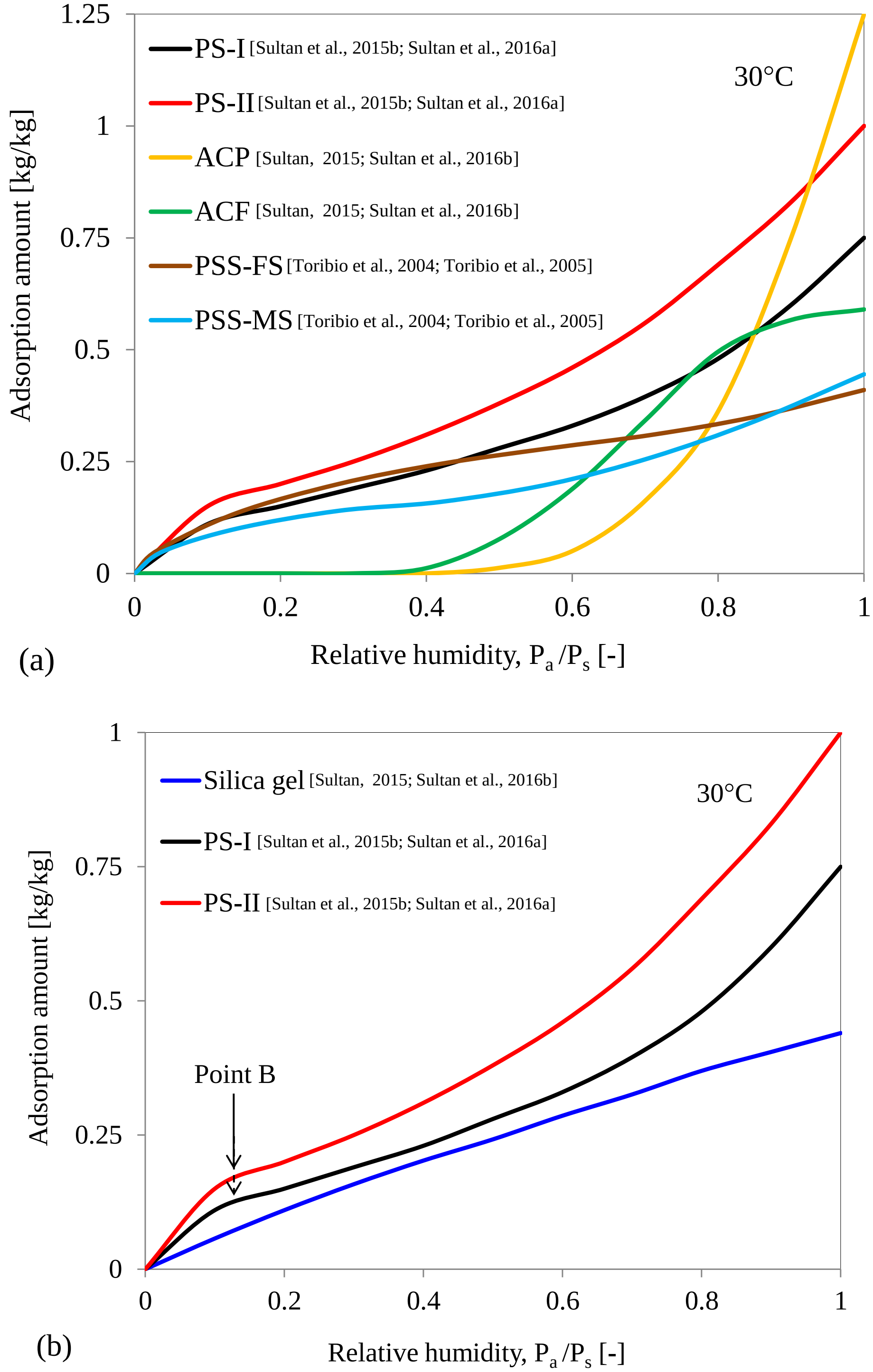

Figure A.4 Comparison of adsorption isotherms at 30°C for (a) PS-I, PS-II, ACP, ACF, PSS-FS, PSS-MS; (b) PS-I and PS-II with Silica-gel.

The full sulfonation of the polystyrene sulfonic acid, sodium salt (PSS) increased its adsorption capacity by 50% than mono sulfonation at particular conditions (Toribio et al., 2004). PSS-MS and PSS-MF enable the similar adsorption trend as achieved by PS-I and PS-II, but possesses considerable lower equilibrium adsorption uptake. Therefore the investigation of nano/micro materials in this study will be focused to PS-I and PS-II. In this regard PS-I and PS-II are compared with conventional silica-gel distinctly as depicting on Figure A.4(b). It can be seen from Figure A.4(b) that the water vapor adsorption uptake by silica-gel at saturation conditions (on 30°C) is about 0.40 kg/kg. But, the adsorption uptake by PS-I and PS-II under the same conditions is almost 2.0 and 2.5 times higher than silica-gel adsorption uptake, respectively (Sultan, 2015). It is noticed that the isotherms of both the PS-I and PS-II have concave shape at the start (lower relative pressure) and then turned to convex shape as the relative pressure increases. Such shape of isotherms is due to two principal classes/fractions of the sorbed water (Smith, 1974; Sultan et al., 2015b). The first is bound water on the inner/outer surface of the adsorbent by the forces in excess of the normal forces which results in concave shape. The second fraction is responsible for convex shape because it is normally condensed within the adsorbent as the function of relative pressure. The summation of these fractions results in sigmoid shape polymer isotherms (Smith, 1974; Sultan et al., 2015b). It is worthy to mention that the sigmoid shape isotherms of polymeric material (PS-I and PS-II) are important for the designing of open cycle desiccant air-conditioning systems. Moreover, the bar graphs shown in Figure A.5 mainly represent the adsorption uptake fraction (Φ) of PS-I and PS-II over silica-gel. The adsorption amounts of PS-I, and PS-II are divided by the adsorption amounts of the silica-gel in order to determine the Φ_{PS-I} and Φ_{PS-II}, respectively. It can be seen from Figure A.5 that Φ_{PS-II} is higher than the Φ_{PS-I} over the entire range of relative humidity. Therefore, it leads towards the better adsorption performance by PS-II under varying relative humidity conditions. The Q_{st} of the polymer adsorbents (PS-I, PS-II) and conventional silica-gel is also compared (Figure A.6) in order to further investigate the insights about these adsorbents. It represents the strength of adsorbent-refrigerant interaction and the performance of the adsorption heat pump systems mainly depend on it (Sultan 2015). The Q_{st} of the silica-gel largely vary as compare to PS-I and PS-II (Figure A.6). However, in case of PS-I and PS-II the isosteric heat of adsorption decreases relatively rapidly than silica-gel. Therefore, it can be concluded from this detailed discussion that the PS-I and PS-II can be potential adsorbents for the optimization of DAC system.

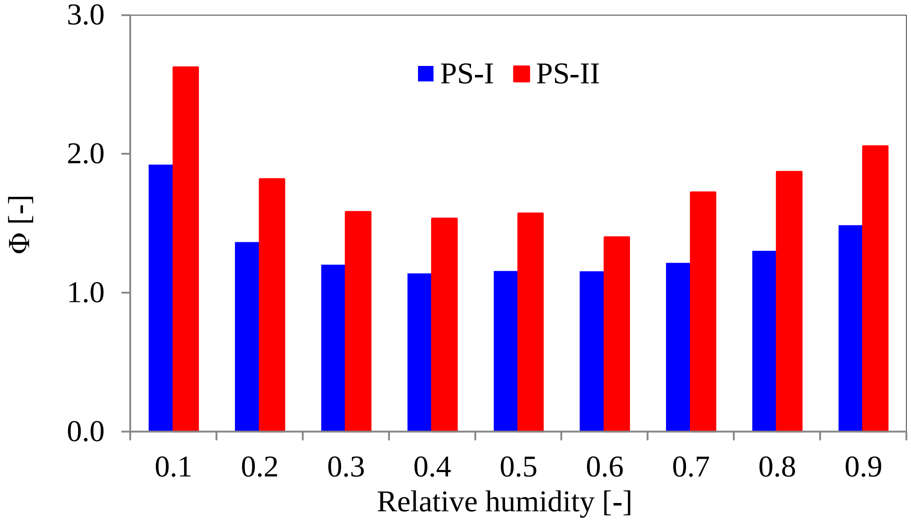

Figure A.5 Adsorption uptake fraction of adsorbents over silica-gel.

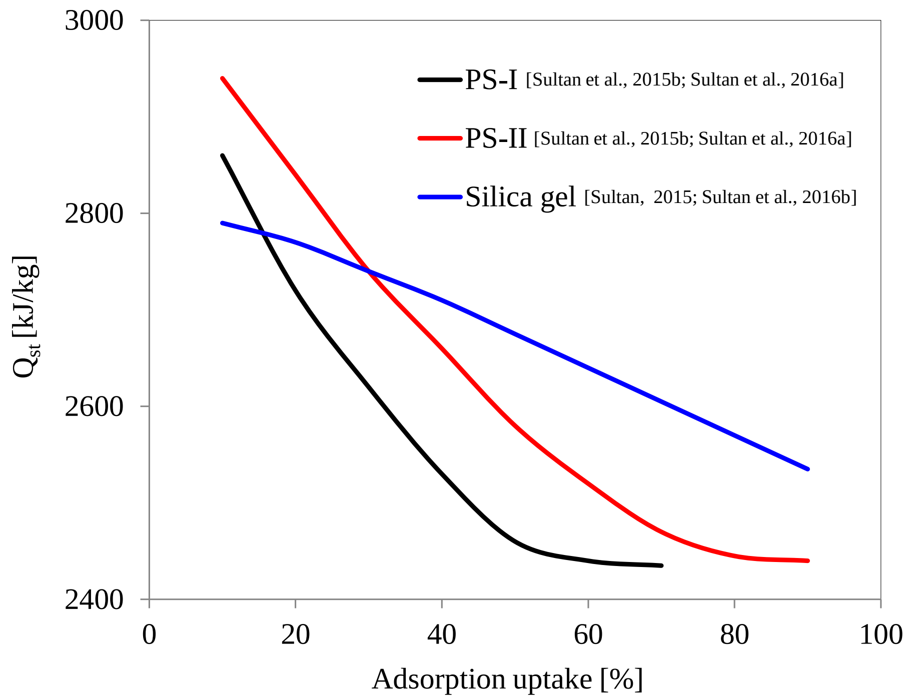

Figure A.6 Comparison of isosteric heat of adsorption of PS-I and PS-II with silica-gel.

