

**SOLAR POWERED HYBRID ADSORPTION DESALINATION-
COOLING SYSTEMS**

BY

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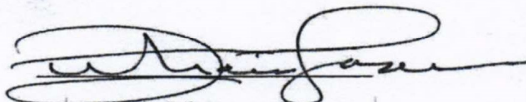
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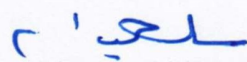
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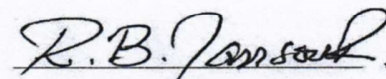
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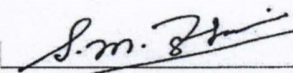

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Dedication

To my mother and my father, I learned and worked hard to be an influential person in the future for their continued encouragement. |

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LIST OF ABBREVIATIONS

GCC	:	Gulf Cooperation Council
MEDAD	:	The hybridization of multi-effect evaporation desalination system with adsorption cycle
SDWP	:	Specific daily water production
BWRO	:	Brackish water reverse osmosis
DWP	:	Daily desalinated water
MVC	:	Mechanical vapor compression
ADD	:	Adsorption desalination
MOF	:	Metal-organic frameworks
HCFCs	:	Hydrochlorofluorocarbons
RD	:	Regular density
RO	:	Reverse osmosis
ED	:	Electrodialysis
MD	:	Membrane desalination
HDH	:	Humidification Dehumidification
MED	:	Multi-effect desalination

MSF	:	Multi-stage flash desalination
TVC	:	Thermal vapor compression
AD	:	Adsorption heat pump
GOR	:	gain output ratio, (-)
COP	:	coefficient of performance, (-)
CC	:	Cooling capacity, kW
BET	:	Brunauer-Emmett-Teller, $\text{m}^2 \text{g}^{-1}$
C_p	:	Specific heat capacity, $\text{J kg}^{-1} \text{K}^{-1}$
h_{fg}	:	Latent heat of evaporation, J kg^{-1}
M	:	Mass, kg
$\dot{m}$	:	Mass flow rate, kg s^{-1}
t	:	Time, s
T	:	Temperature, °C
q	:	Refrigerant amount adsorbed, kg kg^{-1}
Q_{st}	:	Isosteric heat of adsorption/desorption, J kg^{-1}
U	:	Overall heat transfer coefficient, $\text{J m}^{-2} \text{K}^{-1}$
SCP	:	Specific cooling capacity, W kg^{-1}

A	:	Surface area, m ²
P	:	Pressure, Pa
$\dot{Q}_{\text{cooling}}$	:	Cooling output, kW
q^*	:	Equilibrium uptake, kg kg ⁻¹
TBT	:	Top brine temperature, °C
LBT	:	Lower brine temperature, °C
PF	:	Parallel feed
F	:	Feed seawater in each effect, kg s ⁻¹
n	:	Number of effects
D	:	Distillate water in each effect, kg s ⁻¹
B	:	Rejected brine in each effect, kg s ⁻¹
x	:	Water salinity, ppm
LMTD	:	Log mean temperature difference, °C
BPE	:	The boiling point elevation, °C
h_f	:	Enthalpy of saturated liquid water, J kg ⁻¹
h_g	:	Enthalpy of saturated water vapor, J kg ⁻¹

PR : Performance ratio, (-)

sM_{cw} : Specific flow rate of cooling water, (-)

sA : Specific heat transfer area, m² kg⁻¹

P : Pressure, kPa

Subscripts

adt : Adsorbent material

ads : Adsorber

des : Desorber

bed : Adsorbent bed

chill : Chilled water

cond : Condenser

evap : Evaporator

hot : Hot water

f : Heat transfer fluid

hex : Heat exchanger

v : Vapor

w : Refrigerant

sg	:	Silica gel
in	:	Inlet
out	:	Outlet
i	:	Effect number
f	:	Feed seawater
b	:	Rejected brine
v	:	Formed vapor (distillate)
s	:	Steam
c	:	Condenser
e	:	Evaporator
cw	:	Cooling water
dest	:	Distillate water

|

ABSTRACT

Full Name : [Hassan Ali Alkhalifah]

Thesis Title : [solar powered hybrid adsorption desalination-cooling systems]

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In 2015, the world's population was estimated to be 7.3 billion persons. United Nations expected that the world's population would increase by 2.4 billion persons in the next 35 years. This population growth will result in increasing freshwater and cooling demand. So, there is a need to find a cooling system and desalination system that can operate using renewable energy. The proposed work is intended first to study the effect of adsorbent materials (silica gel RD-2060, zeolite 13X, HKUST-1, MIL-100(Fe), and aluminum fumarate) and operating conditions on the two-bed adsorption chiller with mass recovery from the condenser to the evaporator. The results show that using HKUST-1 as an adsorbent is more effective compared to other adsorbents. It improves the COP and cooling capacity by 43% and 75%, respectively, compared to RD silica gel. The second part of the work analyzes the hybrid multiple-effect evaporation desalination system with the adsorption cycle. Rising attention is given to this kind of system since it uses low-grade energy (solar or waste heat energy) to lower the last effect temperature to be lower than the ambient temperature and improve the distillate water production. However, since the work in the literature is focused on conventional adsorbents, this proposal's performance evaluation expands to consider different adsorbents, including the state-of-the-art metal-organic-framework (MOF) that show superior adsorption performance. The results show that distillate water production could increase by more than two folds than the conventional

multiple-effect desalination system. Moreover, the last stage temperature can be as low as 16 °C. More effects can be inserted into the system in comparison with the conventional system.

|

ملخص الرسالة

الاسم الكامل: حسن علي احمد الخليفة

عنوان الرسالة: أنظمة التبريد بالامتزاز و تحلية المياه بالامتزاز الهجينة التي تعمل بالطاقة الشمسية

التخصص: الهندسة الميكانيكية

تاريخ الدرجة العلمية: ديسمبر 2020

في عام 2015 ، قدر عدد سكان العالم بـ 7.3 مليار شخص. توقعت الأمم المتحدة أن يزداد عدد سكان العالم بمقدار 2.4 مليار شخص في السنوات الخمس والثلاثين القادمة. سيؤدي هذا النمو السكاني إلى زيادة الطلب على المياه العذبة والتبريد. لذلك ، هناك حاجة لإيجاد نظام تبريد ونظام تحلية يمكن أن يعمل باستخدام الطاقة المتجددة. يهدف العمل المقترح أولاً إلى دراسة تأثير المواد المازة وظروف التشغيل على مبرد الامتزاز ثنائي السرير مع استعادة الكتلة من مكثف للمبخر. أظهرت النتائج أن استخدام HKUST-1 كمادة مازة أكثر فعالية مقارنة بالامتزازات الأخرى. يحسن معامل الأداء المعياري وقدرة التبريد بنسبة 43% و 75% على التوالي ، مقارنة بسيليكاجيل. الجزء الثاني من العمل يحلل نظام تحلية المياه بالتبخير متعدد التأثيرات المدمج مع نظام الامتزاز. يتم إعطاء اهتمام متزايد لهذا النوع من النظام لأنه يستخدم الطاقة منخفضة الدرجة الحرارية (الطاقة الشمسية أو الطاقة الحرارية المهدرة) لخفض درجة حرارة التأثير الأخير لتكون أقل من درجة الحرارة المحيطة وتحسين إنتاج الماء المقطر. ومع ذلك ، نظرًا لأن العمل في الأدبيات يركز على المميزات التقليدية. يتوسع تقييم الأداء في هذا الاقتراح للنظر في المميزات المختلفة ، بما في ذلك الإطار المعدني العضوي الحديث (MOF) الذي أظهر أداء امتصاص فائقًا. أظهرت النتائج أن إنتاج الماء المقطر يمكن أن يزيد بأكثر من الضعف عن نظام التحلية التقليدي متعدد التأثيرات. علاوة على ذلك ، يمكن أن تكون درجة حرارة المرحلة الأخيرة منخفضة حتى 16 درجة مئوية. يمكن إدخال المزيد من التأثيرات في النظام مقارنة بالنظام التقليدي.

CHAPTER 1

INTRODUCTION

Clean water is an absolute requirement for the lives of all. Unfortunately, for at least one month every year, more than 60 percent of the world's population suffers from water scarcity[1]. A small fraction of all the water on the earth constitutes drinkable water. Although water covers almost 71 percent of the total surface of the planet, just 3 percent of it is drinking water (freshwater). In addition, only 0.5% of freshwater is readily available. The remainder is locked in glaciers, polar ice caps, soil, and the atmosphere [2]. In 2015, the world's population was estimated to be 7.3 billion persons. United Nations expected that the world's population would increase by 2.4 billion persons in the next 35 years [3]. This population increase would result in a rising demand for freshwater. Many desert or semi-desert countries suffer from water shortages caused by high population growth, a decline in underground water, and a rise in the rate of economic development. When the water supply falls below 1000 cubic meters per person per year, water shortage occurs [4].

1.1 Desalination

Processes of desalination of seawater will solve the issue of water scarcity. To obtain freshwater, desalination processes extract dissolved minerals from brackish or seawater

(saline water). These processes need an abundant amount of energy, which is the main reason why saline water desalination costs are high. Around 10000 tons of fossil fuel per year is the energy necessary to generate 1000 cubic meters of water per day [5]. The energy is required to produce freshwater, as shown in Figure 1.1. Renewable energy supplies are used to substitute fossil fuels to reduce the emissions of greenhouse gases and carbon dioxide, the key causes of global warming and climate change.

Thermal and membrane desalination technologies are the primary desalination processes used in the world. Thermal desalination technology is the most popular desalination method used in the countries of the Gulf Cooperation Council (GCC). Strong feed seawater salinity in the Gulf and the repeated occurrences of harmful algae blooms are the key reasons behind it.

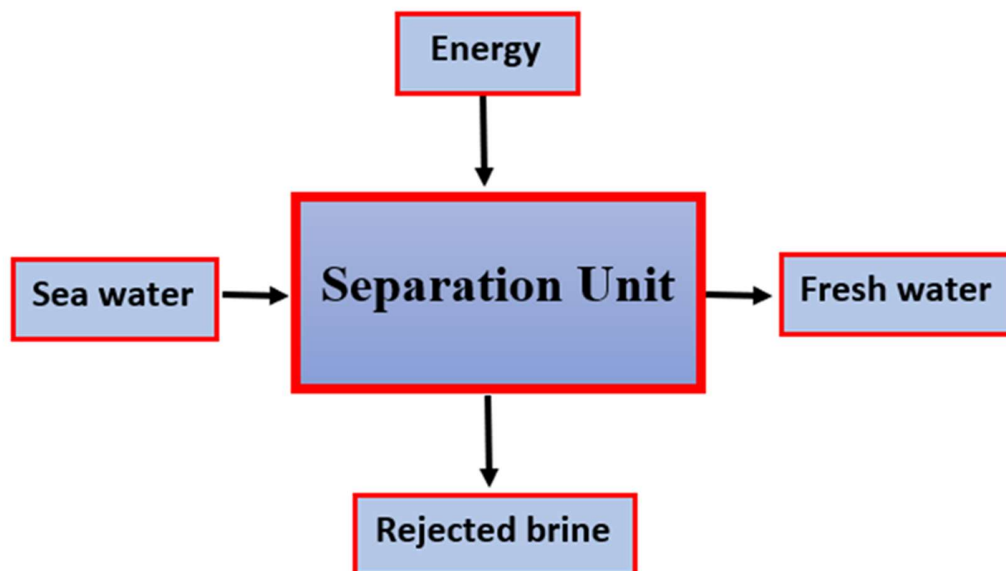


Figure 1-1: Desalination process.

Conventional desalination technologies are split into three main categories:

- Thermal desalination system which utilizes thermal and electric energy.
- Membrane desalination system which uses the electric energy to increase feed pressure.
- Chemical desalination system which utilizes the chemical potential.

The thermal desalination techniques include two major categories:

- Evaporation followed by condensation of the formed water vapor.
- Freezing followed by melting of the formed water ice crystals.

Membrane-based desalination techniques include two major categories:

- Reverse osmosis (RO).
- Electrodialysis (ED).

Figure 1.2 presents the main categories of desalination technologies.

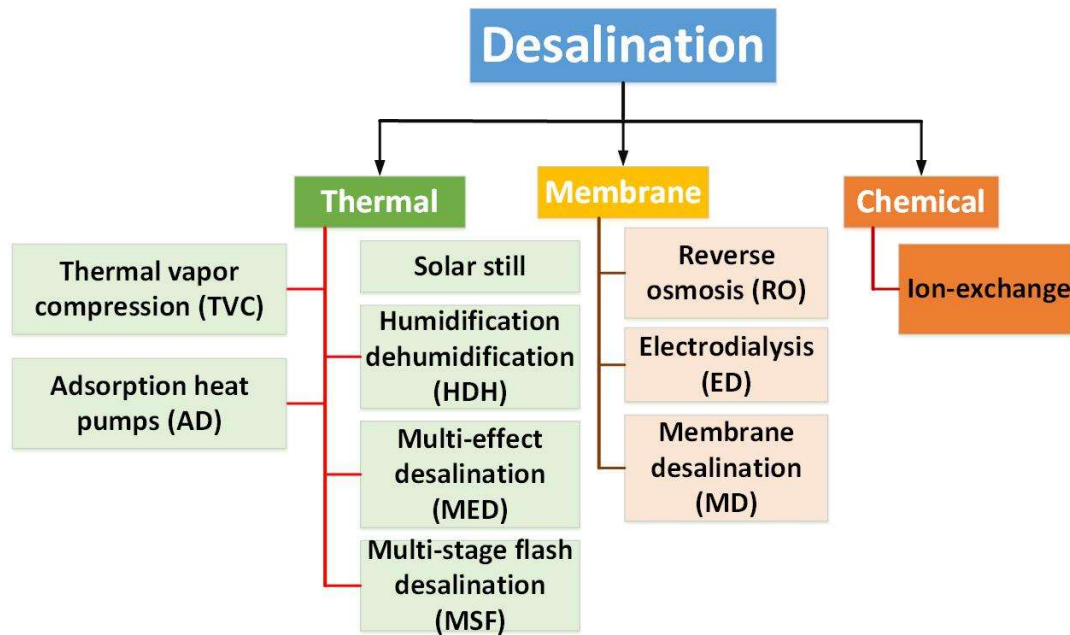


Figure 1-2: Desalination process categories.

1.1.1 Solar Still Desalination System

A solar still (Figure 1.3) desalination process utilizes solar energy to produce fresh water from the brackish water. It is made of an insulated basin (e.g., wood), opaque material base (e.g., black stainless steel), and tilted transparent cover (e.g., glass). Solar radiation, coming from the sun, passes through the glass cover to be absorbed by the basin. Then, it heats the seawater/brackish water and causes evaporation. The formed water vapor condenses on the glazing's inner cold side and drops down by gravity into a trough where it is gathered. The opaque material base is painted black to improve its absorptivity. The glass cover traps the heat inside the basin since longwave radiation cannot pass through it. Solar stills can be divided into two main types, passive and active. Passive stills utilize only solar radiation, while active stills use solar radiation and an external heat source (e.g., an electrical heater). The main advantages of solar still desalination systems are the low cost and maintenance,

they can be used for a small-scale application, and they are environmentally friendly. However, the main drawbacks are low productivity per unit area, and the low gain output ratio (GOR). In the solar still, water heating, evaporation, and condensation processes occur in a single volume, causing a significant thermal inefficiency.

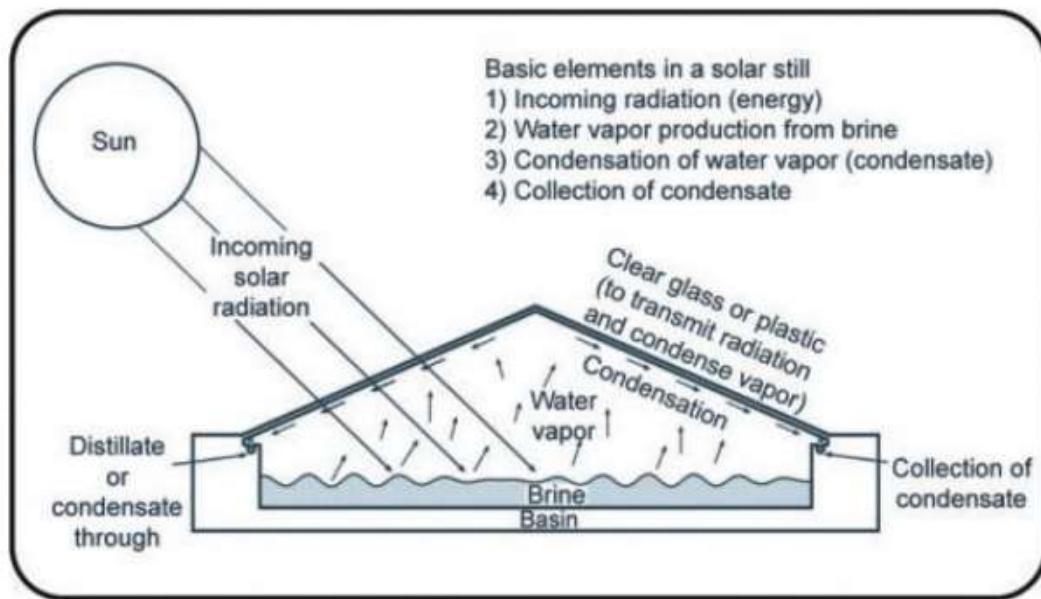


Figure 1-3: Solar still desalination system [6].

1.1.2 Humidification Dehumidification Desalination System

Humidification-dehumidification (HDH) desalination systems imitate nature's rain cycle. It is made of a humidifier, a dehumidifier, and a heater. They are classified into water and air heated cycle, as shown in Figure 1.4. In a water-heated cycle, water leaving the dehumidifier is heated before entering the humidifier using a heater. While in an air-heated cycle, the air is heated before entering the humidifier using an air heater. Dry air is humidified through direct contact with hot water in the humidifier. In the dehumidifier,

humid air is dehumidified through direct contact with a cold surface cooled by flowing water. As a result, water vapor condenses and produces distillate water. The main advantages of the HDH system are lower operating temperatures and the ability to operate over a wide range of feed water quality. They can also be used for small to medium-scale applications and in remote locations. The main drawback of this system is high energy consumptions compared to the other conventional desalination systems.

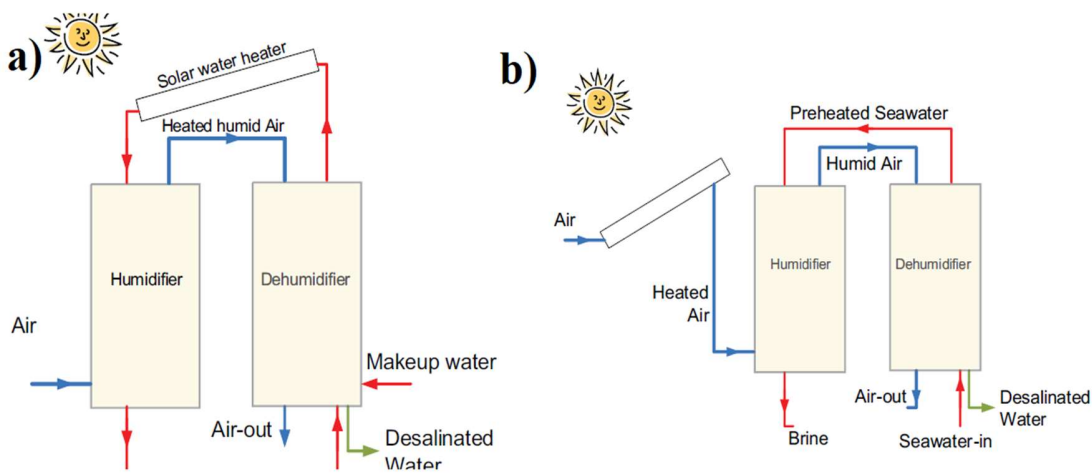
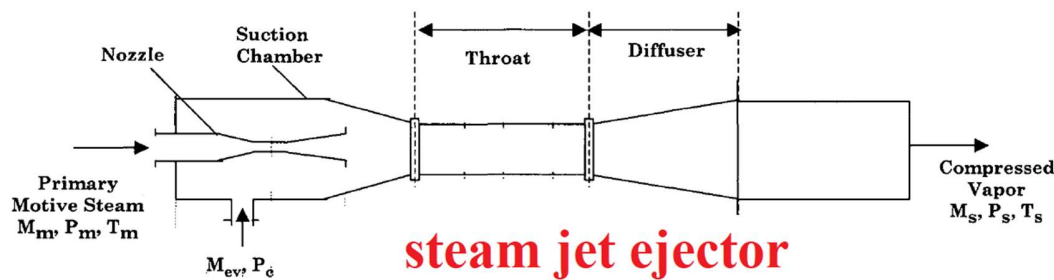


Figure 1-4: HDH a) water-heated cycle b) air heated cycle [8].

1.1.3 Single Effect Thermal Vapor Compression Desalination System

In the thermal vapor compression desalination (TVC) process, vapor's compression provides the required heat to evaporate the feed water instead of the external heat source. Figure 1.5 shows the main components of the single effect thermal vapor compression (TVC) desalination system. In the TVC process, part of the water vapor formed is extracted (called entrained vapor) to the steam ejector. The ejector is used to increase the entrained vapor pressure by utilizing external motive steam. The compressed water vapor condenses in the tubes are placed inside the evaporator and provide the heat of

condensation to heat the feed water sprayed outside the heated tube bundle and evaporate part of it. This system's main advantages are simple and reliable operation, lower power consumption, and the absence of moving parts.



**Single
Effect
with
TVC**

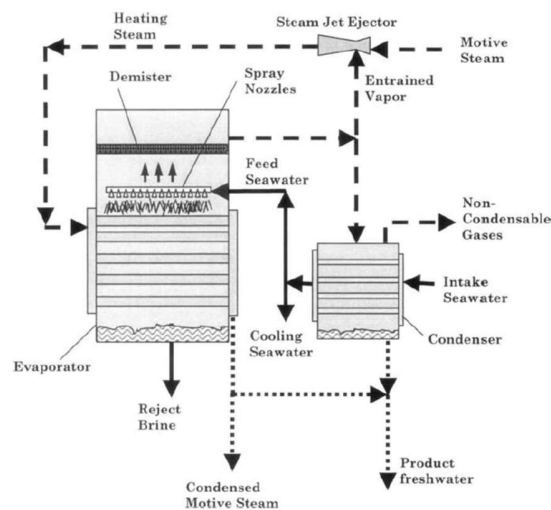


Figure 1-5: Single effect thermal vapor compression desalination system [7].

1.1.4 Multi-stage flash distillation (MSF)

The Multi-stage flash distillation (MSF) is a process that evaporates a part of the brackish water by flashing effect (quick evaporation) through multiple chambers (Figure 1.6). The feedwater is pumped to a higher pressure and passes through a series of chambers where its temperature increases by absorbing the heat released from condensing the formed water vapor. Then it passes through the brine heater, where its temperature rises to the

top brine temperature. Then, it flows into the first flash stage where its pressure decreases, resulting in the water to boil quickly, causing quick flashing, which generates water vapor. The water vapor condenses on the by releasing its latent heat to the incoming feed water. This process continues in each successive stage because the next stage's pressure is lower than the previous stage.

The MSF processes have two main configurations, which are once through and brine circulation. In a once-through process, the feed water is rejected back to the sea after passing through the heater and flash stages once. While in a recycled process, the saline water utilized for cooling purposes is recycled. The main drawback of the MSF process is the high energy required to operate the system.

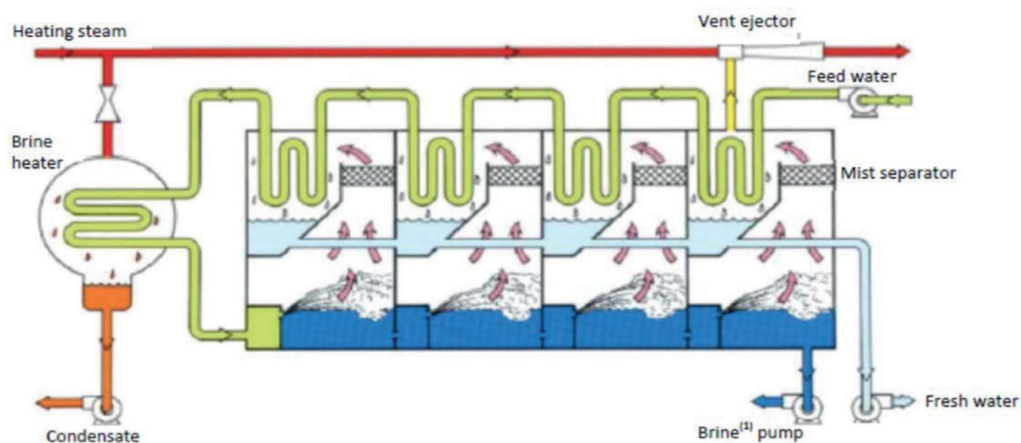


Figure 1-6: Once-through MSF desalination system [9].

1.1.5 Multi-effect distillation (MED)

The multi-effect evaporation (MED) is formed from a series of a single effect. Formed vapor in the first effect is utilized as a heating source for the second effect. Generally, vapor formed in one vessel is used as a heating medium for the next effect. As pressure

decrease along with the effects, the water boils at a lower temperature. Water vapor condenses in the following effects. Vapor reuse in the MED system reduces the rejected brine and the temperature, reducing the amount of energy rejected to the sounding. The MED process can be categorized according to the direction of the feed and brine flow rates as forward, backward, or parallel feed (Figure 1.7). The MED system's advantages are high heat efficiency, suitable to combined with renewable energy systems (e.g., adsorption or absorption), and lower energy consumption and higher performance compared to the MSF system.

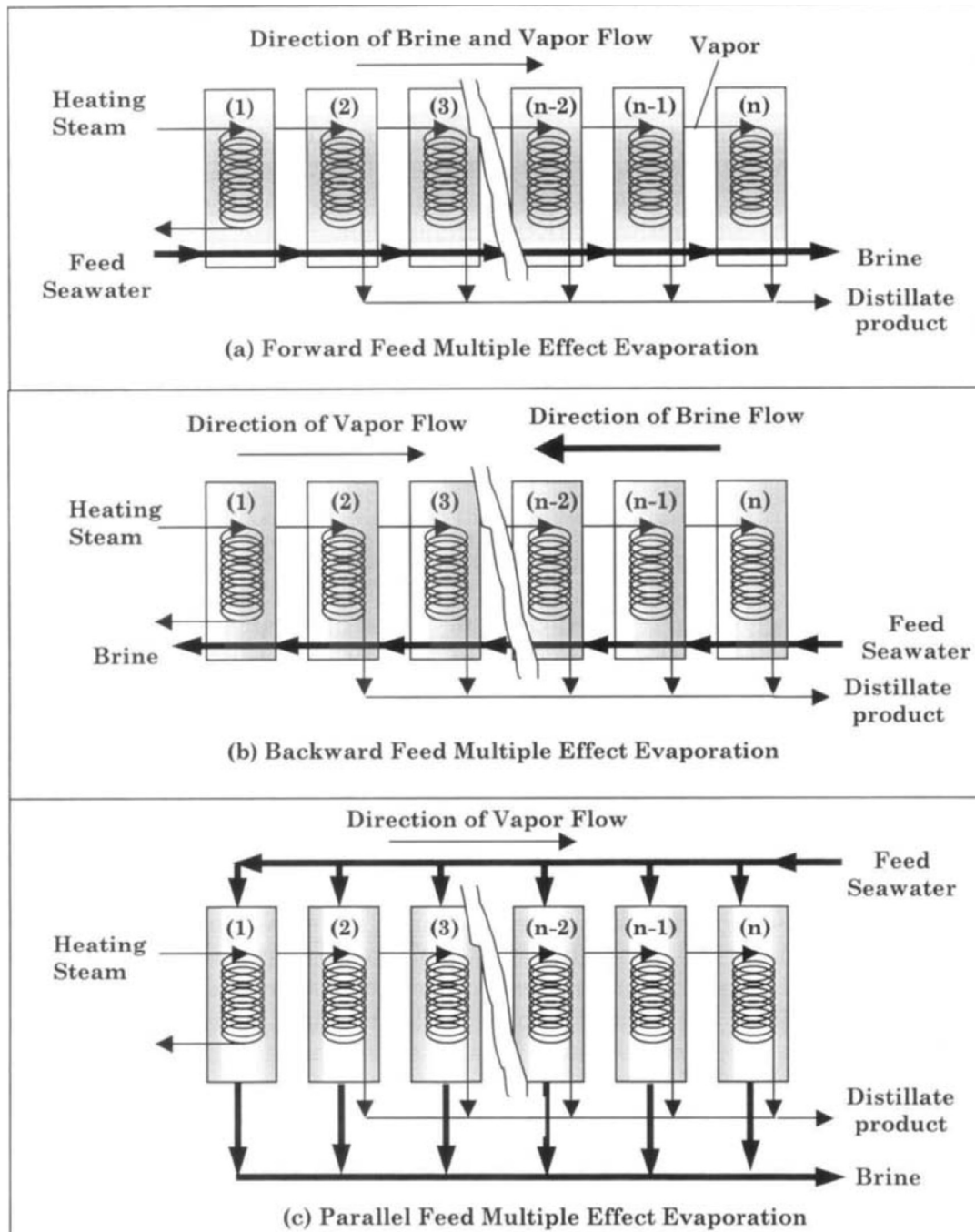


Figure 1-7: Configurations of MED [7].

1.2 Conventional Mechanical Vapor Compression Cooling System

The conventional vapor compression cooling system's main four components are the evaporator, condenser, expansion valve, and mechanical vapor compression machine (compressor). A system's schematic diagram representation and the corresponding thermodynamic cycle are shown in figure 1.8. The heat transfer fluid flowing inside this system is called refrigerant. The refrigerant enters the compressor as a saturated vapor with low pressure. It is compressed to a higher pressure (condenser pressure) and leaves the compressor at a superheated state. This process is isentropic compression. The refrigerant then enters the condenser, where its temperature drops by rejecting its latent heat of condensation, and it leaves as a saturated liquid. This process is a constant-pressure heat rejection. Then, the fluid passes through a throttling valve where its pressure reduces to the evaporator pressure, and it leaves as a saturated liquid-vapor mixture. Then, the fluid enters the evaporator, where its temperature increases by absorbing latent heat of vaporization, and it leaves as a saturated vapor. This process is constant-pressure heat absorption. The refrigeration effect is produced during the vaporization process by removing the heat from the chilled fluid.

The energy exchanges are as follows:

- During the compression process, work is supplied to the mechanical compressor.
- During the condensation process, the refrigerant's latent heat of condensation is rejected to cooling heat transfer fluid (e.g., water).

- During the evaporation process, the refrigerant's latent heat of evaporation is absorbed from chilling heat transfer fluid (e.g., water). This process produces a cooling effect.
- There is no exchange of heat during the throttling process. It occurs at a constant enthalpy.

The cooling system's performance is evaluated using the coefficient of performance (COP) and the cooling capacity.

The coefficient of performance (COP) can be defined as:

$$\text{COP} = \frac{\text{Refrigeration effect}}{\text{Work input}} \quad 1.1$$

The COP of mechanical vapor compression chiller can reach around 9 [13]. The main drawbacks of the conventional vapor compression chillers are the high energy consumption required for mechanical compressors. Moreover, some systems still use HCFC refrigerants, which contribute to the ozone layer depletion and global warming problems.

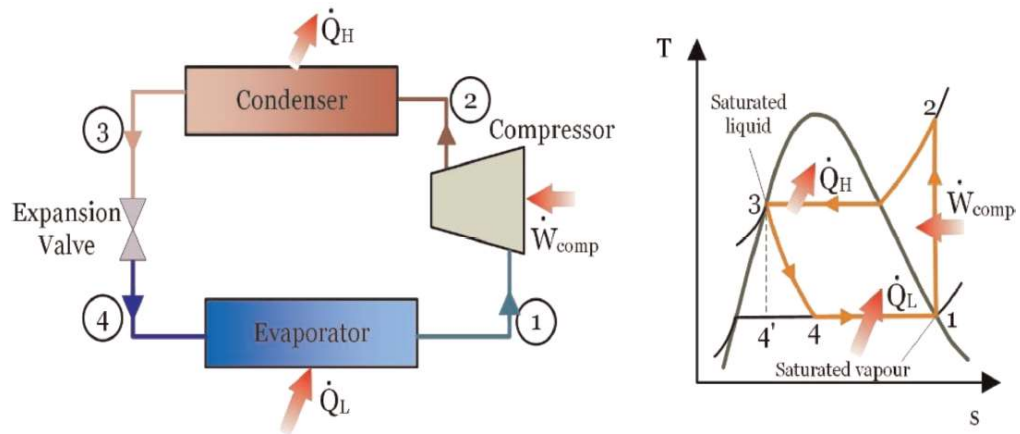


Figure 1-8: The schematic diagram and thermodynamic cycle for conventional vapor compression chiller [14].

1.3 Absorption Cooling System

Absorption chillers (Figure 1.9) use heat energy instead of mechanical energy (compressor). The absorption system's main principle is that a secondary fluid, known as sorbent, is utilized to absorb the primary fluid (e.g., gaseous refrigerant). Usually, absorption chillers are either ammonia-water or lithium bromide cycles. The absorption system's main components are evaporator, expansion valve, condenser, generator, absorber, heat exchanger, and pumps. The formed refrigerant vapor leaves the evaporator and enters the absorber, where the ammonia-water solution absorbs it. The new solution is pumped into the generator, where the solution is separated into liquid sorbent and refrigerant vapor through the heating and rectification process. The absorption chillers required lower energy and can be driven by waste or solar energy. However, these systems' main drawbacks are their large size, heavyweight. Also, they require a large cooling tower.

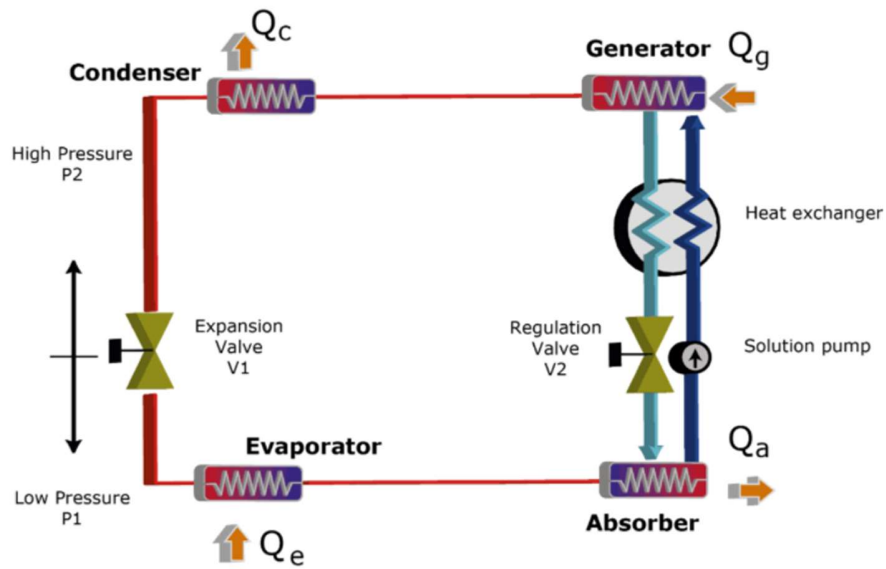


Figure 1-9: The schematic diagram absorption chiller cycle [15].

1.4 Adsorption

1.4.1 Background

The adsorption process (Figure 1.10) is based on a surface phenomenon in which liquid or gas molecules accumulate on the solid interference surface resulting in a higher concentration of the molecules on the surface. While absorption is a bulk phenomenon in which the particles of gas or liquid are spread throughout the bulk (Figure 1.11). The solid that takes up (adsorb) the gas or liquid molecules is called adsorbent, while the liquid or gas molecules accumulated on the adsorbent surface are called adsorbate. The adsorption process is an exothermic process (releasing heat). The desorption process is opposite to the adsorption process. It is the process of removing the adsorbate from the adsorbent. It is an endothermic process (absorb heat).

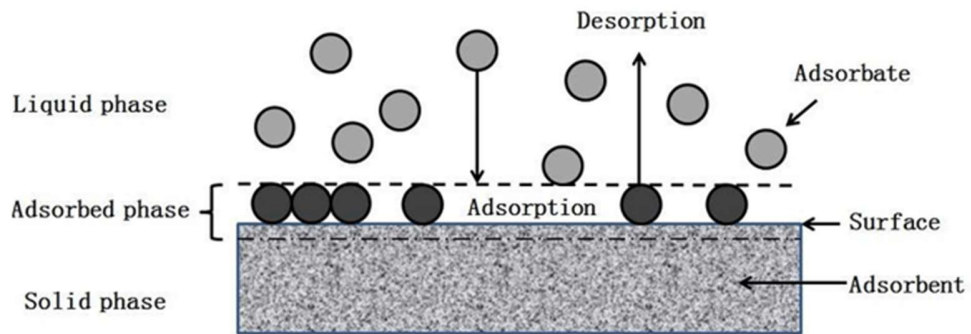


Figure 1-10: Definition of the basic terms of adsorption [11].

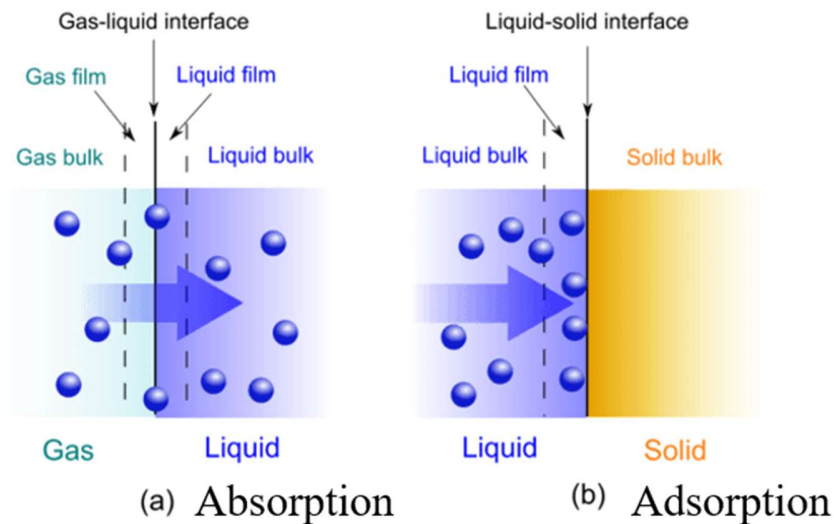


Figure 1-11: Absorption vs. Adsorption [12].

