Thermodynamic Balancing of the Regeneration Process in a Novel Liquid Desiccant Cooling System by Extraction Technology

BY

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This Thesis Is Dedicated To My

Beloved Parents For Their Prayers

And Encouragement, Everything That

I Do Will Come Short Compared To

What You Are Giving Me.

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All praise and gratitude to Allah lord of the worlds for blessing me with the intellect, strength and patience to attain this level of education.

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LIST OF ABBREVIATIONS

GOR : gained output ratio

HDH : humidification-dehumidification desalination system

Symbols

c_p : specific heat capacity at constant pressure (J $kg^{-1}K^{-1}$)

h : specific enthalpy (J kg^{-1})

h^* : specific enthalpy (kJ kg_{da}^{-1})

$\dot{H}$: total enthalpy rate (W)

HCR : heat capacity ratio (-)

$\dot{m}$: mass flow rate (kg s^{-1})

MR : desiccant-to-air mass flow rate ratio (–)

MRR : moisture removal rate (kg s^{-1})

NEG : normalized entropy generation (-)

$\dot{Q}$: heat input rate (W)

RR : recovery ratio (-)

s : specific entropy (J $kg^{-1}K^{-1}$)

$\dot{S}$: total entropy(W K^{-1})

T	:	temperature (°C)

Greek Symbols

Δ	:	difference
ε	:	effectiveness (-)
ξ	:	desiccant solution concentration ($kg_s\ kg_{sol}^{-1}$)
Φ	:	relative humidity (–)
Ψ	:	enthalpy pinch ($kJ\ kg_{da}^{-1}$)
ω	:	humidity ratio ($kg_{H_2O}\ kg_{da}^{-1}$)

Subscripts

1, 2, …	:	desiccant solution stream state points
a, b, …	:	air stream state points
ave	:	average
bot	:	bottom
carnot	:	at carnot efficiency
d'	:	moist air leaving the a condenser with $\Psi_n = 0$
da	:	dry air

ex	:	air state at which air is extracted from the regenerator and injected in the condenser in single extraction case
f	:	dehumidifier
ht	:	heater
hx	:	heat exchanger
i	:	inlet
id	:	ideal
k	:	k = 1 for conventional system and k = 7 for modified system
max	:	maximum
min	:	minimum
n	:	condenser
o	:	outlet
opt	:	optimum
pw	:	freshwater
r	:	regenerator
s	:	desiccant
sol	:	solution

st	:	strong concentration
tan	:	point on air curve at which the slope of its tangent equals slope of regenerator line
tan'	:	correspondent point at regenerator line to the tangent point at air curve
w	:	weak concentration

ABSTRACT

Full Name : Mohamed Ali Mahmoud Ahmed

Thesis Title : Thermodynamic balancing of the regeneration process in a novel liquid desiccant cooling system by extraction technology

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Air conditioning is a necessity in the Kingdom of Saudi Arabia due to extreme temperature and humidity values prevailing during the year. Therefore, the Kingdom is a major consumer of air conditioners with a share reaches up to 60 % in the Middle East. The air conditioning share is up to 70% of electric energy consumption of the Kingdom and it could be doubled by 2030. Therefore, cost-effective, environmentally friendly and energy efficient cooling techniques become essential requirement. In many regions of the Kingdom especially coastal areas such as Eastern and Western Provinces, the moisture content holding in the air is a major problem in air conditioning due to high latent load. This makes the Kingdom one of the most likely candidates for a country to be using liquid desiccant air conditioning systems.

In the conventional liquid desiccant air conditioning system, the scavenging air is thrown into the atmosphere carrying a considerable amount of energy and water vapor. Thus, there is plenty of room to improve the system performance by recover these losses. The proposed cooling system consists of a conventional liquid desiccant cooling system (a dehumidifier, a regenerator, a heater, and a cooler) plus a condenser with applying extraction and injection between the regenerator and the condenser.

The aim of this study is to reduce the energy consumption by recovering the heat from scavenging air using the condenser and also produces freshwater in addition to space cooling. In this research an attempt was made by applying extraction technique between the regenerator and condenser to increase the system performance. Lithium chloride (LiCl) is used as the liquid desiccant for this study. Mathematical formulation of simultaneous heat and mass transfer between the condenser and the regenerator was developed to study the effect of adding a condenser with/without extraction on the cooling system performance.

It is found that the performance of modified system without extraction is 11.25% better than the conventional system. This improvement in system performance vanishes when the condenser effectiveness is less than the critical value. To further improve the system performance, entropy generation in zero extraction cycle is reduced noticeably by bringing heat capacity rate ratio at extraction point to unity. Operationally, heat capacity rate ratio can be varied by changing desiccant-to-air mass flow rate ratio at terminal and intermediate points. This can be done by extract part of recirculated air from the regenerator and inject it to the condenser decreasing local losses through the condenser and regenerator.

Enthalpy pinch (losses through regenerator/condenser) has been introduced for analysis as a replacement to the temperature pinch that usually defined for heat exchangers. The coefficient of performance for both zero and single extraction systems decreases as enthalpy pinch increases (effectiveness decreases). Using the generated model, it is found that at Ψ= 20 kJ/kg dry air which is equivalent to ε = 0.97 for both regenerator and condenser, single extraction cycle performance is 85.7% better than zero extraction cycle and produces 94.2 kg of freshwater per hour as a by-product for the given conditions.

ملخص الرسالة

الاسم الكامل: محمد علي محمود أحمد

عنوان الرسالة: موازنة الديناميكا الحرارية لعملية التجديد في نظام تبريد مبتكر يعتمد على السوائل المجففة بإستخدام تكنولوجيا الإستخراج

التخصص: الهندسة الميكانيكية

تاريخ الدرجة العلمية: مايو، 2017

يعتبر تكييف الهواء ضرورة في المملكة العربية السعودية نسبة لدرجات الحرارة والرطوبة العالية على مستوى المملكة خلال السنة. لذلك، تعتبر المملكة مستهلك أساسي لأجهزة تكييف الهواء بحصة تصل إلى 60 % في الشرق الأوسط. تستهلك أجهزة تكييف الهواء 70 % من مجمل إستهلاك الطاقة الكهربائية في المملكة ومن المحتمل أن يتضاعف هذا الإستهلاك في العام 2030. لهذا، تقنيات تكييف الهواء صديقة البيئة، منخفضة التكلفة والمرشدة للطاقة أصبحت مطلب أساسي. في عديد من مناطق المملكة خاصة الساحلية منها، الرطوبة الموجودة في الهواء تعتبر مشكلة أساسية عند تكييف الهواء نسبة لإرتفاع الحمل غير المحسوس. كل ما سبق ذكره، يجعل المملكة مرشح مناسب كدولة تستخدم أنظمة التكييف المعتمدة على السوائل المجففة.

في نظام التكييف المعتمد على السوائل المجففة التقليدي، يتم رمي الهواء المسؤول من سحب الرطوبة من السائل المجفف بعد خروجه من المجدد إلى الجو مع العلم أنه يحوي كمية معتبرة من الطاقة والرطوبة. لذلك هنالك مجال واسع لتطوير اداء النظام عن طريق استعادة هذه الفقودات. النظام الذي ستتم دراسته في هذه الأطروحة، هو عبارة عن نظام تقليدي (مجفف، مجدد، سخان ومبرد) زائداً مكثف مع تطبيق تقنية الإستخراج والإدخال بين المجدد والمكثف. الهدف من هذه الدراسة، هو تخفيض الطاقة المستهلكة عن طريق إستعادة الحرارة من الهواء المطرود من المجدد بإستخدام المكثف وأيضاً إنتاج مياه صالحة للشرب بالإضافة لتكييف الهواء. في هذا البحث تم إستخدام تقنية الإستخراج بين المجدد والمكثف لزيادة أداء نظام التكييف حيث تم إستخدام كلوريد الليثيوم كسائل مجفف لهذه الدراسة. تم تكوين نظام حسابي لإنتقال الحرارة والكتلة بين المجدد والمجفف لدراسة أثر إضافة المكثف مع/بدون تقنية الإستخراج على اداء نظام التكييف.

لقد وجد أن اداء النظام المعدل بدون تقنية الإستخراج أفضل بنسبة 11.25 % من النظام التقليدي. هذا التحسن في الاداء ينخفض مع إنخفاض كفاءة المكثف حتى ينعدم إذا كانت كفاءة المكثف أقل من القيمة الحرجة. من أجل تحسين اداء النظام المعدل بدون تقنية الإستخراج، العشوائية الحرارية خفضت بصورة ملحوظة عن طريق جعل قيمة نسبة معدل السعة الحرارية عند نقطة الإستخراج مساوية للواحد. من الناحية التطبيقية، يمكن التحكم في نسبة معدل السعة الحرارية عن طريق تغيير نسبة معدل تدفق كتلة السائل المجفف إلى الهواء عند النقاط الخارجية والداخلية. هذه العملية تتم عن طريق أخذ جزء من الهواء المدار من المولد وحقنه في المكثف مما يقلل الفواقد الناتجة من تولد العشوائية الحرارية خلال المولد والمكثف.

الفقد في الطاقة (Enthalpy pinch) عبر المولد أو المكثف أُستخدمت في التحليل الحراري في هذا البحث كبديل للفقد في درجة الحرارة (Temperature pinch) التي تستخدم في العادة لتحليل المبادلات الحرارية. لقد وجد أن معدل الاداء للنظام المعدل مع/بدون تقنية الإستخراج ينخفض كلما زاد الفقد في الطاقة. بإستخدام النظام الحسابي، وجد أنه عند Ψ = 20 kJ/kg dry air حيث انها القيمة الموازية ل 0.97 = ε لكلا من المولد والمكثف، اداء النظام الذي يستخدم تقنية الإستخراج المفرد أفضل بنسبة 85.7 % من النظام التقليدي كما إنه ينتج 94.2 kg من الماء الصالح للشرب في الساعة كمنتج إضافي تحت الظروف التشغيلية المعطاة.

CHAPTER 1

INTRODUCTION

The Kingdom of Saudi Arabia is located between latitudes 16° and 33° N, and longitudes 34° and 56° E. In Kingdom regions, there are extreme seasonal variations in temperature and humidity over the course of the year. The air conditioning is a necessity in the Kingdom. Therefore, the air conditioning shares is up to 70% of electric energy consumption of the Kingdom and due to population increase, it will keep rising up in the future and it could be doubled by 2030 [1].

Freshwater is turning into a scarce resource in many regions around the world. There is about 40% of the world population suffering from the water shortage problems. While expanding population causes an increase in consumption of freshwater in these regions and it is expected to reach 60% by 2025 [2]. Therefore, water purification methods in conjunction with conservation efforts are needed in places where the levels of natural resources of potable water are diminishing [3]. For these reasons, the Kingdom of Saudi Arabia is facing two major problems that affect the existence and the comfort of its population. These problems are, high energy demand and water scarcity.

Therefore, cost-effective and environmentally friendly cooling techniques become essential requirement. These techniques can effectively utilize waste heat, solar energy, etc. The liquid desiccant air conditioning technology is one among them and capable to overcome electric consumption growth [4]. This cooling technology can fulfil the cooling demands controlling the building temperature and humidity efficiently. The liquid desiccant cooling system does not use ozone depleting refrigerants, making it eco-friendly [5].

1.1 Need of Alternative Air Conditioning Systems

Air conditioning (AC) is a process of controlling the air properties with the aim of well distributing of the air to the occupied space.

Air conditioning functions are:

- Temperature control by heating/cooling.
- Humidity control by dehumidification/humidification.
- Air cleaning.

The conventional AC systems which operate on the vapor compression cycle to provide cooling is energy-consuming and this energy is often produced from fossil fuels. However, the using of fossil fuels in the energy production leads to a large environmental damage represented in pollution, acid rain, etc. The conventional AC systems operate efficiently to overcome the sensible load, but it is inefficient for moisture content removal. In these systems, the humid atmospheric air will be cooled until its undesirable moisture is condensed out, often to air temperatures below the desired in the occupied space. Therefore, energy consumption is appreciable due to excess cooling.

In the central regions of Saudi Arabia, the climate is hot and dry, that is, high sensible load and low latent load. In these areas the need for cooling is high but dehumidification is not. Thus evaporative cooling is effective in such climatic conditions. However, plentiful available solar radiation can be effectively used for powering systems such as regenerative cooling, open cycle cooling system, etc.

In the coastal areas, such as Eastern and Western regions of the Kingdom, the moisture content holding on the air is a major problem in AC devices. The requirement of ventilating the spaces in addition to the high moisture content in the outdoor air rise up the latent load. By using the liquid desiccant cooling system, the dehumidification process can be carried out independently, and the sensible cooling process can be performed separately by using a direct/indirect evaporative cooling or a conventional AC system.

Therefore, it is needful to innovate new air cooling technologies. Air cooling systems that use desiccant dehumidification are highly energy efficient and have low impact on the environment. These systems can have good performance comparing to traditional AC systems [6].

The desiccant systems can provide as much as needed dehumidification for the ambient air. This dehumidified air has many applications such as air conditioning especially in humid areas, food preservation, storing chemicals and raw materials in pharmaceutical industry, crops drying, etc. [7]. In the dehumidification process, the vapor pressure of the humid air is higher than the strong liquid desiccant solution and hence moisture transfer takes place [8]. The desiccant vapor pressure is increased as a result of moisture absorption making the liquid desiccant a weak solution. Regeneration is required to re-concentrate the weak desiccant solution to continue the cycle. The dehumidification and regeneration processes are equally important parts of the liquid desiccant system, but regeneration process is the most critical part due to the association of energy consumption with it. Table 1.1 presents a comparison between conventional and liquid desiccant AC systems [9].

Table 1.1 Comparison between conventional and liquid desiccant cooling systems.

Factor	Vapor compression AC	Liquid desiccant AC
Operational cost	High	Low
Energy source	Electricity, Natural gas	Waste heat, solar energy or any low-grade heat
Moisture removal capacity	Average	Accurate
Air quality	Average	High
Installment of the system	Compact and easy	bulky
Effect on environment	Harmful	Eco-friendly

1.2 Innovation Component

The aim of this study is to reduce the AC energy consumption by using the proposed liquid desiccant cooling technology. In this research an attempt was made by applying extraction technique between the regenerator and condenser to increase the system performance. This system provides positive contribution to the environment protection and also produces fresh water in addition to cooling. This system can be widely used in many sectors like supermarkets, schools, office buildings, restaurants, etc.

1.3 Novelty of the Proposed Research

Past studies in humidification dehumidification desalination systems have shown that entropy production is decreased by using the extraction technique. In the proposed research the effect of extracting air from the regenerator and injecting it into the condenser to lower the entropy generation was examined. The aim of the process is bringing the enthalpy rates into balance. Extraction has not been carried out in a desiccant cooling system. The technique is expected to be a giant development in liquid desiccant air conditioning systems. The results are expected to show significant improvement in the desiccant cooling system performance. If this part of study succeeds, the results will be a breakthrough for desiccant air conditioning system and it will be a new dawn for AC. The findings and the outcome of this research are expected to be used as a seed for future research projects for the improvement of the liquid desiccant AC systems.

1.4 Research Objectives

The overall objective of this research is to reduce the energy consumption in AC by using the proposed cooling technology. In this research an attempt was made by applying extraction technique between the condenser and the regenerator in a novel liquid desiccant cooling cycle to increase the system performance and provide positive contribution to the environment protection. The specific objectives of this study are ordered in sequence and can be summarized as follows:

1. State of the liquid desiccant cooling systems.

2. Develop a mathematical model to predict the performance of the regeneration process using enthalpy pinch method.

3. Validation of the proposed model with the results available in the literature.

4. Mathematical formulation of simultaneous heat and mass transfer between the condenser and regenerator with/without extraction.

5. Parametric study for the proposed liquid desiccant cooling system.

CHAPTER 2

LITERATURE REVIEW

2.1 Liquid Desiccant Systems

Within the scope of existing research on desiccant cooling systems, a number of investigations by means of simulation studies have been accomplished. Dunkle [10] published a method of solar AC and describes the use of desiccants as a promising alternative air-conditioning technique, being combined with evaporative coolers and heat exchangers. Nelson et al. [11] carried out simulation studies regarding the feasibility and performance of open cycle desiccant systems using solid desiccants and solar energy. Barlow [12] conducted an analysis of solar desiccant systems and concepts and Duffie and Mitchell [13] performed a component and system evaluation study of solar desiccant cooling systems. Jain and Dhar [14] conducted simulation studies on various cycles for Indian climates. Davangere et al. [15] simulated a solid desiccant air-conditioning system with a backup vapor compression system and evaluated the performance of the system with regard to its feasibility in four cities in the United States. Daou et al. [16] presented an extensive overview on feasibility and performance studies of desiccant cooling systems focussing on system simulation. Schürger [17] carried out investigations into solar powered adsorption cooling systems and developed a simulation model for desiccant wheels. Kohlenbach [18] discussed modelling and simulation of the direct evaporative cooler (DEC) system performance in particular systems or applications.

Gao et al. [19] investigated the performance of a Celdek structured packing dehumidifier using LiCl as a liquid desiccant. The study shows the desiccant inlet parameters have more effect on the performance of the dehumidifier than air parameters. They obtained correlations for enthalpy and moisture efficiencies by using experimental regression. Peng and Zhang [20] studied the simultaneous heat and mass transfer in a liquid desiccant system that uses direct solar regenerator with an air pretreatment unit to reduce regeneration temperature. The simulation results show 70% increase in concentration of outlet solution than traditional direct solar regenerator. Tu et al. [21] proposed and developed a mathematical model for a novel liquid desiccant air conditioning system that utilizes LiCl as liquid desiccant. The system consists of a two packed towers one is used for the desiccant regeneration and the other for the air dehumidification. In their system the dehumidified air is cooled using direct and indirect evaporative coolers. They found that, the system performance is highly increased when the values of the input variables are optimized.

Figure 2.1 shows schematic of the conventional liquid desiccant air conditioning system. This system consists of a dehumidifier, a DEC, a regenerator, a heater, and a heat exchanger [22]. The moisture transfer potential is a function of the desiccant solution vapor pressure which depends on the desiccant temperature and concentration. In this conventional cycle, the process air is cooled in a direct evaporative cooler after the dehumidification process whereas the scavenging (ambient) air is humidified as a result of the regeneration process. The state points of the process air and scavenging air are represented on the psychrometric chart as shown in Fig. 2.2. The moisture from humid process air (state *a*) is removed in the dehumidifier by the strong desiccant solution (state 6). Due to dehumidification process, the process air temperature at state *b* is higher than air at state *a*. The dehumidified air is cooled in the direct evaporative cooler before it is introduced into the space to be conditioned (state *c*). The diluted desiccant (state 1) that leaves the dehumidifier is warm because heat is liberated due to water vapor condensation in the dehumidifier. This diluted desiccant is preheated in the heat exchanger to state 2 and then passed through a heater where its temperature is raised to regeneration temperature (state 3). In the regenerator, the scavenging (ambient) air is exposed to the weak and hot desiccant.

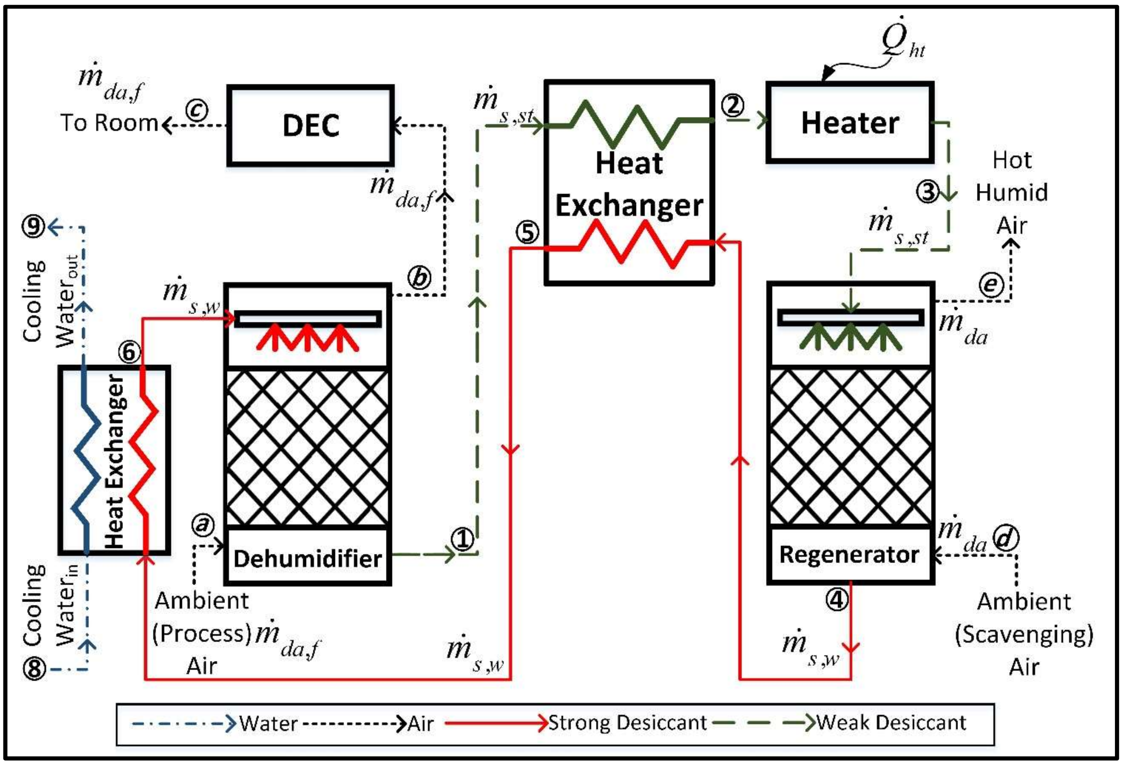


Figure 2.1 Schematic of the conventional liquid desiccant air conditioning system.

At state 3, the desiccant solution vapor pressure is higher than the scavenging air (state *d*) vapor pressure. The moisture content in the weak desiccant solution is transferred to the scavenging air and as a result, air leaves the regenerator (state *e*) hot and humid. In this cycle air at state *e* is thrown into the atmosphere after regeneration carrying a considerable amount of energy and moisture. The concentrated desiccant solution that leaves the regenerator (state 4), is precooled in the heat exchanger to state 5 and is then further cooled in water – desiccant solution heat exchanger (state 8-9). The cool and strong desiccant solution (state 6) is sent to the dehumidifier to continue the cycle.

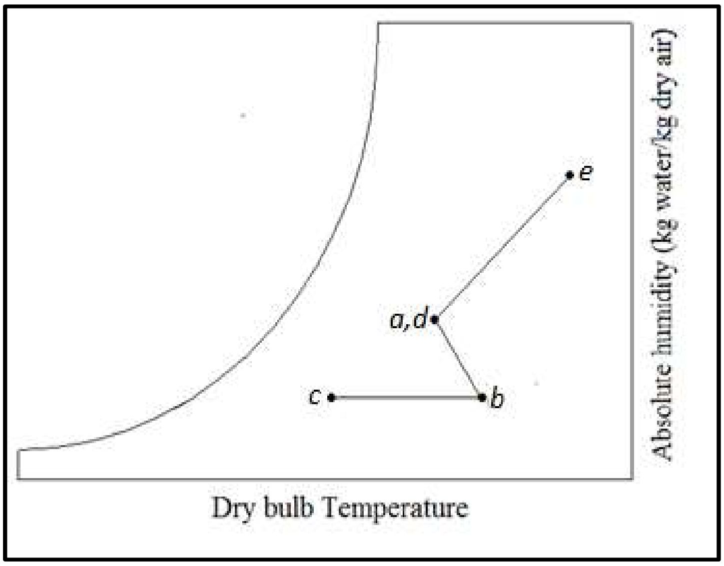


Figure 2.2 Air state points representation on the psychrometric chart.

2.2 Liquid desiccant materials

Nearly any material is a desiccant even glass can collect a small amount of moisture, but desiccants used in commercial equipment are selected for their ability of holding large amount of moisture eagerly. For example, the silica gel packets often sealed into vitamin bottles can hold moisture equal to about 20% of their dry weight. Liquid desiccants can hold even more moisture. As desiccants are used just for commercial purposes, so the definition of desiccant should include only the materials that have high affinity for water vapour due to the vapour pressure difference between desiccant and air. Using desiccants, moisture removal occurs in the vapour phase which means there is no liquid condensate so the process can continue when the dew point of the air is below freezing point. Desiccants can be either liquid or solid, and there are many different materials of both types.

- Solid Desiccants

Solid desiccants such as silica gel, activated alumina, activated bauxite, micro sieves, etc., which have great affinity for water vapour are used for producing dehumidified air. Different types of dehumidifiers and regenerators are available for solid desiccants such as multiple vertical bed, rotating horizontal bed, solid packed tower and rotating honeycombe®. The air to be dehumidified is passed through the desiccant bed, the moisture in the air is condensed out in the pores of the desiccant, and then these beds need to be regenerated so gas, steam or electrical energy are used. The regeneration temperature used for solid desiccant systems is about 80 - 90 ^{0}C.

- Liquid Desiccants

Liquid desiccants have many advantages over solid desiccants, the most important are liquid desiccants require low regeneration temperature which allows the use of the low-grade energy [4] and also concentrated liquid desiccants can be stored to utilize when the thermal energy source is unavailable [23]. Properties such as surface vapor pressure, dynamic viscosity and regeneration temperature control effectiveness of the liquid desiccant. Vapor pressure is the most influential parameter among the aforementioned parameters and it represents the mass transfer potential in the regeneration process [24]. In general terms, liquid desiccants are non-toxic and odorless. Frequently used liquid desiccants are lithium bromide (LiBr), calcium chloride ($CaCl_2$) and lithium chloride (LiCl) [25]. Absorption ability of LiBr is higher than $CaCl_2$ due to its relatively high vapor pressure [26]. However, $CaCl_2$ is common because of its availability and low cost. Liu et al. [27] performed a comparison between the performance of LiCl and LiBr solutions; they found that LiCl performance is better than LiBr due its low vapor pressure. Thus, LiCl is used as the working liquid desiccant solution for this study.

Liquid desiccants advantages are [28]:

- Low pressure drop across the system.
- Liquid desiccants regeneration temperature is low comparing to solid desiccants which allows the using of the low grade energy.
- The liquid desiccant provide purification beside dehumidification for the process air.
- Concentrated liquid desiccant solution can be stored to be utilized when the thermal energy source is unavailable.

2.3 Desiccant regeneration method

A regeneration unit is utilized to reconcentrate the weak liquid desiccant into strong liquid desiccant. Water moisture is transferred from the liquid desiccant to scavenging air [29]. There are three main regeneration methods as shown in Fig. 2.3. The three major regeneration methods are thermal energy regeneration, regeneration by electro dialysis and reverse osmosis (RO) regeneration [30].

