THERMODYNAMIC ANALYSIS OF A SOLAR ASSISTED ADSORPTION COOLING SYSTEM

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CERTIFICATION

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

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ABSTRACT

In this study, an alternative way for using the solar energy in a cooling system was examined. For this research, first an adsorption refrigeration model was designed with solar energy system. The isoster lines are plotted and the required heat of desorption for the adsorber calculated. The COP values were calculated than the solar energy required was determined for this purpose. Finally the system was compared with the conventional cooling system. The study based on only the thermal equations. In this study the calculations were done in an Excel format programming. With this system it is tried to be shown that these types of systems are the future's progressive technology for cooling with all their benefits.

Keywords: Adsorption, Cooling, Solar assisted, Chemical Heat pump, Refrigeration, Isotherms curves

ÖZET

Bu çalışmada, güneş enerjisinden yaralanılarak soğutma amacıyla kullanılabilecek olan alternatif bir yol incelenmiştir. Bu çalışma için, öncelikle güneş enerjisi yardımıyla adsorpsiyonlu (katı soğurmalı) soğutma yapan bir model tasarlanmıştır. İzotermik grafikleri çizilmiş ve desorpsiyon (geri salınım) için gerekli ısı miktarı hesaplanmıştır. Gerekli ısıyı toparlamak için gereken güneş paneli yüzeyi ve sistemin verimlilik katsayısı hesaplanmıştır. Son olarak sistem günümüzde yaygın olan soğutma teknikleri ile kıyaslanmıştır. Hesaplamalar excel formatında yapılmıştır. Bu çalışma ile, bu çeşit soğutma sistemlerinin tüm faydaları ile birlikte geleceğin soğutma sistemleri olduğu gösterilmeye çalışılmıştır.

Anahtar Kelimeler: Adsorpsiyon (Katı Soğurma), Soğutma, Güneş Enerjisi Yardımlı ,Kimyasal Isı Pompaları, Dondurma, İzoterm Eğrileri

ACKNOWLEDGMENTS

The study that named “THERMODYNAMIC ANALYSIS OF A SOLAR ASSISTED ADSORPTION COOLING SYSTEM” is criticized within the every way of the system components, structural schema and thermal analysis. In the study it is greatly focused on the adsorption thermal analysis and on the solar source power of the cycle. For calculating the outputs only the water supply of the solar collector is going to be assumed. The other inputs have been carefully selected with respect to the reference tables.

After designing of the system, some results and recommendations have been done for the sake of the further studies. The sufficient outputs are criticized clearly in the realistic ambient conditions. Also some of the inputs are extremely selected to understand the differences and to show the faults that could possibly be occur in the operating conditions.

I am firstly sincerely very thankful to my supervisor Assist. Prof. Dr. Cemal GÖVSA, who presents all his valuable helps and contributions to my MSc Thesis study and in my higher education period of Mechanical Engineering. I am also sincerely very thankful to my Instructors Prof. Dr. İlkay SALİHOĞLU, Assist. Prof. Dr. Ali EVCİL, Assist. Prof Dr. Hüseyin ÇAMUR in Near East University Mechanical Engineering Department for their valuable labors on me. I am also very grateful to Assist. Prof. Dr. Moghtada Mobedi and Mechanical Engineering Research Assistant Zeynep ELVAN YILDIRIM from İzmir Institute of Technology Department of Mechanical Engineering for their orientations and advices. I am also very grateful to my whole family for their spiritual and financial supports. I also thank to my colleagues for their unlimited confidences to my studies success.

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ABBREVIATIONS

AFP	Advanced flat plate
CLFR	Compact linear Fresnel factor
COP	Coefficient of performance
CPC	Compound parabolic collector
CTC	Cylindrical trough collector
ETC	Evacuated tube collector
FPC	Flat plate collector
HFC	Heliostat field collector
ICPC	Integrated compound parabolic collector
ICS	Integral collector storage
PDR	Parabolic dish reflector
PTC	Parabolic trough collector
LFR	Linear Fresnel reflector

Latin Symbols	**Definition**
h	Enthalpy
W	Work
P	Pressure
P_a	Pressure at point a
P_b	Pressure at point b
P_c	Pressure at point c
P_d	Pressure at point d
T	Temperature
T_a	Temperature at point a
T_b	Temperature at point b
T_c	Temperature at point c

T_d	Temperature at point d
T_C	Temperature at intermediate source-II
T_Z	Temperature at high source
T_L	Temperature at low source
T_A	Temperature at intermediate source-I
T_{eq}	Equivalence temperature (K)
T_∞	Average heating/cooling fluid temperature (K)
Q	Heat
$\dot{Q}$	Heat flux
$\dot{Q}_{ab}$	Heat rate of isosteric heating process (kJ)
$\dot{Q}_{bc}$	Heat rate of isobaric desorption process (kJ)
$\dot{Q}_{cd}$	Heat rate of isosteric cooling process (kJ)
$\dot{Q}_{da}$	Heat rate of isobaric adsorption process (kJ)
$\dot{Q}_L$	Heat rate amount from or to the low temperature source
$\dot{Q}_Z$	Heat rate amount from or the high temperature source
$\dot{Q}_C$	Heat rate amount from or to the intermediate temperature source-II
$\dot{Q}_{CC}$	Heat rate amount from or to the intermediate temperature source-I
x	Mass ratio of the adsorption
x_1	Mass ratio maximum
x_2	Mass ratio minimum
x_a	Mass ratio on point a
x_b	Mass ratio on point b
x_c	Mass ratio on point c
x_d	Mass ratio on point d
x_{ave}	Average value of the mass ratio
C_p	Specific heat (kJ/kgK)
$\dot{m}$	Mass flow rate

m	Mass (kg)
ΔH_a	Heat of adsorption(kJ/kg adsorbate)
ΔH_v	Heat of vaporization(kJ/kg adsorbate)
t	Time (s)
t*	Non dimensional time (t/τ)
r	Radial coordinate (m)
r*	Non-dimensional radial coordinate (r/R)
D	Solid mass diffusivity (m^2/s)
D*	Non-dimensional solid-side mass diffisuvity ($D\tau/R^2$)
R	Effective particle radius of adsorbent (mm)
R^*_{th}	Thermal resistance ratio
A_c	Collector aperture area
A_r	The receiver area
A_s	Surface area between insert and external fluid (m^2)
A_b	Total adsorbent surface for the collector, m^2
T_{wet}	Temperature of the wet adsorbent of the collector
s	Entropy
*	$m_i c_i/m_{ads(dry)} c_{ads(dry)}$
C^*_r	$m_i c_i/m_a c_w \tau_i$
Co	Collector concentration ratio
d	Diameter (m)
a_0, b_0, c_0, d_0	Constants of isotherm formulas
b_0, b_0, b_0, b_0	Constants of isotherm formulas
c_1, c_2, c_3, c_4	Constants for collector efficiency formulas
k_1, k_2, k_3, k_4	Constants for collector efficiency formulas
c_0	Intercept efficiency
G	Solar radiation (W/m^2)
F_r	Heat removal factor
p	Perimeter of flow passage (m)
U	Overall heat transfer coefficient between insert and adsorbent (W/m^2K)

U_L	Overall solar collector heat transfer loss coefficient (W/m^2K)
X_c	Collector length (m)
G_b	Beam solar energy irradiaion(W/m^2)
h_{fr}	Heat transfer coefficient between the receiver and the fluid flowing within receiver
z	Coordinate

Greeks

τ	Mode operating time (s)
γ_1	$(\Delta Ha/C_{pb},K)$
γ_2	$(3/R)(\tau U/ \rho_a Cp_b)$
ρ	Density kg/m^3
λ	Adsorption bond factor
η	Efficiency
η_0	Optical efficiency
ω	Width of collector aperture (m)
κ	Gas constant for adsorbate kJ/kgK
u_0	$(647.27\text{-}T_b)$ (K)
τ^*_i	Non-dimensional time constant (τ_i/τ)
α_1, α_2	Constants for the saturation vapor pressure formulas
ε	Fraction of solar energy reaching surface that is absorbed, absorptivity

Subscripts

ref	Cooling, refrigeration
h	Heating
eva	Evaporator
con	Condenser
ba	Wet adsorbent

i	Insert or inlet
0	Outlet
max	Maximum possible value
min	Minimum possible value
α	Fraction of solar energy reaching surface that is adsorbed, adsorptivity
f	Fluid (water)
st	Storage tank
met	Metal
amb	Ambient
w	Water
z	Zeolite
sat	Saturation
vap	Vapor
ave	Average
sol	Solar
net	Net
bed	Adsorption bed
ads	Adsorbent
vap	Vapor(adsorbate vapor)
tot	Total
g	Regeneration

CHAPTER 1

INTRODUCTION

In our days and our lives with the effect of global heating; the refrigeration processes and most importantly air and room conditioning processes getting much important. Using compressors in the Vapor-Compression Mechanical Refrigeration Cycles have become an element for raising the air pollution. That is because the compressors require electricity for operating. That means, so much petrol consumed for the satisfaction of the electricity demand and that comes with the emission gases which are harmful for surroundings and for living health care. When the goal of the Kyoto Protocol is to enhance renewable energy utilization, and the main aim to reduce the products of coal and petrol combustion which are the main pollutants of the air, water and soil; it is definitely be needed some other techniques to satisfy mankind requirements in air conditioning.

Some other techniques are being tried for a healthy environment, when the mankind is using the refrigeration processes at the mean time. The main reason for developing some other techniques is; to eliminate the compressors which are located in many cycles. In this thesis it is going to be issued that a refrigeration cycle with an alternative way instead of compressor to use in the refrigeration and in the household air conditioning. The system is going to be developed with using adsorption tables and thermal formulas; also the thermodynamics of the system is going to be issued. The system will be driven by the solar power and the solar power requirement amount is calculated.

CHAPTER 2

LITERATURE SURVEY

2.1 Over view

The first foundations of mechanical refrigeration had been done by Prof. Dr. William Cullen from Glasgow University in 1755. He first started his investigations when he realized that his hand had been refreshed by ether and developed an ice maker that is working by the principle of vacuum technology. But the ice maker couldn't go any further from being a laboratory instrument. [1]

An engineer called by Jacop Perkins had gained the first license of practical ice maker machine in 1834. Also studied on a refrigeration system that was going to be operated in the situations without electricity and French Ferdinand Carre discovered the absorption system in 1858. In 1886 an engineer called Windhausen managed to decrease the refrigeration temperature down to -80 °C with a system that operates with CO_2. [1] After many investigations the Kelvinator Company had discovered the first automatically controlled refrigeration in 1918. [1]

2.2 The Classification of Basic Refrigeration Ways

- The vapor-compression cycle systems
- Chemical heat pumps
  - Absorption refrigeration cycle
  - Adsorption refrigeration cycle
- Thermoelectric refrigeration systems

- Vortex tube
- Paramagnetic refrigeration systems
- Sterling cycle
- Gas refrigeration cycle systems
- Vapor-jet refrigeration systems

2.2.1 The Vapor-Compression Refrigeration Cycle

Vapor-compression refrigeration cycles are the most common used refrigeration cycles for refrigeration. A typical vapor compression cycle has been given in Figure 2.1. As the refrigerant passes through the evaporator the heat transferred from the refrigerated space makes the refrigerant liquid vaporized. The refrigerant leaving the evaporator is compressed to a relatively high pressure and temperature also increases with assuming no heat transfer to the compressor or from the compressor. Next the refrigerant vapor passes from the condenser and it condenses. It happens a heat transfer from refrigerant to surroundings. Finally refrigerant fluid enters the expansion valve and expands to the evaporator pressure. “This process is usually modeled as a throttling process, for which the refrigerant pressure decreases in the irreversible adiabatic expansion, and there is an accompanying increase in the specific entropy. The refrigerant exits valve at liquid vapor mixture.” [4]

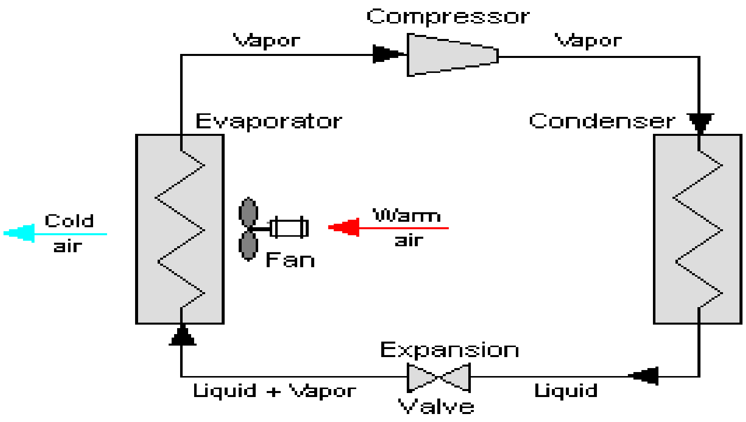


Figure 2.1 Typical single-stage vapor compression refrigeration cycle [40]

If the irreversibilities in the system are ignored which means there are no frictional pressure drops, the refrigerant flows at constant pressure through the two heat exchangers and if the heat transfer amount to the surroundings are ignored, the compression process is an isentropic process. [4] The T-s diagram has been given in Figure 2.2. “All of the processes in the above cycle are internally reversible except for the throttling process.” [4] The COP value for the refrigeration process is defined as the heat removal from the refrigerated space over net work input.

$$COP_{ref}=Q_i/W_{net,i}=(h_1-h_5)/(h_2-h_1) \tag{2.1}$$

“In ideal cycle, the refrigerant leaves the evaporator and enters the compressor as saturated vapor. In practice it may not be possible to control the state of the refrigerant so precisely.” [2] Because of this reason it will be much easier to design the system with the refrigerant is slightly superheated at the compressor inlet. This slight over design is for ensuring that the refrigerant is completely vaporized when it enters the compressor. In ideal case, the refrigerant is assumed to leave the condenser as saturated liquid at the compressor exit pressure.” In actual situations, however, it is unavoidable to have some pressure drop in the condenser as well as in the lines connecting the condenser to the compressor and to throttling valve. The refrigerant is sub cooled some before it enters the throttling valve.” [2]

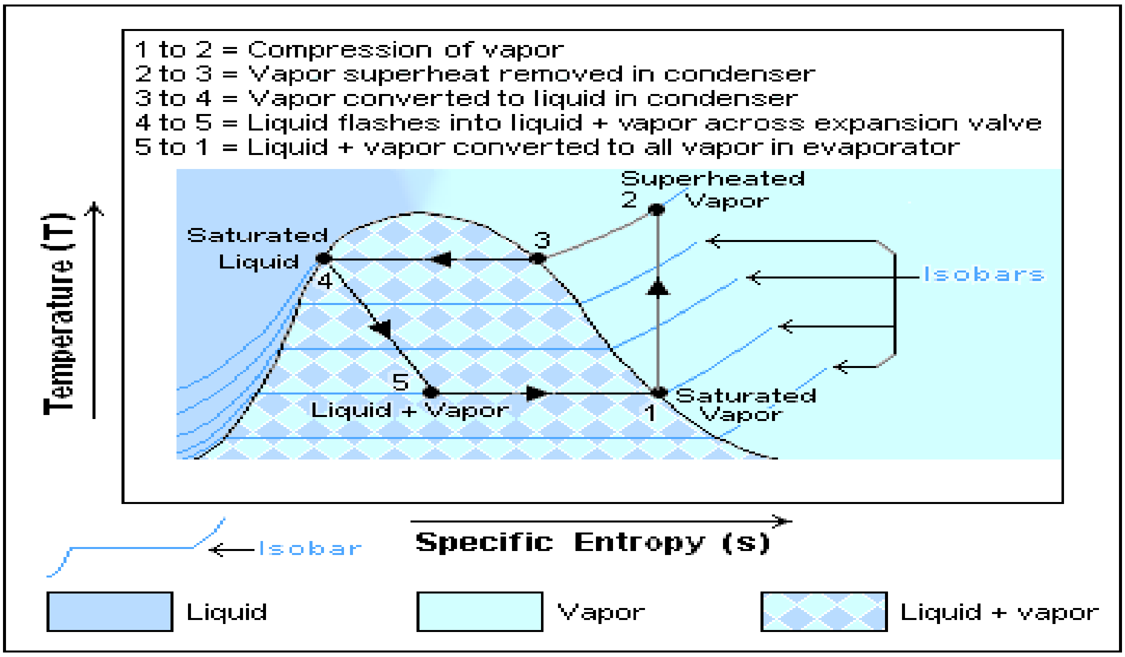


Figure 2.2 The T-s diagram of the ideal vapor-compression refrigeration cycle [40]

2.2.2 Chemical Heat Pumps

The classification of the chemical heat pumps can be seen in Figure 2.3. [18] The absorption and the adsorption heat pumps are the types of chemical heat pumps. As it is indicated in the Figure 2.3, the adsorption heat pumps are classified in the solid adsorption region. They can be direct fired or indirect fired for the purpose of the system. The adsorption and absorption cycles are the most useable methods for the solar assisted cooling and refrigeration purpose. They are classified under the chemical heat pumps because the circulation of the cycle is done by chemically instead of using a physically treatment way on the refrigerant liquid like using compressor.

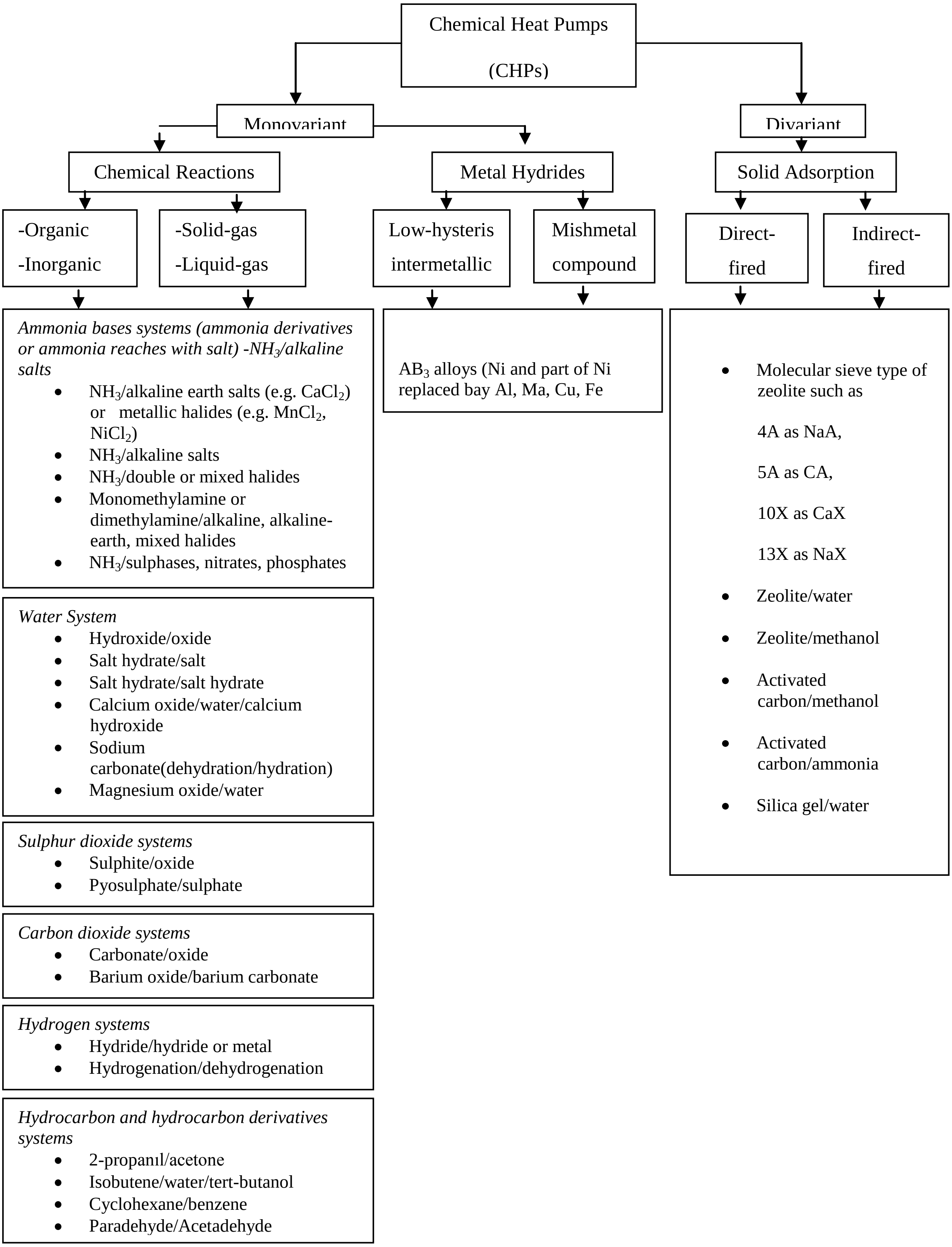


Figure 2.3 Classifications of chemical heat pumps [18]

2.2.2.1 Absorption Refrigeration Cycle

The absorption refrigeration cycle is different from the basic vapor compression cycle in two important ways. In the first way, the cycle has the nature of the compression process instead of compressing the vapor between the evaporator and the condenser with a compressor, the refrigerant of an absorption system is absorbed by a secondary substance called an absorbent, to from a liquid solution. “The liquid solution is then pumped to the higher pressure. Because the average specific volume of the liquid solution is much less than of the refrigerant vapor, significantly less work is required. Absorption refrigeration systems have the advantage that, relatively small work input required compared to vapor-compression systems.” [4] The other main difference between absorption and vapor-compression systems is retrieving the refrigerant vapor from the liquid solution before the refrigerant enters the condenser. This involves heat transfer from a relatively high-temperature source. Otherwise steam or waste heat that is used as the driven source of the system would be discharged to the surroundings and discharging the source heat without using it efficiently will be particularly economical. “Natural gas or some other fuel can be burned to provide the heat source, and there have been practical applications of absorption refrigeration using alternative energy sources such as solar and geothermal energy.” [4]

Liquid refrigerant is vaporized in the evaporator by absorbing heat from the volume which must to be cooled. The suction effect is necessary to draw the vapor through the system which is going to be accomplished by bringing the refrigerant into the absorber. [5] In the Figure 2.4 the NH_3-H_2O system is given. In this figure NH_3 is the refrigerant. In the absorber the refrigerant vapor dissolves and reacts with the water [2]. Absorption is an exothermic process, the released heat must be removed from this portion of the cycle [5]. The amount of refrigerant that can be dissolved in water inversely proportional to the temperature. There for it is necessary to cool the absorber to maintain its temperature as low as possible. [2] The liquid solution of refrigerant and water, which is rich in NH_3 is pumped from the absorber to the generator. In generator heat is added to the solution to vaporize some of the refrigerant out of solution. The vapor, which is rich in refrigerant, passes through a rectifier, which separates the water and returns it to the generator. In the further of the process the high-pressure NH_3 vapor continues its way through the rest of the cycle. The remained hot solution, which is weak in refrigerant, then passes through a regenerator, where it transfers some heat to the rich solution leaving the pump, and it is throttled to the absorber pressure. The pressure of the liquid refrigerant must also be reduced like in the vapor-compression cycle by passing through a throttling device before returning to the evaporator section. [5] The circulation for solar absorption refrigeration cycle diagram has been shown in Figure 2.4.

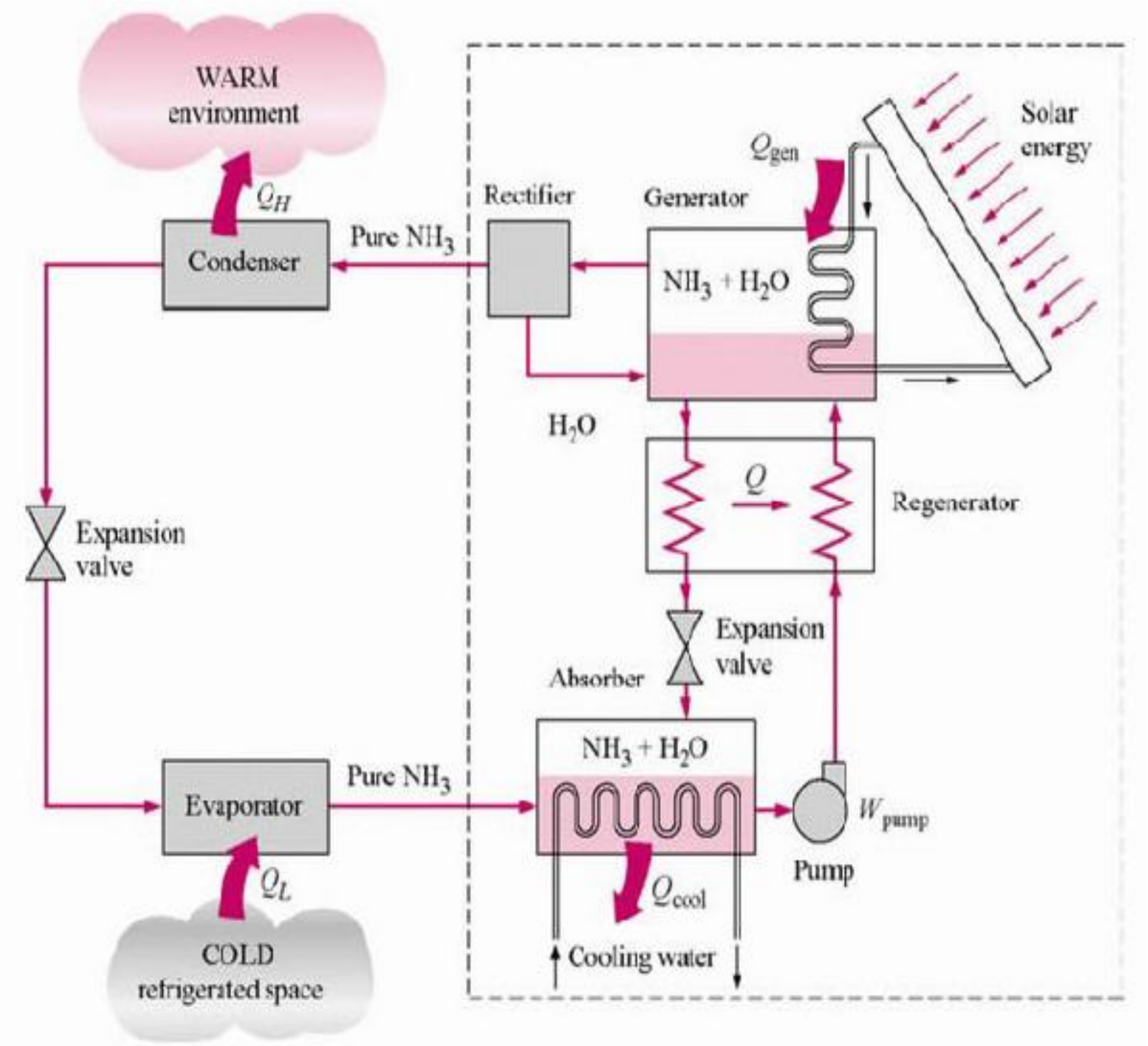


Figure 2.4 A simple absorption refrigeration cycle [2,4]

2.2.2.2 Adsorption Refrigeration Cycle

There are basically two types of adsorption heat pump cycles. The type which is called as heat amplifier is not widely used in industrial applications. They have been realized to recover heat from refuse incineration plants, notably in Sweden and Denmark. “In first type, high temperature heat source is used to transfer heat from low temperature heat source to the intermediate heat source. The amount of heat that is delivered to the intermediate heat source is equal to the sum of high temperature’s heat and low temperature’s heat that are entering to the heat pump.” [17]. The other type has the same main components and working principle as heat amplifier adsorption heat pumps. With this type of heat pumps waste heat can be upgraded, virtually without using an external drive energy. Waste heat of a medium temperature (i.e. between the demand level and the environmental level) is supplied to the evaporator and generator. Useful heat of a higher temperature is given off in the adsorber. “The medium temperature should be higher than condenser temperature.” [17] Also two chamber adsorption cooling system can be examined for strengthening the adsorption refrigeration systems works intermittently. The adsorbate which is being evaporated in the evaporator adsorbed by the adsorbent, while it is being absorbed, it happens no operation in the condenser. It also happens no operation in the evaporator while the refrigerant is being condensed in the condenser. Intermittently operating is the most effective disadvantage for the system. The double reacted adsorption systems neutralize this disadvantage.