Adsorption can be categorized based on the type of forces existing between adsorbent and adsorbate molecules as physisorption (physical adsorption) or chemisorption (chemical adsorption). Chemisorption is the process in which the adsorbate molecules adhere to the surface of the adsorbent by forming a chemical bond. It occurs at high temperatures, and it is an irreversible process because the force of attraction is very strong. Physisorption is the process in which the adsorbate molecules adhere to the surface of the adsorbent by weak Vanderwaal forces of attraction. It is a reversible process (removal of the adsorbate from

the adsorbent material is possible) by heating or decreasing the pressure. Moreover, it occurs at low temperatures. The physisorption process is used in many applications such as air-conditioning and sea water desalination. The adsorbent's materials play an important role in the adsorption process.

The adsorption isotherm (e.g., Freundlich isotherm, Langmuir isotherm) graph is used to study the adsorption process. It is a graph of the amounts of adsorbate adsorbed on the adsorbent's surface as a function of the saturation pressure at a constant temperature. The adsorption equilibrium is the maximum amount of adsorbate adsorbed at a given adsorbate's pressure and adsorbent's temperature. The adsorption isotherm is the adsorption equilibrium at a constant adsorbent's temperature. Adsorption kinetics (e.g., Pseudo first-order rate equation/linear driving force equation) is a measure of the adsorption uptake as a function of time at a constant pressure. The main drawback of the physisorption process is the low cyclic flow rate as results of low adsorption kinetic.

1.4.2 Adsorption cooling system

In the adsorption chiller, the mechanical compressor is replaced by the adsorbent bed driven by heat energy. The adsorption chiller uses an environmentally friendly refrigerant, and it is driven by waste heat or solar energy sources. Therefore, it reduces greenhouse emissions. The adsorption system requires a lower heat source temperature, and it has a lower weight and size than the absorption system. The adsorption chiller's main components are adsorber/desorber beds, evaporator, and condenser, as shown in Figure 1.12. Adsorption refrigeration, unlike absorption and vapor compression systems, is an

inherently cyclic process where multiple adsorbent beds are necessary to provide approximately continuous cooling capacity. The main drawback of the adsorption system is the low COP value.

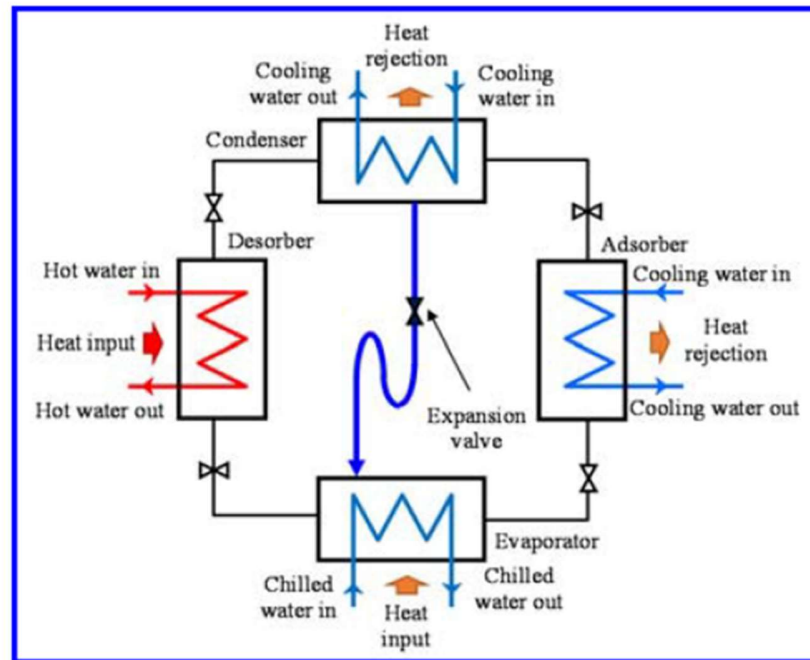


Figure 1-12: 2-Beds adsorption chiller [17].

The working principle of the adsorption cycle process described as flows, as shown in Figure 1.13:

1. Precooling process (C-D)

During this period, there is no adsorption/desorption process. Cooling water flows inside the adsorbent bed to decrease its temperature, reducing its pressure from the condenser pressure to the evaporator pressure. At the end of this process, the valve between the bed and the evaporator is opened.

2. Adsorption process (D-A)

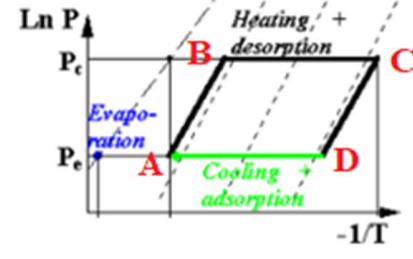
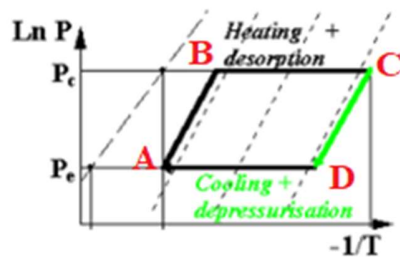
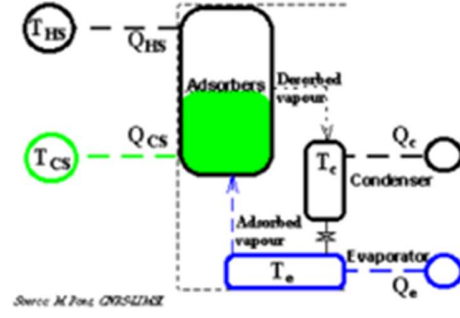
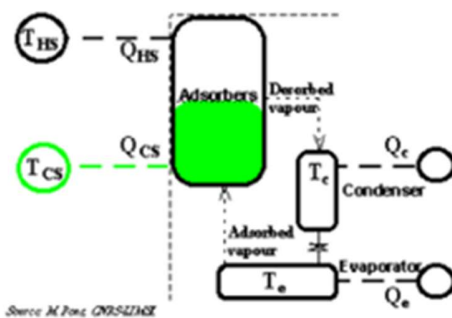
Water vapor is generated inside the evaporator by absorbing its latent heat of evaporation from the heat rejected by the chilled water flowing inside the evaporator heat exchanger. This process results in the cooling effect. The bed adsorbs the water vapor. Since the adsorption process is exothermic, a cooling fluid flows inside the bed to remove this heat. The adsorbent temperature continues to decrease, which induces the adsorption of vapor.

3. Preheating process (A-B)

During this period, there is no adsorption/desorption process. Hot water flows inside the adsorbent bed to increase its temperature, increasing its pressure from the evaporator pressure to the condenser pressure. At the end of this process, the valve between the bed and the condenser is opened.

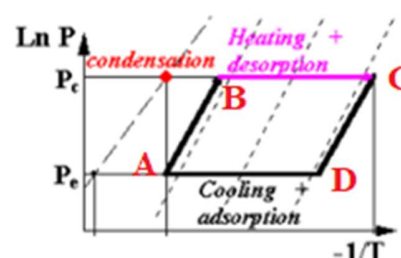
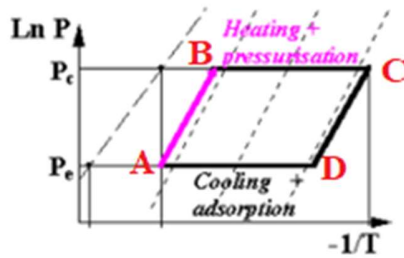
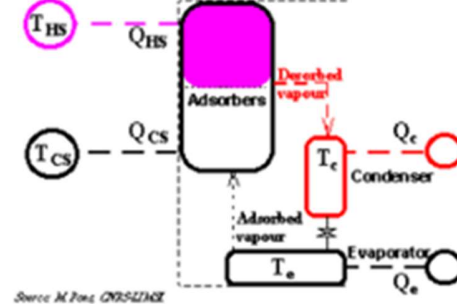
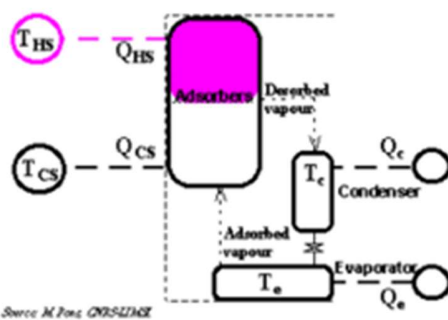
4. Desorption process (B-C)

The bed desorbs water vapor. The hot water flows inside the bed heat exchanger because the desorption process is endothermic. The adsorbent temperature continues to increase, which induces the desorption of vapor. The desorbed vapor is sent to the condenser, where it is condensed by releasing its latent heat of condensation to the cooling fluid flowing inside the condenser. The condensed vapor is then sent back to the evaporator.



1. Precooling process

2. Adsorption process



3. Preheating process

4. Description process

Figure 1-13: Absorption vs. Adsorption [16].

1.4.3 Adsorption desalination system

The single effect adsorption vapor compression desalination developed by El-desouky et al. [7] is shown in Figure 1.14. The main system components are a combined evaporator/condenser unit, adsorption/desorption beds, heat exchanger, and feed preheaters. In the beginning, the first bed is cold and saturated with water, while the second bed is hot and dry. Then, the circulating fluid starts transferring heat from the hot bed to the cold bed. The first bed is heated from T_a to T_{e1} while the second bed is cooled from T_c to T_{e2} simultaneously with no external heat source. Then, the first bed is connected to an external heat source, which increases its temperature from T_{e1} to T_c . Simultaneously, the second bed is connected to an external cooling source, which decreases its temperature from T_{e2} to T_a . Once the first bed pressure becomes higher than the condenser pressure, the bed is opened to the evaporator's tube side, where the generated steam condenses. Simultaneously, once the second bed pressure becomes lower than the evaporator pressure, the bed is opened to the evaporator's shell side where the vapor formed in the evaporator flows to the bed where it is adsorbed. In the second part of the adsorption/desorption process, the previous process reversed between the two beds.

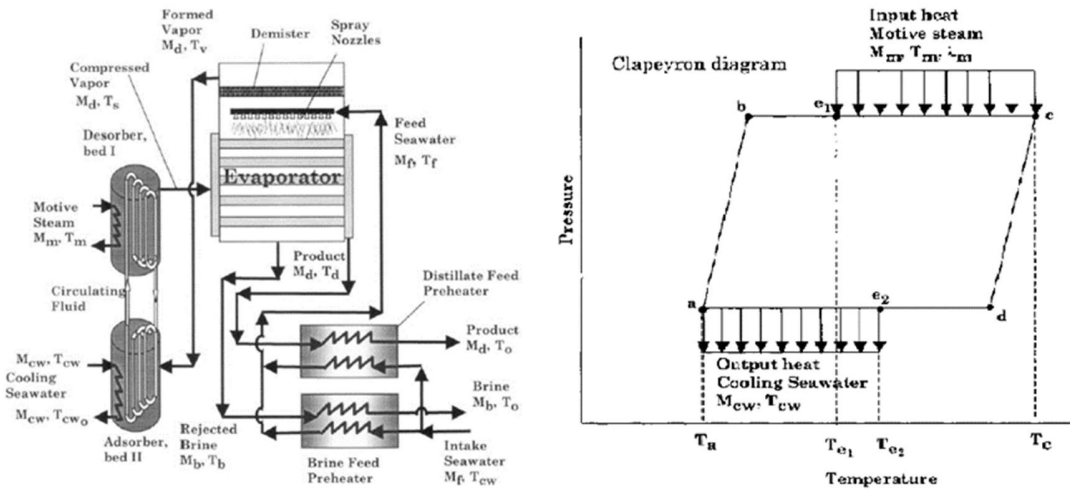


Figure 1-14: The single effect adsorption vapor compression desalination [7].

1.5 Objectives

The objectives of this study are:

- To develop a mathematical model for two beds adsorption chiller with mass recovery between the condenser and evaporator.
 - To validate the model against published results in the open literature.
 - To study the effect of the operation conditions on the system's performance.
 - To study the effect of the adsorbent materials on the system's performance.
 - To identify the most influential parameters that affect system performance.
- To model hybrid multi-effect parallel feed evaporation adsorption desalination system.
 - The model used the parallel feed multiple-effect desalination system hybridized with a two-bed adsorption system.
 - To validate the model against published results in the open literature.

- To evaluate the performance of systems.
- To study the effect of the operating conditions on the integrated system performance.
- To study the effect of the adsorbent materials on the integrated system performance.

CHAPTER 2

LITERATURE REVIEW

2.1 Adsorption Chillers

The adsorption chiller uses low-grade heat sources such as solar or waste heat and an environmentally friendly refrigerant. Moreover, its weight and size are lower than the absorption chiller. Thus, the adsorption chiller tracks scientific attention to replacing conventional mechanical driven chillers and absorption chillers. Various adsorbent-adsorbate pairs are used for adsorption cooling systems depending on the working conditions. The most widely used working pairs are activated carbon-HFC-404A [18], activated carbon-HFC-152a [18], activated carbon-HFC-407C [19], activated carbon-HFC410A [20], activated carbon-HFC-32 [20], activated carbon-CO₂ [21], activated carbon-ammonia [22], activated carbon-R134a [23], activated carbon-ethanol [24], activated carbon-methanol [25], zeolite-water [26] and silica gel-water [27-29]. Silica gel-water and zeolite-water pairs are the most popular working pairs used for the air-conditioning system. In addition, metal-organic frameworks (MOFs) are being used as an adsorbent material because of significant physical characteristics such as high surface area, high pore volume, and uniform pore size [30]. Several studies focused on experimental and analytical investigations based on mass and heat recovery for different system configurations and working pairs to enhance the system's performance because of the adsorption chiller's low performance.

Wang and Chua [36] developed mathematical models (lump-parameter and distributed-parameter) for an adsorption chiller system with heat recovery and two-beds using a silica gel-water working pair. These models were compared to experimental results. The difference between the two models is about 10%. In addition, the computing time of a distributed model is significantly longer than that of the lumped. Therefore, they reported that the lump-parameter model could be utilized to study the performance of adsorption chillers.

Saha et al. [31] performed a simulation study for a two-bed adsorption chiller with silica gel-water as a working pair to study the effect of the operating conditions (flow rates, inlet temperatures, and cycle time) on the system's performance (COP and cooling output). The highest COP, which is 0.535, was obtained at an inlet hot water temperature of 50°C and an inlet cooling water temperature of 20°C. However, under these conditions, the cooling capacity was low. Another study was performed to compare the performance of silica gel-water and silica gel-methanol systems. It is observed that the silica gel-water has a higher COP and cooling capacity [32]. Chua et al. [35] studied the impact of cycle time and switching time on the system's performance. They reported that there is an optimal cycle time and switching time to get the maximum chiller performance. Miyazaki et al. [33] performed a study to enhance the two-bed adsorption chiller's cooling performance by an innovative cycle time allocation. In this configuration, the evaporator is connected to the adsorption beds all the time. In the first half cycle, the first bed is connected to the evaporator all the time. While the second bed is in the pre-heating process until the condenser pressure reached the desorption process occurs, then at the end, it is in the pre-cooling process until the evaporator pressure reach. In the second half cycle, the second

bed is in the adsorption process, while the first bed is in pre-heating, desorption, and pre-cooling modes, respectively. The cooling capacity was improved by 6 %. Saha et al. [34] compared the system performance when CaCl_2 -in-silica gel–water is used instead of silica gel RD–water. It has been identified that the cooling power and COP of CaCl_2 -in-silica gel–water is higher than that of silica gel RD–water by 20% and 25%, respectively.

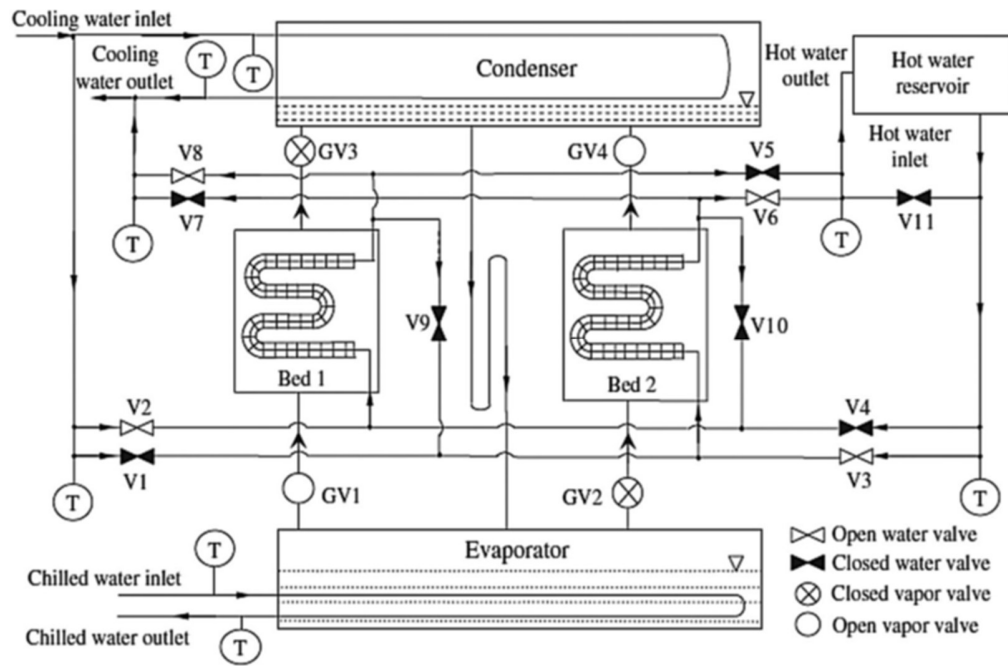


Figure 2-1: A two-bed adsorption chiller [32].

Saha et al. [37] performed a simulation study to predict a three-bed adsorption desalination system's performance, as shown in Figure 2.2, with silica gel–water as a working pair. Moreover, the study compared the waste heat recovery efficiencies of two-bed and three-bed systems. It was found that the efficiency of the three-bed is higher by 10% than the efficiency of the two-bed system.

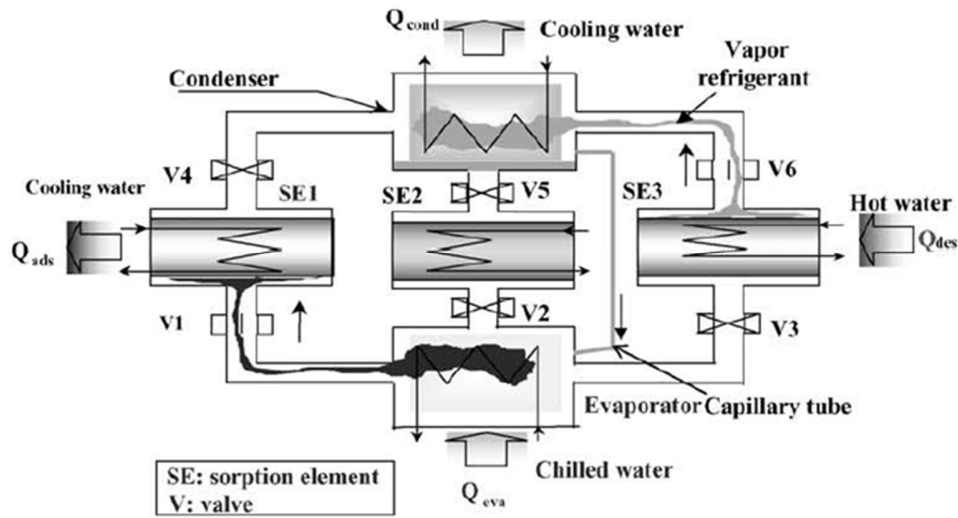


Figure 2-2: A three-bed adsorption chiller [37].

Other studies were performed to investigate the performance of a solar adsorption chiller. Sayfekar and Behbahaninia [38] conducted a simulation study on a two-bed adsorption chiller, as shown in Figure 2.3 with silica gel-water as working pair to predict the effect of operating conditions, solar collector area, and tank volume on the system's performance, solar fraction, and auxiliary heat. The results show that the COP, cooling capacity, and yearly solar fraction and auxiliary heat of this system were 0.575, 8.84 kW, 0.62, and 29427MJ, respectively. Rouf et al. [39] investigated the system's performance improvement when a three-bed adsorption cooling scheme working in a series and four-bed parallel adsorption cooling systems, as shown in Figure 2.4, were used instead of the two-bed system. The results indicated that the specific cooling capacity of the four-bed system is the highest. The maximum cooling capacity of the four-bed system at the peak hours was 150 W/kg.

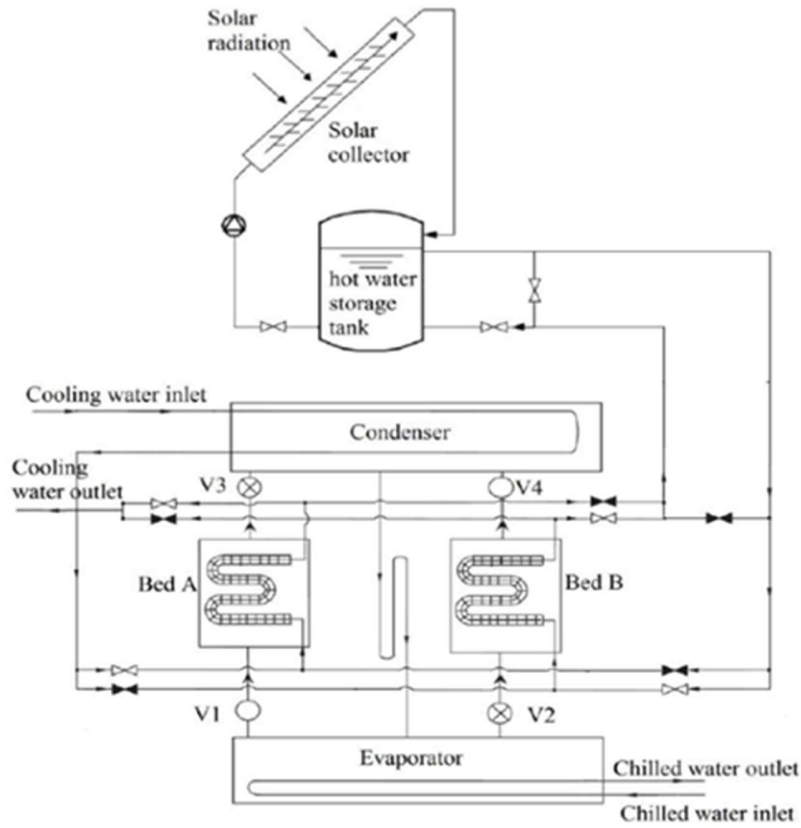


Figure 2-3: A two-bed solar adsorption chiller [38].

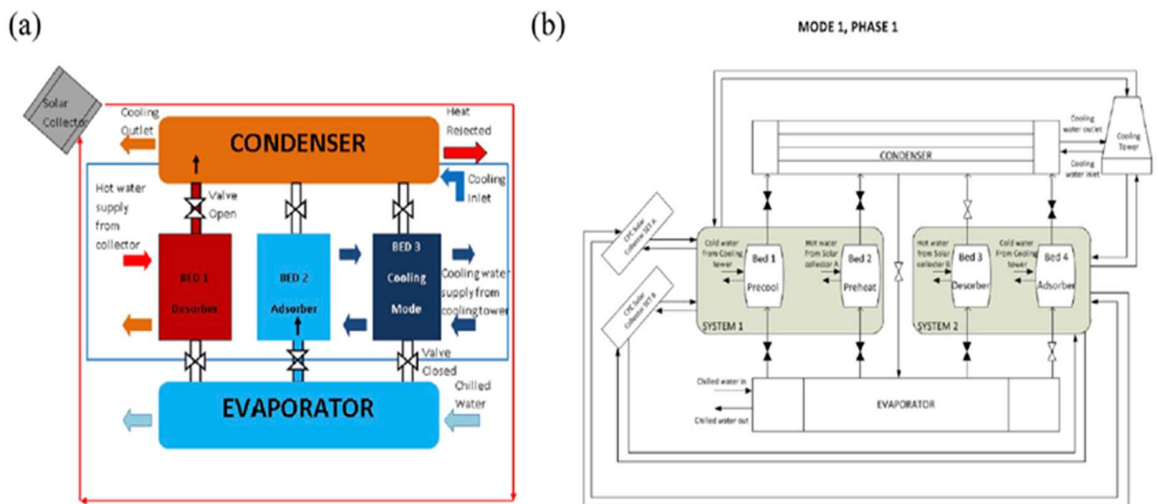


Figure 2-4: a) three-bed adsorption cooling scheme working in a series b) four-bed parallel adsorption cooling system [39]

Other studies used zeolite as an adsorbent instead of the silica gel to improve the system's performance. Li et al. [40] used the zeolite FAM Z01 as an adsorbent. Results showed that the maximum COP they could reach was 0.44 when the hot water inlet temperature was 65°C, and the adsorption/desorption time was 400 seconds. Tso et al. [41] used zeolite 13X/CaCl₂ as an adsorbent. They found that the specific cooling capacity of zeolite 13X/CaCl₂ is higher than that of silica gel-water by 30%. Teo et al. [42] studied the performance of the system with different adsorbent materials. Figure 2.5 shows that zeolite 13X/CaCl₂ has the highest COP.

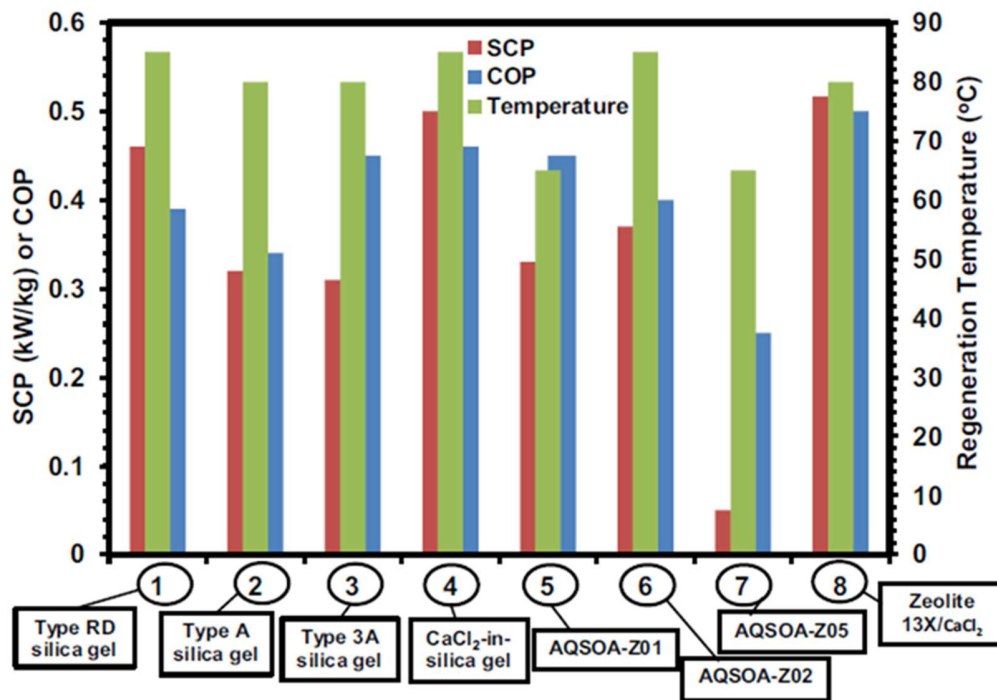


Figure 2-5: Representation of various adsorbents-water systems for comparison purposes in terms of SCP, COP, and the driving heat source temperatures under same amount of adsorbent and optimum conditions and silica gel-water working pairs [42].

A new generation of adsorbent material, called a metal framework (MOF), is used in adsorption chillers to enhance the system's performance and increase the adsorption rate. Rezk et al. [43] conducted an experimental study to predict the adsorption characteristics

and compare the water vapor uptake of different adsorbent materials. These materials were HKUST-1 (Cu-BTC), RD-2060 silica gel, and MIL-100 (Fe-BTC), while the adsorbate was water vapor. The HKUST-1 was found to have the greatest water vapor uptake at low relative pressure, while the MIL-100 had the lowest water uptake. Ali et al. [44] performed an analytical study to compare silica gel-water and HKUST-1-water pairs as working pairs for two-bed adsorption chillers. The results pointed out that the COP and cooling capacity of silica gel were lower than that of the HKUST-1. Elsayed et al. [45] studied an adsorption chiller's performance with aluminum fumarate-water and CPO-27(Ni)-water as working pairs. Results showed that aluminum fumarate is suitable for low evaporation temperature (5°C), while CPO-27(Ni) is ideal for high evaporation temperature (20°C). Qadir et al. [46] performed a numerical study to predict a two-bed solar adsorption chiller's performance with adaptive cycle time using a MIL-100(Fe)/water working pair. Table 1 shows some published results for the MOFs as adsorbent materials with water as the adsorbate.

As far as adsorption cooling system performance assessment is concerned, a careful look at the existing literature would reveal that most of the published works in the open literature do not present a comprehensive study on the effect of various new adsorbent material (MOFs) on the performance of the adsorption chiller and compare it to the conventional adsorbent material (silica gel and zeolite) for appropriate selection by researchers or engineers. The main objective of this work is to investigate the effect of the operating conditions on the performance of a two-bed adsorption chiller. This study compares the cooling performance of different adsorbent materials (silica gel RD-2060, zeolite 13X, HKUST-1, MIL-100(Fe), and aluminum fumarate) at similar operating conditions.

It is important to state that most of the MOFs used in the literature were only utilized for adsorption systems for carbon capture applications. Some studies have addressed moisture capture and desalination as a byproduct. The mass recovery from the condenser to the evaporator was ignored. In this work, we are considering the mass recovery that helps improve system performance.

Table 2-1: some published results for the MOFs as adsorbent materials with water as the adsorbate.

Reference No.	Adsorbent-adsorbate pair	System description
45	Aluminum fumarate - water	Adsorption heat pump and adsorption chiller.
	CPO-27(Ni) - water	
44	HKUST-1 - water	A two-beds adsorption chiller with mass recovery from the condenser to the evaporator.
	RD silica gel - water	
47	copper sulfate - water	A two-beds adsorption chiller with distillate water as byproduct.
48	MIL101-Cr - water	A two-beds adsorption chiller with mass recovery from the condenser to the evaporator.
	Zeolite AQSAOA-Z02 - water	
49	Aluminum fumarate - water	A two-beds adsorption chiller with distillate water as byproduct.
	MIL100(Fe) - water	
	CPO-27(Ni) - water	Integrated adsorption-Organic Rankine Cycle

		system for cooling and electricity
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2.2 Adsorption Desalination Systems

The conventional adsorption desalination (ADD) system configuration, as shown in Figure 2.6, is made up of an evaporator, condenser, and two adsorbent beds, which are filled with adsorbent material and connected to heat exchangers. The ADD's main advantages are desalinating the seawater water at a temperature below the ambient temperature using low heat energy sources. The adsorption heat pump replaces the thermal ejector and mechanical compressor. The main difference between ADD and adsorption chiller is that the condensed water vapor is collected as distillate in ADD, while it is recirculated again to the evaporator for the adsorption chiller.

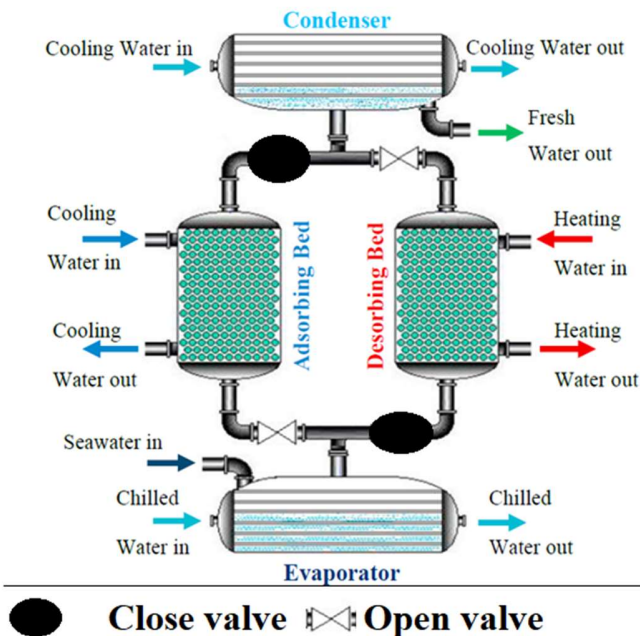


Figure 2-6: Convectional two-bed adsorption desalination system [50].

The thermodynamic cycle of the ADD system is shown in Figure 2.7. It starts with isosteric heating (preheating process) at point 1; at this process, a hot fluid flows inside the isolated saturated bed's heat exchanger until its pressure increases from the evaporator pressure to the condenser pressure at point 2. At point 2, the bed connects to the condenser. It starts isobaric heating to desorb the water vapor from the bed, which will condense in the condenser until the maximum temperature is reached at point 3. At point 3, the bed is isolated, and isosteric cooling (precooling process) starts. At this process, a cold fluid flows inside the isolated dry bed's heat exchanger until its pressure decreases from the condenser pressure to the evaporator pressure at point 4. At point 4, the bed connects to the evaporator. It starts isobaric cooling to adsorb the water vapor from the evaporator until the minimum temperature is reached (point 1). The cooling effect proceeds from the adsorption of the water vapor formed in the evaporator. At the same time, desalination results from the condensation of the desorber water vapor.

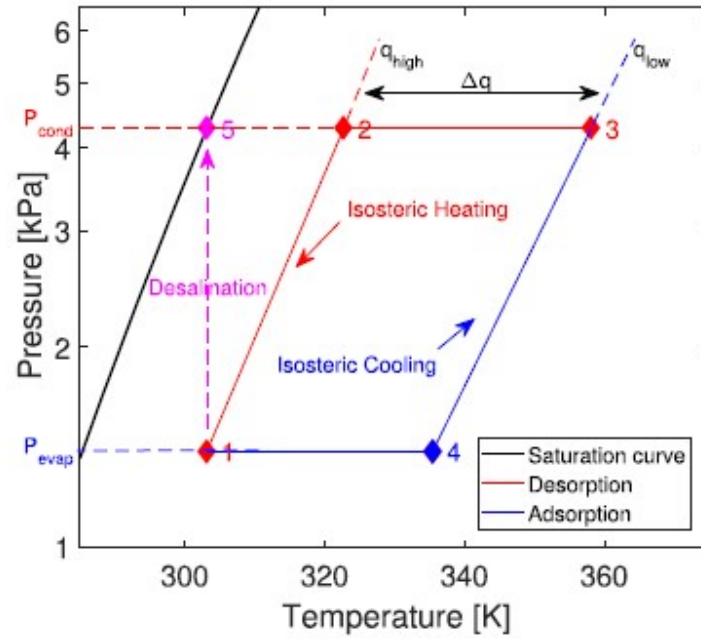


Figure 2-7: The thermodynamic cycle of the ADD system [54].

The available literature on adsorption desalination systems covers several aspects, including different systems configurations, various operating conditions, and the effect of heat recovery circuit. Also, the impacts of varying adsorbent materials as well as combining the adsorption system with a multi-effect desalination system have been analyzed.

Alsaman et al. [66] performed a simulation and an experimental study to investigate a convectional two-bed adsorption desalination (ADD) system's performance with silica gel-water as a working pair. Results show that the specific daily water production (SDWP), the COP, and the specific cooling capacity (SPC) are 4.2 cubic meters per ton of silica gel, 0.451, and 115 watts per kilograms of silica gel, respectively. Rezk et al. [62] performed an optimization study to obtain the optimum operating conditions of heating, cooling, cycle time, and heating water flow rate. Results show that a 70% increase in both SDWP and SCP are reported just by using the right operating conditions with no change in the system design or the used materials. Olkis et al. [65] performed an experiment to study the ADD

performance with heat recovery between the beds. Results show that the ADD system with heat recovery has a 25% reduction of energy consumption than the ADD system without heat recovery. Alsaman et al. [58] investigated the ADD system's performance with the heat recovery circuit between the evaporator and condenser. This study showed the improvement of the ADD system's performance with heat recovery compared to the ADD without heat recovery. The study pointed out that the SDWP for the system with heat recovery is more than twice the amount of distillate water produced by the system without heat recovery. Woo et al. [64] performed an experimental study on the effect of different types of brine feeders in the evaporator (i.e., perforated plate and spray nozzles) on the ADD system's performance. It was also observed that the SDWP of the ADD system with a spray nozzle is 18 % higher than the one with a perforated plate.

Shahzad et al. [61] developed a mathematical model for a pressure swing adsorption cooling and desalination system, which can use low-pressure steam for regeneration utilizing a thermal vapor compressor. The system configuration is shown in Figure 2.8.