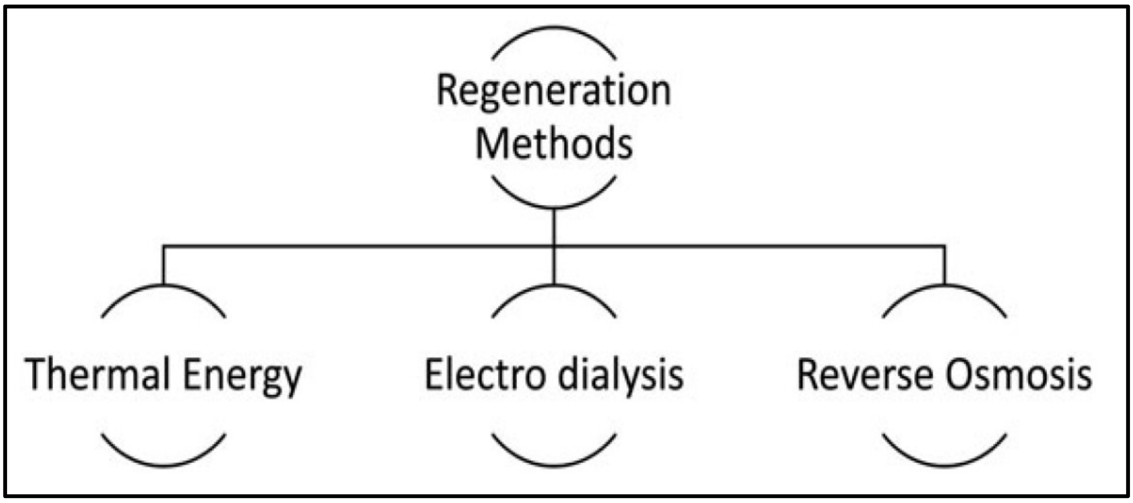


Figure 2.3 Regeneration methods [28].

- Regeneration by electro dialysis:

Since many liquid desiccants can be labeled as electrolyte solutions (such as LiBr, $CaCl_2$ and LiCl), electro dialysis could be a useful measure as a regeneration method [31]. Cheng et al. [32] investigated experimentally the performance of an electrodialysis regeneration unit of the liquid desiccant. In electrodialysis method, electric field transfers ions through a selective membrane. It is found that 55% of the maximum current was utilized by the system.

- Reverse osmosis regeneration:

In seawater desalination reverse osmosis is commonly used. Likewise, diluted liquid desiccant solution can also be converted into a concentrated solution using RO process. Al-Sulaiman et al. [33] proposed RO system with MFI zeolite membrane to separate the added moisture from the weak liquid desiccant solution.

- Thermal energy regeneration:

The most common regeneration method is thermal energy regeneration method [28]. Either air or liquid desiccant or both can be heated using thermal energy regeneration. Regeneration unit works mostly as an adiabatic process. However, Yin and Zhang [34] proposed a system where the regenerator is internally heated and thermal energy is provided by a heating coil in order to keep constant solution temperature along the regenerator.

2.4 Adding condenser to the liquid desiccant cooling system

The recovery of both moisture and heat losses from the scavenging air in the conventional system has not been studied in detail. The recovery of thermal energy has been carried out by many researchers [21], [35-38] while the recovery of moisture in the scavenging air has been studied by Audah et al. [26]. As shown in Fig. 2.4, the system proposed in [26] consists of two major parts; namely a conventional desiccant cooling system and a condenser. The hot and humid scavenging air that leaves the desiccant cooling system is passed through a condenser to capture some of its moisture as a freshwater. They could not be able to recover thermal energy because in their system seawater is used as the cooling stream in the condenser.

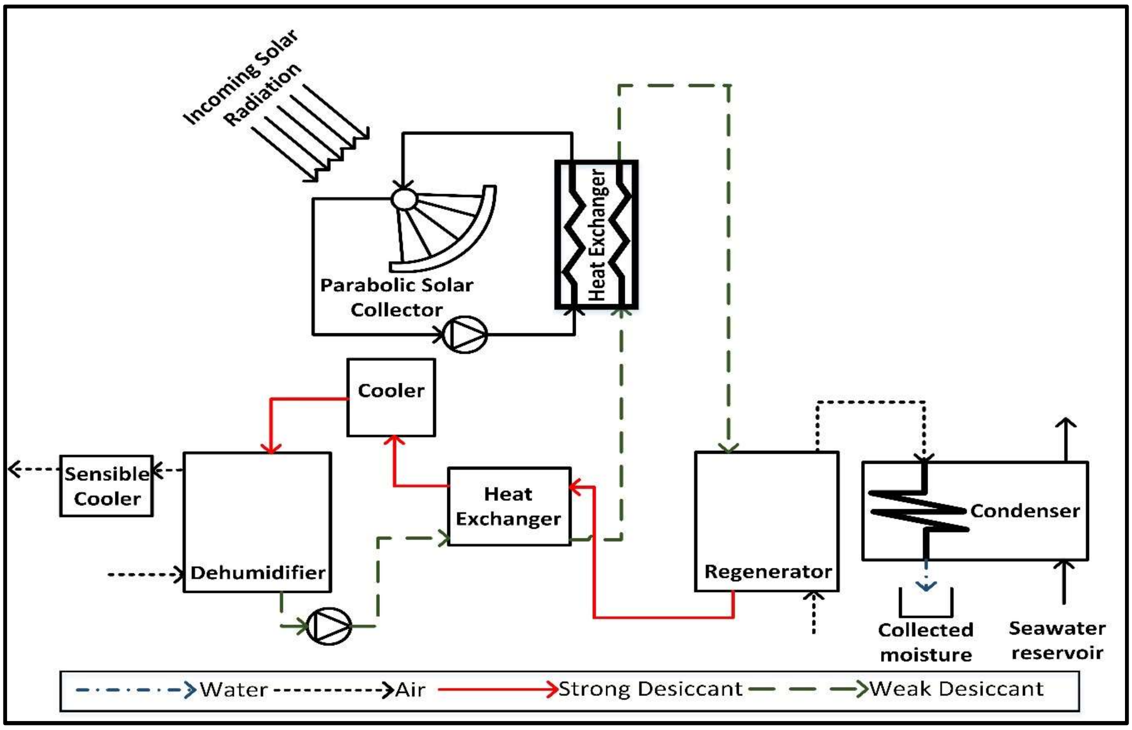


Figure 2.4 Schematic of the liquid desiccant system proposed in [26].

2.5 Extraction and Injection Processes

No research has been carried out to study the effect of extraction technology in liquid desiccant cooling systems. The extraction process has been inspired from desalination field where it was used to improve the humidification dehumidification (HDH) desalination system performance.

Müller–Holst [39, 40], proposed an extraction system by varying the water-to-air mass flow rate ratio continuously in order to achieve thermal balancing of HDH systems. This variation will decrease stream-to-stream temperature difference. He made use of natural convection to circulate the moist air stream through ports in both the humidifier and dehumidifier. This circulation resulted in variation of the water-to-air mass flow rate ratio. After optimization, the system's specific energy consumption was 120 kWh/m^3 (≈ 450 kJ/kg). Another novel approach for thermodynamic balancing was introduced by Zamen et al. [41]. They designed a multi-stage process, in which, HDH processes are executed in a sequence. The brine flow was common for all the stages, while the air flow was separate for each stage. Schlickum [42] and Hou [43] reported a similar design. Zamen et al. [41] have defined the system by the temperature pinch approach, which is commonly used in process industries. The total specific heat consumed by this system was about 800 kJ/kg. The humidifier and dehumidifier both had a temperature pinch of 4 °C, at a top- and bottom-cycle temperature of 70 °C and 20 °C, respectively.

Another novel HDH system driven by forced convection was invented by Brendel [44], [45]. Under balanced temperature profiles, forced convection was used to extract water from the dehumidifier and was injected in the dehumidifier. This extraction process was executed at several points in both the humidifier and dehumidifier. Thiel and Lienhard [46] have stated that the optimization of heat and mass transfer exchanger (HME) devices thermodynamically require considering both the temperature and concentration profiles. They have shown that balancing humidity profile have more significance in the optimization of the system than balancing the temperature profile. Forced convection driven HDH systems with air extraction and injection have also been investigated by Younis et al. [47]. They have succeeded to increase the system efficiency as the specific energy consumption decreased to 800 kJ/kg. In their system, the air was extracted from two points in the humidifier and injected to the dehumidifier. They followed enthalpy-temperature (*h-T*) diagrams, as used in several other publications [39, 41, 44, 48] to illustrate the extraction impact on the design of HDH system.

Thermal balancing by extracting air or water from the humidifier and injected it into the dehumidifier or vice versa has been investigated by Narayan et al. [49]. Mistry et al. [50]

found that reducing the specific entropy would result in minimizing the gained output ratio (GOR). Miller and Lienhard [51] studied the effects of extraction on balancing enthalpy rates in HDH systems. They followed an effectiveness-based methodology. Their main conclusion was that extractions are better for systems that have a high effectiveness in both the humidifier and dehumidifier of an HDH system.

The variation of temperature pinch effect on both the recovery ratio (RR) and GOR has been studied by McGovern et al. [48]. They showed an increase in GOR from 3.5 to 14 by incorporating single water extraction by assuming that effective heat and mass transfer area to be very large. That increase was achieved by using bottom- and top-cycle temperatures of 25 °C and 70 °C, respectively. Furthermore, for a single water extraction and under same operating conditions, they reported an increase in RR from 7% to 11%.

Narayan et al. [52] defined a novel parameter called the enthalpy pinch approach. They used this parameter to balance HME devices since this parameter takes into account both heat and mass transfer processes that are occurring in HDH systems. Balanced systems that have zero extraction, one extraction and an infinite number of extractions were studied using the enthalpy pinch approach. An increase in the GOR from 2.6 to 4.0 for a system with single air extraction has been reported in an experimental study by Narayan et al. [53]. In their experimental study, the enthalpy pinch was 19 kJ/kg of dry air. The bottom- and top-cycle temperature were 25 °C and 90 °C, respectively.

Chehayeb et al. [54] in continuation of the previous work by Narayan et al. [52] have investigated the effect of extractions on the GOR, RR, and the total heat input to the cycle. They examined a finite number of extractions and found that the smaller the enthalpy pinch, the larger would be the impact of balancing. That is, when the heat and mass transfer areas decrease (large enthalpy pinch), the balancing loses its significance. Furthermore, they found that balancing effect on water recovery is not significant compared to its effect on energy efficiency. Chehayeb et al. [55] studied in another investigation the effect of extraction on a fixed-size HDH system with a two-stage HDH processes. A generalized energy effectiveness for heat and mass exchangers (HME) devices was proposed in their study. The model was constructed from a multi-tray bubble-column dehumidifier and a packed-bed humidifier. The results pointed out that thermodynamic balancing maximizes both the GOR and water recovery while keeping entropy generation at minimum levels. Furthermore, it was mentioned that the direction of extraction should always be from the humidifier to dehumidifier to reach a balanced system. It is important to note that the work in the literature regarding air extraction has focused mainly on a water heated, closed air - open water cycle as a basic configuration.

CHAPTER 3

Thermodynamic study of the effect of adding a condenser to the liquid desiccant cooling system

3.1 Summary

The liquid desiccant air conditioning systems are cost-effective, environmentally friendly and energy efficient techniques, especially in coastal areas. In the conventional liquid desiccant air conditioning system, the scavenging air is thrown into the atmosphere carrying a considerable amount of energy and water vapor. Thus, there is plenty of room to improve the system performance by recover these losses. The proposed system consists of a conventional liquid desiccant air conditioning system plus a condenser. The aim of this study is to reduce the energy consumption by recovering the heat from scavenging air using the condenser and also produces freshwater in addition to space cooling. LiCl is used as the liquid desiccant for this study. Mathematical formulation of simultaneous heat and mass transfer between the condenser and the regenerator was developed to establish a comparison between the performance of the conventional and modified systems. Using the generated model, it is found that the modified system performance is 11.25 % better than the conventional system and produces 86.4 kg of freshwater per hour as a by-product for the given conditions.

3.2 Introduction

In the conventional liquid desiccant air conditioning system [22] as depicted in Fig. 2.1, the weak desiccant solution loses some of its energy and moisture contents to the scavenging air. This air is thrown into the atmosphere carrying a considerable amount of energy and water vapor. The recovery of these potential losses is the key to reduction of energy consumption. This chapter proposes a new configuration for the liquid desiccant air conditioning system in which these both losses are recovered.

The objective of this chapter is to examine the reduction of energy consumption in the regeneration system by using the proposed cycle increasing the system performance and provide a positive contribution to the environment protection in addition to supplying potable water.

3.3 Modified liquid desiccant air conditioning cycle description

Figure 3.1 shows the modified desiccant-based air conditioning system containing a dehumidifier, a DEC, a regenerator, a heater, a heat exchanger and a condenser. The difference in components between the conventional and proposed system is that a condenser is added in the modified system. This condenser is used to preheat the diluted desiccant solution from state 7 to 1.

Desiccant solution vapor pressure at various state points of the cooling cycle is shown in Fig. 3.2. After the dehumidification process, the weak desiccant solution (state 7) is preheated in the condenser and heat exchanger to states 1 and 2, respectively. Then, it is heated to regeneration temperature (state *3*) using the heater and the cycle follows the conventional cycle. In the regenerator, the air loop is closed between the regenerator and condenser (states *d* and *e*). As a result of the regeneration process, hot and humid air leaves the regenerator (state *e*) and enters the condenser where it condenses out. The condensate is produced (state *10*) as freshwater which a by-product is. Due to condensation, air leaves the condenser (state *d*) as saturated air with less moisture content and enters the regenerator to close the air loop as shown in Fig. 3.1. The modified system has two advantages over the conventional system; first, in the modified system, the hot and humid air at the regenerator outlet (state *e*) is not thrown into the atmosphere. This air has a relatively high temperature that can be used to preheat the diluted desiccant (state *7-1*) in the condenser and hence, reducing the heater energy consumption; second, air is condensed in the condenser producing freshwater (state *10*) in the modified system.

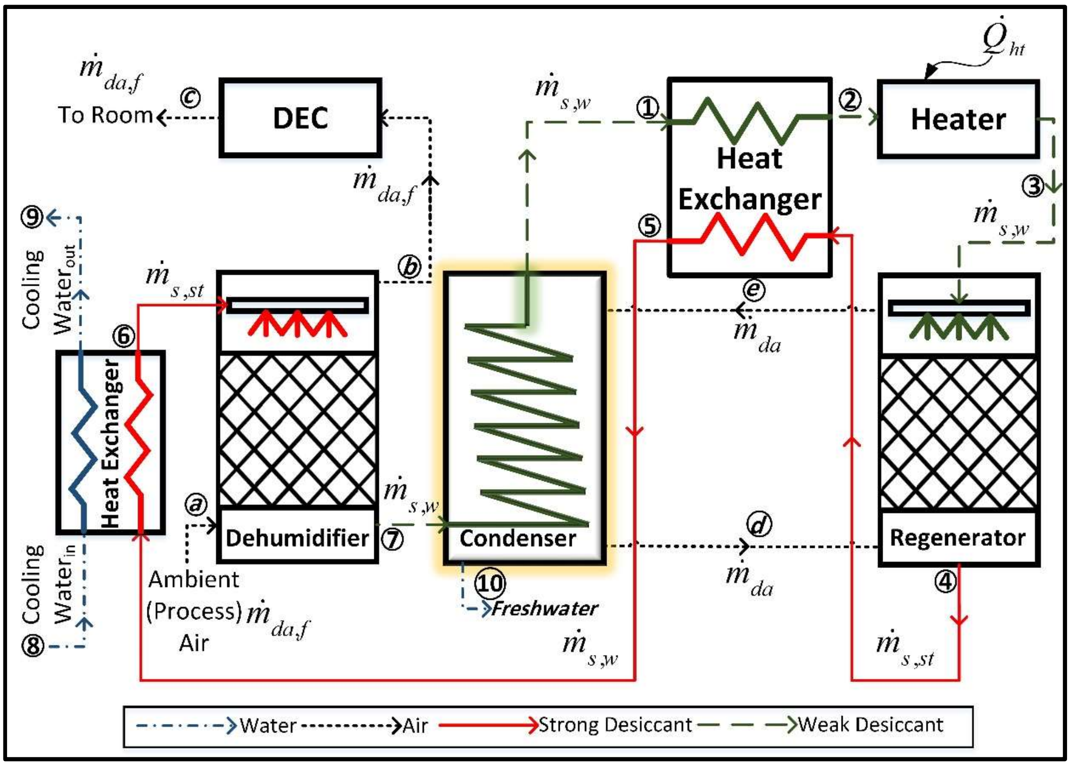


Figure 3.1 Schematic of the modified liquid desiccant air conditioning system.

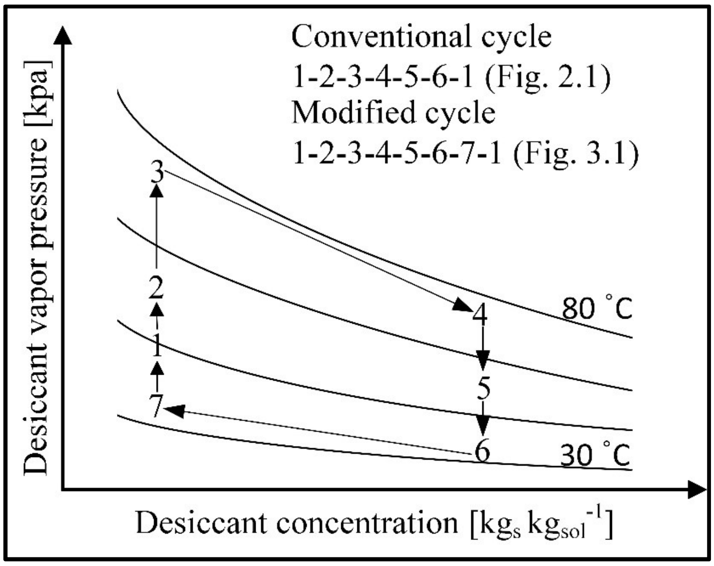


Figure 3.2 Vapor pressure change of the desiccant solution.

3.4 Mathematical modeling

A thermodynamic analysis has been carried out to estimate the performance of conventional and modified systems. In executing the analysis, the following assumptions have been made:

- The working fluid is LiCl solution for the conventional and modified systems.
- All processes involved in the conventional and modified systems work at steady state condition.
- The heat loss from the dehumidifier, the regenerator, heat exchangers or the condenser to the surroundings can be neglected.
- Due to water vapor transfer from the process air to the desiccant solution, the increase in desiccant solution temperature is assumed to be about 2 ^{o}C based on the experimental results [56].

- In the heat exchanger, the phase change is negligible for both hot and cold desiccant streams.
- Comparing to the heater energy input rate, the power required for pumps and blowers is negligible.
- The change in potential and kinetic energies are negligible in the energy balance.
- In the conventional and modified systems, scavenging air at the regenerator outlet is saturated.
- In the modified system, relative humidity of air recirculating between regenerator and condenser are equal.
- In the modified cycle, the freshwater (condensate) leaves the condenser at a temperature equal to the average air temperature in the condenser.

3.4.1. Governing equations

The mass and energy balance equations of the components in the conventional and modified systems are written below:

3.4.1.1. Regenerator:

$$\dot{m}_{s,w} - \dot{m}_{da,r}(\omega_e - \omega_d) = \dot{m}_{s,st} \tag{3.1}$$

$$\dot{m}_{s,w}h_3 - \dot{m}_{s,st}h_4 = \dot{m}_{da,r}(h_e - h_d) \tag{3.2}$$

When the moisture removal rate is given by,

$$\text{MRR} = \dot{m}_{da,r}(\omega_e - \omega_d) \tag{3.3}$$

3.4.1.2. Dehumidifier:

$$\dot{m}_{s,st} + \dot{m}_{da,f}(\omega_a - \omega_b) = \dot{m}_{s,w} \tag{3.4}$$

$$\dot{m}_{s,st}h_6 - \dot{m}_{s,w}h_k = \dot{m}_{da,f}(h_b - h_a) \tag{3.5}$$

3.4.1.3. Heat exchanger:

$$\dot{m}_{s,st}(h_4 - h_5) = \dot{m}_{s,w}(h_2 - h_1) = \dot{Q}_{hx} \tag{3.6}$$

3.4.1.4. Heater:

$$\dot{Q}_{ht} = \dot{m}_{s,w}(h_3 - h_2) \tag{3.7}$$

3.4.1.5. Condenser (modified system):

$$\dot{m}_{10} = \dot{m}_{da,n}(\omega_e - \omega_d) \tag{3.8}$$

$$\dot{m}_{s,w}(h_1 - h_7) + \dot{m}_{10}h_{10} = \dot{m}_{da}(h_e - h_d) \tag{3.9}$$

The set of Eqs. (3.1-3.9) cannot be solved as there are three unknown variables. Therefore, the regenerator, condenser and heat exchanger effectivenesses are defined to solve these equations.

3.4.2. Definition of effectiveness

3.4.2.1. Heat exchanger:

The heat exchanger effectiveness can be defined as, the ratio of actual to maximum heat transfer:

$$\varepsilon_{hx} = \frac{\dot{Q}_{hx}}{\dot{Q}_{max,hx}} \tag{3.10}$$

$$\dot{Q}_{max,hx} = \left(\dot{m}c_p\right)_{min}(T_4 - T_1) \tag{3.11}$$

3.4.2.2. Heat and mass exchangers (regenerator and condenser):

In heat and mass exchangers, the energy transfer is driven by both the concentration and temperature difference. The definition of heat and mass exchanger effectiveness is based on the maximum possible enthalpy difference which can be obtained in a device with 100% effectiveness. The maximum temperature that can be reached by the scavenging air at the regenerator outlet is the desiccant inlet temperature as expressed in Eq. (3.12).

$$\Delta\dot{H}_{air,max} = \dot{m}_{da,r}\left(h_{e,id} - h_d\right) \tag{3.12}$$

The required energy to achieve $\Delta\dot{H}_{air,max}$ is taken from the desiccant stream, which might or might not have the necessary capacity to deliver this amount of energy. If capacity rate of desiccant stream is insufficient, the maximum difference in enthalpy will be that cools down the desiccant to air inlet temperature as shown in Eq. (3.13).

$$\Delta\dot{H}_{s,max} = \dot{m}_{s,w}h_3 - \dot{m}_{s,st}h_{4,id} \tag{3.13}$$

Analogous to heat exchangers, energy effectiveness of heat and mass exchangers is defined as the ratio of actual difference to possible maximum difference in enthalpy rate as stated in Eq. (3.14).

$$\varepsilon_r = \frac{\Delta\dot{H}}{\Delta\dot{H}_{max}} \tag{3.14}$$

Equation (3.14) can be written depending on $\Delta\dot{H}_{max}$ value of both desiccant and air streams as, Case (1), if $\Delta\dot{H}_{air,max} < \Delta\dot{H}_{s,max}$:

$$\varepsilon_r = \frac{h_e - h_d}{h_{e,id} - h_d} \tag{3.15}$$

Case (2), if $\Delta\dot{H}_{air,max} > \Delta\dot{H}_{s,max}$:

$$\varepsilon_r = \frac{\dot{m}_{s,w}h_3 - \dot{m}_{s,st}h_4}{\dot{m}_{s,w}h_3 - \dot{m}_{s,st}h_{4,id}} \tag{3.16}$$

Equations (3.15) and (3.16) can be merged together as,

$$\varepsilon_r = max \left\langle \frac{h_e - h_d}{h_{e,id} - h_d}, \frac{\dot{m}_{s,w}h_3 - \dot{m}_{s,st}h_4}{\dot{m}_{s,w}h_3 - \dot{m}_{s,st}h_{4,id}} \right\rangle \tag{3.17}$$

Analogous to the previous analysis, the energy effectiveness of the condenser can also be defined as,

$$\varepsilon_n = max \left\langle \frac{h_e - h_d}{h_e - h_{d,id}}, \frac{h_1 - h_7}{h_{1,id} - h_7} \right\rangle \tag{3.18}$$

3.4.3. Coefficient of performance (COP) and operating metrics

For a comprehensive understanding of conventional and modified systems, the coefficient of performance (COP) is used as the criterion. The COP is defined as, ratio of the cooling effect to the amount of thermal energy required to produce it and is given by,

$$COP = \frac{\dot{m}_{da,f}(h_a - h_c)}{\dot{Q}_{ht}} \tag{3.19}$$

Any increase in COP results in less capital and operating cost whether the source of heat is solar radiation or fossil fuel. In order to evaluate COP, the following operating parameters are used as input variables:

- Strong desiccant mass flow rate.
- Desiccant-to-air mass flow rate ratio (MR). The ratio of strong desiccant mass flow rate to the dry scavenging air mass flow rate in the regenerator as given by,

$$MR = \frac{\dot{m}_{s,st}}{\dot{m}_{da}} \tag{3.20}$$

- Ambient air temperature and humidity ratio.
- Strong desiccant solution temperature and concentration at dehumidifier inlet.
- Regeneration temperature. The temperature of the weak desiccant solution at regenerator inlet.
- Energy effectiveness (ε). The ratio of actual difference to possible maximum difference in enthalpy. For the regenerator, condenser and heat exchanger, energy effectiveness is defined in section 3.4.2.

The COP of both conventional and modified systems can be predicted by solving Eqs. (3.1-3.7), (3.10-3.11), (3.17) and (3.19–3.20) for the conventional system and Eqs. (3.1-3.11) and (3.17-3.20) for the modified system simultaneously. These equations were solved using Engineering Equation Solver (EES) [57]. EES evaluates moist air properties using Hyland and Wexler formulations [58] and properties of LiCl are calculated using Patek and Klomfar formulations [59]. The mathematical approach to calculate COP of conventional and modified systems is illustrated in the flowchart as shown Fig. 3.3. The comparison between the conventional and modified systems is established taking into account the effect of varying the aforementioned parameters. Typical values of these parameters and their ranges are presented in Table 3.1.

Table 3.1 Basic values and variation of operating parameters.