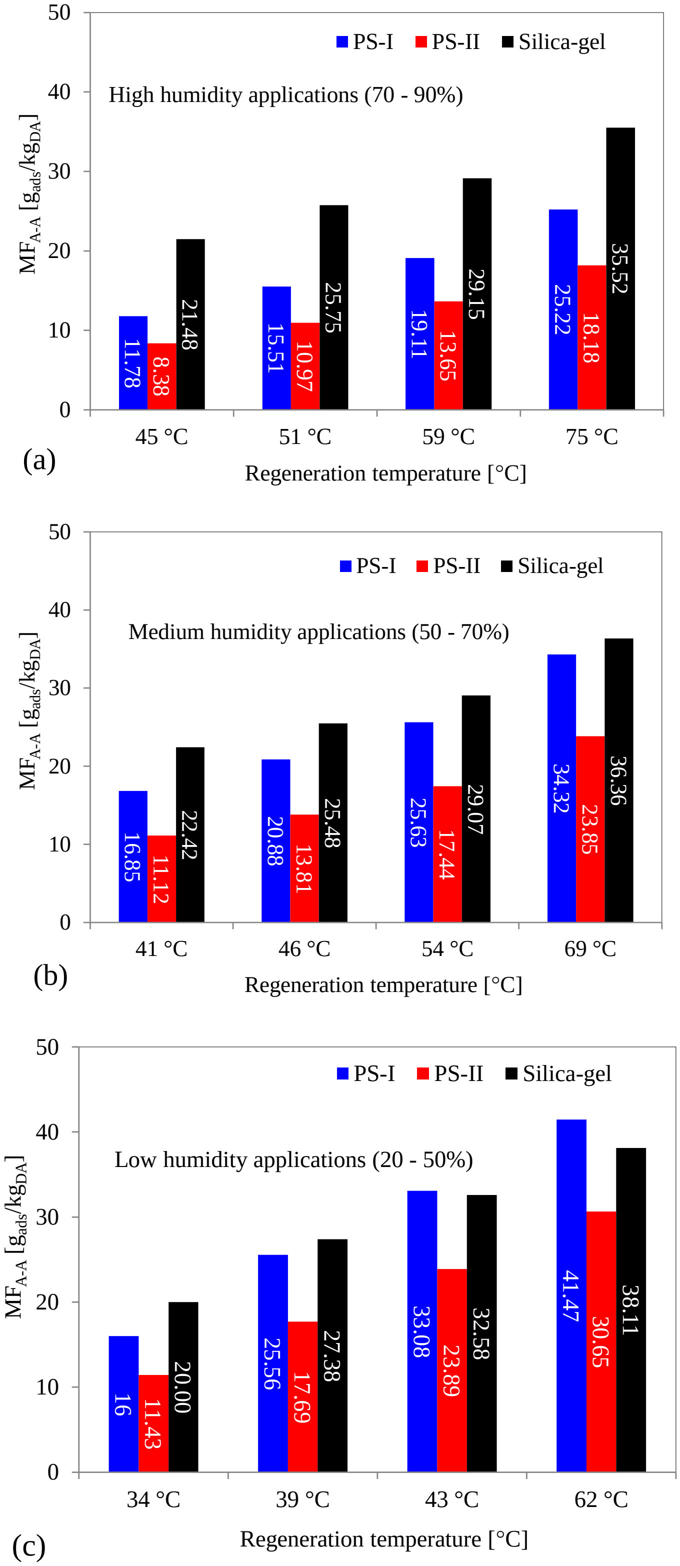

Figure A.7 The adsorbent to air mass fraction under varying regeneration temperature for (a) high; (b) medium; and (c) low humidity conditions.

The contribution of the nano/micro polymeric material (PS-I, PS-II) towards the system size reduction is investigated in comparison to conventional silica-gel using Eq. (1) as explained in methodology section. It can be noticed that the $MF_{A\text{-}A}$ increases with increasing regeneration temperature for all the adsorbents and under all humidity applications as shown in Figure A.7(a),(b),(c). It is determined that the $MF_{A\text{-}A}$ for PS-II is well lower than silica-gel as compare to PS-I. In this regard, it can be envisaged that the PS-II can dehumidify the same amount of air with 2-3 times less adsorbent as compare to silica-gel under particular regeneration temperature. Therefore, the utilization of nano/micro polymeric material (PS-II) can leads towards the reduction in system size by 2-3 times under particular conditions as far as desiccant unit is concerned. Moreover, the nano/micro polymeric materials can be shaped in to honeycomb like structure (desiccant block, desiccant rotor, desiccant coated heat exchangers etc.). The polymeric materials can be regenerated at low temperature. It gives opportunity to harvest the renewable energy sources like solar energy (preferably solar thermal), bio-gas and bio-diesel (etc.) usually amply available at the farm. Therefore, the compact sized DAC system can be operated more economically for different farm applications by performing regeneration through renewable energy sources.

A.5 Conclusions

This study provides water vapor adsorption comparison among different nano/micro materials and conventional hydrophilic adsorbent, silica gel. The studied polymeric and carbon based nano/micro adsorbents include the PS-I, PS-II, PSS-MS, PSS-FS, ACP and ACF. It is investigated that ACP and ACF shows hydrophobic behaviour at low relative pressure range, therefore neglected for the further investigation in this study. However it is important to mention that such kind of water vapor adsorption might interesting for various high humidity based DAC applications e.g. agricultural greenhouses, storage of fresh fruits and likewise. On the other hand, PSS-MS and PSS-MF enable the similar adsorption trend as achieved by PS-I and PS-II, but possesses considerable lower equilibrium adsorption uptake. Therefore the investigation of nano/micro materials in this study mainly focused to

PS-I and PS-II. Moreover, the following generalized conclusions can be highlighted on the basis of this study.

- ✓ Carbon based adsorbents enable adsorption uptake at very high relative humidity range due to their adsorption isotherms of IUPAC type III and V.
- ✓ PSS-FS and PSS-MS enable IUPAC type II isotherm similar to PS-I and PS-II, however enable lower equilibrium adsorption uptake.
- ✓ Polymeric nano/micro adsorbents (PS-I and PS-II) enable higher water vapor adsorption uptake with IUPAC adsorption isotherm type II.
- ✓ The nano/micro polymeric can be well shaped into channels and honeycomb like structure (desiccant block, desiccant rotor, desiccant coated heat exchangers etc.).
- ✓ Polymeric nano/micro adsorbents facilitate low temperature regeneration, providing an opportunity to run the system on renewable energy (like solar, bio-gas, bio-diesel etc.).
- ✓ PS-II reduces the desiccant unit size (in the DAC system) by 2-3 times as compare to conventional silica-gel under particular conditions.

The study provides fundamental guidelines of polymer nano/micro materials application in air-conditioning, and therefore will be worthy for the researchers/scientists working for the development of advance polymer based nano or micro materials.

A.6 Nomenclature

AC	air-conditioning
ACF	activated carbon fiber
ACP	activated carbon powder
DAC	desiccant air-conditioning
M	adsorption uptakes [g_{H2O}/g_{ads}]
MF_{A-A}	adsorbent to air mass fraction [g_{ads}/kg_{DA}]
P_a	partial vapor pressure [kPa]
P_s	saturation vapor pressure [kPa]
PS	polymer sorbent
PSS-FS	polystyrene sulfonic acid, sodium salt - full sulfonated

PSS-MS polystyrene sulfonic acid, sodium salt - mono sulfonated

Q_{st} isosteric heat of adsorption [kJ/kg]

RH relative humidity [%] or [-]

T temperature [°C or K]

W humidity ratio [g_{H2O}/kg_{DA}]

Φ adsorbents adsorption uptake fraction over silica-gel

Subscripts

DA dry air

ads adsorbent

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APPENDIX B

STRENGTHS, WEAKNESSES, OPPORTUNITIES, THREATS (SWOT) ANALYSIS OF LOW COST COOLING TECHNOLOGIES FOR AGRICULTURAL PRODUCT STORAGE

APPENDIX B

STRENGTHS, WEAKNESSES, OPPORTUNITIES, THREATS (SWOT) ANALYSIS OF LOW COST COOLING TECHNOLOGIES FOR AGRICULTURAL PRODUCT STORAGE

This chapter investigates the prospective and challenges associated with low cost, energy and environmental friendly appropriate cooling technologies for the storage of agricultural products. In this regard, strengths, weaknesses, opportunities, threats (SWOT) analysis is carried out for the purpose. It is used to investigate the strengths, weaknesses, opportunities and threats related to the evaporative cooling, desiccant air-conditioning and compressor based air-conditioning technologies in general and in particular when employed for the storage of agricultural products. The study comprises on extensive literature review, experimental and theoretical studies published in the literature and discussion with the researchers working in the parallel field. The resultant performance by all kinds of air-conditioning systems were based on ambient air conditions as well as nature of storage application. However according to the findings, it has been concluded that the one or other technology (evaporative cooling or desiccant air-conditioning) could yield low cost air-conditioning system for the storage of agricultural products.

B.1 Introduction

The global production of agricultural products (fruits and vegetables) has crossed 1.8 billion tons (www.fruitlogistica.de). However, the large amount of fruits and vegetables becomes lost due to postharvest losses. The postharvest losses (PHL) are the losses of the quantity and quality of the agricultural product throughout the postharvest chain. The PHL are accounted about 35-55% of the fresh agricultural produce depending upon the countries' geo-economic condition (Sanzani et al., 2016). The most adopted physical treatment to combat such losses is the cold storage (Usall et al., 2016). The most of the cold storages available today are equipped with refrigeration machines. Such systems are not only degrading the environment and consuming high primary energy but also cannot provide the optimal storage conditions (temperature and relative humidity) particularly to the tropical agricultural products and dry fruits (Mahmood et al., 2015; Mahmood et al., 2016b). It is because of diversified nature of agricultural products, chilling injury, and complex mechanism of transpiration, respiration and fermentation etc. In this perspective, low cost environmental friendly evaporative cooling technologies like direct evaporative cooling (DEC), indirect evaporative cooling (IEC)/M-Cycle evaporative cooling (MEC) have shown potential to provide storage conditions for particular agricultural products (Lal Basediya et al., 2013; Mahmood et al., 2016c). However, these technologies cannot be used effectively for the storage of agricultural products under largely varying ambient conditions (particularly humid) due to limited cooling performance (Lal Basediya et al., 2013). The scope of DEC and IEC/MEC for the storage of wide range of agricultural products under varying environmental condition can be extended by the integration of desiccant dehumidification. The desiccant air-conditioning (DAC) comprises of desiccant dehumidification cum evaporative cooling has ability to deal the latent and sensible load of air-conditioning distinctly. Such ability of the DAC system makes it promising to provide optimal conditions for the storage of agricultural products without affecting their quantitative, qualitative and nutritive attributes. The objective of this study is to find out the low cost, energy and environmental friendly appropriate cooling technologies for the storage of agricultural products and for other parallel agricultural applications.