To analyze the cycle, the cycle must be plotted on a Clapeyron Diagram. The various characteristics could be obtained by the Clapeyron diagrams of the chemical heat pumps and it is clearly be seen the isosteric and isothermic characteristics of the cycle. “By plotting the system’s cycle along the equilibrium lines of the clapeyron diagram, the operating pressure, the range of temperature upgrade, mass of the working pairs required, amount of power consumed and heat released, etc. could be predicted.” [18] The working of an intermittent solid adsorption cycle can be also represented in a Clapeyron Diagram by knowing the relation between vapor pressure of working fluid (e.g water) and the adsorbent (e.g. NaX zeolite) temperature equilibrium [18].

Adsorption refrigeration cycle is not too much different from absorption refrigeration cycle. It contains mainly condenser, evaporator, expansion valve, adsorbent bed, some adsorbent and some adsorbate. “The main adsorbent and adsorbate pairs are Activated Carbon/Ammonia, Silica gel/methanol, Silica gel/water and Zeolite/water. In this system the compressor is replaced by a thermal compressor which is operated by heat instead of a mechanical energy.” [7] In evaporator the adsorbate evaporates by having heat from surroundings. Adsorbate adsorbed by the dry adsorbent in the adsorbent bed. The heat is transferred in to adsorbent bed for desorption. The desorbed material continues cycling in to the condenser. After condensation adsorbate expands in the expansion valve and arrives to evaporator. The ideal adsorption cooling cycle on a schematic vapor pressure diagram is given in Figure 2.5 and also a simple adsorption cooling unit schema has been given in Figure 2.6.

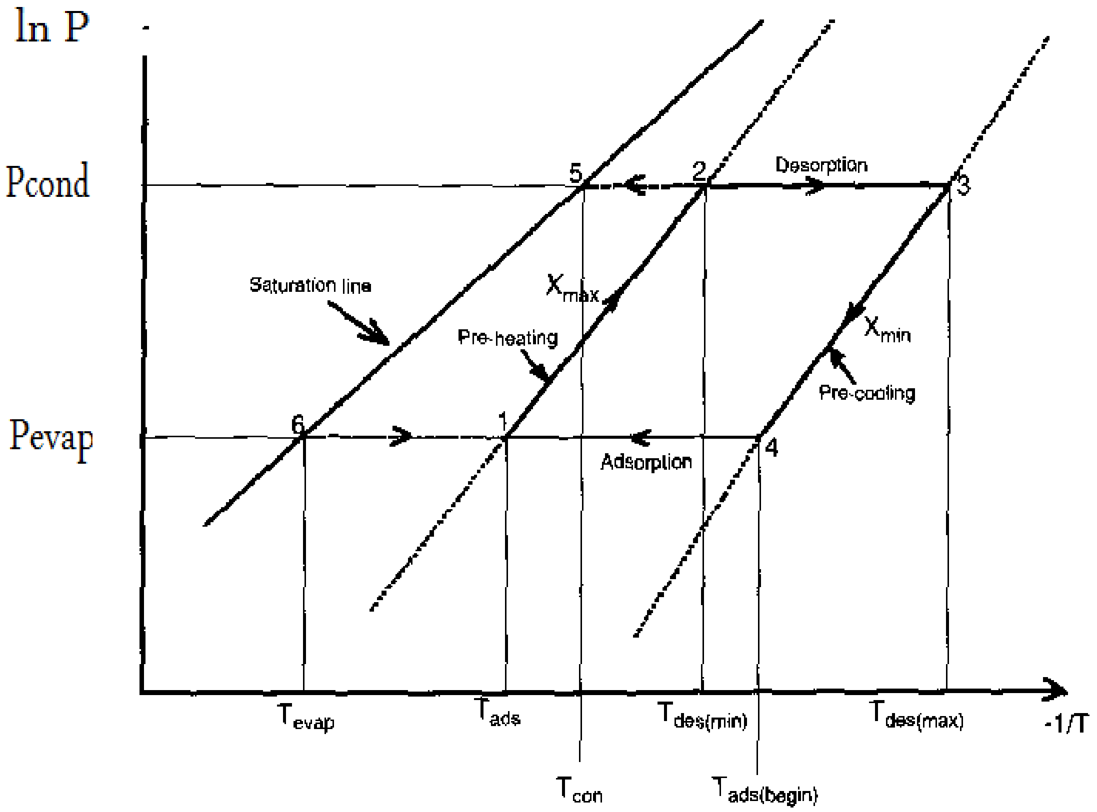


Figure 2.5 The ideal adsorption cooling cycle on a schematic vapor pressure diagram [10]

In Figure 2.5, from point 1 to point 2 the adsorbent bed's temperature increases from T_1 to T_2 with transferring heat from outside in to the system. In preheating the vapor pressure is steady without desorption. From point 2 to 3 the heat transfer to the adsorbent bed goes on. But in this period from point 2 to 3, desorption begins and the desorbed water vapor condenses in steady condensation pressure. Point 3 to 4 region begins after the maximum bed temperature reached and desorption finished the bed temperature decreases to the T_4. An expansion valve also helps the pressure decrease. From 4 to 1 the heat transfer from bed to surroundings goes on. The adsorbate that evaporates in the evaporator adsorbed by the adsorbent. This is a heat loosing process.

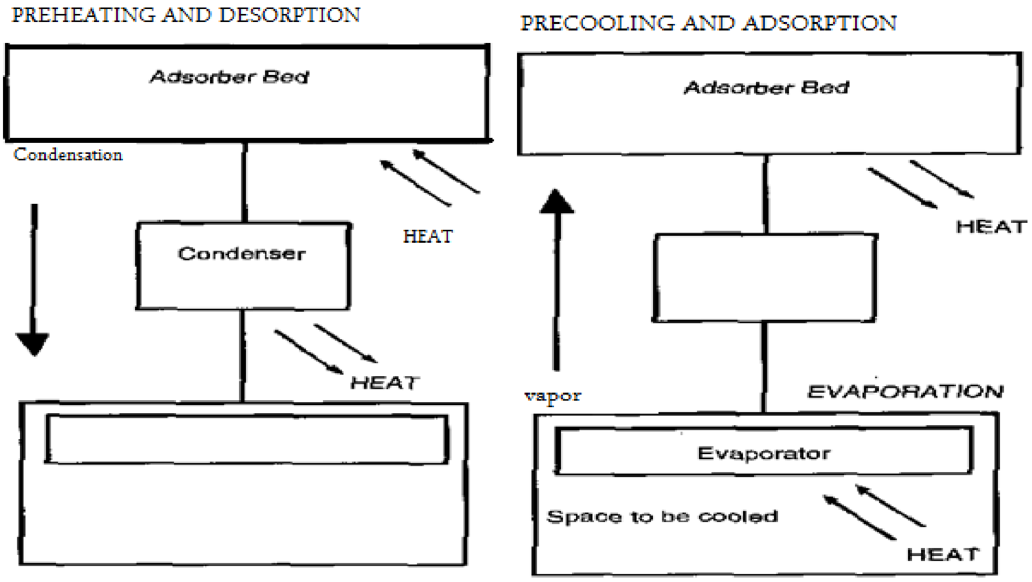


Figure 2.6 A simple adsorption cooling unit schema [10]

The system's working principles, components and thermodynamic Analysis of the system will be much sufficiently analyzed in Chapter 3 and Chapter 4.

CHAPTER 3

METHODOLOGY

3.1 The Adsorption and Adsorption Pairs

Vapor or vapor liquid adsorption specified as a material holt on the surface of a solid material by a physical or chemical reaction. Adsorption is classified in two types as, physical adsorption and chemical adsorption. In chemical adsorption the chemical bond (covalent bond) holds the adsorbate on the absorbent surface. The main characteristic of the chemical adsorption is; it is an endothermic process and it is not reversible which means it will not happen any desorption in the process. In physical adsorption the physical bond (Van der Waals, dipol-dipol) holds the adsorbate on the adsorbent surface The physical adsorption's efficiency is going to be decreased with increasing the environments temperature because the physical adsorption process is an exothermic process and this released heat called the adsorption heat. The process is reversible, desorption occurs when the heat applied to the absorbent and this is the heat which is going to be acquired from the solar collector. Like it had been said before the adsorption process that is used in the heat pump systems are physical adsorption and the physical structure of the absorbent is highly effective on the adsorption process and on the COP of refrigeration. In Figure 3.1 the photo shows the natural clinoptilolit zeolite absorbent's chemical structure that is pictured by electron scanning microscope by focusing for 5000 times greater. [6] "Zeolites are the crystal structured form of the water added aluminum silicate that comes from alkali and ground alkali elements. General chemical formulas are written as $M_{x/n}[(AlO_2)_x(SiO_2)_y]_2H_2O$ (M, represents the cations) and they have many natural and reproduced types.

They have 3-10 A^{o} stoma degree and have a high surface area (1000 m^{2}/gr synthetic zeolit) so they have highly adsorption capacity. Zeolites are able to adsorp gases with respect to approximately 30% percent of their weight". [6]

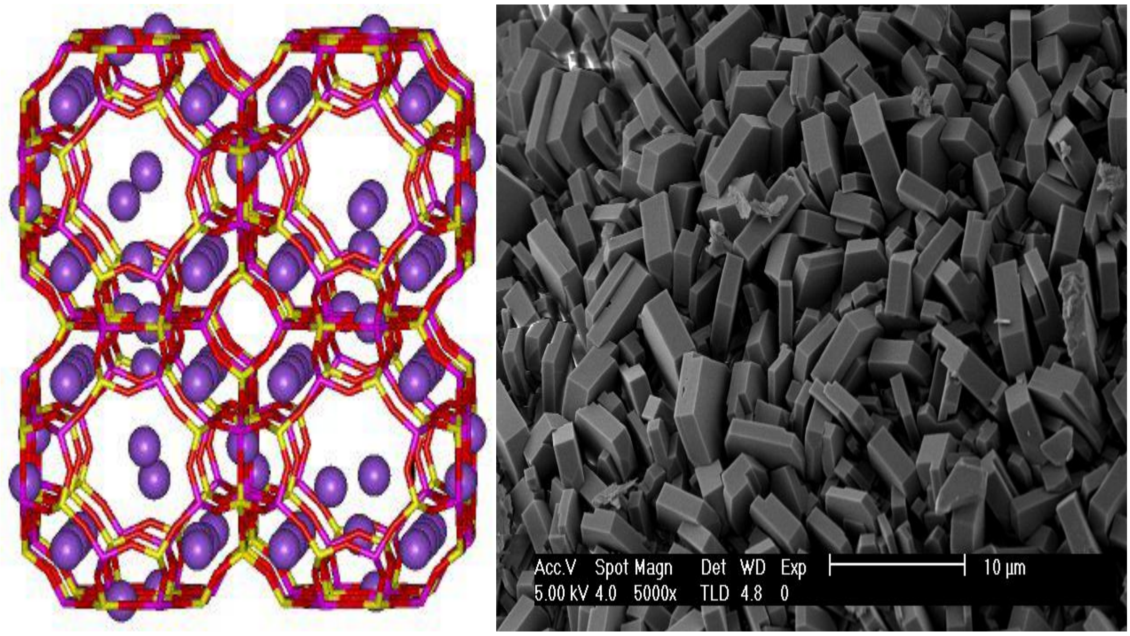


Figure 3.1 Clinoptilolit zeolite absorbent's chemical structure that is pictured by electron microscope [6]

"The adsorbent–adsorbate pair, which must be compatible with the environment, is one of the important parts of adsorption heat pump system." [14] Main requirements of the adsorbate are high latent heat, non-corrosive, non-toxicity and good thermal and chemical stability within the working conditions (temperature and pressure ranges). "On the other hand, adsorbents should have high adsorption capacity, high thermal conductivity, low cost. Zeolite–water, active carbon–methanol, silica gel-water, and carbon–ammonia are some of the common adsorbent–adsorbate pairs used in adsorption heat pump systems." [14] They must be non-polluting and non-flammable, their vapor pressure must be near atmospheric level, in the temperature range between 263 and 353 K." [8] Table 3.1 shows some physical properties of several refrigerants [8]

Name	**Formula**	**Normal Boiling Point (K)**	**Heat of Vaporization (J/g)**
Ammonia	NH_3	239	1368
Water	H_2O	373	2258
Methanol	CH_3OH	338	1102

Table 3.1 Physical properties of some refrigerants [8]

The characteristics of these refrigerants (adsorbates) are given in Table 3.2 [8]. “The suitable adsorbents are porous materials and they adsorb large amounts of refrigerant fluids in vapor phase and have the following characteristics like; wide concentration change in small temperature range, reversibility of adsorption process for many cycles, low cost, good thermal conductivity.” [8]

Ammonia	**Methanol**	**Water**
– Toxic – Flammable in some concentration – Not compatible with copper – High operating pressure – Good latent heat – Thermally stable – Non polluting	– Toxic – Flammable – Not compatible with copper at high temperature – Unstable beyond 393 K – Low pressure – Good latent heat	– Perfect, except for very low operating pressure – At low pressure does not oxidize copper and only partially stainless steal – Not suitable for cold climate zone

Table 3.2 Characteristics of some refrigerants [8]

“The smaller the pore diameter means the higher the adsorption energy and the regeneration temperature.” [8] Suggested applications for adsorption pairs are given in the below Table 3.3 [8].

Freezing (T<253 K)	**Refrigeration (T≈273 K))**	**Air Conditioning (T=278-288K)**	**Space heating (T≈333K)**
Zeolite- NH_3	A.Carbon- CH_3OH	Zeolite- H_2O	Zeolite- H_2O
		A.Carbon- NH_3	A.Carbon- NH_3
		A.Carbon- CH_3OH	Silica-Gel- H_2O
		Silica-gel- H_2O	

Table 3.3 Suggested applications for adsorption pairs [8]

“For many adsorbent–adsorbate pairs, the adsorption heat pump cycle operates under high vacuum. It is difficult to maintain the operation pressure in a high vacuum for a long time. This requires high vacuum technology like using special materials and gaskets but they increase the cost of adsorption heat pump and cause the use of heavier containers.” [14] The high vacuum is required for operating under low pressure because of the physical properties of the working pairs which are; vaporizing and condensing pressures and temperature values. High vacuum is also required for maintaining and stabilizing these properties in operational range for the cooling propose. In Table 3.4, it is given the illustration of the comparison of adsorbent–adsorbate pairs according to maximum adsorbate capacity, heat of adsorption values, adsorbent specific heat, energy density, and operating temperature range [14]. For selecting the proper and convenient adsorbent-adsorbate pair by considering the adsorption capacity and also for operating temperature ranges; reversibilities and cost analysis must also be considered for the selection of the pair [14]. “The heat transfer through packed beds is slow, limiting the refrigeration effect and also most conventional adsorbents require a regeneration temperature in the range of 200 °C to 250 °C, restricting the coefficient of performance. Issues related to the choice of adsorbate including flaming ability, toxicity, stability and gas dynamics." [22] "The specific volume of adsorbate also limits performance, either because of choking or frictional resistance. Generally, water is not useful for high heat loads." [22] Alternatively, ammonia is toxic and corrodes copper and brass fittings. "Most alcohols are environmentally friendly, but they are dehydrated and catalytically decomposed during desorption at 150 °C to 200 °C." [22]

Adsorbate–adsorbent	Maximum adsorbate capacity (kg/kg)^a	Average heat of adsorption (kJ/kg adsorbate)^a	Adsorbent specific heat (kJ/kg)^a	Energy density (kJ/kg adsorbate)	Temperature range (°C)
Water–zeolite 4A	0.22	4400	1.05	1250	30–350
Water–zeolite 5A	0.22	4180	1.05	1200	30–350
Water–zeolite MgA	0.29	3400	1.06	800	60–250
Water–zeolite 13X	0.30	4400	0.92	1290	30–350
Water–zeolite 13X	0.27		0.84	930	20–300
Water–zeolite 10A	0.20	4000		897	50–250
Water–zeolite 13X	0.27	3400	1.06	1200	30–350
Water–clinoptilolite	0.12	3000	1.11	480	20–240
Water–mordenite	0.11	4000		419	30–350
Water–chabazite	0.17	3000	1.08	700	30–250
Water–charcoal	0.40	2320	1.09	1200	30–250
Water–ac. alumina	0.19	2480	1.00	660	30–250
Water–silica gel	0.37	2560	0.88	1000	30–150
Water–silica gel	0.20	2500	1.045	600	20–130
Methanol–zeolite 13X	0.20	2400	1.07		
Methanol–zeolite 4A	0.16	2300	1.07		
Methanol–zeolite 5A	0.17	2300	1.07		
Methanol–zeolite 5A	0.17	2300	1.07		
Methanol–ac. carbon	0.32	1400	0.9	590	20–140

Table 3.4 Comparisons of adsorbent–adsorbate pairs [14]

The differences between the cyclic behaviors of the single and binary working fluid systems are shown in Figure 3.2 [14]. The operation pressure of water-zeolite system is low and requires high vacuum in the cycle like shown in Figure 3.2 (A), and the NH_3-Zeolite cycle operates at higher pressure which is 4-11 times higher than the ambient pressure like shown in Figure 3.2 (B). With the appropriate mixing of ammonia and water, a cycle which operates with a pressure close to ambient pressure can be obtained as shown in Figure 3.2 (C). “This improvement may solve one of the problems of adsorption heat pump which is working under high vacuum.” [14]

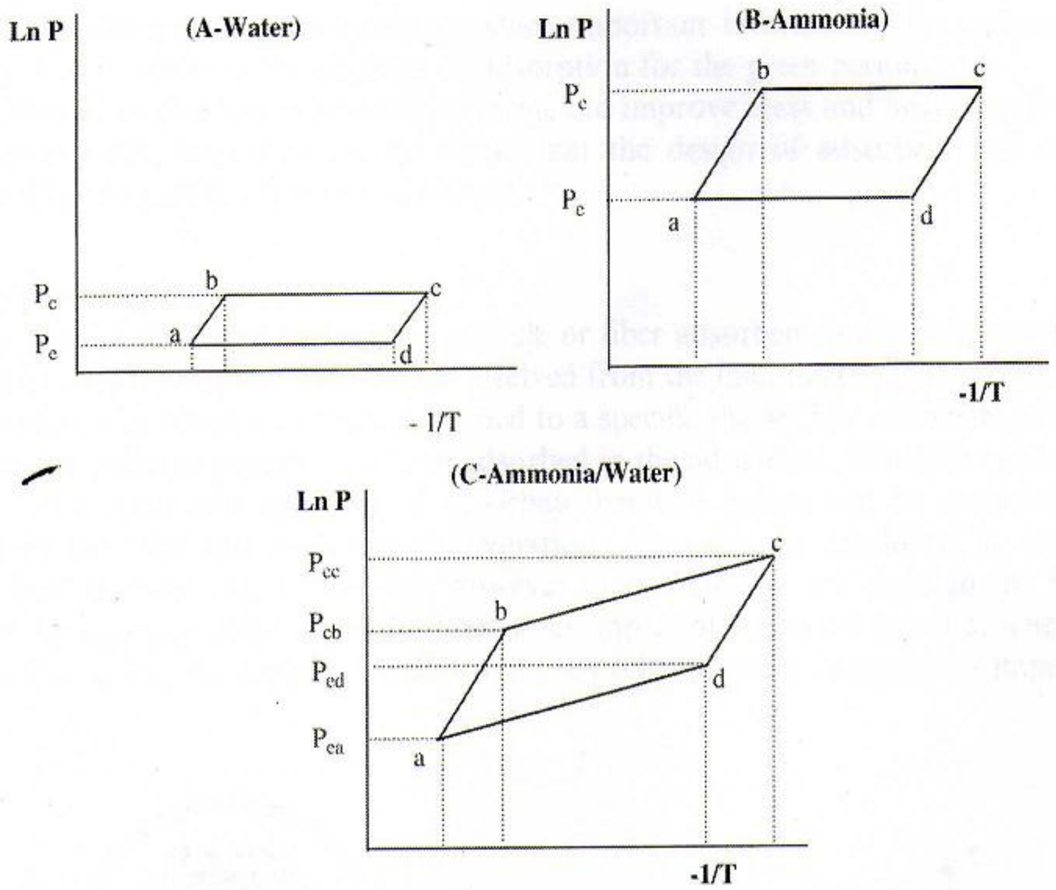


Figure 3.2 Ideal cycle of the system on isosteric graphs for single and binary working fluid [14]

The working pairs were carefully selected to satisfy the adsorption refrigeration and cooling cycle’s requirements and carry on the adsorption refrigeration much sufficiently. These are also valid for resorption refrigeration with a high desorption temperature during the regeneration temperature [35]. “The choice of the working pairs that are suitable for the proposed cycle, was based on the thermodynamics characteristics of the Clausius-Clapeyron diagram (isoster diagrams).” [35] The adsorbent particle diameter plays an important role in mass transfer and heat transfer characteristics. If the particle sizes increase, the mass transfer resistance in micro-pores increases while the mass transfer in macro-pores decreases [31]. “The performances of compressed adsorbent particles which are with the motivation of higher performance are based on higher equivalent thermal conductivity of the bed.” [31] Compression of the adsorbent particles will lead us to the mass transfer resistance in macro-pores of the bed which is also increased [31].

3.2 Principles of Adsorption Cycle for Refrigeration

3.2.1 Heating and Pressurization

This is the period from a to b as shown in Figure 3.2. During this period, the adsorber receives heat while being closed. The adsorbent temperature increases, which induces a pressure increase from the evaporation pressure up to the condensation pressure. “This period is equivalent to the "compression" in compression cycles.”[16]

3.2.2 Heating, Desorption and Condensation

This is the period form point b to c as shown in Figure 3.2. During this period, the adsorber continues receiving heat while being connected to the condenser, which now superimposes its pressure. The adsorbent temperature continues increasing, which induces desorption of vapor. This desorbed vapor is liquefied in the condenser. The condensation heat is released to the second heat sink at intermediate temperature. “This period is equivalent to the “condensation” in vapor compression cycles.” [16]

3.2.3 Cooling and Depressurization

This is the period from point c to d as shown in Figure 3.2. During this period, the adsorber releases heat while being closed. The adsorbent temperature decreases, which induce the pressure decrease from the condensation pressure down to the evaporation pressure. “This period is equivalent to the “expansion” in vapor compression cycles.” [16]

3.2.4 Cooling, Adsorption and Evaporation

This is the period from point d to a as shown in Figure 3.2. During this period, the adsorber continues releasing heat while being connected to the evaporator, which now superimposes its pressure. The adsorbent temperature continues decreasing, which induces adsorption of vapor. This adsorbed vapor is vaporized in the evaporator. The evaporation heat is supplied by the heat source at low temperature. “This period is equivalent to the "evaporation" in vapor compression cycles.” [16]

3.3 Adsorbent Beds

They are used for keeping the adsorbent materials inside of the bed. They also called as reactors in some fields of their applications. The adsorption and desorption processes occurs in these adsorbent keepers and the designing of adsorbent bed is the other important and difficult point in these type of systems. Adsorbent bed requires a special design for controlling mass and heat transfer between the adsorbate and adsorbent pairs. “Since thermal conductivity of adsorbents is generally low, heat is transferred slowly through the adsorbent bed as well as the periods of adsorption and desorption processes become longer.” [14] “The mass transfer depends on adsorbate flow through the bed (interparticle flow) and through the adsorbent (intraparticle diffusion due to concentration differences, molecular diffusion, Knudsen diffusion, and surface diffusion).” [14] In order to determine temperature and concentration properties in the adsorbent bed, the heat and mass transfer equations have to be solved for the adsorber. The governing equations for the transportation of heat and mass are coupled and should be solved simultaneously and the results produce information for designing of adsorbent bed for estimating the capacity of adsorption for the given period. “Experimental studies had also been performed to improve mass and heat transfer rates in adsorbent beds. The design of adsorbent bed must also be classified according to the form of adsorbent.” [14] There are some important parameters that affect the system efficiency. The adsorbent bed thickness has a great influence on the performance of the system. “The COP increases with an increase in adsorbent thickness.” [31] Although a larger thickness means more adsorbate can be driven in to cycling so the thermal resistance is going to enlarge. It leads us to a longer cycle time and a reduction in the SCP [31].