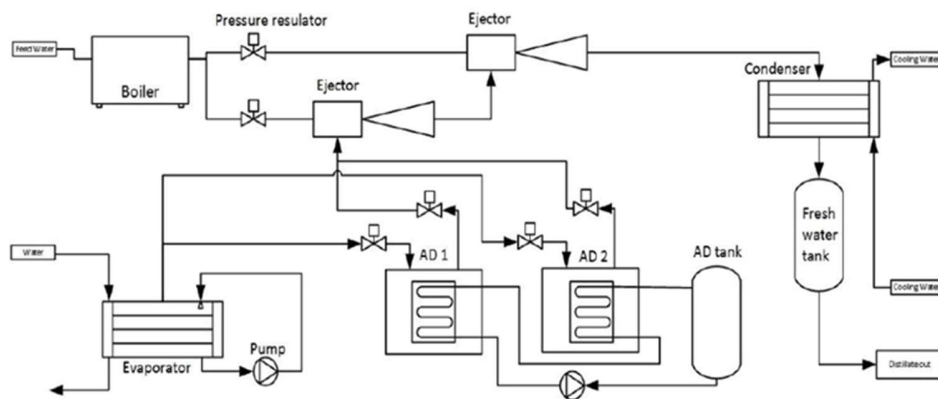


Figure 2-8: A pressure swing adsorption cooling and desalination system with thermal vapor compressor [61].

Askalany [60] developed a mathematical model of an innovative system that combined two beds adsorption system with mechanical vapor compression (MVC-AD) to produce both

cooling effect and potable water, as shown in Figure 2.9. The daily desalinated water (DWP) for the MVC-AD system is higher than the DWP of ADD.

Thu et al. [63] performed a two-bed adsorption desalination simulation study with an integrated evaporator-condenser unit, as shown in Figure 2.10. The results showed that the specific water daily water production increases by more than 300% compare to the convectional system configuration. Also, this system has a lower pumping power consumption.

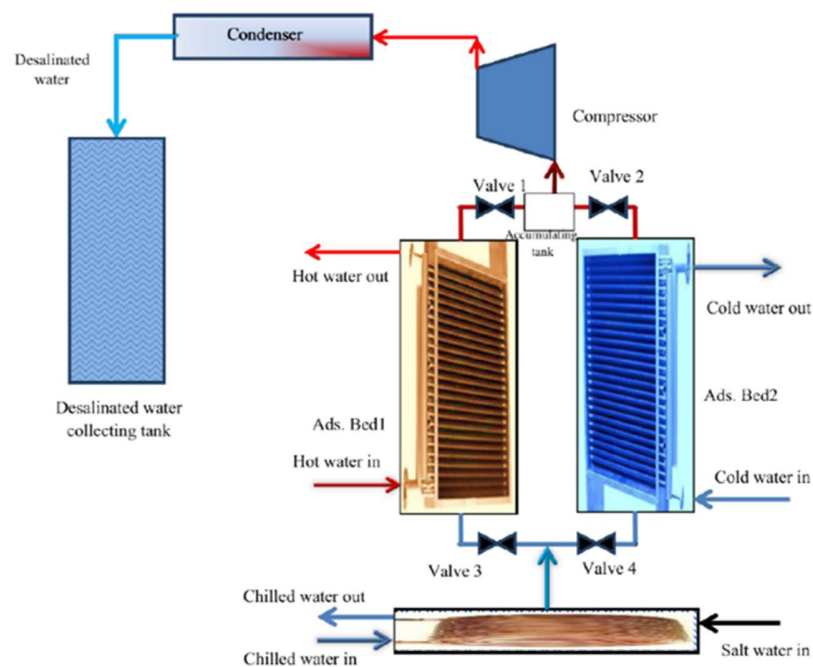


Figure 2-9: The MVC-AD system [60].

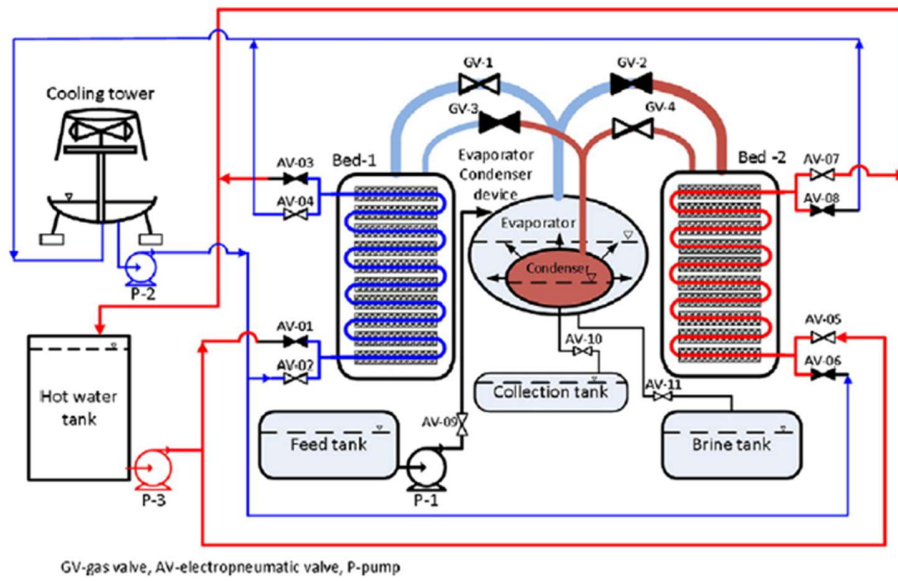


Figure 2-10: An adsorption desalination system with an integrated evaporator–condenser [63].

Thu et al. [67] performed a simulation study of a three-bed, two-evaporator, and one-condenser adsorption desalination system, as shown in Figure 2.11. The results show that the COPs for low-pressure and high-pressure evaporators are 0.33 and 0.51, respectively.

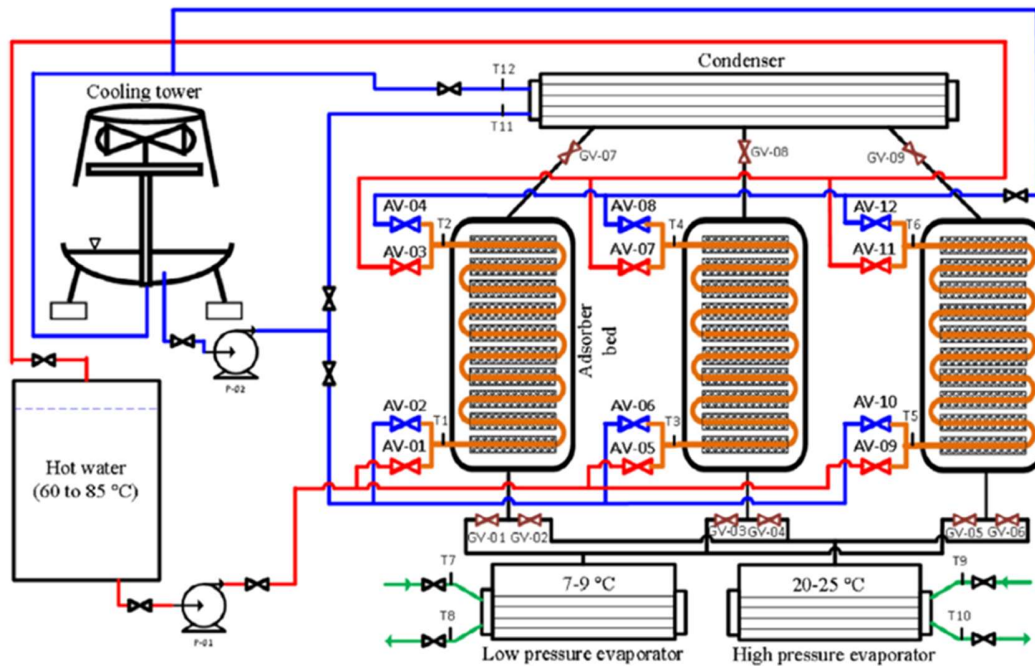


Figure 2-11: A 2-evaporator, 3-bed adsorption desalination system [67].

Other studies have been performed to study the effect of replacing the two-bed with a four-bed adsorption system. Thu et al. [70] conducted an experimental study to compare system performance with two-bed and four-bed systems. The four-bed system can improve the SDWP by about 14%. Ng et al. [68] developed a mathematical model for a waste heat-driven adsorption cycle (AD) with four beds, as shown in Figure 2.12. The model was developed to produce distillate water and cooling effect. The model studies the impact of the hot water's inlet temperature on the system performance (e.g., the daily water production and the cooling effect). Results were validated using the data generated from the experimental setup. Other studies were performed to study the cycle time effect on the system's performance [69,74]. Further improvement was implemented by adding a heat exchanger between the cooling water and chilled water. Results show that SDWP improved by about 22% [59].

Thu et al. [72] developed a mathematical model of a four-bed adsorption desalination system with heat recovery between the evaporator and condenser. A simulation study and an experiment were performed to study the operating conditions' effect on the system's performance. Another analysis was performed to compare the system's performance with and without a heat recovery circuit between the evaporator and condenser. The heat recovery can improve SDWP by about 90% [73].

Chen and Chua [71] developed a mathematical model for a four-bed two-evaporator adsorption chiller with distillate water as a byproduct, as shown in Figure 2.13. A simulation study was performed to study the effect of the operating conditions on the system's performance.

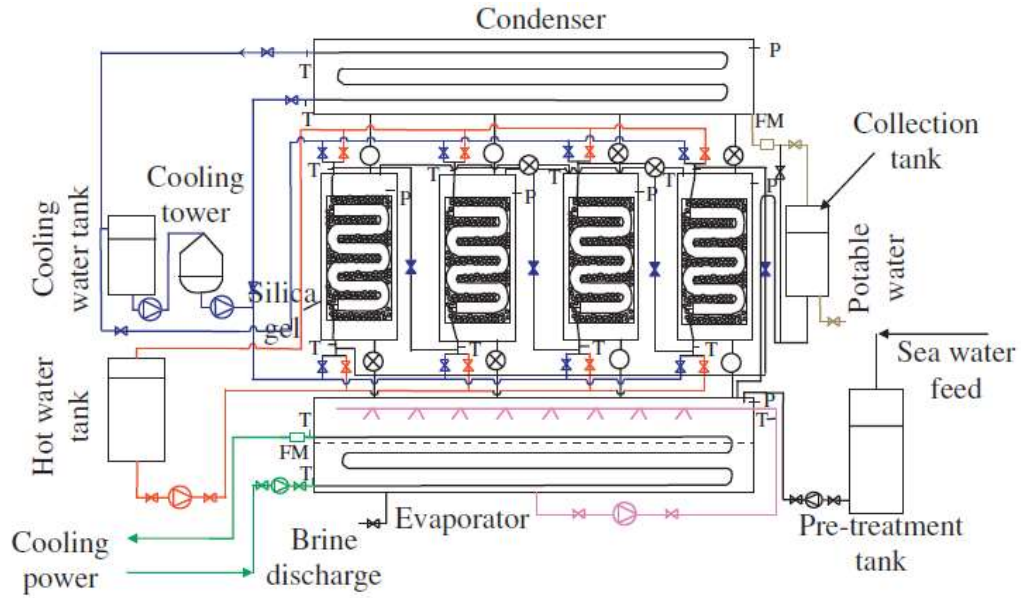


Figure 2-12: A 4-beds adsorption desalination-cooling system [68].

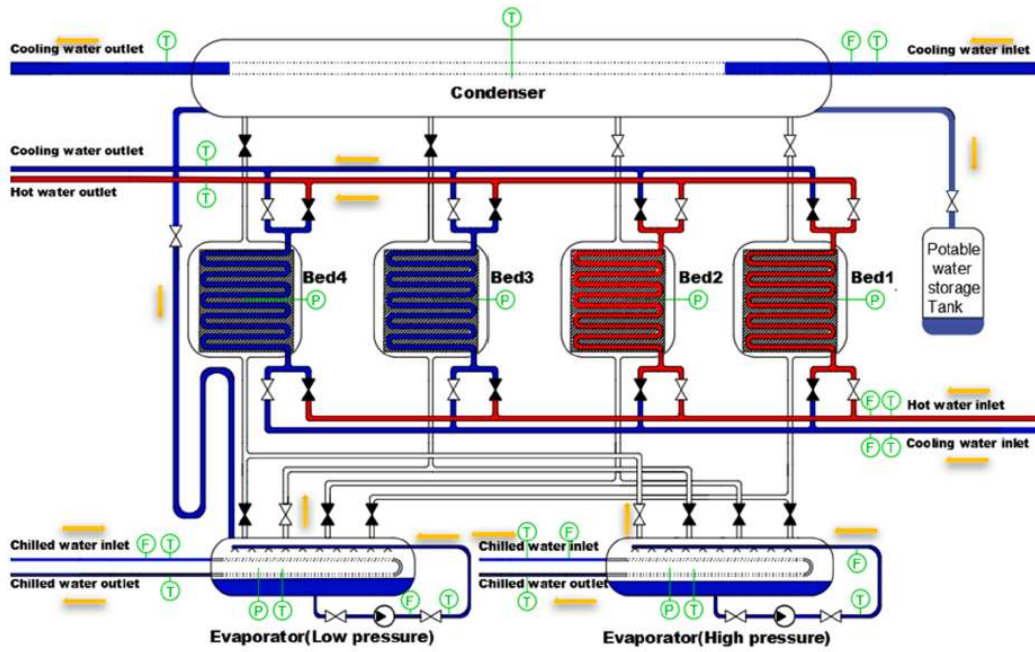


Figure 2-13: A four-bed, two-evaporators adsorption desalination & cooling system [71].

Youssef et al. [75] developed a mathematical model for a dual cooling-adsorption system with an integrated evaporator-condenser and heat recovery circuit between the first stage's evaporator and the second stage's condenser, as shown in Figure 2.14. The adsorbent-adsorbate pair was the AQSOA-ZO2-water pair. A simulation study was performed to

compare this system performance with a conventional two-bed adsorption desalination system. The results show that the SDWP can be more than twice the SDWP of the two-bed system.

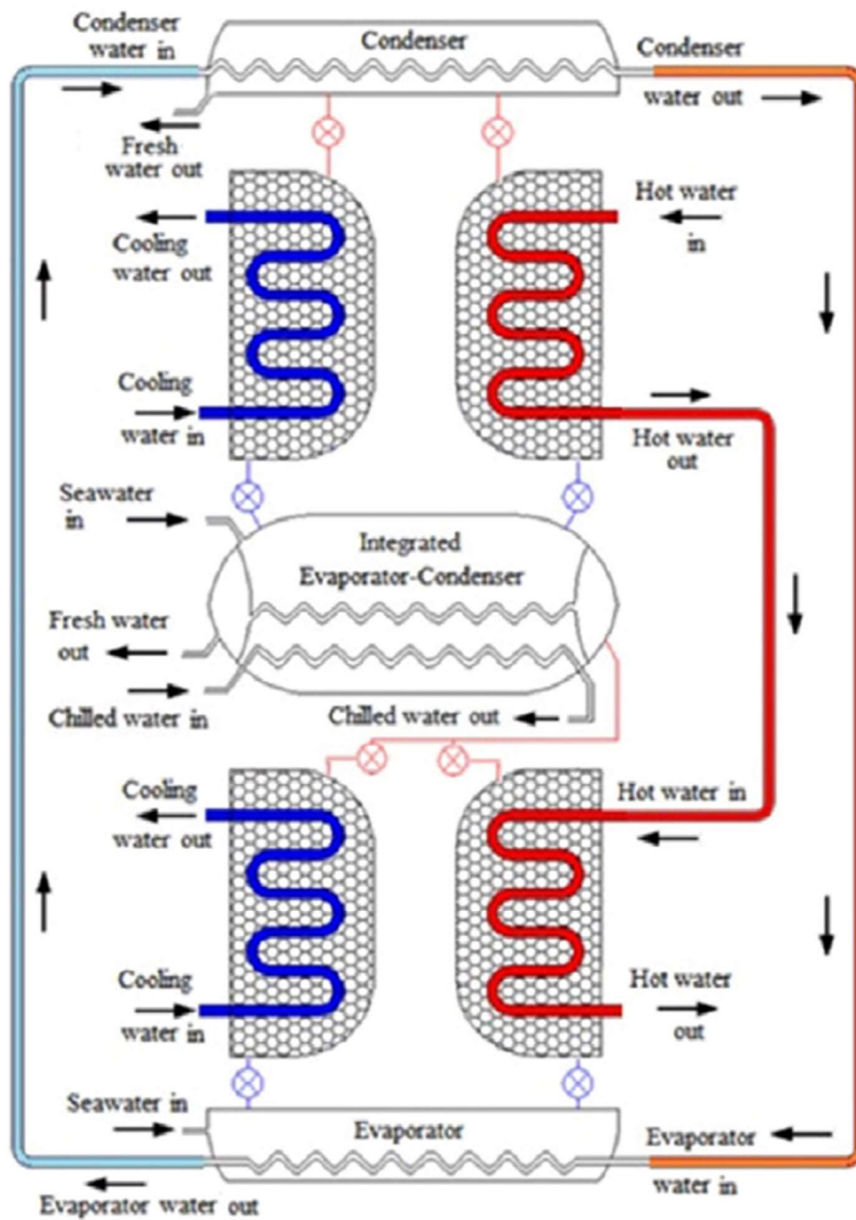


Figure 2-14: A multi-cycle adsorption desalination & cooling system with integrated evaporator/condenser [75].

In another study, a numerical study for a dual cooling-adsorption system with a heat recovery circuit was performed, as shown in Figure 2.15. The system consists of two

stages; the first stage performed cooling and desalination while the second stage was made only for desalination. The heat recovery is placed between the first and second stages' condenser and the second stage's evaporator. The adsorbent-adsorbate pair in this study was zeolite (AQSOA-Z02) -water pair. The results showed that the cooling capacity increased by more than 44% and the specific daily water production by more than 45% in comparison with convectional systems [76].

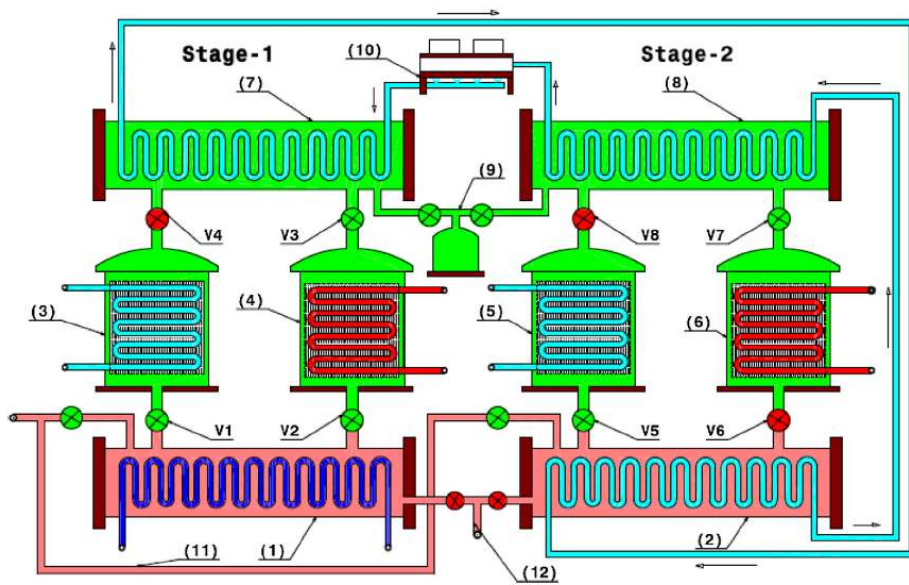


Figure 2-15: multistage adsorption desalination & cooling system [76].

Researchers use metal-organic frameworks (MOFs) as adsorbent materials instead of the silica gel to improve the adsorption desalination system's performance. Youssef et al. [50] performed a simulation study to predict adsorbent materials' influence on a two-bed adsorption system's performance. The adsorbent materials selected for this study were silica gel, zeolite AQSOA-Z02, and Aluminum Fumarate. Results show that the Aluminum Fumarate specific daily water production is 345% and 200% higher than the AQSOA-Z02 and silica gel, respectively. Another study was performed using the CPO-27(Ni) as an adsorbent material [56]. Another analysis was performed using the copper sulfate as an

adsorbent material [47]. Elsayed [57] conducted an experimental study to predict the water vapor uptake of MIL-101(Cr) and CaCl₂ composite.

2.3 Hybridization of Adsorption System with Convectional

Desalination Systems

Different hybrid configurations with reverse osmosis (RO) and multi-effect desalination (MED) were studied to utilize both systems' advantages and overcome their limitations. One of the MED system's limitations is the bottom brine temperature; the last effect's temperature should be higher than the ambient temperature. Thu et al. [81] developed a mathematical model of hybridization of the adsorption cycle with the multi-effect distillation (MED-AD) with a backward feeding configuration, as shown in Figure 2.16. Results showed that the final condensing temperature can as low as 5 °C, which is below the ambient temperature. Another study has performed to study the performance of a backward feeding configuration MED-AD with heat recovery between MED last effect and AD condenser [80]. Shazad et al. [79] designed a mathematical model of a parallel feed MEDAD system, as shown in Figure 2.17. A simulation study was performed to compare the MEDAD against the conventional MED system in terms of the water production rate. The results show a significant water production improvement (more than three folds) by hybridization with the same heat input and high side temperature. Another study was performed to predict the performance of a novel hybrid MEDAD+RO system as shown in Figure 2.18 [78]. The results show that this system's water production was double the water production of the MED system.

Atab [82] performed a simulation study of hybridization of brackish water reverse osmosis (BWRO) desalination technology and an adsorption cycle (AD), as shown in Figure 2.19. The results showed that the RO-AD system's cost of water production is lower than the RO stand-alone system by more than 30%.

Qasem and Zubair [83] performed a numerical study of adsorption humidification-dehumidification (air-heated) desalination and cooling system (HDHAD) with two layouts, as shown in Figure 2.20. In the first one, feed water was precooled in the adsorption evaporator before entering the humidification-dehumidification (HDH) system. In contrast, in the other design, the seawater cooled the adsorption bed. The results showed that gained output ratios are 7.8 and 7.6 for the first design and the second design, respectively. Also, the COP of the adsorption system is higher than 0.45. Table 2.2 summarizes the work found in the literature review in terms of the COP, SDWP, and SCP of the adsorption desalination and cooling system.

It is noted that little attention has been given to the MED-AD systems with parallel feed configuration system. Only the work reported by Shahzad has been performed for this system. This work, however, has only considered silica gel as an adsorbent material and did not include the effect of the adsorption system's operating conditions on the system performance. Thus, it is intended in this thesis to investigate the impact of the operating conditions on the system's performance. Moreover, the HKUST-1 adsorbent material is utilized instead of the silica gel due to its superior performance characteristics that are discussed in the next few chapters.

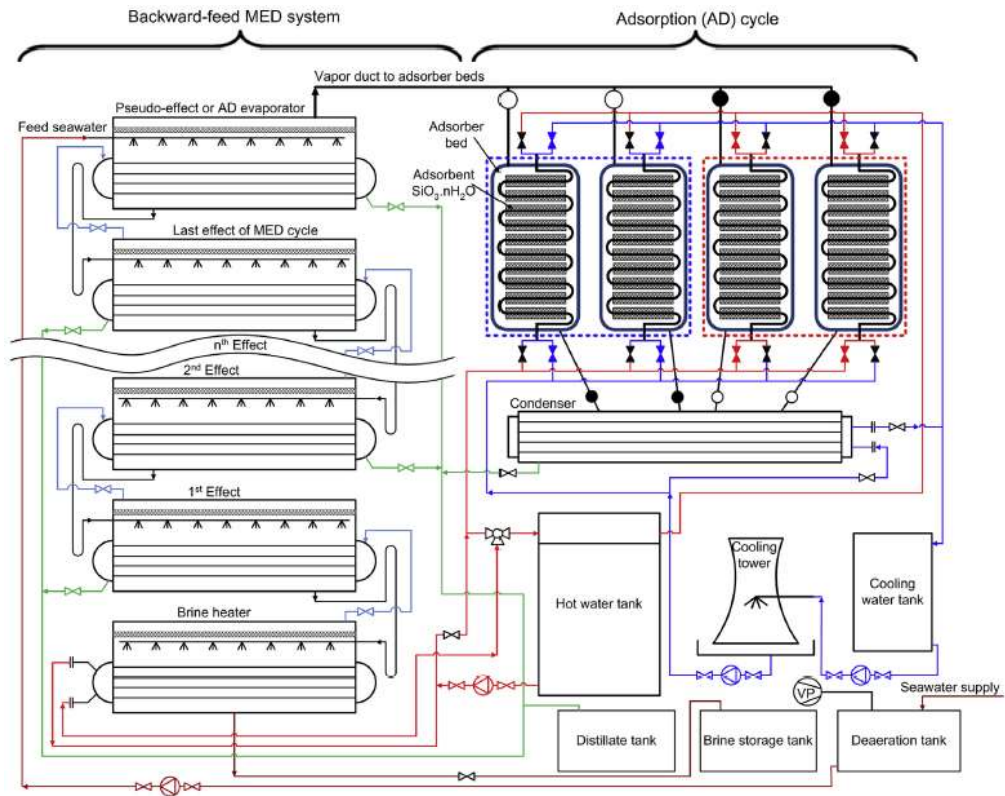


Figure 2-16: A backward feeding MEDAD [81].

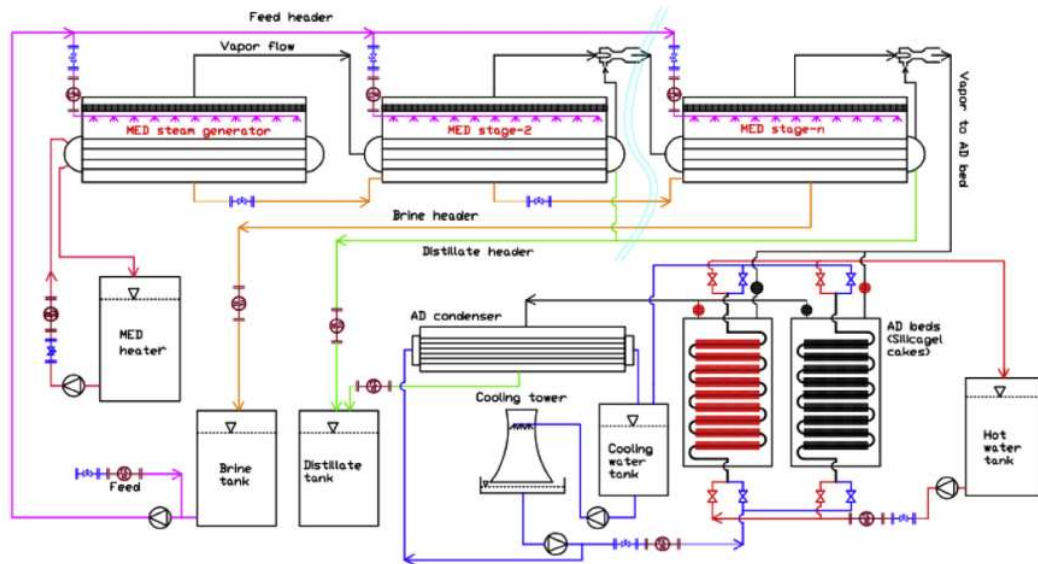


Figure 2-17: A parallel feeding MEDAD [79].

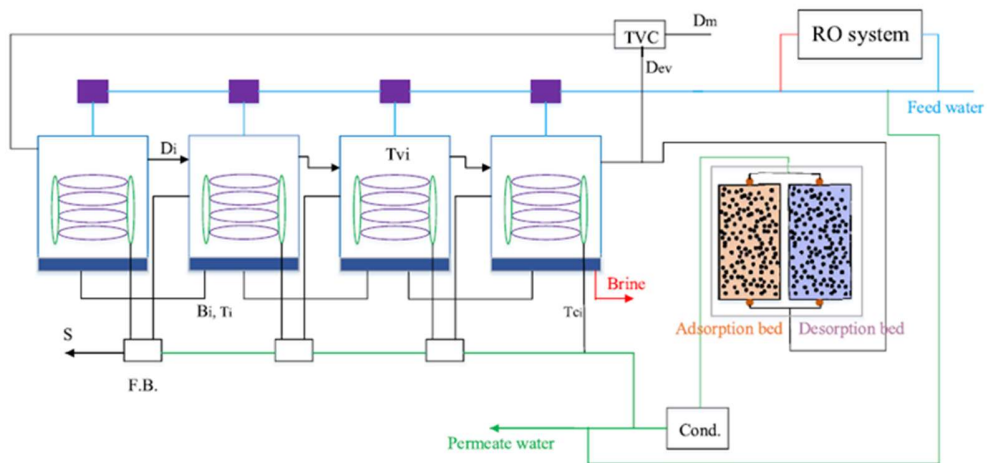


Figure 2-18: A parallel feeding MEDAD [78].

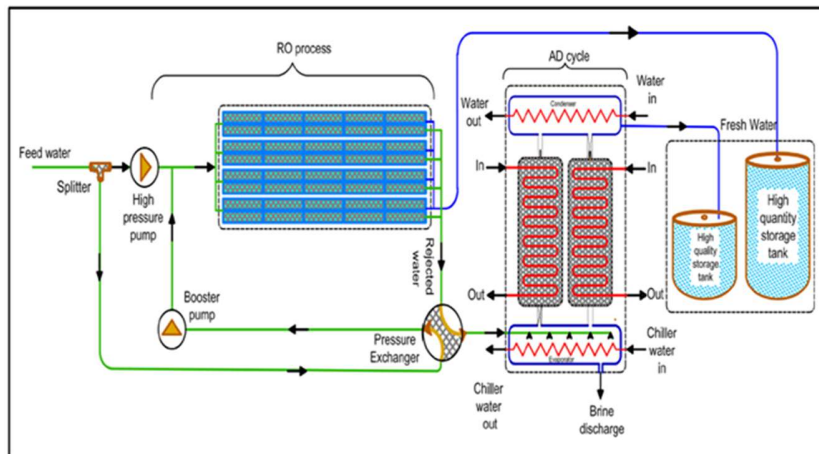


Figure 2-19: A RO+AD system [82].

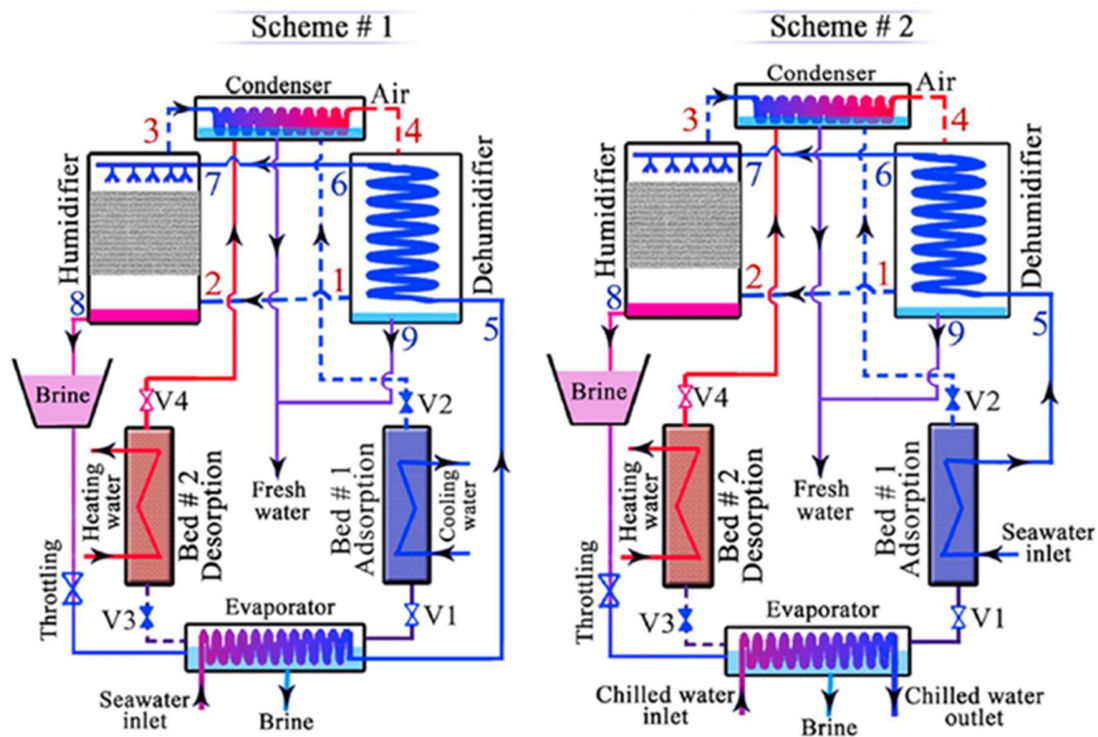


Figure 2-20: A HDHAD system [83].

Table 2-2: Summary of the literature review.

Reference No.	System description	Study	DWP	Cooling capacity	COP	PR	GOR
55	• Single effect evaporation desalination with adsorption heat pump. • Desalination system. • Zeolite – water.	Numerical	1 kg per second	-	-	7.25	-
62	• Two beds, single evaporator, and single condenser. • Cooling system with desalination as a byproduct. • Silica gel – water.	Numerical	6.9 m ³ per ton of adsorbent	191 W per kg of adsorbent	0.961	-	-
	• Two beds, single evaporator, and single condenser.						

58	- Cooling system with desalination as a byproduct. - Silica gel – water.	Numerical & Experimental	4 m ³ per ton of adsorbent	112 W per kg of adsorbent	0.45	-	-
64	- Two beds, single evaporator, and single condenser with spray nozzle. - Cooling system and desalination as byproduct. - Silica gel – water.	Numerical	6.76 m ³ per ton of adsorbent	37.05 Rton per ton of adsorbent			
65	- Two beds, single evaporator, and single condenser with heat recovery between the evaporator and the condenser. - Desalination system. - Silica gel – water.	Experimental	7.7 kg per kg of adsorbent	-	-	0.6	-

63	- Two beds with integrated evaporators-condenser device. - Desalination system. - Silica gel – water.	Numerical	26 m ³ per ton of adsorbent	-	-	-	-
73	- Two beds, single evaporator, and single condenser with heat recovery between the evaporator and the condenser. - Desalination system. - Silica gel – water.	Numerical	13.46 m ³ per ton of adsorbent	-	-	0.755	-
59	Two beds, single evaporator, and single condenser with heat exchanger between the chilled water coming out from the	Experimental		-	-	0.65	-

	evaporator and cooling water supplied to the adsorber bed. • Cooling system with desalination as a byproduct. • Silica gel – water.		8.2 m ³ per ton of adsorbent				
60	• Two beds, single evaporator, and single condenser with mechanical vapor compression between the condenser and desorber. • Cooling system with desalination as a byproduct. • Silica gel – water.	Numerical	14 m ³ per ton of adsorbent	0.21kW per kg of adsorbent	0.6	-	-
	• Two beds, single evaporator, and single condenser with						

61.	thermal vapor compression between the condenser and desorber. - Cooling system with desalination as a byproduct. - RD Silica gel – water.	Numerical	-	-	-	-	-
70	- Two beds, single evaporator, and single condenser. - Desalination system. - Silica gel – water.	Experimental	8.8 m ³ per ton of adsorbent	-	-	0.573	-
	- Four beds, single evaporator, and single condenser. - Desalination system. - Silica gel – water.		10 m ³ per ton of adsorbent	-	-	0.607	-

68	• Four beds, single evaporator, and single condenser. • Cooling system with desalination as a byproduct. • Silica gel – water.	Numerical & Experimental	8 m ³ per ton of adsorbent	51.6 Rton per ton of adsorbent	-	-	-
69	• Four beds, single evaporator, and single condenser. • Cooling system with desalination as a byproduct. • Silica gel – water.	Experimental	4.7 kg per kg of adsorbent	-	0.38	-	-
74	• Four beds, single evaporator, and single condenser. • Cooling system with desalination as a byproduct. • Silica gel – water.	Numerical	2.3 kl per kg of adsorbent	18.5 Rton per ton of adsorbent	0.4	-	-

71	• Four beds, two evaporators, and single condenser. • Cooling system with desalination as a byproduct. • Silica gel – water.	Numerical	5.24 m ³ per ton of adsorbent	19 kW	0.65	-	-
72	• Four beds, single evaporator, and single condenser with heat recovery between the condenser and the evaporator. • Desalination system. • Silica gel – water.	Numerical	9.24 m ³ per ton of adsorbent	-	0.77	-	-
67	• Three beds, two evaporators, and single condenser. • Cooling system with desalination as a byproduct.	Numerical		10.8 kW for HPE and 7.1	0.82	0.67	-

	Silica gel – water.		6.67 m ³ per ton of adsorbent	kW for LPE			
76	- Two stages adsorption system. First stage, cooling system with desalinate water as byproduct. Second stage, desalination system with heat recovery between the evaporator and the condenser. - Cooling and desalination system. - AQSOA-Z02 in stage-1 and silica gel in stage-2	Numerical	5.62 m ³ per ton of adsorbent for stage-1 and 10.14 m ³ per ton of adsorbent for stage-2	47.82 Rton per ton of adsorbent for stage-1	-	-	-

75	- Two 2-adsorber bed cycles linked with integrated evaporator/condenser, one cycle uses the integrated evaporator/condenser as evaporator (upper) and the second one uses it as a condenser (lower). - Cooling and desalination system. - Zeolite AQSOA-Z02 – water.	Numerical	15.4 m ³ per ton of adsorbent	46.6 Rton per ton of adsorbent	-	-	-
	- Two beds, single evaporator, and single condenser. - Cooling system with desalination as a byproduct.				-	-	-

50	Al-fumarate – water.	Numerical	11.3 m ³ per ton of adsorbent	90.9 Rton per ton of adsorbent			
	- Two beds, single evaporator, and single condenser. - Cooling system with desalination as a byproduct. - AQSOA-Z02 – water.		6.4 m ³ per ton of adsorbent	50.5 Rton per ton of adsorbent	-	-	-
	- Two beds, single evaporator, and single condenser. - Cooling system with desalination as a byproduct. - Silica gel – water.		8.4 m ³ per ton of adsorbent	62.4 Rton per ton of adsorbent	-	-	-

45	- Two beds, single evaporator, and single condenser. - Cooling system with desalination as a byproduct. - CPO-27Ni – water.	Numerical	22.8 m ³ per ton of adsorbent	220 Rton per ton of adsorbent	-	-	-
47	- Two beds, single evaporator, and single condenser. - Cooling system with desalination as a byproduct. - Copper sulfate – water.	Numerical	8.2 m ³ per ton of adsorbent	227 W per kg of adsorbent	0.57	-	-
82	- Hybrid reverse osmosis/adsorption desalination and cooling plant. - Cooling and desalination system.	Numerical	6.3 m ³ per ton of adsorbent	75 Rton per ton of adsorbent	-	-	-

	• Silica gel – water.						
83	• Hybrid humidification-dehumidification (air-heated) system with an adsorption desalination system. • Desalination system. • Silica gel – water.	Numerical	22 kg per hours	-	-	-	8
	• Hybrid humidification-dehumidification (air-heated) system with an adsorption desalination system. • Cooling and desalination system. • Silica gel – water.		21.5 kg per hours	-	-	0.57	7.6

81	- Hybridization of adsorption cycle with the multi-effect distillation (MED) -Backward feeding with 8 effect. - Desalination system. - Silica gel – water.	Numerical	0.541 kg/s	-	-	7.87	6.63
80	- Hybridization of adsorption cycle with the multi-effect distillation (MED) -Backward feeding with 7 effect with heat recovery loop between the AD condenser and MED first effect. - Desalination system. - Silica gel – water.	Numerical	25 m ³	-	-	6.5	5.1

79	- Hybridization of adsorption cycle with the multi-effect distillation (MED) -parallel feeding with 8 effect. - Desalination system. - Silica gel – water.	Numerical	5 liter per minutes	-	-	-	-
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CHAPTER 3

ADSORPTION CHILLER

Adsorption chiller performance depends on the adsorbent-adsorbate pair; therefore, employing an adsorption working pair with high water uptake is an effective way to boost system performance [51]. The selection of adsorbent-adsorbate pairs depends on particular desirable characteristics, including their chemical, physical and thermodynamic properties, availability and costs. An excellent solid adsorbent must have a reasonably wide surface area and a volume of micropore. The adsorbate selected for this study is water because it has a high latent heat of vaporization. The adsorbent materials chosen for this study are RD silica gel, zeolite 13X, MIL-100(Fe), HKUST-1, and aluminum fumarate. The mathematical model of a two-bed adsorption chiller is built in this chapter. A simulation analysis is carried out to study the impact of operating conditions and adsorbent materials on the performance of the system.