B.2 Procedural Details

Strengths, Weaknesses, Opportunities, Threats (SWOT) analysis is carried out in this study. It is used to investigate the strengths, weaknesses, opportunities and threats

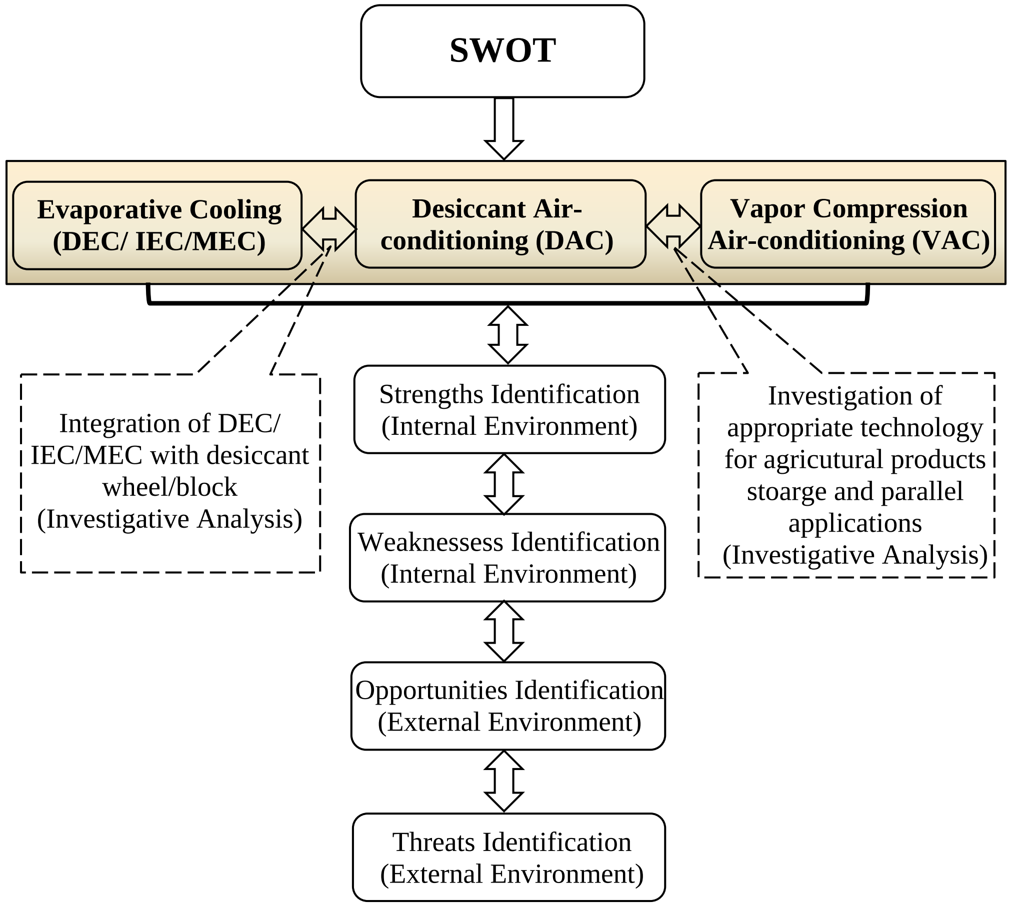


Figure B.1 Hierarchy of SWOT analysis of three air-conditioning dilemma for agricultural products storage.

associated with the evaporative cooling (DEC, IEC/MEC), desiccant air-conditioning (DAC) and compressor based air-conditioning (VAC) technologies in general and in particular when employed for the storage of agricultural products. SWOT is a strategic analysis tool and consists of internal and external factors/environment (Oxizidis and Papadopoulos, 2008). The internal factors examine the strengths and weaknesses of the technology with respect to its users and competitive systems, whereas the external factors (opportunities and threats) represent the individual perceptions about the technology future footings (Oxizidis and Papadopoulos, 2008). The key purpose of the SWOT analysis is to achieve the study objective by considering external and internal factors. Therefore, in the present study, SWOT analysis is carried out on the basis of extensive literature review

(Ming et al., 2014; Oxizidis and Papadopoulos, 2008; Paschalidou et al., 2016), experimental and theoretical studies published in the literature (Lal Basediya et al., 2013; Mahmood et al., 2015a; Mahmood et al., 2016a; Mahmood et al., 2016b; Mahmood et al., 2016c; Sultan et al., 2015) and discussion with the researchers working in the parallel field. The hierarchy of the SWOT analysis adopted in this study is shown Figure B.1.

B.3 Results and Discussion

In the present study the SWOT analysis is preformed to identify the prospects and challenges associated with agricultural products storage. The procedural details about SWOT analyses are explained in heading B.2. The strengths, weaknesses, opportunities and threats of different air-conditioning technologies are briefly elaborated in the Table B.1. The major strengths of the evaporative cooling technologies for storage of agricultural products include its suitability for the storage of most of agricultural products without causing chilling injury, discoloration, off-flavor (Lal Basediya et al., 2013; Mahmood et al., 2016a), low cost, and supply of 100% fresh air [Mahmood et al., 2016a; Mahmood et al., 2016c]. Consequently fermentation is avoided by means of ventilation which maintains the quantitative, qualitative and nutritive attributes of the stored products (Mahmood et al., 2016a; Mahmood et al., 2016b). However, these technologies consume water to supply the fresh cooled air therefore the availability of water at site must be ensured during the storage period. The direct evaporative cooling also cannot be promptly used to provide storage conditions for the dry agricultural products. Moreover, the cooling performance of evaporative cooling technologies depends on the ambient conditions. In human thermal comfort, mostly DEC and IEC are considered suitable technologies when ambient air wet bulb temperature is ≤ 20°C and ≤ 25 °C, respectively (Lal Basediya et al., 2013). However, the ideal design conditions for agricultural product storage may vary depending upon the nature of the storage. According to the literature MEC can work efficiently when ambient air dry bulb temperature and humidity ratio are up to 45°C and 11.2 g/kgDA, respectively (Mahmood et al., 2016c). Their (DEC/IEC/MEC) effectiveness decreases greatly in moderate and humid conditions. In humid areas the potential of all kind of evaporative cooling systems is limited, however, desiccant air-conditioning system could be a feasible

Table B.1 SWOT analysis of air-conditioning technologies.

1. Evaporative cooling

Quantitative Estimates

DEC (Direct Evaporative Cooling)

$$\left(T_{supply}\right)_h = \left(T_{inlet} - \varepsilon_{wb}\left(T_{inlet} - T_{inlet,wb}\right)\right)_h$$

$$h_{inlet} = h_{supply} = h\ (isenthalpic\ cooling)$$

$$T_{wb} \leq T_{supply} \leq T_{inlet,wb}$$

IEC (Indirect Evaporative Cooling)

$$\left(T_{supply}\right)_x = \left(T_{inlet} - \varepsilon_{wb}\left(T_{inlet} - T_{inlet,wb}\right)\right)_x$$

$$X_{inlet} = X_{supply} = X\ (sensible\ cooling)$$

$$T_{wb} \leq T_{supply} \leq T_{inlet,wb}$$

MEC (M-Cycle Evaporative Cooling)

$$\left(T_{supply}\right)_x = \left(T_{inlet} - \varepsilon_{wb}\left(T_{inlet} - T_{inlet,wb}\right)\right)_x$$

$$X_{inlet} = X_{supply} = X\ (dewpoint\ sensible\ cooling)$$

$$T_{dp} \leq T_{supply} \leq T_{inlet,dp}$$

Strengths (Internal Factor)

a) Huge primary energy savings and environmental friendly
b) Supply of 100 % fresh air, simple in construction and operation
c) Low installation, operational and maintenance cost
d) Possibility of making with local materials
e) Easy integration into constructed structures
f) Suitable for remote areas farming applications using renewable energy and storage of agricultural products
g) Suitable for on-farm precooling of agricultural products
h) Affordable by small/medium scale farming community
i) Higher dew point effectiveness and cooling (MEC only)

Weaknesses (Internal Factor)

a) Need of water availability,
b) higher cooler size and water consumption
c) Suitability depends on ambient conditions
d) Limited scope at humid areas (e.g. $X_{MEC} \leq 11.2$ g/kgDA, RH_{DEC} and $RH_{IEC} > 75\%$)
e) Danger of pad (media) blockage due to contaminated water
f) Not suitable for very high sensible loads
g) Short term storage of agricultural products
h) Not promptly suitable for dry fruit storage (DEC only)

$$\Delta P_{DEC} \leq \Delta P_{IEC} \leq \Delta P_{MEC}$$

$$\Delta P\ (pressure\ drop) = P_{inlet} - P_{supply}$$

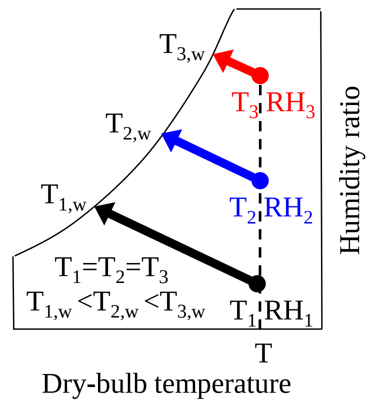
Opportunities (External Factor)

a) Strengthening of local small/medium industries

DEC

$$\varepsilon_{DEC} = \frac{T_{inlet} - T_{supply}}{T_{inlet} - T_{inlet,wb}}$$

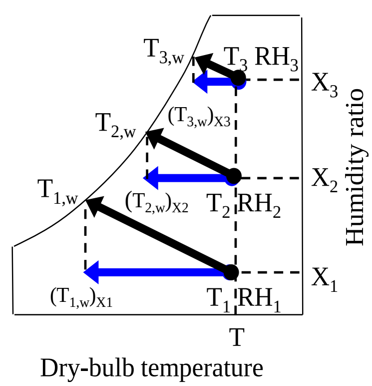
$$\varepsilon_{DEC} = 0.75 - 0.95$$



IEC

$$\varepsilon_{IEC} = \frac{T_{inlet} - T_{supply}}{T_{inlet} - T_{inlet,wb}}$$

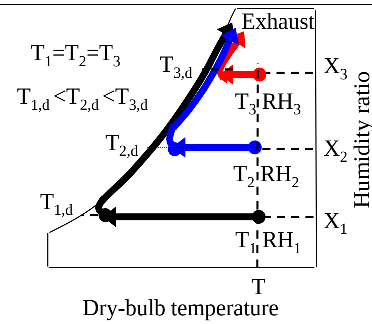
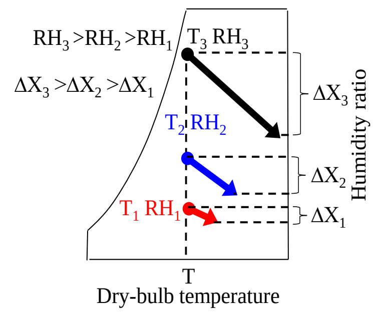
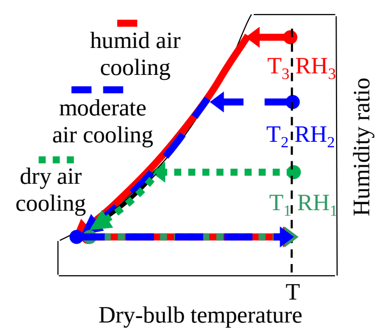
$$\varepsilon_{IEC} = 0.5 - 0.65$$