The adsorbers are the most important components for an adsorption cooling system.” [30] “The performance of the adsorbers determines the capacity of an adsorption cooling system to a great extent.” [30] So, the adsorber must have good performance of the heat and mass transfer. But the working pair Zeolite 13X and water have a poor thermal conductivity of 0.009-0.15 $Wm^{-1}K^{-1}$ with respect to the other cooling pairs. For this purpose, the enhancement of the heat transfer inside the adsorbers must be considered [30]. The systems which have single adsorption bed, can only provide cooling intermittently and it has a lower system performance in terms of COP and SCP. To increase the performance in terms of COP and SCP multi bed systems must be used [33]. “Regenerative process with temperature front (also called thermal wave process), regenerative process with heat and mass recovery and involving rotating adsorbers technology have been extensively developed to improve the process’s continuous manner and stability.” [33]

3.3.1 Uncoated Type Adsorbers

In this type of adsorbent bed, pellet, granule or fiber adsorbents are generally used and these adsorbents are not used as they are received from the manufacturer. “However, there are some studies in which adsorbent are formed to a specific shape.”[14] The adsorbate moves in voids between pellet or granule and then adsorbed in the adsorbent which is based on porosity of the bed, convection and diffusion of adsorbate between pellets can be considered or removed in the heat and mass transfer equations. Fins can be used to increase heat transfer rate in the bed. However, mass transfer rate through the bed can be improved by creating voids in the bed. Some examples of uncoated type adsorbers are shown in Figure 3.3. Figure 3.3.a, shows slim thin wall shell tube adsorber designed for improving heat transfer rate. [14] The activated carbon used as adsorbent which is placed among the tubes that are used for heating and cooling. “The rib pieces on tubes increase heat transfer rate.” [14] Activated carbon fibers have higher total pore volume, surface area and adsorption capacity than silica gel particles. Moreover, adsorption/desorption isotherm shows that activated carbon fibers do not have adsorption/desorption hysteresis. Activated carbon fibers are packed tightly inside oxygen-free copper fins as shown in Figure 3.3.b.[14] The developed prototype of a fast cycle adsorption refrigerator that is composed of laminate of monolithic carbon discs and aluminum fins is shown in Figure 3.3.c. “The monolithic carbon is mixed with organic binder, compressed and fired.” [14]

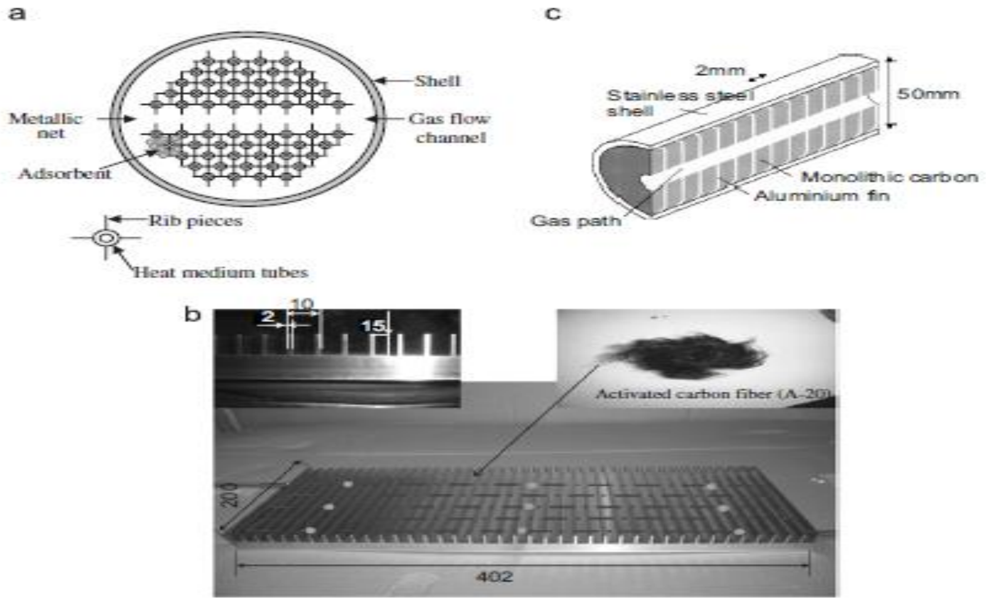


Figure 3.3 Photograph of untreated type of adsorbent bed designs [14]

3.3.2 Coating Type Adsorbers

The adsorbent is coated around a pipe, fin or in metal foam and these types of adsorbent beds generate high speed heat and mass transfer with respect to the other types. Diffusion in the adsorbent is accepted as the main mechanism of mass transfer since there is no void in the coated type of adsorbers. Figure 3.4.a, shows a coated stainless steel tube with adsorbent for improving heat and mass transfer rate in bed. “This method allows obtaining high specific power adsorption heat pump.” [14] “An adsorbent bed which is made of finned tubes covered with SWS-1L (CaCl2 in mesoporous silica gel) adsorbent as shown in Figure 3.4.b.” [14] “The optimal cycle time of system is 20 to 40 minutes and a cooling COP is varies between 0.17 and 0.48 was prepared open-cell copper foam as metal support for adsorbent bed as shown in Figure 3.4.c.” [14] “The zeolite adsorbent is grown by hydrothermal synthesis on this metal support as shown in Figure 3.4.d.” [14]

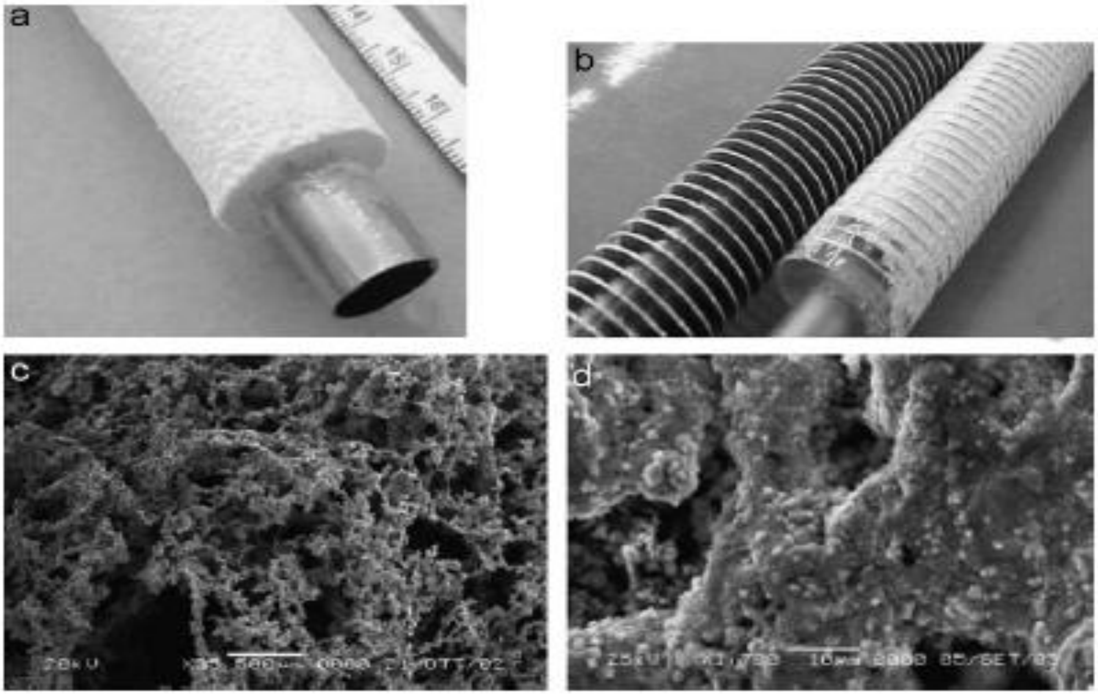


Figure 3.4 Photograph of coated type of adsorbent bed designs [14]

3.3.3 Extended Surfaces for Heat Exchangers

Several types of extended surfaces are used for finned tubes, plate heat exchangers, plate-fin heat exchangers. “The drawback of the extended surfaces is that they increase the thermal capacity of the adsorber, and therefore the extended surfaces heat exchangers require advanced cycle (heat recovery) to use heat source efficiently.” [29]

3.3.4 Consolidated Adsorbers

“Another way to get high heat transfer coefficients is to develop consolidated adsorbent beds with higher thermal conductivity. This is particularly interesting when the simple powder beds are not suitable.” [29] This approach has been developed for a long period of usage. For metal hydrides and researchers found that consolidation of the adsorbent beds with expanded natural graphite will be helpful for the heat transfer intensification [29]. “The optimal result is the 3000W/m^2 wall heat transfer coefficient.” [29] “The utilization of aluminum powder foam as a heat conduction matrix in consolidate compound is another method to intensify the heat thermal conductivity.” [29]

3.3.5 Heat Pipe Technology

It is deemed that the use of heat pipe will enable us to enhance the heat exchange efficiency in adsorber with respect to the high heat transfer coefficient that could be obtained by the condensation and the evaporation of the heat transfer fluid inside the heat pipe [29]. “The condensation of the heat transfer fluid releases the necessary heat to regenerate the adsorber and the vaporization of the former absorbs the necessary heat to cool down the adsorber.” [29]

It is given some types of the adsorption beds (reactors) in Figure 3.5 [17]. The bed amounts and the chamber amount can be increased with respect to the purpose of the cooling and main load of the cooling for the high improvement of the COP of refrigeration.

Figure 3.5 Example of reactor designs [17]

Figure 3.6 shows a double reacted adsorption adsorbent bed which is developed by a firm. “As it is sown in the Figure the first adsorption process has finalized the communication between the adsorbent bed and evaporator is interrupted and the heating procedure begins. At the same time the other adsorbent bed’s communication established, this type of double reacted adsorbent beds keep the hole system in continues operating.” [6]

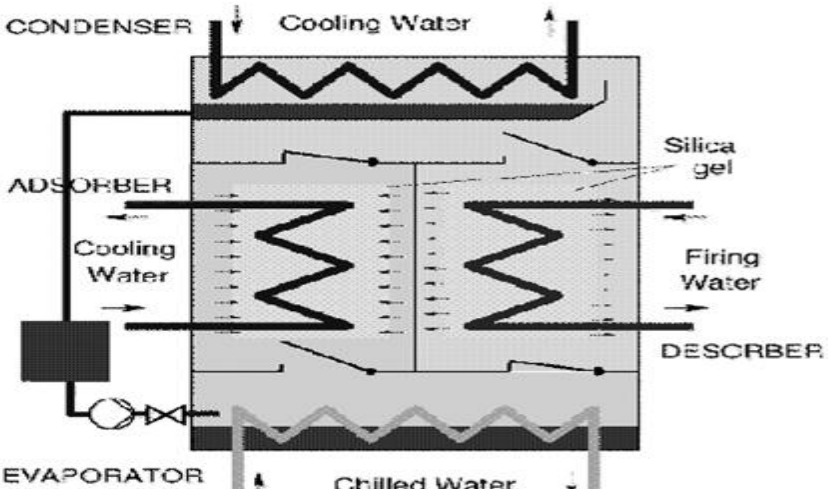


Figure 3.6 Two chamber adsorption cooling system [6,7]

3.4 The Solar Source of the Cycle

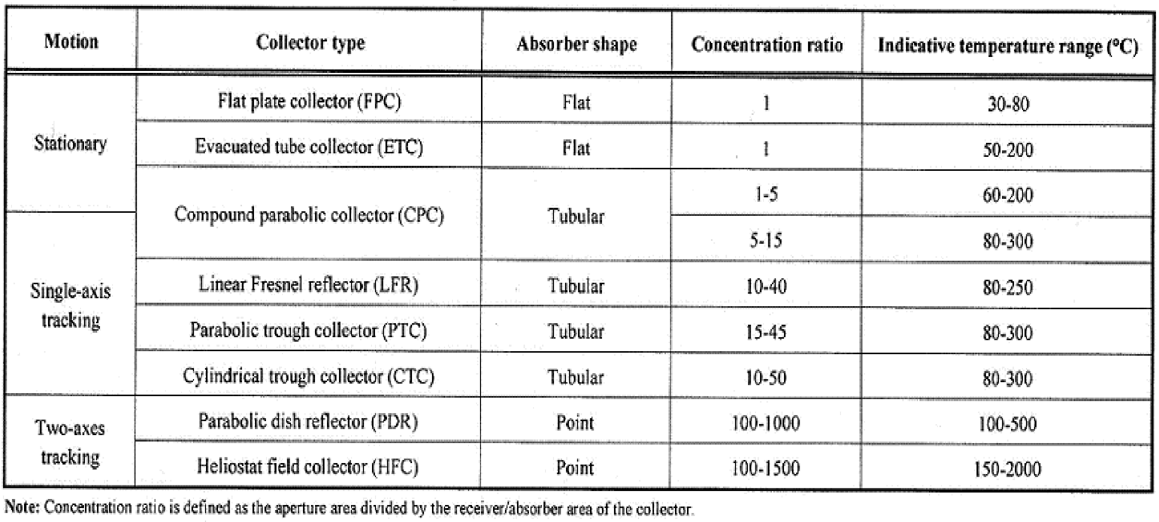
“The adsorption and absorption heat pumps and cooling systems are operated with respect to the heat supplied in to the separators of the system. This is generally 80-130 °C hot/superheated water or water vapor in 1 Atm in single effected systems.” [24] These types of systems’ COP doesn’t occurs above 0,7 in our days and they are much effective if there are enough of rejected heat amount in the plant. “The underworld thermal waters recourses, the exhaust emission gases of some processes, some cogeneration systems’ and manufacturing systems and superheated water that are heated in Heat Exchangers are used for rejected type of heat source for the adsorption and absorption heat pump systems.” [24] Solar power is renewable and much clean. They can be used for driving the adsorption heat pumps and absorption heat pumps and adsorption and absorption cooling systems instead of rejected heat types of sources [24]. The researches in literature and in operational marketing applications show that, it will be much applicable and economical especially in the cooling systems. “Because the increased demand of cooling is parallel to the solar power radiation increasing.” [24] “The fundamental operational problem with solar collectors is the collection and delivery of solar energy to users with minimum looses.” [23] “The optimum operating conditions for solar collectors can be investigated using different modes of performance.” [23] The main aim is to optimize the thermal efficiency of any collector, which is defined as the ratio of “useful energy output to the input” during the same time period [23]. The performance of solar collectors can be examined from the standpoint of exergy. The amount of useful energy delivered by solar collectors is found to be affected by heat transfer irreversibility between the sun and the collector, between the collector and the ambient air and in-side the collector. “The rate at which exergy is collected by a solar collector can be increased by increasing the mass flow rate of the flowing fluid inside of the collector.” [23] “Since the collector is also a expensive part of the thermal system, it is required advanced technology to build it.” [23] This means the collector area had been optimized which leads us to the optimization of hole system’s capital cost and also the optimization of the mass flow rate of the fluid inside the collector is necessarily required with respect to exergy and the adsorption rate of the adsorption bed. [23]

3.4.1 Solar Power

Solar power is the technology for gathering power and energy from sunlight. “The power comes from the sun with respect to the hydrogen gases that are converted in to helium gases on the sun.” [27] The sun power is approximately 1370 W/m^2 in the outer space but it is changing from 0 to 1100 W/m^2 in the atmosphere. A few amount of this energy that is arriving on to earth is much greater than mankind’s required. “The investigations and applications showed that the solar power is a clean and environmental power source for mankind.” [27]

3.4.2 Solar Collector Types

There are basically two types of solar collectors which are none concentrating or stationary and concentrating and also be called as absorbing and reflecting absorbing. [27] “Solar energy collectors are basically distinguished by their motion, i.e., stationary, single axis tracking and two-axes tracking, and their operating temperature. A large numbers of solar collectors are available in the market.” [27] In Table 3.5, the comprehensions’ of the solar collectors can be seen.



Motion	Collector type	Absorber shape	Concentration ratio	Indicative temperature range (°C)
Stationary	Flat plate collector (FPC)	Flat	1	30-80
	Evacuated tube collector (ETC)	Flat	1	50-200
	Compound parabolic collector (CPC)	Tubular	1-5	60-200
Single-axis tracking			5-15	80-300
	Linear Fresnel reflector (LFR)	Tubular	10-40	80-250
	Parabolic trough collector (PTC)	Tubular	15-45	80-300
	Cylindrical trough collector (CTC)	Tubular	10-50	80-300
Two-axes tracking	Parabolic dish reflector (PDR)	Point	100-1000	100-500
	Heliostat field collector (HFC)	Point	100-1500	150-2000

Note: Concentration ratio is defined as the aperture area divided by the receiver/absorber area of the collector.

Table 3.5 Solar energy collectors [27]

3.4.3 Flat Plate Solar Collectors

Flat-Plate collectors are made of an insulated and weatherproof box containing a dark absorber plate under one or more transparent or translucent (semi transparent) covers inside of the box. Water or some other conducting fluid passes through pipes which are located below the dark absorber plate and the fluid is heated up while it is flowing inside of the pipe. “They are still the most common type of collector in many countries.” [25] “Flat plate collectors are the most widely used kind of collector in the world for domestic solar water heating and solar space heating applications; they are durable and effective.” [26] “Flat-Plate collectors are usually employed for low temperature applications up to 80 °C.” [27] A flat plate type solar collector has been shown in Figure 3.7.

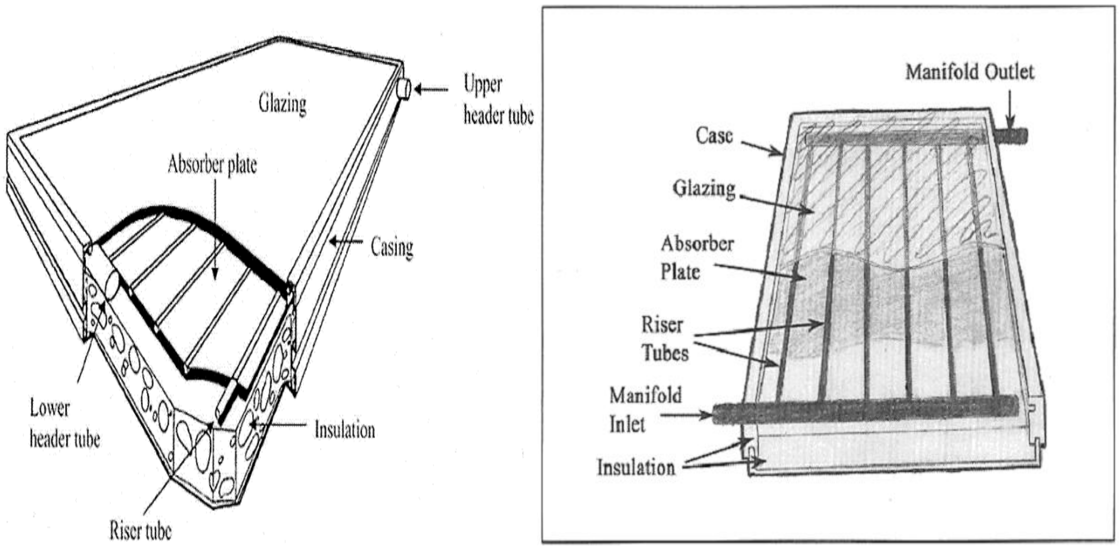


Figure 3.7 A flat plate type collector and its details [26, 27 and 37]

3.4.4 Evacuated Tube (Vacuum Piped) Solar Panels

Evacuated tube collectors are constructed of a number of glass tubes which are simultaneously lined. Each tube is made of annealed glass and has an absorber plate within the tube. During the manufacturing process, a vacuum is created inside the glass tube like as the other types of the solar collectors. “The absence of air in the tube creates excellent insulation, allowing higher temperatures to be achieved at the absorber plate.” [26] “The vacuum envelope reduces convection and conduction losses, so the collectors can operate at higher temperatures (~150 °C). Both direct and diffuse radiation can be collected.” [27] There are several types of evacuated tubes;

3.4.5 Glass-Glass tubes

“They consist of two glass tubes which are fused together at one end.” [26] The inner tube is coated with a selective surface for absorbing the solar energy much sufficiently but they inhibit radioactive heat loss. The air is withdrawn ("evacuated") from the space between the two glass tubes to form a vacuum and they eliminate conductive and convective heat loss and they also perform very well in overcast conditions as well as low temperatures. “Because the tube is 100% glass, the problem with loss of vacuum due to a broken seal is greatly minimized. Glass-glass solar tubes may be used in a number of different ways, including direct flow, heat pipe, or U pipe configuration.” [26]

3.4.6 Glass-Metal tubes

They consist of a single glass tube which a flat or curved aluminum plate is attached to a copper heat pipe or water flow pipe. The aluminum plate is generally coated with Tinox, or similar selective coating. These types of tubes are very efficient but the loss of vacuum can cause some problems. “This is primarily due to the fact that their seal is glass to metal. The heat expansion rates of these two materials. Glass-glass tubes although not quite as efficient glass-metal tubes are generally more reliable and much cheaper.” [25]

3.4.7 Glass-glass - water flow path tubes

“They incorporate a water flow path into the tube itself. The problem with these tubes is that if a tube is ever damaged water will pour from the collector onto the roof and the collector must be "shut-down" until the tube is replaced.” [25] An evacuated type of solar collector has been shown in Figure 3.8.

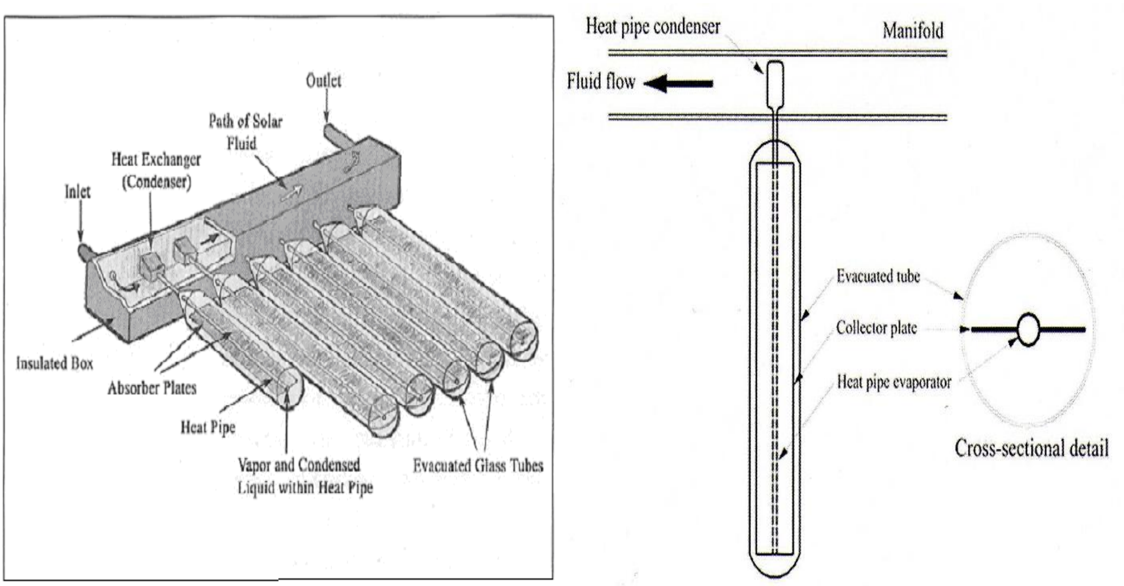


Figure 3.8 An evacuated tube solar collector and its schematic diagram [26, 27]

3.4.8 Concentrating Type of Collectors

A concentrating collector utilized with a reflective parabolic-shaped surface to reflect and concentrate the solar energy to a local point where the absorber of the energy is located. The reflectors must track the sun for high efficiency. These types of solar collectors can achieve very high temperatures because the diffusion of the solar resource is concentrated on a small area like a focusing magnifier. In fact, on the earth's surface the hottest temperatures ever measured have been located at the focal point of a massive concentrating solar collector. “Concentrating collectors have been used to make steam that spins an electric generator in a solar power station. This is sort of like starting a fire with a magnifying glass on a sunny day.” [26]

Many designs have been considered for concentrating collectors. “Concentrators can be reflectors or refractors, can be cylindrical or parabolic and can be continues or segmented.” [27] Receivers can be convex, concave, cylindrical or flat and can also be covered with glazing or they can be uncovered which is meant to be without glazing. “Increasing concentration ratios mean increased temperatures at which energy can be delivered but consequently these collectors have increased requirements for precision in optional quality and positioning of the optical system.” [27]

3.4.9 Parabolic Through Collectors

“Parabolic through collectors are made by bending a sheet or reflective material in to a parabolic shape and a metal black tube, covered with a glass tube to reduce heat losses, is placed along the focal line of the receiver. These types of collectors produce heat approximately from 50 °C to 400 °C for solar thermal electricity generation or process heat applications.” [27]

3.4.10 Linear Fresnel Reflector

“They relie on an array of linear mirror strips which concentrate light on to fixed receiver mounted on a linear tower. They can be imagined as a broken-up parabolic through reflector but unlike parabolic troughs, it doesn’t have to be of a parabolic shape, large absorbers can be constructed and the absorber doesn’t move.” [27]

3.4.11 Parabolic dish reflector

They are point focus collectors and they also truck the sun in two axes for focusing the solar energy onto a receiver that located at the focal point of the dish. “The dish structure must track fully the sun to reflect the beam into the thermal receiver.” [27] These reflectors can achieve temperatures approximately in excess of 1500 °C [27].

3.4.12 Heliostat field collector

These types are used for extremely high inputs of radiant energy by using many of flat mirrors, or heliostats and using altazimuth mounts can be used to reflect their incident direct solar radiation onto a common focusing target [27]. “These systems allow working at relatively high temperatures of more than 1500 °C and to integrate thermal energy in more efficient cycles." [27] The concentrating types of collectors and their schematics have been shown in Figures 3.9, 3.10, 3.11, 3.12 and 3.13.