3.1 Mathematical Model of a Two-Bed Adsorption Chiller

3.1.1 Energy and Mass Balance

The lumped parameter model was used to build a mathematical two-bed adsorption chiller (shown in Figure 3.1) model, with the following assumptions:

- Temperature, pressure, and the mass adsorbed in the adsorber beds are uniform.

- The difference in pressure between the adsorber bed and the condenser or the evaporator and desorber bed is insignificant.
- The impact of convection heat transfer within the beds is negligible.
- The adsorbed/desorbed water's specific heat capacity is the same value as for the bulk liquid water.
- The loss of heat surrounding is ignored.

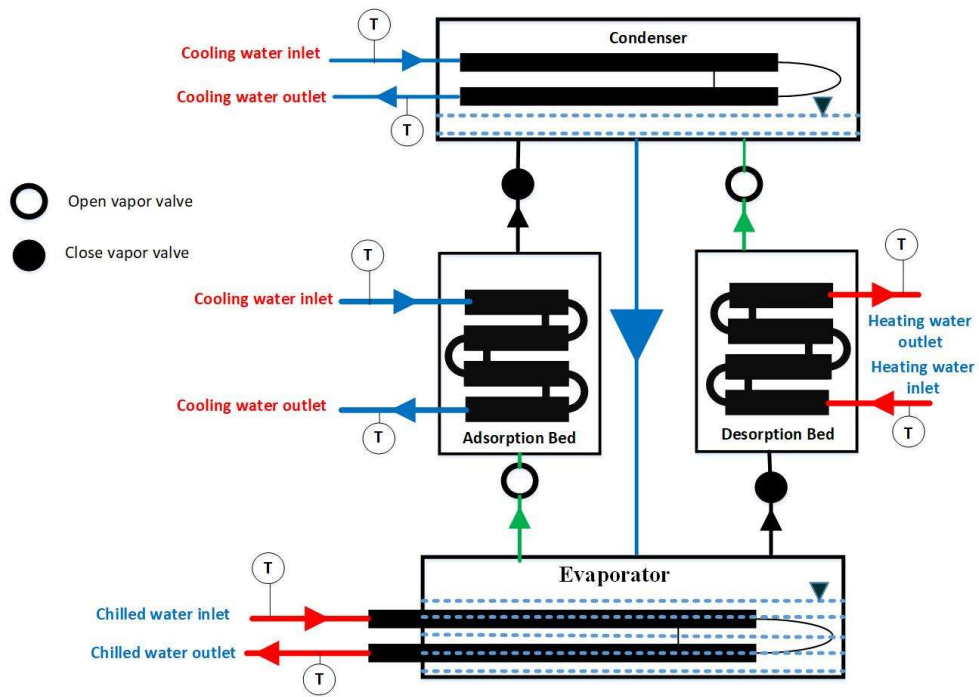


Figure 3-1: Two-bed adsorption chiller.

The adsorber/desorber energy balance equation can be written as [31]:

$$\begin{aligned}
 & (M_{adt}C_{p,adt} + M_{adt}q_{bed}C_{p,w} + M_{hex}C_{p,hex}) \frac{dT_{bed}}{dt} \\
 & = \dot{m}_f C_{p,f} (T_{in,bed} - T_{out,bed}) + M_{adt} Q_{st} \frac{dq_{bed}}{dt}
 \end{aligned} \tag{3.1}$$

$$T_{out,bed} = T_{bed} + (T_{in,bed} - T_{bed}) \exp \left[-\frac{(UA)_{bed}}{\dot{m}_f C_{p,f}} \right] \quad (3.2)$$

where $z=0$ for desorbing beds and $z=1$ for adsorbing beds. The equation's (3.1) left side reflects the sensible heat of the adsorbent material, the adsorbate within the bed, and the heat exchanger. The right-hand side reflects the sensible heat emitted or absorbed by the heat transfer fluid plus the heat of the adsorption/desorption phase in addition to the sensible heat required to overheat the incoming adsorbate vapor from the evaporator temperature to the bed temperature. Equation (3.2) is used to measure the temperature of the fluid outlet based on the log mean temperature difference of the adsorber bed heat exchanger.

The condenser energy balance equation can be written as [31]:

$$\begin{aligned} (M_{hex,cond} C_{p,hex,cond}) \frac{dT_{cond}}{dt} \\ = \dot{m}_{f,cond} C_{p,f} (T_{in,cond} - T_{out,cond}) - M_{adt} h_{fg} \frac{dq_{des}}{dt} \\ + M_{adt} C_{p,v} (T_{cond} - T_{bed}) \frac{dq_{des}}{dt} \end{aligned} \quad (3.3)$$

$$T_{out,cond} = T_{cond} + (T_{in,cond} - T_{cond}) \exp \left[-\frac{(UA)_{cond}}{\dot{m}_{f,cond} C_{p,f}} \right] \quad (3.4)$$

The equation's (3.3) left side defines the metal heat exchanger's sensible heat inside the condenser. The first term on the right-hand side defines the sensible heat absorbed by the heat transfer fluid. The second term is the heat of the condensation process, while the third term is the sensible heat of the adsorbate formed by the desorber. Equation (3.4) is used to

calculate the fluid outlet temperature based on the condenser heat exchanger's log mean temperature difference.

The evaporator energy and mass balance equations can be written as [31]:

$$\begin{aligned}
 & (M_{\text{evap}}C_{p,w} + M_{\text{hex,evap}}C_{p,\text{hex,evap}}) \frac{dT_{\text{evap}}}{dt} \\
 & = \dot{m}_{f,\text{chill}}C_{p,f}(T_{\text{in,chill}} - T_{\text{out,chill}}) - M_{\text{adt}}h_{fg} \frac{dq_{\text{ads}}}{dt} \\
 & + M_{\text{adt}}C_{p,w}(T_{\text{evap}} - T_{\text{cond}}) \frac{dq_{\text{des}}}{dt}
 \end{aligned} \tag{3.5}$$

$$T_{\text{out,chill}} = T_{\text{evap}} + (T_{\text{in,chill}} - T_{\text{evap}}) \exp \left[-\frac{(UA)_{\text{evap}}}{\dot{m}_{f,\text{chill}}C_{p,f}} \right] \tag{3.6}$$

$$\frac{dM_{\text{evap}}}{dt} = -M_{\text{adt}} \left(\frac{dq_{\text{ads}}}{dt} + \frac{dq_{\text{des}}}{dt} \right) \tag{3.7}$$

The terms in the left side of the equation (3.5) describe the sensible heat of the liquid refrigerant and the metal heat exchanger within the evaporator, respectively. The right-hand side is the sensible heat emitted by the chilled water (cooling effect), the heat of the evaporation phase and the sensible heat of the adsorbate leaving the condenser, respectively. Equation (3.6) computes the heat transfer fluid outlet temperature based on the log mean temperature difference of the evaporator heat exchanger. Equation (3.7) represents the rate change of the refrigerant within the evaporator.

The performance parameters used to predict the adsorption chiller's performance are the coefficient of performance (COP) and cooling capacity.

The COP of the chiller is given by [31]:

$$\text{COP} = \frac{\dot{m}_{f,\text{chill}} C_{p,f} \int_0^{t_{\text{cycle}}} (T_{\text{in,chill}} - T_{\text{out,chill}}) dt}{\dot{m}_{f,\text{hot}} C_{p,f} \int_0^{t_{\text{cycle}}} (T_{\text{in,hot}} - T_{\text{out,hot}}) dt} \quad (3.8)$$

The cooling capacity of the refrigerator is calculated by [31]:

$$\dot{Q}_{\text{cooling}} = \dot{m}_{f,\text{chill}} C_{p,f} \int_0^{t_{\text{cycle}}} (T_{\text{in,chill}} - T_{\text{out,chill}}) dt \quad (3.9)$$

3.1.2 Adsorption Isotherms

Adsorption isotherms define the amount of refrigerant adsorbed and the conditions of equilibrium. The updated Freundlich model was used to measure the adsorption isotherms for the silica gel RD-2060-water pair [31].

$$q^* = A(T_{\text{bed}}) * \left(\frac{P_{\text{cond/evap}}}{P_{\text{bed}}} \right)^{B(T_{\text{bed}})} \quad (3.10)$$

$$\text{where } A(T_{\text{bed}}) = A_0 + A_1 * T_{\text{bed}} + A_2 * T_{\text{bed}}^2 + A_3 * T_{\text{bed}}^3 \quad (3.11)$$

$$\text{and } B(T_{\text{bed}}) = B_0 + B_1 * T_{\text{bed}} + B_2 * T_{\text{bed}}^2 + B_3 * T_{\text{bed}}^3 \quad (3.12)$$

where the constants are listed in Table 3.1.

The modified Dubinin-Astakhov (modified D-A) is used to measure the adsorption isotherm for zeolite 13X-water [53].

$$q^* = 0.27 * \exp \left(-5.63 * \left(\frac{T_{\text{bed}}}{T_{\text{cond/evap}}} - 1 \right)^{1.73} \right) \quad (3.13)$$

Langmuir model has is to calculate the adsorption isotherms for HKUST-1 (Cu-BTC (copper benzene-1,3,5-tricarboxylate))-water [44].

$$q^* = \frac{0.6576 * \left(4.548 * \frac{P_{\text{cond/evap}}}{P_{\text{bed}}}\right)}{1 + \left(4.548 * \frac{P_{\text{cond/evap}}}{P_{\text{bed}}}\right)} \quad (3.14)$$

The adsorption isotherms of aluminum fumarate is modelled using polynomial and exponential expressions [45].

$$A(T_{\text{bed}}) = 8.314 * T_{\text{bed}} * \ln\left(\frac{P_{\text{bed}}}{P_{\text{cond/evap}}}\right) \quad (3.15)$$

if $A(T_{\text{bed}}) < 2900$

$$x^* = 0.5948 - 3.12 \times 10^{-4} A(T_{\text{bed}}) + 1.68302 \times 10^{-7} A(T_{\text{bed}})^2 - 4.06 \times 10^{-1} A(T_{\text{bed}})^3 \quad (3.16)$$

if $2900 \leq A(T_{\text{bed}}) \leq 3987$

$$x^* = 2.36129 - 9.93768 \times 10^{-4} A(T_{\text{bed}}) + 1.05709 \times 10^{-7} A(T_{\text{bed}})^2 \quad (3.17)$$

if $A(T_{\text{bed}}) > 3987$ (3.18)

$$x^* = 0.111993 \exp(-0.000258797 * A(T_{\text{bed}}))$$

The Sips model was used to measure the adsorption isotherm for MIL-100(Fe)-water [43].

$$q^* = \frac{0.3571 * \left(3.508 * \frac{P_{\text{cond/evap}}}{P_{\text{bed}}} \right)^{3.2395}}{1 + \left(3.508 * \frac{P_{\text{cond/evap}}}{P_{\text{bed}}} \right)^{3.2395}} \quad (3.19)$$

Table 3-1: Values of constants used in equations 3.11 and 3.12.

Constant	Value	Constant	Value
A₀	-6.5314 kg	B₀	-15.587
A₁	0.072452 kg/K	B₁	0.15915 K ⁻¹
A₂	-0.23951x10 ⁻³ kg/K ²	B₂	-0.50612x10 ⁻³ K ⁻²
A₃	0.25493x10 ⁻⁶ kg/K ³	B₃	-0.5329x10 ⁻⁶ K ⁻³

3.1.3 Adsorption Kinetics

The rate of adsorption is proportional to the difference between the concentration of the equilibrium and the concentration at any instance. The rate of adsorption is calculated using the linear driving force of the kinetic equation.

$$\frac{dq}{dt} = k_1 * \exp\left(-\frac{k_2}{T_{\text{bed}}}\right) * (q^* - q) \quad (3.20)$$

where k_1 and k_2 are constants that depend on the adsorbent-adsorbate pairs. Their values are given in Table 3.2 for different working pairs.

3.1.4 Saturated Vapor Pressure of Water (Antoine's Equation)

The saturated vapor pressure of water is calculated based on Antoine's equation and given as:

$$P = 133.32 * \exp\left(18.3 - \frac{3820}{(T + 273.15) - 46.1}\right) \quad (3.21)$$

Table 3-2: Values of constants used in equation (3.20).

Working pair	k₁ (1/s)	k₂ (K)	Reference No.
RD silica gel -water	7558.02	5051.72	31
Zeolite 13X-water	40.04 X10 ⁻³	905.8	52
MIL-100(Fe)-water	4.208 X10 ⁷	3247.53	43
HKUST-1-water	6.375 X10 ⁷	6002	44
Aluminium fumarate-water	1.29	2168.15	45

3.2 Solution Method of a Two-Bed Adsorption Chiller

MATLAB is used to perform numerical integration using a set of coupled ordinary differential equations of the evaporator, the condenser, and the adsorption/desorption beds. The numerical integration is based on an explicit method, where the dependent variable can be separated. Runge-Kutta method is used with a time step of one second and a tolerance of 10⁻³ with the initial conditions given in Table 3.3. The solution begins with assuming initial temperature, water mass in the evaporator, and water uptakes. The program assumes some initial values; it then carries on an explicit numerical integration to simulate the system's cyclic quasi-steady-state conditions. A flowchart of the computer program is shown in Figure 3.2. The processes that the adsorbent beds undergo in one complete cycle is shown in Table 3.4.

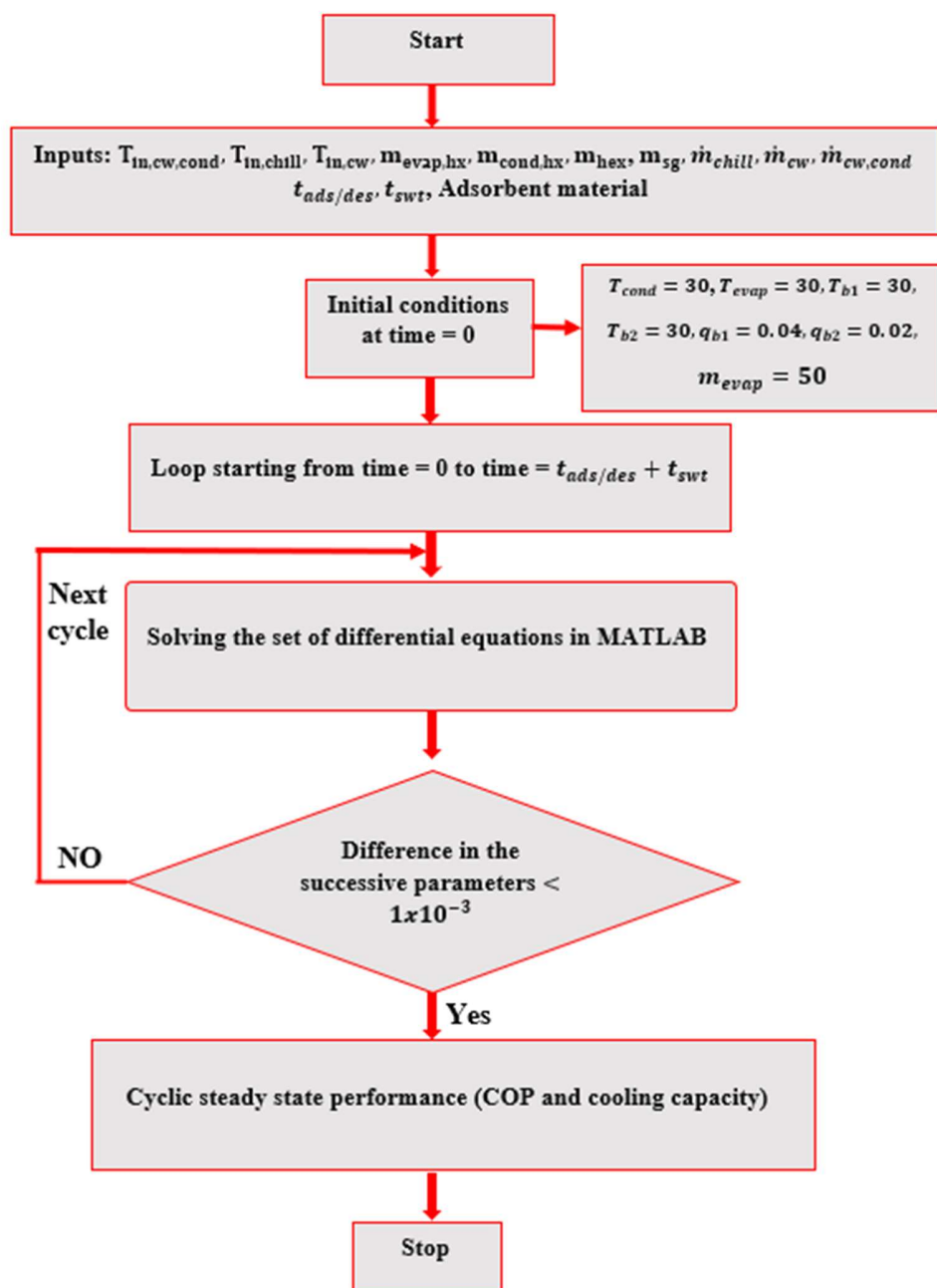


Figure 3-2: Flowchart of the computer program for a Two-bed adsorption chiller.

Table 3-3: Values of the initial conditions.

Symbol	Value	Unit
T_{cond}	30	$^{\circ}\text{C}$
T_{evap}	30	$^{\circ}\text{C}$
T_{b1}	30	$^{\circ}\text{C}$
T_{b2}	30	$^{\circ}\text{C}$
q_{b1}	0.04	$\text{kg}_{\text{adsorbed}}/\text{kg}_{\text{adsorbent}}$
q_{b2}	0.02	$\text{kg}_{\text{adsorbed}}/\text{kg}_{\text{adsorbent}}$
m_{evap}	50	kg

Table 3-4: The processes that the adsorbent beds undergo in one complete cycle.

Bed 1	Desorption	Pre-cooling	Adsorption	Pre-heating
Bed 2	Adsorption	Pre-heating	Desorption	Pre-cooling

3.3 Simulation Results and Discussion of a Two-Bed Adsorption

Chiller

The results of the present work are validated against the previous work reported by Saha et al. [31]. The findings of the current study indicate a very strong agreement with Saha et al. [31] as seen in Figure 4, where the maximum percentage difference is 2.4%.

A parametric analysis is carried out to study the impact of operating conditions on the performance of the cooling system in terms of the coefficient of performance (COP) and the cooling capacity of the different adsorbent-adsorbate pairs.

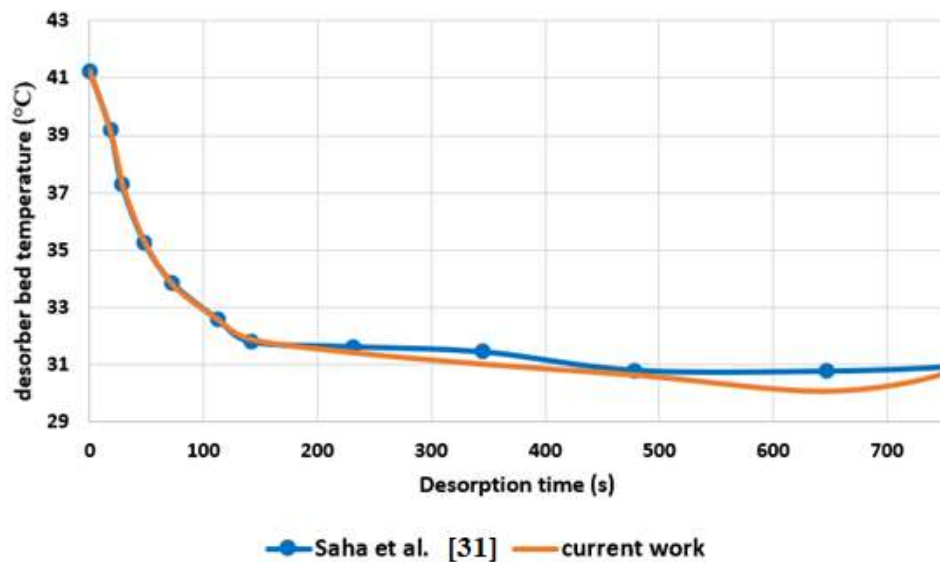


Figure 3-3: Validation of the current work.

3.3.1 The Transient Temperature Profile of The Main Heat Exchanger Components of a Two-Bed Adsorption Chiller

The transient temperature curves of the evaporator, condenser, adsorber, and desorber heat exchangers with the operating conditions defined in Table 3.5 are shown in Figure 3.4. RD silica gel-water and HKUST-1-water are used as working pairs. The cooling of the adsorption bed is slower and more complex than the heating of the desorption bed. As a result, the temperature of the bed at the end of the desorption process is almost reaching the temperature of hot water. However, the temperature of the bed in the adsorption process stays within 3 or 4 °C above the temperature of the cooling water. The condenser temperature reaches its highest at the beginning of each cycle due to an increase in the

temperature of the cooling water. The evaporator temperature reaches its peak at the end of each cycle due to an increase in the temperature of the chilled water.

The temperature profile of the heat transfer fluid at the outlet section is shown in Figure 3.5. At the end of the adsorption/desorption period, it is shown that the temperature of the hot water outlet approaches the temperature of the hot water inlet. In contrast, the heat transfer fluid's outlet temperature, which is used to cool the adsorber bed, remains within two-Celsius degrees higher than its inlet temperature. During the precooling/preheating process, between 800 and 830 seconds, the temperature of the hot water outlet and the adsorber cooling water outlet temperatures change rapidly due to switching from adsorber to desorber and vice versa. The temperature of the chilled water outlet drops significantly at the start of the adsorption process since the adsorber is dry, which means that there is no adsorbate within the bed, and the rate of adsorption is high. The temperature of the condenser cooling water outlet rises dramatically at the beginning of the desorption process because the desorber is saturated and the rate of desorption is high. HKUST-1 has a lower chilled outlet temperature due to higher adsorption rates, resulting in higher evaporation rates. The lower temperature results in a higher cooling effect.

Table 3-5: Physical property and operating conditions values.

Symbol	Value	Unit
$t_{ads/des}$	800	S
t_{swt}	30	S
$T_{in,hot}$	85	°C
$T_{in,cw}$	25	°C
$T_{in,cw,cond}$	25	°C
$T_{in,chill}$	14	°C
$\dot{m}_{hw}$	1.3	Kg/s
$\dot{m}_{cw}$	1.3	Kg/s
$\dot{m}_{chill}$	0.7	Kg/s
$\dot{m}_{cw,cond}$	1.3	Kg/s
M_{adt}	47	kg
$M_{evap,hx}C_{p,evap,hx}$	4805.7	J/K
$M_{cond,hx}C_{p,cond,hx}$	9372.08	J/kg

$M_{\text{hex}}C_{p,\text{hex}}$	7719.4	J/K
$C_{p,\text{sg}}$	924	J/kg/K
$C_{p,\text{w}}$	4180	J/kg/K
$C_{p,\text{f}}$	4180	J/kg/K
$C_{p,\text{v}}$	1890	J/kg/K
$C_{p,\text{cw,cond}}$	4180	J/kg/K
$C_{p,\text{cw}}$	4180	J/kg/K
$(UA)_{\text{ads}}$	3500	W/K
$(UA)_{\text{des}}$	3500	W/K
$(UA)_{\text{cond}}$	15330	W/K
$(UA)_{\text{evap}}$	4870	W/K
Q_{st}	2.8×10^6	J/kg
h_{fg}	2.5×10^6	J/kg

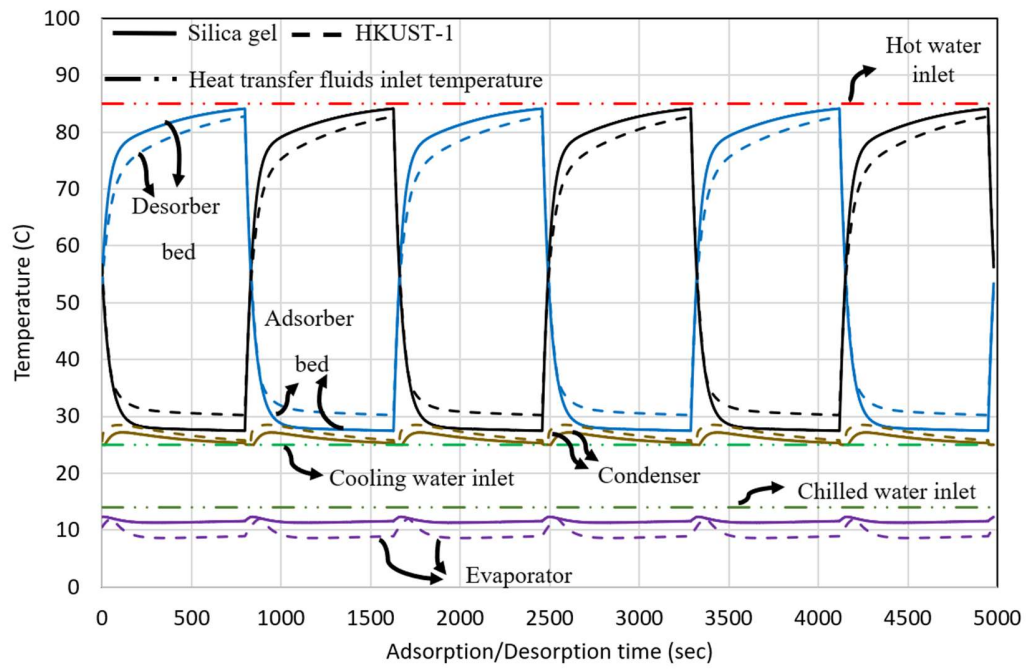


Figure 3-4: Transient temperature profile of the major components heat exchangers (RD silica gel -water and HKUST-1-water).

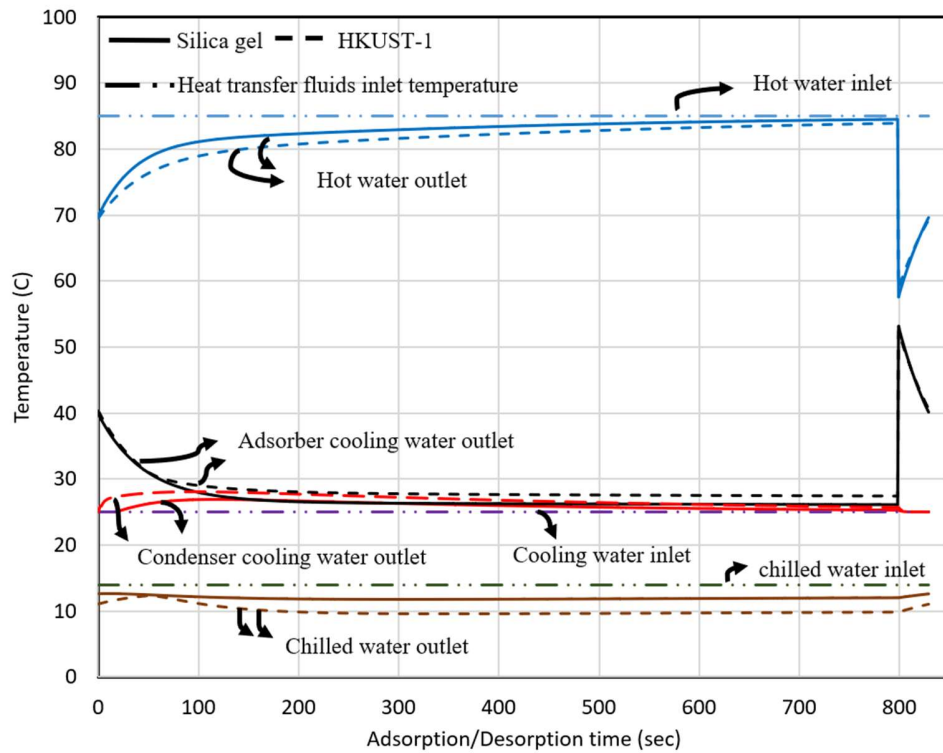


Figure 3-5: The outlet temperature profile of the heat transfer fluids (RD silica gel -water and HKUST-1-water).

3.3.2 The Impact of The Operating Conditions on The Performance of The Adsorption Cooling System

3.3.2.1 Impact of operating temperatures

Figure 3.6 reveals the influence of hot water inlet temperature on the system's COP. As the hot water inlet temperature increases, the silica gel's COP and aluminum fumarate's COP decrease gradually in the temperature range between 60 °C and 70 °C. Then, it drops sharply. The main reason for that is a higher driving temperature increases the heating power and irreversible losses. Moreover, the heating power is no more effective and becomes relatively higher than the cooling effect generated. As the hot water inlet temperature rises, the COP of HKUST-1 and MIL100(Fe) increases until it reaches a peak value, then it starts to decrease. The reason is that when heat source temperature is higher than 70 °C for MIL100(Fe) and 85 °C for HKUST-1, desorption energy is no more effective. The zeolite's COP increases as the hot water inlet temperature increases. The rise in driving temperatures, which increase the desorbed refrigerant rate, increases the cooling capacity. A higher desorption rate would guarantee a higher refrigerant recirculation and hence increases the cooling effect. As a result, as hot water inlet temperature increases, the system's cooling effect rises for all the adsorbent materials, as shown in Figure 3.7.

The MOFs are less hydrophilic than silica gel and zeolite. As a result, they desorbed higher water vapor than the silica gel and zeolite at the same partial pressure. Moreover, MOFs have a higher Brunauer Emmett–Teller (BET) surface area. Thus, MOFs perform better due to the higher desorption rate. The most significant parameter affecting the system performance is the adsorbent water vapor uptake. The adsorption cooling process occurs at low partial pressure (supplied vapor pressure/ saturation pressure). Thus, the material

with higher water vapor uptake at low relative pressure will perform better. For the silica gel, most of the water vapor uptake accrues at a high partial pressure ratio. Thus, the silica gel system has a lower performance. The water vapor uptake at low partial pressure was highest for HKUST-1 and lowest for silica gel [43,49]. A higher water vapor uptake results in a higher evaporation rate causing a higher cooling effect higher refrigerant recirculation rate. Figures 5 and 6 show the impact of different adsorbent materials on the system's performance. The HKUST-1 has the highest COP and cooling capacity due to the high amount of water vapor adsorbed by HKUST-1. Table 3.6 shows the percentage of improvement in the COP and cooling capacity (CC) at a hot water inlet temperature of 85 °C in comparison to silica gel RD 2060. As the HKUST-1 used instead of silica gel RD 2060, the COP would be improved by more than 43% and the cooling capacity by more than 75% in this case.

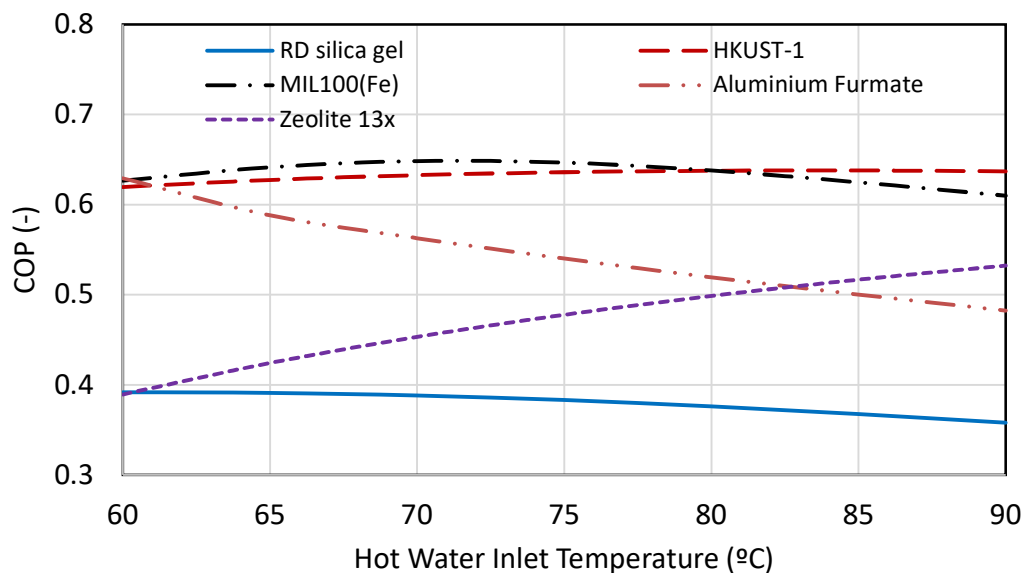


Figure 3-6: The impact of the hot water inlet temperature on the system's COP.

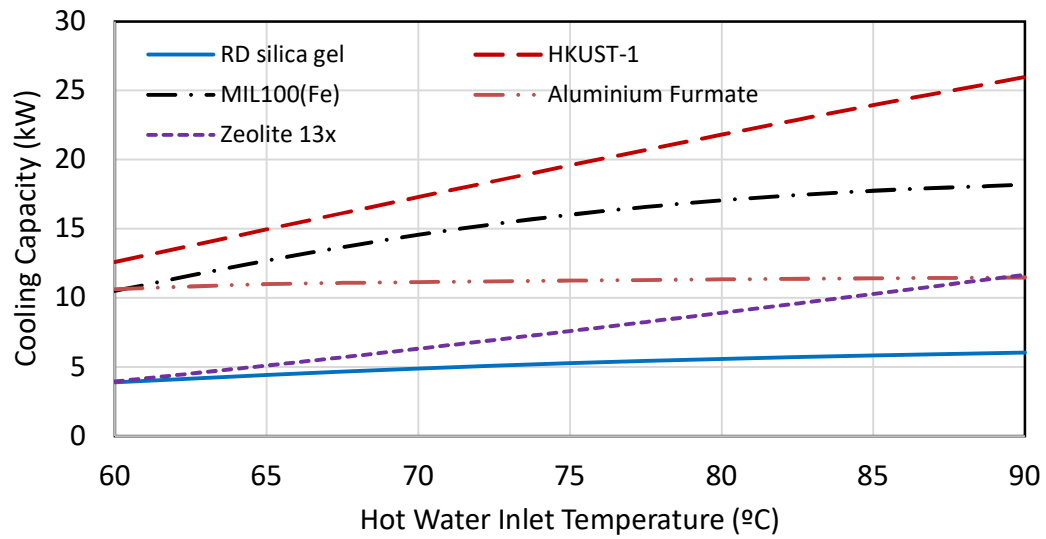


Figure 3-7: The impact of the hot water inlet temperature on the system's cooling capacity.

Table 3-6: Percentage Improvement in COP due using various pairs.

Adsorbent material	$\frac{COP_i - COP_{RD2060}}{COP_i} * 100$	$\frac{CC_i - CC_{RD2060}}{CC_i} * 100$
Zeolite 13X	28.6 %	43 %
MIL100(Fe)	41.6 %	67.3 %
HKUST-1	43 %	75.8 %
Aluminium Fumarate	26.7 %	48.8 %

Figures 3.8 and 3.9 show the chilled water inlet temperature's influence on the system's COP and cooling capacity for all adsorbent materials. As the refrigerated water inlet temperature increases, both COP and cooling capacity increase. This behavior happens due to the higher rate of evaporation of refrigerant in the evaporator. Zeolite 13X performed

better than the aluminum fumarate at chilled water inlet temperature lower than 12 °C. This is because the water uptake at a very low partial pressure of zeolite 13X higher than the aluminum fumarate.

Figures 3.10 and 3.11 show the influence of the cooling water's inlet temperature on the system's COP and cooling capacity. Raising the cooling water inlet temperature leads to a decrease in the COP and cooling capacity due to the lower amount of refrigerant adsorbed and slower condensation of the desorbed refrigerant.

The effects of operating temperatures on different adsorbent materials cooling capacities are shown in Table 3.7. The most influential operating temperatures are the chilled and hot water inlet temperatures. The effect of hot water inlet temperature (regeneration temperature) is more pronounced in the zeolite 13X due to the higher water uptake. The impact of chilled water inlet temperature is more cleared in aluminum fumarate.