MEC

$$\varepsilon_{MEC} = \frac{T_{inlet} - T_{supply}}{T_{inlet} - T_{inlet,dp}}$$

$$\varepsilon_{MEC} = 0.5 - 0.6$$

b) Reduction in peak grid loads and dependence on fossil fuels c) Higher reduction in postharvest losses due to rapid pre-cooling provisions and improvement in farmers' livelihood d) Sustainable supply of products and market growth <u>Threats</u> a) Product price b) Competition with existing storage technologies	
2. Desiccant air-conditioning <u>Strengths (Internal Factor)</u> a) Savings in primary energy utilization and environmental friendly b) Distinct control of latent and sensible loads c) Most suitable for humid ambient conditions d) Driven by thermal energy (solar thermal etc.) e) Desiccant integrated MEC facilitates the dew point evaporative cooling under wide range of ambient conditions f) Flexible integration of desiccant with DEC/IEC/MEC according to the requirements of product storage zone g) Hybrid system design options h) Overall suitable for storage of dry and fresh agricultural products <u>Weaknesses (Internal Factor)</u> a) Bigger system size b) Higher initial investment and long payback period <u>Opportunities (External Factor)</u> a) Accelerate the research on desiccant materials b) Awareness of desiccant technologies <u>Threats (External Factor)</u> a) Barriers in market penetration & VAC market saturation	DAC 
3. Vapor compression air-conditioning <u>Strengths (Internal Factor)</u> b) Compatible system size c) Suitable for higher sensible loads a) <u>Weaknesses (Internal Factor)</u> b) Higher initial investment and requirements of primary energy c) Requirements of skilled labor d) Difficult to maintain the product storage zone conditions e) Not suitable for tropical agricultural products storage f) Less supply of fresh air (15% only) may affect the stored products quality g) Overall not effectively suitable for agricultural applications <u>Opportunities (External Factor)</u> a) Growth of VAC market and job employments <u>Threats (External Factor)</u> a) Product quality and environmental pollution	VAC 

choice in such scenario. In other words, one or other technology (evaporative cooling or desiccant air- conditioning) could yield low cost air-conditioning system. Studies have shown the potential of DAC system in humid and semi humid climatic conditions. It also gives opportunity to utilize low grade waste heat and renewable energy options particularly solar thermal energy. Lots of practical air-conditioning systems based on desiccant has been established and working all over the world. Mostly studied desiccant air-conditioning systems are standalone DAC systems with aid of simplified heat exchangers, direct and/or indirect evaporative cooling assisted DAC systems, vapor compression based hybrid DAC systems, solar energy operated DAC systems, bio-gas operated DAC systems, single/multi stage DAC systems, solid/liquid based DAC systems, wheel/block based DAC systems, heat and pressure regeneration based DAC systems and many more. The detail insights about DAC system can be found from reference (Sultan et al., 2015). The theoretical and experimental evaluation of the DAC system has been performed at low regeneration temperature (50-60°C) for storage of agricultural products under varying climatic conditions (Mahmood et al., 2015; Mahmood et al., 2016a, Sultan et al., 2014). It has been found in these studies that DAC can achieve the latent and sensible load of air–conditioning for the storage of agricultural products. Moreover, in case of higher sensible loads hybrid DAC system can be employed. In contrast to DAC and DEC/IEC/MEC, the refrigeration machines are being mostly used in the cold storages. Such systems, aside from high energy consumption and environmental degradation, cannot be used for the storage of agricultural products. It is due to uneven control of ambient relative humidity which ultimate effects quality attributes of the agricultural products. In addition due to excessive cooling chilling injury may occur in the product.

B.4 Conclusions

This study analyses the strengths, weaknesses, opportunities, threats (SWOT) related to the evaporative cooling, desiccant air-conditioning and compressor based air-conditioning technologies in general and in particular when employed for the storage of agricultural products. The resultant performance by all kinds of air-conditioning systems were based on ambient air conditions as well as nature of storage application. However according to the findings, it has been concluded that the one or other technology (evaporative cooling or desiccant air-conditioning) could yield low cost air-conditioning system for the storage of agricultural products. Moreover, these technologies (evaporative cooling or desiccant air-conditioning) can be used for the storage of agricultural products without causing adverse effects on the product qualitative and nutritive attributes.

B.5 Nomenclature

ΔP	pressure drop
DAC	desiccant air-conditioning
DEC	direct evaporative cooling
ε	effectiveness [-]
h	enthalpy [kJ/kg]
IEC	indirect evaporative cooling
MEC	M-Cycle evaporative cooling
PHL	postharvest losses [%] or [-]
RH	relative humidity [%]
T	temperature [°C or K]
VAC	vapor compression air-conditioning
X	humidity ratio [g/kgDA]

Subscript

d/dp	dew-point
w/wb	wet-bulb

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APPENDIX C

ENERGY RECOVERY IN CONDENSERS AND COOLING TOWERS

APPENDIX C

ENERGY RECOVERY IN CONDENSERS AND COOLING TOWERS

C.1 Maisotsenko Cooling Tower

Cooling tower is a heat rejection device which rejects waste heat to the atmosphere through the cooling of a water stream to a lower temperature (He et al., 2015; Tyagi et al., 2008; Tyagi et al., 2012). The most common applications for cooling towers are providing cooled water for AC, manufacturing and power generation processes. The schematic diagrams for conventional (CCT) and Maisotsenko (MCT) cooling towers are shown in Figure C.1(a)-(d) (Anisimov et al., 2014c; Gillan et al., 2011b; Khalatov et al., 2011; Morosuk et al., 2012). In addition, the corresponding psychrometric processes are elaborated on Figure C.2(a) and C.2(b) (Anisimov et al., 2014c; Gillan et al., 2011b), respectively. It can be seen that the CCT cools the hot water from the cooling tower to the ambient air wet-bulb temperature using DEC technique (Anisimov et al., 2014c; Qureshi and Zubair, 2006). On the other hand the MCT cools the water stream towards the dew-point of the ambient air (Anisimov et al., 2014c; Maisotsenko et al., 2005a; Morozyuk and Tsatsaronis, 2011; Morosuk et al., 2012; Wicker, 2003).

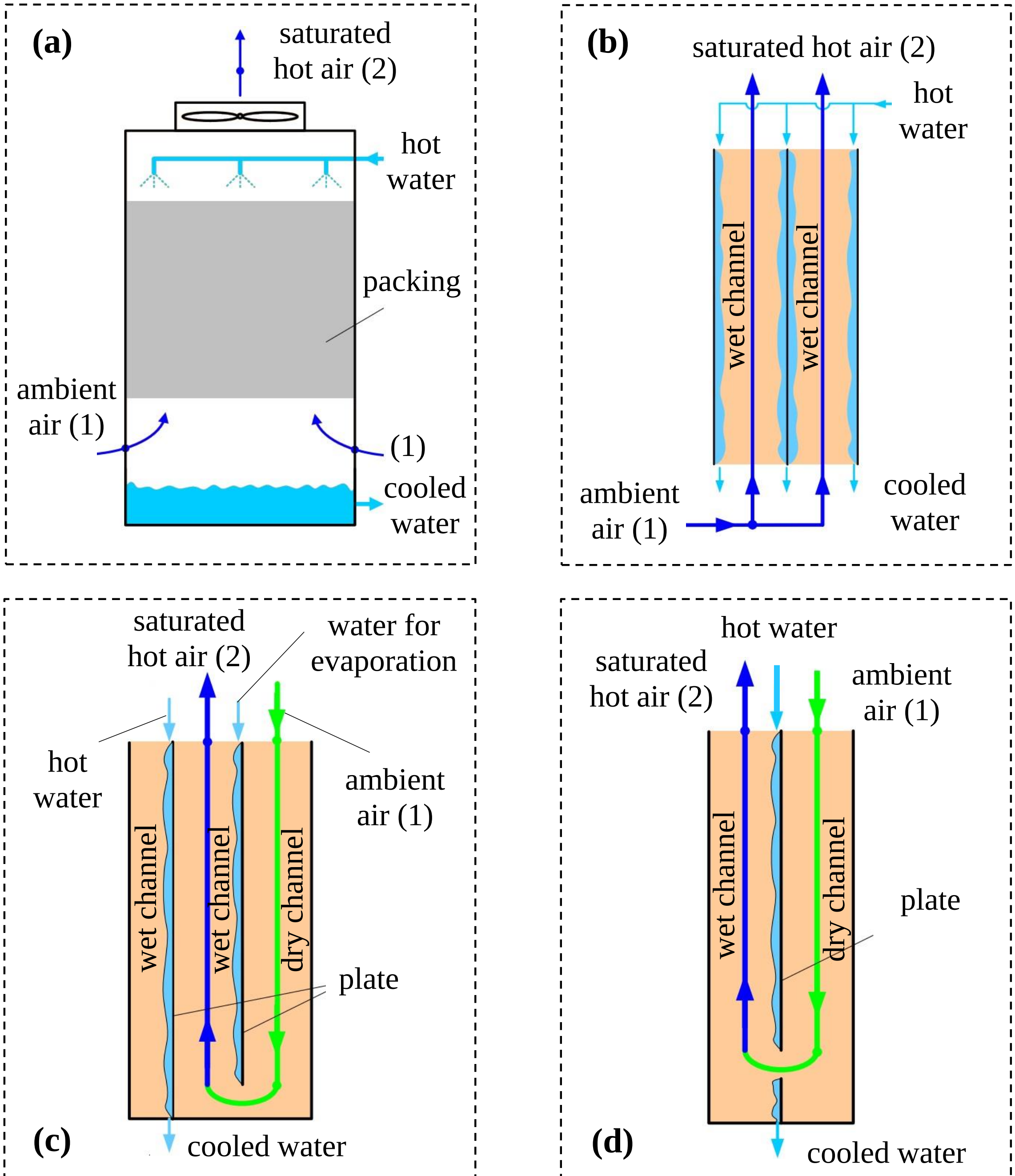

Figure C.1 Schematic diagram for: (a) general cooling tower flow scheme, (b) CCT, (c) close circuit MCT, and (d) open circuit MCT (reproduced from Anisimov et al., 2014c; Gillan et al., 2011b; Khalatov et al., 2011; Morosuk et al., 2012).

In both open and close circuit based MCTs, the air in dry-channel is sensibly cooled before it enters to the wet-channel for evaporative cooling which enables it to cool the water near to the dew-point temperature of the ambient air (Maisotsenko et al., 2005a; Morozyuk and Tsatsaronis, 2011; Sverdlin et al., 2011; Wicker, 2003). Furthermore, it also helps to reduce the pressure drop and fan power. The major differences between CCT and MCT are air flow route and water distribution through the fill (packing). Therefore, MCT fill can be designed as retrofit to the operational CCT in order to substantially decrease cooled water temperature.

The open and close circuit based MCTs are compared with the CCTs in order to analyze the M-Cycle applicability in cooling towers (Gillan et al., 2011b). In comparison with open circuit, the close circuit MCT has no direct contact of air with the process fluid i.e. water or a glycol/water mixture. Results showed that the both MCTs cooled the water at lower temperature than the CCT and enable double evaporation rate. Furthermore, the open circuit MCT has been preferably proposed by the authors because of its simple fill structure, and has been analyzed by many other researchers (Anisimov et al., 2014c; Khalatov et al., 2011; Morozyuk and Tsatsaronis, 2011; Sverdlin et al., 2011). Morosuk et al., 2012, simulated the COP of the MCT for various ambient air conditions. According to the results, the MCT yields the COP from 0.20 to 0.90 depending on the available conditions. Unlike the CCTs the COP of the MCT increases with the increase in ambient air temperature which distinguishes its applicability in hot climates. In another simulation based study (Anisimov et al., 2014c) the COP of the MCT was found up to 0.86 along with 2.20% evaporation rate. Furthermore, Sverdlin et al., 2011, analyzed the M-Cycle base cooling tower using a simulation program which was validated on the basis of real field data from existing cooling towers. It was concluded that the M-Cycle based cooling towers can conditionally reduce the water temperature up to the dew-point of the ambient air.