Parameter		Basic value	Range
Strong desiccant flow rate	kg s^{-1}	1	-
Desiccant-to-air mass flow rate ratio	-	Opt	0.5-3
Ambient air temperature	°C	30	24-40
Ambient air humidity ratio	$\mathrm{kg_{H_2O}\ kg_{da}^{-1}}$	0.02	0.015-0.025
Strong desiccant temperature	°C	30	24-40
Strong desiccant concentration	$kg_s\ kg_{sol}^{-1}$	0.3	0.25-0.35
Regeneration temperature	°C	80	70-90
Energy effectiveness	-	0.8	0.4-0.9

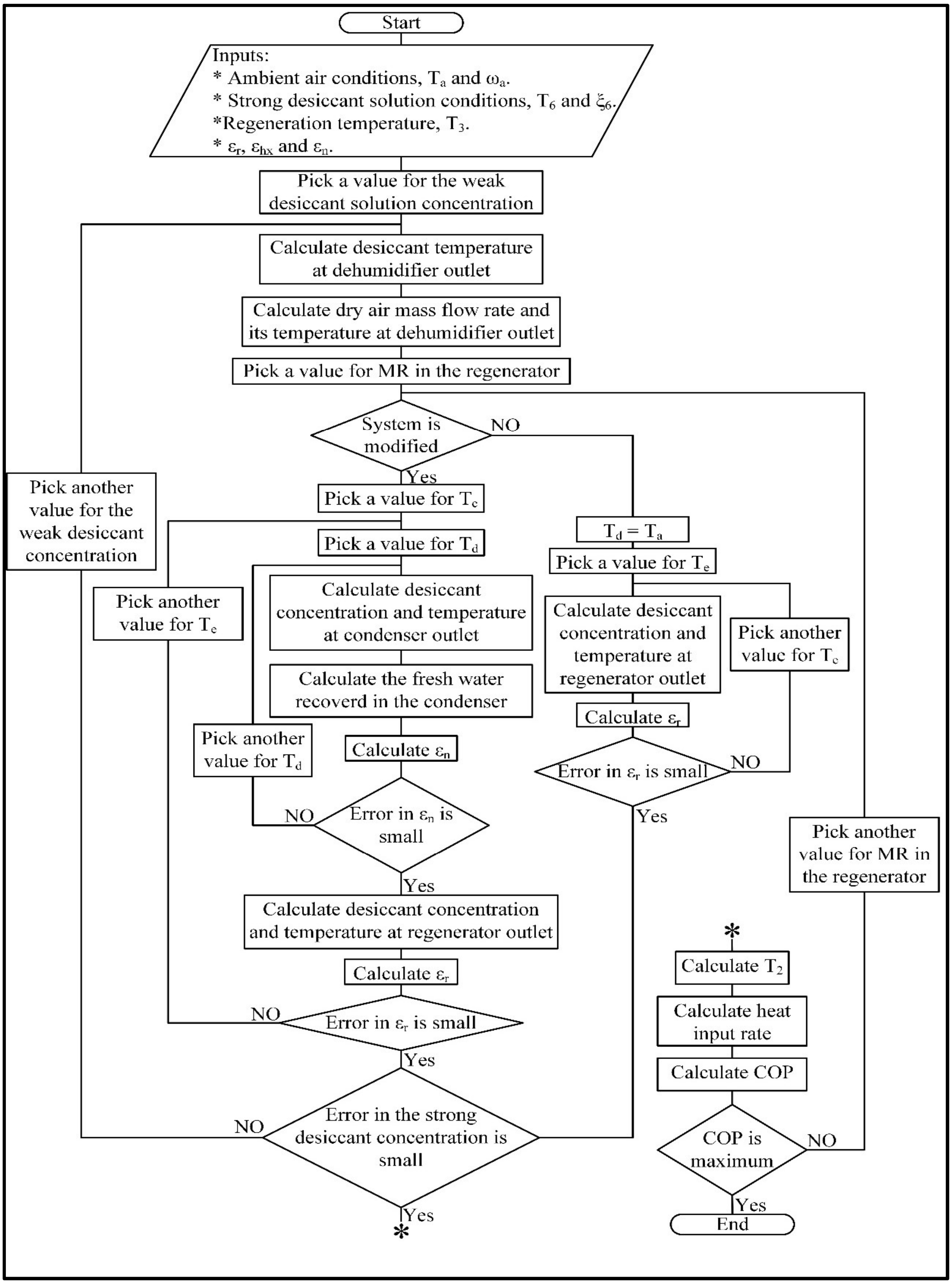


Figure 3.3 Performance prediction algorithm for conventional and modified systems flowchart.

3.4.4. Model validation

In order to use the mathematical model of the current study with confidence, a comparison of the predicted values with the experimental results was established for validation. Fumo and Goswami [56] presented reliable results from a study of the performance of a packed tower absorber and regenerator for an aqueous lithium chloride desiccant system. The tower is constructed from a 25.4 cm (24 cm I.D.) diameter acrylic tube. The height of the tower is constant and equal to 60 cm. The packings used were 2.54 cm (1 in.) polypropylene Rauschert Hiflow® rings with specific surface area of 210 m^2/m^3. These characteristics are represented in the energy effectiveness (ε_r). The rates of regeneration process were assessed under the effects of variables such as air and desiccant flow rates, air temperature and humidity, and desiccant temperature and concentration. These variables with the energy effectiveness are used as input data in the validation.

The results obtained from the current regenerator model are compared with the experimental results reported in [56] and presented in Table 3.2. The energy effectiveness of the regenerator is calculated from the experimental data and found it varies between 0.93 to 0.95. Nonetheless, the value of 0.94 is used as the represented energy effectiveness for the whole validation as shown in Table 3.2. The model results of the regenerator show good agreement with the experimental results and the discrepancy is between 0 to 8.6 %.

Table 3.2 Validation of the current study results of the regenerator with the experimental data [56].

	$T_{air,o}$ [ºC]	$\omega_{air,o}$ $[g_{H_2O}\ kg_{da}^{-1}]$	$T_{s,o}$ [ºC]	$\xi_{s,o}$ $[kg_s\ kg_{sol}^{-1}]$	water evaporation rate [g s^{-1}]
Case (1): $\dot{m}_{air,i}$ = 0.83 kg $m^{-2}s^{-1}$, $T_{air,i}$ = 30.4 ºC, $\omega_{air,i}$ = 18.3 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.46 kg $m^{-2}s^{-1}$, $T_{s,i}$ = 65 ºC, $\xi_{s,i}$ = 0.34 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	58.9	57.9	58.6	0.3450	1.55
Current study	58.2	52.5	59.8	0.3415	1.34
Percentage difference	1.2	7.6	2.0	1.0	8.4
Case (2): $\dot{m}_{air,i}$ = 1.1 kg $m^{-2}s^{-1}$, $T_{air,i}$ = 30.1 ºC, $\omega_{air,i}$ = 18 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.21 kg $m^{-2}s^{-1}$, $T_{s,i}$ = 65.1 ºC, $\xi_{s,i}$ = 0.341 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	59.3	53.2	57.8	0.3480	1.81
Current study	58.2	52.5	57.8	0.3431	1.78
Percentage difference	1.9	1.3	0.0	1.4	1.7
Case (3): $\dot{m}_{air,i}$ = 1.44 kg $m^{-2}s^{-1}$, $T_{air,i}$ = 29.8 ºC, $\omega_{air,i}$ = 17.7 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.48 kg $m^{-2}s^{-1}$, $T_{s,i}$ = 65.1 ºC, $\xi_{s,i}$ = 0.345 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	57.5	48.8	56.6	.3520	2.10
Current study	58.2	52.3	55.8	0.3476	2.35
Percentage difference	1.2	7.2	1.4	1.2	8.1
Case (4): $\dot{m}_{air,i}$ = 1.1 kg $m^{-2}s^{-1}$, $T_{air,i}$ = 35.1 ºC, $\omega_{air,i}$ = 18 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.4 kg $m^{-2}s^{-1}$, $T_{s,i}$ = 65.1 ºC, $\xi_{s,i}$ = 0.334 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	58.5	55.1	57.4	0.3410	1.91
Current study	58.4	53.1	58.3	0.3360	1.81
Percentage difference	0.1	3.6	1.5	1.5	5.3

Case (5): $\dot{m}_{air,i}$ = 1.1 kg m^{-2}s^{-1}, $T_{air,i}$ = 40 ºC, $\omega_{air,i}$ = 17.8 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.35 kg m^{-2}s^{-1}, $T_{s,i}$ = 65 ºC, $\xi_{s,i}$ = 0.336 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	58.9	54.8	57.6	0.3420	1.91
Current study	58.5	53.4	58.4	0.3381	1.84
Percentage difference	0.7	2.6	1.4	1.1	3.7
Case (6): $\dot{m}_{air,i}$ = 1.13 kg m^{-2}s^{-1}, $T_{air,i}$ = 30.2 ºC, $\omega_{air,i}$ = 14.3 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.37 kg m^{-2}s^{-1}, $T_{s,i}$ = 65.2 ºC, $\xi_{s,i}$ = 0.34 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	57.6	51.3	57.2	0.3470	1.97
Current study	57.9	51.7	57.5	0.3422	1.998
Percentage difference	0.5	0.8	0.4	1.4	1.4
Case (7): $\dot{m}_{air,i}$ = 1.1 kg m^{-2}s^{-1}, $T_{air,i}$ = 29.4 ºC, $\omega_{air,i}$ = 21 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.44 kg m^{-2}s^{-1}, $T_{s,i}$ = 65.5 ºC, $\xi_{s,i}$ = 0.336 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	58.5	54.1	58.3	0.3420	1.70
Current study	58.8	54.0	58.7	0.3379	1.698
Percentage difference	0.5	0.1	0.7	1.2	0.1
Case (8): $\dot{m}_{air,i}$ = 1.12 kg m^{-2}s^{-1}, $T_{air,i}$ = 30.3 ºC, $\omega_{air,i}$ = 18.2 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 5.19 kg m^{-2}s^{-1}, $T_{s,i}$ = 65.4 ºC, $\xi_{s,i}$ = 0.344 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	57.6	50.7	57.0	0.3490	1.71
Current study	58.4	53.2	56.4	0.3466	1.83
Percentage difference	1.4	4.9	1.1	0.7	7.0
Case (9): $\dot{m}_{air,i}$ = 1.1 kg m^{-2}s^{-1}, $T_{air,i}$ = 29.9 ºC, $\omega_{air,i}$ = 18 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 7.54 kg m^{-2}s^{-1}, $T_{s,i}$ = 65.2 ºC, $\xi_{s,i}$ = 0.343 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	59.0	55.6	57.9	0.3490	1.95
Current study	58.3	52.7	59.1	0.3447	1.80
Percentage difference	1.2	5.2	2.1	1.2	7.7

Case (10): $\dot{m}_{air,i}$ = 1.11 kg m^{-2}s^{-1}, $T_{air,i}$ = 30 ºC, $\omega_{air,i}$ = 18.7 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.25 kg m^{-2}s^{-1}, $T_{s,i}$ = 60.3 ºC, $\xi_{s,i}$ = 0.344 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	55.8	44.7	54.2	0.3480	1.36
Current study	54.9	44.3	54.6	0.3455	1.34
Percentage difference	1.6	0.9	0.7	0.7	1.5
Case (11): $\dot{m}_{air,i}$ = 1.08 kg m^{-2}s^{-1}, $T_{air,i}$ = 29.7 ºC, $\omega_{air,i}$ = 18.4 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.32 kg m^{-2}s^{-1}, $T_{s,i}$ = 70 ºC, $\xi_{s,i}$ = 0.345 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	62.6	66.6	60.0	0.3530	2.45
Current study	61.6	62.3	61.2	0.3476	2.24
Percentage difference	1.6	6.5	2.0	1.5	8.6
Case (12): $\dot{m}_{air,i}$ = 1.1 kg m^{-2}s^{-1}, $T_{air,i}$ = 29.7 ºC, $\omega_{air,i}$ = 17.7 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.4 kg m^{-2}s^{-1}, $T_{s,i}$ = 64.8 ºC, $\xi_{s,i}$ = 0.328 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	57.6	54.2	56.8	0.3340	1.89
Current study	58.0	51.9	57.9	0.3299	1.765
Percentage difference	0.6	4.3	2.0	1.2	6.6
Case (13): $\dot{m}_{air,i}$ = 1.12 kg m^{-2}s^{-1}, $T_{air,i}$ = 30.3 ºC, $\omega_{air,i}$ = 18.2 $g_{H_2O}\ kg_{da}^{-1}$, $\dot{m}_{s,i}$ = 6.43 kg m^{-2}s^{-1}, $T_{s,i}$ = 65 ºC, $\xi_{s,i}$ = 0.349 $kg_s\ kg_{sol}^{-1}$.					
Experimental results	57.9	50.1	57.5	0.3540	1.67
Current study	58.2	52.4	57.8	0.3510	1.795
Percentage difference	0.5	4.6	0.6	0.8	7.5

3.5 Results and discussion

3.5.1. Effect of desiccant-to-air mass flow rate ratio

Figure 3.4 illustrates the effect of desiccant-to-air mass flow rate ratio on the performance of conventional and modified systems. The COP of both systems increases as MR increases up to an optimum value of MR = 1.27 and 1.46 for conventional and modified systems, respectively and then COP starts to decrease. To comprehend COP behavior, heat input rate and cooling effect are analyzed and shown in Fig 3.5. From this figure, it can be seen that up to the optimum point, the cooling effect increases while heat input rate remains constant and hence, COP increases as shown in Fig. 3.4.

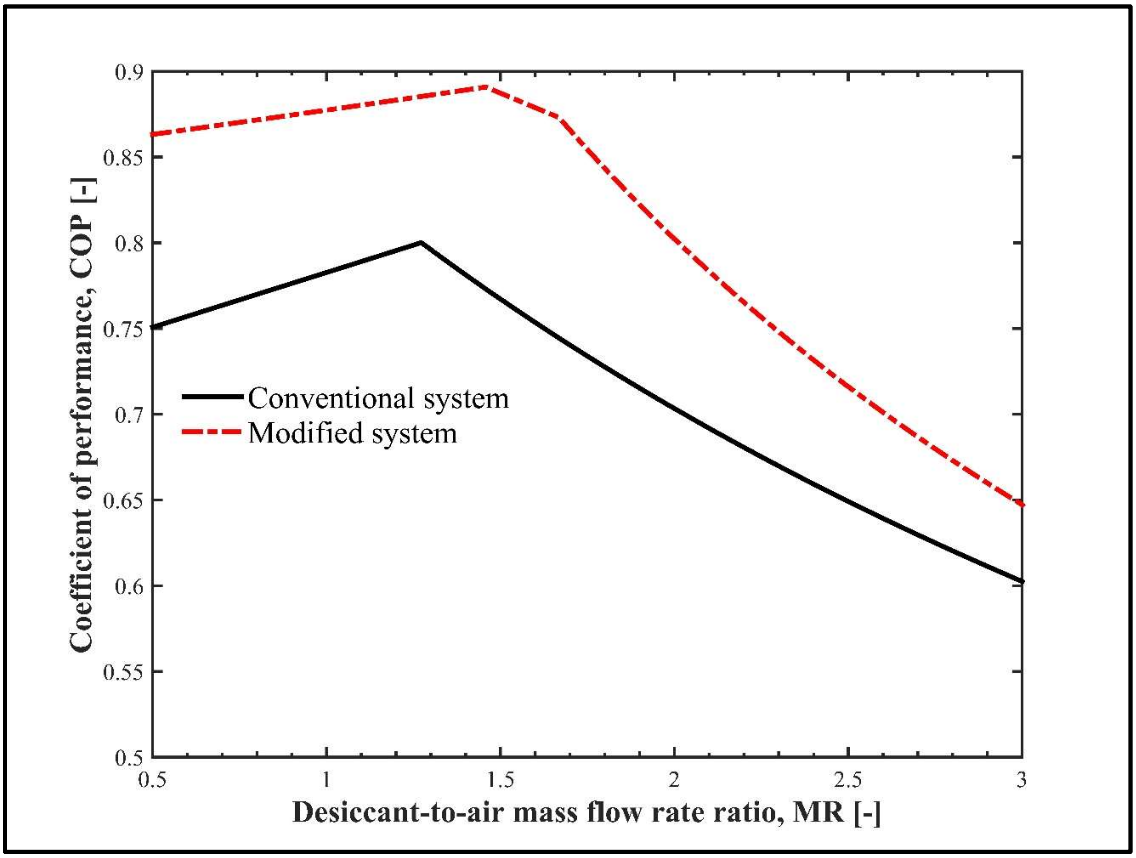
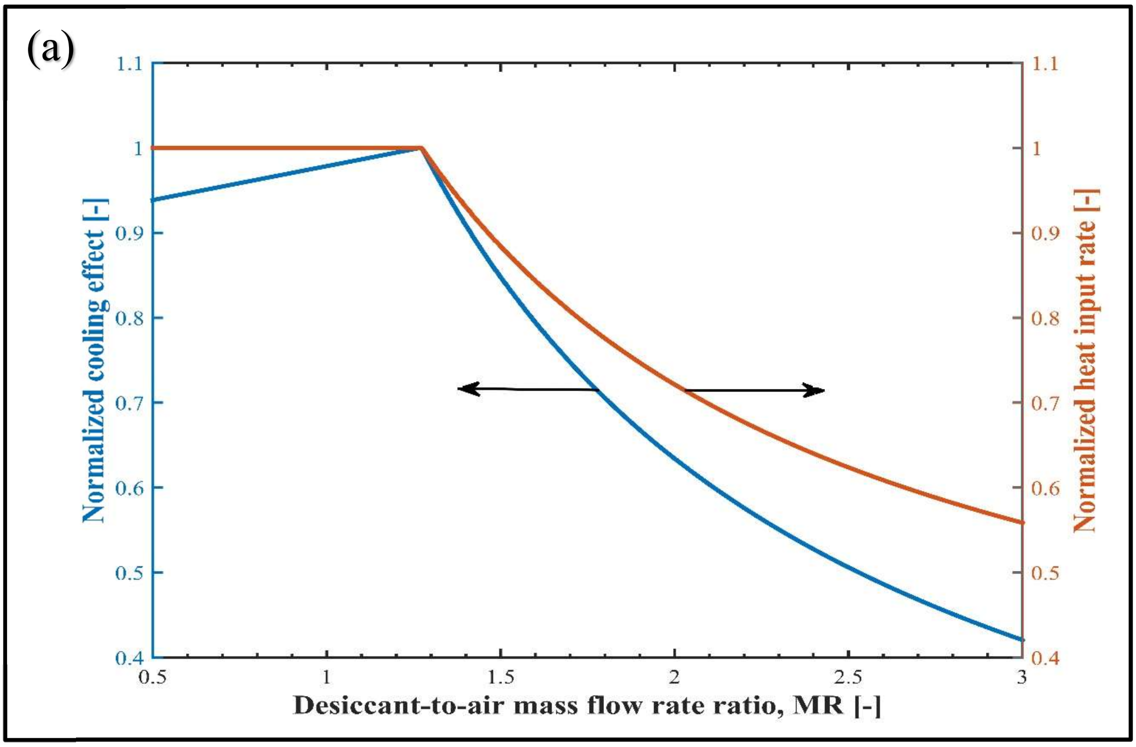


Figure 3.4 Effect of desiccant-to-air mass flow rate ratio on COP.



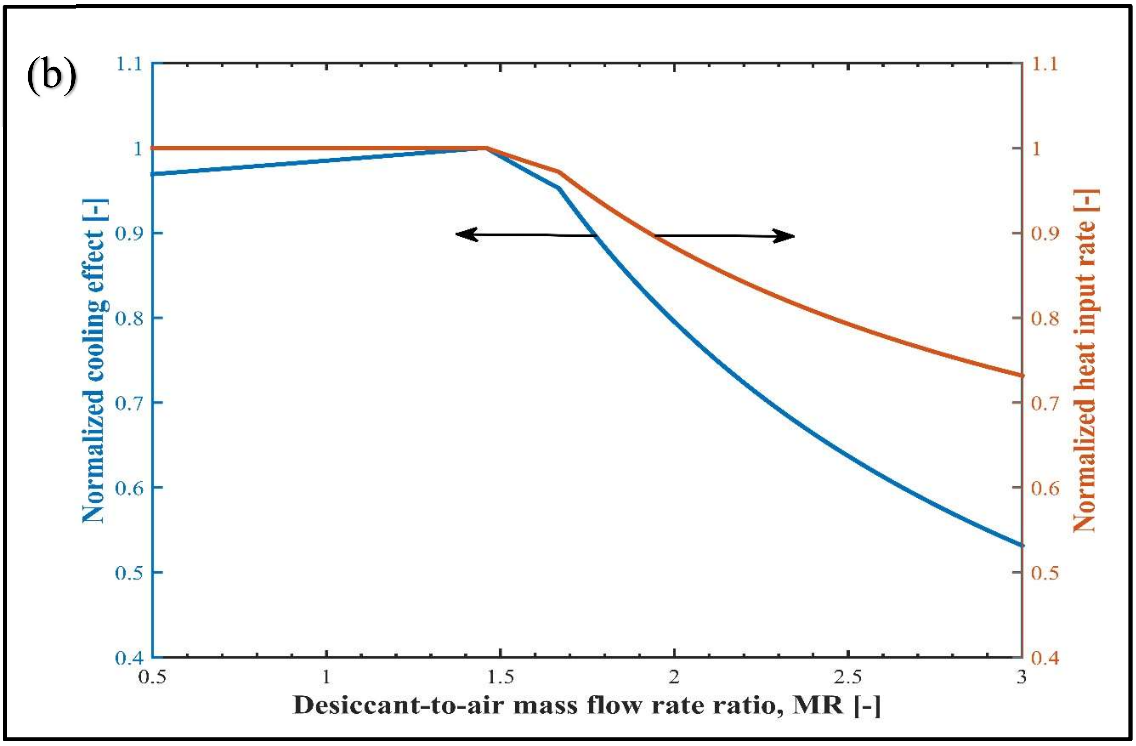


Figure 3.5 Effect of desiccant-to-air mass flow rate ratio on heat input rate and cooling effect. (a) Conventional system. (b) Modified system.

After the optimum point, both heat input rate and cooling effect decrease as MR increases. However, the decrease in the cooling effect is higher than heat input rate which justifies the decrease in COP.

From Fig. 3.5, heat input rate is constant up to an optimum value of MR and then decreases dramatically as MR increases. At constant desiccant mass flow rate, any increase in MR necessitates a decrease in air mass flow rate. From the energy balance, at a lower mass flow rate of air passing through the regenerator, air and desiccant solution leave the regenerator with higher temperature. This additional energy in the strong desiccant solution is used to preheat the weak desiccant solution in the heat exchanger (state 1-2 as in Figs. 2.1 and 3.1) resulting in less heat input rate in the heater. Referring to Eq. (3.3), moisture removal rate is governed by the scavenging air flow rate and its humidity ratio at the regenerator outlet. Up to the optimum point, as MR increases, outlet air temperature increases resulting in higher humidity ratio which increases the moisture removal rate and accordingly, cooling effect increases as shown Fig. 3.5. However, after the optimum point, the decrease in air mass flow rate results in a dramatic decrease in moisture removal rate and consequently, less cooling effect is produced.

It is to be noted that the optimum value of MR is higher in the modified system than the conventional system as shown in Figs. 3.4 and 3.5. In the case of the modified system, the air loop between the regenerator and condenser is closed and as a result, air enters the regenerator at a higher temperature. Therefore, lower mass flow rate of air is needed for the regeneration.

3.5.2. Effect of ambient air conditions

The effect of ambient air conditions on the performance of the conventional system is presented in Figure 3.6 (a). It may be pointed out that at higher ambient air temperature, the desiccant temperature increases, reducing the mass transfer potential in the dehumidifier. However, as ambient (scavenging) air temperature increases, the desiccant temperature at regenerator outlet also increases. This will enhance the heat recovery which results in less heat input rate and consequently, COP increases.

In the regenerator, as air inlet humidity ratio increases, the water evaporation rate decreases. This is because, at higher humidity ratio, air vapor pressure is higher and accordingly the mass transfer potential is lower. However, in the dehumidifier as air inlet humidity ratio increases, air vapor pressure increases and as a result, the moisture removal rate increases. Therefore, the cooling effect increases and hence, COP increases.

The performance of the modified system with respect to ambient air conditions on is presented in Fig. 3.6 (b). In the modified system, the temperature and humidity ratio of ambient air have no effect on the regeneration process since the air loop between the regenerator and condenser is closed as shown in Fig. 3.1. While it influences the dehumidification process and hence the decrease in the modified system performance as the ambient air temperature increases.

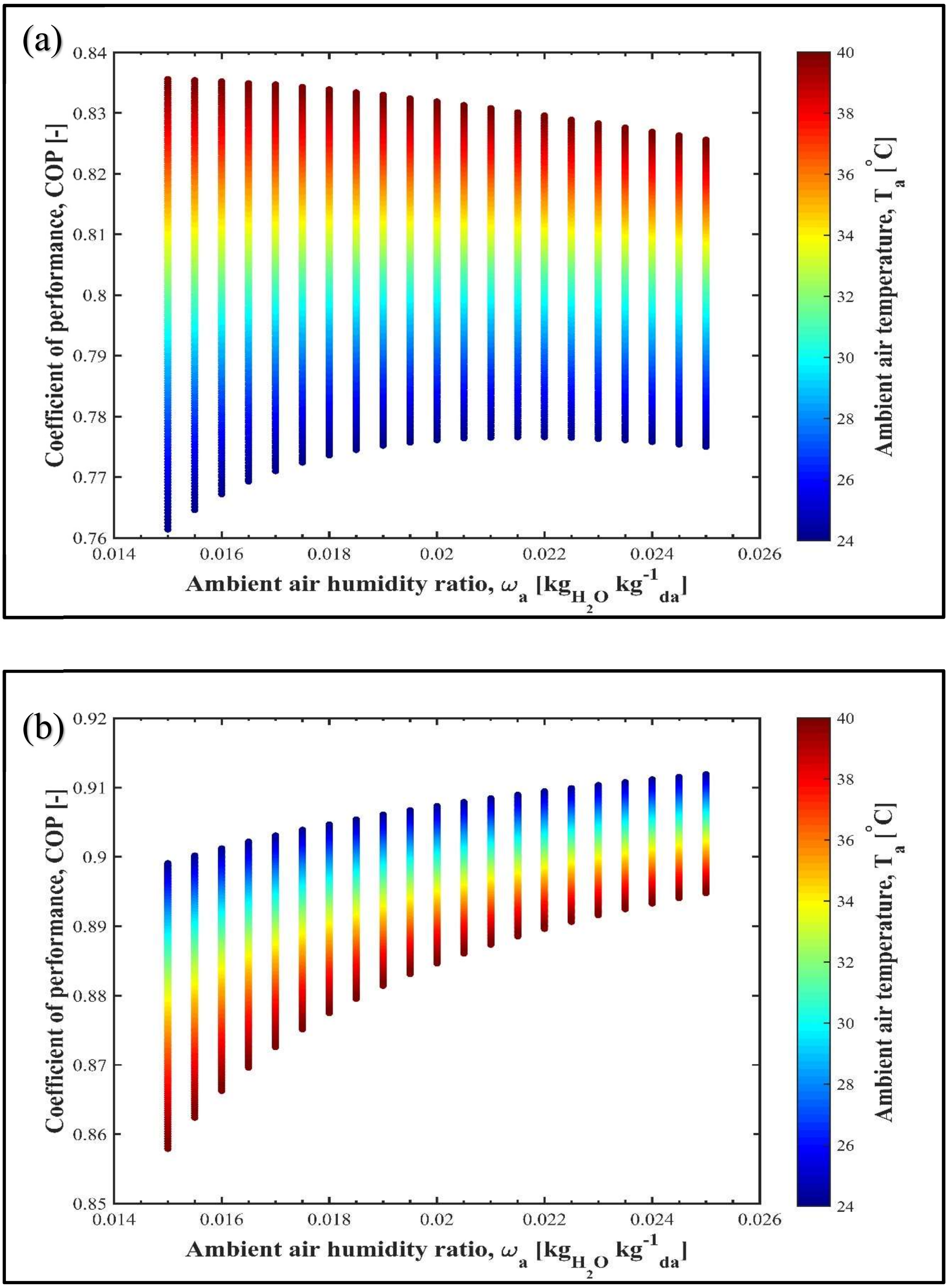


Figure 3.6 Effect of ambient air condition on COP. (a) Conventional system. (b) Modified system.