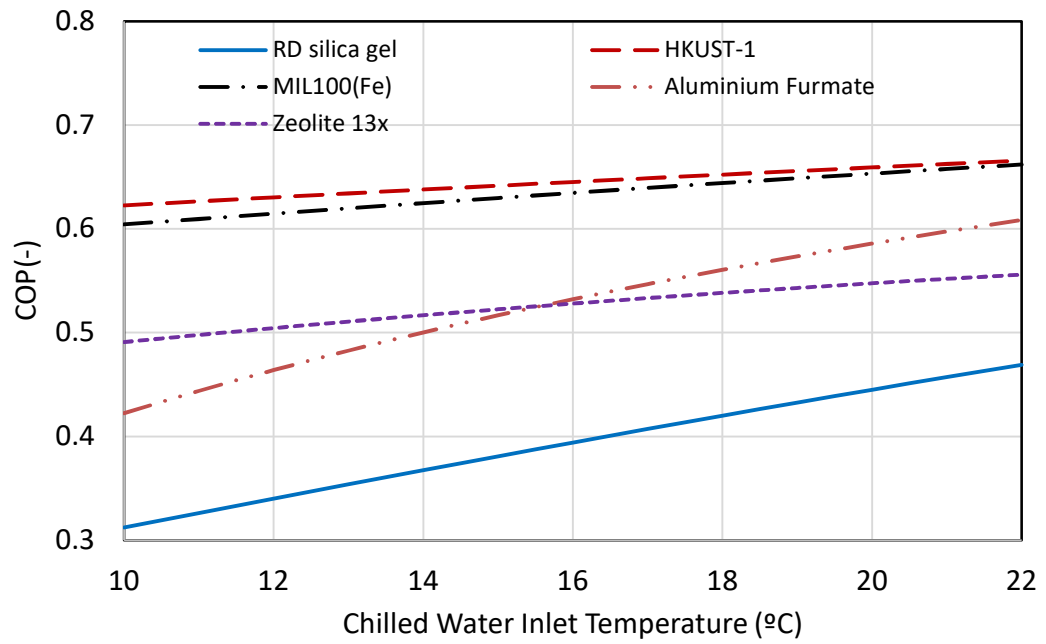


Figure 3-8: The impact of the chilled water inlet temperature on the system's COP.

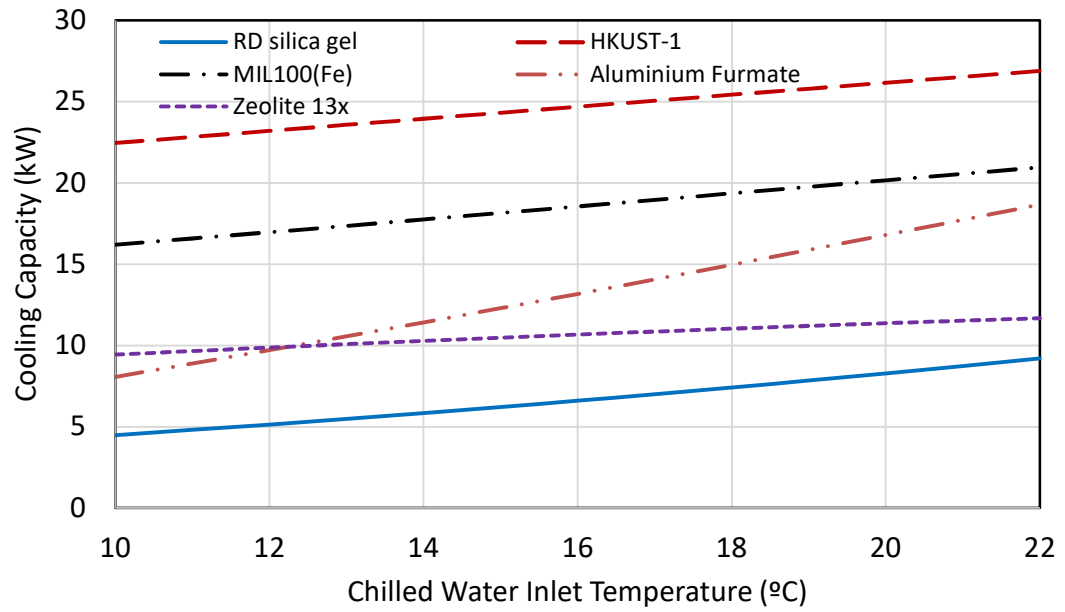


Figure 3-9: The impact of the chilled water inlet temperature on the system's cooling capacity.

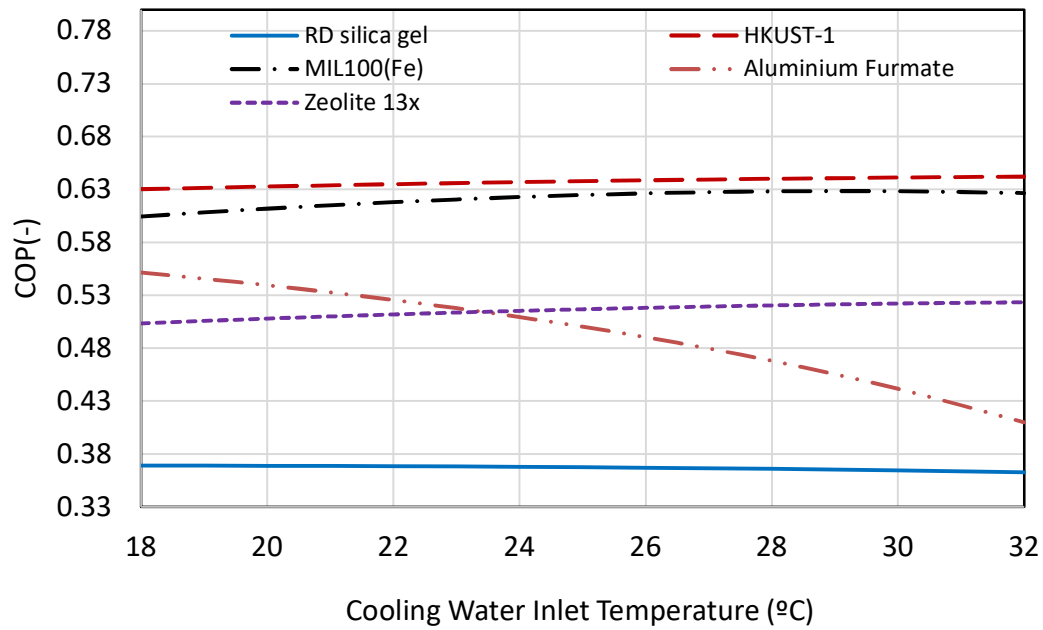


Figure 3-10: The impact of the cooling water inlet temperature on the system's COP.

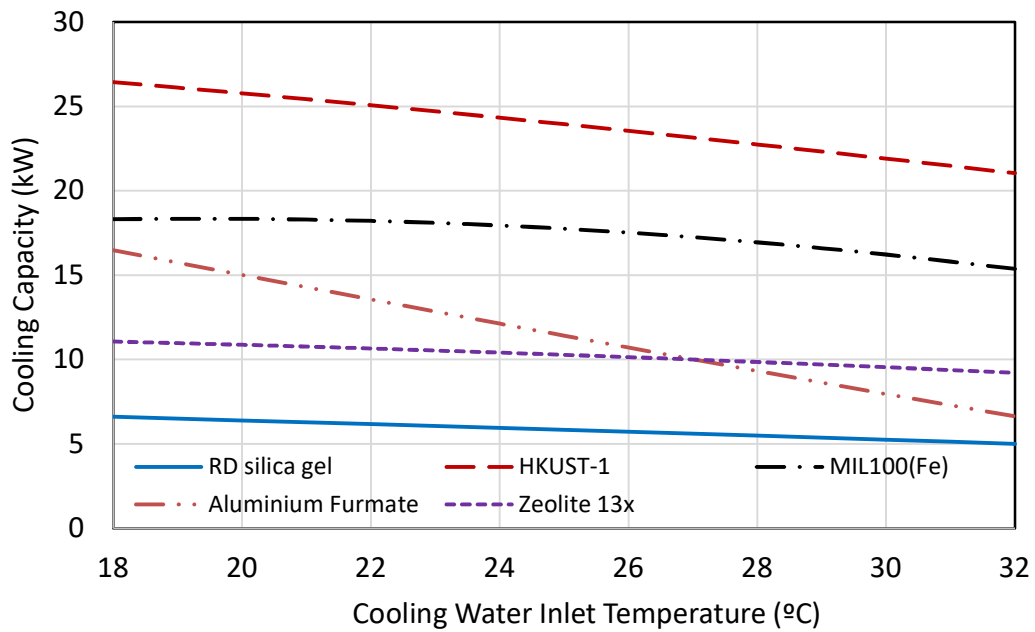


Figure 3-11: The impact of the cooling water inlet temperature on the system's cooling capacity.

Table 3-7: Impact of operating temperatures on the cooling capacity.

Increasing the temperature (°C) (From- To)	Increases/ decreases	The cooling capacity change (%)				
		Silica gel	Zeolite	HKUST-1	MIL-100 (Fe)	Fumarate
Hot water (60-90)	Increases	55	224	106	73	8
Chilled water (12-22)	Increases	105	24	20	29	130
Cooling water (18-32)	Decreases	22	17	23	15	60

3.3.2.2 Impact of heat transfer fluid mass flow rates

Figures 3.12 and 3.13 show the impact of the chilled water flow rate on the system's COP and cooling capacity. COP and cooling capacity increase sharply as the mass flow rate of heat transfer fluid increases in a range of 0.2 kg/s to 0.8 kg/s due to a higher heat transfer rate. Then, the increase is insignificant because the temperature difference between the chilled water inlet and outlet temperature reduces slowly at a higher chilled water flow rate. The effect of chilled water mass flow rate is more pronounced in the aluminum fumarate. However, the impact of the operating temperatures is more significant than the operating flow rates.

Figures 3.14 and 3.15 show that COP and cooling capability increase slightly as the rate of hot water flow increases. As this flow rate increases, the amount of refrigerant desorbed also increases, while the heating power also increases.

Figure 3.16 shows the impact of the flow rate of adsorber cooling water on the performance of the silica gel-water system. Increasing the cold-water flow rate of the adsorption bed

increases the COP and cooling power due to higher refrigerant adsorption rates. This rise, however, is negligible.

The impact of operating mass flow rates on various adsorbent cooling capacity materials are shown in Table 3.8. The cooled water flow rate is the most significant heat transfer fluid mass flow rate. The effect of cooled water mass flow is more pronounced in the aluminum fumarate. However, the effect of operating temperatures is more important than the operating flow rate.

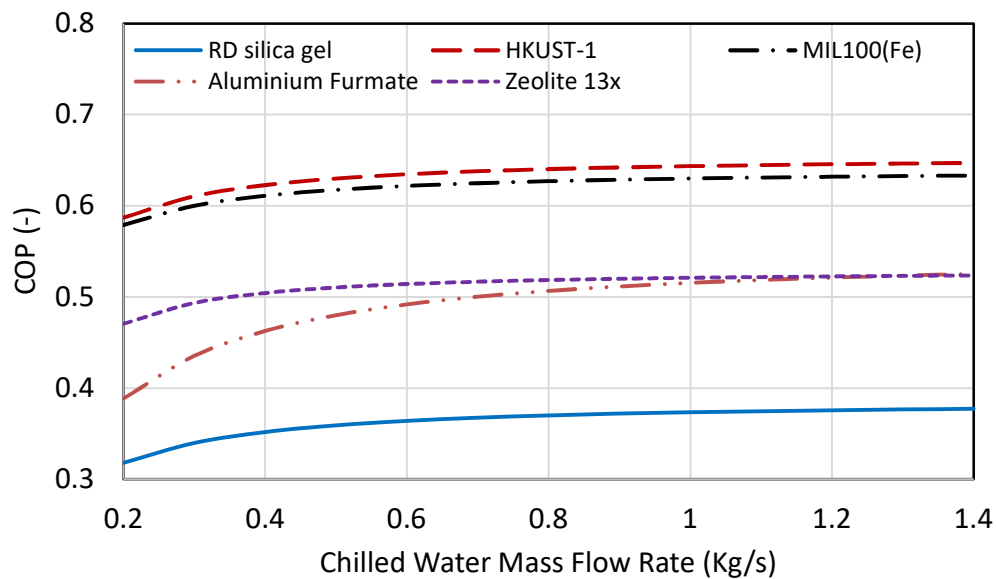


Figure 3-12: The impact of the chilled water flow rate on the system's COP.

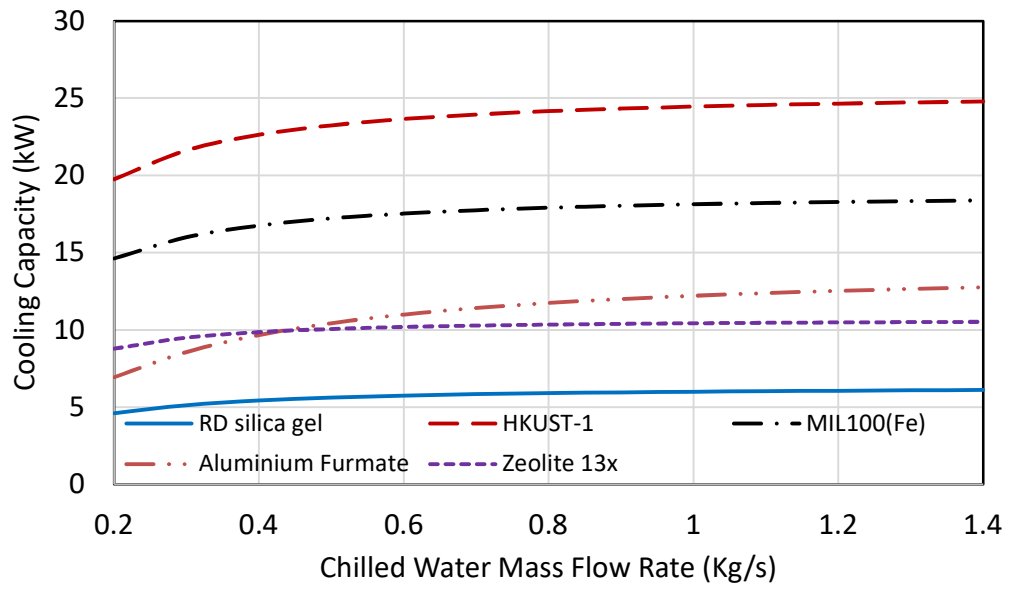


Figure 3-13: The impact of the chilled water flow rate on the system's cooling capacity.

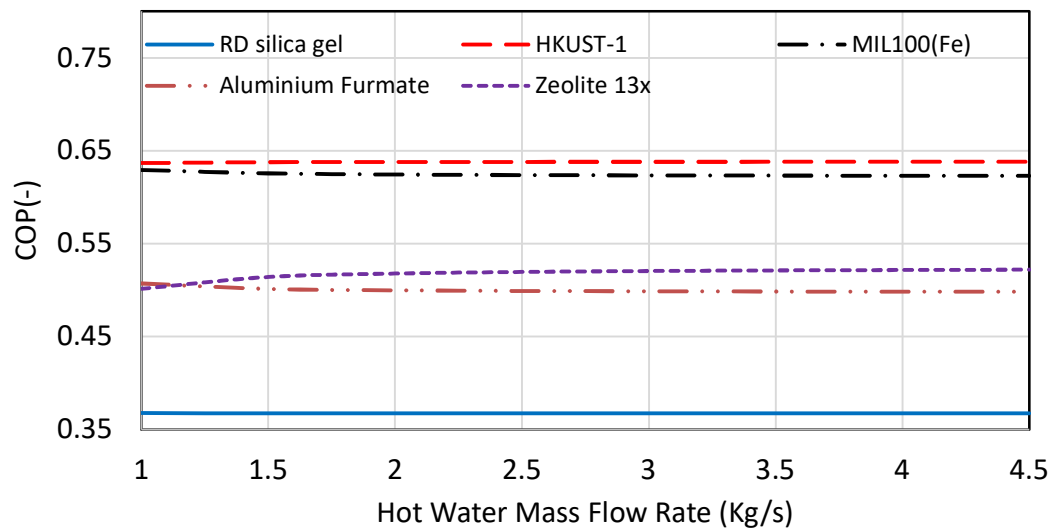


Figure 3-14: The impact of the hot water flow rate on the system's COP.

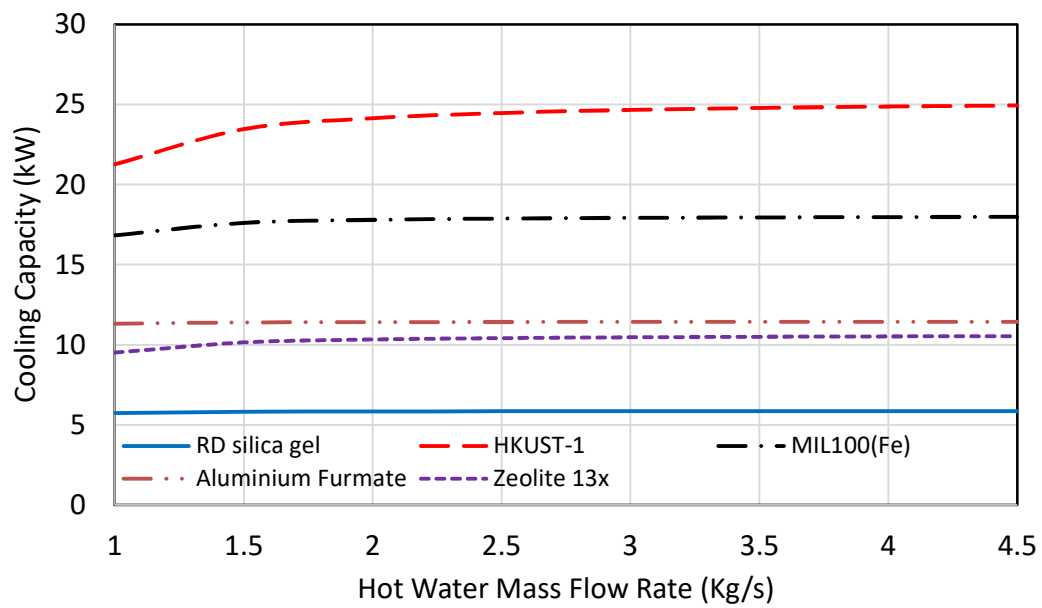


Figure 3-15: The impact of the hot water flow rate on the system's cooling capacity.

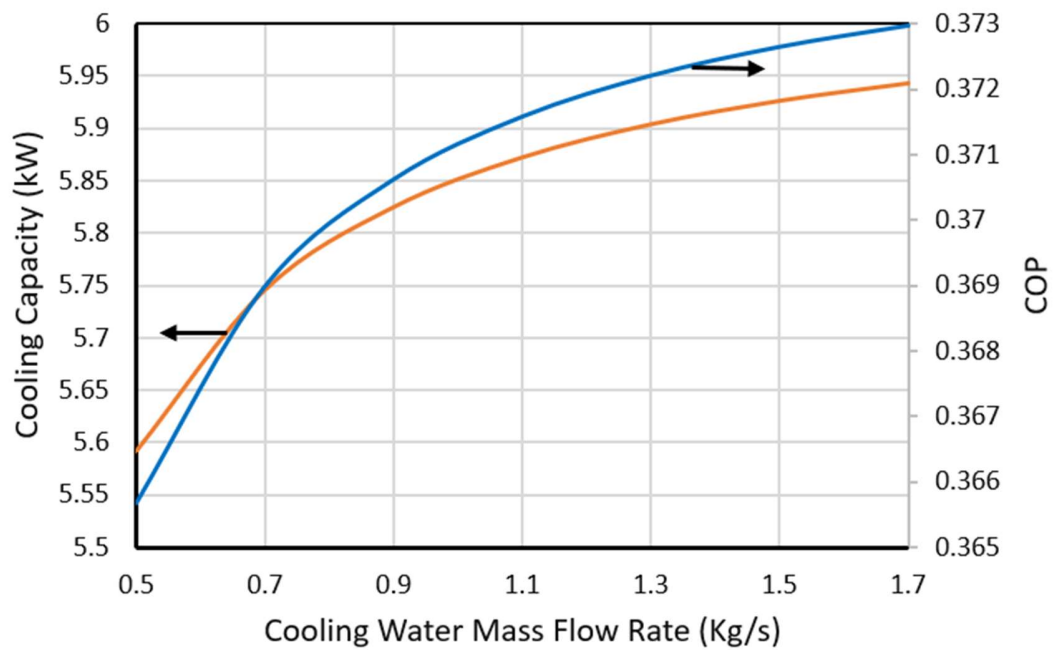


Figure 3-16: The impact of the adsorber cooling water flow rate on the system's performance (RD silica gel - water).

Table 3-8: Impact of operating mass flow rates on the cooling capacity.

Increasing the mass flow rate (kg/s) (From- To)	Increases/ decreases	The cooling capacity change (%)				
		Silica gel	Zeolite	HKUST-1	MIL- 100 (Fe)	Fumarate
Hot water (1-4.5)	Increases	2	11	17	7	1
Chilled water (0.2-1.4)	Increases	32	20	25	26	84

3.3.2.3 Impact of adsorbent mass

Figure 3.17 reveals that as the adsorbent mass increases from 20 kg to 80 kg, the COP rises sharply. This rise is because more refrigerant is adsorbed and desorbed, which produces a higher cooling effect. A more rise in adsorbent mass results in a minor improvement in COP because the adsorber bed energy consumed does not contribute to the desorption but only increases its component's sensible heat. Figure 3.18 shows that the specific cooling power decreases as the mass of silica gel increases due to the heat transfer's slowdown, and accordingly, desorption would be incomplete.

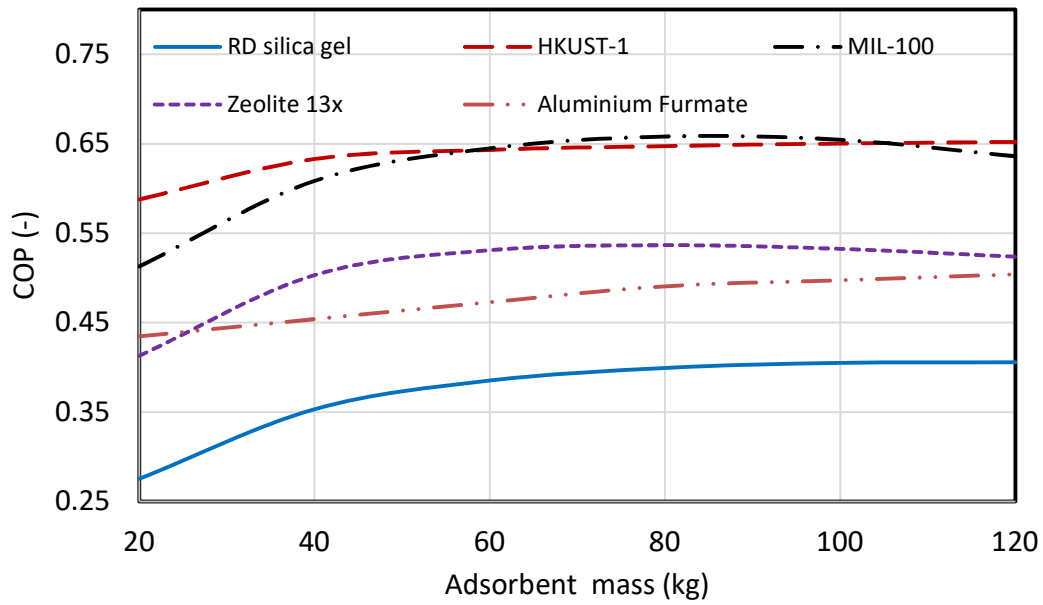


Figure 3-17: The impact of adsorbent mass on the system's COP.

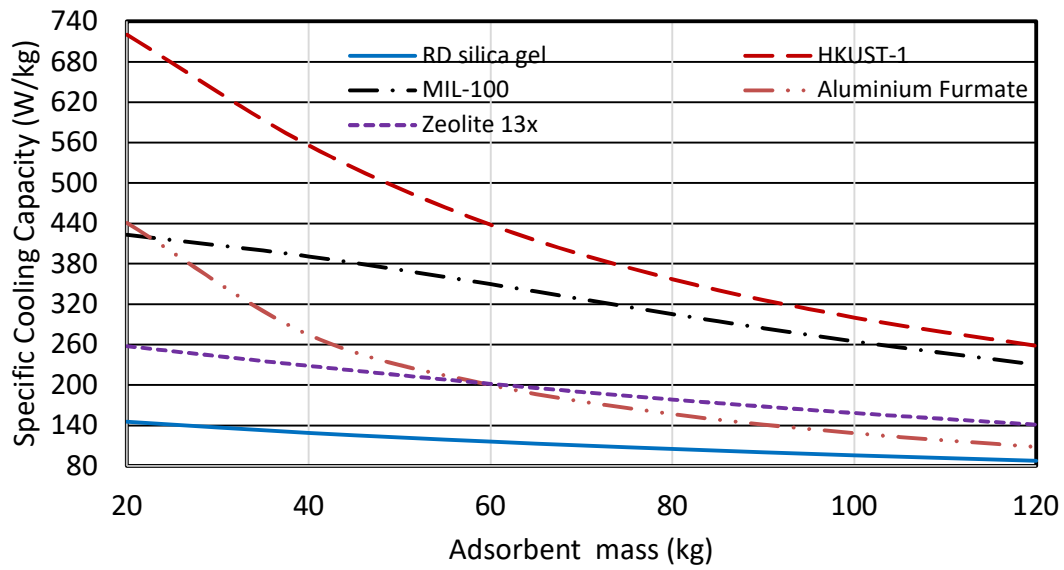


Figure 3-18: The impact of adsorbent mass on the system's specific cooling capacity.

3.3.2.4 Impact of adsorption/desorption and switching time

Figures 3.19 and 3.20 show the impact of adsorption/desorption time on the system's performance. The cooling capacity rises as the adsorption/desorption time rises until it reaches the highest value at an optimal adsorption/desorption time. Then, it starts to decrease. When the adsorption/desorption time is shorter, time isn't sufficient for desorption and adsorption to happen completely. However, when the time is longer, the rate of adsorption decreases. The COP increases as the adsorption/desorption time increases due to reducing the cycle's irreversible losses and the heat source consumption.

Figures 3.21 and 3.22 show the effect of switching time (pre-cooling/pre-heating time) on system performance. The cooling ability increases as the switching time increases until the maximum value is reached at an optimum switching time. After that it's beginning to decline. If the switching time is shorter, there is not enough time to cool the hot bed enough. In addition, longer time is not favorable because the pressure of the adsorber exceeds the pressure of the evaporator. This decreases the cooling power as the full adsorption takes place at the pressure of the evaporator.

The optimal adsorption/desorption and switching times for adsorbent materials are shown in Table 3.9. The effects of operating times on the different cooling capacity of adsorbent materials are shown in Table 3.10. It is clear that the most influential operation is adsorption/desorption time. The effect of adsorption/desorption time in the MIL-100(Fe) is more pronounced.

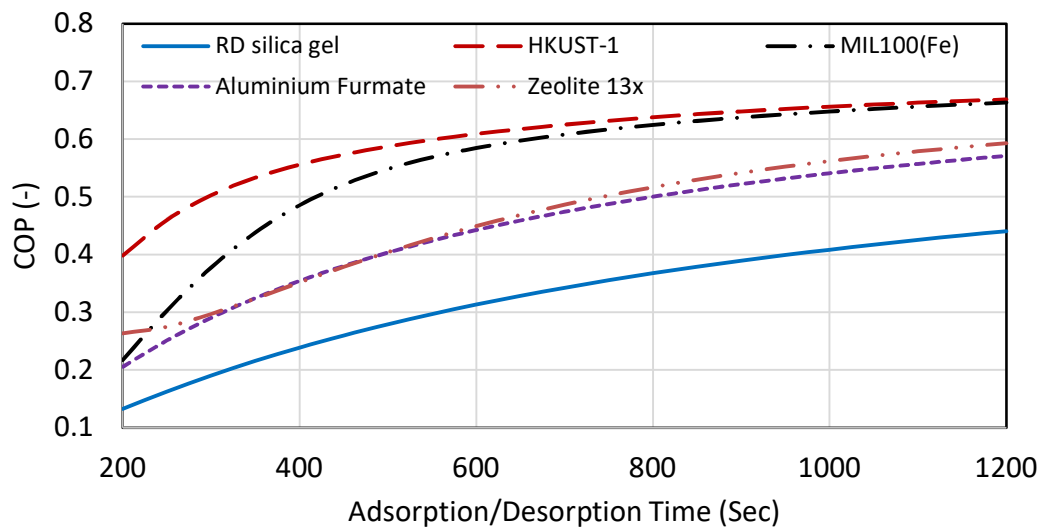


Figure 3-19: The impact of adsorption/desorption time on the system's COP.

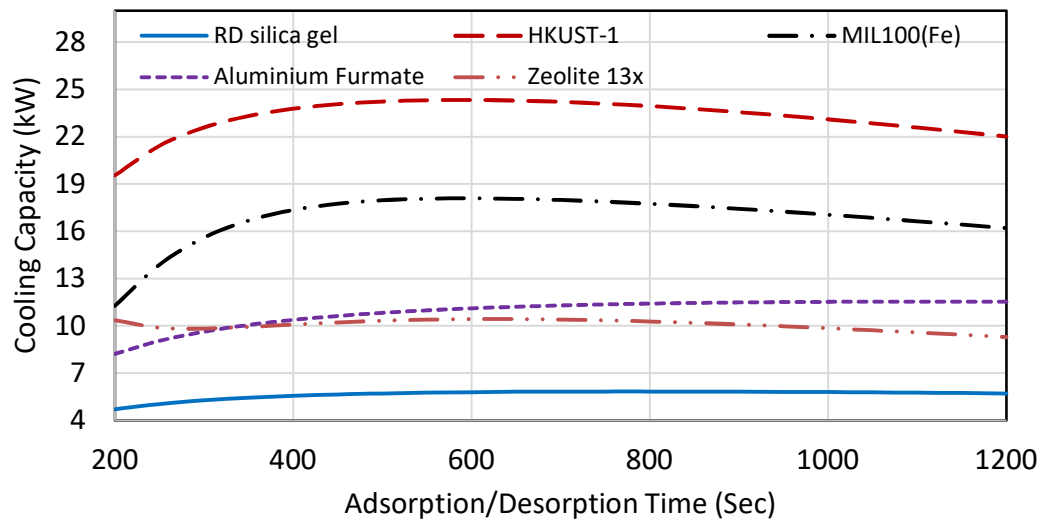


Figure 3-20: The impact of adsorption/desorption time on the system's cooling capacity.

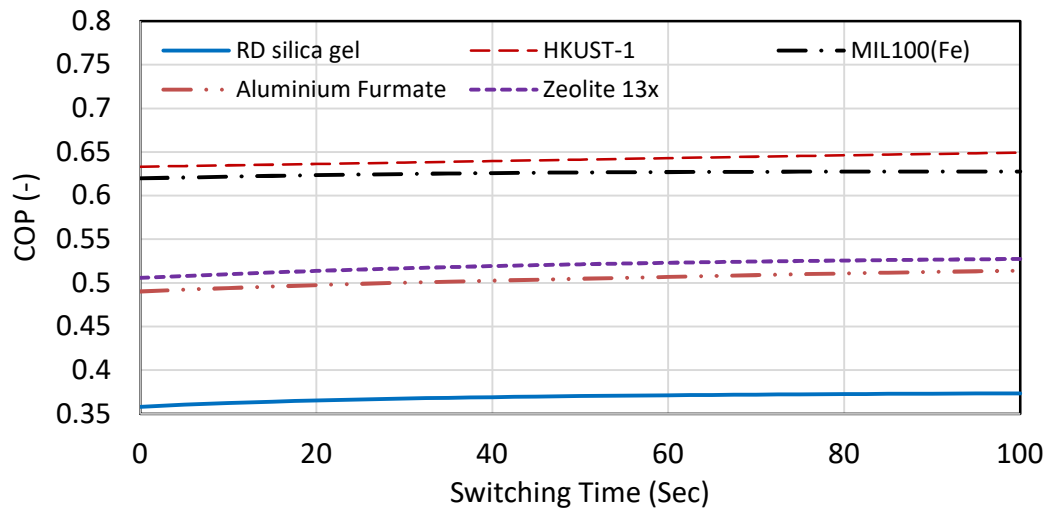


Figure 3-21: The impact of switching time on the system's COP.

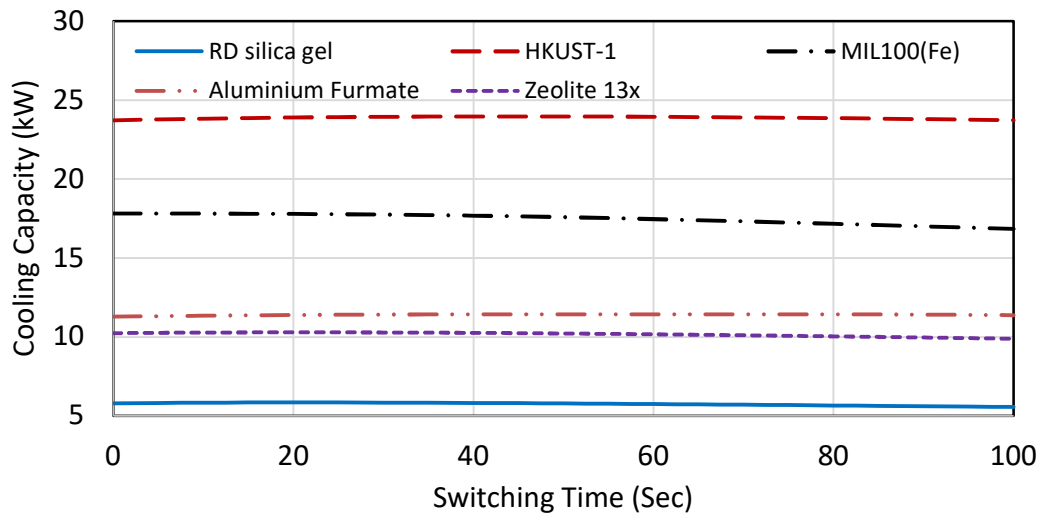


Figure 3-22: The impact of switching time on the system's cooling capacity.

Table 3-9: The optimal adsorption/desorption and switching times for adsorbent materials.

Adsorbent material	The optimal adsorption/desorption time (Seconds)	The optimal switching time (Seconds)
Silica gel	800	25

HKUST-1	600	45
Aluminum fumarate	900	50
Zeolite 13X	650	30
MIL100(Fe)	700	20

Table 3-10: Impact of operating times on the cooling capacity.

Increasing the time (sec) (From- To)	Increases/ decreases	The cooling capacity change (%)				
		Silica gel	Zeolite	HKUST- 1	MIL- 100(Fe)	Fumarate
Adsorption/desorption time (200-1200)	Increases	21	10	13	43	40
Switching time (0-100)	Decreases	4	3	0.008	0.8	0.5

CHAPTER 4

PARALLEL FEED MULTI-EFFECT DESALINATION SYSTEM (PF-MED)

The multiple-effect desalination (MED) system utilizes the thermal energy to produce fresh water from the seawater. In the MED system, the vapor formed in one effect is used as a heat source for the next effect. The vapor reuse results in the reduction of primary energy consumption. Moreover, this system can operate with low-grade waste heat and solar energy (low-temperature heat source). The MED system is the most thermodynamically efficient and economical in terms of power consumption in thermal desalination processes. Moreover, it can be hybridized with another system (e.g., adsorption system or absorption system) [7].

The temperature of the first effect is called the top brine temperature (TBT) and the temperature of the last effect is called the temperature of the lower brine (LBT). The maximum gap for the operation of the conventional MED system is the temperature difference between the TBT and LBT. This gap represents the number of vapor heat recoveries in terms of the number of effects, which affect the system's performance. The MED TBT's maximum value is around 70 C due to the scaling, fouling effect. The MED LBT is controlled by the cooling water condenser, which is at ambient temperature. The minimum values of the MED LBT should be higher than the ambient temperature so that in the condenser, the vapor created by the last impact can be condensed.

4.1 Parallel Feed Multi-Effect Desalination System (PF-MED)

In the parallel feed MED system, the vapor flows from the first effect to the last effect in the direction of pressure decrease. The feed water flows from the main header to each effect in a perpendicular direction, as shown in Figure 4.1. The feed water flows in all effects with the same temperature and salinity. Then, its temperature is increased to the saturation temperature inside each effect. This process is followed by evaporation and an increase in salinity of rejected brine. The MED system used in this study uses the horizontal falling film tubes distinguished by their ability to handle seawater scaling.

The seawater intake flows into the down condenser, whereby absorbing the latent heat of condensation of the vapor produced in the last effect, its temperature increases to the feed temperature. Back to the sea, part of the warm seawater called cooling water is rejected. The cooling water is used to eliminate the excess heat applied to the system. The feed seawater is sprayed onto the horizontal tubes' outer side. The feed seawater temperature increases from the feedwater temperature to the boiling temperature corresponding to the vapor space pressure before a small portion of water vapor is formed. The heat required for preheating and evaporation is provided by condensing the steam in the first effect and the formed vapor in the previous effect for the remaining effects. Saturated steam is supplied to the system from an external boiler.