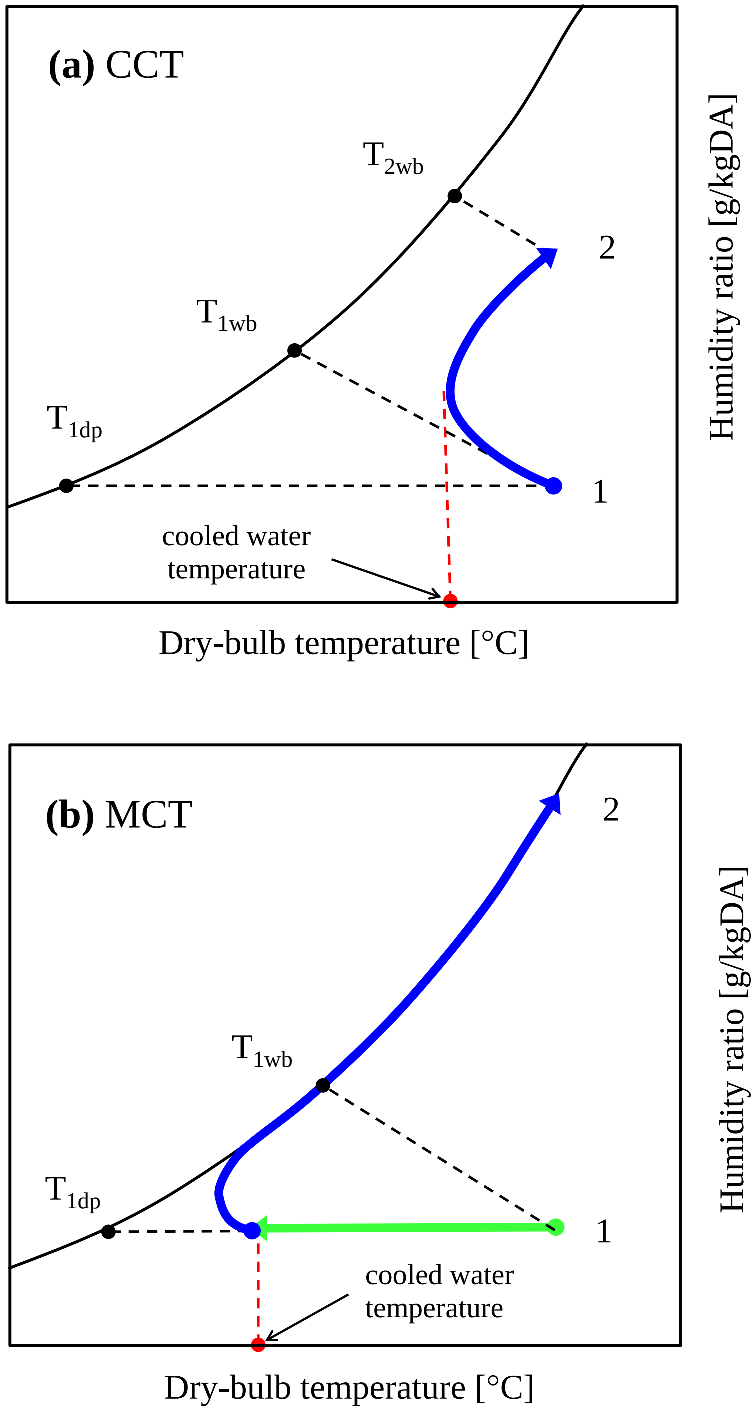

Figure C.2 Psychrometric representation of cooling tower operation for: (a) CCT; and (b) close/open circuit MCT (reproduced from Anisimov et al., 2014c; Gillan et al., 2011b).

C.2 Maisotsenko Condenser

One of the key component in vapor compression, absorption cooling and refrigeration system is the condenser, which could be air-cooled, water-cooled or evaporative condenser (Hajidavalloo and Eghtedari, 2010; Hosoz and Kilicarslan, 2004; Hwang et al., 2001; Li et al., 2009). The M-Cycle as an advance IEC process can be efficiently applied to the condensers of the air-conditioning, refrigeration and power producing systems (Maisotsenko et al., 2003; Maisotsenko et al., 2007). The Maisotsenko condenser (M-Condenser) is an evaporative condenser that can considerably enhance the energy efficiency of the system by rejecting the heat efficiently (Gorshkov, 2012; Khalatov et al., 2011; Maisotsenko, 2006; Idalex Technologies, 2006; Wani and Ghodke, 2011; Wani et al., 2012).

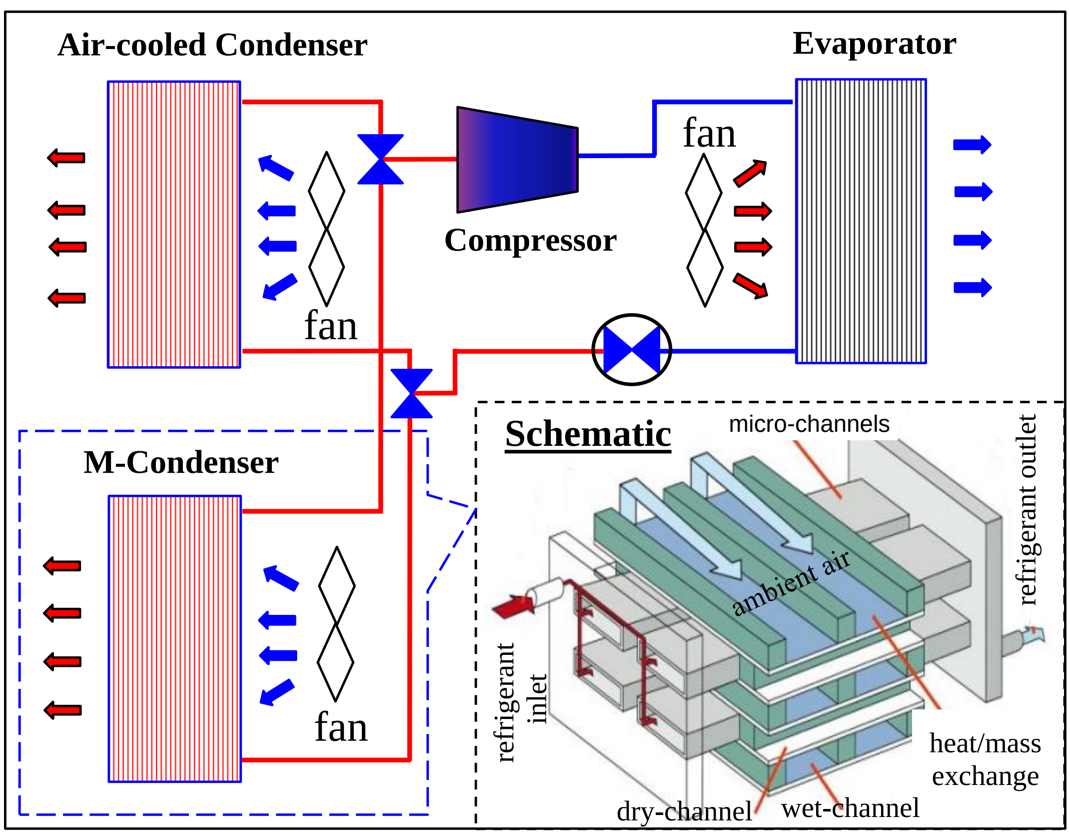

Figure C.3 Experimental setup for performance comparison between the air-cooled and M-Condenser (reproduced from Gillan et al., 2011a; Idalex Technologies, 2006; Maisotsenko, 2006).

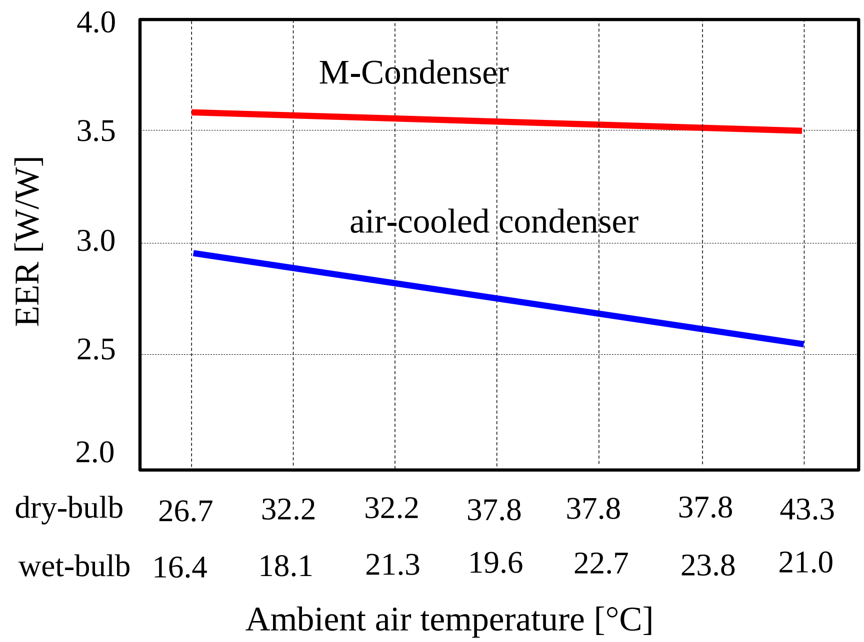

Figure C.4 Effect of ambient air conditions on the performance of air-cooled and M-Condenser (reproduced from Gillan et al., 2011a). Lines are obtained from the best fit of experimental data.

Gillan et al., 2011a, performed an experimental study on M-Condenser in order to compare its performance with the conventional air-cooled condenser. Figure C.3 (Gillan et al., 2011a; Maisotsenko, 2006; Idalex Technologies, 2006) shows the experimental setup which mainly consists of: (1) compressor, (2) air-cooled condenser, (3) M-Condenser, (4) fans, and (5) valves. The arrangement of values facilitated in determination of performance parameters of air-cooled and M-Condenser independently. The M-Condenser was composed of aluminum based micro channels for refrigerant flow, whereas the product and working channels are realized by cellulose sheets and plastic coating as shown in Figure C.3. Effect of ambient conditions on energy efficiency ratio (EER) was determined for air-cooled and M-Condenser as shown in Figure C.4 (Gillan et al., 2011a). It can be noticed that the EER by the M-Condenser remains stable by increasing the ambient air temperature from 26.7°C to 43.3°C whereas it decreases in case of air-cooled condenser. Furthermore, the M-Condenser obtained 30% higher EER (on an average) as compared to air-cooled condenser with a minimum of 9% and a maximum of 58%.