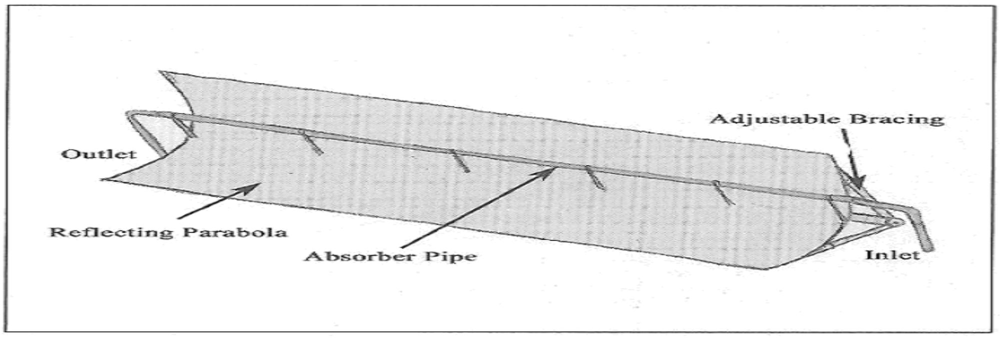


Figure 3.9 A concentrating type of solar collector [26]

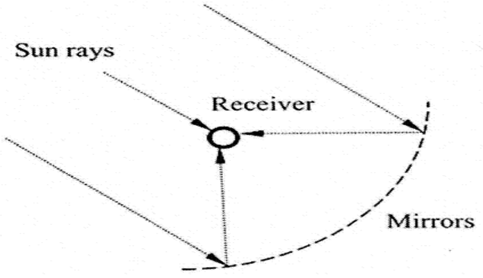


Figure 3.10 Schematic diagram of a Fresnel type parabolic through collector [37]

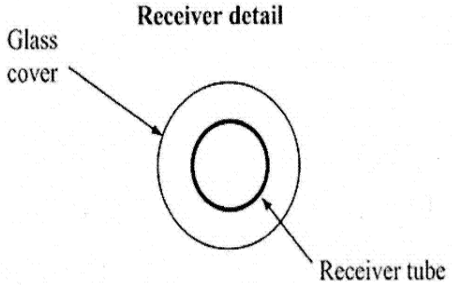


Figure 3.11 Schematic of a parabolic through collector [37]

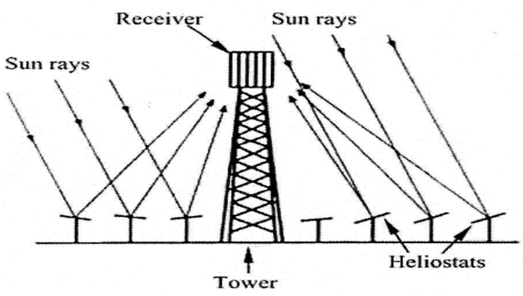


Figure 3.12 Schematic diagram of heliostat field collector [37]

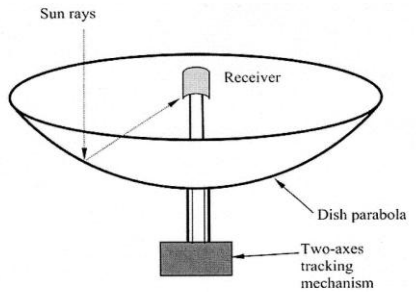


Figure 3.13 Schematic diagram of a parabolic dish collector [37]

3.4.13 ICS Collectors

In an integral collector storage unit (ICS), the hot water storage tank is the solar absorber of the unit. The tank or tanks are mounted in an insulated box with glazing on one side and are painted black or are coated with a selective surface for high absorption of the solar energy. “The sun shines through the glazing and hits the black tank, warming the water inside the tank. Some models feature a single large tank (113-189 liters) while others feature a number of metal tubes plumbed in series (113-189 liters total capacity)” [26] The single tanks are generally made of steel, while the tubes are generally made of copper. “These collectors weigh 125 kg to 205 kg when full, so wherever they are mounted, the structure has to be strong enough to carry this significant weight.” [26] The Figure 3.14. shows a tank-type ICS collector and Figure 3.15 shows a tube-type ICS collector.

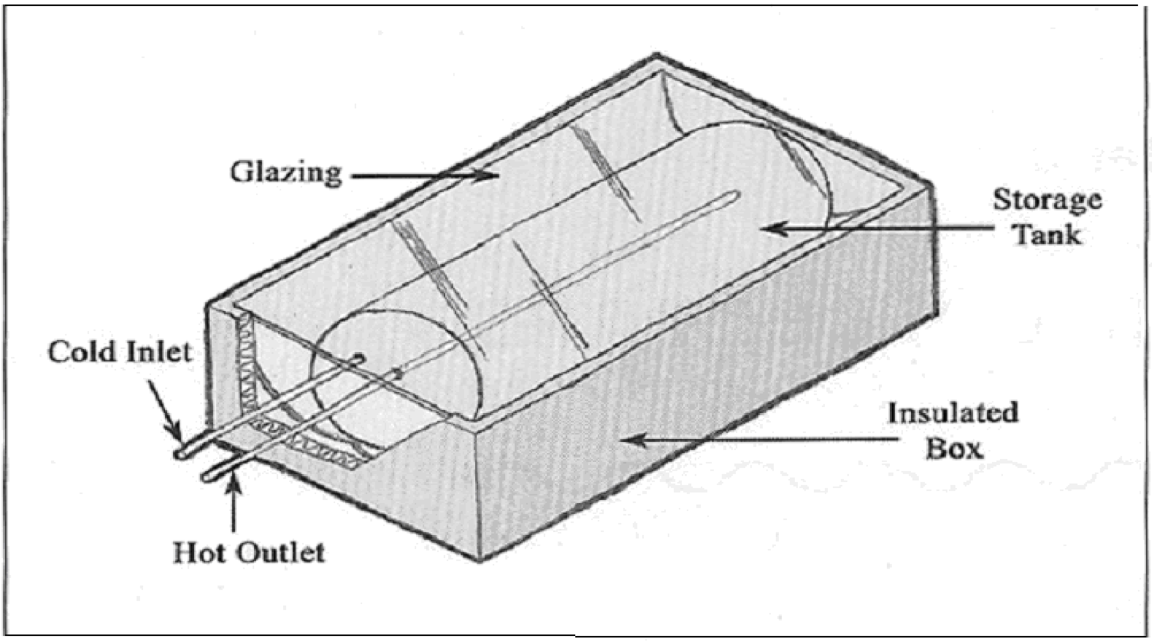


Figure 3.14 A tank-type ICS collector [26]

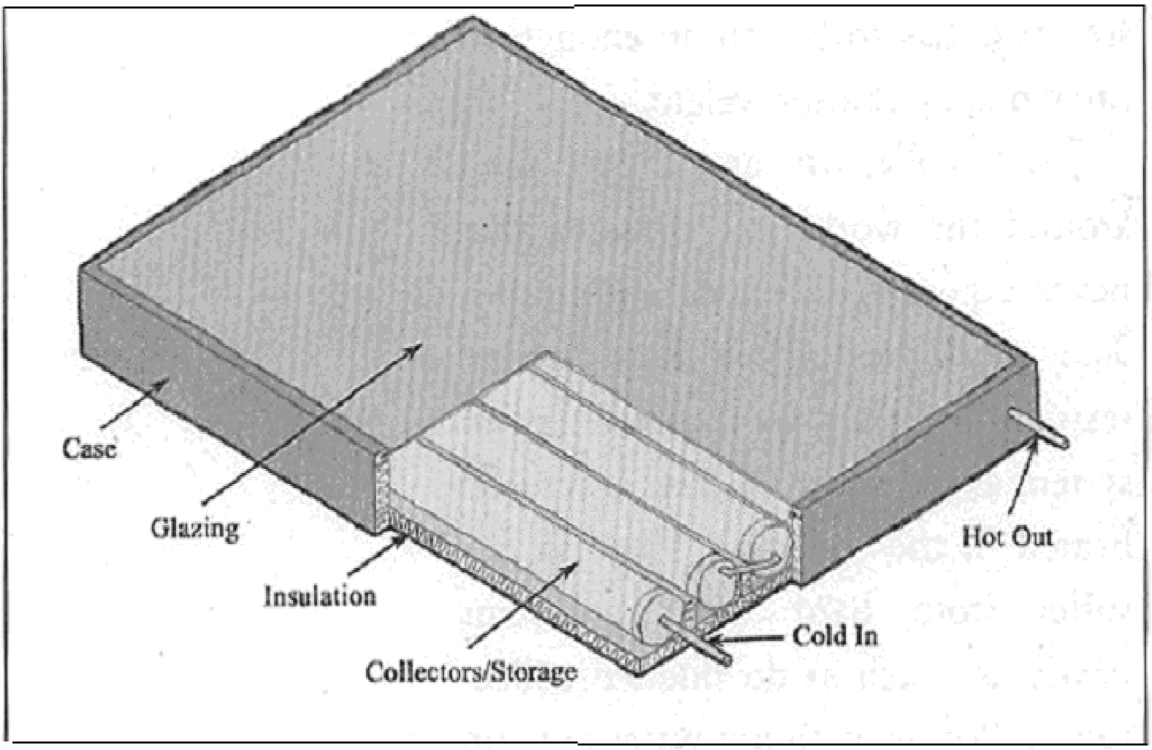


Figure 3.15 A tube-type ICS collector [26]

3.4.14 Air Collectors

Air can also be used as the heat transfer fluid in a solar collector. Air collectors are flat plate type solar collectors. “Instead off an absorber plate made of copper piping and copper fins, the absorber plate in an air collector is typically made of a solid sheet of aluminum.” [26] The aluminum absorber plate can be coated with a selective surface or can be paint black paint to increase efficiency. When the sun shines on the absorber plate, it gets hot. “Air is drawn from the building and it is blown across the back of the absorber plate and heated. The hot air is then delivered to the building through ductwork. A blower circulates the air through the system.” [26] A simple type of an air collector is shown in Figure 3.16.

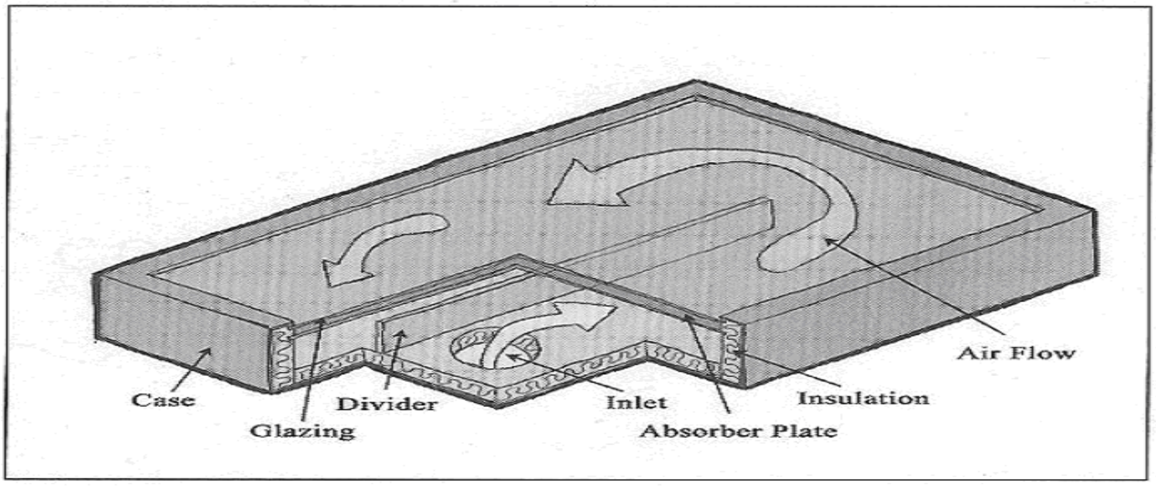


Figure 3.16 An air collector type [26]

3.4.15 Pool Collectors

They are generally used for heating the swimming pools by solar heating systems. They don't have to be glazed and made of a special copolymer plastic. They are only be used when it is warm outside. "These collectors cannot withstand freezing conditions."[26] Figure 3.17 shows a pool collector type.

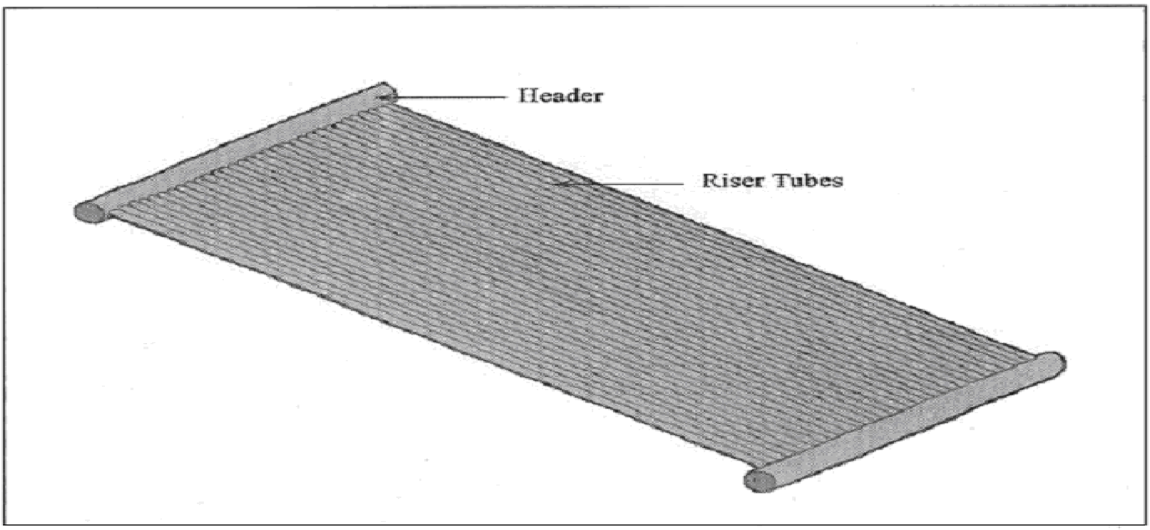


Figure 3.17 Pool collector type [26]

3.4.16 Types of Solar Water Heating Systems for Delivering the Solar Energy Source in to the Adsorber Bed

These systems can be either active or passive and either open-loop or closed-loop. An active system uses electric pumps, valves, and controllers to circulate water or other heat-transfer fluids through the collectors for efficiency and continuously. They are usually more expensive than passive systems. “Active systems are often easier to retrofit than passive systems because their storage tanks do not need to be installed above or close to the collectors.” [25]

In “Open-Loop Active Systems” a pump is used to circulate water through the collectors in these systems. It is efficient and it lowers the operating costs but is not appropriate if water is hard or acidic. Because without a filtration system the scale and corrosion will gradually disable the system. “Open-loop active systems are popular in regions that don’t experience subzero temperatures. Flat plate open-loop systems should never be installed in climates that experience sustained periods of subzero temperatures.” [25]

In “Closed-Loop Active Systems” The heat-transfer fluids pumped (usually a glycol-water antifreeze mixture) through the solar water heater. Heat exchangers transfer the heat from the fluid to the water that is stored in tanks. Double-walled heat exchangers or twin coil solar tanks prevent contamination of household water. Closed-loop glycol systems are popular in areas that have extended to subzero temperatures because they offer good freeze protection. However, glycol antifreeze systems are more expensive to purchase and install Also the glycol must be checked each year and changed every few years, depending on glycol quality and system temperatures. Draining back systems use water as the heat-transfer fluid in the collector loop. A pump circulates the water through the solar water heater. If the pump is turned off, the solar water heater drain of water, which ensures freeze protection and also allows the system to turn it off if the water in the storage tank becomes too hot. A problem with the draining back systems is that the solar water heater installation and plumbing must be carefully located to allow complete drainage. The pump must also have sufficient head pressure to pump the water up to the collector each time the pump starts. Electricity usage is therefore slightly higher than a sealed closed or open loop system. [25]

“An open-loop system circulates household (potable) water through the collector. A closed-loop system uses a heat-transfer fluid (water or diluted antifreeze) to collect heat. A heat exchanger used to transfer the heat to the household water. A disadvantage of closed looped system is that efficiency is lost during the heat exchange process.” [25]

“A passive system has no pump and relies on thermo-siphoning to circulate water.” [25] The amount of hot water and the maximum temperature of the water depends on the type and size of the system and also it depends on the amount of solar energy available at the location. The correct installation angle and orientation of the systems carries much importance on the efficiency. They have the advantage of working when electricity cut age and electric pump breakdowns. This makes passive systems generally more reliable, easier to maintain and much longer working life than active systems. “Passive systems are often less expensive than active systems, but also they are generally less efficient due to slower water flow rates through the system.” [25]

A thermosiphon system relies on warm water rising and it is highly important for the circulation of the water through the solar absorber and through to the water tank. In this type of systems, the water tank must be located above the absorber solar collector. “As water in the absorber heated up, it becomes lighter and naturally rises into the tank above.” [25] Meanwhile, cooler water in the tank flows downwards into the collector that causes circulation of the system. “The disadvantages of this design are the poor aesthetics of having a large tank on the roof and the misses with structural integrity of the roof. Often the roof must be tough enough to carry the water tank.” [25] This system is generally used with both flat plate and evacuated tube solar collectors.

The batch heaters are also simple passive systems. They consist of one or more storage tanks that are placed in an insulated box that has a glazed side absorbing the sun. “Batch heaters are inexpensive and have few components, but only perform efficient in summer when the weather is hot.” [25]

3.5 The Efficiency and Typical Characteristics of the Solar Collectors

"The efficiency of the collectors varies depending on the solar radiation, outside temperature, and collector fluid temperature." [36] The plotting in Figure 3.18 shows the efficiency curves for three types of collectors. The vertical axis shows efficiency. The horizontal axis shows difference between the average collector temperature and the ambient air temperature. It is clear that efficiency is not a constant, it depends toughly on the temperature difference between the collector and the outside air. "As it is expected, the collectors show better efficiency when there is not much difference between the collector temperature and ambient (because the collector losses are greater when it is colder out.)" [36] Figure 3.19 shows the solar energy applications and the types of collectors used for the applications.

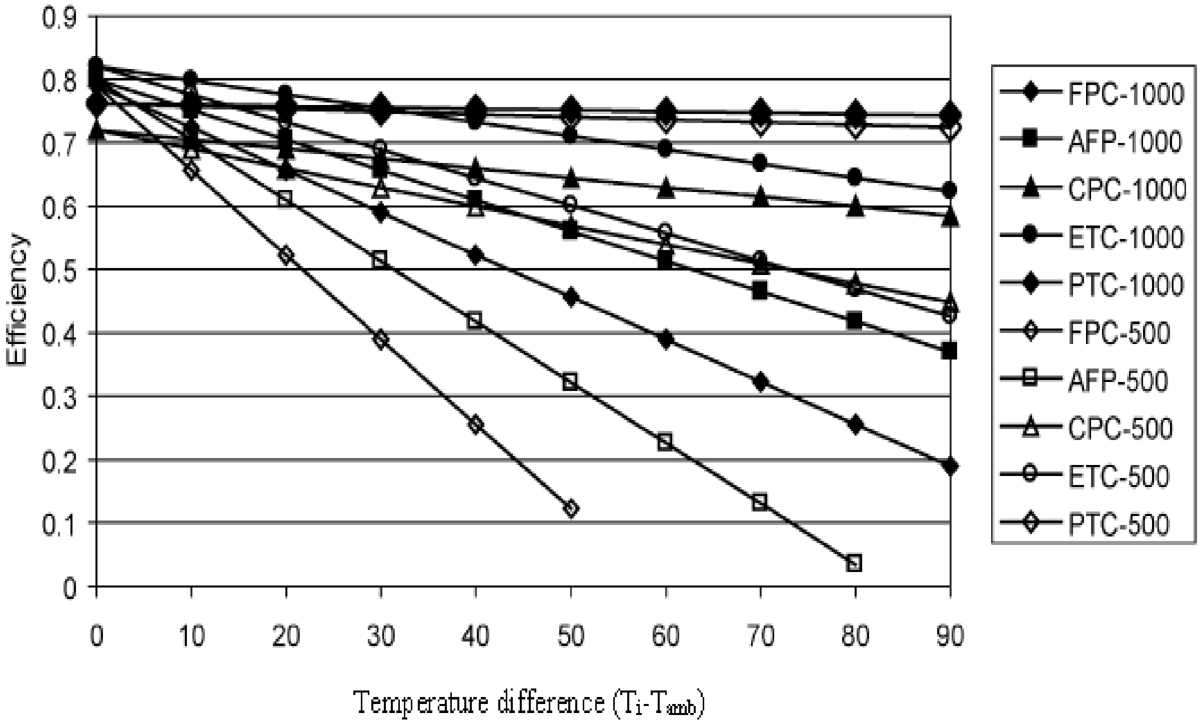
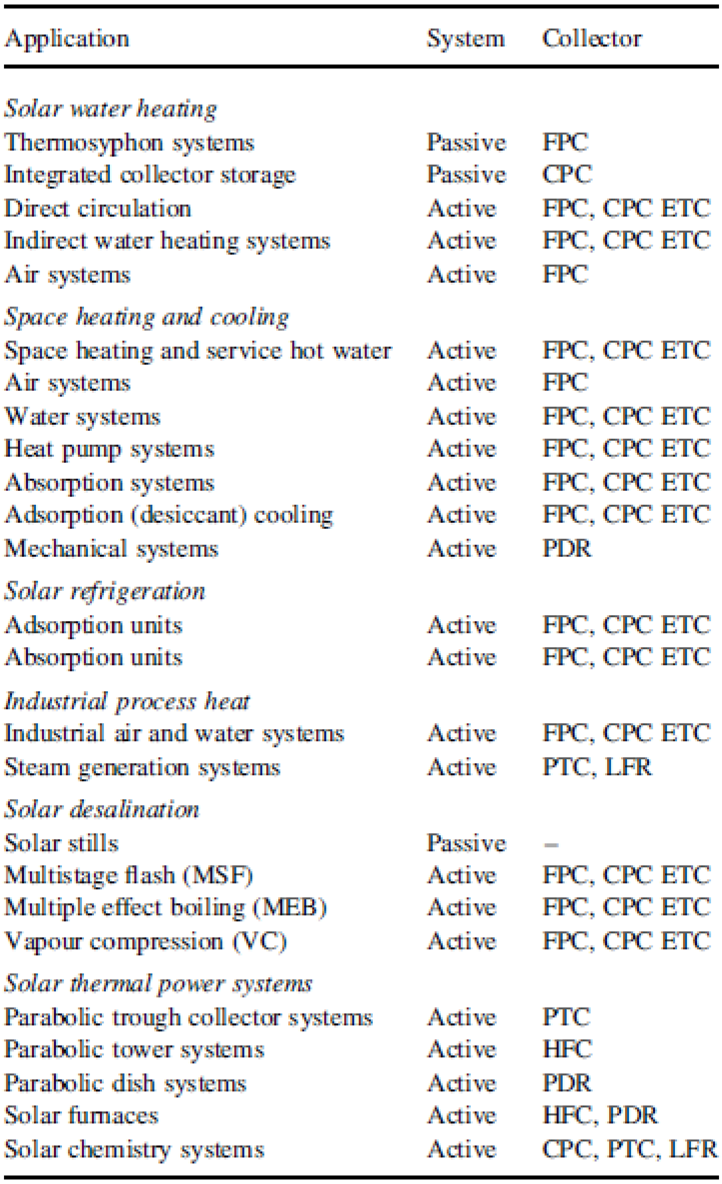


Figure 3.18 Efficiency plotting of different types of solar collectors at two irradiation levels 500 and 1000 W/m^2 (T is in °C) [37]



Application	System	Collector
Solar water heating		
Thermosyphon systems	Passive	FPC
Integrated collector storage	Passive	CPC
Direct circulation	Active	FPC, CPC ETC
Indirect water heating systems	Active	FPC, CPC ETC
Air systems	Active	FPC
Space heating and cooling		
Space heating and service hot water	Active	FPC, CPC ETC
Air systems	Active	FPC
Water systems	Active	FPC, CPC ETC
Heat pump systems	Active	FPC, CPC ETC
Absorption systems	Active	FPC, CPC ETC
Adsorption (desiccant) cooling	Active	FPC, CPC ETC
Mechanical systems	Active	PDR
Solar refrigeration		
Adsorption units	Active	FPC, CPC ETC
Absorption units	Active	FPC, CPC ETC
Industrial process heat		
Industrial air and water systems	Active	FPC, CPC ETC
Steam generation systems	Active	PTC, LFR
Solar desalination		
Solar stills	Passive	–
Multistage flash (MSF)	Active	FPC, CPC ETC
Multiple effect boiling (MEB)	Active	FPC, CPC ETC
Vapour compression (VC)	Active	FPC, CPC ETC
Solar thermal power systems		
Parabolic trough collector systems	Active	PTC
Parabolic tower systems	Active	HFC
Parabolic dish systems	Active	PDR
Solar furnaces	Active	HFC, PDR
Solar chemistry systems	Active	CPC, PTC, LFR

Figure 3.19 Solar energy applications and types of collectors used [37]

“The approximate efficiency of solar panel is defined as; Heat Output divided by Solar Radiation on Panel Where: Heat Output is the value you measured in BTU/hr or watts/hr [36].” The yearly irradiation level received for Cyprus is given in Figure 3.20 as a schema. The typical characteristics of the solar collators can be taken from Tables 3.21, 3.22, 3.23 and 3.24.