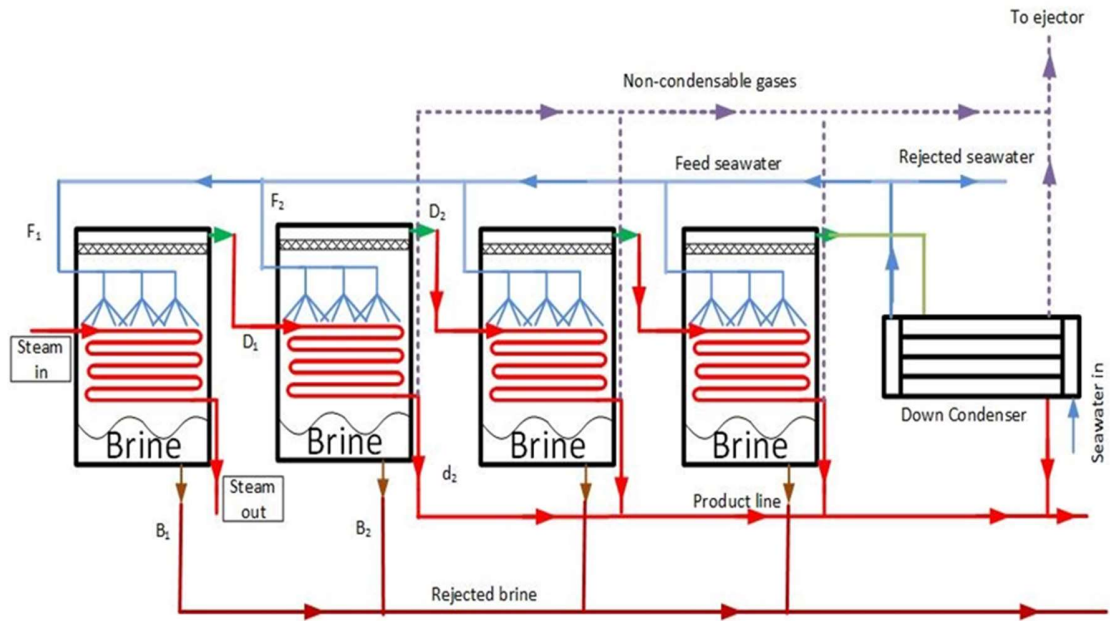


Figure 4-1: PF-MED system.

4.2 Mathematical Modeling of Parallel Feed Multi-Effect

Desalination System (PF-MED)

The transient model of the PF-MED system will be discussed in the next chapter.

In this chapter, the steady-state model of the PF-MED system will be discussed in detail.

4.2.1 Steady State Model

The steady-state mathematical model was developed for the PF-MED with the following assumptions:

- Steady-state operating conditions.
- Neglecting the depression in temperature corresponding to the decrease in pressure in the demister, vapor transmission lines and during the period of condensation.

- The feed water is distributed among the effects equally.
- The salinity of the vapor is zero (does not contain salt).
- No loss of energy between effects and surrounding areas.
- For all effects, the brine temperature difference between effects is set to equal.

Material and salt balance equations

- **Feed seawater flow rate in each effect i**

$$F_i = \frac{\dot{m}_f}{n} \quad (4.1)$$

Where n the number of effects.

- **Total mass flow rate balance**

$$\dot{m}_f = \dot{m}_b + \dot{m}_v \quad (4.2)$$

Where f: feed seawater, b: rejected brine, v: formed vapor (distillate).

- **Mass flow rate balance in effect i**

$$F_i = B_i + D_i \quad (4.3)$$

- **Total salt balance**

$$x_f \dot{m}_f = x_b \dot{m}_b + x_v \dot{m}_v \quad (4.4)$$

Where $\dot{m}$ the mass flow rate and x the salinity of the water, where $x_v = 0$.

- **Salt balance in effect i**

$$x_f F_i = B_i x_{b,i} \quad (4.5)$$

- **Total distilled water**

$$\dot{m}_v = \sum_{i=1}^n D_i \quad (4.6)$$

Energy equations

- **First Effect**

$$\dot{m}_s h_{fg,s} = F_1 c_{p,f} (T_1 - T_f) + D_1 h_{fg,v,1} \quad (4.7)$$

By condensing the saturated steam coming from an external boiler, the heat applied to the first effect is equal to the heat obtained by the feed seawater to boost its temperature from the feed temperature to the boiling temperature of the brine plus the amount of heat required to create vapor within the effect.

Evaporator heat transfer area in effect 1:

$$\dot{m}_s h_{fg,s} = U_1 A_1 (T_s - T_1) \quad (4.8)$$

- **Effect i >1**

$$D_{i-1} h_{fg,v,i-1} = F_i c_{p,f} (T_i - T_f) + D_i h_{fg,v,i} \quad (4.9)$$

In addition to the amount of heat needed to produce the vapor inside the effect, the heat added to the effect by condensing the vapor generated in the previous effect is in thermal equilibrium with the heat obtained by the feed seawater to increase its temperature from the feed temperature to the boiling temperature of the brine.

Evaporator heat transfer area in effect i:

$$D_{i-1} h_{fg,v,i-1} = U_i A_i (T_{v,i-1} - T_i) \quad (4.10)$$

- **The down condenser**

$$D_n h_{fg,v,n} = (\dot{m}_f + \dot{m}_{cw}) c_{p,f} (T_f - T_{cw}) \quad (4.11)$$

The heat added to the effect by condensing the vapor created in the last effect equals the heat gained by the seawater intake to boost its temperature from the temperature of the cooling water to the temperature of the feed.

Condenser heat transfer area:

$$D_n h_{fg,v,n} = U_c A_c (LMTD)_c \quad (4.12)$$

Where log mean temperature difference given as:

$$(LMTD)_c = \frac{(T_f - T_{cw})}{\ln \left(\frac{(T_{v,n} - T_{cw})}{(T_{v,n} - T_f)} \right)} \quad (4.13)$$

- **The brine temperature difference between effects**

$$\Delta T_i = \frac{T_s - T_n}{n} \quad (4.14)$$

- **The brine temperature of the first effect**

$$T_1 = T_s - \Delta T_i \quad (4.15)$$

- **The brine temperature of effect $i > 1$**

$$T_i = T_{i-1} - \Delta T_i \quad (4.16)$$

- **Vapor temperature in effect i**

$$T_{v,i} = T_i - BPE_i \quad (4.17)$$

Where BPE the boiling point elevation.

4.2.2 The Water and Steam Properties

The steam's latent heat is given as a function of the temperature of the steam, and the vapor's latent heat is given as a function of the temperature of the vapor. As a function of the brine boiling temperature, the overall heat transfer coefficient for the evaporators is given, and the overall heat transfer coefficient is given as a function of the vapor temperature of the last effect. As a function of the brine temperature and concentration of the brine salt, the boiling point elevation is given.

- **Seawater specific heat at a constant pressure [7]**

$$c_p = A_1 + B_1 * T + C_1 * T^2 + D_1 * T^3 \quad (4.18)$$

Where

$$A_1 = 4206.8 - 6.6197 * s + 1.2288 \times 10^{-2} * s^2 \quad (4.19)$$

$$B_1 = -1.2162 + 5.4178 \times 10^{-2} * s - 2.2719 \times 10^{-4} * s^2 \quad (4.20)$$

$$C_1 = 1.2026 \times 10^{-2} - 5.3566 \times 10^{-4} * s + 1.8906 \times 10^{-6} * s^2 \quad (4.21)$$

$$D_1 = 6.8777 \times 10^{-7} + 1.517 \times 10^{-6} * s - 4.4268 \times 10^{-9} * s^2 \quad (4.22)$$

Where c_p in $\frac{J}{kg \cdot ^\circ C}$, T in $^\circ C$ and s water salinity in gm/kg.

- **Enthalpy of saturated liquid water [7]**

$$h_f = -0.033635409 + 4.2057557011 * T - 6.200339 \times 10^{-4} * T^2 + 4.459374 \times 10^{-6} * T^3 \quad (4.23)$$

Where h_f in kJ/kg and T in $^\circ C$.

- **Enthalpy of saturated water vapor [7]**

$$h_g = 2501.689845 + 1.806916015 * T + 5.087717 \times 10^{-4} * T^2 - 1.1221 \times 10^{-5} * T^3 \quad (4.24)$$

Where h_g in kJ/kg and T in °C.

- **Latent heat enthalpy of water [7]**

$$h_{fg} = 2501.897149 - 2.407064037 * T + 1.192217 \times 10^{-3} * T^2 - 1.5863 \times 10^{-5} * T^3 \quad (4.25)$$

Where h_{fg} in kJ/kg and T in °C.

- **Boiling point elevation (BPE) [7]**

$$BPE = A_2 * X + B_2 * X^2 + C_2 * X^3 \quad (4.26)$$

Where

$$A_2 = 8.325 \times 10^{-2} + 1.883 \times 10^{-4} * T + 4.02 \times 10^{-6} * T^2 \quad (4.27)$$

$$B_2 = -7.625 \times 10^{-4} + 9.02 \times 10^{-5} * T - 5.2 \times 10^{-7} * T^2 \quad (4.28)$$

$$C_2 = 1.522 \times 10^{-4} - 3 \times 10^{-6} * T + 3 \times 10^{-8} * T^2 \quad (4.29)$$

Where BPE in °C, T in °C and s water salinity in gm/kg.

- **Condenser overall heat transfer coefficient [7]**

$$U_c = 1617.5 - 0.1537 * T + 0.1825 * T^2 - 0.00008026 * T^3 \quad (4.30)$$

Where U_c in $\frac{W}{m^2 \text{ } ^\circ C}$ and T in $^\circ C$.

- **Evaporators overall heat transfer coefficient [7]**

$$U_e = 1939.4 + 1.40562 * T - 0.0207525 * T^2 + 0.0023186 * T^3 \quad (4.31)$$

Where U_c in $\frac{W}{m^2 \text{ } ^\circ C}$ and T in $^\circ C$.

4.2.3 The System Performance

- **The performance ratio (PR)**

The PR is used as a performance indicator of MED system. It is directly related to the number of effects.

$$PR = \frac{\dot{m}_v}{\dot{m}_s} \quad (4.32)$$

- **The specific flow rate of cooling water**

$$sM_{cw} = \frac{\dot{m}_{cw}}{\dot{m}_v} \quad (4.33)$$

- **The specific heat transfer area**

$$sA = \frac{\sum_{i=1}^n A_i + A_c}{\dot{m}_v} \quad (4.34)$$

4.3 Solution Method of a PF-MED Model

4.3.1 Steady-State PF-MED Model

An iterative solution is used to solve the model equations. The model equations are solved simultaneously by Newton's method, also known as the Newton–Raphson method, with an iteration error of 1×10^{-4} . The model equations (mass, salt, and energy balance equations) have been solved using Engineering equation solver software (EES). The solution starts with defining the following input parameters: number of effects, steam temperature, intake seawater temperature, feed water salinity, feedwater temperature, and last effect temperature. The model equations are solved simultaneously to calculate the system performance parameters. Table 4.1 shows the parameters used for the steady-state PF-MED system simulations.

Table 4-1: The parameters used for steady-state PF-MED model simulations.

Parameter	Value	Unit
T_s	70	$^{\circ}\text{C}$
n	4	-
T_f	35	$^{\circ}\text{C}$
T_{cw}	25	$^{\circ}\text{C}$
T_n	40	$^{\circ}\text{C}$
$\dot{m}_v$	1	kg/s
$\dot{m}_f$	2.5	kg/s

x_f	42000	ppm
-------	-------	-----

4.3.2 Transient PF-MED Model

MATLAB is used to perform numerical integration using coupled ordinary differential equations of the mass, salt, and energy balance. The numerical integration is based on an explicit method, where the dependent variable can be separated. Runge-Kutta method is used with a time step of one second and a tolerance of 1×10^{-3} . The solution begins with assuming initial temperatures, mass, and salinity of brine water in the effects. The program assumes some initial values; it then carries on an explicit numerical integration to simulate the system's cyclic quasi-steady-state conditions. A flowchart of the computer program is shown in Figure 4.2. Table 4.2 shows the parameters used for the transient PF-MED system simulations.

Table 4-2: The parameters used for the transient PF-MED system simulations.

Parameter	Value	Unit
T_s	70	$^{\circ}\text{C}$
n	4	-
T_f	35	$^{\circ}\text{C}$
T_{cw}	25	$^{\circ}\text{C}$
$\dot{m}_s$	0.22	kg/s
$\dot{m}_{f,i}$	0.625	kg/s
$\dot{m}_{b,i}$	0.39	kg/s

x_f	42000	ppm
A_i	33	m^2

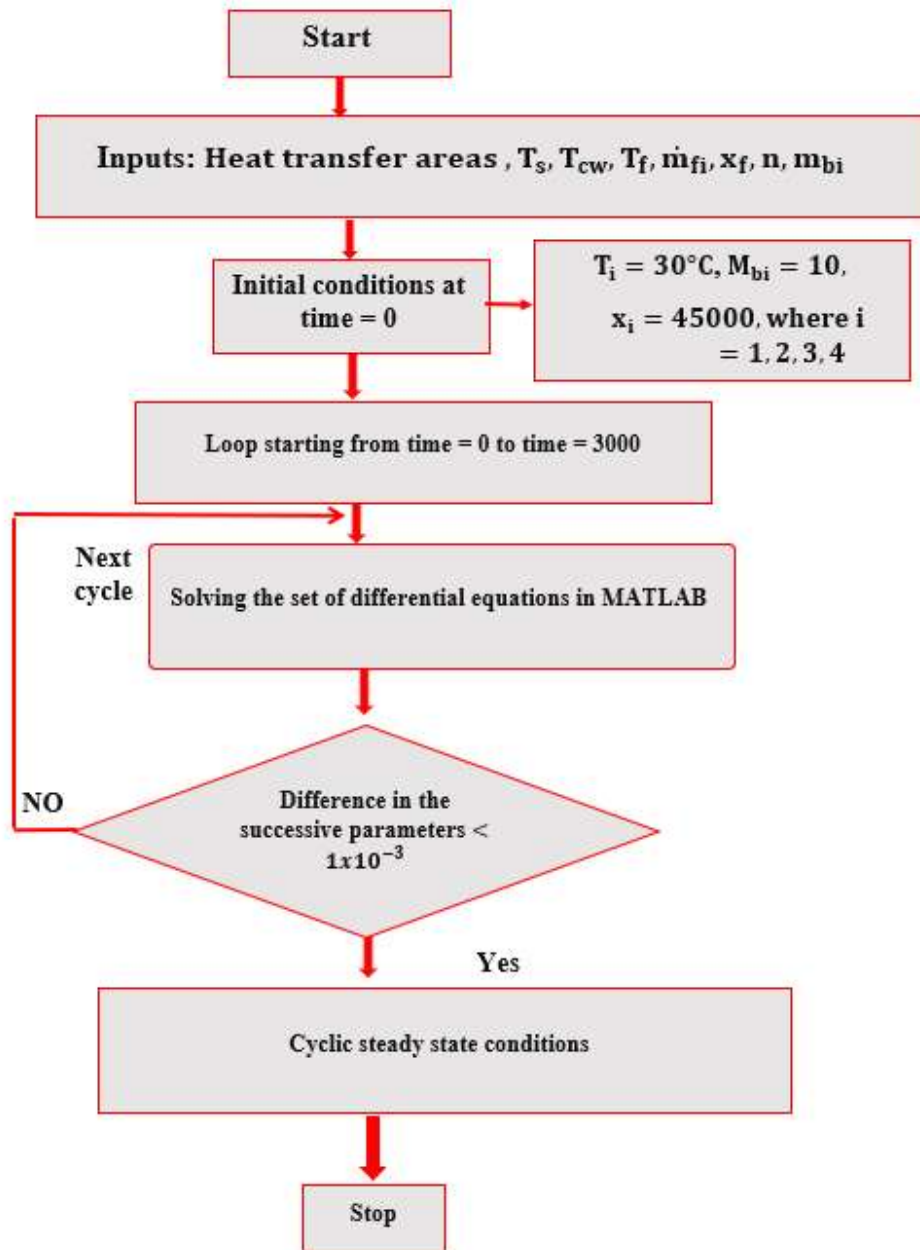


Figure 4-2: Flowchart of the computer program of transient PF-MED model.

4.4 Simulation Results and Discussion of PF-MED Model

4.4.1 Steady-State PF-MED Model

The results of the current work are validated against previous work reported by Khalid [84]. As shown in Figure 4.3, the results of the current study show an excellent agreement with Khalid [31], where the maximum percentage difference is 1%.

As a function of steam temperature, Figure 4.3 shows the PR of the PF-MED system. As the heating steam temperature increases, the system performance ratio (PR) reduces. At higher steam temperatures, the steam's latent heat decreases. In addition, because the feedwater temperature is kept constant, the sensible heat required to increase feedwater temperature to higher temperature increases at higher steam temperature. Both factors lead to higher consumption of steam, leading to lower PR.

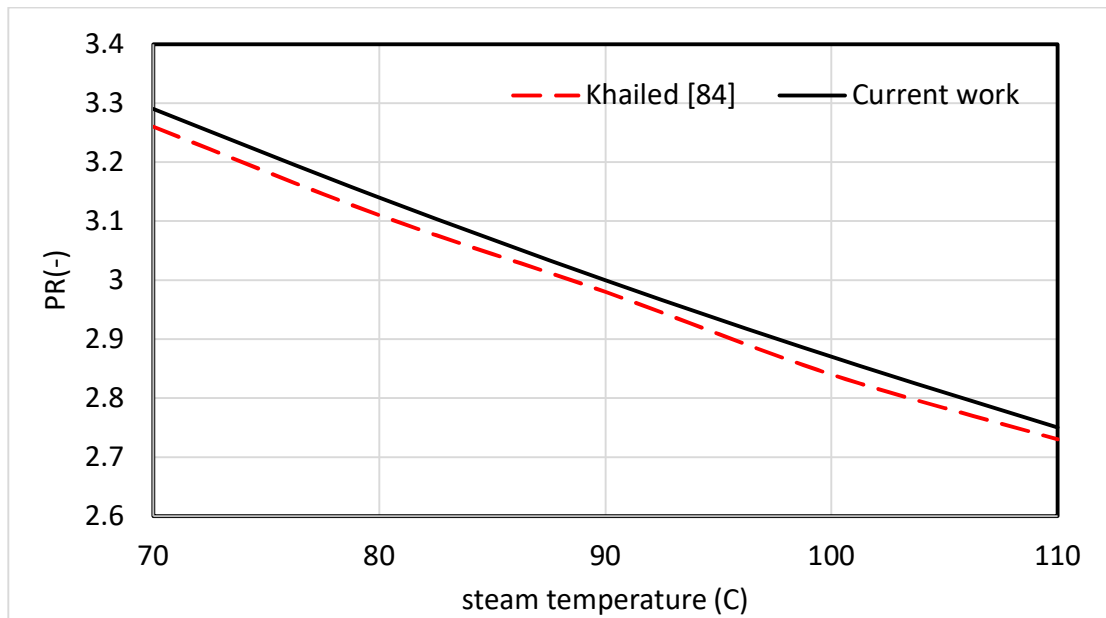


Figure 4-3: Validation of the steady-state PF-MED system model.

4.4.2 Transient PF-MED Model

Against the steady-state model, the results of the transient model are validated. The current study results show a very good agreement with the steady-state model as shown in Table 4.3, where the maximum percentage difference is 0.94 %.

Table 4-3: Comparison between the steady-state and transient model.

Effect Temperature	Steady state model	Transient model	% difference
1	62.5	62.7	0.32
2	55	54.99	0.02
3	47.5	47.59	0.2
4	40	40.38	0.94

CHAPTER 5

THE HYBRIDIZATION OF A PARALLEL FEED MULTI-EFFECT EVAPORATION DESALINATION SYSTEM WITH ADSORPTION CYCLE (MED-AD)

5.1 Background

Hybridization of a multi-effect evaporation desalination system with an adsorption system (MED-AD) is a novel system to overcome both systems' limitations to enhance the systems' performance. It is predicted that hybridization improved water production by many folds with the same heat input. Also, it is expected that the last effect's temperature can be lower than the ambient temperature [80].

In the MED-AD system, the vapor produced by the last stage is adsorbed by the adsorbent bed of the adsorption cycle (AD). The TBT and LBT limit the number of the effect of the conventional MED system. The soft scaling agents (e.g., sulfate and magnesium) are more active at a temperature higher than 70 °C resulting in system degradation. So, the TBT is limited to be lower than 70 °C. The condenser working with the cooling tower limits the LBT to be above the temperature of the ambient (40 °C).

The solution of the limited LBT is the MED-AD system because the AD cycle can work with a temperature lower than the ambient temperature (typically around five-degree Celsius). Reducing the LBT increases the operational gap, which increases the number of effects that can be installed.

In this study, the parallel feed MED system is coupled with a two-beds AD system. The MED-AD system is shown in Figure 5.1. The last stage of the MED system replaced the AD system evaporator. In the thermal integration of these systems, the last effect is called the pseudo-effect. This stage's temperature is controlled by the water adsorption of the adsorbent bed and the heat input from the previous effect.

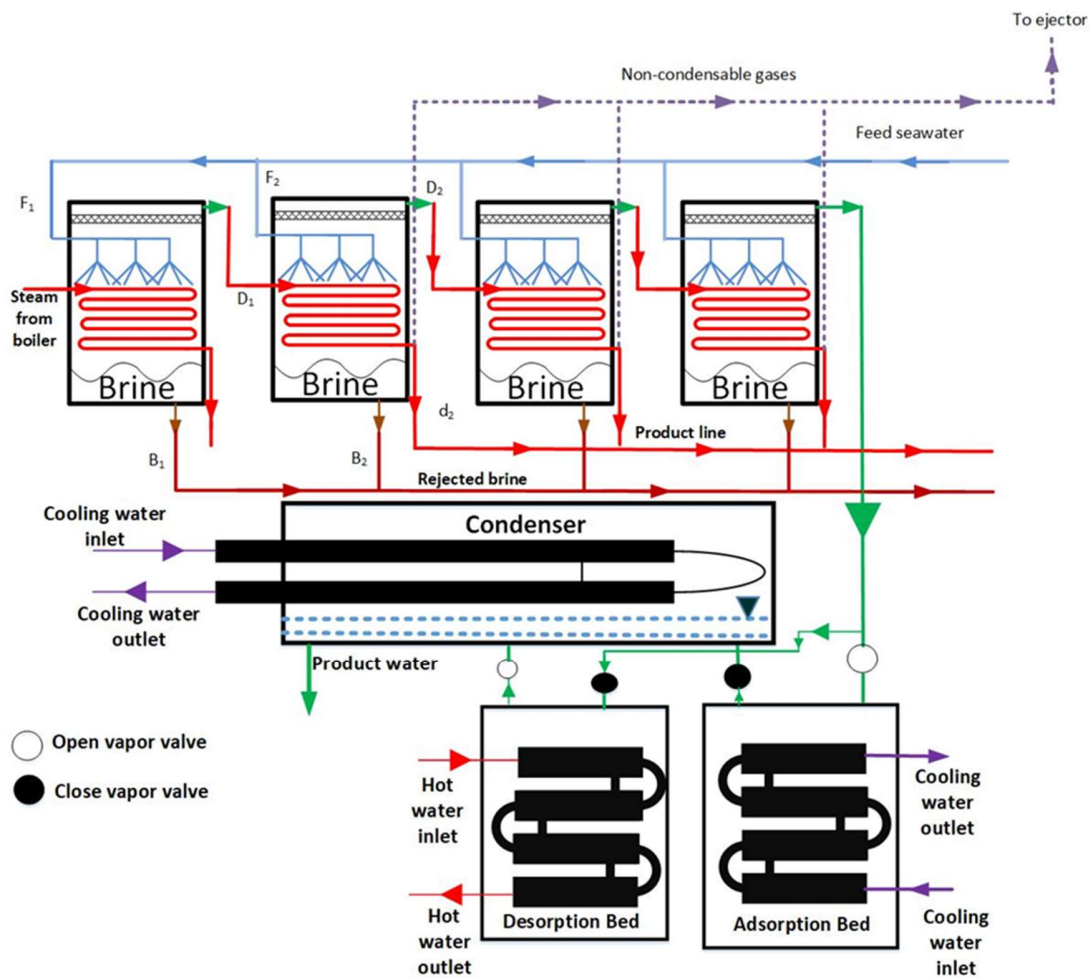


Figure 5-1: MED-AD system.

5.2 Hybridization of A Parallel Feed Multiple-Effect Desalination

System with A Two-Beds Adsorption System (MED-AD)

In the MED first effect, the steam coming from an external boiler condenses inside the tube. In order to raise the temperature to the boiling temperature and to evaporate some of it the feeding seawater absorbs the latent heat of the condensation. Generated vapor from one effect is re-used as a heat source for the next stage. The last effect of the MED system is used as an evaporator for the AD system. The adsorbent bed adsorbs the formed vapor from the last effect until it becomes saturated (adsorption process). Simultaneously, the second bed, which is initially saturated with water vapor, release the water vapor (desorption process). The desorbed vapor condensed within the AD condenser. As the adsorption process is exothermic, cold water circulates within the heat exchanger of the adsorption bed to preserve the adsorption of water vapor. In contrast, the desorption process is endothermic, so hot water circulates through the heat exchanger of the desorber bed to sustain the desorption of water vapor. When the adsorption bed becomes saturated, preheating and precooling processes occur simultaneously. Preheating the adsorption bed happens to increase its pressure from evaporator pressure to the condenser pressure. The desorption bed's precooling happens to reduce its pressure from condenser pressure to evaporator pressure during the switching time. Simple valves are used to change over adsorbent beds' duties from adsorption to desorption or vice versa.

5.3 Mathematical Model of PF-MED-AD

5.3.1 Transient Model of PF-MED System

Since the adsorption process is cyclic, the transient model of the PF-MED system is used.

The transient mathematical model was developed for the PF-MED with the following assumptions:

- The areas of heat transfer are equal and constant for all the stages.
- Neglecting the depression of temperature corresponding to the pressure drop in the demister, vapor transmission lines, and during the process of condensation.
- The feed water is distributed among the effects equally.
- The vapor salinity is zero (does not contain salt).
- No energy loss between the effects and the surrounding.
- Vapor is at the temperature of saturation.

- Mathematical model for the first effect ($i=1$)

As shown in Figure 5.2, first effect is used to produce the MED system's initial vapor.

It is a simple shell and tube heat exchanger with tubes arranged horizontally. Inside the tubes, the saturated steam is condensed, and seawater spattered on the tube's outer side.

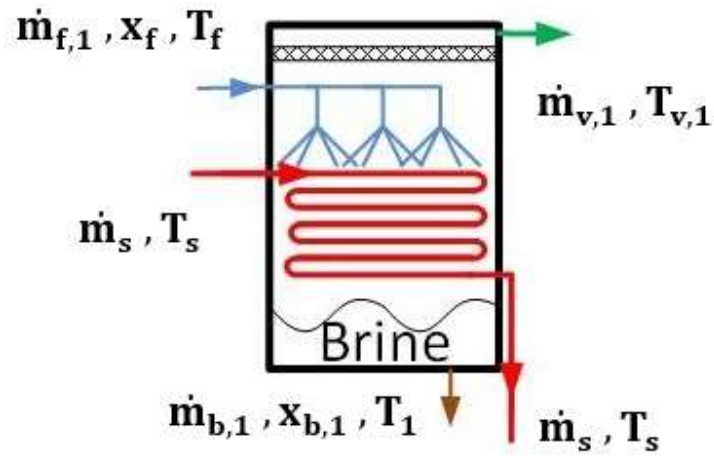


Figure 5-2: First effect.

Mass balance: Instantaneous change in liquid balance in effect can be found by mass balance equation as:

$$\frac{dM_{b,1}}{dt} = \dot{m}_{f,1} - \dot{m}_{b,1} - \dot{m}_{v,1} \quad (5.1)$$

Where M_b the brine water accumulation in the effect, $\dot{m}_f$ the feed seawater flow rate, $\dot{m}_b$ the rejected brine water flow rate and $\dot{m}_v$ the vapor flow rate.

Salt balance: Instantaneous change in the concentration of liquid salt inside the effect can be calculated by the salt balance equation as:

$$M_{b,1} \frac{dx_{b,1}}{dt} = x_f \dot{m}_{f,1} - x_{b,1} \dot{m}_{b,1} \quad (5.2)$$

Where x_b the brine water salt concentration, x_f the feed seawater salt concentration.

Energy balance: Transient brine boiling temperature of the stage can be found by the energy balance equation as:

$$(M_{b,1} c_{p,b,1} + M_{HX,1} c_{p,HX,1}) \frac{dT_1}{dt} = h_{f,T_f} \dot{m}_{f,1} - h_{f,T_1} \dot{m}_{b,1} - h_{g,T_{v,1}} \dot{m}_{v,1} + \dot{m}_s h_{fg,T_s} \quad (5.3)$$

Where h_{f,T_f} the feed seawater enthalpy, h_{f,T_1} the rejected brine water enthalpy, $h_{g,T_{v,1}}$ the formed vapor enthalpy, h_{fg,T_s} the latent heat entholpy, $\dot{m}_s$ the steam flow rate, M_{HX} the mass of the heat exchanger.

The vapor formed in the effect:

$$\dot{m}_{v,1} = \frac{U_2 A_2 (T_{v,1} - T_2)}{h_{fg,T_{v,1}}} \quad (5.4)$$

The vapor temperature:

$$T_{v,1} = T_1 - BPE_1 \quad (5.5)$$

- **Mathematical model for the intermediate effects ($1 < i < n$)**

As shown in Figure 5.3, intermediate effects are used to reuse the previous effect's vapor. It is a simple shell and tube heat exchanger with tubes arranged horizontally. Inside the tubes, the saturated vapor is condensed, and seawater spattered on the tube's outer side.

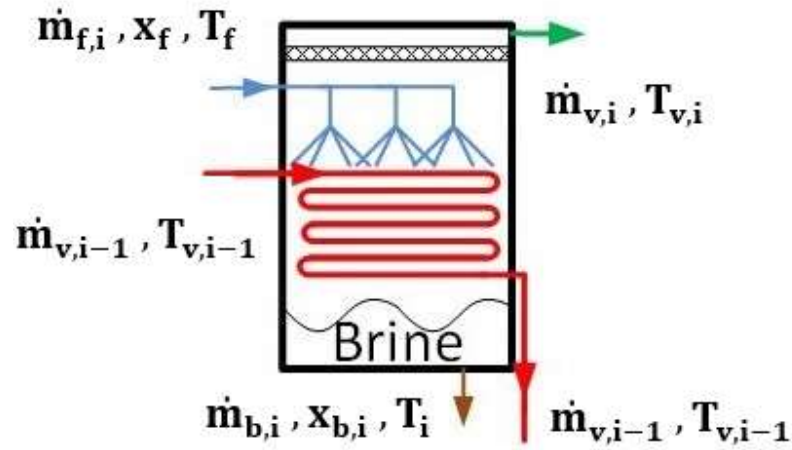


Figure 5-3: the intermediate effects.

Mass balance: Instantaneous change in liquid balance in effect can be found by mass balance equation as:

$$\frac{dM_{b,i}}{dt} = \dot{m}_{f,i} - \dot{m}_{b,i} - \dot{m}_{v,i} \quad (5.6)$$

Salt balance: Instantaneous change in the concentration of liquid salt inside the effect can be calculated by the salt balance equation as:

$$M_{b,i} \frac{dx_{b,i}}{dt} = x_f \dot{m}_{f,i} - x_{b,i} \dot{m}_{b,i} \quad (5.7)$$

Energy balance: Transient brine boiling temperature of the effect can be found by the energy balance equation as:

$$\begin{aligned}
& (M_{b,i} c_{p,b,i} + M_{HX,i} c_{p,HX,i}) \frac{dT_1}{dt} \\
& = h_{f,T_f} \dot{m}_{f,i} - h_{f,T_i} \dot{m}_{b,i} - h_{g,T_{v,i}} \dot{m}_{v,i} \\
& + \dot{m}_{v,i-1} h_{fg,T_{v,i-1}}
\end{aligned} \tag{5.8}$$

The vapor formed in the effect:

$$\dot{m}_{v,i} = \frac{U_{i+1} A_{i+1} (T_{v,i} - T_{i+1})}{h_{fg,T_{v,i}}} \tag{5.9}$$

The vapor temperature:

$$T_{v,i} = T_i - BPE_i \tag{5.10}$$

- **Mathematical model for MED last effect attached to AD beds (i=n)**

Mass balance: Instantaneous change in liquid balance in effect can be found by mass balance equation as:

$$\frac{dM_{b,n}}{dt} = \dot{m}_{f,n} - \dot{m}_{b,n} - M_{adt} \frac{dq_{ads}}{dt} \tag{5.11}$$

Salt balance: Instantaneous change in the concentration of liquid salt inside the effect can be calculated by the salt balance equation as:

$$M_{b,n} \frac{dx_{b,n}}{dt} = x_f \dot{m}_{f,n} - x_{b,n} \dot{m}_{b,n} \tag{5.12}$$

Energy balance: Transient brine boiling temperature of the effect can be found by the energy balance equation as:

$$\begin{aligned}
& (M_{b,n} c_{p,b,n} + M_{HX,n} c_{p,HX,n}) \frac{dT_n}{dt} \\
& = h_{f,T_f} \dot{m}_{f,n} - h_{f,T_n} \dot{m}_{b,n} - h_{g,T_{v,n}} M_{adt} \frac{dq_{ads}}{dt} \\
& + \dot{m}_{v,n-1} h_{fg,T_{v,n-1}}
\end{aligned} \tag{5.13}$$

The vapor formed in the effect:

$$\dot{m}_{v,n} = M_{adt} \frac{dq_{ads}}{dt} \tag{5.14}$$

The vapor temperature:

$$T_{v,n} = T_n - BPE_n \tag{5.15}$$

5.3.2 Mathematical model of AD system

5.3.2.1 Energy balance

The lumped parameter model was applied to develop a mathematical model of a two-bed AD with the following assumptions:

- Temperature, pressure, and the mass adsorbed in the adsorber beds are uniform.
- There is a negligible pressure differential between the adsorber and the condenser or desorber and the last stage of the MED.
- Convection heat transfer effect inside the beds is negligible.
- The adsorbed/desorbed water's specific heat capacity is the same as for the bulk liquid water.
- The surrounding heat loss is neglected.

The adsorber/desorber energy balancing equation can be written as [31]:

$$\begin{aligned}
& (M_{\text{adt}}C_{p,\text{adt}} + M_{\text{adt}} q_{\text{bed}}C_{p,w} + M_{\text{hex}}C_{p,\text{hex}}) \frac{dT_{\text{bed}}}{dt} \\
& = \dot{m}_f C_{p,f} (T_{\text{in,bed}} - T_{\text{out,bed}}) + M_{\text{adt}} Q_{\text{st}} \frac{dq_{\text{bed}}}{dt}
\end{aligned} \tag{5.16}$$

$$T_{\text{out,bed}} = T_{\text{bed}} + (T_{\text{in,bed}} - T_{\text{bed}}) \exp \left[-\frac{(UA)_{\text{bed}}}{\dot{m}_f C_{p,f}} \right] \tag{5.17}$$

The sensible heat of the adsorbent material, the adsorbate inside the bed, and the heat exchanger are defined on the equation's left-hand side (5.16). Sensible heat emitted or absorbed by the heat transfer fluid plus the heat of the adsorption/desorption process is depicted on the right side. Equation (5.17) is used on the basis of the log mean temperature the difference of the adsorber bed heat exchanger to estimate the fluid outlet temperature.

The AD-condenser's energy balance equation can be written as [31]:

$$\begin{aligned}
& (M_{\text{hex,cond}}C_{p,\text{hex,cond}}) \frac{dT_{\text{cond}}}{dt} \\
& = \dot{m}_{f,\text{cond}} C_{p,f} (T_{\text{in,cond}} - T_{\text{out,cond}}) - M_{\text{adt}} h_{fg} \frac{dq_{\text{des}}}{dt} \\
& + M_{\text{adt}} h_{f,T_{\text{cond}}} \frac{dq_{\text{des}}}{dt}
\end{aligned} \tag{5.18}$$

$$T_{\text{out,cond}} = T_{\text{cond}} + (T_{\text{in,cond}} - T_{\text{cond}}) \exp \left[-\frac{(UA)_{\text{cond}}}{\dot{m}_{f,\text{cond}} C_{p,f}} \right] \tag{5.19}$$

The sensible heat of the metallic heat exchanger within the condenser is expressed by the left-hand side of equation (5.18). The sensible heat consumed by the heat transfer fluid reflects the first term on the right-hand side. The heat of the condensation process is expressed by the second term, while the third term is the sensible heat of the distillate water

leaving the condenser. Equation (5.19) is used on the basis of the log mean temperature the difference of the condenser heat exchanger to estimate the fluid outlet temperature.

5.3.2.2 Adsorption isotherms

The amount of refrigerant uptake and equilibrium conditions are defined by the adsorption isotherms. To measure the adsorption isotherms for silica gel RD-2060-water pairs, the modified Freundlich model was used [31].

$$q^* = A(T_{bed}) * \left(\frac{P_{cond/evap}}{P_{bed}} \right)^{B(T_{bed})} \quad (5.20)$$

$$\text{where } A(T_{bed}) = A_o + A_1 * T_{bed} + A_2 * T_{bed}^2 + A_3 * T_{bed}^3 \quad (5.21)$$

$$\text{and } B(T_{bed}) = B_o + B_1 * T_{bed} + B_2 * T_{bed}^2 + B_3 * T_{bed}^3 \quad (5.22)$$

where the constants are listed in Table 5.1.

The model for Langmuir is to measure the isotherms of adsorption for HKUST-1-water [44].

$$q^* = \frac{0.6576 * \left(4.548 * \frac{P_{cond/evap}}{P_{bed}} \right)}{1 + \left(4.548 * \frac{P_{cond/evap}}{P_{bed}} \right)} \quad (5.23)$$

Table 5-1: Values of constants used in equations 5.21 and 5.22.

Constant	Value	Constant	Value
A_o	-6.5314 kg	B_o	-15.587
A_1	0.072452 kg/K	B_1	0.15915 K ⁻¹

A₂	$-0.23951 \times 10^{-3} \text{ kg/K}^2$	B₂	$-0.50612 \times 10^{-3} \text{ K}^{-2}$
A₃	$0.25493 \times 10^{-6} \text{ kg/K}^3$	B₃	$-0.5329 \times 10^{-6} \text{ K}^{-3}$

5.3.2.3 Adsorption kinetics

In any case, the rate of adsorption is proportional to the difference between equilibrium concentration and concentration at any instance. Using a linear driving force kinetic equation, the adsorption rate is calculated.

$$\frac{dq}{dt} = k_1 * \exp\left(-\frac{k_2}{T_{\text{bed}}}\right) * (q^* - q) \quad (5.24)$$

where k_1 and k_2 are constants that depend on the adsorbent-adsorbate pairs. Their values are given in Table 5.2 for different working pairs.