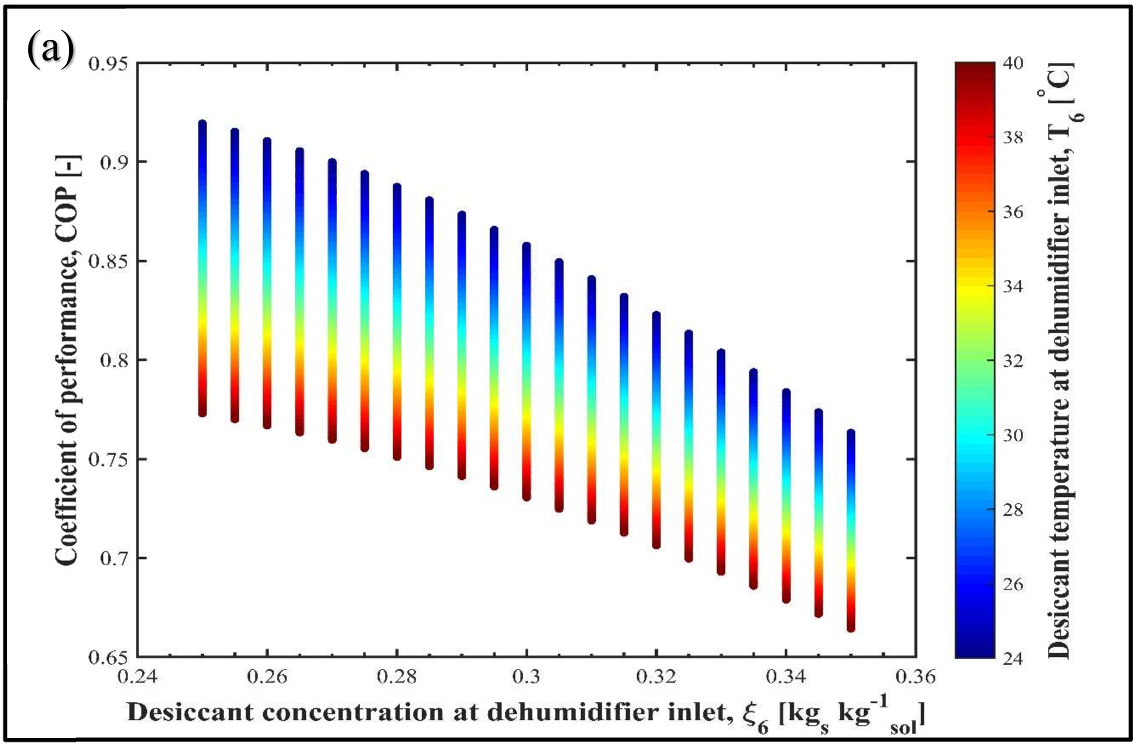
3.5.3. Effect of desiccant solution inlet conditions

The effect of strong desiccant solution temperature at the inlet of the dehumidifier on the performance of both systems is shown in Fig. 3.7. The increase in desiccant temperature at the dehumidifier inlet lowers the mass transfer potential in the dehumidifier lowering the moisture condensation rate and hence, the cooling effect is less. As a result, COP decreases in both systems.

Figure 3.7 also shows the effect of inlet desiccant solution concentration on the performance of both systems. It can be seen that COP decreases as desiccant concentration increases. In the dehumidifier, as desiccant concentration increases, water condensation rate increases. This is due to the fact that at higher desiccant concentration, the potential for mass transfer increases and hence, the water condensation rate is higher. However, as the desiccant solution concentration increases, its vapor pressure decreases in the regenerator, thus, the water evaporation rate decreases. Therefore, the mass transfer potential in the regenerator is less and as a result, more energy is required to overcome this deficit in mass transfer which reduces the COP of the systems.

3.5.4. Effect of regeneration temperature

The performance of both systems with respect to the regeneration temperature is presented in Fig. 3.8. From this figure, COP increases as regeneration temperature increases. As shown in Table 3.3, both heat input rate and cooling effect increase as the regeneration temperature increases. The percentage increase of cooling effect is higher than the required heat input rate and hence, COP increases. It is to be noted that the increase in COP is significant in the case of the modified system compared with the conventional system due to the incorporation of the condenser in heat recovery.



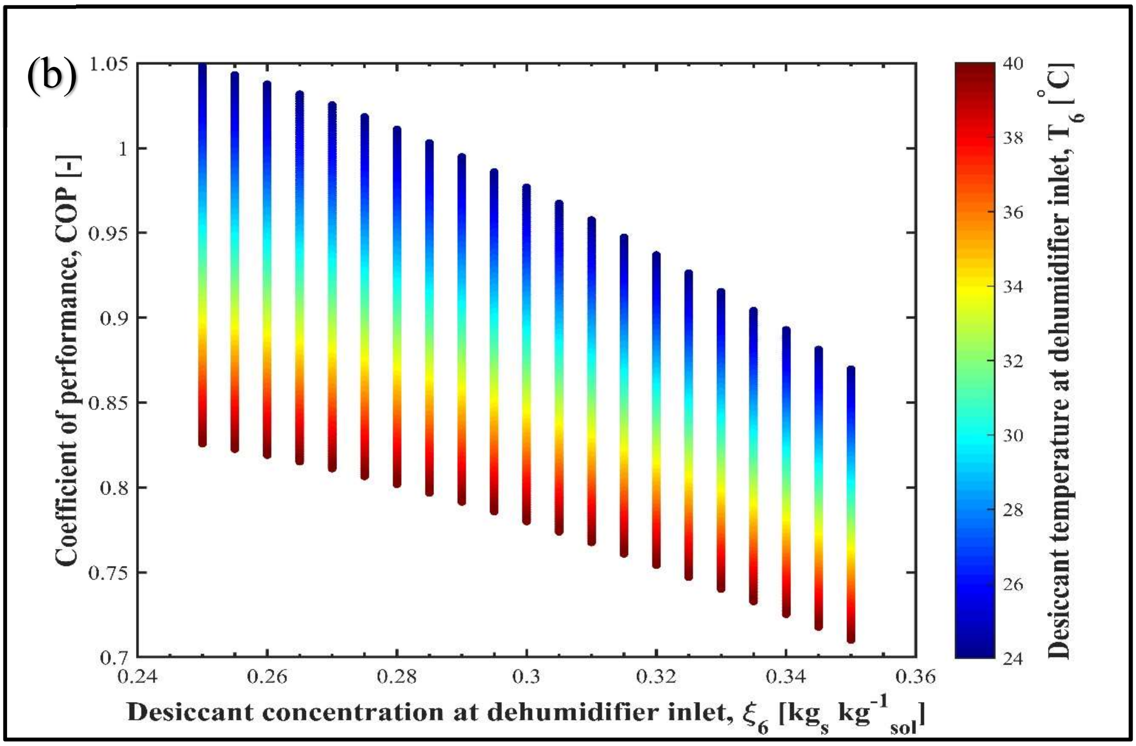


Figure 3.7 Effect of desiccant solution condition at dehumidifier inlet on COP. (a) Conventional system. (b) Modified system.

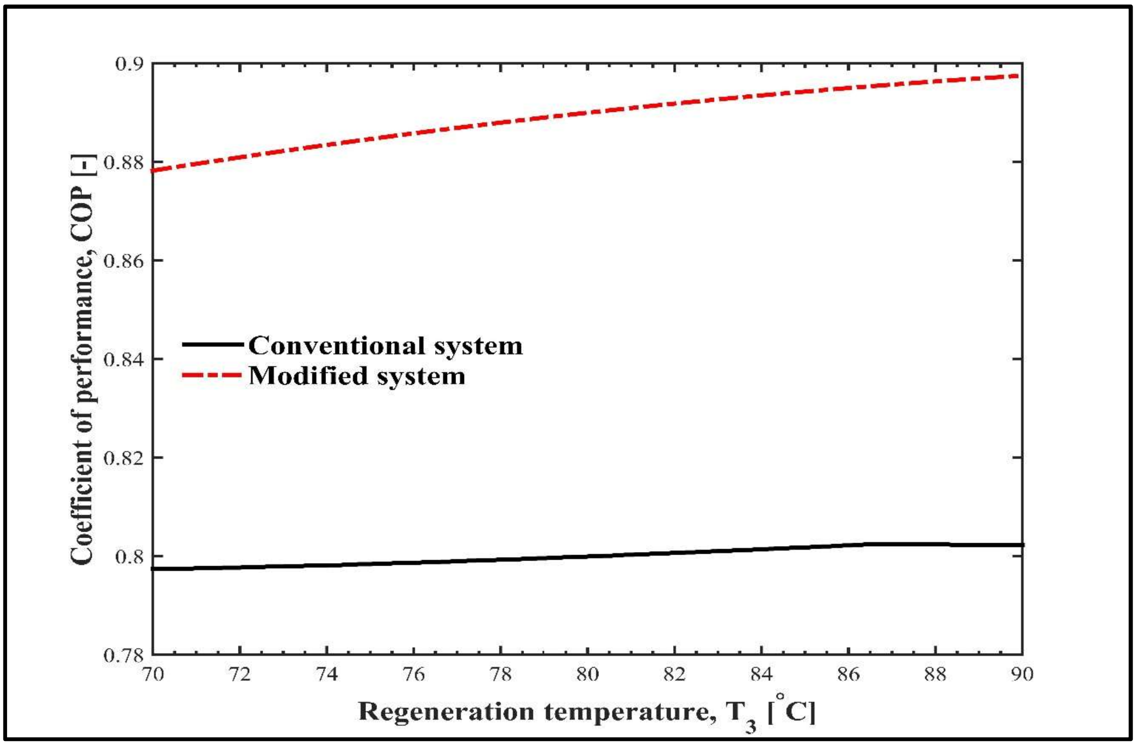


Figure 3.8 Effect of regeneration temperature on COP.

Table 3.3 Effect of regeneration temperature on heat input rate and cooling effect.

	Conventional system			Modified system		
	T_3 =		Percentage change	T_3 =		Percentage change
	70 °C	90 °C		70 °C	90 °C	
Heat input rate [kW]	98.3	148.2	50.8	54.3	81.8	50.6
Cooling effect [kW]	78.4	118.9	51.7	47.7	73.5	54.1
COP [-]	0.797	0.802	0.63	0.878	0.898	2.28

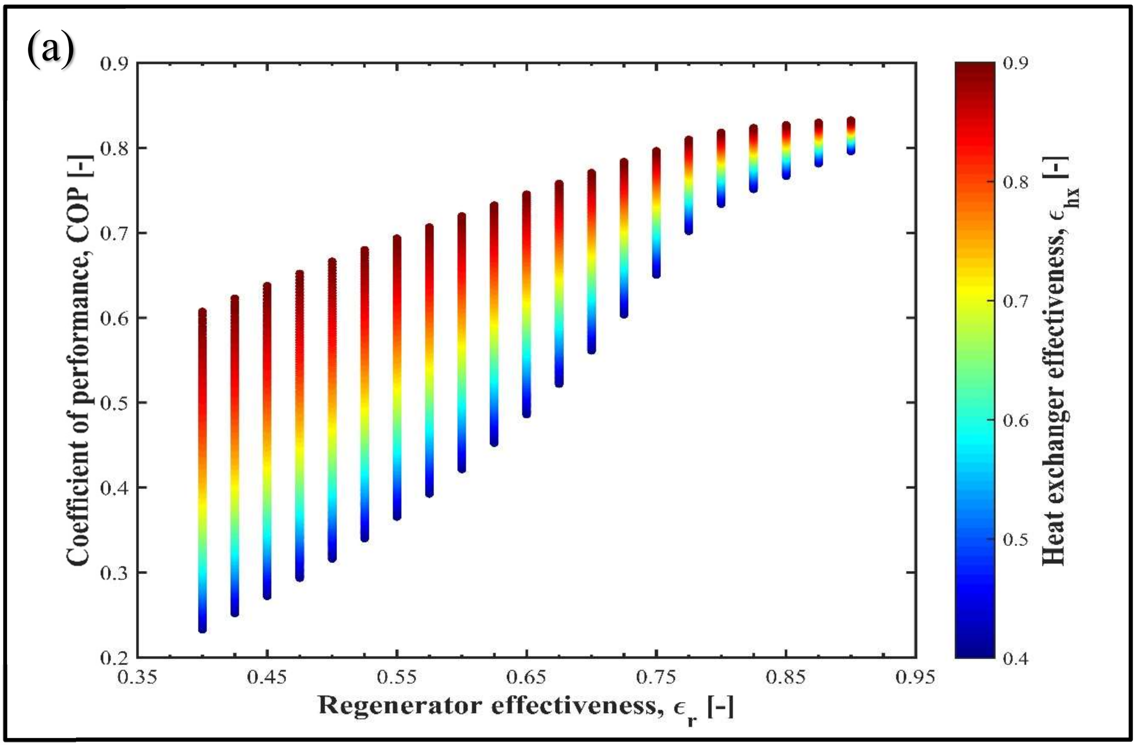
3.5.5. Effect of the components energy effectivenesses

The effect of regenerator and heat exchanger effectivenesses on the COP is exhibited in Fig. 3.9. In both systems, COP increases as heat exchanger effectiveness increases. Using a heat exchanger with higher effectiveness increases the efficiency of heat recovery and that reduces the required heat input rate in the heater.

Figure 3.9 also shows that COP increases as regenerator effectiveness increases. In the case of a regenerator with lower effectiveness, the potential for heat and mass transfer is less. Therefore, the desiccant solution leaves the regenerator with increase in temperature and decrease in concentration. This increase in temperature is used to preheat the weak desiccant solution in the heat exchanger and hence, less heat input rate is required. On the other hand, the decrease in desiccant concentration reduces the moisture removal rate in the dehumidifier and consequently, the cooling effect is reduced. This indicates the decrease in COP as regenerator effectiveness decreases.

The desiccant temperature at the outlet of regenerator increases as the effectiveness decreases. That is, more energy is recovered in the heat exchanger. Therefore, the heat exchanger has more influence on the performance of both systems at lower regenerator effectiveness as shown in Fig. 3.9.

The behavior of COP in the modified system with respect to condenser effectiveness is presented in Fig. 3.10. It shows that, at lower condenser effectiveness, the performance of the modified system is lower than the conventional system. As condenser effectiveness increases, the COP increases and it exceeds the conventional system COP when the effectiveness is greater than 0.57.



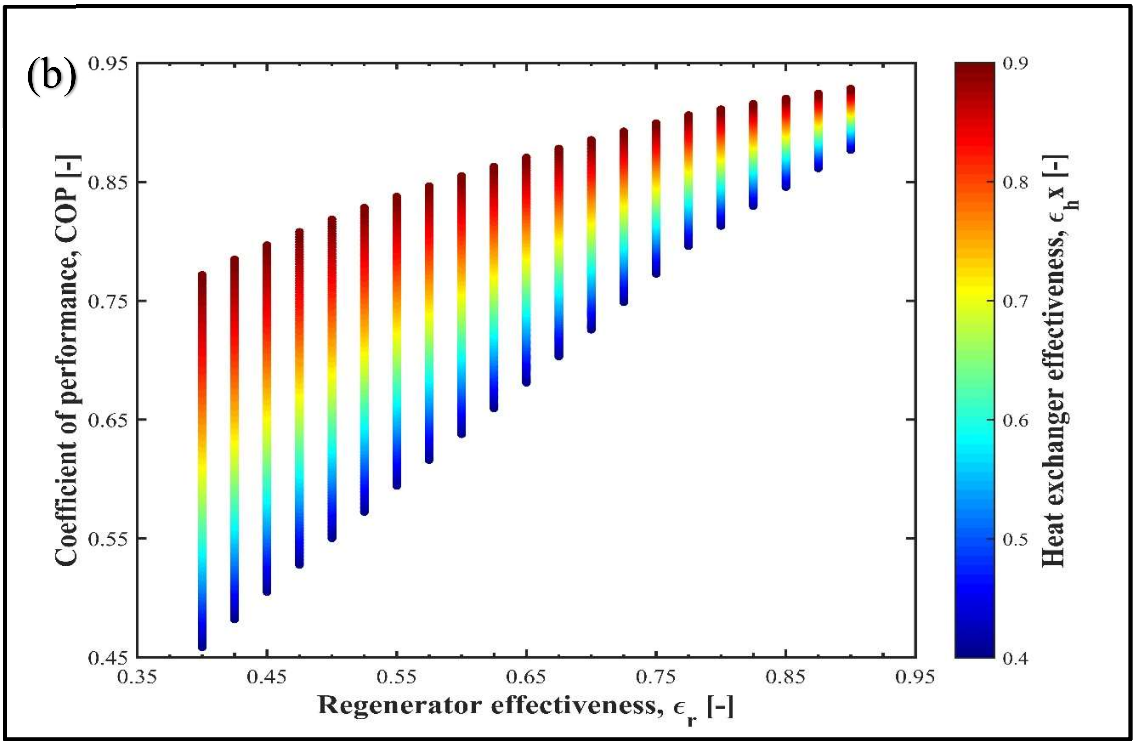


Figure 3.9 Effect of regenerator and heat exchanger effectivenesses on COP. (a) Conventional system. (b) Modified system.

This is because, as condenser effectiveness increases, the weak desiccant temperature (state 1 in Fig. 3.1) increases and hence the required heat input rate is less. It is to be noted that using a condenser with effectiveness less than the value of 0.57 is counterproductive.

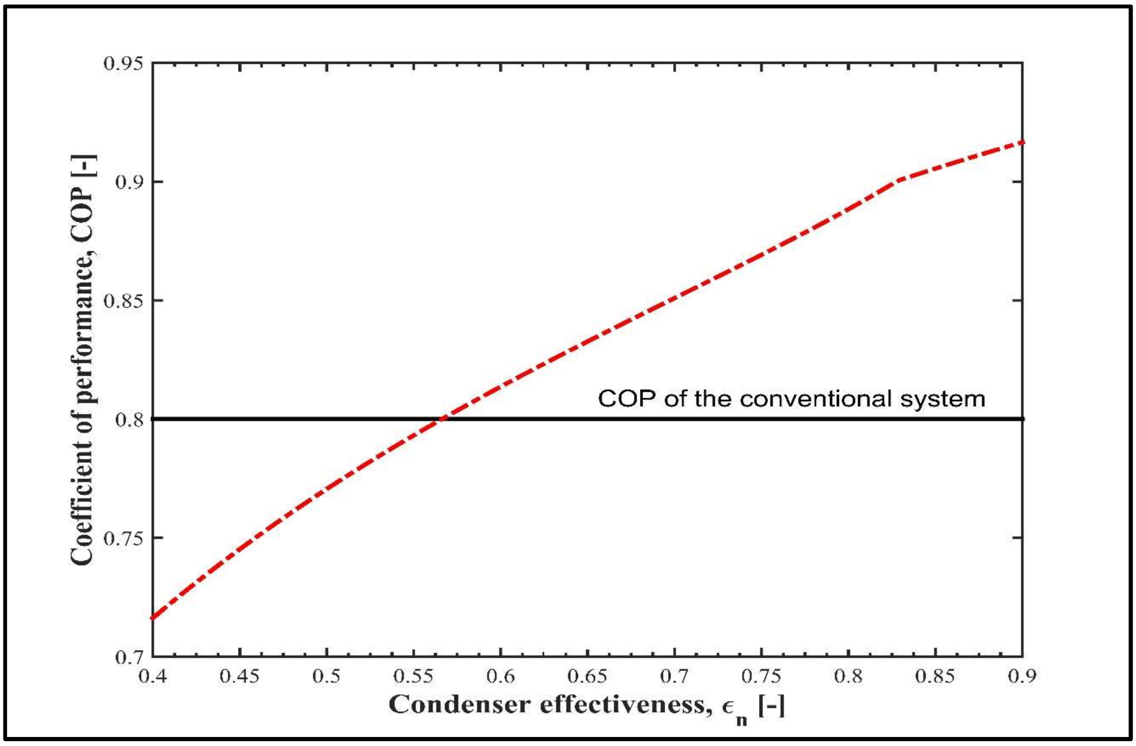


Figure 3.10 Effect of condenser effectiveness on COP of the modified system.

3.5.6. Freshwater produced in the modified system

The effect of regeneration temperature on rate of freshwater production with respect to desiccant solution conditions at dehumidifier inlet is shown in Fig. 3.11. From this figure, as regeneration temperature increases, freshwater produced increases due to increase in mass transfer potential in the regenerator. As a result, air humidity ratio at regenerator outlet increases and hence, more freshwater is recovered in the condenser.

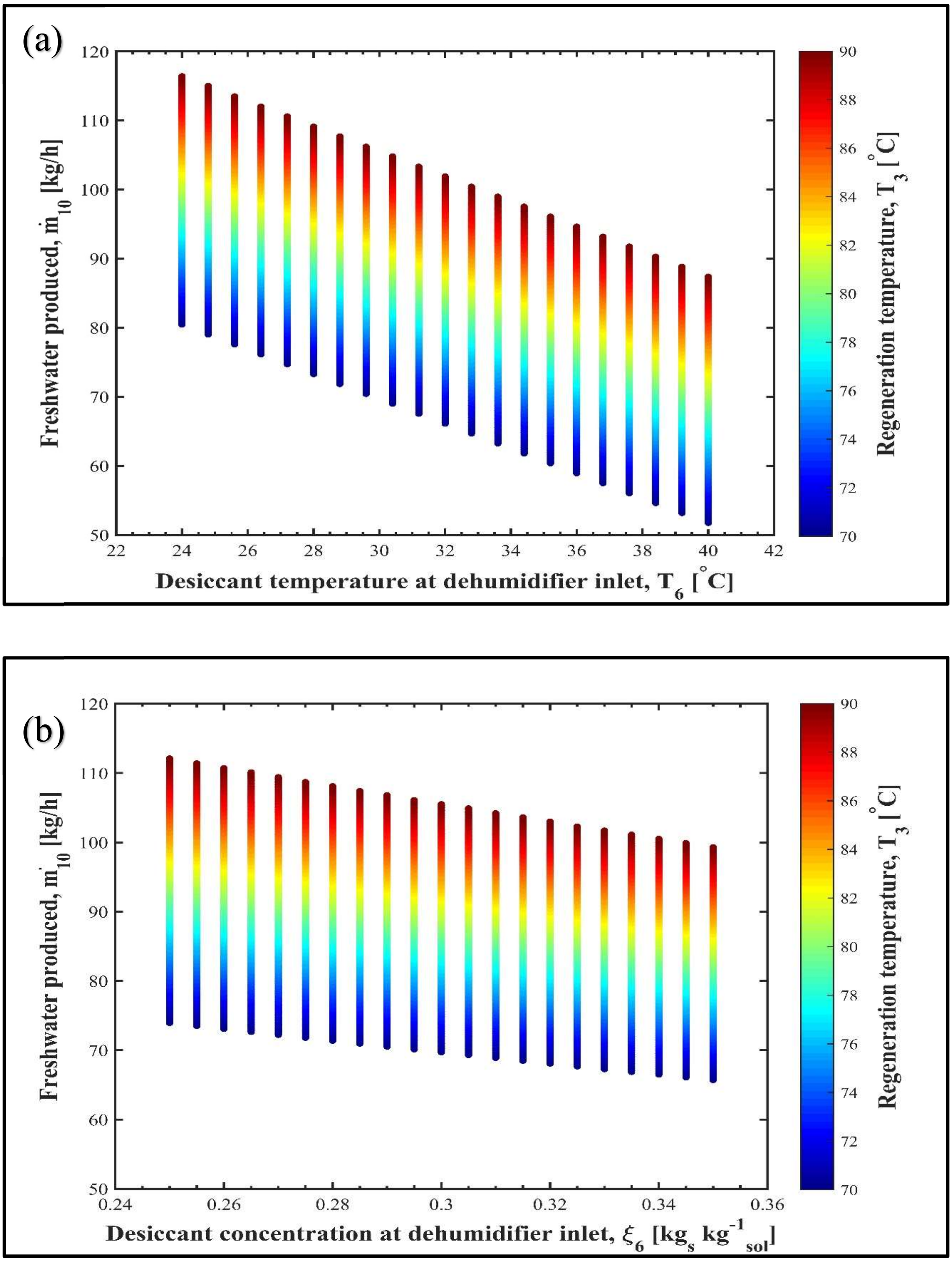


Figure 3.11 Freshwater produced in the modified system vs. regeneration temperature. (a) Effect of desiccant temperature at dehumidifier inlet. (b) Effect of desiccant concentration at dehumidifier inlet.

The desiccant solution that leaves the dehumidifier is used as the coolant in the condenser and as coolant temperature increases, the mass transfer potential in the condenser decreases and hence, the decrease in freshwater produced as shown in Fig. 3.11 (a). The concentration of the weak desiccant solution that leaves the dehumidifier is used as the inlet stream in the regenerator. As the weak desiccant solution concentration increases, the mass transfer potential in the regenerator decreases. Therefore, water evaporation rate also decreases and consequently, less freshwater is recovered in the condenser as shown in Fig. 3.11 (b).

3.5.7. Comparison between the conventional and modified systems

In order to establish an unbiased comparison, the COP of both conventional and modified systems is estimated based on all variations of operating conditions at an optimum desiccant-to-air mass flow rate ratio as shown in Figs. 3.4 - 3.10.

Table 3.4 Comparison between the conventional and modified systems.

	Conventional system	Modified system	Percentage change
COP [-]	0.8	0.89	11.25
Produced freshwater [kg h^{-1}]	-	86.4	-
Critical condenser effectiveness [-]	-	0.57	-

Table 3.4 shows the comparison between the conventional and modified systems based on the basic operating conditions as listed in Table 3.1. The performance of the modified system is better than the conventional system due to additional heat recovery using the condenser. The modified system performance is 11.25% better than the conventional

system when the condenser effectiveness is 0.8. This improvement decreases as condenser effectiveness decreases until it vanishes at 0.57. It is to be noted that, the modified system produces 86.4 kg of freshwater per hour as a by-product.

3.6 Conclusions

The effect of operating conditions on the performance of the conventional and modified desiccant air conditioning systems have been examined. The COP is used as a criterion to compare the performance of the systems. From the aforementioned discussion, the following conclusions can be drawn:

1. Main factors that influence the performance of both systems are desiccant-to-air mass flow rate ratio, desiccant solution and process air inlet conditions, and energy effectiveness of the system components.
2. The performance of both systems increases as the energy effectiveness increases. Moreover, the increase in regenerator effectiveness has the highest effect on the system performance while heat exchanger effectiveness has the lowest effect.
3. The modified system performance is 11.25% better than the conventional system. This comparison is executed at optimum desiccant-to-air mass flow rate ratio using basic operating conditions given in Table 3.1. This improvement in modified system vanishes when the condenser effectiveness is less than 0.57.
4. The amount of water evaporated in the regenerator is recovered using the condenser as freshwater. The modified system produces 86.4 kg of freshwater per hour as a by-product.
5. Using the condenser for heat and mass recovery in the modified system is promising with plenty of room to improve its performance.

CHAPTER 4

Minimization of entropy generation concept

4.1 Summary

This chapter details a procedure to study the effect of entropy generation in the heat and mass exchangers (regenerator and condenser) and how it can be minimized. Using the second law, the highest COP achievable (COP when the system is reversible i.e. zero entropy production) in the modified system is evaluated. It is found that the upper limit of the COP of the modified system is 112.2.

The heat capacity rate ratio (HCR) is presented as the main parameter to describe the entropy generation. It is found that in the regenerator and condenser, entropy generation is minimized at a fixed energy effectiveness once the HCR is equal to unity regardless the value of other operating conditions.

4.2 Introduction

The dehumidification and regeneration processes are equally important parts of the liquid desiccant system, but regeneration process is the most critical part due to the association of energy consumption with it. Therefore, a major share of the entropy produced in the liquid desiccant air conditioning system is due to the mass and heat processes taking place in the regenerator and condenser. The irreversibilities of the regenerator and condenser must be minimized in order to optimize the performance of the modified system. A small number of researchers [60 - 62] have attempted previously to study the local irreversibilities of heat and mass exchangers with the aim of optimizing its second law design. The expression for entropy generation rate in heat and mass exchangers as a sum of three terms have been presented by Carrington and Sun [60]. These terms are heat transfer term, mass transfer term and coupled heat and mass transfer term as in,

$$\dot{S}'''_{gen} = \frac{k}{T^2}(\nabla T)^2 + \frac{2\rho^2 \bar{R} D'}{M_A M_B c}(\nabla T)(\nabla x_A) + \frac{\rho^2 \bar{R} D_{AB}}{M_A M_B x_A x_B c}(\nabla x_A)^2 \tag{4.1}$$

On the right hand side of Eq. (4.1), the first term is irreversibility due to heat transfer which is the only term that causes irreversibility in heat exchangers. Therefore, entropy production in heat exchangers can be minimized by balancing stream to stream temperature. However, this does not apply on heat and mass exchangers, since the second and third terms as in Eq. (4.1) (due effect of mass transfer) can contribute a major part in the entropy generation. To evaluate the entropy generation in heat and mass exchangers based on Eq. (4.1), a volume integral should be performed and it requires knowing the local temperature and concentration through the heat and mass exchangers. Thus, using a simple approach is useful to evaluate entropy generation. This chapter presents a control volume process that facilitate the entropy generation minimization. This process uses heat capacity ratio and energy effectiveness for the analysis in the regenerator and condenser.