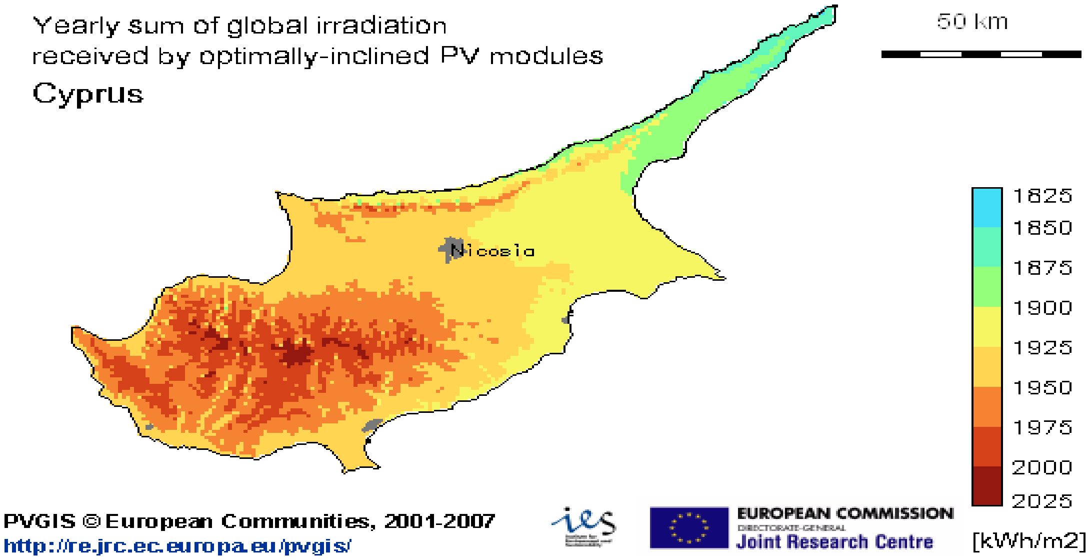
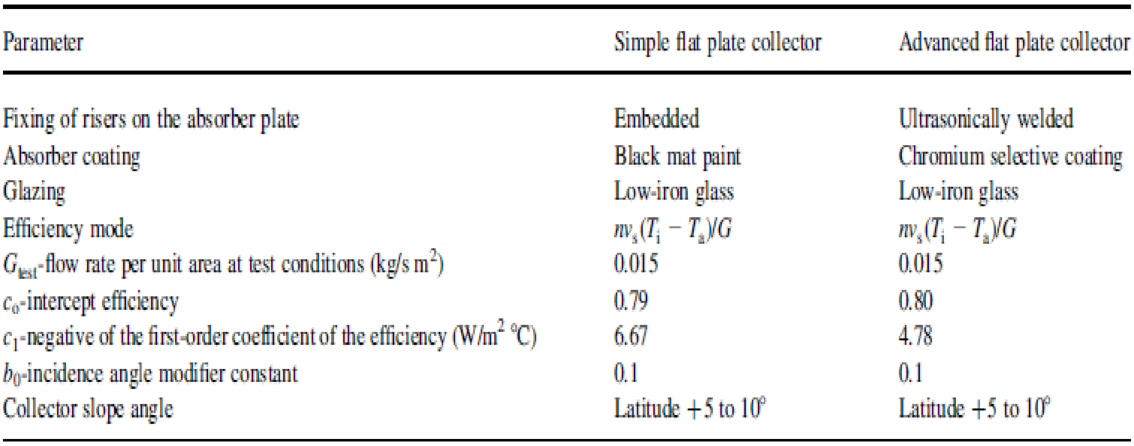
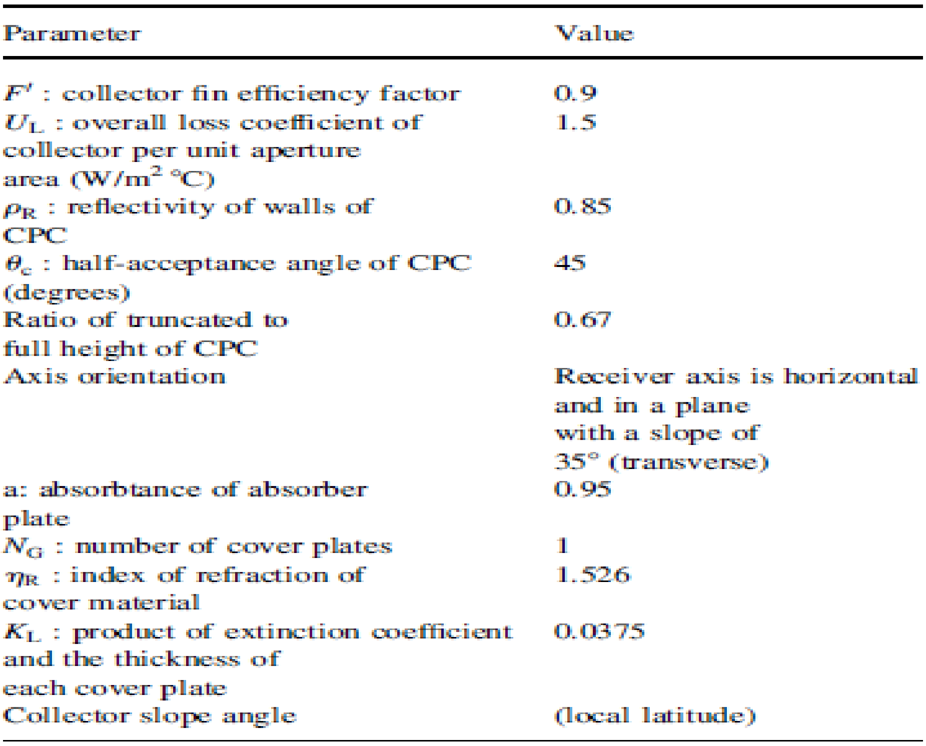


Figure 3.20 Yearly sum of global irradiation received by optimally inclined PV modules for Cyprus [39]



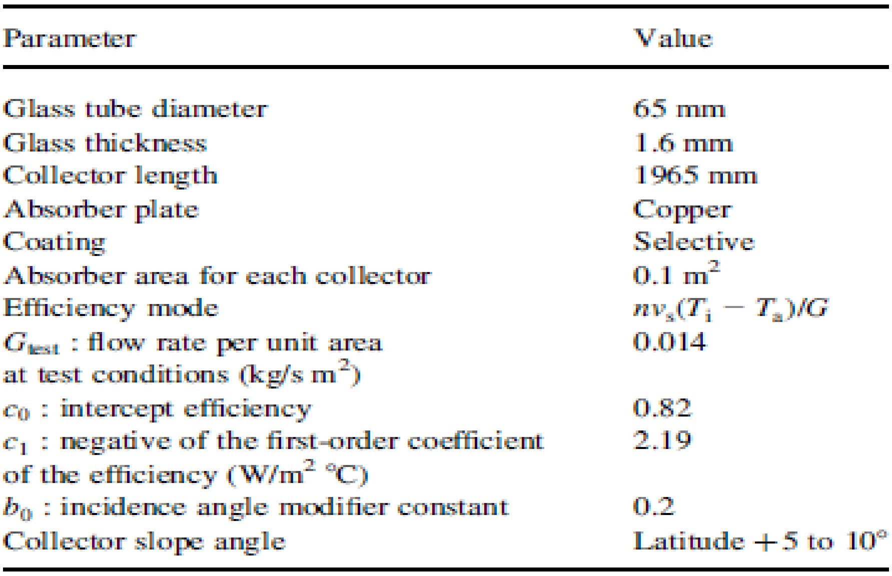
Parameter	Simple flat plate collector	Advanced flat plate collector
Fixing of risers on the absorber plate	Embedded	Ultrasonically welded
Absorber coating	Black mat paint	Chromium selective coating
Glazing	Low-iron glass	Low-iron glass
Efficiency mode	$nv_s(T_i - T_a)/G$	$nv_s(T_i - T_a)/G$
G_{test}-flow rate per unit area at test conditions (kg/s m^2)	0.015	0.015
c_0-intercept efficiency	0.79	0.80
c_1-negative of the first-order coefficient of the efficiency (W/m^2 °C)	6.67	4.78
b_0-incidence angle modifier constant	0.1	0.1
Collector slope angle	Latitude +5 to 10°	Latitude +5 to 10°

Figure 3.21 Characteristics of a typical FPC [37]



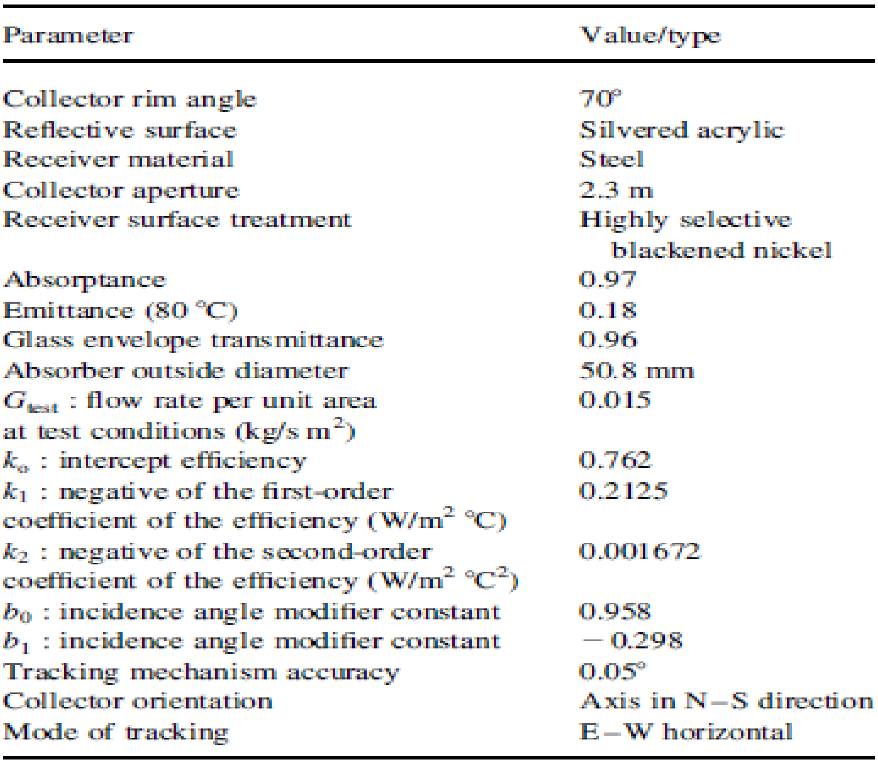
Parameter	Value
F' : collector fin efficiency factor	0.9
U_L : overall loss coefficient of collector per unit aperture area (W/m^2 °C)	1.5
ρ_R : reflectivity of walls of CPC	0.85
θ_c : half-acceptance angle of CPC (degrees)	45
Ratio of truncated to full height of CPC	0.67
Axis orientation	Receiver axis is horizontal and in a plane with a slope of 35° (transverse)
a: absorbtance of absorber plate	0.95
N_G : number of cover plates	1
η_R : index of refraction of cover material	1.526
K_L : product of extinction coefficient and the thickness of each cover plate	0.0375
Collector slope angle	(local latitude)

Figure 3.22 Characteristics of a typical CPC [37]



Parameter	Value
Glass tube diameter	65 mm
Glass thickness	1.6 mm
Collector length	1965 mm
Absorber plate	Copper
Coating	Selective
Absorber area for each collector	0.1 m^2
Efficiency mode	$n v_s (T_i - T_a)/G$
G_{test} : flow rate per unit area at test conditions (kg/s m^2)	0.014
c_0 : intercept efficiency	0.82
c_1 : negative of the first-order coefficient of the efficiency (W/m^2 °C)	2.19
b_0 : incidence angle modifier constant	0.2
Collector slope angle	Latitude + 5 to 10°

Figure 3.23 Characteristic of typical ETC [37]



Parameter	Value/type
Collector rim angle	70°
Reflective surface	Silvered acrylic
Receiver material	Steel
Collector aperture	2.3 m
Receiver surface treatment	Highly selective blackened nickel
Absorptance	0.97
Emittance (80 °C)	0.18
Glass envelope transmittance	0.96
Absorber outside diameter	50.8 mm
G_{test} : flow rate per unit area at test conditions (kg/s m^2)	0.015
k_o : intercept efficiency	0.762
k_1 : negative of the first-order coefficient of the efficiency (W/m^2 °C)	0.2125
k_2 : negative of the second-order coefficient of the efficiency (W/m^2 °C^2)	0.001672
b_0 : incidence angle modifier constant	0.958
b_1 : incidence angle modifier constant	-0.298
Tracking mechanism accuracy	0.05°
Collector orientation	Axis in N–S direction
Mode of tracking	E–W horizontal

Figure 3.24 Characteristics of typical IST PTC [37]

3.6 Evaporators

“In a refrigeration system, the evaporator is used for vaporizing the refrigerant by transferring heat from surroundings (mainly from the refrigeration desired space) to the refrigerant liquid which it contains.” [1] The evaporator types are given below.

- Dry Type Evaporators
- Liquid Type(liquid overflowed) Evaporators
- Air Cooling Evaporators
- Naked Pipe Evaporators
- Lamella Evaporators
- Plate Type Evaporators
- Fined Pipes Evaporators
- Liquid Cooled Evaporators
- Double Piped Evaporators

- Naked Piped Serpentine Evaporators
- Spiral Piped Evaporators
- Pool Type Evaporators
- Injection Type Evaporators.
- Carcass-piped Cluster Evaporators

3.7 Condensers

They are the mechanism that works for converting the high pressurized and heated vapor in to liquid form by transferring the observed heat to the surroundings [1]. The condensers types are given below.

- Air cooled Condensers
- Water Cooled Condenser Types
- Tube in Tube Type Condensers
- Spiral Type Condensers
- Vertically Located Carcass Piped Condensers
- Horizontally Located Carcass Piped Condensers.
- Atmospheric Type Condensers
- Evaporative-Water Injection Type Condensers

3.8 Throttling (Expansion) Valves

“Expansion valves are used for decreasing the refrigerant liquid’s velocity and pressure to desired velocity and pressure.” [1] The expansion valve types are given below.

- Manuel Type Valves
- Automatic Type Valves
- Thermostatic Type Valves
- Valves Work with Electricity
- Capillary Pipes

3.9 Designing parameters

Basically, the cycle is intermittent because cold or heat production is not continuous. Cold or heat production proceeds only during part of the cycle. When there are two adsorbtion beds in the unit, they can be operated separately and the cold or heat production is quasicontinuous. When there are two adsorbers or more, other types of cycles can be processed and that enhance the cycle performance [16].

The design of the system depends on the temperature differential between the condenser and evaporator and the temperature of the third heat supply. The adsorbent and adsorbate pairs and also pressure variables are chosen with respect to the temperature differentials and the third heat supply’s temperature. When the adsorption capacity of the adsorbent bed increases the efficiency of the system also increases. The adsorbate and adsorbent pair’s specific heat, heat transfer coefficient, density and also the relation properties with each other are very important. The adsorption relation between each other must be powerful for higher efficiency. But if the adsorption is so much powerful than the desired adsorption the regeneration gets harder and this not a valuable factor for the system operating. The negative effects on the environment and on the healthcare also the cost prices must be considered.

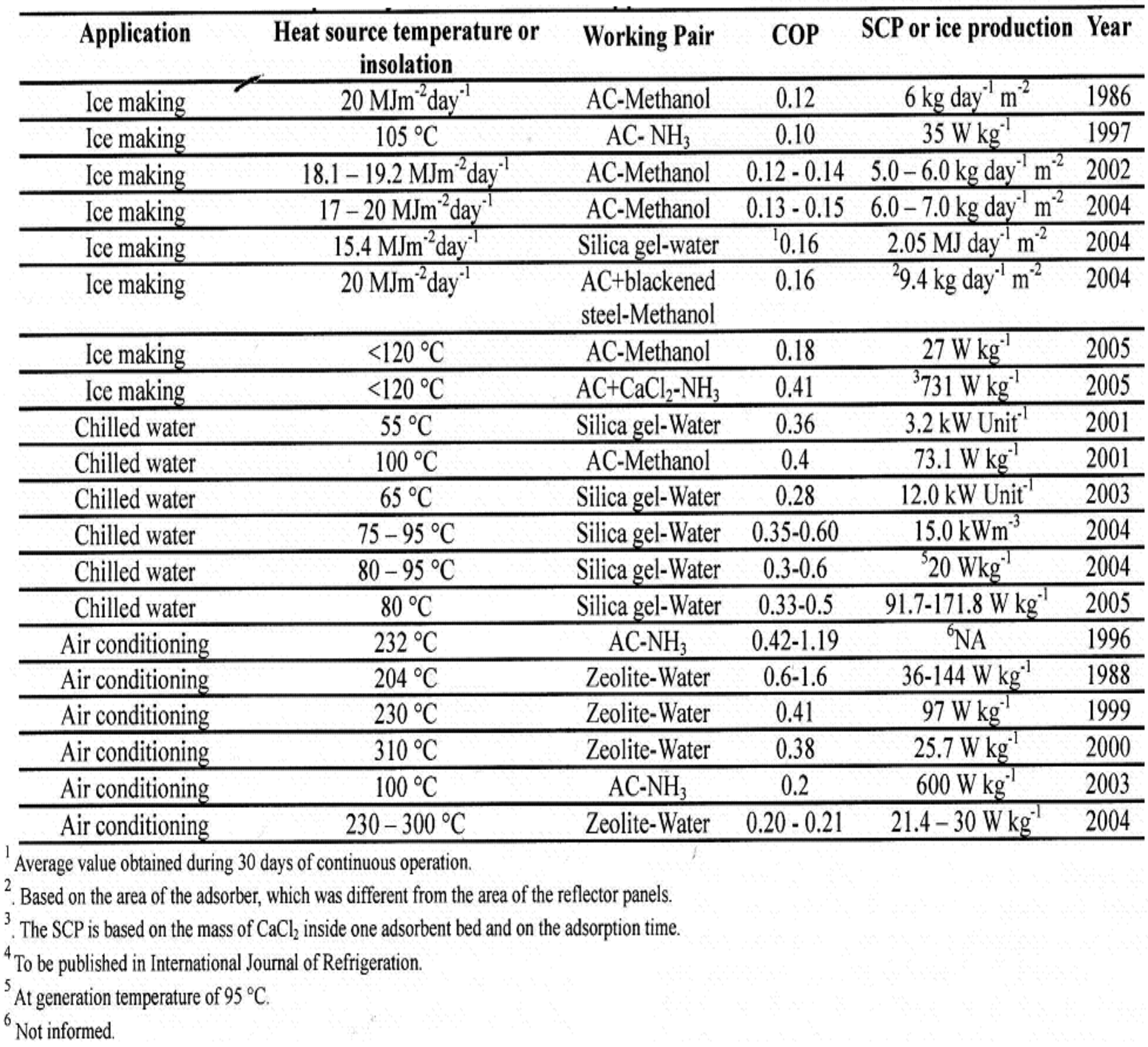
When designing the adsorption adsorbent bed, the main problem is; it is required a highly efficient designing for controlling the mass and heat transfer rates in the adsorption beds. The lowered level of heat transfer capacity in the adsorbent cause’s lower heat transfer rate and it makes the adsorption and desorption period longer. “The low working pressure in evaporation period causes some leakage problems occur. These problems shortened the valuable life of adsorption and reduce the performance. This problem can only be blocked by using high capacity vacuum technology.”[16] High vacuum technology requires high set up and maintains cost. Adsorption cycled refrigeration systems does not require any mechanically worked power. But in this types of developed systems the valves which are located after evaporator and before condenser are turned on or off manually. Automatically controlled types of systems can also be considered for further studies to reduce the cycle time. “In double reacted adsorption adsorbent beds these automatically controlled systems are highly required for operating the system serially.” [6]

There are two main aims for the development of adsorption refrigeration, one is for low efficiency applications with simple system structure, such as locomotive air conditioners and solar powered adsorption ice makers and the other is for high efficiency applications [29]. “SJTU developed an adsorption chiller powered by low temperature heat source and it is commercialized. It could be coupled with solar powered water heating system to work as a solar air conditioner.” [29]

With respect to the energy and environment, for air conditioning, the cooling COP must higher than 1 and possibly higher than 1.2 to compete favorably with a mechanical vapor compression unit (COP=3) powered with electricity provided by a thermal power station. [29] “The whole energy consumption for heating in winter and for cooling in summer should not exceed the consumption of energy just for heating in winter time with a convectional heating method if the heat recovery efficiency could reach 0.75.” [29]

“At the end of the 1980’s a solar adsorption air conditioning system presented that had 20 m^2 of solar panel area and Zeolite-water pair used for working pair. This system designed for refrigerate a 12 m^3 room for food preservation.” [32] “When the insolation received by the solar collectors was about 22 Mjm^{-2}, the cold room could store 1000 kg of vegetables with rotation of 130 kg refrigerant per day for a temperature difference of 20 °C between the ambient temperature and interior temperature and the COP was 0.10.” [32] Most solar refrigeration systems especially the icemaker prototypes have a daily ice production between 4 and 7 kg per m^2 of solar collector, with a solar COP between 0.10-0.15 [32]. In the Table 3.6 it is given some of the best performances obtained by different prototypes manufactured during in the last 25 years. The prototypes of theoretical technologies for adsorption refrigeration have been extensively studied in academic as well as industry sectors. A large amount of patents have been taken in different countries around the world, which lead to substantial improvements on system COP and SCP, and novel application in various area [33]. In Figures 3.25, 3.26 and 3.27 the numbers of cited patents by countries, number of cited patents by leader organizations in this sector and number of cited patents issued in the period of 2000-2005 are given. “For adsorption refrigeration system, significant achievements have been obtained on the use of various technologies to reduce the complexity of system structure, initial cost, to increase the system operation reliability as well as energy performance.” [33] Heat pipe technology is a useful tool in improving the heat transfer and simplifies the structure in system level, further study should be conducted to utilize this technology with increased ratability and lower initial investment [33]. “Based on optimized control operation via computer technology, high system performance can be achieved.”[33] Hence, combining information technology into adsorption system should be an important direction to improve adsorption refrigeration system [33]. “Various heat and mass recovery strategies are proved to be effective on improving the energy performance in terms of COP. However intensive research is still needed to enhance the amount of recovered thermal energy, without adding complexity in to the design of the system and operation of the system and also still needed to optimize the recovered mass rate to achieve the best effect.” [33]

“In order to become a realistic alternative to the vapor compression refrigeration systems, solid-gas sorption refrigeration systems must exhibit comparably high performances, and the low COP is the main drawback for the solid-gas adsorption or desorption refrigeration systems, which has been limiting the wide application of sorption refrigeration technology; a good heat and mass transfer are crucial for the high performance of sorption refrigeration systems.” [35]



Application	Heat source temperature or insolation	Working Pair	COP	SCP or ice production	Year
Ice making	20 $MJm^{-2}day^{-1}$	AC-Methanol	0.12	6 kg day^{-1} m^{-2}	1986
Ice making	105 °C	AC- NH_3	0.10	35 W kg^{-1}	1997
Ice making	18.1 – 19.2 $MJm^{-2}day^{-1}$	AC-Methanol	0.12 - 0.14	5.0 – 6.0 kg day^{-1} m^{-2}	2002
Ice making	17 – 20 $MJm^{-2}day^{-1}$	AC-Methanol	0.13 - 0.15	6.0 – 7.0 kg day^{-1} m^{-2}	2004
Ice making	15.4 $MJm^{-2}day^{-1}$	Silica gel-water	[1]0.16	2.05 MJ day^{-1} m^{-2}	2004
Ice making	20 $MJm^{-2}day^{-1}$	AC+blackened steel-Methanol	0.16	[2]9.4 kg day^{-1} m^{-2}	2004
Ice making	<120 °C	AC-Methanol	0.18	27 W kg^{-1}	2005
Ice making	<120 °C	AC+$CaCl_2$-NH_3	0.41	[3]731 W kg^{-1}	2005
Chilled water	55 °C	Silica gel-Water	0.36	3.2 kW $Unit^{-1}$	2001
Chilled water	100 °C	AC-Methanol	0.4	73.1 W kg^{-1}	2001
Chilled water	65 °C	Silica gel-Water	0.28	12.0 kW $Unit^{-1}$	2003
Chilled water	75 – 95 °C	Silica gel-Water	0.35-0.60	15.0 kWm^{-3}	2004
Chilled water	80 – 95 °C	Silica gel-Water	0.3-0.6	[5]20 Wkg^{-1}	2004
Chilled water	80 °C	Silica gel-Water	0.33-0.5	91.7-171.8 W kg^{-1}	2005
Air conditioning	232 °C	AC-NH_3	0.42-1.19	[6]NA	1996
Air conditioning	204 °C	Zeolite-Water	0.6-1.6	36-144 W kg^{-1}	1988
Air conditioning	230 °C	Zeolite-Water	0.41	97 W kg^{-1}	1999
Air conditioning	310 °C	Zeolite-Water	0.38	25.7 W kg^{-1}	2000
Air conditioning	100 °C	AC-NH_3	0.2	600 W kg^{-1}	2003
Air conditioning	230 – 300 °C	Zeolite-Water	0.20 - 0.21	21.4 – 30 W kg^{-1}	2004

[1] Average value obtained during 30 days of continuous operation.

[2]. Based on the area of the adsorber, which was different from the area of the reflector panels.

[3]. The SCP is based on the mass of $CaCl_2$ inside one adsorbent bed and on the adsorption time.

[4] To be published in International Journal of Refrigeration.

[5] At generation temperature of 95 °C.

[6] Not informed.

Table 3.6 Performance of adsorption systems for different applications [32]

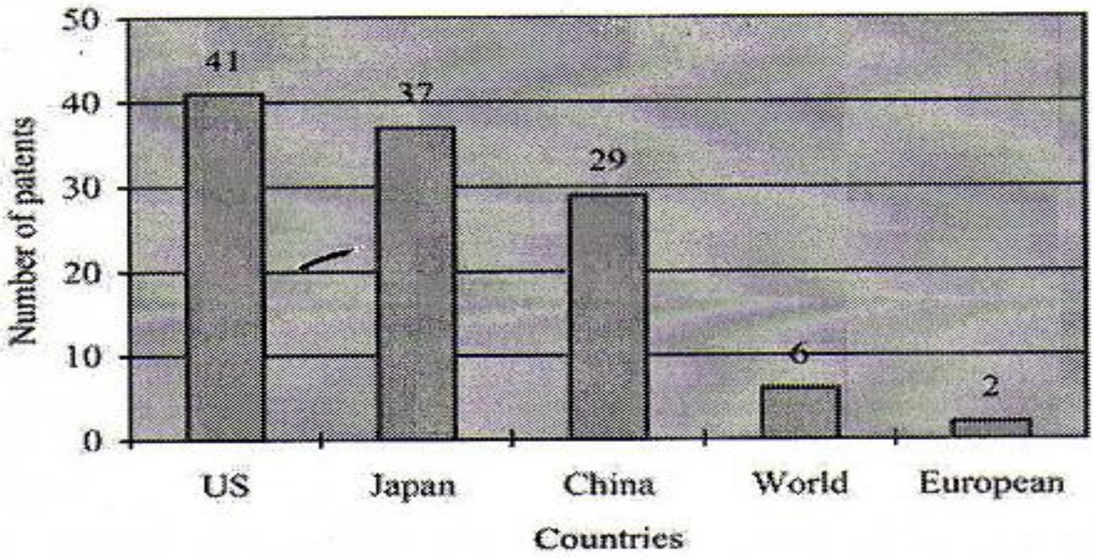


Figure 3.25 Number of citied patents by countries [33]

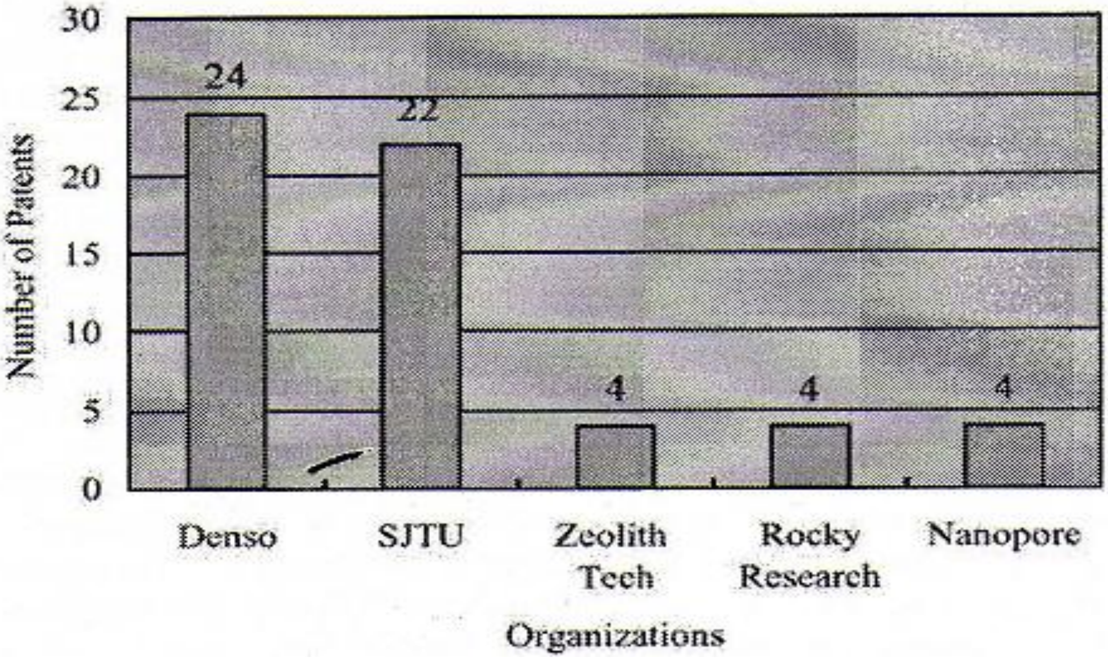


Figure 3.26 Number of citied patents by organizations [33]

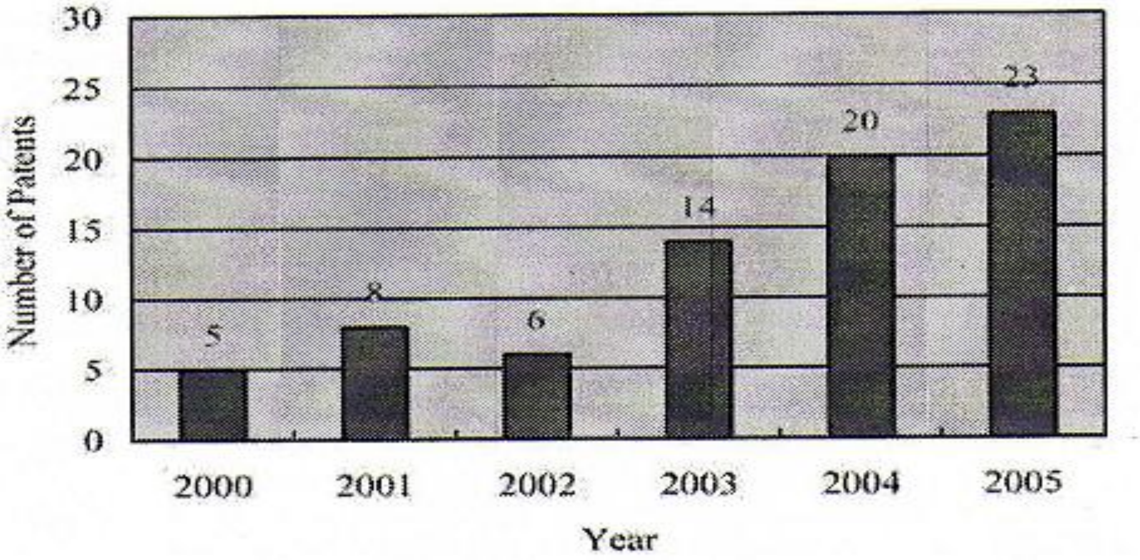


Figure 3.27 Number of citied patents issued in the period of 2000-2005 [33]

To increase the value of COP and to gather much sufficient cooling for the required propose a combined system can be considered like shown in Figure 3.28 while design the whole system.