Table 5-2: Values of constants used in equation (5.24).

Working pair	k₁ (1/s)	k₂ (K)	Reference No.
RD silica gel -water	7558.02	5051.72	31
HKUST-1-water	6.375	2393.55	44

5.3.2.4 The system performance

MED first effect heat input: saturated steam circulated through the first effect's heat exchanger to produce intimal vapor. This energy supplied is calculated as:

$$Q_s = \dot{m}_s h_{fg,T_s} \quad (5.25)$$

Desorber bed heat input: hot water circulated through the desorber bed's heat exchanger to supply desorption heat. This energy supplied is calculated as:

$$Q_{des} = \dot{m}_{hw} C_{p,hw} (T_{hw,in} - T_{hw,out}) \quad (5.26)$$

Heat rejected the adsorber bed: cooling water circulated through the heat exchanger of the adsorber bed to absorb the heat of the adsorption. The energy supplied shall be measured as:

$$Q_{ads} = \dot{m}_{cw} C_{p,cw} (T_{cw,out} - T_{cw,in}) \quad (5.27)$$

Heat rejected the AD condenser: cooling water circulated through the heat exchanger of the condenser to release the residual heat of the condensation. The energy supplied shall be measured as:

$$Q_{cond} = \dot{m}_{cw,cond} C_{p,cw,cond} (T_{cw,cond,out} - T_{cw,cond,in}) \quad (5.28)$$

AD-condenser distillate water: desorbed water vapor condensed in AD condenser is calculated as:

$$\dot{m}_{dest,AD,cond} = \frac{(UA)_{cond} (LMTD)}{h_{fg,T_{cond}}} \quad (5.29)$$

$$\text{where } (LMTD) = \frac{T_{cw,cond,out} - T_{cw,cond,in}}{\ln \left(\frac{T_{cond} - T_{cw,cond,in}}{T_{cond} - T_{cw,cond,out}} \right)} \quad (5.30)$$

The MED effect distillate water: water produced from each MED effect is calculated as:

$$\dot{m}_{dest,i} = \frac{(UA)_{i+1}(T_{v,i} - T_{i+1})}{h_{fg,T_{v,i}}} \quad (5.31)$$

Total distillate water is calculated as:

$$\dot{m}_{dest,total} = \sum_{i=1}^n \dot{m}_{dest,i} + \dot{m}_{dest,AD,cond} \quad (5.32)$$

The system performance ratio is calculated as:

$$PR = \frac{\dot{m}_{dest,total} h_{fg}}{Q_{des} + Q_s} \quad (5.33)$$

5.4 Solution Method of a MED-AD System

MATLAB is used to perform numerical integration using a set of coupled ordinary differential equations of the AD system (AD-condenser, the adsorption/desorption beds) and PF-MED system (mass, salt, and energy balance equations). The numerical integration is based on an explicit method, where the dependent variable can be separated. Runge-Kutta method is used with a time step of one second and tolerance of 10^{-3} . The solution begins with assuming initial temperatures, masses, and water uptakes. The program

assumes some initial values, it then carries on an explicit numerical integration to simulate the system's cyclic quasi-steady-state conditions. The computer program flow chart is shown in Figure 5.4.

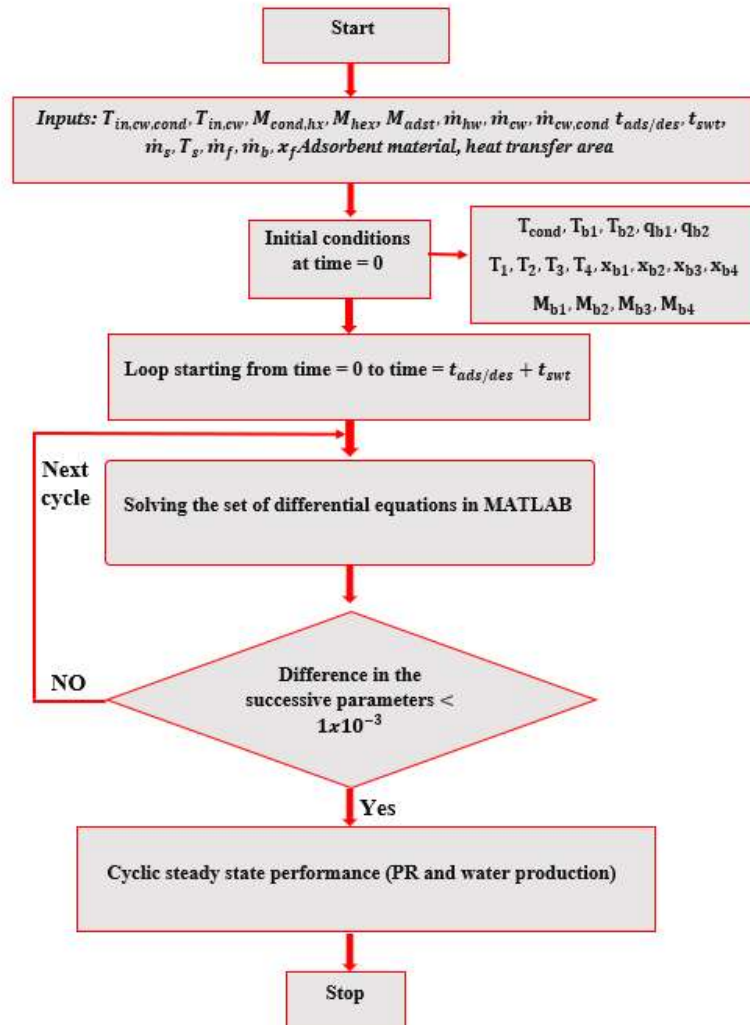


Figure 5-4: flowchart of the computer program.

5.5 Results and Discussion of a MED-AD system

Current work results are validated against previous work reported by Son et al. [77].

Current work results show a very good agreement with Son et al. [77] as shown in Table 5.3, where the maximum percentage difference is 1.65 %.

Table 5-3: validation table.

Effect average temperature	Son et al. [77]	Current work	% difference
1	44.49	43.75	1.65
2	40.1	40.04	0.04
3	34.61	35.05	1.26
4	26.82	26.44	1.43

A simulation analysis was conducted under the operating condition described in Table 5.4 to predict the effect of the operating conditions on the performance of the MED-AD system.

Table 5-4: The MED-AD system operating conditions.

Symbol	Value	Unit
$t_{ads/des}$	800	S
t_{swt}	30	S
$T_{in,hot}$	85	°C
$T_{in,cw}$	25	°C

$T_{in,cw,cond}$	25	°C
$\dot{m}_{hw}$	1.25	Kg/s
$\dot{m}_{cw}$	1.25	Kg/s
$\dot{m}_{cw,cond}$	1.25	Kg/s
M_{adt}	100	kg
$M_{cond,hx}C_{p,cond,hx}$	9372.08	J/kg
$M_{hex}C_{p,hex}$	7719.4	J/K
$C_{p,sg}$	924	J/kg/K
$C_{p,w}$	4180	J/kg/K
$C_{p,f}$	4180	J/kg/K
$C_{p,v}$	1890	J/kg/K
$C_{p,cw,cond}$	4180	J/kg/K
$C_{p,cw}$	4180	J/kg/K
$(UA)_{ads}$	3500	W/K
$(UA)_{des}$	3500	W/K
$(UA)_{cond}$	15330	W/K

Q_{st}	2.8×10^6	J/kg
$h_{fg,cond}$	2.5×10^6	J/kg
T_s	50	°C
n	4	-
T_f	30	°C
$\dot{m}_s$	0.01	kg/s
$\dot{m}_f$	0.8	kg/s
$\dot{m}_{b,i}$	0.1	kg/s
x_f	42000	ppm
A_i	4	m ²

The MED-AD system's total energy consumption calculation considers only the payable energy related to the steam boiler (the steam that to be used for the first effect). In contrast, the energy needed for adsorbent regeneration is considered non-payable energy (solar energy).

Figure 5.5 displays the temporary temperature profiles of the AD cycle components (adsorber and desorber heat exchangers and condenser) with an inlet hot water temperature (desorption temperature) 85 °C and an inlet cooling water temperature 25 °C and operating conditions given in Table 5.4. The working pair used was RD silica gel-water. The cooling

of the adsorption bed is longer and more complicated than the heating of the desorption bed. Thus, the bed temperature at the end of the desorption process remains within 5 °C lower than the heating water temperature. However, the adsorption process's bed temperature remains within 7 or 8 °C higher than the cooling water temperature. The condenser temperature reaches a peak at the start of each cycle due to the increase in the cooling water temperature. The sharp increase of condenser cooling water outlet temperature at the beginning of the desorption process is high because the desorber is saturated, and the desorption rate is high.

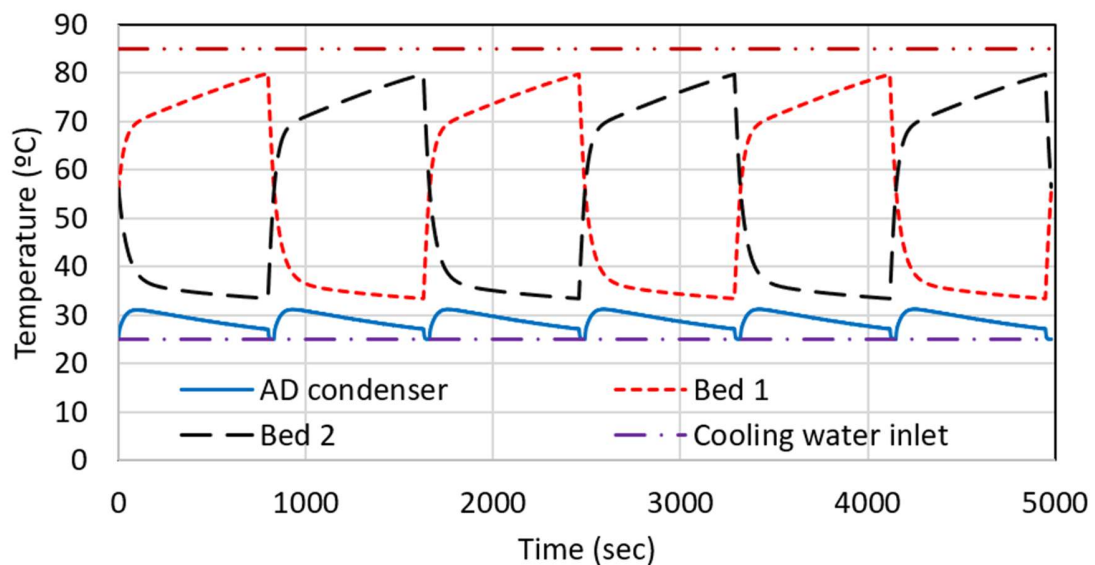


Figure 5-5: Transient temperature profile of the major components heat exchangers (RD silica gel -water).

Figure 5.6 demonstrates the transient temperature profiles of the MED effects. After a half-cycle time, the MED last effect and AD condenser are disconnected from adsorbent beds (no adsorption or desorption processes occur), and precooling and preheating processes occur to switch the beds from adsorption bed to desorption bed. As a result, the last effect temperature and condenser temperature increase sharply due to the comment water vapor

uptake process during the switching period. The MED effect temperature fluctuation is detected at the MED last effect due to the AD cyclic operation nature. It propagates to the middle effects until it is damping down due to the upstream drive effect.

One of the main drawbacks of the MED cycle is limiting the bottom brine temperature (last effect temperature) to be greater than the ambient temperature (40 °C) due to cooling tower operation. Thus, the hybridization of MED and AD by connecting the last MED effect to the adsorption bed results in lowering the MED last effect temperature below the ambient temperature due to water vapor uptake by AD adsorbent bed. The AD bed adsorbs the water vapor from the last stage. The temperature drop is caused by the mass transfer effect resulting from water vapor uptake. In the MED-AD system, it clear that most of the stages' temperatures below ambient temperature. As a result, the operational gap increases. Thus, more stages can be inserted, resulting in more distillate water production and better recovery of heat input. It is clear from Figure 5.6 that the last effect temperature with silica gel as adsorbent material is around 26 °C, while it is about 17 °C using the HKUST-1 instead of the silica gel as adsorbent material. So, this means the last effect temperature is reduced by 35 %. Thus, more effects can be inserted compare with the silica gel system; hence, a better system's performance can be achieved.

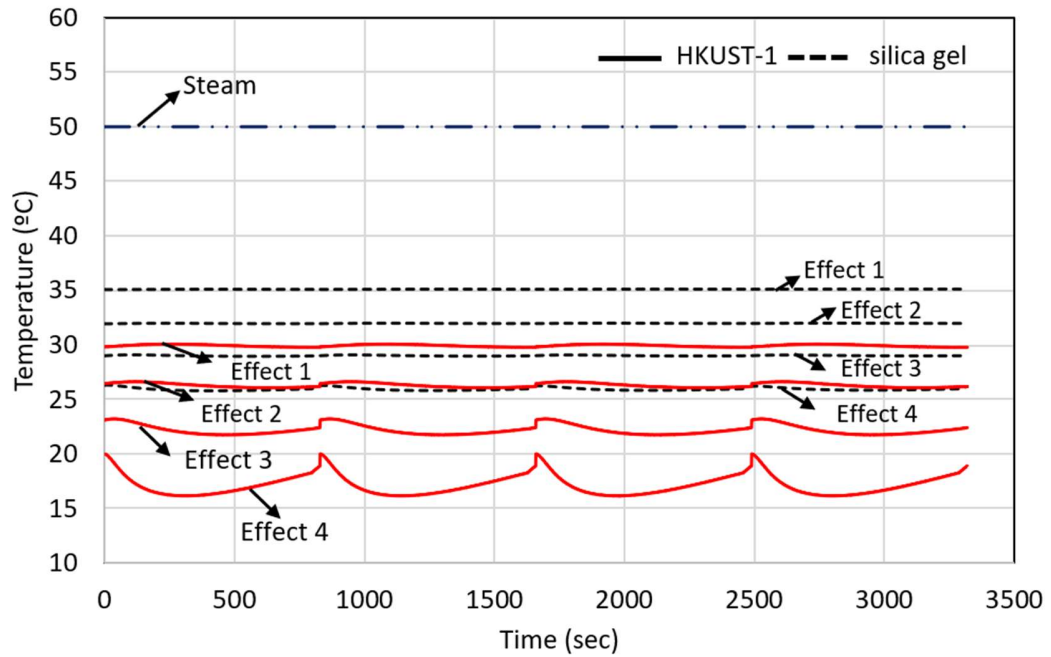


Figure 5-6: The transient temperature profile of the MED stages (RD silica gel -water and HKUST-1-water).

In the conventional four effect MED system with a steam temperature of 50 °C, the temperature difference between two consecutive effects is 2.5 °C. However, in the MED-AD system, the temperature difference varies from 3 C to 5 °C. This higher temperature difference results in a higher evaporation rate caused by higher heat flux, thus higher distillate water production. The cyclic distillate water production from the AD condenser results in water production fluctuation, as shown in Figure 5.7. The distillate water production has a minimum value during the switching period since there is no adsorption and desorption processes occur. The MED-AD average distillate water production increases by two to three times compared to the conventional MED cycle. This is because of the higher temperature difference and a higher water vapor formed in the last stage due to the adsorbent bed's water uptake. It is clear from Figure 5.7 that the average distillate water production with HKUST-1 as adsorbent material is significantly higher than average distillate water production with silica gel by around 40% improvement.

Most of the water vapor uptake occurs at a high relative pressure (adsorbent bed pressure/saturation pressure) for silica gel. Thus, the silica gel's water vapor adsorption is low since the AD system's adsorption process occurs at low relative pressure. The HKUST-1 has a significantly higher water vapor uptake compared to the silica gel. This because the HKUST-1 has an adsorption isotherm that shows that most water vapor uptake occurs at low relative pressure. Moreover, the HKUST-1 desorbed more water vapor since it is less hydrophilic than silica gel. Thus, the HKUST-1 performed better than the silica gel since higher water vapor uptake results in lower effects temperature and higher distillate water production.

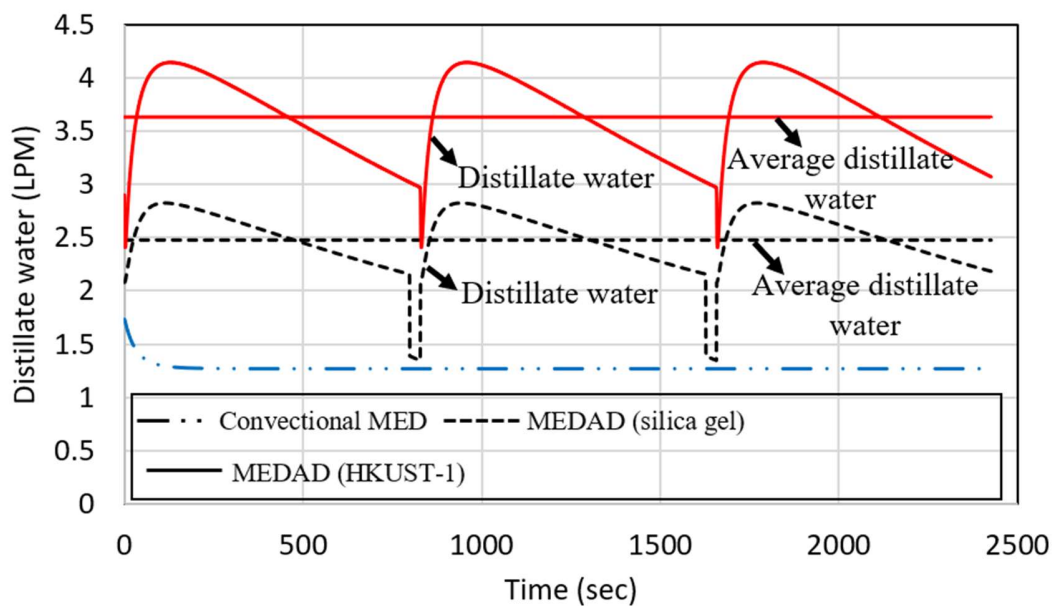


Figure 5-7: The transient distillate water profile (RD silica gel -water and HKUST-1-water).

5.5.1 The Impact of The MED System Operating Conditions on The MED-AD System Performance.

Figure 5.8 shows the impact of the flow rate of the mass on the performance ratio and the average of total distillate water. The total distillate water increases with increasing the flow rate of the steam mass due to the higher evaporation rate resulting from higher energy input to the first effect. In contrast, the performance ratio decreases with increasing the flow rate of the steam mass. This because as the flow rate of the steam mass increases, the energy consumption increases. As the flow rate of the mass increases by 140 %, the energy consumption increases by 140 % while the total distillate water's energy only increases by 70%. As a result, the PR would be lower as the mass of steam is increased.

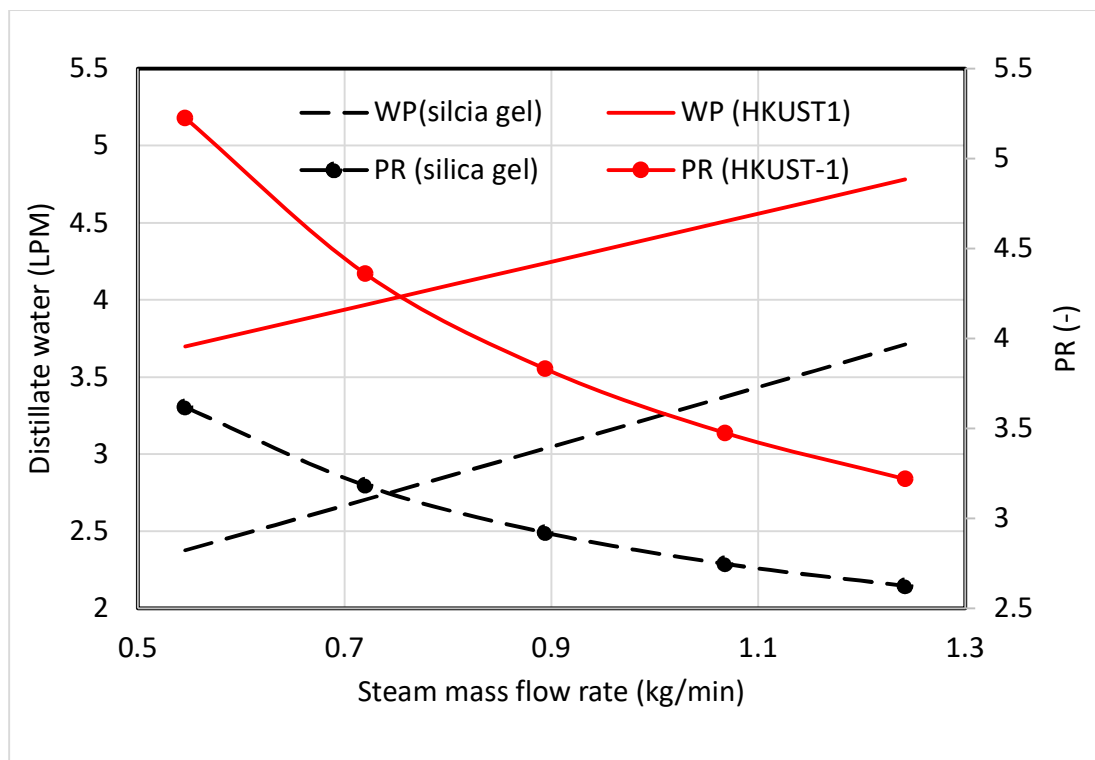


Figure 5-8: The effect of the steam flow rate on the system performance (RD silica gel -water and HKUST-1-water).

Figure 5.9 shows the influence of the feed flow rate on the output of the system. It is obvious that as the feedwater flow rate increases, the average distillate water production decreases. This is due to the lower rate of evaporation resulting from rising the requisite sensible heat to lift its temperature from the temperature of the feed to the temperature of boiling. The lower output of distillate water at a higher flow rate of feed water results in a decrease in PR.

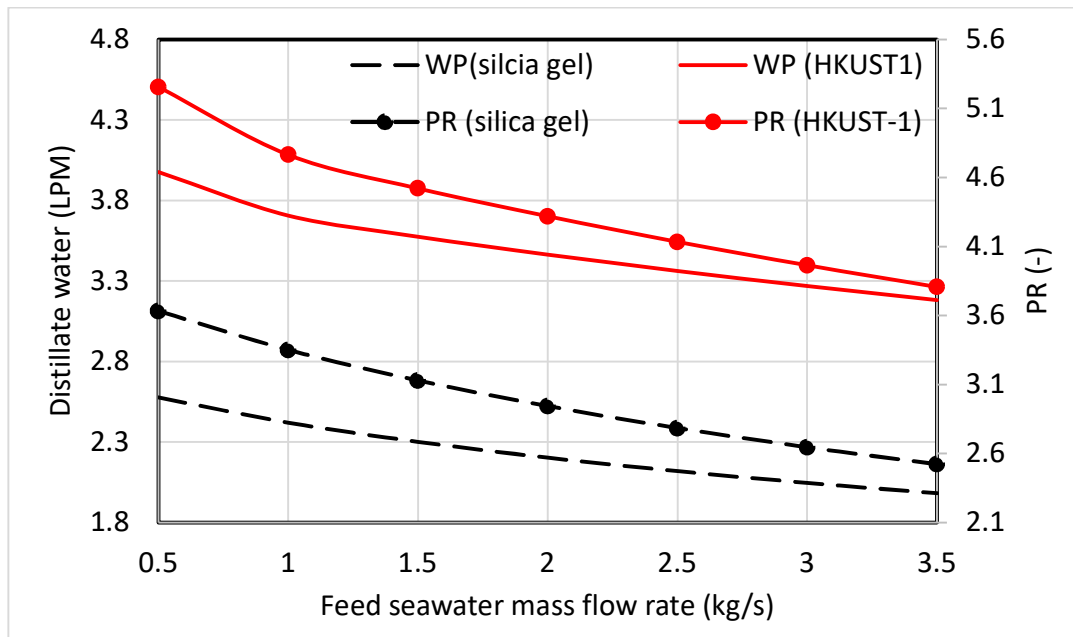


Figure 5-9. : The effect of the feed seawater flow rate on the system performance (RD silica gel -water and HKUST-1-water).

5.5.2 The Impact of The AD Cycle Operating Conditions on The MED-AD System Performance.

The impact of hot water inlet temperature (desorption temperature) on system performance is demonstrated in Figure 5.10. As the temperature of the hot water inlet rises, the output of distillate water increases due to the desorption rate enhancement. Before the adsorption mode, the rise in the desorption rate causes more water vapor adsorbed in the adsorption

mode. Also, the PR increases as the desorption temperature increases due to the higher distillate water production. It is clear that increasing the desorption temperature from 60 °C to 90 °C increases the distillate water production of silica gel and HKUST-1 by 21 % and 61 %, respectively. This is because HKUST-1 is less hydrophilic, which means that it releases more water vapor at the same partial pressure. Moreover, the HKUST-1 has a higher water vapor uptake due to the smaller particle size and higher BET surface area, as shown in Table 5.5.

Table 5-5: Physical properties of adsorbent materials [43].

Property	Silica gel RD 2060	HKUST-1
Average radius of adsorbent particle	16 μm	0.71 μm
BET surface area	840 m^2/g	1500-2100 m^2/g

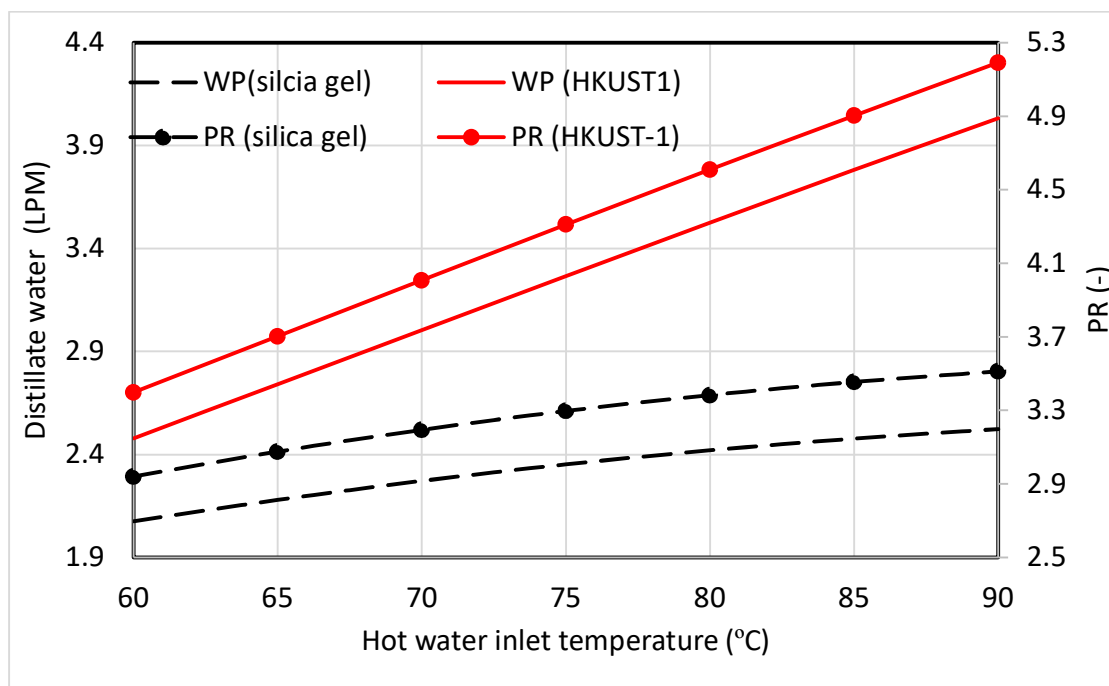


Figure 5-10: The effect of the hot water inlet temperature on the system performance (RD silica gel -water and HKUST-1-water).

Figure 5.11 displays the impact of the cooling water inlet temperature on system performance. Increasing the temperature of the inlet of cooling water contributes to a decrease in the PR and distillate water production due to the lower amount of refrigerant adsorbed and slower condensation of the desorbed refrigerant. The slower condensation causes insufficient drying of the adsorbent bed. Thus, it results in a lower adsorption rate in the next half-cycle. The desorption temperature effect is more pronounced than the cooling water inlet temperature for both adsorbent materials.

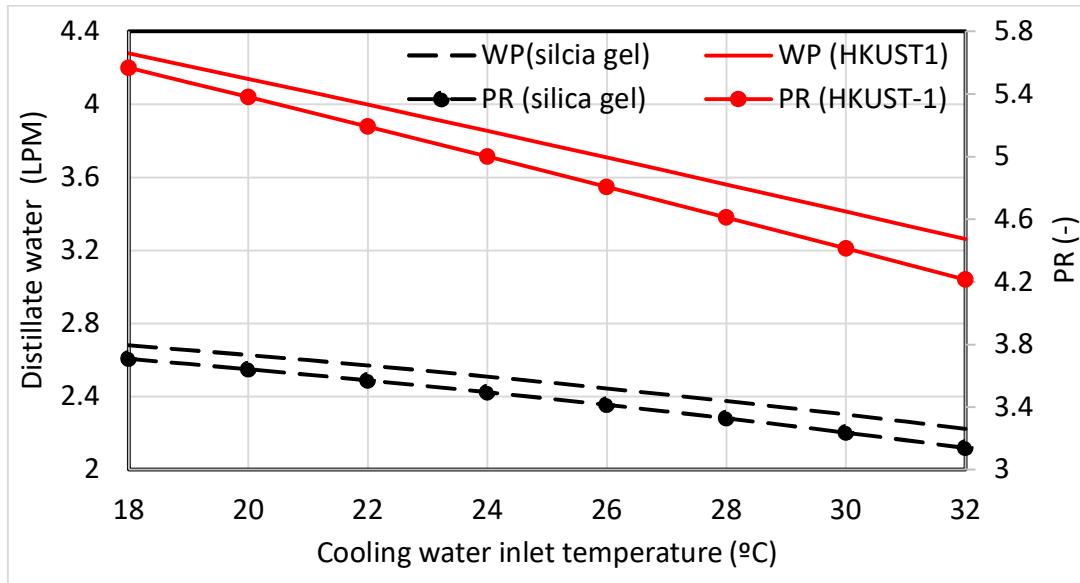


Figure 5-11: The effect of the cooling water inlet temperature on the system performance (RD silica gel -water and HKUST-1-water).

Figure 5.12 displays the effect of adsorbent material mass on system performance. As the adsorbent mass increases from 20 kg to 80 kg, the PR rises sharply. This rise is because more water vapor is adsorbed and desorbed. A further increase in adsorbent mass results in an insignificant rise in PR because the adsorber bed energy consumed does not contribute to the desorption but only increases its component's sensible heat. The distillate water production increases as the adsorbent mass increase due to the more water vapor adsorbed and desorbed.

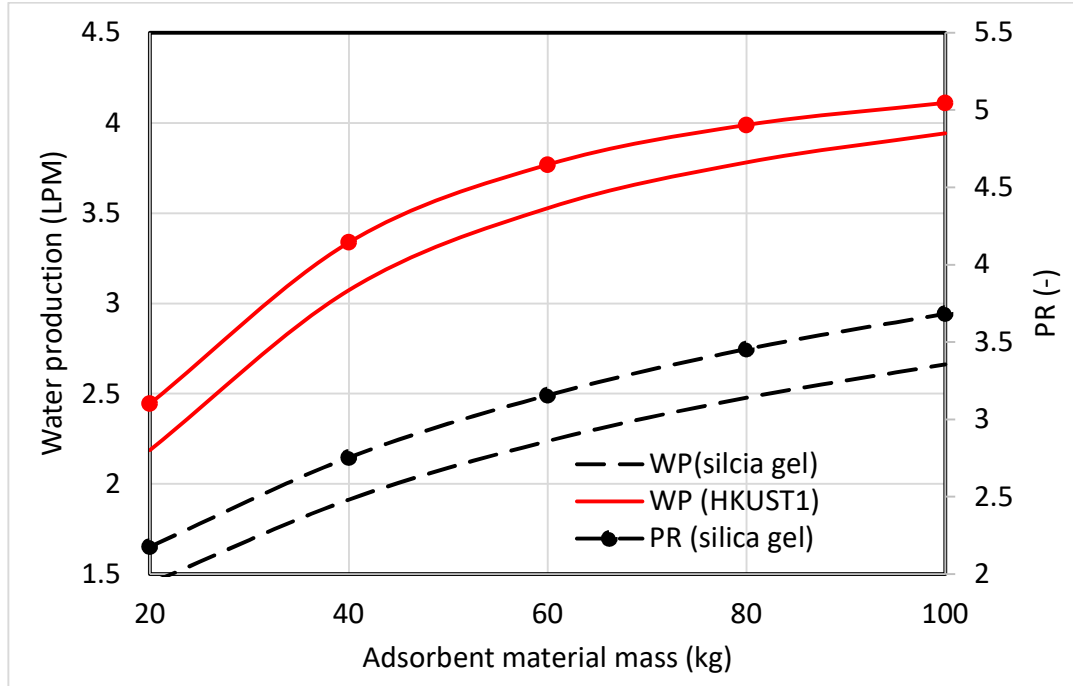


Figure 5-12: The effect of the adsorbent material mass on the system performance (RD silica gel -water and HKUST-1-water).

Figure 5.13 displays the effect of adsorption/desorption time on the system's performance. The PR and the distillate water production increases as the adsorption/desorption time increases until it reaches a maximum value at an optimal adsorption/desorption time of 750 seconds. Then, it starts to decrease. When the adsorption/desorption time is shorter, time is not enough for desorption and adsorption to occur thoroughly. Thus, the adsorbent bed's affinity for the water vapor uptake in the next cycle is reduced. On the other hand, when the time is longer, the rate of adsorption decreases. Longer desorption time results in additional heat input to the desorption bed that only heats the bed components and has no effect on the desorption rate.

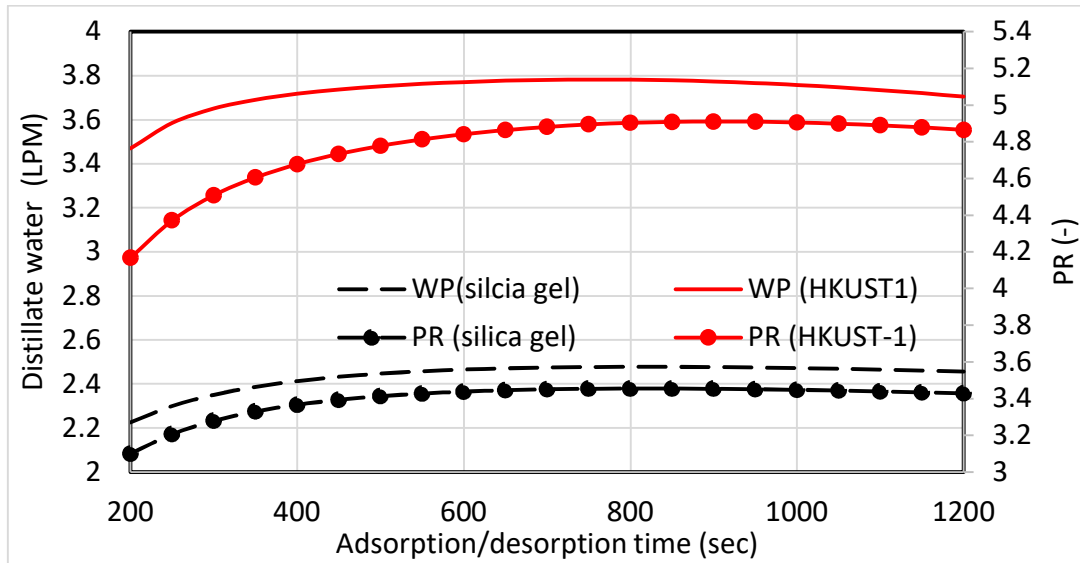


Figure 5-13; The effect of the adsorption/desorption time on the system performance (RD silica gel -water and HKUST-1-water).

Figures 5.14 shows the effect of switching time (precooling/preheating time) on the system's performance. The distillate water production increases as the switching time increases until it reaches a maximum value at an optimal switching time of 10 seconds. After that, it starts to decrease. When the switching time is shorter, time is not enough for the hot bed to be cooled sufficiently. Furthermore, a longer time is not favorable because the adsorber's pressure exceeds the evaporator's pressure. This means that a part of the adsorption process occurs at variable pressure. This leads to a reduction in distillate water production since the maximum adsorption takes place at the evaporator's pressure.

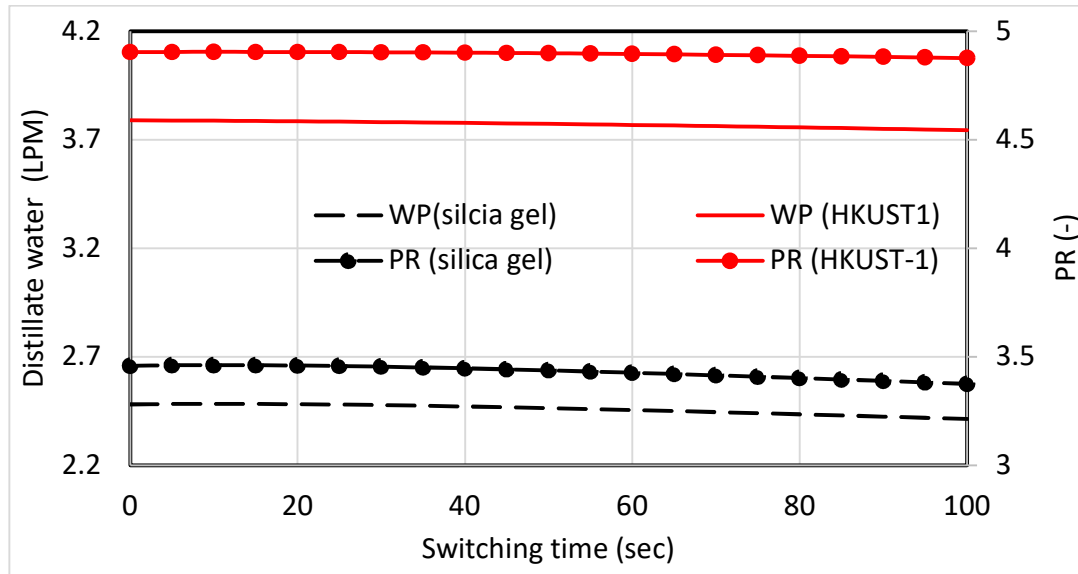


Figure 5-14: The effect of the switching time on the system performance (RD silica gel -water and HKUST-1-water).