C.3 Nomenclature

CCT	conventional cooling tower
COP	coefficient of performance
DEC	direct evaporative cooling
EER	energy efficiency ratio
IEC	indirect evaporative cooling
MCT	Maisotsenko cooling rower

Subscript

dp	dew-point
wb	wet-bulb

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APPENDIX D

ADVANCES OF PSYCHROMETRIC RENEWABLE ENERGY IN POWER TURBINES

APPENDIX D

ADVANCES OF PSYCHROMETRIC RENEWABLE ENERGY IN POWER TURBINES

D.1 Maisotsenko Cycle Conception in Gas Turbines

The gas turbine usually consists of: (1) a compressor, (2) a combustion chamber, (3) a turbine, and (4) a generator (Cengel and Boles, 1994; Cohen et al., 1996). A simple layout of open cycle gas turbine operating on ideal Brayton cycle is shown in Figure D.1(a). Referring to T-S diagram shown in Figure D.1(b), the ideal Brayton cycle is made up of four completely irreversible processes i.e. (1➔2) isentropic compression; (2➔3) constant pressure heat addition; (3➔4) isentropic expansion; and (4➔1) constant pressure heat rejection. The detail insight of the operational principle can be found from the ref. [Yee et al., 2008].

The gas turbine with the working fluid of air-water mixture enables high electric efficiency and high specific power output to specific investment cost below that of combined cycles (Jonsson and Yan, 2005; Poullikkas, 2005). Various humidified gas turbine cycles have been proposed in the literature (e.g. direct water-injected cycles, steam-injected cycles, and evaporative cycles with humidification towers etc.), though only few are available commercially. In addition, the Maisotsenko gas turbine conceptions based on humidified gas turbine cycles have been recently proposed (Buyadgie et al., 2015; Gillan and Maisotsenko, 2003; Maisotsenko et al., 2004; Maisotsenko et al., 2005; Maisotsenko et al., 2006; Saghafifar and Gadalla, 2015a; Saghafifar and Gadalla, 2015b; Saghafifar and

Gadalla, 2015c). The details about particular gas turbine type can be found from the cited references. The key features of the humidified gas turbine can be explained as (Jonsson and Yan, 2005). Addition of water or steam increases the combustion chamber efficiency as well as mass flow rate passing through the turbine, and consequently enhances the specific power output. In this regard, the compressor work remains constant (if the water/steam is added after the compressor), and least work is required to increase the pressure of a liquid as compared to a gas. The overall cycle efficiency increases by utilizing the turbine exhaust heat for generating hot water and/or steam in order to preheat the combustion air in the recuperator. Addition of water before combustion reduces the compressed air temperature at the inlet to the recuperator, which improves the energy recovery rate. Furthermore, humidifying the gas turbine working fluid helps in reduction of NO_x formation during the combustion process.

From the above prospective the M-Cycle roles in the improvement of gas turbine efficiency can be listed as follows:

- ✓ To supply hot and humid air into the combustion chamber that improves the cycle efficiency, and consequently augments the power generation.
- ✓ To recover the thermal energy from the exhaust gasses in order to improve the cycle efficiency.
- ✓ To provide cooled air to the compressor for turbine inlet air cooling which improves the compressor efficiency.
- ✓ To reduce the NO_x formation during combustion process by providing excessive moist air.

In order to achieve the above mentioned M-Cycle roles, the coming headings explain the recently proposed three different Maisotsenko gas turbine cycles. It is worthy to mention that the M-Cycle is a heat recovery conception, and is not limited to the discussed cycles only. Its applications could be more versatile, though the study focuses only few cases because of the limited literature.

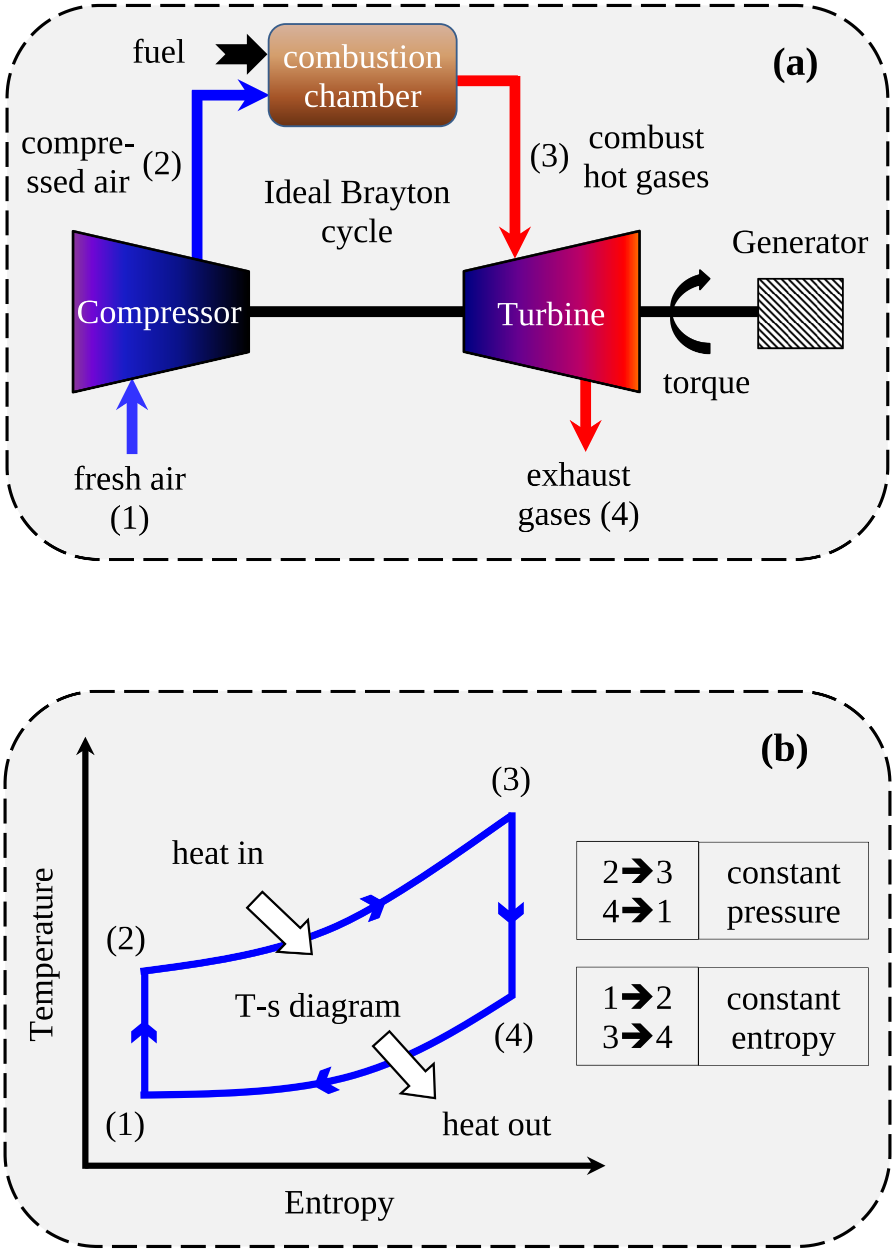

Figure D.1 Open cycle gas turbine operating on ideal Brayton cycle: (a) simple layout; (b) T-s diagram.

D.2 Maisotsenko Humid Air Turbine Cycle

The evaporative gas turbine or humid air turbine (HAT) cycle involves in injection of water in the humidification tower with a water loop recirculation system (Saghafifar and Gadalla, 2015a; Jonsson and Yan, 2005). The importance of HAT cycle is well-known in the literature and considered as one of the most efficient humidified gas turbine cycle (Jonsson and Yan, 2005). Gallo, 1997, compared the HAT cycle performance with the other gas turbine power cycles. According to the results the HAT cycle obtained optimum efficiency at moderate pressure ratio whereas the specific work output increases with the increase in pressure ratio (Saghafifar and Gadalla, 2015a; Gallo, 1997). Furthermore, it achieved the highest efficiency of 54.8% among the other studied cycles when the turbine inlet temperature and pressure ratio was about 1300°C and 12, respectively. Similarly another study (Lazzaretto and Segato, 2002) showed the maximum efficiency of 54.6% for the total pressure ratio of 20.

The Maisotsenko humid air turbine (M-HAT) cycle is one of recently proposed humidified gas turbine cycle (Gillan and Maisotsenko, 2003; Jenkins et al., 2014a; Jenkins et al., 2014b; Khalatov et al., 2011; Maisotsenko et al., 2004; Maisotsenko et al., 2005; Maisotsenko et al., 2006; Reyzin, 2011; Saghafifar and Gadalla; 2015a; Saghafifar and Gadalla; 2015b). The simplified schematic diagram of the M-HAT cycle is shown in Figure D.2(a). It works on evaporative gas turbine principle in which the humidification tower is replaced by the M-Cycle air saturator assembly (Khalatov et al., 2011; Reyzin, 2011; Saghafifar and Gadalla; 2015a) e.g. shell and tube air saturator (Gillan and Maisotsenko, 2003; Saghafifar and Gadalla; 2015a), as shown in Figure D.2(a). The operational scheme of the M-Cycle shell and tube air saturator is shown in Figure D.2(b) [Gillan and Maisotsenko, 2003]. Referring to Figure D.2, air at state (1) is compressed adiabatically to state (2) by the compressor. The compressed air enters into the M-Cycle air saturator where it is heated and humidified at state (3) by utilizing the waste heat from the turbine exhaust gases, and consequently supplied to the combustion chamber. The turbine exhaust gases enter into the M-Cycle air saturator at state (5) and exit at state (6). The detailed operational procedure can be found from (Gillan and Maisotsenko, 2003; Saghafifar and Gadalla; 2015a).