4.3 Highest COP achievable in the modified system

As mentioned earlier, the entropy is mainly generated in the regeneration process (regenerator and the condenser), hence, in this section the regeneration part will be idealized (100% reversible i.e. zero entropy production) where the dehumidifier is not. The modified system (Fig. 3.1) is converted into black box as shown in Fig. 4.1.

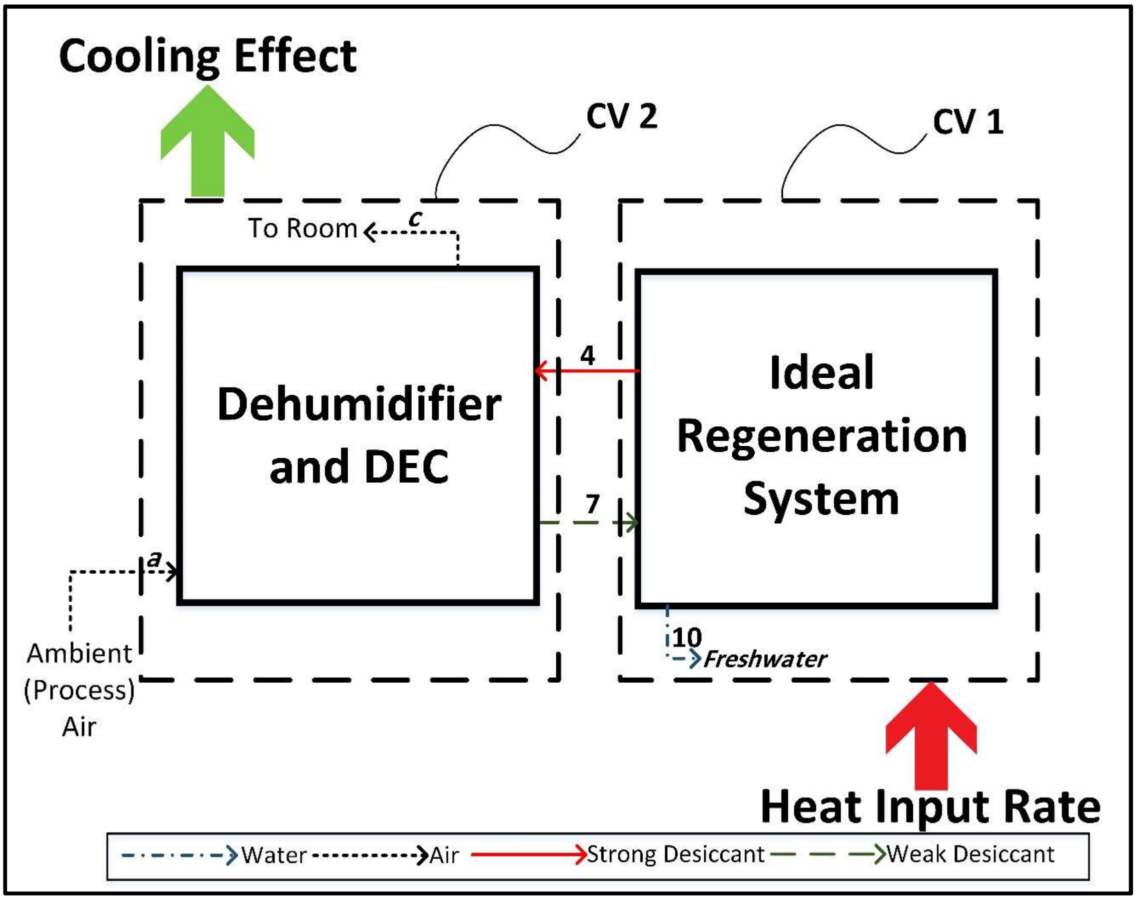


Figure 4.1 Schematic diagram for the modified system for calculating the highest COP achievable.

Figure 4.1 shows the first and second laws application on the modified system. To derive the Carnot COP expression for the modified system, the following assumptions are used to calculate the maximum theoretical cooling effect and the least energy consumption in the modified system:

1. The recovery ratio (ratio of moisture removal rate in the regenerator to the weak desiccant mass flow rate at regenerator inlet) is taken as 10%.

$$RR = \frac{MRR}{\dot{m}_{s,w}} \tag{4.2}$$

2. The strong desiccant solution concentration is taken as $0.3 kg_s\ kg_{sol}^{-1}$.
3. The ambient air condition at the dehumidifier inlet is 30° C and 0.02 $kg_{H_2O}\ kg_{da}^{-1}$.
4. The operating pressure is constant at atmospheric and the ambient surroundings is 30° C.
5. The regeneration temperature is 90° C and the bottom desiccant temperature is 30° C.
6. The analysis is executed for the weak desiccant mass flow rate of 1 $kg\ s^{-1}$.

Applying the first and second laws on CV 1 gives,

$$\dot{Q}_{ht} + \dot{m}_{s,w}h_7 = \dot{m}_{10}h_{10} + \dot{m}_{s,st}h_4 \tag{4.3}$$

$$\frac{\dot{Q}_{ht}}{T_3} + \dot{m}_{s,w}s_7 + \dot{S}_{gen} = \dot{m}_{10}s_{10} + \dot{m}_{s,st}s_4 \tag{4.4}$$

Using these equations,

$$\dot{Q}_{ht} - \dot{Q}_{ht}\frac{T_{bot}}{T_3} - \dot{S}_{gen}T_{bot} = \dot{m}_{10}(h_{10} - T_{bot}s_{10}) + \dot{m}_{s,st}(h_4 - T_{bot}s_4) - \dot{m}_{s,w}(h_7 - T_{bot}s_7) \quad (4.5)$$

$$\dot{Q}_{ht} = \frac{\dot{m}_{10}(h_{10} - T_{bot}s_{10}) + \dot{m}_{s,st}(h_4 - T_{bot}s_4) - \dot{m}_{s,w}(h_7 - T_{bot}s_7) + \dot{S}_{gen}T_{bot}}{1 - \frac{T_{bot}}{T_3}} \quad (4.6)$$

The minimum heat required is at $\dot{S}_{gen} = 0$,

$$\dot{Q}_{ht,min} = \frac{\dot{m}_{10}(h_{10} - T_{bot}s_{10}) + \dot{m}_{s,st}(h_4 - T_{bot}s_4) - \dot{m}_{s,w}(h_7 - T_{bot}s_7)}{\left(1 - \frac{T_{bot}}{T_3}\right)} \quad (4.7)$$

From CV2,

$$Cooling\ effect = \dot{m}_{da,f}(h_a - h_c) \quad (4.8)$$

So, the Carnot COP is given by,

$$COP_{Carnot} = \frac{\dot{m}_{da,f}(h_a - h_c)}{\dot{Q}_{ht,min}} \quad (4.9)$$

Based on the aforementioned assumptions,

$$COP_{Carnot} \quad (4.10)$$

$$= \frac{\dot{m}_{da,f}(h_a - h_c)}{\left(\frac{\dot{m}_{10}(h_{10} - T_{bot}s_{10}) + \dot{m}_{s,st}(h_4 - T_{bot}s_4) - \dot{m}_{s,w}(h_7 - T_{bot}s_7)}{1 - \frac{T_{bot}}{T_3}}\right)}$$

$$= 112.2$$

Figure 4.2 shows the effect of regeneration temperature on rate of highest COP achievable in the modified system with respect to desiccant solution temperature at condenser inlet. From this figure, COP increases as regeneration temperature increases. This is because, mass transfer potential in the regenerator increases as regeneration temperature increases.

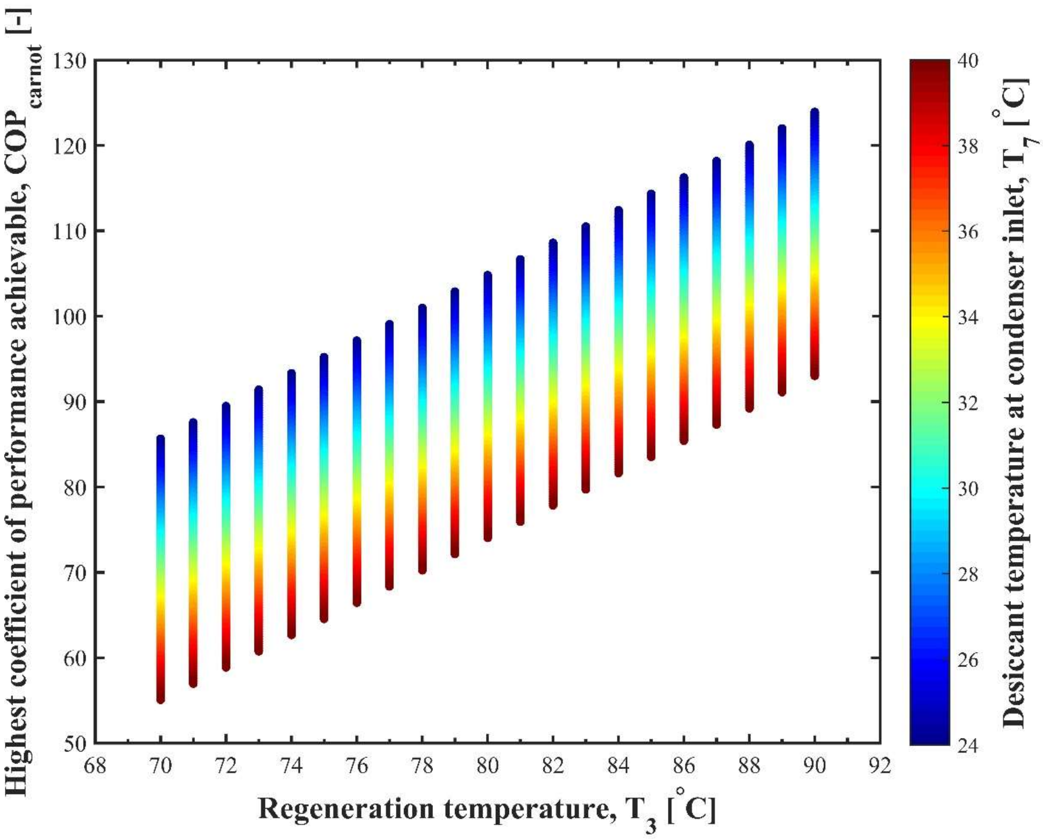


Figure 4.2 Effect of desiccant temperature at condenser inlet and regeneration temperature on COP$_{carnot}$.

As a result, desiccant leaves the regenerator at higher concentration resulting in more air dehumidification and hence, more cooling effect is produced. The desiccant solution that enters the condenser is used as the coolant in the condenser and as coolant temperature increases, the mass transfer potential in the condenser decreases and hence, the decrease in the system performance.

4.4 Heat exchangers

The entropy generation in heat exchangers have been studied previously by several researchers [63 - 70] . Based on this literature, a comprehensive understanding is developed on how to minimize entropy generation in heat exchangers.

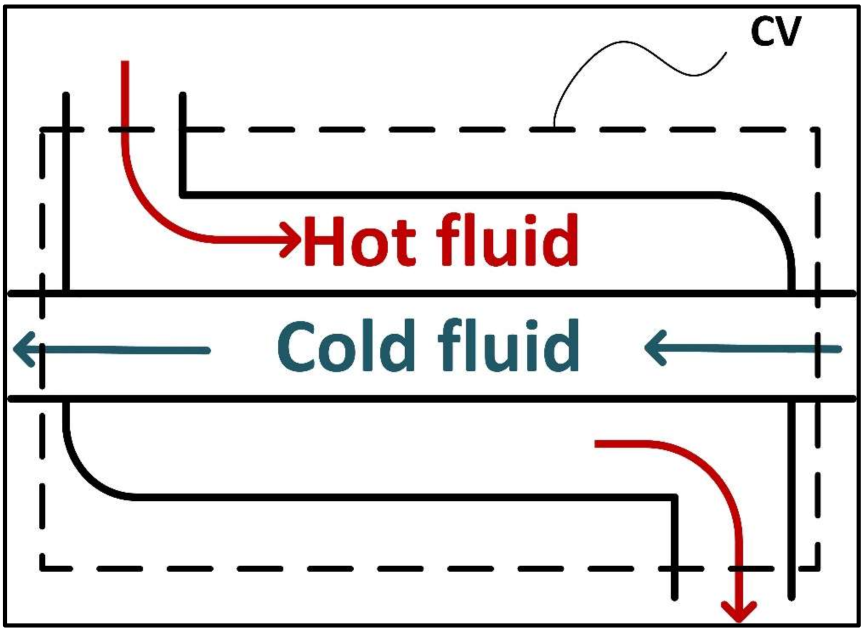


Figure 4.3 Counter flow heat exchanger control volume.

A simple case of a heat exchanger as shown in Fig. 4.3 is considered as the base to explain the concept of reduction of entropy production (thermodynamic balancing) in heat and mass exchangers. It is to be assumed that the phase change is negligible for both hot and cold desiccant streams and heat loss to the surroundings can be neglected.

Applying the first law on the control volume in Fig. 4.3 is expressed by,

$$\dot{m}_{hot}\left(h_{hot,i} - h_{hot,o}\right) = \dot{m}_{cold}(h_{cold,o} - h_{cold,i}) \tag{4.11}$$

Assuming constant specific heat capacity at constant pressure,

$$\dot{m}_{hot}c_{p,hot}\left(T_{hot,i} - T_{hot,o}\right) = \dot{m}_{cold}c_{p,cold}(T_{cold,o} - T_{cold,i}) \tag{4.12}$$

Applying the second law on the control volume in Fig. 4.3 is expressed by,

$$\dot{S}_{gen} = \dot{m}_{hot}\left(s_{hot,o} - s_{hot,i}\right) - \dot{m}_{cold}(s_{cold,o} - s_{cold,i}) \tag{4.13}$$

Or,

$$\dot{S}_{gen} = \dot{m}_{hot}c_{p,hot}ln\left(\frac{T_{hot,o}}{T_{hot,i}}\right) - \dot{m}_{cold}c_{p,cold}(\frac{T_{cold,o}}{T_{cold,i}}) \tag{4.14}$$

The heat capacity rate ratio is defined as in equation (4.15),

$$HCR = \frac{\left(\dot{m}c_p\right)_{cold}}{\left(\dot{m}c_p\right)_{hot}} \tag{4.15}$$

Using this definition and heat exchanger effectiveness (Eqs. 3.10 and 3.11), normalized entropy generation (NEG) can be calculated as in (4.16) and (4.17), Case 1: $\left(\dot{m}c_p\right)_{cold} < \left(\dot{m}c_p\right)_{hot}$:

$$\frac{\dot{S}_{gen}}{\left(\dot{m}c_p\right)_{cold}} = \frac{1}{HCR} * ln\left(1 - HCR * \varepsilon_{hx} * \left(1 - \frac{T_{cold,i}}{T_{hot,i}}\right)\right) + ln\left(1 + \varepsilon_{hx} * \left(\frac{T_{hot,i}}{T_{cold,i}} - 1\right)\right) \tag{4.16}$$

Case 2: $\left(\dot{m}c_p\right)_{hot} < \left(\dot{m}c_p\right)_{cold}$:

$$\frac{\dot{S}_{gen}}{\left(\dot{m}c_p\right)_{hot}} = HCR * ln\left(1 + \frac{1}{HCR} * \varepsilon_{hx} * \left(\frac{T_{hot,i}}{T_{cold,i}} - 1\right)\right) + ln\left(1 + \varepsilon_{hx} * \left(1 - \frac{T_{cold,i}}{T_{ho\ ,i}}\right)\right) \tag{4.17}$$

Figure 4.4 shows the change in entropy generation with respect to HCR at different values for energy effectiveness of the heat exchanger at constant top and bottom temperatures. It shows that normalized entropy generation decreases as HCR increases until it reaches its minimum at HCR = 1 and then starts to increase. At this point (HCR = 1), the heat transfer process in the heat exchanger is reversible i.e. the heat exchanger is thermally balanced.

In a heat exchanger with infinite area (that is $\varepsilon_{hx} = 1$), the entropy generation in such devices is totally due to irreversibility residual. This is directly related with operating conditions upon which heat capacity rate of the cold and hot streams are not equal. It is to be noted that the entropy generation in a thermally balanced heat exchanger with infinite area (HCR = 1, $\varepsilon_{hx} = 1$) is zero as shown in Fig. 4.4.

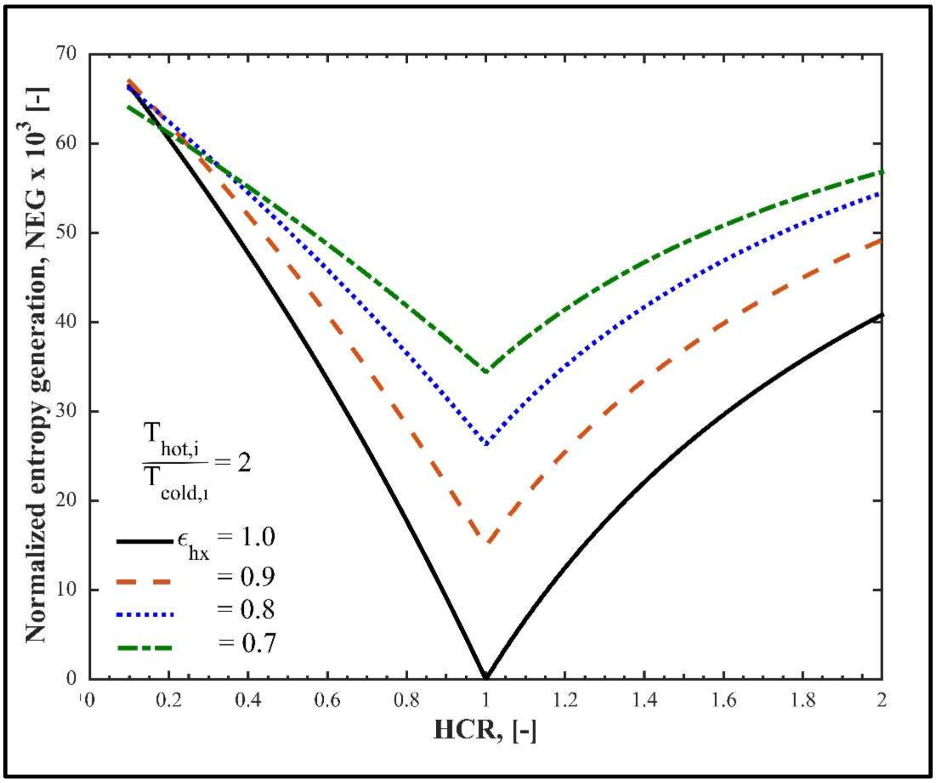


Figure 4.4 Effect of heat exchanger energy effectiveness on entropy generation.

The effect of HCR on entropy generation in the heat exchanger for different top to bottom temperature ratios is presented in Fig. 4.5. It shows that as the ratio increases entropy generation increases. This is because as top to bottom temperature increases the gap between the temperatures increases which justifies the increase in entropy production. From Figs. 4.4 and 4.5, it can be concluded that entropy generation is minimum in a heat exchanger with operating condition at which the fluid streams have constant heat capacity rate; that is, HCR = 1, is thermally balanced i.e., reversible (or zero remnant irreversibility).

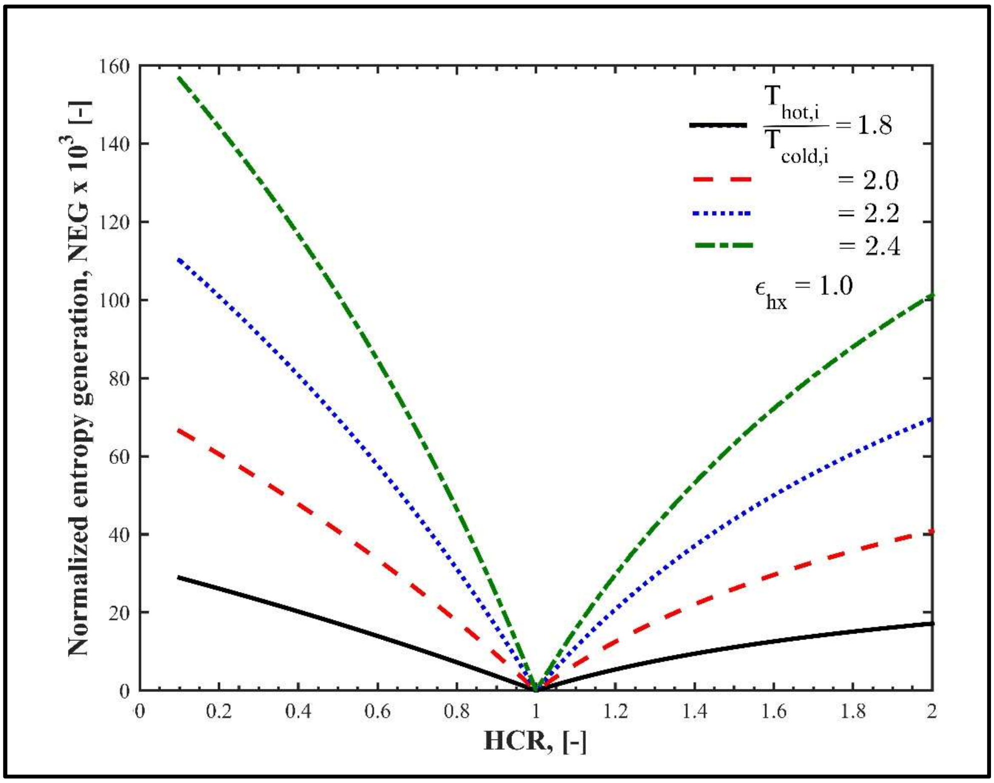


Figure 4.5 Effect of the ratio of top to bottom temperature on entropy generation.

4.5 Heat and mass exchangers

4.5.1 Control volume balancing and modified HCR

The driving force for energy transfer in a combined heat and mass exchanger is a combination of both the temperature and concentration differences. An example of a desiccant regenerator is taken to explain how the entropy generation can be minimized by balancing these driving forces. It is important to note that the heat exchanger terminology that is used in the previous section is not directly applicable to combined heat and mass exchangers. Taking this into consideration, the regenerator effectiveness should be carefully defined (refer to Eq. (3.17)) before considering of balancing these devices.

Applying first and second laws of thermodynamics to the control volume in Fig. 4.6, entropy production in a heat and mass exchanger can be expressed as,

$$(\dot{m}\,h)_{s,i} - (\dot{m}\,h)_{s,o} = (\dot{m}h)_{air,o} - (\dot{m}h)_{air,i} \tag{4.18}$$

$$\dot{S}_{gen} = (\dot{m}\,s)_{s,o} - (\dot{m}\,s)_{s,i} + (\dot{m}s)_{air,o} - (\dot{m}s)_{air,i} \tag{4.19}$$

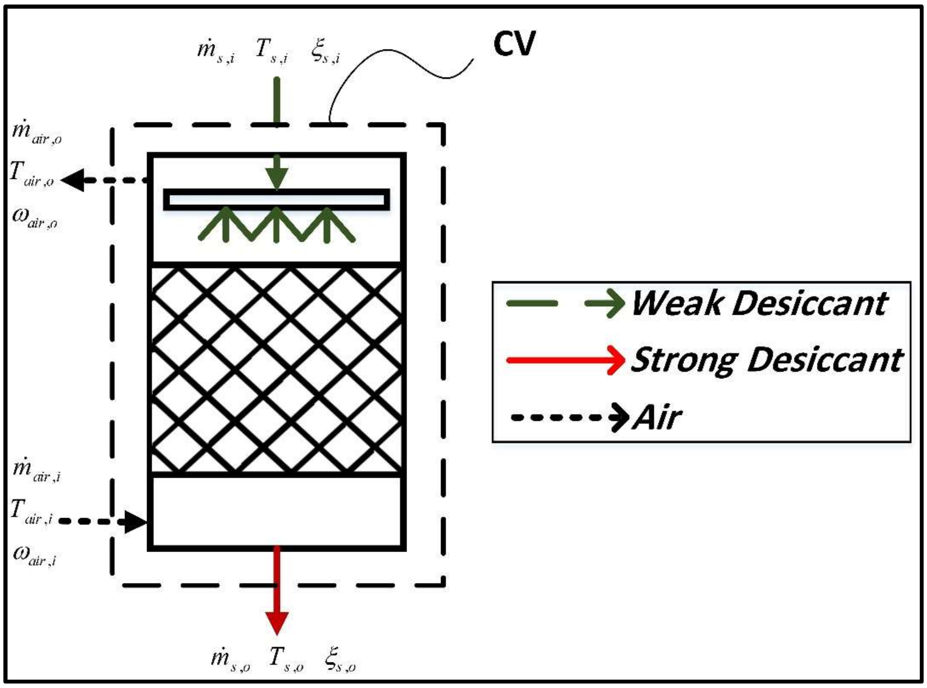


Figure 4.6 Regenerator control volume.

Where mass balance of the control volume in Fig. 4.6 gives,

$$\dot{m}_{s,i} - \dot{m}_{air,i} = \dot{m}_{s,o} + \dot{m}_{air,o} \tag{4.20}$$

Analogous to heat exchangers, a NEG is defined for the regenerator as in Eqs. (4.21) and (4.22),

Case 1: $\Delta\dot{H}_{max,s} < \Delta\dot{H}_{max,a}$:

$$\frac{\dot{S}_{gen}}{(\dot{m}c_p)_{min}} = \frac{\dot{S}_{gen}}{\frac{\dot{m}_{s,i} + \dot{m}_{s,o}}{2}\ c_{p,s}} \tag{4.21}$$

Case 2: $\Delta\dot{H}_{max,s} > \Delta\dot{H}_{max,a}$:

$$\frac{\dot{S}_{gen}}{(\dot{m}c_p)_{min}} = \frac{\dot{S}_{gen}}{(\dot{m}c_p)_{min}} = \frac{\dot{S}_{gen}}{\frac{\dot{m}_{s,i} + \dot{m}_{s,o}}{2}\ c_{p,s}\ HCR} \tag{4.22}$$

These equations cannot be solved as there is one extra unknown. A definition for HCR should be expressed to close the set of equations. As mentioned earlier, due to difference between 'heat exchange' and 'heat and mass exchange' devices a modified HCR should be defined. Based on heat exchanger analysis, this HCR can be expressed as follows,

$$HCR_{hx} = \frac{(\dot{m}c_p)_{cold}}{(\dot{m}c_p)_{hot}} \tag{4.23}$$

Since maximum temperature difference is the same for cold and hot stream in the heat exchanger i.e. $\Delta\dot{H}_{max,state} = (\dot{m}c_p)_{state}(T_{hot,i} - T_{cold,i})$, where state = cold or hot. Therefore, HCR for heat exchanger can be given as,

$$HCR_{hx} = \frac{\Delta\dot{H}_{max,cold}}{\Delta\dot{H}_{max,hot}} \tag{4.24}$$

In like manner, Eq. (4.24) can be used for heat and mass exchange devices taking in consideration a new definition for$\Delta\dot{H}_{max}$. Hence, HCR for the regenerator can be defined, as the ratio of the maximum difference in enthalpy rate of the air stream to that of the desiccant stream as in,

$$HCR_r = \frac{\dot{m}_{air,o} h_{air,o,id} - \dot{m}_{air,i} h_{air,i}}{\dot{m}_{s,i} h_{s,i} - \dot{m}_{s,o} h_{s,o,id}} \tag{4.25}$$

4.5.2 Solution method

The functional dependence of entropy generation on other variables has been shown by Eqs. (3.17), (4.18 – 4.22) and (4.25). The entropy generation in the regenerator can be predicted by solving aforementioned equations simultaneously. Hence, the calculations in this research are executed by EES which uses accurate properties of both LiCl solution and air-water vapor mixture (moist air). The EES software has been used for analysis of several publications [71-77].