In the conventional MED system, the top brine temperature and bottom brine temperature are limited to 70 °C and 40 °C, respectively. When the steam temperature is 50 °C, and the temperature difference between the effects is 2.5 °C, the maximum number of stages is four effects. When the MED system is connected to the AD cycle, the number of stages increased to nine effects that can further improve steam input energy utilization. This is due to the lowering of the last stage's temperature below the ambient temperature. Table 5.6 shows the stages' temperature and the temperature difference between the effects.

Table 5-6: The stages' temperature and the temperature difference between the effects.

Stage Number	Stage temperature (°C)	$T_i - T_{i+1}$ (°C)	
1	43.8	$T_i - T_{i+1}$ (°C)	
2	40.66	$T_1 - T_2$	3.14
3	37.79	$T_2 - T_3$	2.87

4	35.11	$T_3 - T_4$	2.68
5	32.55	$T_4 - T_5$	2.56
6	30.05	$T_5 - T_6$	2.5
7	27.54	$T_6 - T_7$	2.51
8	24.94	$T_7 - T_8$	2.6
9	22.19	$T_8 - T_9$	2.75

The solar system is used to heat the hot water flowing inside the desorption bed in order to reduce greenhouse gas emissions and improve the system performance ratio.

CHAPTER 6

THERMAL SOLAR SYSTEM FOR AD CYCLE

The system performance for liquid heating systems is calculated in terms of f , which is the fraction of the total heating load given by solar energy. Klein and Beckman originally created the f chart method. Figure 6.1 demonstrates the methods used for this study. In the f -chart method, the primary design variable is the collector area. The secondary variable can be storage capacity, collector type, thermal load, collector heat exchanger size, and flow rate. A typical heating system using liquid heat transfer fluids covered by the f -chart method is shown in Figure 6.2. This method typically uses water or an antifreeze solution as the heat transfer fluid for the collector. Either air or liquid can be heated by the collector. If the heat transfer fluid in the collector is not water, a heat exchanger between the collector and the tank is used. In this device, the energy collector loop is transferred directly or through an external heat exchanger from the collector fluid to the pre-heat storage tank.

The resulting correlations give as a function of two-dimensional variables the fraction of the monthly heating load (for space heating and hot water) given by solar energy (f); X : the ratio of collector losses to heating loads and Y : the ratio of absorbed solar radiation to heating loads.

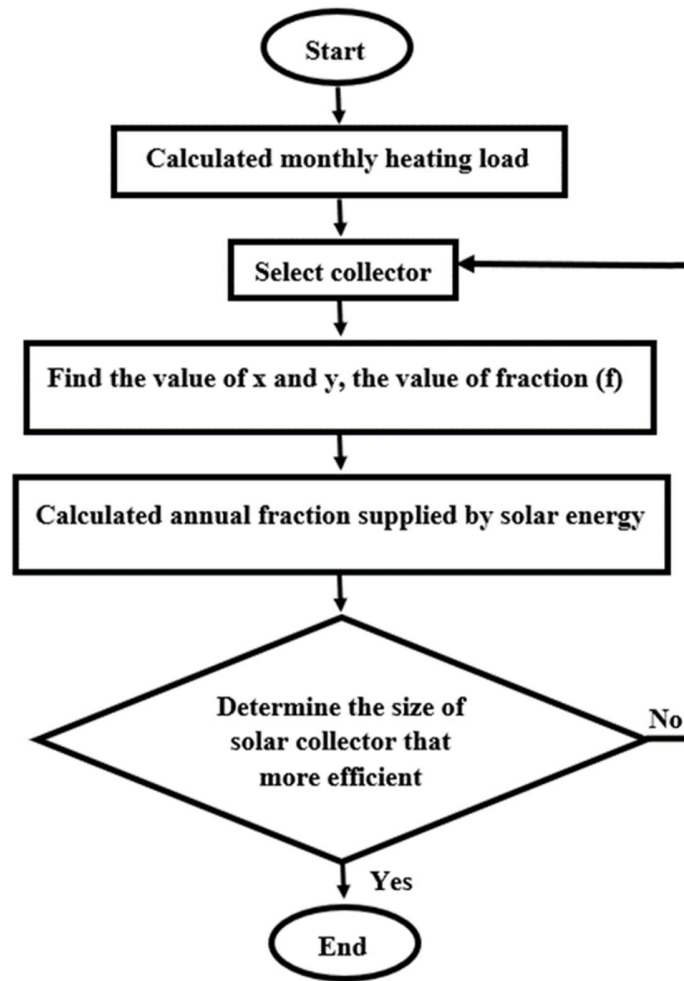


Figure 6-1: Standard water heating configuration only [85].

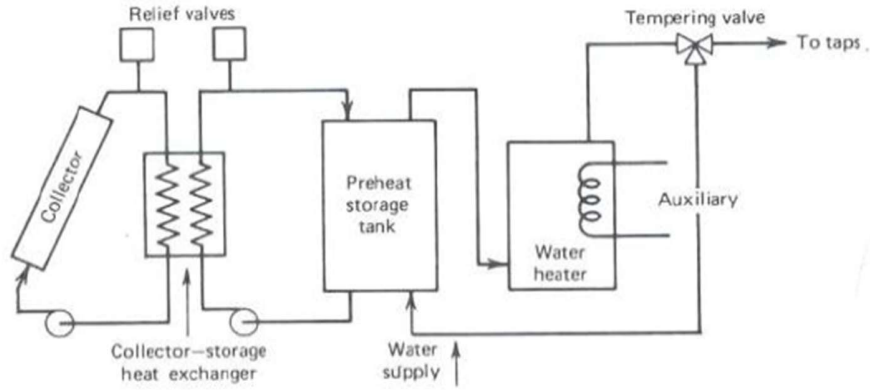


Figure 6-2: Overall methodology of the analysis [85].

To define the solar collector, which is the thermal output curve slope of the solar collector ($F_R U_L$ and intercept ($F_R * (\tau\alpha)_n$), the F-chart method needs two values. A flat-plate solar collector was chosen to be studied in this analysis. The characteristics of the water heating collector are $F_R U_L = 4.082 \text{ W} * \text{m}^{-2} * \text{C}^{-1}$ which refers to the collector efficiency on the collector's top, bottom, and edge surface, $F_R * (\tau\alpha)_n = 0.718$ and $(\tau\alpha) / (\tau\alpha)_n = 0.96$ and, as calculated from standard collector measurements, the collector heat exchanger correction factor $F'_R / F_R = 0.97$. A solar heating system is to be designed in Dharan, Saudi Arabia (Latitude of 26.3° N , Longitude of 50.2° E , elevation above sea level 17 m). The location and climate data are shown in Figure 6.3. Table 6.1 showed the range of variable collectors.

Location

	Unit	Climate data location	Facility location
Name		Saudi Arabia - Dhahran Intl (Mil)	Saudi Arabia
Latitude	N°	26.3	26.3
Longitude	E°	50.2	50.2
Climate zone		+B - Extremely hot - Dry	+B - Extremely hot - Dry
Elevation	m	17	17

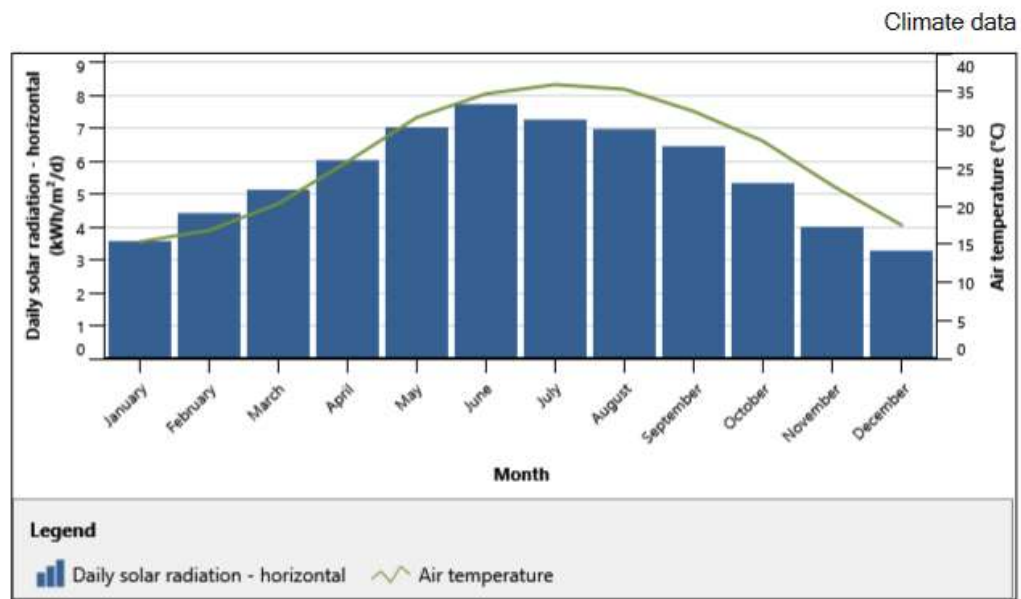


Figure 6-3: The location and climate data.

Table 6-1: Ranges of design variables used in developing f -charts for liquid and air systems [85].

Parameter	Value
Transmittance-absorptance product at normal incidence- ($\pi\alpha$) _n	0.6 - 0.9
Product of heat removal factor with heat exchanger and collector area - $F'_R A_C$	5 – 120 m ²
Collector heat loss coefficient - U_L	2.1 – 8.3 W/m ² °C
Collector slope from horizontal - β	30 - 90°
Overall energy loss coefficient and area product for a building used in a degree day space heating load - (UA) _h	83- 667 W/°C

The monthly cumulative load for hot water heating (L):

$$L = \dot{m}_{hw} C_{p,hw} (T_{hw,in} - T_{hw,out}) * \Delta t \quad (6.1)$$

The collector losses variables [85]:

$$X = \frac{A_C F_R U_L \left(\frac{F'_R}{F_R} \right) (T_{ref} - \bar{T}_a) \Delta t}{L} \quad (6.2)$$

The incident solar variables [85]:

$$Y = \frac{A_C F_R * (\tau\alpha)_n \left(\frac{F'_R}{F_R} \right) \left(\frac{\tau\alpha}{(\tau\alpha)_n} \right) (\bar{H}_T) N}{L} \quad (6.3)$$

The monthly solar fraction, f can be determined from the following relation with X and Y [85]:

$$f = 1.029Y - 0.065X - 0.245Y^2 + 0.0018X^2 + 0.0215Y^3 \quad (6.4)$$

The annual load heating supplied by solar energy:

$$F = \frac{\sum f_i L_i}{\sum L_i} \quad (6.5)$$

Where,

Δt = Total number of seconds in the month

A_C = Area of solar collector (m^{-2})

F_R = Collector heat removal factor (%)

U_L = Collector overall energy loss coefficient ($W * m^{-2} * C^{-1}$)

F'_R = Collector-heat exchanger efficiency factor (%)

$\tau\alpha$ = Monthly average transmittance-absorptance product (%)

$(\tau\alpha)_n$ = Normal transmittance-absorptance product (%)

T_{ref} = Reference temperature (100 °C)

$\bar{T}_a$ = Monthly average ambient temperature (°C)

$\bar{H}_T$ = Monthly averaged daily radiation incident on collector surface per unit area (MJ*m⁻²)

N = Number of days in the month.

The monthly cumulative load for hot water heating system with mass flow rate of 1.52 kg/s and an outlet temperature of 85 °C, an inlet temperature of 75 °C, and operating 24 hours per day is given as:

$$\begin{aligned} L &= \dot{m}_{hw} C_{p,hw} (T_{hw,in} - T_{hw,out}) * \Delta t = 1.52 * 4180 * (85 - 75) * 3600 * 24 \\ &= 5.48 \times 10^9 \text{ J} = 5.48 \text{ GJ} \end{aligned}$$

This power can be obtained from 8 flat plate collectors that provide a solar fraction of 47.3% and a payback period of 9.5 years or 15 evacuated tube collectors that provide a solar fraction of 56.7% with a payback period of 14.5 years.

The solar collector selected for this study is glazed flat plate solar collector manufactured by Caleffi with model number SolarFlat NAS10408A. The aperture area per solar collector is 2.8 m² and total number of collectors is six. Table 6.2 summarized the necessary load provided by the 16.8 m² solar thermal system (by applying the f-chart method).

Table 6-2: Monthly and Annual Performance of Liquid Heating System.

Month	Number of days (Days)	Air temperature (°C)	Averaged daily radiation incident (MJ/m ²)	L (GJ)	X	Y	f	fL (GJ)
January	31	15.3	12.852	5.48	2.75	0.82	0.52	2.87
February	29	16.8	15.912	5.48	2.53	0.95	0.62	3.39
March	31	20.3	18.468	5.48	2.59	1.17	0.75	4.10
April	30	25.8	21.708	5.48	2.33	1.33	0.85	4.64
May	31	31.6	25.308	5.48	2.22	1.61	0.97	5.34
June	30	34.7	27.828	5.48	2.05	1.71	1.03	5.62
July	31	35.9	26.136	5.48	2.08	1.66	1.00	5.50
August	31	35.3	25.092	5.48	2.10	1.59	0.98	5.35
September	30	32.4	23.22	5.48	2.13	1.43	0.90	4.94
October	31	28.5	19.188	5.48	2.32	1.22	0.79	4.32
November	30	22.7	14.4	5.48	2.43	0.89	0.59	3.21
December	31	17.5	11.808	5.48	2.68	0.75	0.48	2.64
Total				65.8				51.93

The fraction of the annual heating load supplied by solar energy:

$$F = \frac{\sum f_i L_i}{\sum L_i} = \frac{51.93}{65.8} = 0.79$$

RETscreen simulation software was used to design the solar system used to heat the hot water flowing inside the adsorption cycle (desorption bed). The simulation of this load with

the same solar collector model and design details showed that the solar fraction was 0.765. The standard error between the RETScreen simulations and the f-Chart results was about 3.3%.

A simulation study was performed to predict the amount of greenhouse gas emissions by non-renewable driven adsorption system. The base case uses electricity as a fuel with a fuel rate of 0.1 \$/kWh. The proposed case uses natural gas as a fuel with a fuel rate of 0.4 \$/m³. Table 6.3 showed the fuel consumption, fuel cost and greenhouse gas emissions (GHG emission) for both cases.

Table 6-3: The fuel consumption, fuel cost and greenhouse gas emissions (GHG emission) for both cases.

	Fuel consumption kWh	Fuel cost \$	GHG emission tCO ₂
Base case	18,641	1,864	15.1
Proposed case	9,825	370	1.8
Savings %	8,817 47.3%	1,494 80.2%	13.4 88.3%

A simulation study was performed to predict the best solar collector type to be used to provide the needed heating load with storage system. The selected solar collectors for this study were glazed flat plate collector (Caleffi - solarFlat NAS10406) and evacuated tube collector (Advanced Thermal Solar - ATS-20). The solar tracking mode is fixed with slope angle 25° and azimuth angle 0°.

Table 6-4: Comparison between flat plate and evacuated solar thermal collector.

	solarFlat NAS10406	ATS-20
--	--------------------	--------

Aperture area per solar collector (m ²)	2.197	0.943
Number of collectors	8	15
Total area (m ²)	18.8	27.1
Solar fraction (%)	47.3	56.7
Fuel saved (kWh)	8817	10571
Payback period (years)	9.5	14.5
Total initial cost (\$)	15800	26461
Storage capacity (Liters)	1318	1061

Therefore, a glazed flat plate solar collector (Caleffi - solarFlat NAS10406) will be used to provide the needed heating for the desorption process since the total initial cost and the required collector area are lower. The details are provided in appendix B.

Figures 6.4 and 6.5 show the effect of the number of solar collectors on the solar fraction and payback period for both flat plate and evacuated solar tube collector. The solar fraction increases as the number of solar collectors increase for both types. Since the system has a storage tank, then as the number of solar collectors increases, the additional energy will be stored to be used during the dark time. So, increasing the number will increase the stored energy. Therefore, this will reduce the required energy by auxiliary heaters. In the case of the flat plate, the payback period decreases as the number of collectors increases. However, in the case of the evacuated tube, it increases as the number increases.

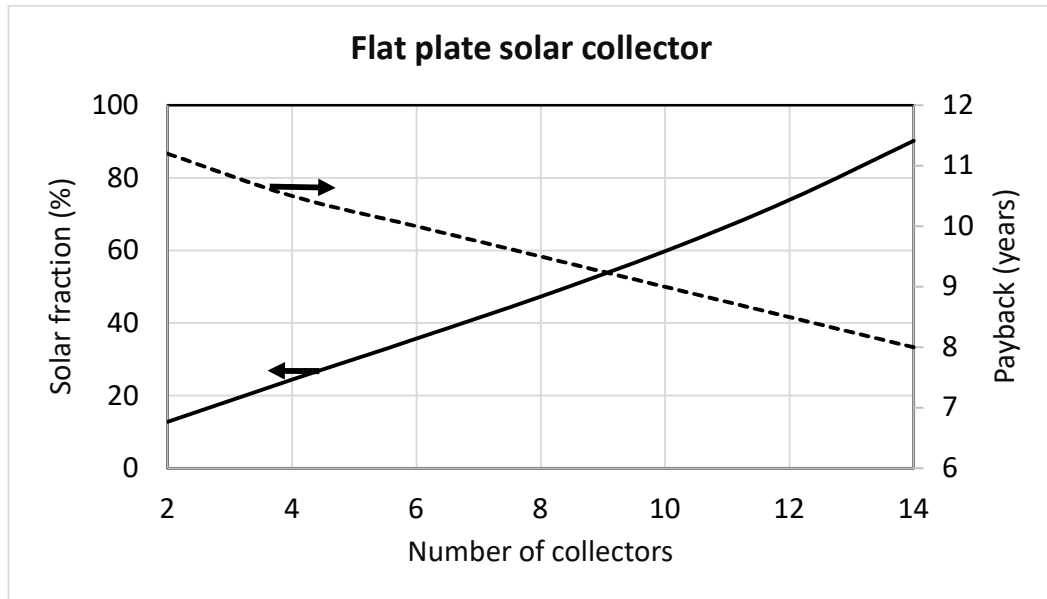


Figure 6-4: The effect of the number of flat plate solar collector on the solar fraction and payback period.

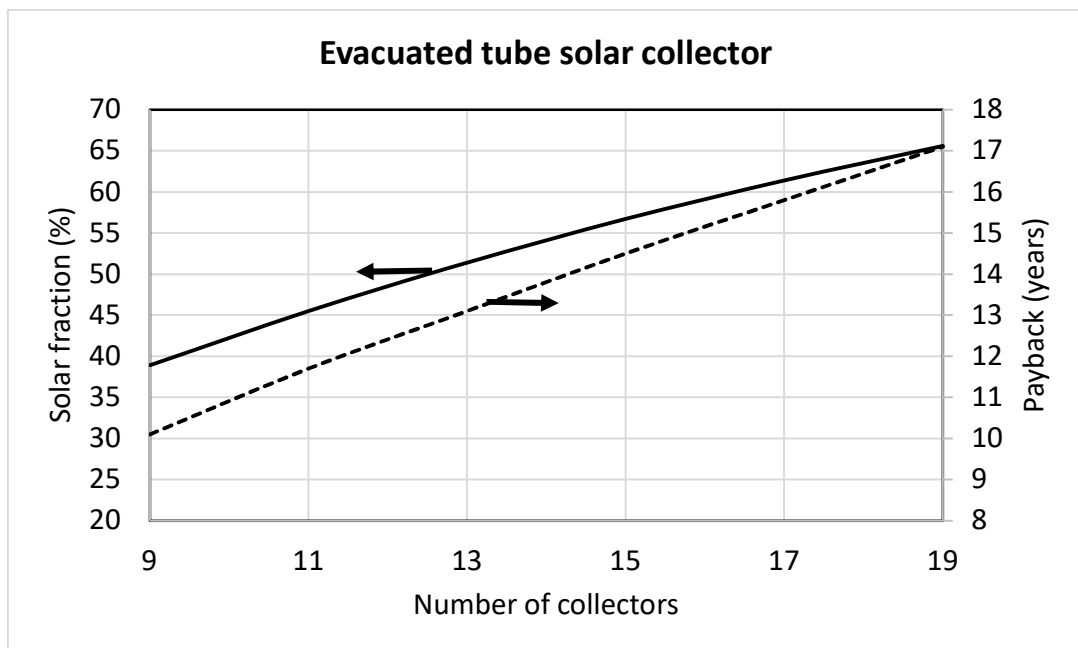


Figure 6-5: The effect of the number of evacuated tube solar collector on the solar fraction and payback period.

CHAPTER 7

CONCLUSION

This study predicts a two-bed adsorption chiller's cooling performance, including a mass recovery from the condenser to the evaporator, by utilizing a lumped transient mathematical model. This work's main objective was to study how the operating parameters affect the adsorption chiller performance. The second objective was to compare different adsorbent materials' performance (zeolite 13X, HKUST-1, aluminum fumarate, MIL-100(Fe), and silica gel RD-2060) at similar operating conditions. The two most critical parameters to predict a cooling system's performance are the COP and the cooling capacity. The major findings of this study can be summarized in the following points:

- The impact of the regeneration temperature (the temperature of the inlet of hot water) is more pronounced in the zeolite 13X due to the higher water uptake.
- The impact of the change in operating temperatures of the hot, chilled, and cooling water has a more noticeable impact on the adsorption chiller's performance than that of the mass flow rate of heat transfer fluids.
- As the adsorbent mass increases from 20 kg to 80 kg, the COP increases sharply.
- For silica gel, the maximum cooling power is 5.95 kW at an adsorption/desorption time of 800 seconds and a switching time of 25 seconds.
- The cooling power for HKUST-1 has an optimum value of 24 kW at an adsorption/desorption time of 600 seconds and a switching time of 45 seconds, resulting in a shorter cycle time compared to the silica gel.

- Using HKUST-1 as an adsorbent improves the COP and cooling capacity by 43% and 75%, respectively, compared to RD silica gel.

A transient mathematical model has been developed to predict the performance of the hybrid multi-effect parallel feed evaporation adsorption desalination system. The objective is to analyze the impact of operational conditions on the performance of the system. The second objective is to compare the performance of various adsorbent materials (silica gel RD-2060 and HKUST-1) at similar operating conditions of the entire system. The distillate water production and the PR are the two most critical performance parameters of desalination system. The major findings of this study can be summarized in the following points:

- The integrated system allowed lowering the MED last effect temperature below the ambient temperature due to water vapor uptake by AD adsorbent bed. As a result, the operational gap (temperature difference between the temperature of the steam and the temperature of the last effect) increases. Thus, more effects can be inserted, resulting in more distillate water production and better recovery of heat input.
- Using the HKUST-1 instead of the silica gel as adsorbent material results in reducing the last effect temperature by 35 %. This led to the use of more effects in the MED system compared with the case of using silica gel in the adsorption sub-system. Hence, a better system performance can be achieved.
- The HKUST-1 system improves distillate water production and PR by 40 % and 60 %, respectively.
- The MED-AD average distillate water production increases by two to three times compared to the conventional MED cycle.

- Total distillate water increases with an increase in the steam mass flow rate due to the higher evaporation rate resulting from higher energy input to the first effect.
- If the temperature of the hot water inlet rises, the output of distillate water increases as the rate of desorption increases.
- Distillate water production increases as the adsorption/desorption period increases until the maximum value is reached at an optimal adsorption/desorption time of 750 seconds. Then, it starts to decrease. Therefore, an optimum adsorption/desorption time do exist, and it should be considered in design/operation of the integrated system.

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APPENDIX A: OVERALL HEAT TRANSFER

COEFFICIENT CALCULATION

The used overall heat transfer coefficient in the model is calculated using El-Dessouky correlation (equation 4.31). The following are the heat resistance equations to calculate the overall heat transfer coefficient to find the percentage of error in the El-Dessouky correlation.

The overall heat transfer coefficient based on the outside surface area U_o is related to the individual thermal resistance by the following expression:

$$\frac{1}{U_o} = \frac{1}{h_i} * \frac{r_o}{r_i} + R_{f,i} \frac{r_o}{r_i} + \frac{r_o \ln \left(\frac{r_o}{r_i} \right)}{k_w} + R_{f,o} + \frac{1}{h_o} \quad (A.1)$$

Where h is the heat transfer coefficient, R_f is the fouling resistance, k_w is the thermal conductivity of tube material, and r is the radius. The subscripts i and o refer to the inner and outer tube surface, respectively.

Shell side:

Han and Fletcher [7] developed the following experimental correlation to calculate the boiling heat transfer coefficient h_o for thin water film flowing over the outside of smooth horizontal tubes:

$$h_o \left(\frac{\mu^2}{\rho^2 g k^3} \right)^{\frac{1}{3}} = 0.0004 \text{Re}^{0.2} \text{Pr}^{0.65} q''^{0.4} \quad (\text{A.2})$$

In the above equation Re and Pr are Reynolds and Prandtl numbers, respectively, q'' is the heat flux, μ is the viscosity, ρ is the density, and k is the thermal conductivity of the fluid.

Tube side:

Shah [7] correlation used to calculate the heat transfer coefficient of condensation inside a horizontal tube:

$$\frac{h_i}{h_f} = 1 + \frac{3.8}{Z^{0.95}} \quad (\text{A.3})$$

The parameter Z is defined as:

$$Z = \left(\frac{1}{x} - 1 \right)^{0.8} \text{Pr}^{0.4} \quad (\text{A.3})$$

Where x is the vapor mass fraction. The local superficial heat transfer coefficient h_f is calculated from:

$$h_f = h_l (1 - x)^{0.8} \quad (\text{A.4})$$

Where h_l is the heat transfer coefficient, when all the flowing mass is liquid, and is calculated by the well-known Dittus-Bolter equation:

$$h_i = 0.023Re^{0.8}Pr^{0.4} \left(\frac{k_f}{d_i} \right) \quad (A.5)$$

In the above equation Re and Pr are Reynolds and Prandtl numbers, respectively, d_i is the inside diameter and k is the thermal conductivity of the fluid

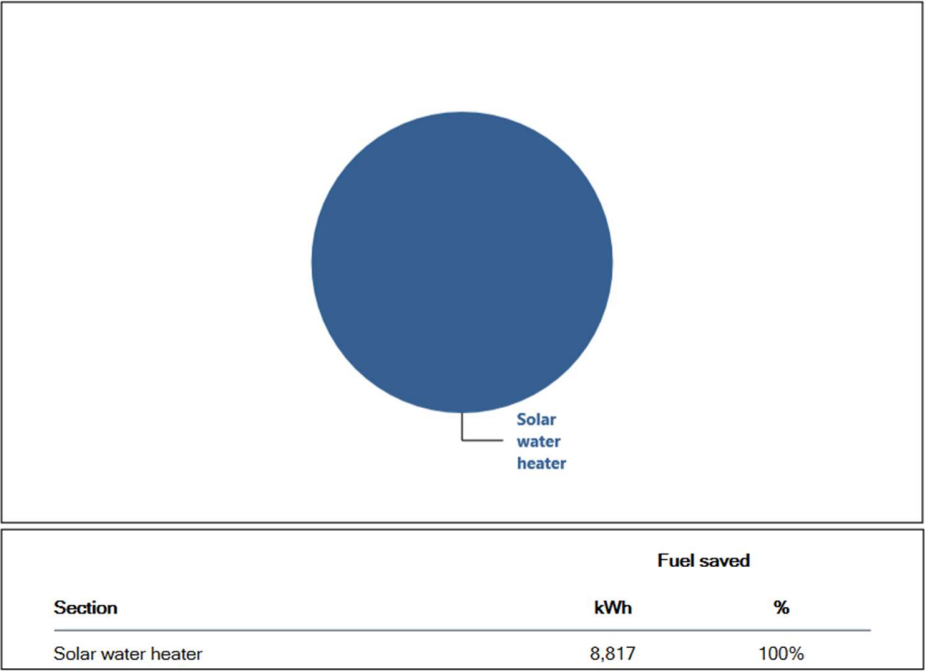
The calculated U-value using El-Dessouky correlation was 2.498 Kw/m²C, where it was 2.62 Kw/m²C using heat resistance equations with 4.7 % of error, which is acceptable in the problem solution.

APPENDIX B: UTILIZATION OF SOLAR ENERGY TO PROVIDE THE REQUIRED ENERGY TO RUN THE SYSTEMS

Use of Flat plate Collectors to provide necessary heating

Solar water heater		
Type		Glazed
Manufacturer		Caleffi
Model		SolarFlat NAS10406
Gross area per solar collector	m ²	2.355
Aperture area per solar collector	m ²	2.197
Fr (tau alpha) coefficient		0.706
Fr UL coefficient	(W/m ²)/°C	4.91
Temperature coefficient for Fr UL	(W/m ²)/°C ²	0
Number of collectors - suggested		8
Number of collectors		8
Solar collector area	m ²	18.8
Capacity	kW	12.3
Balance of system & miscellaneous		
Storage	yes/no	Yes
Storage capacity / solar collector area	L/m ²	75
Storage capacity	L	1,318
Initial costs	\$	14,800
Summary		
Electricity - pump	kWh	0
Energy saved	kWh	8,817
Solar fraction	%	47.3%

Fuel saved



Financial parameters

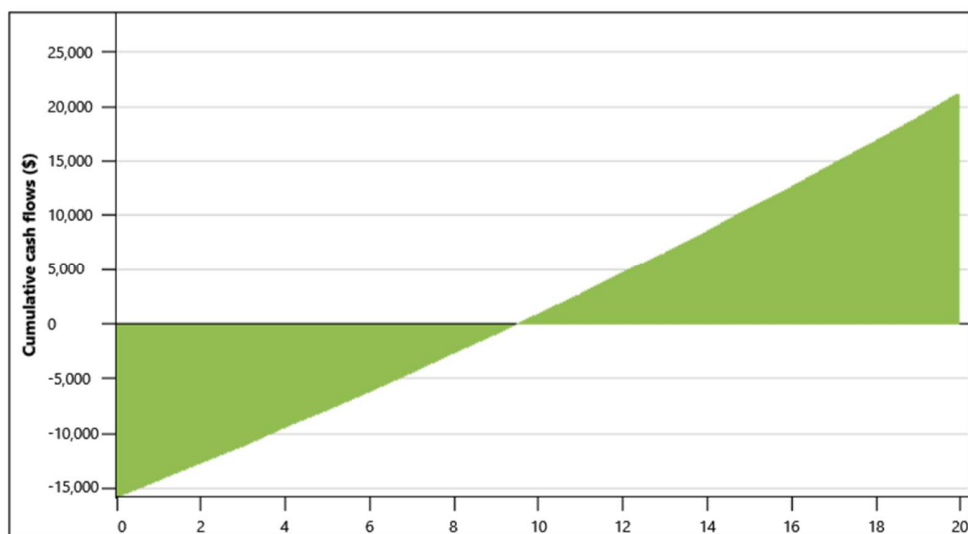
Inflation rate	%	2%
Project life	yr	20

Costs | Savings | Revenue

Initial costs			
Incremental initial costs	100%	\$	15,800
Total initial costs	100%	\$	15,800
Yearly cash flows - Year 1			
Annual costs and debt payments			
O&M costs (savings)		\$	0
Fuel cost - proposed case		\$	370
Debt payments		\$	0
Total annual costs		\$	370
Annual savings and revenue			
Fuel cost - base case		\$	1,864
GHG reduction revenue		\$	0
Other revenue (cost)		\$	0
Total annual savings and revenue		\$	1,864
Net yearly cash flow - Year 1		\$	1,494

Financial viability

Pre-tax IRR - assets	%	9.2%
Simple payback	yr	10.6
Equity payback	yr	9.5

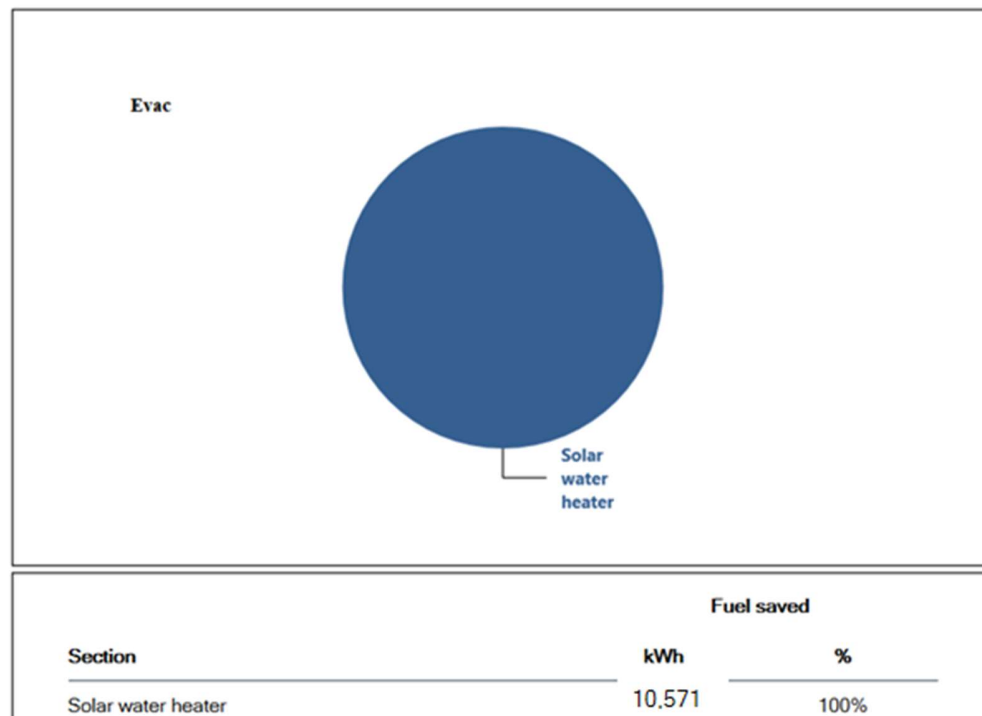


Use of Evacuated Tube Collectors to provide necessary heating

Solar water heater	
Type	Evacuated
Manufacturer	Advanced Thermal Solar
Model	ATS-10
Gross area per solar collector	m ² 1.807
Aperture area per solar collector	m ² 0.943
Fr (tau alpha) coefficient	0.371
Fr UL coefficient	(W/m ²)/°C 1.225
Temperature coefficient for Fr UL	(W/m ²)/°C ² 0
Number of collectors - suggested	15
Number of collectors	15
Solar collector area	m ² 27.1
Capacity	kW 9.9
Balance of system & miscellaneous	
Storage	yes/no Yes
Storage capacity / solar collector area	L/m ² 75
Storage capacity	L 1,061
Initial costs	\$/m ² -aperture 1,425
Summary	
Electricity - pump	kWh 0
Energy saved	kWh 10,571
Solar fraction	% 56.7%

End-use

Fuel saved



Financial viability

Financial parameters

Inflation rate	%	2%
Project life	yr	20

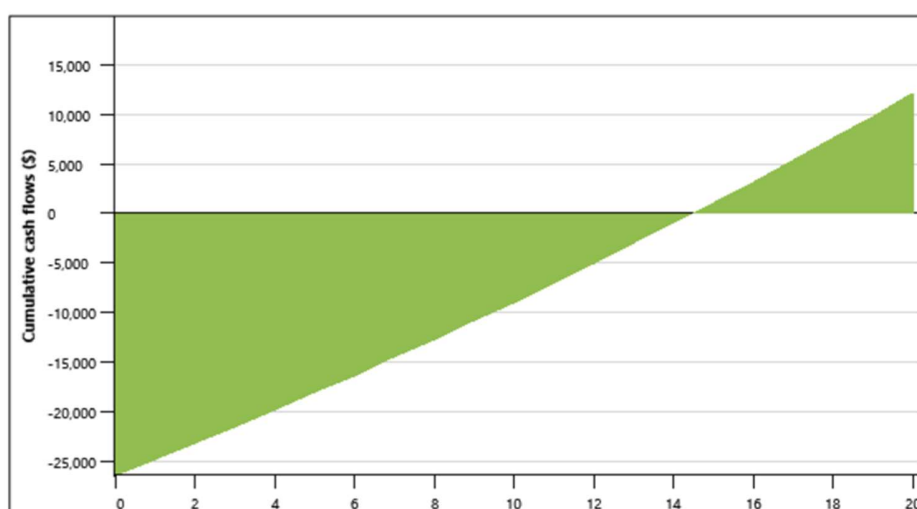
Costs | Savings | Revenue

Initial costs			
Incremental initial costs	100%	\$	26,461
Total initial costs	100%	\$	26,461
Yearly cash flows - Year 1			
Annual costs and debt payments			
O&M costs (savings)		\$	0
Fuel cost - proposed case		\$	304
Debt payments		\$	0
Total annual costs		\$	304
Annual savings and revenue			
Fuel cost - base case		\$	1,864
GHG reduction revenue		\$	0
Other revenue (cost)		\$	0
Total annual savings and revenue		\$	1,864
Net yearly cash flow - Year 1		\$	1,560

Financial viability

Pre-tax IRR - assets	%	3.7%
Simple payback	yr	17
Equity payback	yr	14.5

Cumulative



Vitae

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