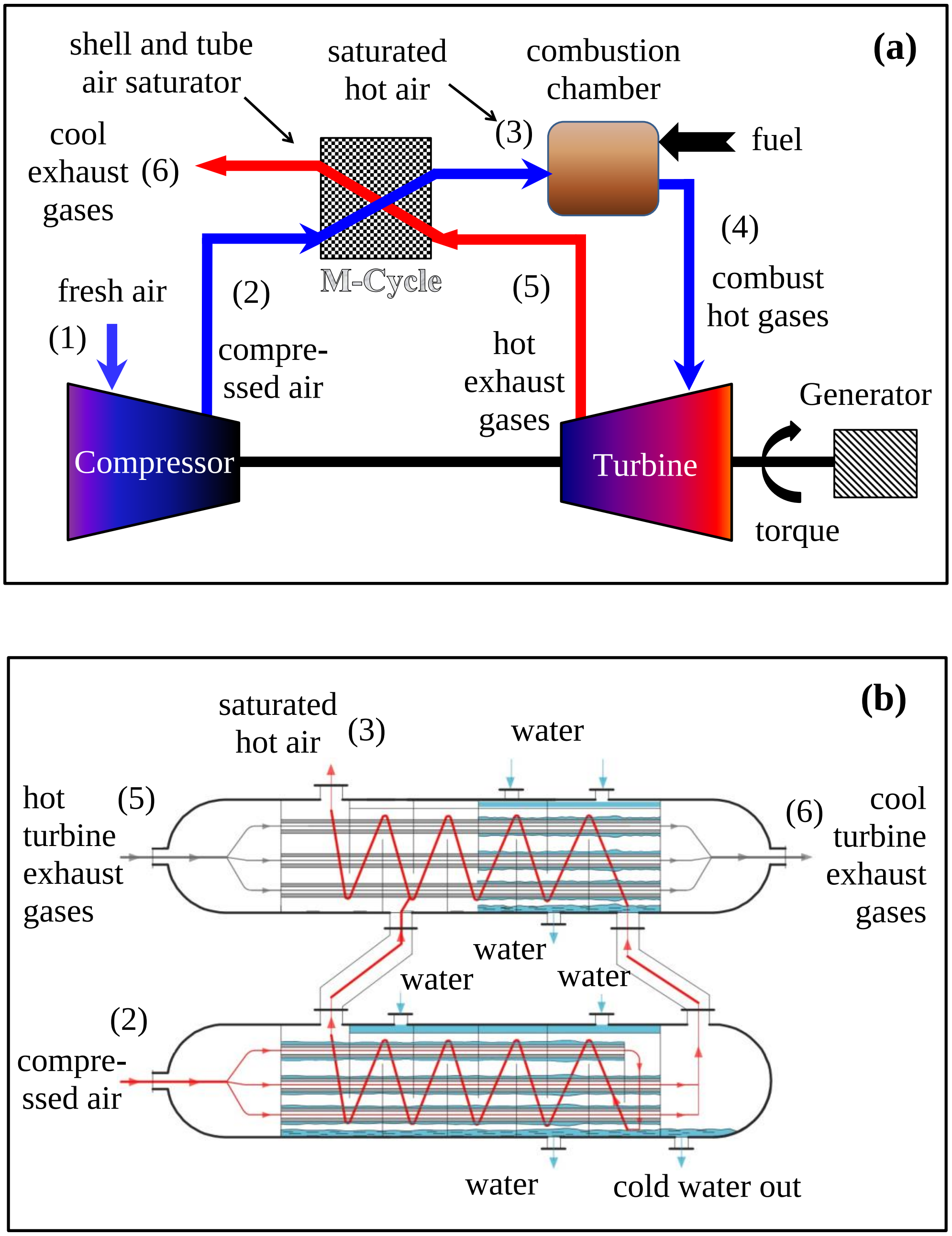

Figure D.2 Maisotsenko humid air turbine (M-HAT): (a) simplified schematic diagram of M-HAT cycle; (b) operational scheme of M-Cycle shell and tube air saturator (Gillan and Maisotsenko, 2003) in M-HAT cycle.

Saghafifar and Gadalla, 2015a, performed the detailed analysis of M-HAT power cycle with a comprehensive model of air saturator. The comparative analysis between HAT and M-HAT cycles has been conducted by the authors in order to optimize the effect of compressor pressure ratio, inlet air temperature, combustor outlet temperature, water inlet temperature, and air saturator degree of humidification. Maximum efficiency by the M-HAT was achieved when water addition in the upper section of the air saturator (Figure D.2b) was limited to cool the exhaust gases. It has been concluded that air saturator in comparison with conventional heat exchanger can increase the plant efficiency and specific work output by 7% points and 44.4%, respectively. In addition, the improvement results in 13,000 tonnes of natural gas fuel saving per year. Moreover, the M-HAT cycle possesses greater efficiency than the HAT cycle at higher pressure ratios. It is worth mentioning that the specific work output by the M-HAT cycle was continuously greater than one achieved by the HAT cycle regardless of compression pressure ratio. Similarly various studies (Gillan and Maisotsenko, 2003; Jenkins et al., 2014a; Jenkins et al., 2014b; Khalatov et al., 2011; Reyzin, 2011) conclude that the thermodynamic efficiency of M-HAT cycle is significantly higher than any humidified gas turbine cycle including the HAT cycle. Consequently, the present study concludes that the M-HAT cycle has challenged the HAT cycle for the title of optimal humidified gas turbine cycle.

D.3 Maisotsenko Air Bottoming Cycle

Conventional combined power cycles operating on gas turbine topping cycle and a steam turbine bottoming cycle are considered as the most efficient combined power cycles (Cengel and Boles, 1994; Cohen et al., 1996; Saghafifar and Gadalla, 2015b). However, it has been reported that the fact is not applicable for the small scale power plants with capacity of ≤ 50MWe (Poullikkas, 2005), because of the condenser and heat recovery steam generator in the steam bottoming cycle (Saghafifar and Gadalla, 2015b). In this regard, organic Rankine cycle (ORC) (Bianchi and De Pascale, 2011; Chacartegui et al., 2009; Clemente et al., 2013;; Kaikko et al., 2001) and air bottoming cycle (ABC) (Ghazikhani et al., 2014; Korobitsyn, 2002; Pierobon and Haglind, 2014; Saghafifar and Poullikkas, 2014) has been extensively studied in order to replace the steam bottoming cycle. The ABC cycle was proposed in the late 1980s and is well-known in the literature due to its simplicity and compact design (Saghafifar and Gadalla, 2015b).

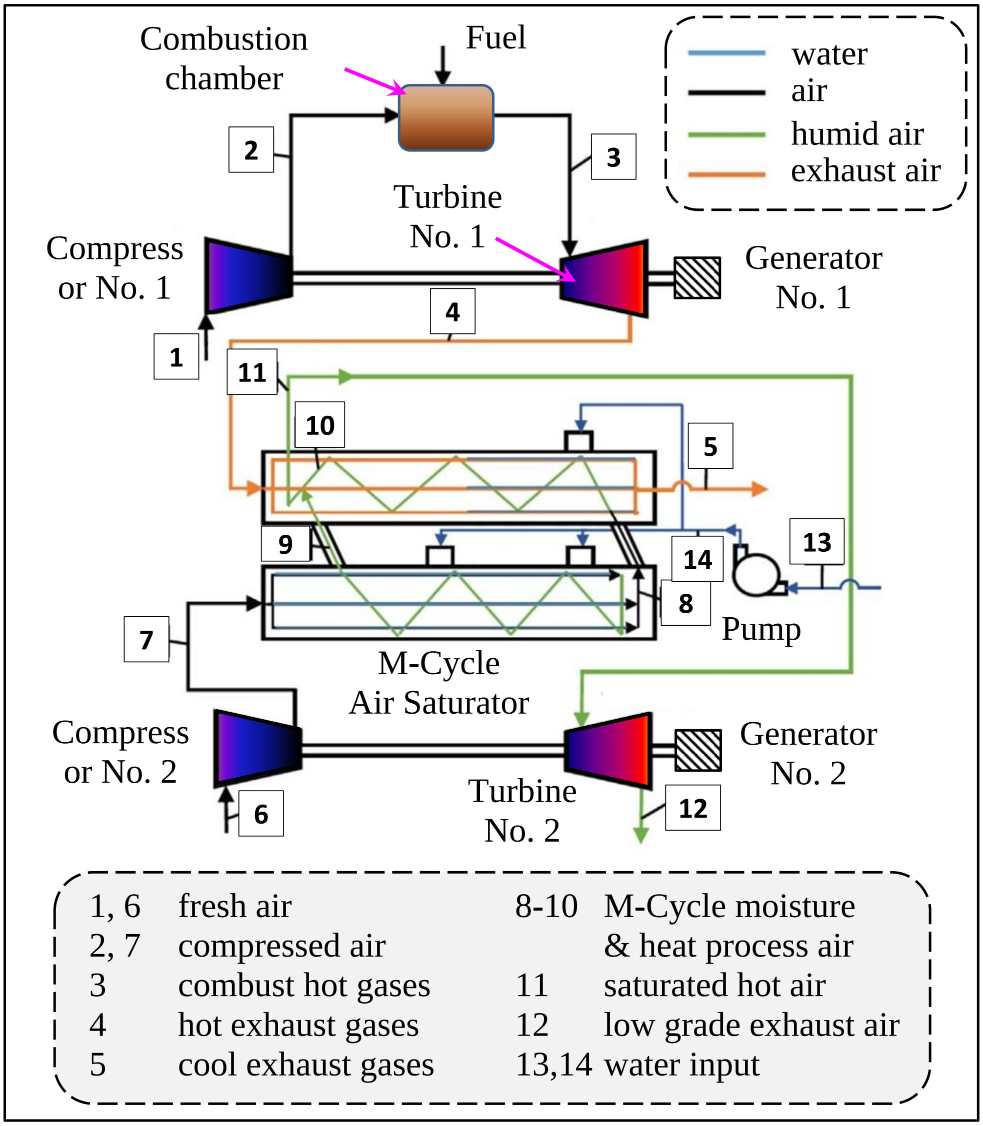

Figure D.3 Schematic diagram of the Maisotsenko air bottoming cycle (M-ABC) (reproduced from Saghafifar and Gadalla, 2015b).

Maisotsenko air bottoming cycle (M-ABC) (Saghafifar and Gadalla, 2015b) is a recently proposed ABC cycle which is an integration of Maisotsenko gas turbine cycle (Saghafifar and Gadalla, 2015a) as a bottoming cycle to a topping simple gas turbine cycle (Cengel and Boles, 1994; Cohen et al., 1996). In the M-ABC, the conventional air heat exchanger of the ABC is replaced by the M-Cycle air saturator cum heat exchanger. The schematic diagram of the M-ABC cycle is shown in Figure D.3 (Saghafifar and Gadalla, 2015b). It can be seen that the hot exhaust gases from turbine No. 1 enters into the M-Cycle air saturator cum heat exchanger at state (4) where it transfer its heat using the M-Cycle principle and exit at state (5). On the other hand the compressed air at state (7) is heated and humidified simultaneously by means of M-Cycle principle, and supplied to the turbine No. 2 at state (11). A comprehensive analysis of M-ABC is reported by Saghafifar and Gadalla 2015b. The authors developed a thermodynamic model for M-ABC with the detailed air saturator model. It has been concluded that the M-ABC enables higher efficiency as compared to conventional ABC at the optimum operating conditions. The specific work output by the M-Cycle air saturator was 43% higher than the conventional heat exchanger. Consequently, the efficiency was enhanced by 3.7%, which is equivalent to 2600 tonnes of natural gas fuel saving per year. Furthermore, the authors reported that the replacement of conventional ABC power plants with the M-ABC can yield savings of 0.655 US$M every year for the plant operating hours of 8000 hr/year and fuel cost of 5.5 US$/GJ. Hence, it has been concluded that the M-ABC has thermodynamic superiority over ABC, which can setup an advance combined power cycle.