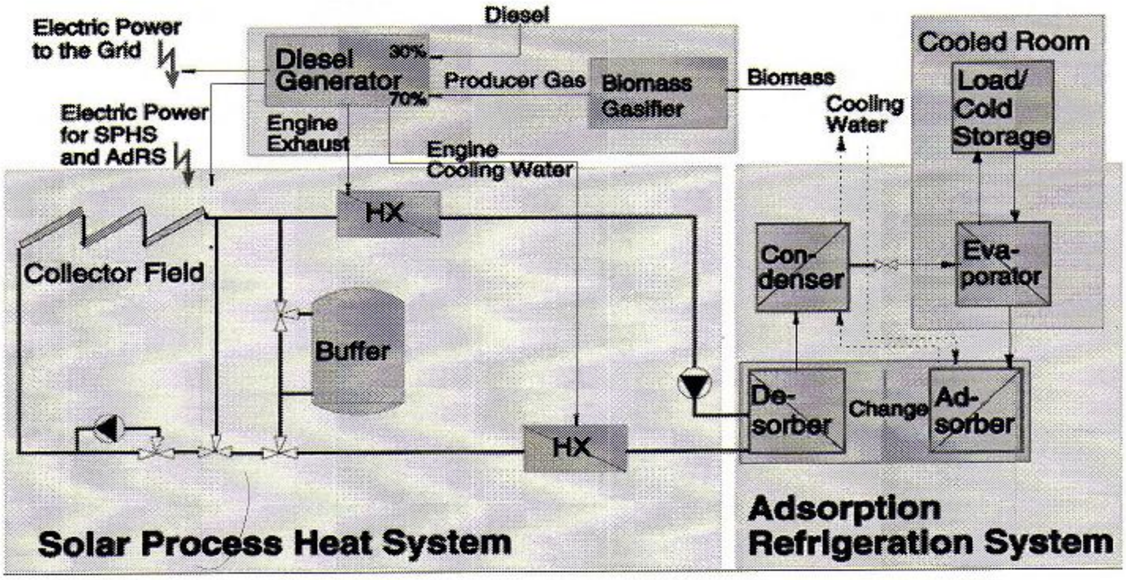


Figure 3.28 A combined adsorption refrigeration system [7]

3.9.1 Advantages and Disadvantages of Adsorption Refrigeration

The advantages of the solar adsorption cooling systems are given as;

- They do not require moving parts for circulation of working fluid
- They have long life time
- They operate without noise and vibration
- They have simple principle of working
- They do not require frequent maintenance
- They are environmental friendly since do not contain any hazardous materials for environment
- They can be employed as thermal energy storage device
- They can operate with thermal driving energy sources such as waste heat, solar, and geothermal energies, etc. [14]

The disadvantages of the solar adsorption cooling systems are given as;

- They have low COP values
- They intermittently working principles
- They require high technology and special designs to maintain high vacuum
- They have large volume and weight relative to traditional mechanical heat pump systems [14]

CHAPTER 4

IMPLEMENTATION

4.1 Thermal Processing and Thermal Analysis of the System

The system's basic thermal analysis's going to be introduced with the help of the Figure 4.1 (Schematic of a basic adsorption heat pump thermodynamic cycle in Clapeyron Diagram and in a basic structural schema) and Figure 4.2 (Heat transfer configuration of ideal adsorption heat pump cycle) [14]. The adsorption heat pump cycle can be considered as two separated cycles. The first cycle is a reversible heat pump in which the working fluid is vaporized in evaporator by taking heat (Q_L) from the low-level temperature source and releasing (Q_{CC}) heat to the first intermediate temperature source. The second cycle is a heat engine, which receives heat (Q_Z) from the high-temperature source and releases heat (Q_C) to the second intermediate temperature source. The transfer of heat (Q_C) to the second intermediate temperature source occurs during the condensation of working fluid in condenser. This cycle represents desorption process." [14] "It is assumed that the work obtained in the heat engine is employed to run the heat pump." The temperatures of intermediate sources which are T_C and T_{CC}, are generally close to each other" [14] Because of this reason three temperature levels can be considered for an adsorption heat pump instead of four level temperatures. [14] The ideal coefficient of performance (COP) of an adsorption heat pump cycle can be calculated as with respect to the reference [14]:

For cooling:

$$COP_{ref} = \frac{Q_L}{Q_Z} = 1 + \frac{1-(T_C / T_Z)}{(T_C / T_L) - 1} \tag{4.1}$$

For heating:

$$COP_h = \frac{Q_C}{Q_Z} = 1 + \frac{1-(T_C/T_Z)}{(T_C/T_L)-1} \qquad (4.2)$$

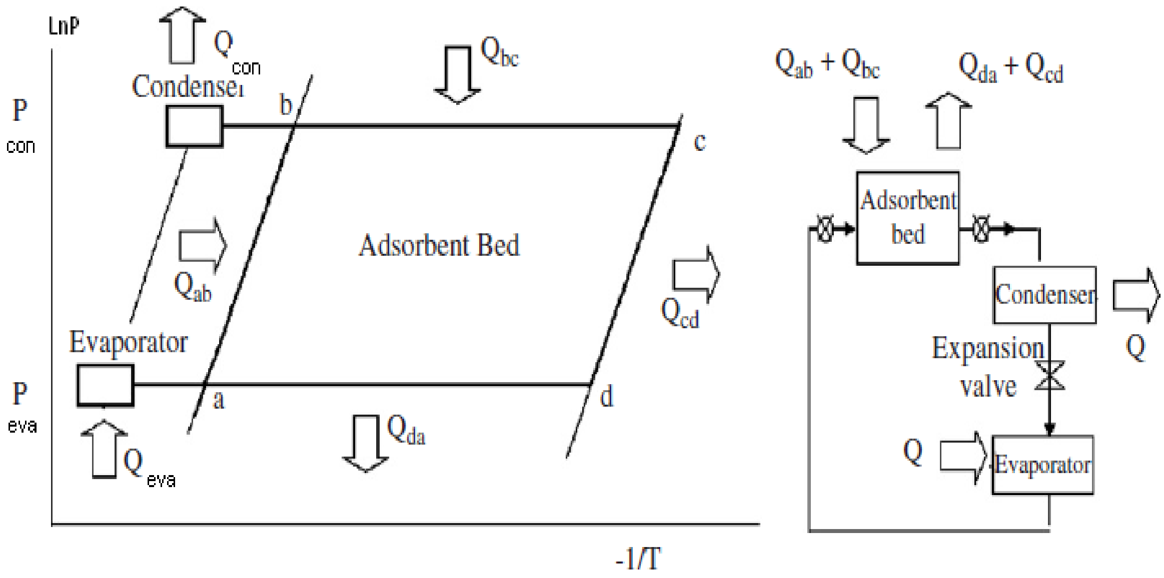


Figure 4.1 Thermodynamic cycle of a basic adsorption heat pump [14]

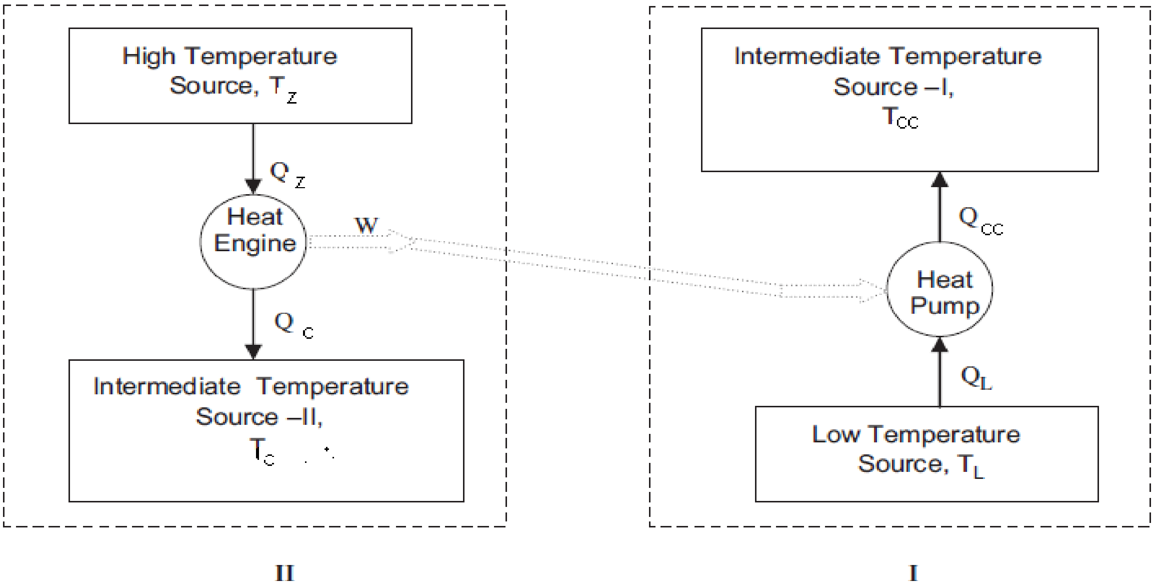


Figure 4.2 Heat transfer configuration of ideal adsorption heat pump cycle [14]

The specific cooling power can also be performance criteria for the systems. The specific cooling power was defined as the cooling capacity produced in an hour for a kilogram of adsorbent. If regeneration and preheating is used in the system, the coefficient of performance was defined as the specific cooling power divided by the total amount of heat input to the adsorber during the preheating and regeneration periods for a kilogram of the adsorbent in the basic of an one-hour operation with respect to the reference [12].

In the Figure 4.1, the period from a to b is called as “Isosteric heating”. “The valves between the adsorbent bed and the condenser and the evaporator are closed. The temperature of adsorbent bed is increased from T_a to T_b by heating the adsorbent bed without desorption.” [14] The amount of heat, which should be transferred to the adsorbent bed to increase temperature of the adsorbent from T_a to T_b is given by the following relation. [14]

$$Q_{ab} = [m \; C_{p,z} + xC_{p,w} \; + m_{bed}C_{p,bed}]. \; T_b - T_a \qquad (4.3)$$

After the “Isosteric Heating” period the “Isobaric Desorption” period begins. This period is represented from b to c in Figure 4.1. The heating process is continued. “The valve between the adsorbent bed and condenser is opened. Desorption process is started and water vapor condensed in the condenser. The pressure of the period remains constant. In the equation 4.4, the first part of the equation causes to increase the temperature of adsorbate-adsorbent pair and adsorbent bed while the other part causes the desorption process”[14]

$$Q_{bc} = \left[m \; C_{p,z} + xC_{p,w} \; + m_{bed}C_{p,bed}\right]. \; T_c - T_b \; + m\Delta H_a \; x_c - x_b \qquad (4.4)$$

After the “Isobaric Desorption” period the “Isosteric Cooling” period begins. This period is represented from c to d in Figure 4.1. The valve between the condenser and adsorbent bed is closed and the temperature of adsorbent bed (T_c), which is the maximum temperature of the cycle, is decreased to T_d. During the process the pressure of the adsorbent is decreased to evaporator value. [14] The following relation is given for calculation of the “Isosteric Cooling” period’s rejected heat amount.

$$Q_{cd}=[m\,(C_{p,z}+xC_{p,w})+m_{bed}C_{p,bed}].(T_d-T_c) \qquad (4.5)$$

After the “Isosteric Cooling” period the “Isobaric Adsorption” period begins. This period is represented from d to a in Figure 4.1. “The valve between the adsorbent bed and evaporator is opened and vaporization of the adsorbate in the evaporator is started. During adsorbing of the adsorbate in the adsorbent, heat is released due to heat of adsorption. This generated heat should be removed from the adsorbent bed and the temperature of adsorbate-adsorbent pair and container should be decreased to T_a.” [14] This period’s transferred heat amount value is given by the following relation.

$$Q_{da}=[m\,(C_{p,z}+xC_{p,w})+m_{bed}C_{p,bed}].(T_a-T_d)+m\Delta H_a(x_a-x_d) \qquad (4.6)$$

The Q_{con} (heat released because of the condensation) and Q_{eva} (the amount of heat required for evaporation) is determined by the formulas with respect to the reference [14,8]

$$Q_{eva}=m\Delta x\Delta H_v+m\Delta xC_{p,w}(T_{eva}-T_{con}) \qquad (4.7)$$

$$Q_{con}=m\Delta x\Delta H_v \qquad (4.8)$$

“The x represents an equilibrium curve where the weight ratio of adsorbate and dry adsorbent is constant ($m_{adsorbate}/m_{adsorbent}$).” [8] The x curve lines which are called the isoster lines; can be seen in Figure 4.1. The isosteric lines respectively shift from one to the other while the “Isobaric” periods are processing. They are constant in “Isosteric” periods in as seen in Figure 4.1.

The COP of the basic adsorption heat pump cycle can also be determined as with respect to the reference [14];

For Cooling:

$$COP_{ref}=Q_{eva}/(Q_{ab}+Q_{bc}) \qquad (4.9)$$

For Heating:

$$COP_{h}=(Q_{con}+Q_{cd}+Q_{da})/(Q_{ab}+Q_{bc}) \qquad (4.10)$$

4.2 Mass and Energy Balance

When we assume there are seven basic components, including four adsorbers, one evaporator, one condenser and one expansion valve in the system which also shows and proves the system can also be designed as a multi-bed adsorption heat pump as shown in Figure 4.3. “During this cycle, the adsorbers consecutively and orderly proceed to four different modes of adsorption, preheating, regeneration and precooling.” [12] In the Figure 4.3. while the (A) is in adsorption mode, (B),(C) and (D) would be in preheating, regeneration and precooling modes in an order, respectively. When the (A) switches to the preheating mode, the (B),(C) and (D) would switch to the regeneration, precooling and adsorption modes in an order, respectively [12]. When the adsorber (A) accomplishes the four modes, the system would proceeds to the next cycle from the beginning [12]. We can also drive the cycle without precooling, preheating and regeneration modes or we may install one or two of the modes in to the system respectively. But the adsorption mode is cannot be neglected and it is the primary mode for the cycle. This time only a one adsorber bed can make preheating, desorption, precooling and adsorption sections in a sequence mode.

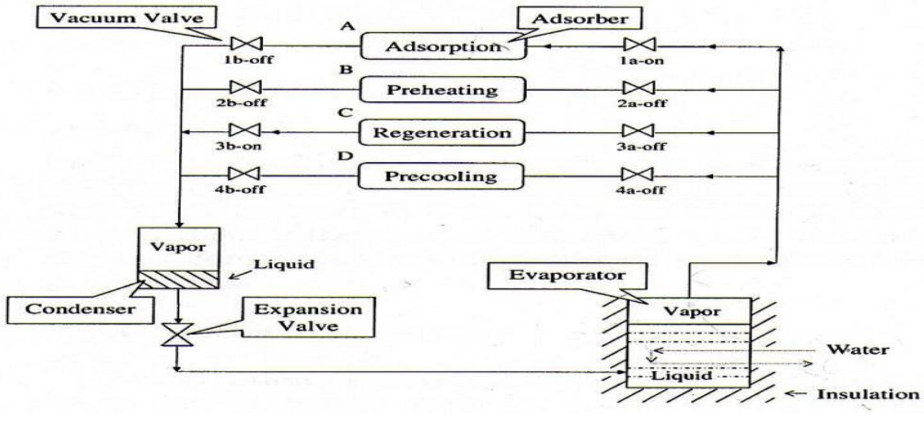


Figure 4.3 A multi-bed adsorption heat pump [12]

“A Solid-side resistance model was used to analyze the heat and mass transfer in the adsorber. For convenience of the analysis, the adsorbents were considered as spherical particle with an effective radius” [12]. And the governing equations can be expressed as follows with respect to the reference [12]:

$$\frac{\partial x}{\partial t^*} = \frac{1}{r^{*2}} \frac{\partial}{\partial r^*}[r^{*2} D^* \frac{\partial x}{\partial r^*}] \tag{4.11}$$

The Initial condition is going to be: x(r*,t*=0)=x(r*) with respect to the reference [12].

Boundary conditions are going to be as seen below [12]:

$$\frac{\partial x}{\partial r*}|_{r*} = 0$$

In adsorption time: $x(r^*=1)=x(P_{eva}, T_{wet})$. (4.12)

Regeneration time: $x(r^*=1)=x(P_{con}, T_{wet})$.

Preheating/Precooling time: $m_{vap} + m_{ads}=j$ constant.

Average adsorbate content is going to be as seen below with respect to the reference [12]

$$x_{ave} = 3\int_0^1 (r^{*2} x) dr^* \tag{4.13}$$

Energy balance of wet adsorbent with respect to the reference [12]:

$$\frac{dT_b}{dt^*} = \gamma_2 (T_i - T_b) + \gamma_1 (\frac{\partial x_{ave}}{\partial t^*}) \tag{4.14}$$

Initial condition for the above equation (4.14) is going to be: T_b (t*=0) = T_b with respect to the reference [12].

Energy balance of the insert;

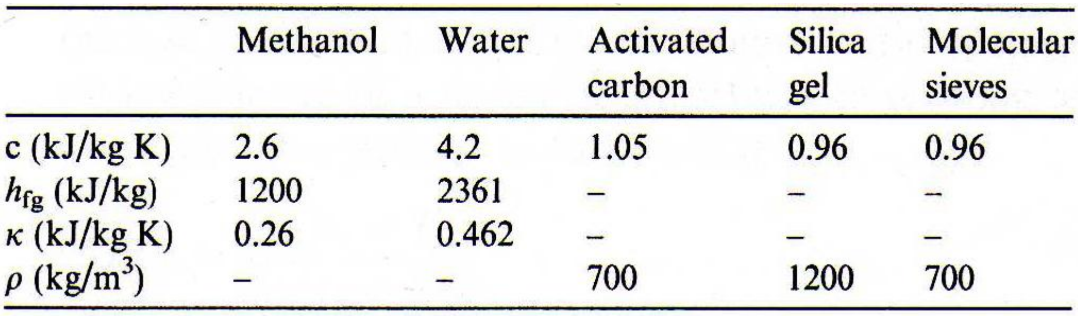
$$\frac{dT_i}{dt^*} = (\frac{1}{\tau_i^*})(1+R_{th}^*)(T_{eq} - T_i) \tag{4.15}$$

Where;

$$\mathrm{T_{eq}} \equiv (\mathrm{T_\infty} + \mathrm{R_{th}^*} \mathrm{T_{wet}})(1+\mathrm{R_{th}^*}), \mathrm{R_{th}^*} \equiv \mathrm{UA_b} / \mathrm{hA_S} \tag{4.16}$$

Initial condition: T_i ($t^*=0$)=T_i [12] (4.17)

Where in above equations, the parameters γ_1 and γ_2 are a function of the specific heat of wet adsorbent (c_b). “The c_b value was evaluated through the following relationship: $c_b=c_{ads}x_{ave}+c_j$” [12]. “T represents the average temperature of the fluid in the flow passage. This temperature was considered to vary with time.” [12] In this work, three adsorption pairs were considered. The corresponding physical properties are given in the Table 4.1.



	Methanol	Water	Activated carbon	Silica gel	Molecular sieves
c (kJ/kg K)	2.6	4.2	1.05	0.96	0.96
h_{fg} (kJ/kg)	1200	2361	–	–	–
κ (kJ/kg K)	0.26	0.462	–	–	–
ρ (kg/m^3)	–	–	700	1200	700

Table 4.1 Physical properties of adsorbates and adsorbents [12]

The instantaneous energy balance of the fluid in the flow passage by using a one dimensional model can be expressed as follows with respect to the reference [12]:

$$\frac{dT_w}{dz} = \frac{ph}{\dot{m}c_w}(T_i - T_w) \qquad (4.18)$$

Initial conditions are going to be as given below:

Adsorption/precooling model: $T_{w,i}(z=0)=T_C$,
Regeneration/preheating mode: $T_{w,i}(z=0)=T_G$, (4.19)

The exiting fluid's temperature ($T_{w,o}$) can be calculated by solving the above equation and it is expressed as seen below with respect to the reference [12]:

$$T_{w,o} = T_i - (T_i - T_{w,i}).\exp(-C_r^*) \qquad (4.20)$$

In the above equation C^*_r is defined as, [12]

$$(m_i c_i / \dot{m}\, c_w \tau_i) \qquad . \qquad (4.21)$$

During the heat transfer process the increase of energy in the fluid is equal to the energy delivered from the insert to fluid. [12]

$$T_\infty = T_i - (1/C_r^*)(T_{w,o-}T_{w,i}) \qquad (4.22)$$

"In the above equation, T_i and $T_{w,o}$ are unknowns. Hence the average temperature of the heating/cooling fluid (T_∞) in the flow passage cannot be solved explicitly. In order to solve the T_∞ value, an iteration scheme was adopted. In iteration scheme, an initial guessed T_∞ value is reassumed in every time step . This guessed value is substituted into Equation to solve a T_i value. Then substituting the T_i value into further more equations, a $T_{w,o}$ value and a new T_∞ value can be obtained. Using this new T_∞

value and repeating the steps over and over again then it is going to be seen that the solution would be more specific." [12]

4.3 Adsorption Isoster Lines and Heat of Adsorption

"Adsorption isoter lines were used to evaluate the mass concentration of adsorbate on the surface of the adsorbent for different vapor pressure and temperatures." [12] The equations that are given below are used for plotting the isoster diagrams.

For activated carbon-methanol pair and 13X molecular sieves-water pair [12];

$$Ln(P)=a(x)+[b(x)/T_b] \tag{4.23}$$

Where;

$$a(x)=a_0+a_1x+a_2x^2+a_3x^3 \tag{4.24}$$

$$b(x)=b_0+b_1x+b_2x^2+b_3x^3 \tag{4.25}$$

For silica gel-water pair; [12]

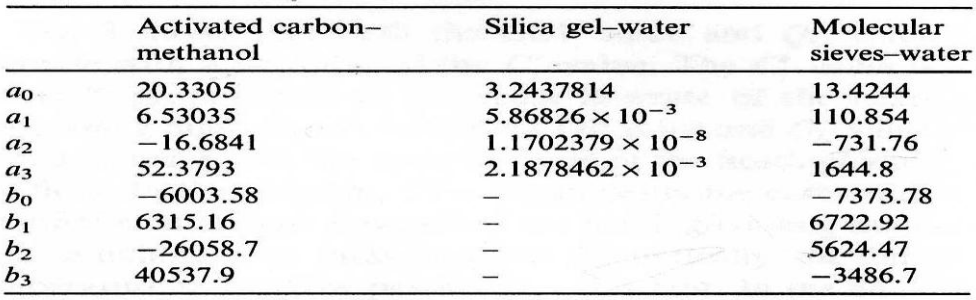
$$P/P_{sat}=(2.112x)^{[1+0.2843\exp(-10.28x)]}x(29.91P_{sat})^{[0.2843\exp(-10.28x)]} \tag{4.26}$$

Where;

$$\log_{10}(P_{sat}/218.167)=-\frac{u_0}{T_b}\left(\frac{a_0+a_1u_0+a_2u_0^3}{1+a_3u_0}\right)$$

$$u_0=647.27-T_b \tag{4.27}$$

In the equation (4.23), P is in mbar; and in the equation (4.26), P is in atm. In both these equations, T_b is in Kelvin. [12] The coefficients can be taken from Table 4.2 [12]. "Furthermore the heat of adsorption (ΔH_a) is considered as constant and its value is approximately equal to the 1.2 times of latent heat.” [12] Or it can be picked up from the Table 4.1.