4.5.3 Code correctness check

The EES code were tested against several extreme situations. For instance, at ε_r=1, minimum terminal change in temperature for both air and desiccant streams were zero; and the moisture removal rate was maximum. At ε=0, the output of any parameter (e.g. mass flow rate, temperature and concentration) of both desiccant and air had the same values of input variable; and the moisture removal rate was zero. As well, the calculations were conducted repeatedly many times to ensure consistency.

4.5.4 Effect of desiccant inlet temperature

Figure 4.7 shows the effect of HCR on NEG at fixed values of air inlet temperature and regenerator effectiveness with respect to different values of desiccant inlet temperature. From this figure two observations can be made: first, regardless the value of desiccant inlet temperature the NEG is minimum at HCR = 1.; second, the relation between desicant inlet temperature and NEG is non-monotonic and it differs with respect to HCR.

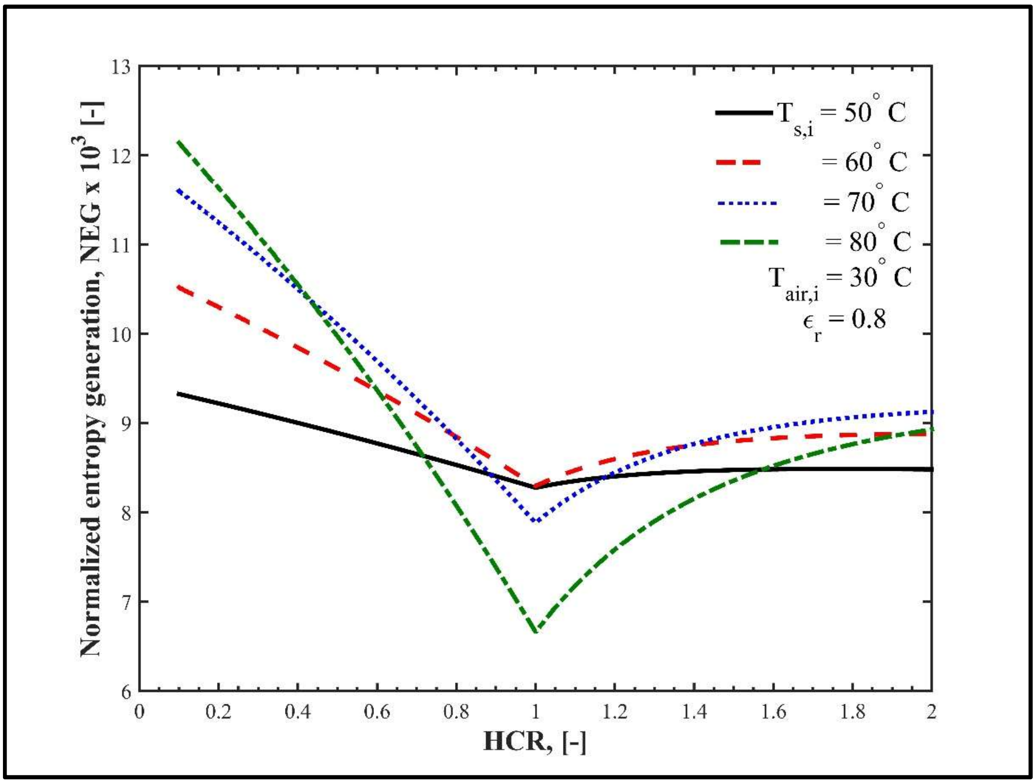


Figure 4.7 Effect of desiccant inlet temperature on entropy generation in the regenerator.

4.5.5 Effect of air inlet temperature

Figure 4.8 illustrates the effect of HCR on NEG at fixed values of desiccant inlet temperature and regenerator effectiveness with respect to different values of air inlet temperature. As in section 4.5.4, the NEG is minimized at HCR = 1 at all values of inlet air temperatures. Furthermore, it is to be observed that NEG relation with the air inlet temperature is non-monotonic.

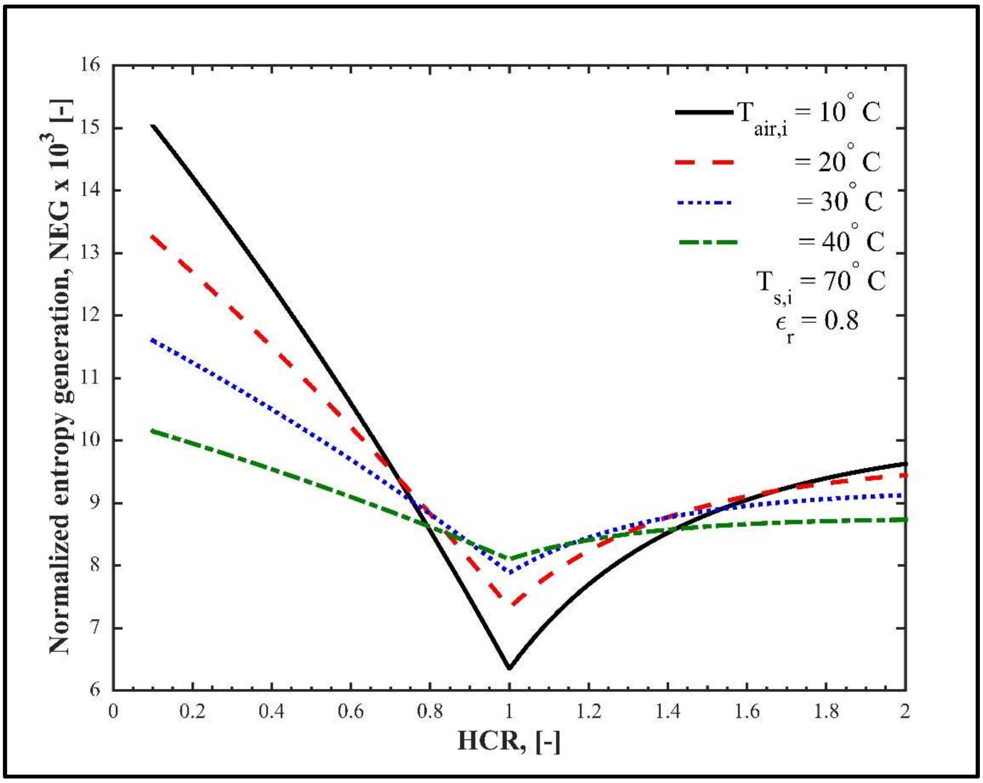


Figure 4.8 Effect of air inlet temperature on entropy generation in the regenerator.

4.5.6 Effect of regenerator energy effectiveness

Based on the definition of regenerator energy effectiveness as in Eq. (3.17), its effect of NEG with respect to HCR is shown in Fig. 4.9. For all values of energy effectiveness, the minimum NEG is reached at HCR = 1. As it is predicted, in the case of a regenerator with higher effectiveness (large heat and mass exchange area), the NEG is less than the case of a regenerator with lower effectiveness at the same values of desiccant and air inlet temperatures.

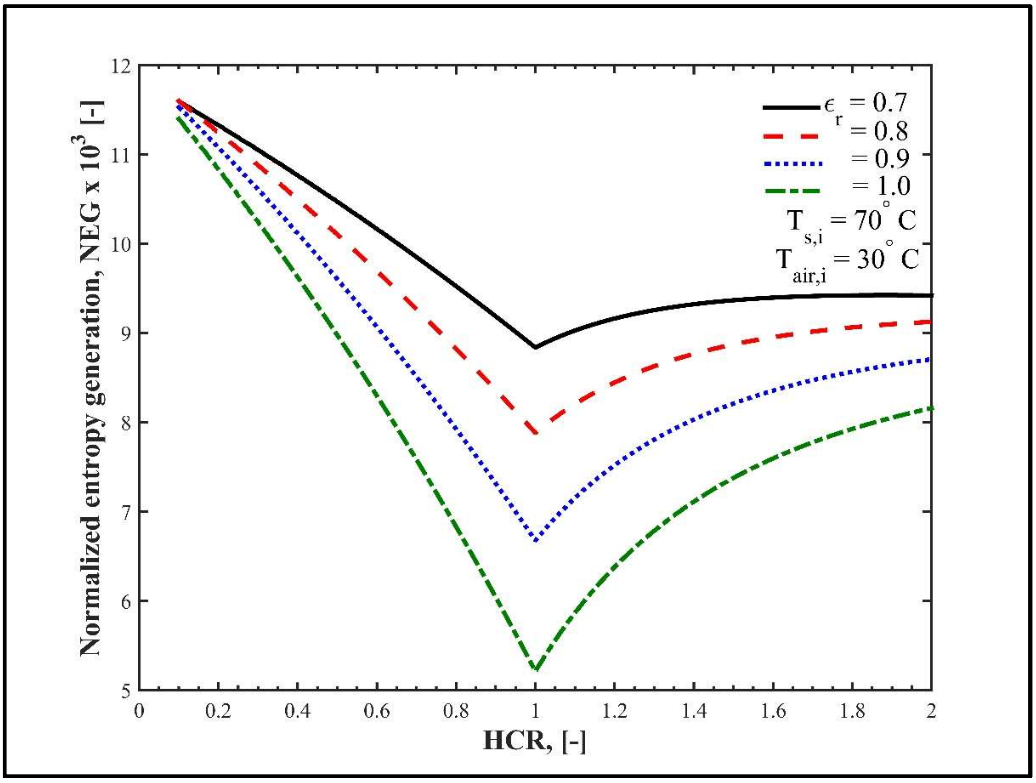


Figure 4.9 Effect of regenerator energy effectiveness on entropy generation.

4.5.7 Effect of HCR between 'heat' and 'heat and mass' exchangers

Figures 4.7-4.9 show the effect of desiccant inlet temperature, air inlet temperature and regenerator effectiveness on NEG plotted against heat capacity rate ratio. The factor of interest in these figures to be studied is HCR; it is important to be noted that, the NEG is always minimum at HCR=1 regardless of the values of operating conditions. This is an important result and based on that, it could be said, the balanced state is defined by HCR being unity for heat and mass exchangers regardless of whether it is a fixed operating conditions or a fixed effectiveness.

Operationally, HCR can be varied by only changing the desiccant-to-air mass flow rate ratio because the energy effectiveness and inlet temperatures cannot be controlled. It is important to be noted that the entropy generation in a thermally balanced heat exchanger with infinite heat exchange area (HCR = 1, ε_{hx} = 1) is zero as shown in Fig. 4.4; but it is still higher than zero as shown in Fig. 4.9 in a thermally balanced regenerator with infinite heat and mass exchange area (HCR = 1, ε_r = 1). This is because, in the balanced heat exchanger the stream-to-stream temperature difference is constant throughout the length of heat exchanger. While in the balanced regenerator the stream-to-stream temperature and concentration difference is constant at the terminal points only. This is a major difference between balancing the heat and mass exchanger and heat exchanger.

However, this is a 'control volume' balanced state in which the system does not include injections or extractions i.e. zero extraction and this could be extended to concept of complete thermodynamic balancing in heat and mass exchangers by varying mass flow rate ratio through the regenerator or condenser processes path (infinity extraction) or to discrete balancing (single extraction).

4.6 Conclusions

The effect of HCR and the operating conditions on the NEG in the heat and heat and mass exchanger have been examined. To understand and optimize the second law performance of the regenerator and condenser, a comprehensive study has been carried out. From the aforementioned discussion, the following conclusions can be drawn:

1. HCR is defined for the regenerator as the ratio of the maximum possible change in total enthalpy rate of the air to the maximum possible change in total enthalpy rate of the desiccant.

2. Main factors that affect NEG are HCR, energy effectiveness and desiccant and air inlet temperatures. Operationally, HCR can be varied by only changing the desiccant-to-air mass flow rate ratio because the energy effectiveness and inlet temperatures are hard to be controlled
3. Bringing HCR to unity reduces entropy generation and improve system performance noticeably.
4. In the balanced heat and mass exchangers, the stream-to-stream temperature and concentration difference is constant at the terminal points only. Therefore, varying desiccant-to-air mass flow rate ratio using single mass extraction and injection brings HCR at extraction point to unity. Hence, local irreversibilities through the regenerator and condenser can be tremendously decreased i.e. leads to better system performance.

CHAPTER 5

Thermodynamic balancing of a novel regeneration process by mass extraction and injection technique

5.1 Summary

The idea of the modified system as proposed in chapter 3 is promising. The roadblock to commercialize this system is that the increase in the system performance due to modification is relatively modest. In this chapter, thermodynamic balancing of the regeneration process in the modified system is proposed by using mass extraction and injection technique. This technique is considered as a potential mean to reduce energy consumption and increase the modified system performance dramatically. Thermodynamic balancing reduces the production of entropy generation in the regeneration process by balancing stream-to-stream temperature and concentration difference at terminal and extraction points. The proposed cooling (single extraction) system is the modified (zero extraction) system (Fig. 3.1) with applying mass extraction and injection between the regenerator and the condenser. The mathematical procedure to model the system was outlined to study the effect of the extraction on the cooling system performance. Using the generated model, it is found that, at Ψ= 20 kJ/kg dry air; single extraction system performance is 85.7 % better than zero extraction system and produces 94.2 kg of fresh water per hour as a by-product.

5.2 Introduction

Optimum design system is one in which entropy generation is minimum at fixed cost and size. The efficiency of such systems is optimized by using finite time thermodynamics. In this chapter, this principle is applied to the regeneration process of the modified system (regenerator and condenser) to improve the performance of the modified air conditioning system. In the modified system, the carrier air that shuttles heat and moisture between the regenerator and condenser. It is found as mentioned in chapter 4 that reduction of the entropy production in the modified system improves the performance (COP) of the system. Also, it is found that extracting air from the regenerator and injecting it into the condenser can bring enthalpy rates into balance and lower the entropy generation. In this chapter, extensive analysis is reported to study the effect of mass extraction and injection on the modified air conditioning system. This analysis depends on the idea that at constant regenerator and condenser effectivenesses, the modified system operates optimally for any given operating conditions at sole desiccant-to-air mass flow rate ratio. Design algorithm is developed to balance the regeneration process in the modified system. It is to be noted that thermodynamic balancing has not been carried out in a liquid desiccant cooling system and it was a giant development in liquid desiccant AC performance as expected.

5.3 Modified liquid desiccant air conditioning cycle with single mass extraction and injection of the carrier air

Figure 5.1 shows the modified desiccant-based air conditioning system with single mass extraction and injection of the carrier air.

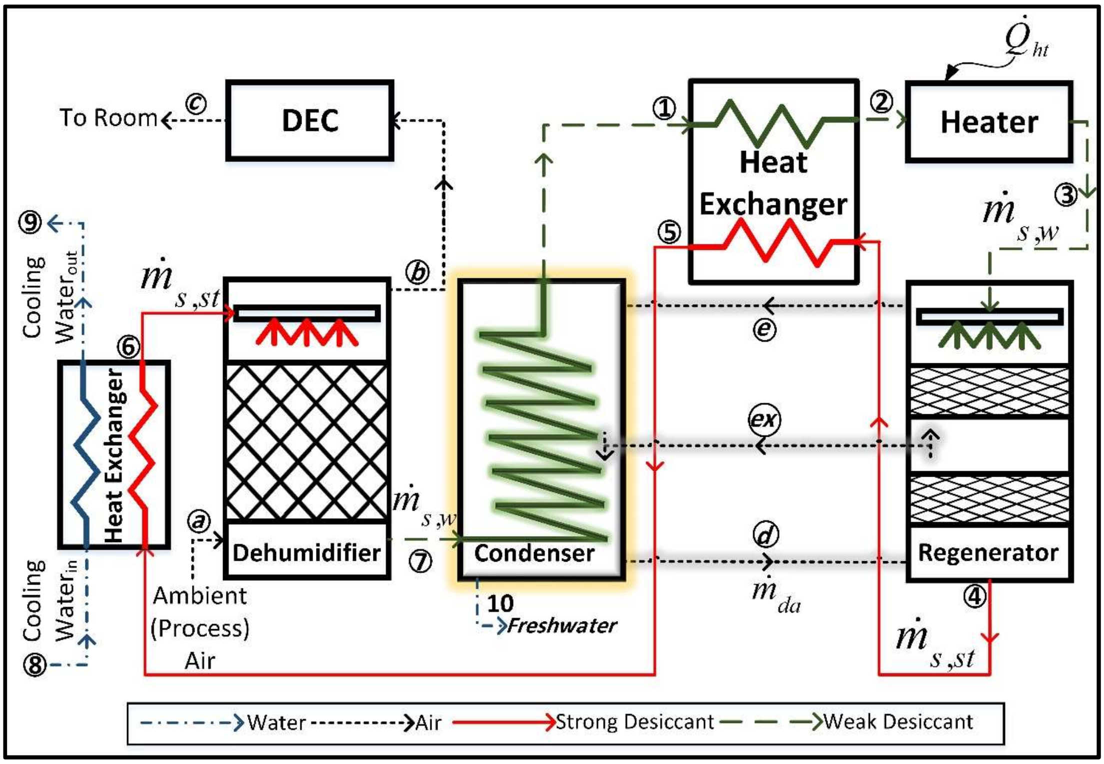


Figure 5.1 Schematic of the modified liquid desiccant air conditioning system with single mass extraction and injection of the carrier air.

This embodiment contains a dehumidifier, a DEC, a regenerator, a heater, a heat exchanger and a condenser. The difference in components between the modified and proposed system is that carrier air is extracted and injected between the regenerator and the condenser. That makes the single extraction cycle a two stage zero extraction cycles. The extracted air from the regenerator (state *ex*) will be injected in the condenser to a point with the same state of the extraction point to avoid entropy generation due to mixing.

5.4 Thermodynamic balancing of the regenerator and condenser

In thermal system, entropy is produced due to its finite size and the heat and mass transfer operations occur in its components. In the liquid desiccant cooling system most of the thermal operations occur in the regeneration process. Therefore, a major part of the entropy is generated in the regenerator and condenser. The entropy generation in the regenerator and condenser should be addressed in order to improve the cooling system performance. The control volume method that is developed in section 4.5 is revisited and extended for single balancing by extraction and injection. In this section, for regenerator and condenser modelling, enthalpy pinch is used as an appropriate alternative to temperature pinch [43], [48], [78] and energy effectiveness [79-81].

5.4.1 Enthalpy pinch

A temperature versus enthalpy diagram is plotted as in Fig. 5.2 in order to visualize simultaneous heat and mass transfer processes in both the regenerator and condenser.

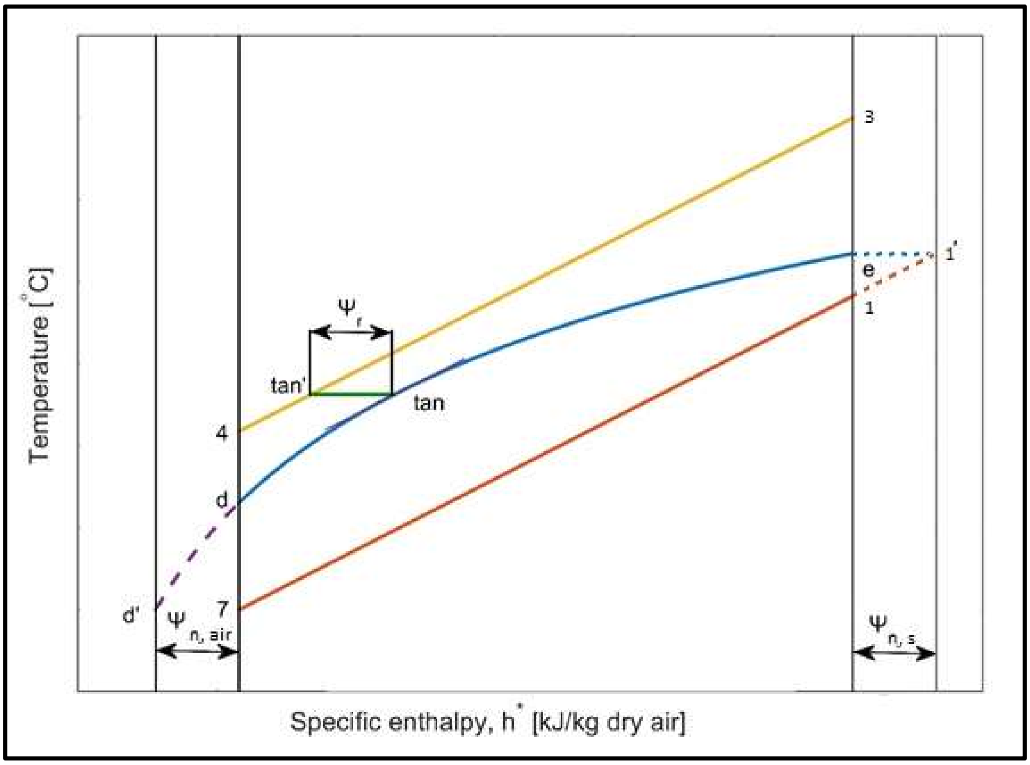


Figure 5.2 Temperature-enthalpy diagram for a desiccant system without extraction.

The curved line d-e represents the air process through the regenerator and the condenser. The line 7-1 and the line 3-4 represent the desiccant process through the condenser and the regenerator, respectively.

States *d'* and *1'* represent the ideal states that desiccant and carrier air streams, respectively, would have reached in infinite size condenser. Therefore, (h^*_d - $h^*_{d'}$) is represented as $\Psi_{n, air}$ and ($h^*_{1'}$ - h^*_1) is represented as $\Psi_{n, s}$ and they can be defined as the loss in enthalpy rate (per kg of dry air circulating between the regenerator and condenser) due to finite-sized regenerator and condenser. This loss cannot be reduced by thermal balancing without increasing the heat and mass transfer area (energy efectiveness). It can be concluded that this loss (enthalpy pinch) is the opposite of energy effectiveness.

Normalization of enthalpy pinch by the quantity of dry air recirculating in the system was proposed by McGovern et al. [48] so the thermodynamic processes as shown in Fig. 5.2 is presented. This concept is used through this chapter by dividing Eq. (3.14) numerator and denominator by this amount of dry air,

$$\varepsilon = \frac{\Delta h^*}{\Delta h^*_{max}} = \frac{\Delta h^*}{\Delta h^* + \Psi_{\mathrm{TD}}} \tag{5.1}$$

Where Ψ_{TD} is the minimum loss in enthalpy at terminal points due to size limit in heat and mass exchangers and is defined as follow,

$$\Psi_{\mathrm{TD}} = min[\Psi_{\mathrm{n,s}}, \Psi_{\mathrm{n,air}}] = min\left[\frac{\Delta h^*_{max,s}}{\dot{m}_{da,r}} - \Delta h^*, \frac{\Delta h^*_{max,air}}{\dot{m}_{da,r}} - \Delta h^*\right] \tag{5.2}$$

In a heat exchanger, Ψ_{TD} is similar to terminal temperature pinch which can be defined as the minimum loss in temperature due to a finite size limit in the heat exchanger. Based on that, it could be said Ψ for heat and mass exchangers is analogous to temperature pinch for a heat exchange device and can be used to define the enthalpy rate loss at any point of heat and mass exchanger. The energy effectiveness is usually used as performance metric for 'heat exchangers' and 'heat and mass exchangers'. But, this parameter accounts only for terminal differences. In order to design for balancing by altering desiccant-to-air mass flow rate ratio through the process path, local differences are needed to be considered. From Fig. 5.2, it is to be noted that the regenerator pinch point does not occur at the terminal locations but rather at an intermediate point. This behavior cannot be captured using energy effectiveness.

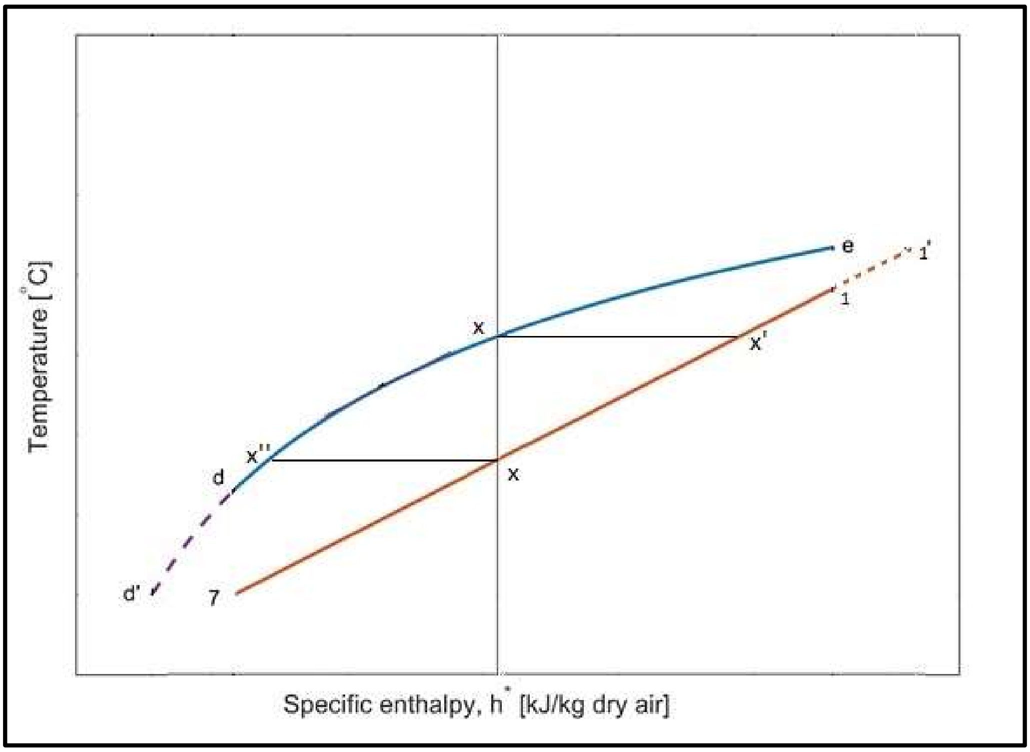


Figure 5.3 Temperature-enthalpy chart for a condenser underlining local enthalpy pinch.