D.4 Maisotsenko Sub-Atmospheric Brayton Cycle

The modern gas turbine engines are based on well-known open Brayton cycle (see Figure D.1) (Cengel and Boles, 1994; Cohen et al., 1996), which begins with air compression (in the compressor) followed by constant pressure heat addition (in the combustion chamber) and finally terminates in the gas turbine at the ambient pressure conditions. The overall disadvantage of the cycle is the significant amount of waste heat discharged into the atmosphere which results in poor thermal efficiency (Khalatov et al., 2015). In this regard, inverse Brayton cycle (Wilson and Korakianitis, 1984) have been investigated with different configuration in order to increase the overall cycle performance (Agnew et al., 2003; Alabdoadaim et al., 2006; Besarati et al., 2010; Zhang et al., 2009; Zhang et al., 2012a, Zhang et al., 2012b). In this cycle heated working medium at atmospheric pressure is initially expanded in the gas turbine. After that the working medium heat is recovered by the heat exchanger, and finally the cooled gas is sucked by the compressor to the atmospheric pressure. The cycle works below the atmospheric pressure and referred as sub-atmospheric cycle (Khalatov et al., 2015). It is believed that the reverse

Brayton cycle is not commercially feasibly because of the greater compressor size and employed higher operational energy (Khalatov et al., 2015). On the other hand, the M-Cycle as an innovative humidifying recuperator can significantly improve the cycle performance by providing extremely saturated hot air to the combustion chamber (before turbine) and cooled air to the compressor (after turbine) simultaneously (Buyadgie et al., 2015; Khalatov et al., 2015; Maisotsenko et al., 2004; Maisotsenko et al., 2006). Consequently, it will improve the fuel combustion efficiency as well as compressor efficiency at the same time. Furthermore, simple designs of atmospheric combustion chamber and cheaper materials could be employed in the turbine industry (Khalatov et al., 2015).

The Maisotsenko sub-atmospheric Brayton (M-SAB) cycle conception was realized recently by (Maisotsenko et al., 2004; Maisotsenko et al., 2006) in which the authors proposed various possible configurations of M-SAB cycle. On the basis of available literature, the present study discuses two kinds of M-SAB cycle which are based on: (1) compressor (Khalatov et al., 2015), and (2) ejector (Buyadgie et al., 2015). The compressor based M-SAB cycle is similar to the conventional reverse Brayton cycle. However, the compressor is replaced by the steam-air ejector in case of ejector based M-SAB cycle. The coming headings discuss these cycles one by one.

D.4.1 Compressor based System

Khalatov et al. 2015, analyzed the compressor based M-SAB cycle while recovering the turbine waste heat. The schematic diagram of the cycle is shown in Figure D.4 (Khalatov et al. 2015; Maisotsenko et al., 2004; Maisotsenko et al., 2006). The cycle configuration is similar to the (Maisotsenko et al., 2004; Maisotsenko et al., 2006), however an additional solar energy utilization unit is proposed for pre-heating (process 1➔2) in order to improve the cycle efficiency. The air is heated and humidified simultaneously by the M-Cycle assembly (process 2➔3) while recovering the turbine waste heat (process 5➔6). The saturated hot air improves the combustion efficiency as well as reduces the NO_x emission when used in combustion chamber (process 3➔4). The combust hot gases at atmospheric conditions are expanded in the gas turbine (process 4➔5). The energy from the hot gases at state (5) is recovered before it goes to the compressor by means of M-Cycle assembly (process 5➔6) and an additional heat exchanger (process 6➔7), which consequently improves the compressor efficiency. Analysis showed that the M-SAB cycle can achieve the thermal efficiency of 0.45 to 0.82 at pre-heating (T_2) and combustion temperature (T_4) of 40-90 °C and 160-340 °C, respectively. It is worth mentioning that the pre-heating shows significant improvement in thermal efficiency by the M-SAB cycle because of the versatile features of M-Cycle at higher temperature as explained in heading D.2. Unlike open Brayton cycle the higher regeneration rate promotes the thermal efficiency of the M-SAB cycle. The study concluded that the M-SAB cycle possesses higher efficiency as compared to conventional open Brayton cycle at certain conditions.

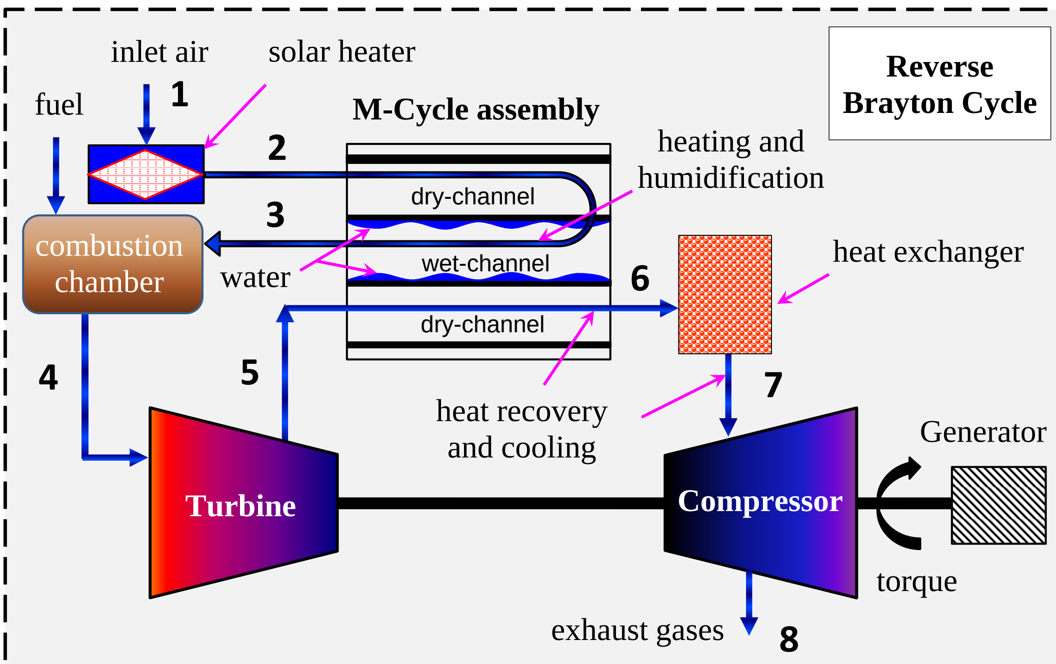

Figure D.4 Schematic diagram of the compressor based M-SAB cycle (reproduced from Khalatov et al., 2015; Maisotsenko et al., 2004; Maisotsenko et al., 2006).

D.4.2 Ejector based System

Buyadgie et al. 2015, proposed the ejector based M-SAB cycle and investigated its performance for various applications. The schematic diagram of the turbo-ejector based M-SAB cycle is shown in Figure D.5 (Buyadgie et al. 2015).

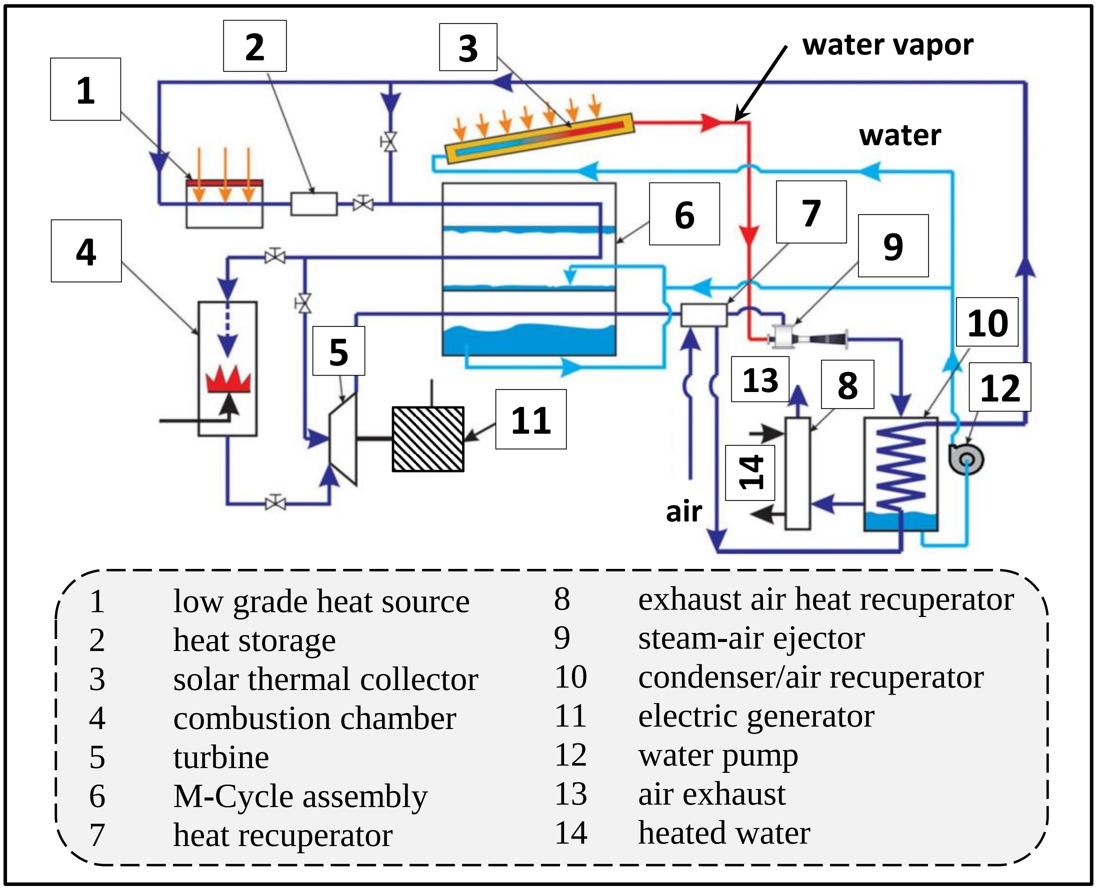

Figure D.5 Schematic diagram of the ejector based M-SAB cycle (reproduced from Buyadgie et al., 2015).

The principle operation of the cycle is similar the one based on compressor as explained in Figure D.4 (Khalatov et al. 2015; Maisotsenko et al., 2004; Maisotsenko et al., 2006), though the compressor is replace by the steam-air ejector. Each process of the cycle is labelled on the Figure D.5 which gives the detail insight of the cycle. According to the results the replacement of mechanical compressor with the steam-air ejector results in 2-4 times higher power generation, and yields 15% to 20% capital cost reduction of the system. In addition, the electricity used to operate the fans for the M-Cycle assembly decreases two times per power unit. The authors concluded that the turbo-ejector M-SAB cycle design is the optimum choice when the electricity price is high and heat price is low. Furthermore, it is more beneficial when the power generation and low temperature cooling is required simultaneously despite of the available heat cost.

From the above prospective the present study concludes that the M-Cycle possesses huge energy recovery potential in various power producing gas turbines. It addition to provide hot and humidified air for combustion, the M-Cycle recovers the turbine waste heat efficiently as compared to conventional heat exchangers. Furthermore, the nature of the M-Cycle helps to provide the cooled air to the compressor simultaneously, which increases the compressor efficiency. Another silent feature of the M-Cycle is the pollution control by reducing NO_x formation during combustion which can lead towards an environment friendly gas turbine power cycle.

D.5 Nomenclature

M-Cycle	Maisotsenko Cycle
HAT	humid air turbine
M-HAT	Maisotsenko humid air turbine
ABC	air bottoming cycle
M-ABC	Maisotsenko air bottoming cycle
M-SAB	Maisotsenko sub-atmospheric Brayton

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