	Activated carbon–methanol	Silica gel–water	Molecular sieves–water
a_0	20.3305	3.2437814	13.4244
a_1	6.53035	5.86826×10^{-3}	110.854
a_2	−16.6841	1.1702379×10^{-8}	−731.76
a_3	52.3793	2.1878462×10^{-3}	1644.8
b_0	−6003.58	–	−7373.78
b_1	6315.16	–	6722.92
b_2	−26058.7	–	5624.47
b_3	40537.9	–	−3486.7

Table 4.2 Coefficients for adsorption isotherms [12]

The maximum adsorption values that can be reached as a ratio (x) are given in the Figure 4.4 (a)-(c) [12].

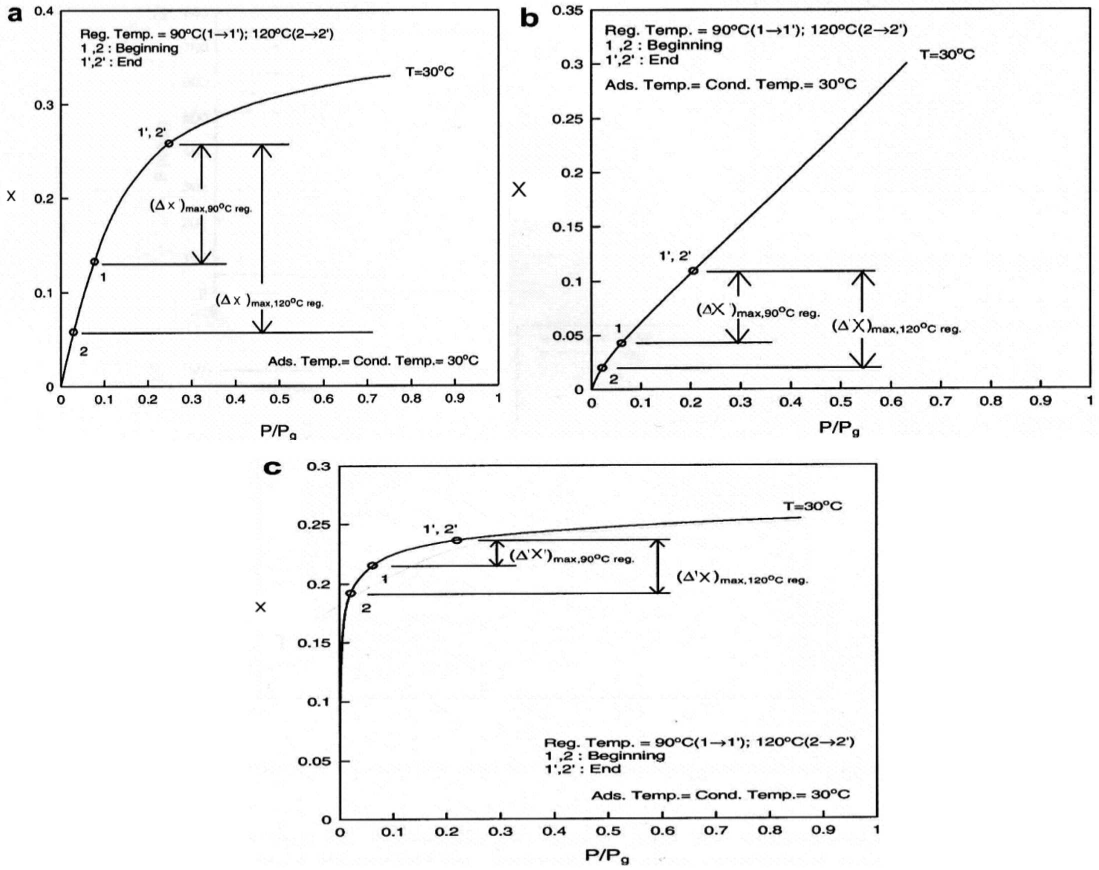


Figure 4.4 Δx_{max} for T_L=5C^o: (a) activated carbon-methanol; (b) silica gel-water; (c) 13X molecular sieves-water [12].

4.4 Saturation Vapor Pressure

For water the saturation vapor pressure in the evaporator and condenser was evaluated using the equation given below;

$$P_{sat}=\exp[\alpha_1 -(\alpha_2/T)] \tag{4.28}$$

"In which P is in mbar and T is in Kelvin, for methanol: α_1=20.84 and α_2=4696; for water α_1=20.5896 and α_2=5098,26" [12]. Thermodynamic property tables and charts of the adsorbates' can also be used for plotting the saturation line of the adsorbates on the isoster diagrams.

4.5 Solid-Side Mass Diffusivity

The solid-side mass diffusivity can be expressed as follows [12]

$$D=(D_o/\tau_t)\exp(-0.45\Delta H_a/\lambda\kappa T_b) \tag{4.29}$$

"The D_0 equals to 1.6×10^{-6} m^2/s; λ is 1.0; τ_t stands for the tortuosity of micropores, its value was selected as 3.0."[12]

4.6 The Time Variable of the System

The laboratory test results of silicagel and Zeolite 13X are given in the Figures 4.5. and 4.6 with respect to the reference [20]. There two linear curves are seen in both the Figure 4.5 and Figure 4.6 and also there is a breakeven point in the two Figures 4.5 and 4.6. The dynamic behavior of Zeolite 13X and Silicagel 490 under adsorption of water vapour is different. As shown in the related Figures, Zeolite 13X gives a higher water vapour sorption rate than Silicagel 490-water pair [20]. The breakeven point is approximately 300 for Silicagel 490 and 400 for Zeolite 13X as seen in the related

Figures. “For a t>1 h measurement the weight m(t) increases as a function of time (t) for the two sorbent materials Silicagel and Zeolite.” [20] “But it is reduced when compared to the previous time variables like t<1 h.” [14] “6 g of water on Silicagel 490 and 12 g of water on Zeolite 13X in the first 300 s for silicagel and in the first 400 s for Zeolite are adsorbed on the surface of the sorption materials.” [20] “These results have been considered in the design of the heat exchangers are the key components of a thermo-chemical energy storage systems.” [20] “In the experimental design, spherical Zeolite 13X particles with an average diameter of 1 mm and granular Silicagel 490 with a particle size distribution of 3-5 mm were use as sorbent materials the bulk densities are 650 kg/m^3 for Zeolite 13X and 470 kg/m^3 for Silicagel 490. The bet surface of Zeolite 13X is in the range of 500 m^2/g and 800m^2/g and for Silicagel 490 it is 400m^2/g. The sorbent material fixed bed was dried at a temperature of T=100 °C by the electrical heating and simultaneously pumping with the turbo-molecular and the mechanical vacuum pumps.” [15] The Schematic of the experimental set-up is given in Figure 4.7 [20]

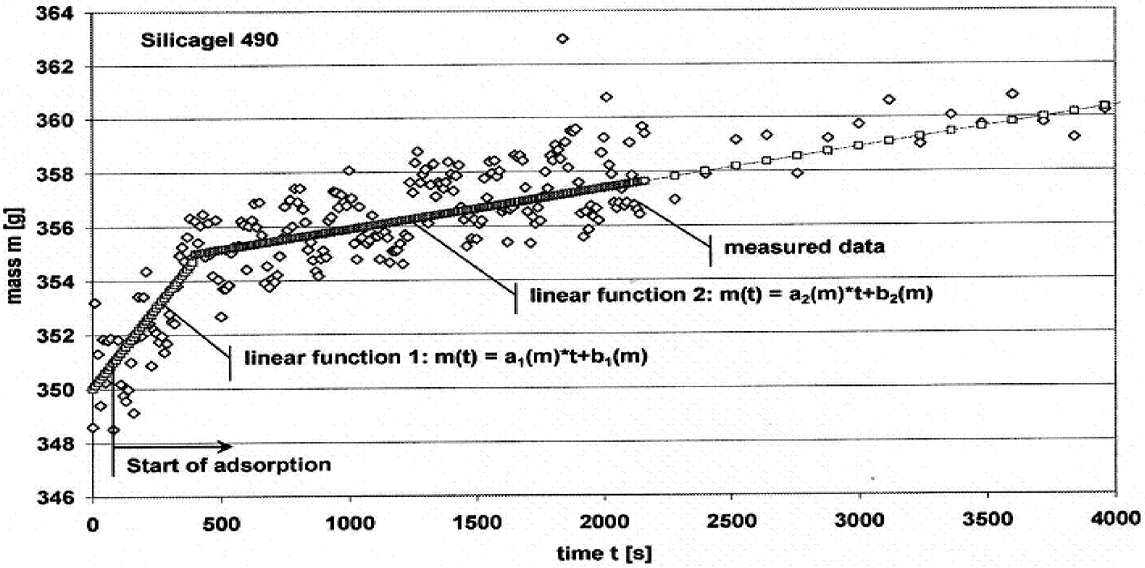


Figure 4.5 “Increase of mass function of time in the fixed bed of silicagel 490 by adsorption of water vapor at a pressure of P(H_2O, T=22 °C)=24.8 mbar.

In the Figure 4.5 the first t=300 s; the data points can be fitted by a linear function (1) with a slope of dm/dt =2x10^{-2} g/s, while the measured data at a time t >300 s can be approximated by a linear function (2) with a slope of dm/dt=1.7x13^{-3} g/s." [20]

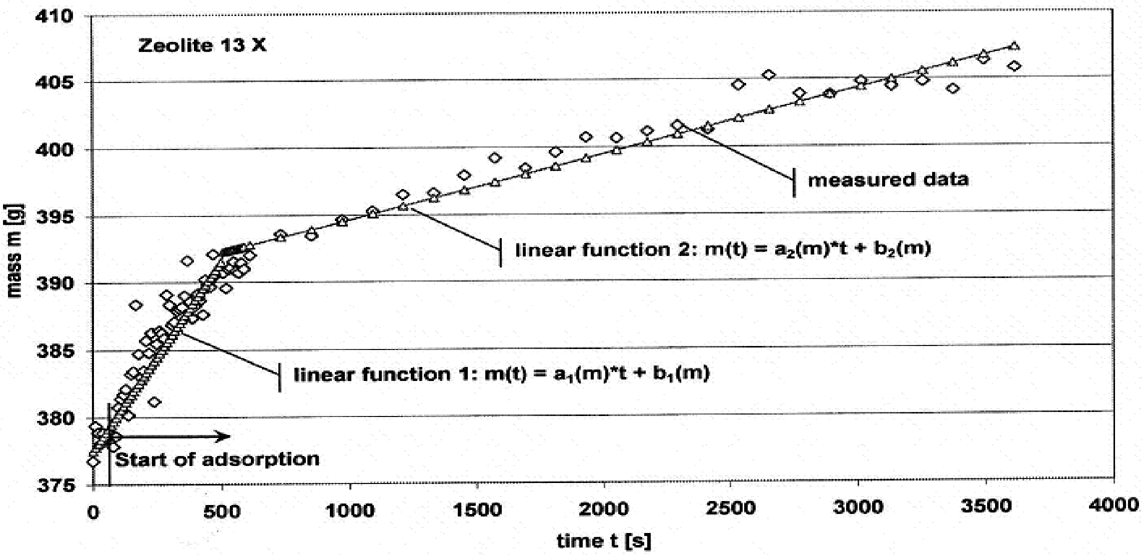


Figure 4.6 "Increase of mass function of time in the fixed bed of Zeolite 13X by adsorption of water vapor at a pressure of P(H_2O, T=22 °C)=24.8 mbar.

In the Figure 4.6 the first t=400 s; the data points can be fitted by a linear function (1) with a slope of dm/dt =3x10^{-2} g/s, while the measured data at a time t >400 s can be approximated by a linear function (2) with a slope of dm/dt=5x13^{-3} g/s." [20]

In the Figure 4.7, the experimental set up that is used for calculating the sorption rate has been given with respect to the reference [20]

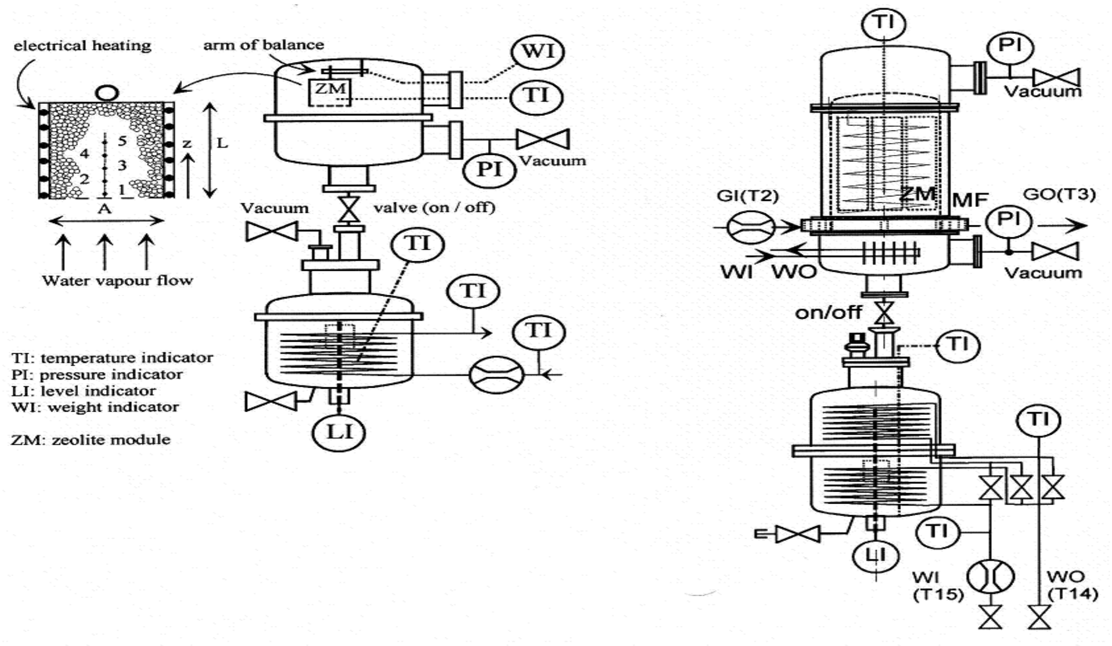


Figure 4.7 The experimental set up [20]

“Left of Figure 4.7 is the schematic of the experimental set-up with two vacuum chambers to measure time dependent mass adsorption on granular Zeolite and Silicagel and the temperature development as a function of time and position z (1 cm to 5 cm) in the fixed bed (ZM) of solid sorbent material (left). On the right it is seen a schematic of the laboratory sorption unit (with new rib heat exchanger) to measure temperature profiles and power out put is shown. In both schematics the lower chamber contains the sorbate water and the upper chamber shows the fixed sorbent bed container. In the schematics: PI pressure Indicator, TI Temperature Indicator, LI Level Indicator, GI Glycol In, GO Glycol Out, WI Water In, WO Water Out.” [20]

“With the help of another experiment on Zeolite with respect to the reference [21], it is seen the drying curves of Zeolite at different temperatures in the Figure 4.8. It is obtained at atmospheric pressure with small samples of Zeolite that are put in an electric stove at different temperatures. [21] From these curves we can understand that it is not suitable that to reduce the humidity from its initial original humidity at about 30-35 % at saturation to the below humidity at about 5%. “Further drying would result only a small improvement of the capacity of the Zeolite to adsorb water, but at the expense of a big amount of additional energy and time required.” [21] “Therefore 250 °C is an appropriate regeneration temperature. At this temperature one gets in 2,5 hours zeolite with 5% humidity (with 84% of the water extracted) [21]. At 200 °C, one could need 6 hours to reduce the humidity to 10% (extracting 66 % of the water).” [21] This high regeneration temperature of zeolite is a short coming of this adsorber and it is making it difficult to regenerate it either simple solar equipment.” [21]

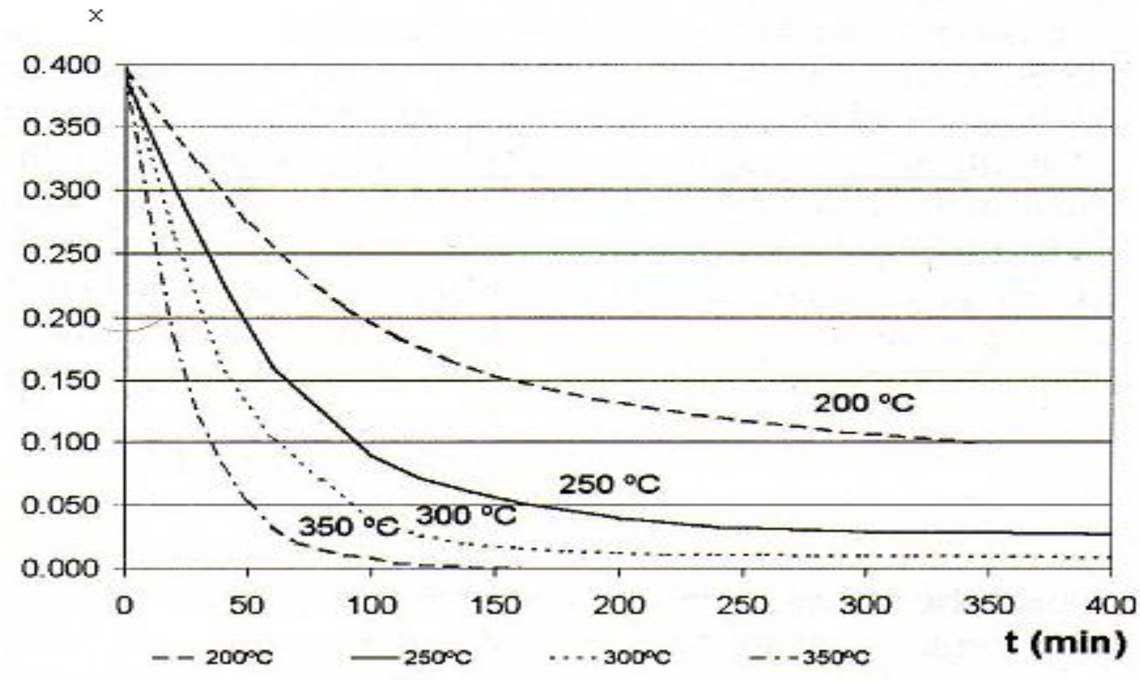


Figure 4.8 Drying curves of zeolite at different temperatures (humidity in dry base) [21]

4.7 Solar Collector Efficiency and Performance Analysis

The solar performance coefficient is defined as the ratio of the cooling power to the incident global irradiance during the whole day and it is given as seen below [28]:

$$COP_{sol} = Q_{ref}/G_{tot} \quad (4.30)$$

Q_{ref} is the cooling power which is produced at evaporator level and which can be written as [28]:

$$Q_{ref}=m_{ads}\Delta m[L(T_{eva})-C_{adsorbate}(T_{con}-T_{eva})] \quad (4.31)$$

Where, the amount of adsorbate circulating in the system Δm, should be known and it is defined as the difference between total adsorbed mass during the heating isoster and the total adsorbed mass during isoster. It is calculated as [28]:

$$\Delta m=m_{max}-m_{min}=m(T_{ads},P_{eva})-m(T_{con},P_{con}) \quad (4.32)$$

Where m_{max} is the adsorbed mass that is corresponding to the adsorption temperature Ta and evaporation pressure P_{eva}. m_{min} is the adsorbed mass that is corresponding to the regenerating temperature T_g and condensation pressure P_{con} [28].

G_{tot} is the total solar energy absorbed by the collector during the whole day. With respect to the reference [28] it can be calculated by:

$$G_{tot}=\int_{sunrise}^{sunset} G(t)dt \quad (4.33)$$

In the final stage with respect to the Reference [37], the efficiency for FPC types can be written as;

$$F_r\varepsilon\tau-c_1(T_i-T_{amb})/G_{tot}-c_2(T_i-T_{amb})^2/G_{tot}=c_0-c_1\Delta T/G_{tot}-c_2Gt(\Delta T/G_{tot})^2=\eta \quad (4.34)$$

And In the final stage with respect to the Reference [37], the efficiency for concentrating collectors types can be written as;

$$F_r\eta_0-c_1(T_i-T_{amb})/(C_oG_b)-c_2(T_i-T_{amb})^2/(C_oG_b)=k_0-k_1\Delta T/G_b-k_2G_b(\Delta T/G_b)^2=\eta \quad (4.35)$$

The values of the constants in equation (4.34) and (4.35) can be taken from Tables 3.20, 3.21, 3.22 and 3.23. Usually the second-order terms are neglected in which case c_2=0 and k_2=0 for simplifying the calculations. [37]

4.7.1 The PTC Modeling

“The thermal efficiency of a parabolic through collector is defined as the ratio of the useful energy produced in any time period to the beam radiation incident on the collector aperture in the same period.” [38] It can be calculated from the ‘Hottel-Whillier-Bliss’ equation under steady state condition [38]

4.7.2 Hot water storage tank modeling

The storage tank temperature is evaluated with the aid of the energy balance equation in the tank as seen below; [38]

$$(M_{st}C_{p,f}+M_{met}C_{p,met})\frac{dT_{st}}{dt}=\dot{m}_f C_{p,f}(T_0-T_i)+UA_{st}(T_{amb}-T_{st}) \quad (4.36)$$

“Where $C_{p,f}$, M_{st} and T_{st} are, respectively, the specific heat, mass and temperature of water in the storage tank. C_{pmet} is the specific heat of the storage tank metal, and M_{met} is its mass, while (UA_{st}) is the loss coefficient-area product of the storage tank. In the right side of the equation (4.38), the first term designates the thermal input from the parabolic trough collector [38]. We can also determine the heat quantity Q_{st} that could be stored in the tank without any heat removal during the day from the start point (sunrise) to the final point, when the tank water reaches its maximum temperature.” [38] It is given by the following equation (4.37). [38] Where $T_{st,i}$ and $T_{st,max}$ are the initial temperature and maximum temperature that are respectively reached by hot water in the tank [38].

$$Q_{st}=M_{st}C_{p,f}\;\;T_{st,max}-T_{st,i} \quad (4.37)$$

CHAPTER 5

CALCULATIONS OF THE STUDY

For the study, a Microsoft Excel Program has been established for the calculations. The Zeolite 13X-Water pair has been selected as the adsorbent-adsorbate pair in a adsorption solar air conditioning system. With respect to the input and output temperature values of the cycle, a suitable solar collector has been selected for the whole cooling system. The required solar collector area has been calculated with respect to the required heat amount for total adsorption in one hour and effciency of solar collector. A parabolic trough type solar collector (PTC) has been selected for simplifying the collector selection. In the calculations the heating values and mass values are going to be implemented as rates (heat flow rate and mass flow rate) for calculating one hour working values. In the study the COP of heating isn't going to be calculated because the study is focused on cooling only and the heating of the system isn't required for the main aim.

In the calculations the input parameters are; T_{eva} as the evaporation temperature, T_{con} as the condensation temperature, $x_1=x_{min}$ as the minimum concentration ratio of the working pair, $x_2=x_{max}$ as the maximum concentration ratio of the working pair and $\dot{Q}_{eva}$ as the evaporation heat amount for one hour. The input parameters are going to be reassigned one by one and the critics are going to be issued with respect to the output values. The output values are; $\dot{m}_z$: as the required mass amount for the Zeolite 13X for one hour operating, $\dot{m}_w$: as the mass flow rate of the water for one hour operating, Q_{ab}: as the required heat amount in "Isosteric Heating" period,Q_{bc}: as the required heat amount in "Isobaric Desorption" period, Q_{cd}: as the required heat amount in "Isosteric Cooling" period, Q_{da}: as the required heat amount in "Isobaric Adsorption" period, COP_{ref}: as the performance value of the cooling process, $\dot{Q}_{con}$: as the rejected heat amount because of the condensation, T_a, T_b, T_c, T_d are the temperature values in points a, b, c and d in the Clapeyron Diagram of the cycle, $Ln(P_{eva})$ and $Ln(P_{con})$ are the pressure values in the evaporation and in the condensation points. The water mass sorpted in one second by Zeolite 13X is also be calculated. The used formulas for calculating the inlet and outlet heats are given below.

For inlet heat rates while "Isosteric Heating" and "Isobaric Desorption" Periods;

$$Q_{ab}=[m\,(C_{p,z}+xC_{p,w})+m_{bed}C_{p,bed}].(T_b-T_a)$$

$$Q_{bc}=[m\,(C_{p,z}+xC_{p,w})+m_{bed}C_{p,bed}](T_c-T_b)+m\Delta H_a(x_c-x_b)$$

For outlet heat rates while "Isosteric Cooling" and "Isobaric Adsorption" Periods;

$$Q_{cd}=[m\,(C_{p,z}+xC_{p,w})+m_{bed}C_{p,bed}](T_d-T_c)$$

$$Q_{da}=[m\,(C_{p,z}+xC_{p,w})+m_{bed}C_{p,bed}](T_a-T_d)+m\Delta H_a(x_a-x_d)$$

In the above formulas which are used for calculating the heat transfer rates, the "$(m_{bed}C_{p,bed})$" terms are going to be neglected for simplifying the calculations. This term can be replaced in to the formulas after all off the calculations for much specific values. That's because, first off all the heat transfer rates of the bed and mass balance rates of the working pair must be calculated, after with respect to these values the adsorption bed's dimensions, type and the other specifications are going to be decided. After all, with respect to the adsorption bed's specifications the calculations may be recalculated for improving the calculations. After neglecting the "$(m_{bed}C_{p,bed})$" terms in the above formulas, for calculating the one hour of working demands the above formulas are going to be used as follows;

For inlet heat rates while "Isosteric Heating" and "Isobaric Desorption" Periods;

$$\dot{Q}_{ab}=\dot{m}\,(C_{p,z}+xC_{p,w})(T_b-T_a)$$

$$\dot{Q}_{bc}=\dot{m}\,(C_{p,z}+xC_{p,w})(T_c-T_b)+\dot{m}\Delta H_a(x_c-x_b)$$

For outlet heat rates while “Isosteric Cooling” and “Isobaric Adsorption” Periods;

$\dot{Q}_{cd}=\dot{m}(C_{p,z}+xC_{p,w})(T_d-T_c)$

$\dot{Q}_{da}=\dot{m}(C_{p,z}+xC_{p,w})(T_a-T_d)+\dot{m}\Delta H_a(x_a-x_d)$

For the above formulas the coefficients are given below for the calculations.