Another design issue, that high values of effectiveness in the regenerator in extreme case might lead to cross of concentration or temperature. Enthalpy pinch does not have this problem due to being a local parameter. Therefore, it is used in this chapter as a defining parameter of performance for heat and mass exchange devices (regenerator and condenser). Figure 5.3 shows infinitesimal small volume represented by the line that goes through point x. This point has two ideal states x' and x''. The local enthalpy pinch (Ψ) at location x is defined as follows,

$$\Psi = min[(h^*_{x'} - h^*_{x}), (h^*_{x} - h^*_{x''})] \tag{5.3}$$

5.4.2 Control volume balancing (zero extraction)

As stated earlier, a balanced system could be defined as a system in which entropy generation is minimized at a fixed energy effectiveness when the modified HCR is equal to one. This could be extended to enthalpy pinch by dividing both numerator and denominator in Eq. (4.24) by mass flow rate of dry air as in,

$$HCR = \frac{\Delta \dot{H}_{max,s}}{\Delta \dot{H}_{max,air}} = \frac{\Delta h^* + \Psi_{\mathrm{n,s}}}{\Delta h^* + \Psi_{\mathrm{n,air}}} \tag{5.4}$$

From Eq. (5.4), for HCR to be equal unity in the condenser, $\Psi_{\mathrm{n,s}} = \Psi_{\mathrm{n,air}}$ is needed (varying desiccant-to-air mass flow rate ratios). It should be noted, these enthalpy pinch points are located at the inlet and outlet of the condenser, and at a single intermediate location in the regenerator.

5.4.3 Mass extraction and injection-based balancing (single extraction)

In the case presented in previous section (control volume balancing), HCR is brought to unity by varying desiccant-to-air mass flow rate ratios at terminal points. For such case, HCR is not equal to unity at every locations in the heat and mass exchanger. However, the control volume balancing method could be extended by varying desiccant-to-air mass flow rate ratios through the device by mass extraction and injection of air which results in HCR equal to unity at every point of the device (i.e. Ψ is constant). This method is called 'complete thermodynamic balancing' (where number of extractions and injections tends to ∞) which is impossible to achieve. On the other hand, discrete extraction and injection is practical and single extraction is extensively investigated in this chapter. The local enthalpy pinch through the condenser with/without extraction is illustrated in Fig. 5.4. It can be observed that single extraction brought local enthalpy pinch to minimum (HCR = 1) at the injection point as well as the terminal points which leads to tremendous decrease in entropy generation as shown in Fig. 5.5. From this figure, the general trend in both zero and single extraction devices is that, entropy generation increases as the enthalpy pinch increases; because enthalpy pinch is the inverse of energy effectiveness. However, entropy generation in single extraction device is much less than zero extraction up to certain point after which single extraction has no effect. The reason of this behavior is explained at length in the results and discussion section.

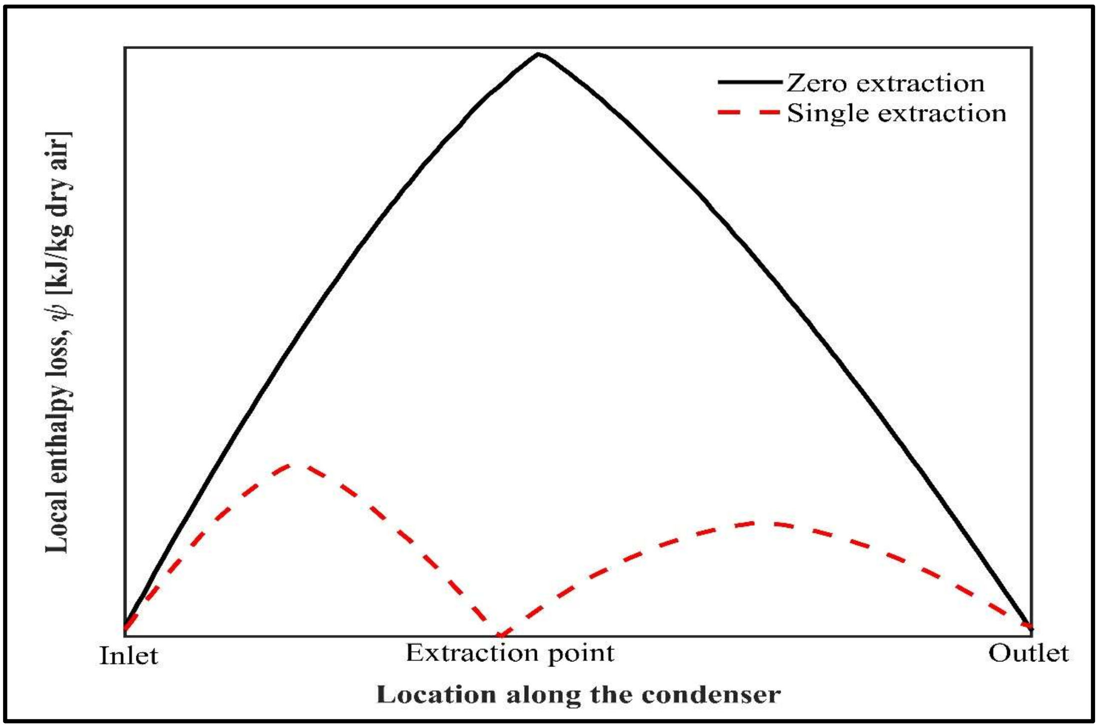


Figure 5.4 Local enthalpy pinch through the condenser for zero and single extractions.

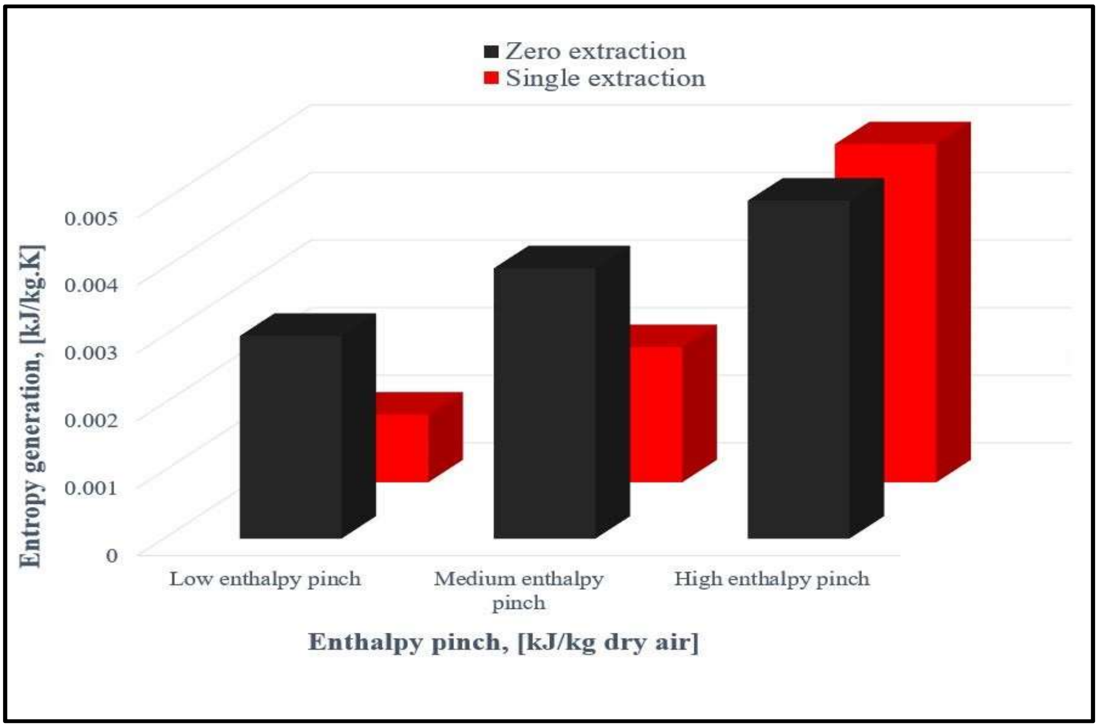


Figure 5.5 Effect of single extraction on reduction of entropy production.

5.5 Modeling of the proposed systems

In this section, concepts of thermodynamic balancing developed for heat and mass exchange devices is used and applied to the proposed systems design. An embodiment for systems to be studied is already illustrated in Fig. 5.1.

5.5.1 Zero extraction system

To evaluate desiccant-to-air mass flow rate ratio, its relation with desiccant lines slope ('7-1' and '3-4) in both the condenser and regenerator, respectively, as in temperature-enthalpy pinch diagram (Fig. 5.2) can be used. Figure 5.6 shows a control volume that contains the heater, section from the condenser and the corresponding section in the regenerator.

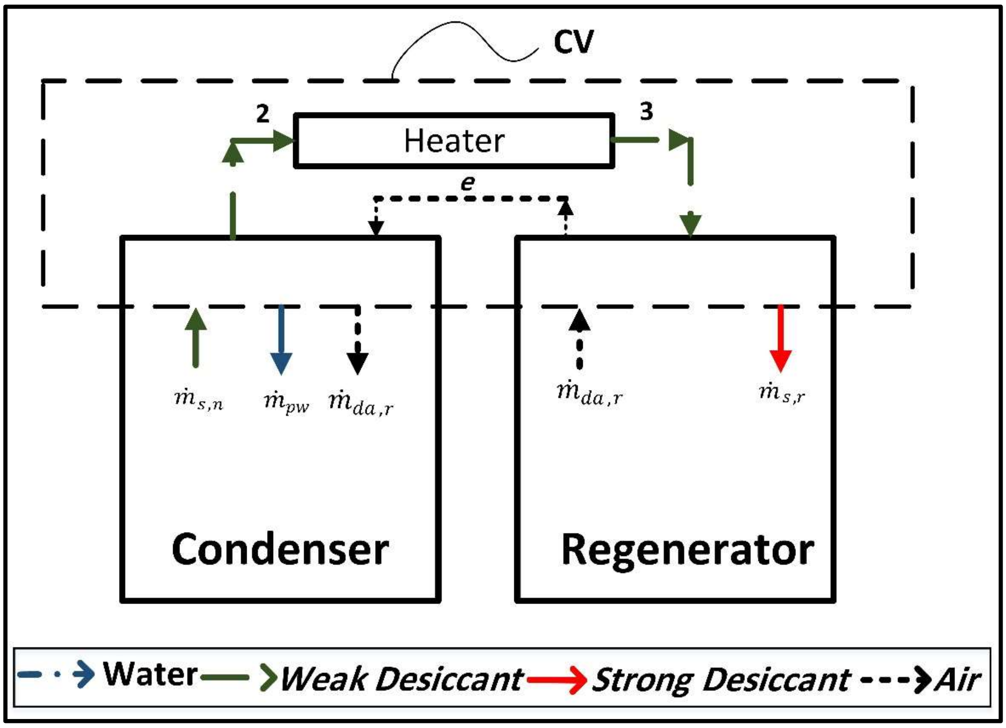


Figure 5.6 A control volume containing the heater, a section of the condenser and the corresponding section of the regenerator.

An energy balance on the control volume as shown in Fig. 5.6 can be expressed as:

$$(\Delta\dot{H})_s - \left(\Delta\dot{H}\right)_{pw} = (\Delta\dot{H})_{da} \tag{5.5}$$

Or,

$$\dot{m}_s\left(C_{p,ave}\Delta T\right)_s - \dot{m}_{pw}\left(C_{p,ave}\Delta T\right)_{pw} = \dot{m}_{da}(\Delta h^*)_{da} \tag{5.6}$$

For an infinitesimally small control volume, the change in enthalpy rate of pure water is mainly due to temperature change, so the change in mass flow rate of pure water inside the control volume can be neglected. Assuming that, at any location in the condenser, the condensed water is at the same temperature as the desiccant, and that specific heat capacity of desiccant is constant, the energy balance in Eq. (5.6) simplifies to,

$$(\dot{m}_s - \dot{m}_{pw})C_{p,ave}\Delta T = \dot{m}_{da}(\Delta h^*)_{da} \tag{5.7}$$

Taking the limit as Δh^* tends to zero, the following expression is obtained for the slope of process path followed by the desiccant stream in the condenser on the temperature-enthalpy diagram,

$$\frac{dT}{dh^*} = \frac{1}{m_r C_{p,ave}} \tag{5.8}$$

Where,

$$\mathrm{C_{p,ave}} = \frac{(\mathrm{C_{p,ave}})_\mathrm{s} + (\mathrm{C_{p,ave}})_\mathrm{pw}}{2} \tag{5.9}$$

Based on Eq. (5.8), a comprehensive algorithm to evaluate the proposed system performance using regeneration temperature, the desiccant temperature at the condenser inlet and enthalpy pinch of both the regenerator and condenser as input data is presented in Fig. 5.7.

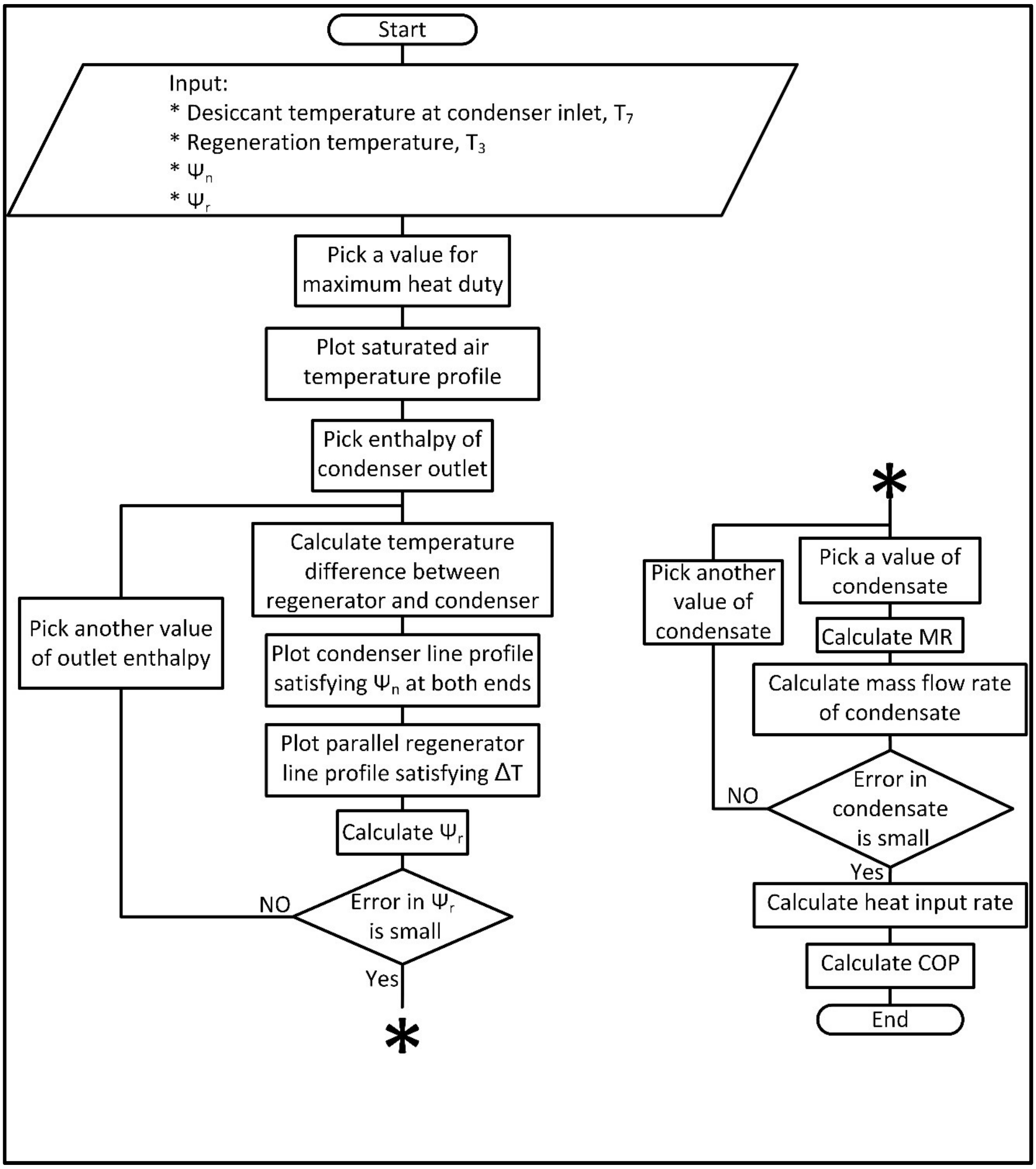


Figure 5.7 Zero extraction algorithm flowchart.

5.5.2 Single extraction system

Figure 5.8 illustrates a temperature-enthalpy chart of a system with a single injection and extraction. In this case, the air is extracted from the regenerator at the state '*ex*' and is injected to the corresponding location in the condenser with the same state '*ex*' to avoid generating additional entropy due to mixing. The major variation in the single-extraction model is that thermal balancing process is executed in two stages. The concept established in this chapter results mainly in generating of Fig. 5.8. Visualization and modeling of this system depends on this figure.

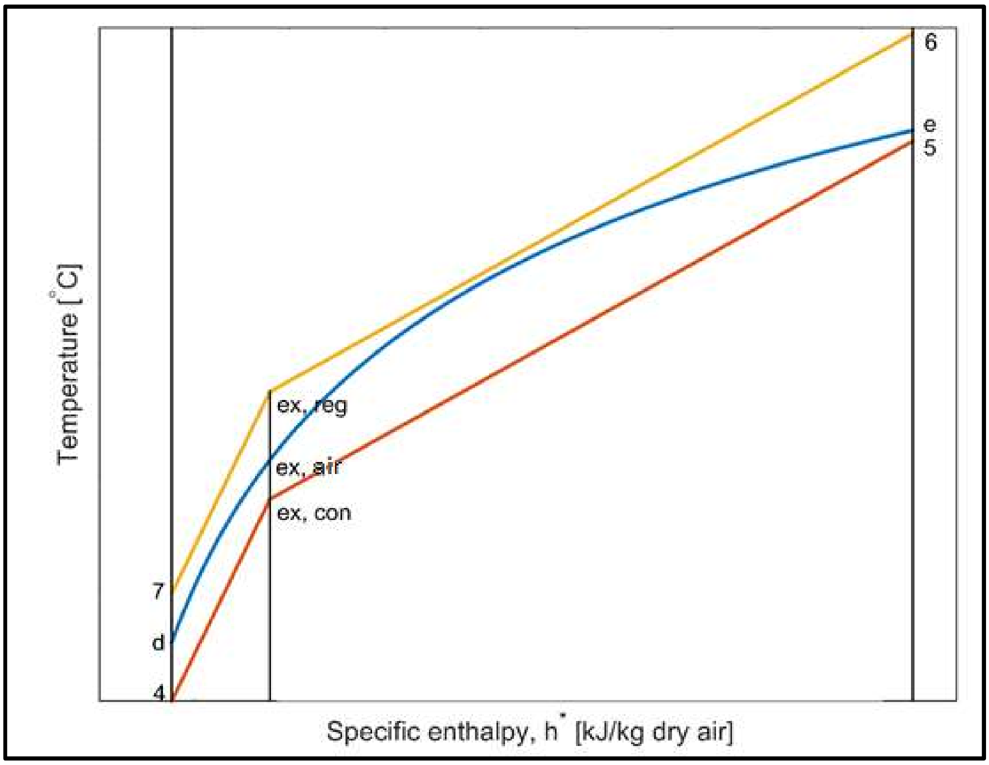


Figure 5.8 Temperature-enthalpy diagram for a desiccant system with single extraction.

The idea is to convert this figure into a mathematical model to evaluate the performance of the single extraction system. The detailed procedure to model the single extraction system is shown in Fig. 5.9.

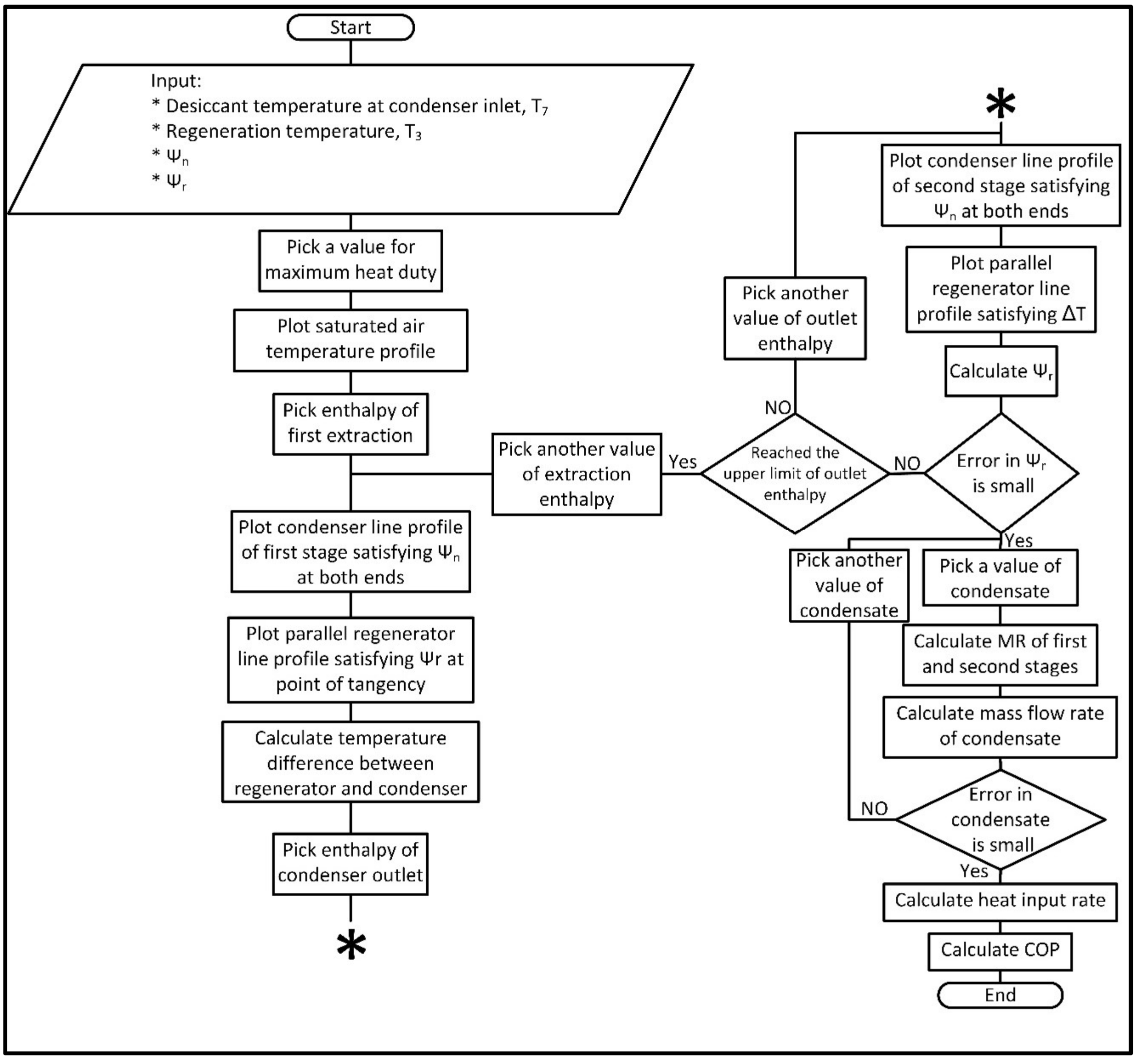


Figure 5.9 Single extraction algorithm flowchart.

5.5.3 Operating conditions

The extraction technique is executed only in the regeneration process, hence, the inputs that affect extraction are ones those affect this process, as follow: the weak desiccant temperature and concentration at condenser inlet, the regeneration temperature, the regenerator enthalpy pinch and the condenser enthalpy pinch. The comparison between the zero and single extraction systems is established taking into account the effect of varying the aforementioned parameters. Typical values of these parameters and their ranges are presented in Table 5.1.

Table 5.1 Basic values and variation of operating parameters.

Parameter		Basic value
Weak desiccant flow rate	kg s^{-1}	1
Weak desiccant temperature	°C	25
Regeneration temperature	°C	80
Enthalpy pinch	kJ kg_{da}^{-1}	20

5.5.4 Model validation

In order to use the mathematical model developed in the chapter with confidence, the predicted values is compared with the results from literature for validation. Applying extraction on desiccant cooling system has not been studied before while it is well studied in the desalination literature and hence, the current model will be validated with results of desalination system in the literature. Chehayeb et al. [54] presented a reliable data for GOR in zero and single extraction desalination system. Narayan et al. [52] reported data of terminal point temperatures of the zero and single extraction desalination system. The results obtained from the current study model are compared with the results reported in [54] and [52] and presented in Table 5.2 and 5.3, respectively. The current model results show good agreement within $\pm$ 2.2 %.

Table 5.2 Validation of the current results with the water heated HDH system [54] at different enthalpy pinch for the zero and single extraction system.

	Zero extraction			Single extraction		
Enthalpy pinch (kJ kg^{-1}_{da})	Ref. [54] GOR [-]	Current work GOR [-]	Error (%)	Ref. [54] GOR [-]	Current work GOR [-]	Error (%)
0	2.83	2.85	0.9	8.48	8.46	0.2
5	2.73	2.71	0.7	7.08	7.05	0.5
10	2.71	2.7	0.3	6.00	5.98	0.4
15	2.69	2.69	0.0	4.92	4.9	0.5
20	2.45	2.44	0.4	4.16	4.14	0.5
25	2.43	2.4	1.2	3.87	3.85	0.4
30	2.26	2.26	0.0	2.33	2.31	1.1
35	2.18	2.19	0.6	2.32	2.3	0.6
40	1.88	1.85	1.8	1.88	1.85	1.8

Table 5.3 Comparison between Ref. [52] and the current work for temperatures at the terminal points.

	Zero extraction			Single extraction		
Terminal points	Ref. [52] (ºC)	Current work (ºC)	Error (%)	Ref. [52] (ºC)	Current work (ºC)	Error (%)
Dehumidifier outlet	62.6	64	2.2	70.2	69.9	0.4
Humidifier outlet	36	35.9	0.3	29.8	29.6	0.7
Bottom air temperature	25.5	25.2	1.2	25.5	25.2	1.2
Top air temperature	64.3	65.7	2.2	71.1	70.9	0.3
Dehumidifier extraction	-	-	-	38.5	38.1	1.0
Air extraction	-	-	-	41.9	41.6	1.0
Humidifier extraction	-	-	-	48.2	47.7	1.0

5.6 Results and discussion

5.6.1 Effect of enthalpy pinch

Figure 5.10 represents the effect of enthalpy pinch on the performance of the proposed system with zero and single extractions. It can be seen from this figure the general trend is that the COP decreases as enthalpy pinch increases for both zero and single extractions system. This is because enthalpy pinch is the inverse of effectiveness. Region 'A' is considered as the high effectiveness region with enthalpy pinch ranges from 0 to 50 kJ/kg dry air, which is equivalent to effectiveness varying from 100% to 91% for both the regenerator and condenser.

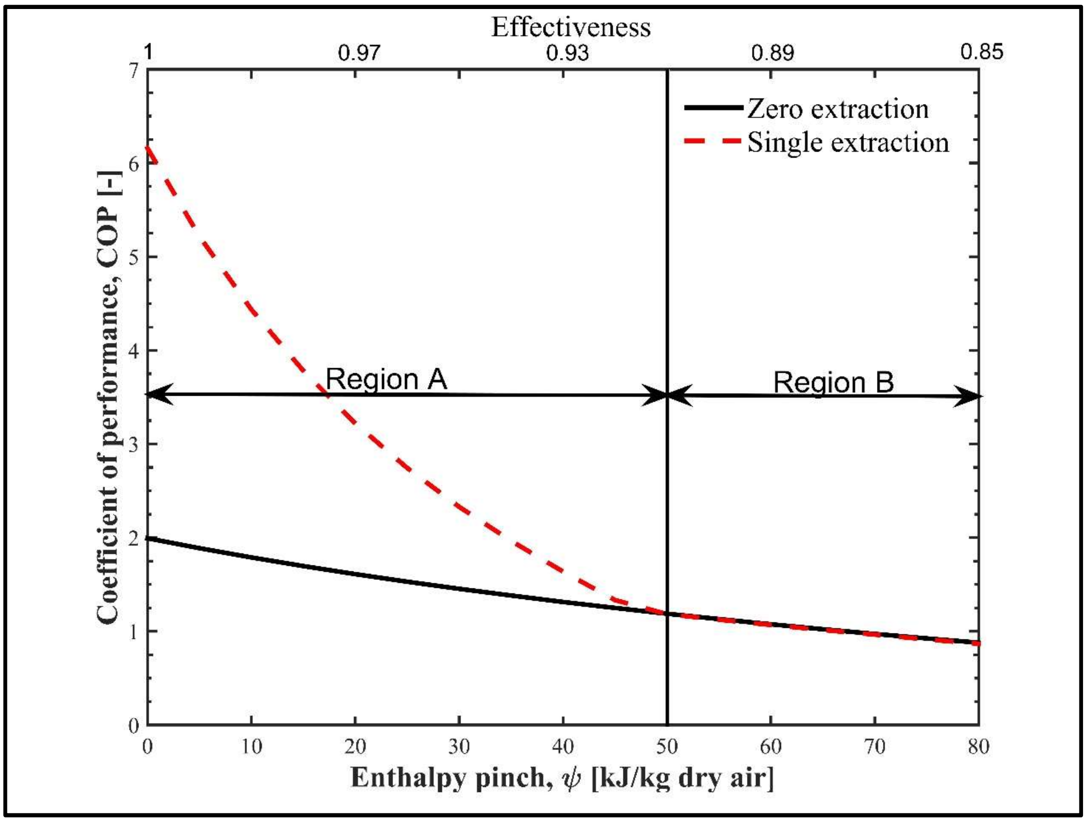


Figure 5.10 Effect of enthalpy pinch on COP.

The single extraction has higher COP in region 'A' than zero extraction for the proposed system. It starts with a COP of 6.2 and then decreases to COP slightly above 1. The reduction in entropy generation due to thermal balancing associated with single extraction leads to a tremendous increase in COP.

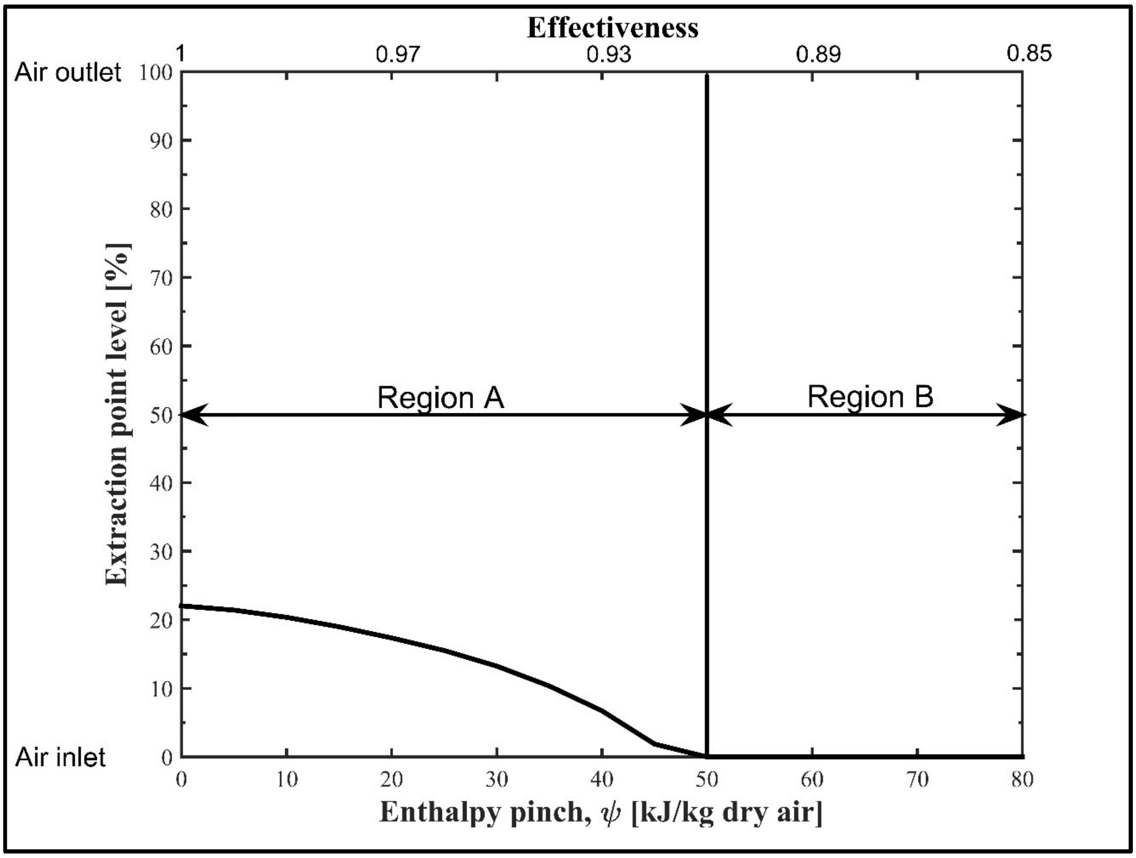


Figure 5.11 Effect of enthalpy pinch on extraction point position.

In region 'B' the COP of the proposed system with zero and single extractions coincides i.e. single extraction has no effect after this point. For this region, the enthalpy pinch starts from 50 kJ/kg dry air, which is equivalent to effectiveness value from 91%. This is because, in a single extraction cycle, air is extracted from a point placed between the inlet and outlet of regenerator and closer to air inlet; as enthalpy pinch increases, extraction point level gets closer to the inlet point up to the coincide point (Ψ= 50 kJ/kg dry air); at this point air extraction point is merged with air inlet i.e. single extraction becomes zero extraction cycle as shown in Fig. 5.11.

The results of Fig. 5.10 are summarized in Table 5.4 to indicate the best configuration for each effectiveness region.

Table 5.4 Recommended configuration for each effectiveness region.

Region	Enthalpy Pinch ($kJ\ kg_{da}^{-1}$)	Effectiveness (%)	Recommended configuration
A	0 - 50	100 - 91	Single Extraction
B	> 50	< 91	Zero Extraction

5.6.2 Effect of desiccant temperature at the condenser and regenerator inlets

Figure 5.12 illustrates the effect of the regeneration temperature on the proposed system with zero and single extractions at zero enthalpy pinch (that is, 100% component effectiveness). It is clear that as the regeneration temperature increases the COP decreases for both cases. Since the desiccant temperature at condenser inlet is fixed for two cases at 20 °C and 30 °C, respectively. It is found that the increase in regeneration temperature leads to increase in the total heat input. Thus, the COP drops accordingly. In addition, it is to be noted that at 20 °C the COP is higher than at 30 °C. This is because of the fact that the ability of cold desiccant to condensate more air in the condenser.

The reduction in entropy generation due to the thermal balance associated with single extraction leads to tremendous increase in COP, which is almost tripled compared to zero extraction at 80 °C regeneration temperature for temperatures of 20 °C, and 30 °C at condenser inlet.

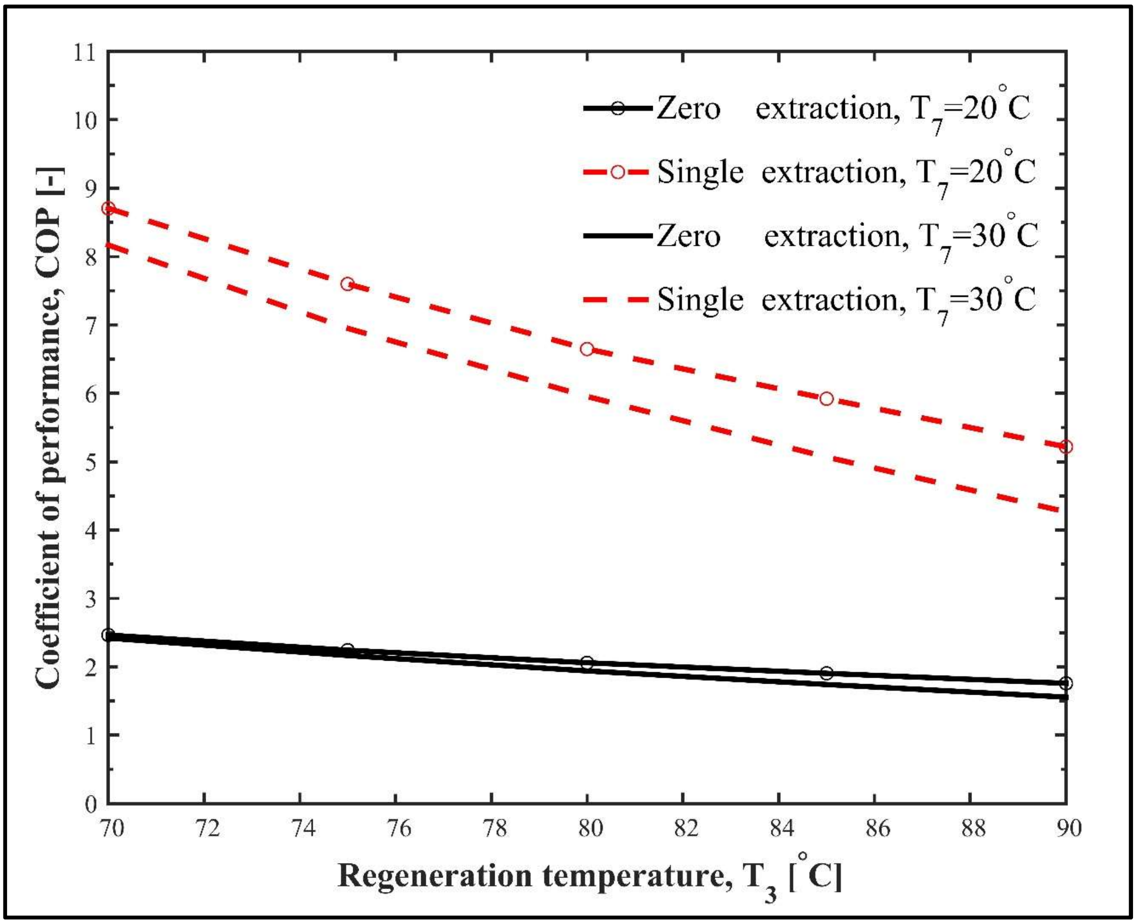


Figure 5.12 Effect of the regeneration temperature on performance: $\Psi_n = \Psi_r = 0$ kJ/kg dry air.

Figure 5.13 illustrates the behavior of this system under the same operating conditions but at 20 kJ/kg dry air enthalpy pinch (that is, about 97% component effectiveness). The COP for the 20 °C condensation temperature case increases (with and without extraction) as regeneration temperature increases. This increase may be associated with the increase in the regeneration temperature which will result in a better evaporation process in the regenerator and consequently more vapor is condensed in the condenser. The COP for zero extraction almost doubled compared to zero extraction at 80 °C regeneration temperature which is mainly due to the thermal balancing.

For the case where the condensation temperature is 30 °C, the COP for zero extraction decreases as the regeneration temperature increases due to the increase in the amount of total heat input rate. Thermal balancing through single extractions for this case increases the COP until regeneration temperature approaches 85 °C and then starts to drop. This behavior is associated with the increase in the regeneration temperature which means better evaporation until a certain point at which the effect of increase in total heat input rate surpasses the effect of moisture removal rate generated through increasing the regeneration temperature and thus the COP drops. By using extraction, COP increases by more than double for the regeneration temperature of 85 °C due to the reduction in entropy generation associated with the extraction process.

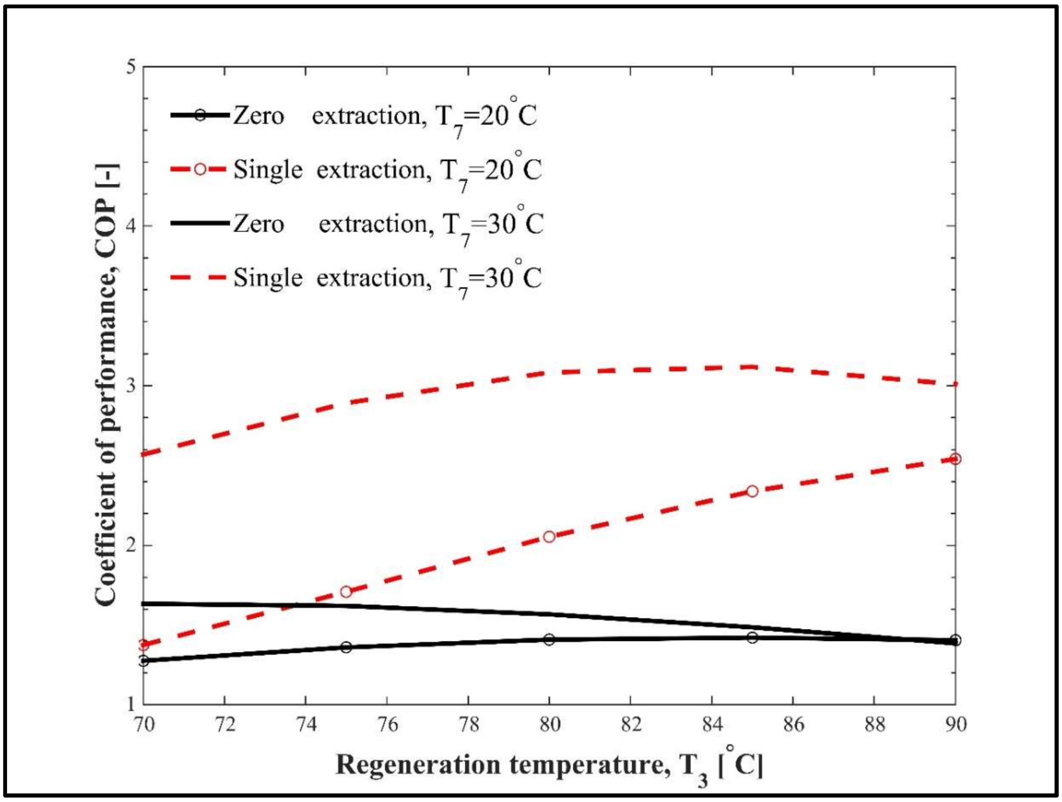


Figure 5.13 Effect of the regeneration temperature on performance: Ψn = Ψr = 20 kJ/kg dry air.

The COP for 30 °C condensation temperature case is higher when compared to the 20 °C case for zero and single extraction which is mainly because of the less amount of heat input rate required for 30 °C case.

At regeneration temperature of 70 °C, the COP for single extraction for the case where the 30 °C condensation temperature is double of the 20 °C condensation temperature case. Then this difference in COP decreases as regeneration temperature increases until it reaches a difference of almost 0.4 at regeneration temperature 90 °C. This decrease in the difference may be associated with the decrease in the total heat input required compared to the effect of coolant desiccant's ability to condensate more moisture in the condenser. After reaching a regeneration temperature of 89 °C, it is noticed that the COP for the 20 °C condensation temperature case with zero extraction exceeds the 30 °C case. At this point, the effect of the coolant desiccant to condensate the air exceeds that of the total heat input rate required.

Figures 5.14 and 5.15 show the impact of running the system under the same operating conditions at an enthalpy pinch of 50 and 80 kJ/kg dry air, respectively (that is, 91% and 85% component effectiveness). From both figures, it is clear that there is no effect of extraction on the COP with increasing the regeneration temperature as the thermal balancing effect appears clearly for high effectiveness components. Figure 5.14 illustrates that increase in the regeneration temperature increases the COP for zero and single extraction. This increase in COP is due to the amount of evaporation in the regenerator due to the increase of the regeneration temperature which results in a higher amount of moisture removal rate and thus COP is higher.

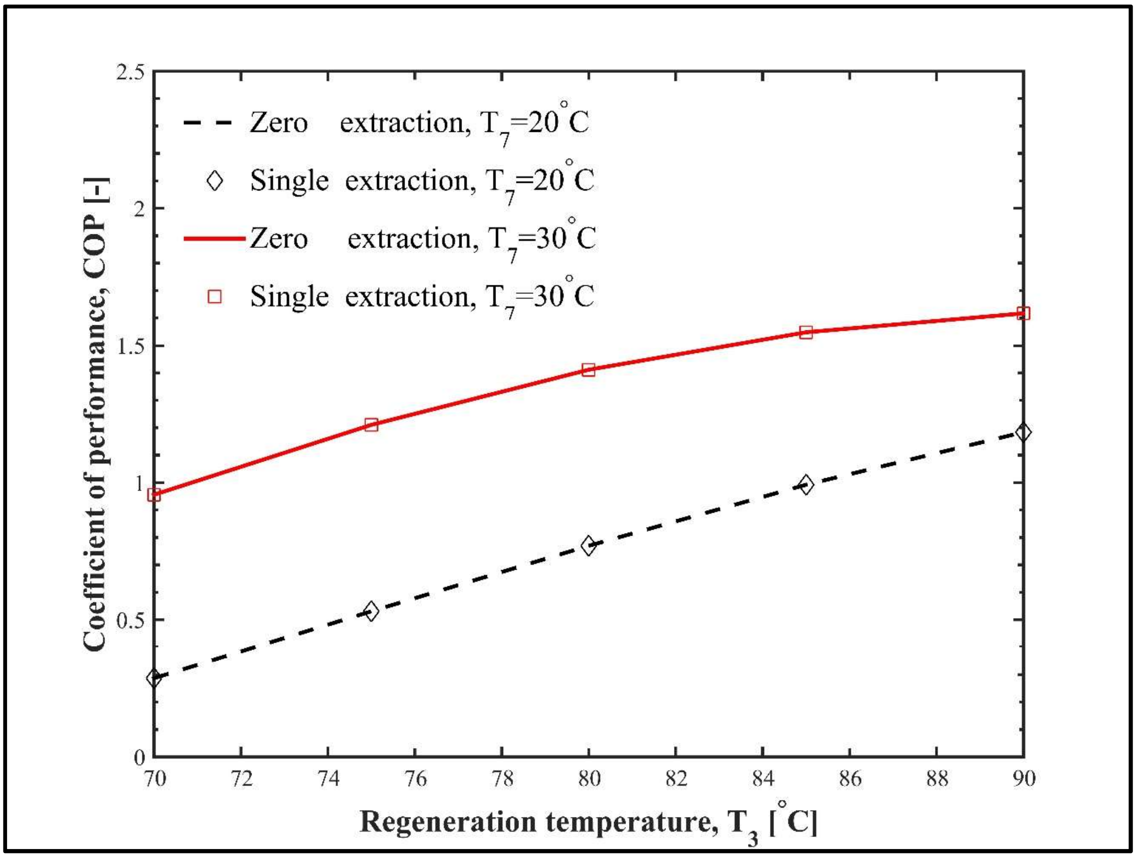


Figure 5.14 Effect of the regeneration temperature on performance: Ψn = Ψr = 50 kJ/kg dry air.

Figure 5.15 shows the increase in regeneration temperature for the case with T_7= 20 °C and 30 °C, in T_7 = 20 °C case, the COP is very low until the regeneration temperature reaches 80 °C. After this point, the COP starts to increase sharply from 0.1 to slightly more than 1. This COP is low due to the low effectiveness of the components. After 80 °C, the increase in regeneration temperature starts to impact the performance as it results in an adequate amount of evaporation which means more condensation in the condenser and that results in an increase in COP. For the case of 30 °C condensation temperature, the COP increases as the regeneration temperature increases. This increase of moisture removal rate is associated with the increase in regeneration temperature.

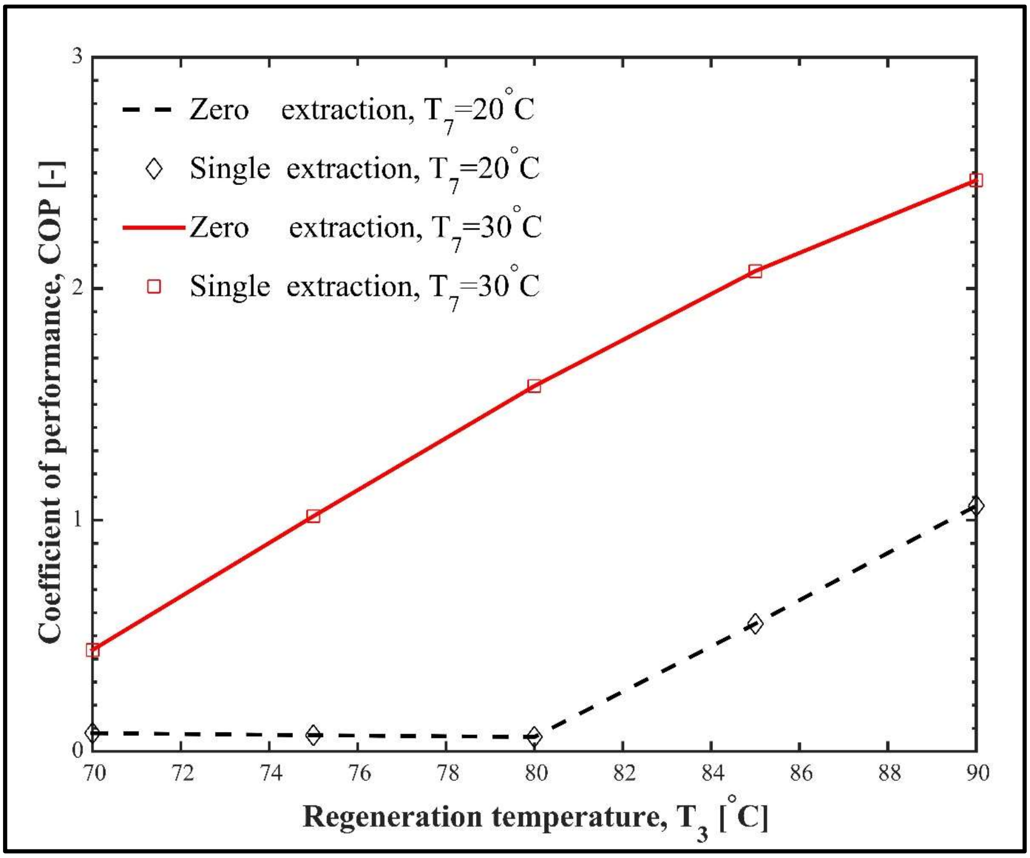


Figure 5.15 Effect of the regeneration temperature on performance: Ψn = Ψr = 80 kJ/kg dry air.

5.7 Conclusions

In this chapter, the performance of the proposed desiccant cooling system is evaluated with zero and single extractions. The effect of operating conditions on the systems performance have been examined. The COP is used as the benchmark to study the effect of single extraction. Based on the aforementioned analysis and discussion presented in the chapter, the following findings can be drawn:

1. Enthalpy pinch has been introduced for analysis of heat and mass exchangers as a replacement to the temperature pinch that usually defined for heat exchangers.

Enthalpy pinch (kJ/kg of dry air) combines temperature and concentration, and is inversely proportional to energy effectiveness.

2. Varying desiccant-to-air mass flow rate ration using single extraction and injection brings HCR at extraction point to unity. Hence, local enthalpy pinch through the condenser and regenerator is tremendously decreased i.e. leads to better system performance.
3. The coefficient of performance for both zero and extraction systems decreases as enthalpy pinch increases (or effectiveness decreases).
4. Performance of single extraction is much better than zero extraction cycle up to certain enthalpy pinch and after this point, the effect of single extraction coincides with the zero extraction performance i.e. single extraction has no effect after this point.
5. The coefficient of performance for both zero and extraction systems increases as regeneration temperature increases up to a certain point at which it reaches its optimum value and starts to decrease after it.
6. At $\Psi = 20$ kJ/kg dry air which is equivalent to $\varepsilon = 0.97$ for both regenerator and condenser; single extraction system performance is 85.7% better than zero extraction.
7. Due to addition of condenser in both zero and single extraction cycles, amount of water evaporated in the regenerator can be extracted using the condenser as a fresh water. Both zero and single extraction cycles produce 94.2 kg of fresh water per hour as a by-product.

CHAPTER 6

Conclusions and Recommendations

In this thesis, the performance of the conventional and proposed (zero and single extractions) liquid desiccant cooling system is evaluated. The effect of operating conditions on the systems performance have been examined. The COP is used as the benchmark to compare the performance of all systems. To understand and optimize the performance of the proposed system, a comprehensive study has been carried out. Based on the aforementioned analysis and discussion presented in this thesis, the following findings can be drawn:

1. Enthalpy pinch has been introduced for analysis of heat and mass exchangers as a replacement to the temperature pinch that usually defined for heat exchangers. Enthalpy pinch (kJ/kg of dry air) combines temperature and concentration of the desiccant solution, and is inversely proportional to energy effectiveness.
2. Main factors that influence the performance of all systems are desiccant-to-air mass flow rate ratio, HCR, desiccant solution and process air inlet conditions, and enthalpy pinch (energy effectiveness) of the system components.
3. The performance of both systems increases as the energy effectiveness increases. Moreover, the increase in regenerator effectiveness has the highest effect on the system performance while heat exchanger effectiveness has the lowest effect.
4. The system performance with zero extraction is 11.25% better than the conventional system. This comparison is executed at optimum desiccant-to-air mass flow rate ratio

using basic operating conditions given in Table 3.1. This improvement in modified system vanishes when the condenser effectiveness is less than 0.57.

5. A modified HCR is defined for the regenerator as the ratio of the maximum possible change in total enthalpy rate of the air to the maximum possible change in total enthalpy rate of the desiccant. Operationally, HCR can be varied by only changing the desiccant-to-air mass flow rate ratio because the energy effectiveness and inlet temperatures are hard to be controlled. Bringing HCR to unity reduces entropy generation and improve system performance noticeably.
6. In the balanced heat and mass exchangers, the stream-to-stream temperature and concentration difference is constant at the terminal points only. Therefore, varying desiccant-to-air mass flow rate ratio using single mass extraction and injection brings HCR at extraction point to unity. Hence, local irreversibilities through the regenerator and condenser can be tremendously decreased i.e. leads to better system performance.
7. Performance of single extraction is much better than zero extraction cycle up to certain enthalpy pinch and after this point, the effect of single extraction coincides with the zero extraction performance i.e. single extraction has no effect after this point.
8. The COP for both zero and extraction systems increases as regeneration temperature increases up to a certain point at which it reaches its optimum value and starts to decrease after it.
9. At Ψ= 20 kJ/kg dry air which is equivalent to ε = 0.97 for both regenerator and condenser; single extraction system performance is 85.7% better than zero extraction.
10. Due to addition of condenser in both zero and single extraction cycles, amount of water evaporated in the regenerator can be extracted using the condenser as a fresh water.

Zero and single extraction cycles produce 91.4 and 94.2 kg of fresh water per hour as a by-product, respectively.

Recommendations

Using the condenser for heat and mass recovery in the modified system is promising with plenty of room to improve its performance. The following recommendations are offered to improve the proposed system further as a promising technology in the field of liquid desiccant air cooling systems:

1. Applying mass extraction and injection on the desiccant stream instead of the carrier air.
2. The proposed system is operated at atmospheric pressure only. From psychrometric chart, humidity ratio is higher at low pressure, hence, running the system where the regenerator at sub-atmospheric pressure is recommended.
3. To get a better performance in the proposed system, air should leave the condenser at low humidity ratio. Therefore, increase operating pressure in the condenser is favorable. The pressure can be increased by using mechanical or thermo compressor.

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