ΔH_a=3400 kJ/kg taken from Table 3.4
Cp_z=1,06 kJ/kgK taken from Table 3.1
Cp_w=4,2 kJ/kgK taken from Table 3.1
ΔH_v=2258 kJ/kg taken from Table 3.1
hfg_w= 2361 kJ/kg taken from Table 4.1

For calculating the evaporation and condensation heat rates are:

$\dot{Q}_{eva}=\dot{m}\Delta x\Delta H_v+\dot{m}\Delta xC_{p,w}(T_{eva}-T_{con})=\dot{m}_{adsorbent}(x_1-x_2)h_{fg,eva}$

$\dot{Q}_{con}=\dot{m}\Delta x\Delta H_v=\dot{m}_{adsorbent}(x_1-x_2)h_{fg,con}$

In fact in the above formulas which are given for calculating the “$\dot{Q}_{eva}$” and “$\dot{Q}_{con}$”, the “$\dot{m}\Delta x\Delta H_v$” part of the equations are used for the both two equations. Because the phase change of the refrigerant is valid in the both two equations. The “$\dot{m}\Delta xC_{p,w}(T_{eva}-T_{con})$” part of the above formula is only valid if evaporation period continues after the phase change. The “$\dot{m}\Delta xC_{p,w}(T_{eva}-T_{con})$” part of the $\dot{Q}_{eva}$’s equation can also be valid for the $\dot{Q}_{con}$’s equation if the condensation period continues in theory. But in the cycle “$\dot{m}\Delta xC_{p,w}(T_{eva}-T_{con})$” part of the formulas is only valid in “$\dot{Q}_{eva}$”. That is because; the refrigerated space’s initial conditions can be taken different with respect to each other. That means the initial temperature of the refrigerated space can be so different from the evaporator inlet temperature of the refrigerant liquid. But in the condensation period only phase change of the refrigerant is valid and the temperature difference doesn’t occur. To sum up, in condensation the saturated vapor enters the condenser and saturated liquid exits the condenser. But in the evaporation, saturated adsorbate can be preheated in the evaporator by using the refrigeration desired space's heat until the adsorbate reaches the saturated liquid phase. In the above formulas the "h_{fg}" can be read from appendix 1. $\dot{Q}_{eva}$ is the main input variable and all off the calculations begin from here. The amount of water sorpted is going to come from the equation of "x" (concentration ratio, $m_w=m_z.x_{max}$)

The performance coefficient's formula is;

$COP_{ref}=\dot{Q}_{eva}/(\dot{Q}_{ab}+\dot{Q}_{bc})$

The used formulas to plot the isosteric diagram for molecular sieves (Zeolite 13X)-water pair are:

$Ln(P)=a(x)+[b(x)/T_b]$

$a(x)=a_0+a_1x+a_2x^2+a_3x^3$

$b(x)=b_0+b_1x+b_2x^2+b_3x^3$

The coefficients for the above equations are acquired from Table 4.2 for molecular sieves-water pair are given as;

$a_0=13.4244$

$a_1=110.854$

$a_2=-731.76$

$a_3=1644.8$

$b_0=-7373.78$

$b_1=6722.92$

$b_2=5624.47$

$b_3=-3486.7$

The thermodynamic property table values (Appendix 1; saturated water-Temperature table [2]) is considered while plotting the saturation line of water in isosteric diagram (Clasius-Clapeyron Chart).

The collector efficiency and the required collector area are going to be calculated as assuming it is going to be used a sun tracking parabolic trough collector with respect to the below formula.

$$\eta = F_r\eta_0 - c_1(T_i - T_{amb})/(CG_b) - c_2(T_i - T_{amb})^2/(CG_b) = 0.762 - 0.2125\Delta T/G_b - 0.001672G_b(\Delta T/G_b)^2$$

The constants for the collector's efficiency formula, are taken from Figure 3.23 with neglecting the second order terms "$((CG_b) - c_2(T_i - T_{amb})^2/(CG_b))$" to simplify the calculations. In the above formula $T_i = 20$ °C is assumed as the initial temperature of the water in the collector inlet. But after the first cycle of the water in the collector, the inlet temperature of the water in the collector entrance is going to be increased if it is closed loop collector cycle. That also means the collector efficiency is going to be increased. But it is a open loop cycle, the inlet temperature always stays in the approximate value of the water supplier's conditions like it was assumed. It may be used a preheating technology before the collector's entrance. But unless they are not reinforced with other renewable energy sources, it will be nonsense to use preheating technology while the main aim of the study is to reduce the main pollutants of the atmosphere.

With help of the Microsoft Excel program which is established for the calculations, the following calculations had been done. In the calculations and diagrams, the temperatures are given in Kelvin, the heats are in (kJ/h), the pressures are in mbar and the mass rates are in (kg/h). To remind, $\dot{m}_z$ indicates the Zeolite amount for one hour operating. After one hour adsorbent won't be able to adsorp anymore water.

5.1 Case 1

Inputs

T_{eva}	20	°C
T_{con}	60	°C
$x_1=x_{min}$	0,05	
$x_2=x_{max}$	0,25	
$\dot{Q}_{eva}$	12000 (12660.67)	btu/h(kJ/h)

Table 5.1 Input values for case 1

Outputs

$\dot{m}_z$	30,28868	kg/h
$\dot{m}_w$	7,572171	kg/h
$\dot{Q}_{ab}$	2444,927	kJ/h
$\dot{Q}_{bc}$	36818,46	kJ/h
COP_{ref}	0,322455	
$\dot{Q}_{cd}$	3074,511	kJ/h
$\dot{Q}_{da}$	32447,01	kJ/h
$\dot{Q}_{con}$	20596,31	kJ/h
T_a	300,8465	K
T_b	341,0061	K
T_c	582,2605	K
T_d	495,5024	K
$Ln(P_{eva})$	3,167776	mbar
$Ln(P_{con})$	5,279959	mbar
Required Heat for total desorption.	39263,38862	kJ/h
Effciency of PTC	0,659553585	
Required Heat for total desorption with respect to the solar collector effciency	59530,24818	kJ/h
Required Solar Collector area	21,12499935	m^2
Water mass sorpted per second	2,103380848	g/s

Table 5.2 Output values for case 1

Classius-Clapeyron Diagram of Case 1

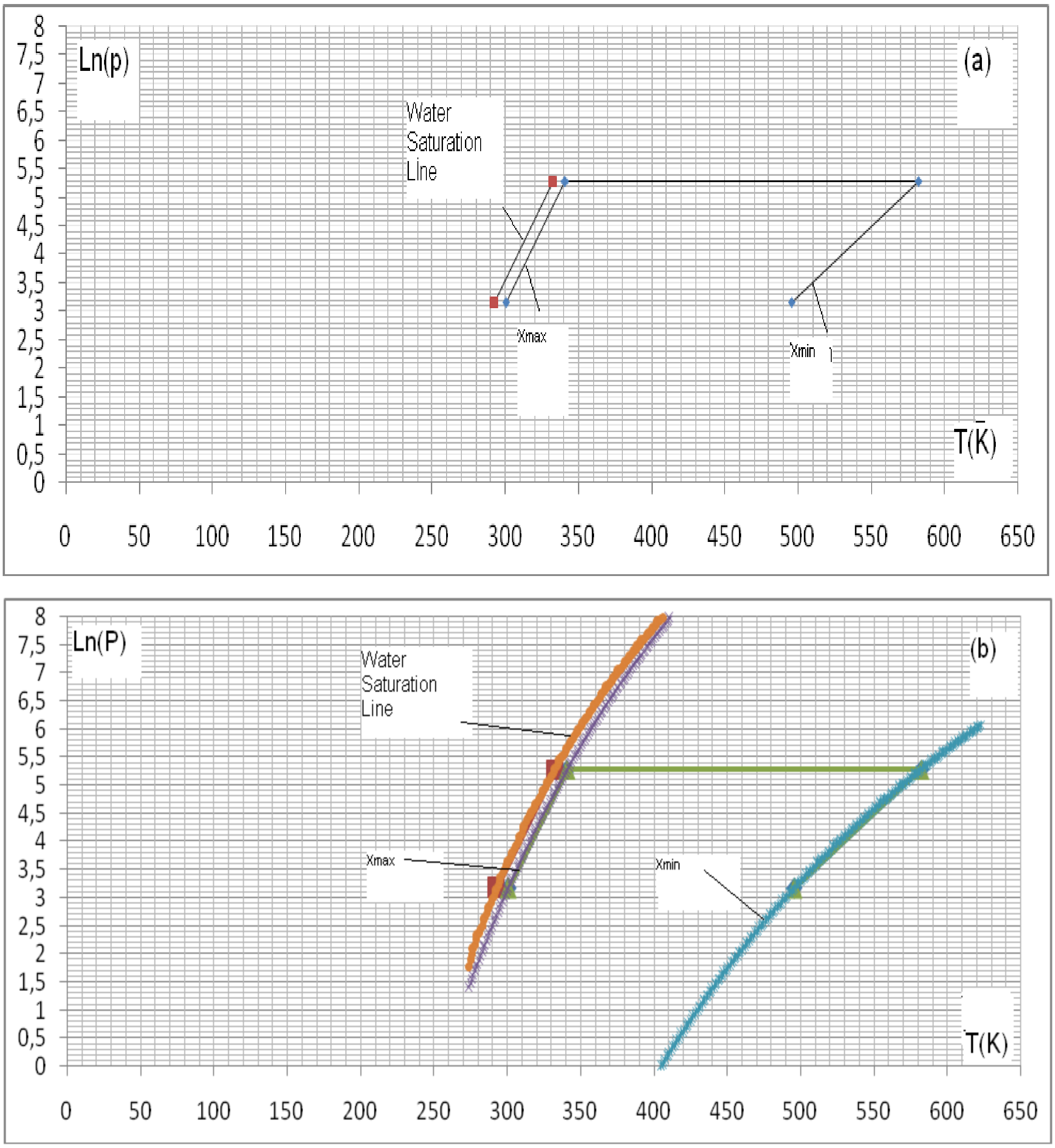


Figure 5.1 Ln(P)-T Diagram for case 1 a)For the important points can be clearly seen b) For representing the important points on the isoster lines, water saturation line

5.2 Case 2 (The Change of the T_{con})

Inputs

T_{eva}	20	°C
T_{con}	40	°C
$x_1=x_{min}$	0,05	
$x_2=x_{max}$	0,25	
$\dot{Q}_{eva}$	12000(12660.67)	btu/h(kJ/h)

Table 5.3 Input values for case 2

Outputs

$\dot{m}_z$	29,11838	kg/h
$\dot{m}_w$	7,279594	kg/h
$\dot{Q}_{ab}$	1207,968	kJ/h
$\dot{Q}_{bc}$	33881,16	kJ/h
COP_{ref}	0,360815	
$\dot{Q}_{cd}$	1492,403	kJ/h
$\dot{Q}_{da}$	31193,31	kJ/h
$\dot{Q}_{con}$	19800,5	kJ/h
T_a	300,8465	K
T_b	321,4857	K
T_c	539,3084	K
T_d	495,5024	K
$Ln(P_{eva})$	3,167776	mbar
$Ln(P_{con})$	4,319199	mbar
Required Heat for total desorption.	35089,13207	kJ/h
Effciency of PTC	0,674765791	
Required Heat for total desorption with respect to the solar collector effciency	52001,94277	kJ/h
Required Solar Collector area	18,45349282	m^2
Water mass sorpted per second	2,022109463	g/s

Table 5.4 Output values for case 2

Classius-Clapeyron Diagram of Case 2

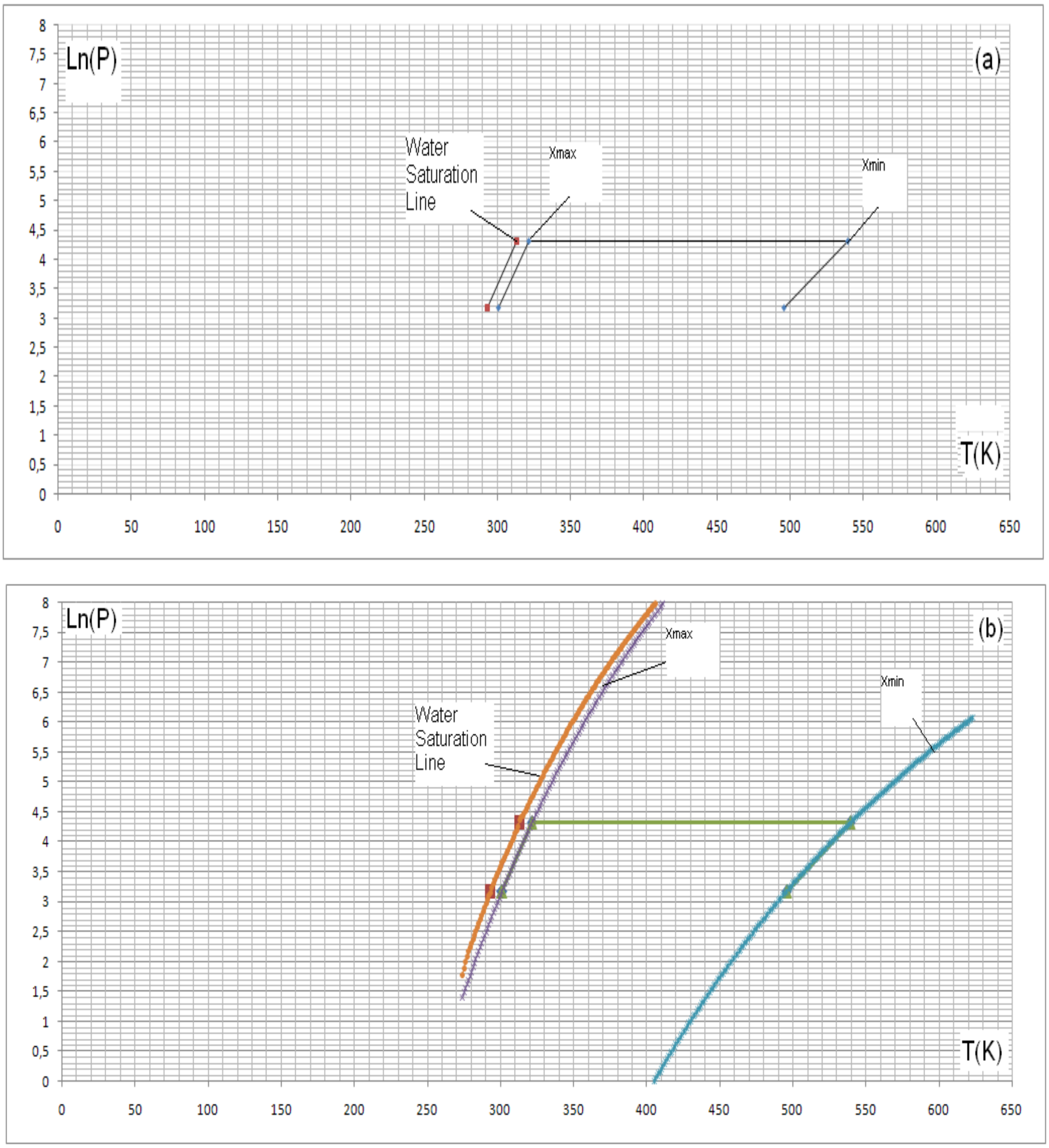


Figure 5.2 Ln(P)-T Diagram for Case 2 a) For the important points can be clearly seen b) For representing the important points on the isoster lines, water saturation line

5.3 Case 3 (The Change of T_{eva})

Inputs

T_{eva}	10	°C
T_{con}	60	°C
$x_1=x_{min}$	0,05	
$x_2=x_{max}$	0,25	
$\dot{Q}_{eva}$	12000(12660.67)	btu/h(kJ/h)

Table 5.5 Input values for case 3

Outputs

$\dot{m}_z$	30,90984	kg/h
$\dot{m}_w$	7,72746	kg/h
$\dot{Q}_{ab}$	3133,338	kJ/h
$\dot{Q}_{bc}$	37573,53	kJ/h
COP_{ref}	0,311021	
$\dot{Q}_{cd}$	3904,828	kJ/h
$\dot{Q}_{da}$	32432,57	kJ/h
$\dot{Q}_{con}$	21018,69	kJ/h
T_a	290,5731	K
T_b	341,0061	K
T_c	582,2605	K
T_d	474,2864	K
$Ln(P_{eva})$	2,53367	mbar
$Ln(P_{con})$	5,279959	mbar
Required Heat for total desorption.	40706,86633	kJ/h
Effciency of PTC	0,659553585	
Required Heat for total desorption with respect to the solar collector effciency	61718,81594	kJ/h
Required Solar Collector area	21,90163802	m^2
Water mass sorpted per second	2,146516588	g/s

Table 5.6 Output values for case 3

Classius-Clapeyron Diagram of Case 3

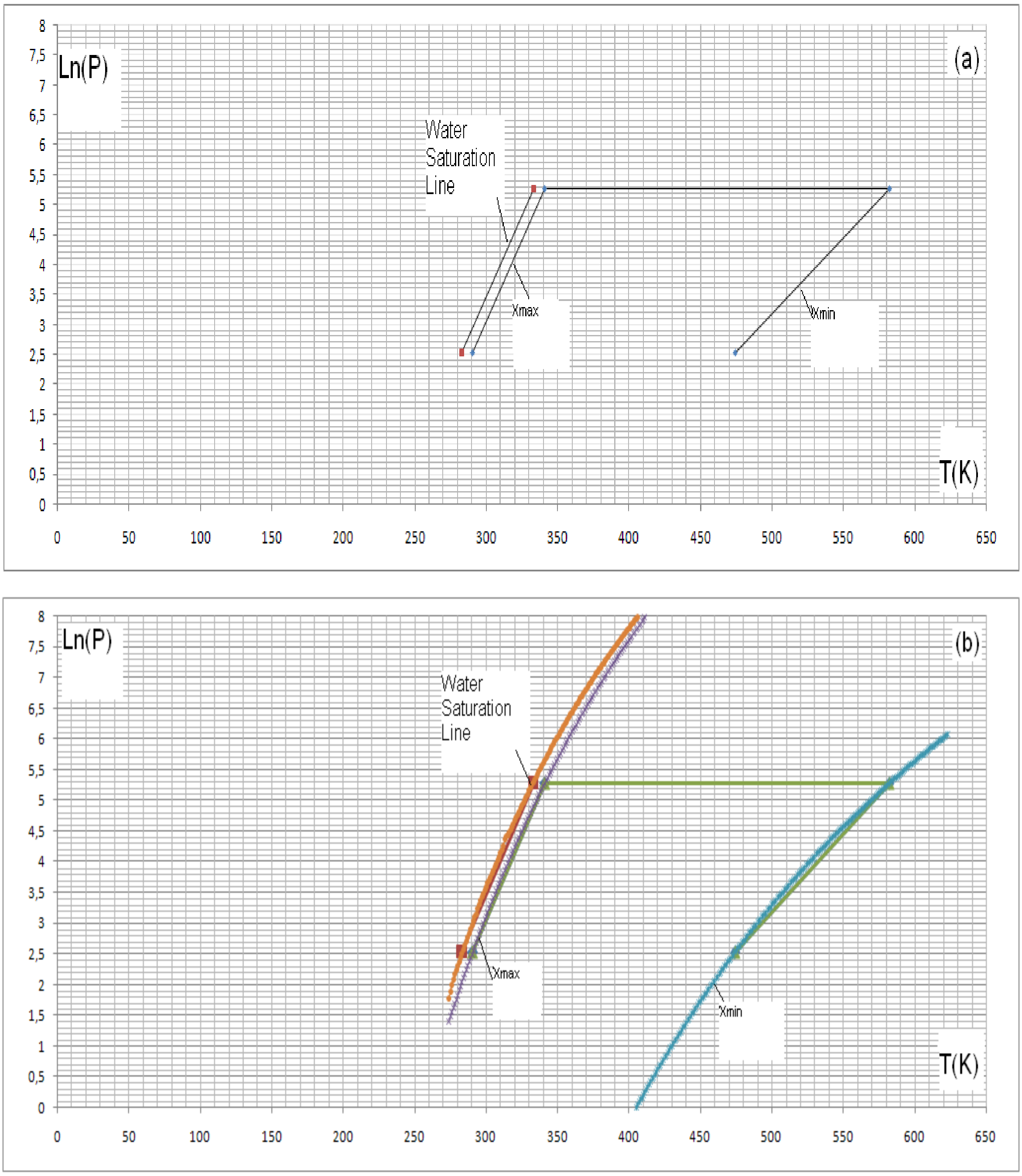


Figure 5.3 Ln(P)-T Diagram for case 3 a) For the important points can be clearly seen b) For representing the important points on the isoster lines, water saturation line

5.4 Case 4 (The Change of x_{max})

Inputs

T_{eva}	20	°C
T_{con}	60	°C
$x_1=x_{min}$	0,05	
$x_2=x_{max}$	0,2	
$\dot{Q}_{eva}$	12000(12660.67)	btu/h(kJ/h)

Table 5.7 Input values for case 4

Outputs

$\dot{m}_z$	40,38491	kg/h
$\dot{m}_w$	8,076982	kg/h
$\dot{Q}_{ab}$	3864,137	kJ/h
$\dot{Q}_{bc}$	34529,95	kJ/h
COP_{ref}	0,329756	
$\dot{Q}_{cd}$	4099,348	kJ/h
$\dot{Q}_{da}$	30631,65	kJ/h
$\dot{Q}_{con}$	20596,31	kJ/h
T_a	357,4511	K
T_b	410,6082	K
T_c	582,2605	K
T_d	495,5024	K
$Ln(P_{eva})$	3,167776	mbar
$Ln(P_{con})$	5,279959	mbar
Required Heat for total desorption.	38394,08904	kJ/h
Effciency of PTC	0,659553585	
Required Heat for total desorption with respect to the solar collector effciency	58212,23613	kJ/h
Required Solar Collector area	20,65728748	m^2
Water mass sorpted per second	2,243606238	g/s

Table 5.8 Output values for case 4

Classius-Clapeyron Diagram of Case 4

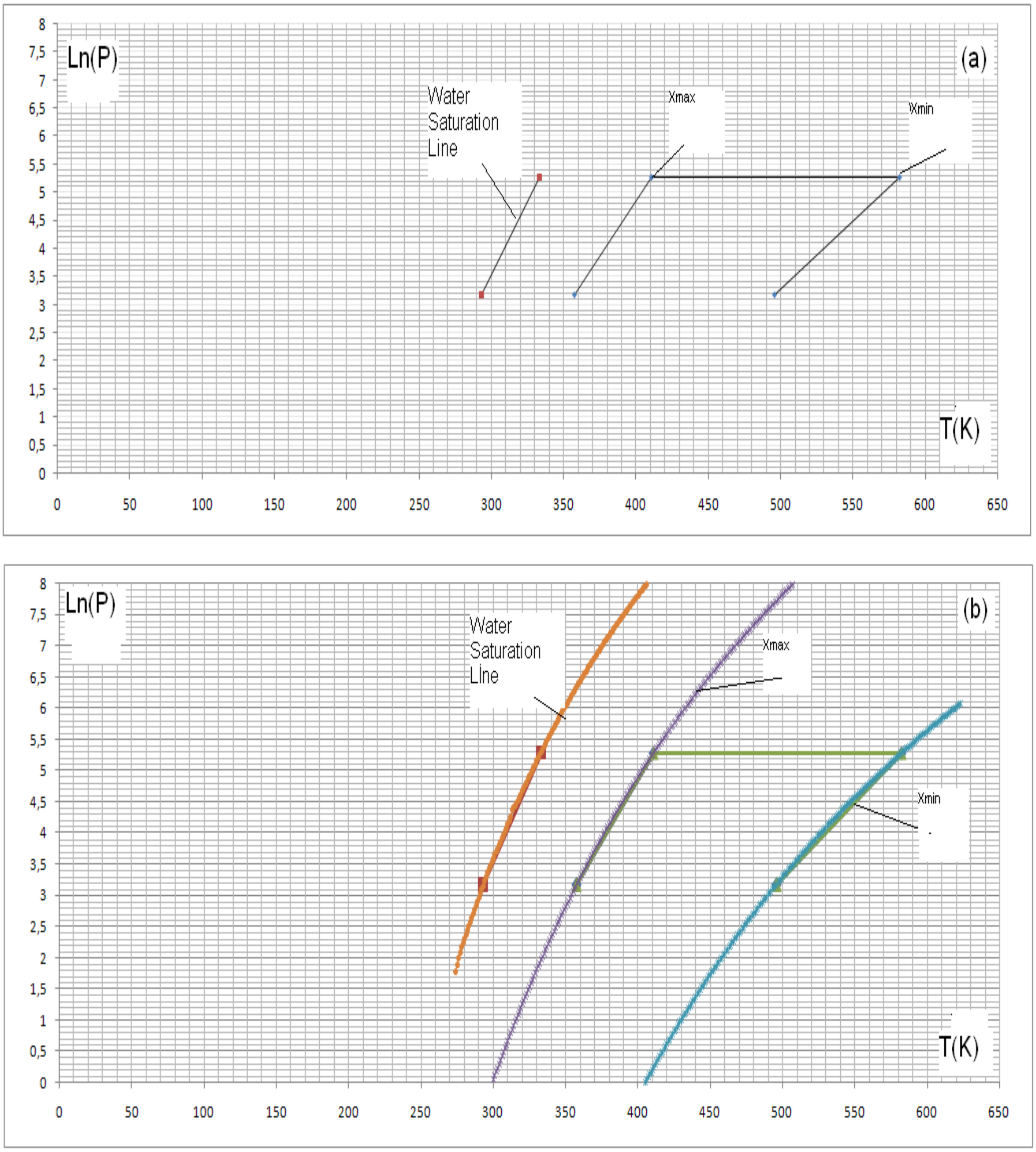


Figure 5.4 Ln(P)-T Diagram for case 4 a) For the important points can be clearly seen b) For representing the important points on the isoster lines, water saturation line

5.5 Case 5 (The Change of x_{min})

Inputs

T_{eva}	20	°C
T_{con}	60	°C
$x_1=x_{min}$	0,1	
$x_2=x_{max}$	0,25	
$\dot{Q}_{eva}$	12000(12660.67)	btu/h(kJ/h)

Table 5.9 Input values for case 5

Outputs

$\dot{m}_z$	40,38491	kg/h
$\dot{m}_w$	10,09623	kg/h
$\dot{Q}_{ab}$	3259,902	kJ/h
$\dot{Q}_{bc}$	35259,58	kJ/h
COP_{ref}	0,328682	
$\dot{Q}_{cd}$	3684,3	kJ/h
$\dot{Q}_{da}$	30618,83	kJ/h
$\dot{Q}_{con}$	20596,31	kJ/h
T_a	300,8465	K
T_b	341,0061	K
T_c	490,425	K
T_d	424,3166	K
$Ln(P_{eva})$	3,167776	mbar
$Ln(P_{con})$	5,279959	mbar
Required Heat for total desorption.	38519,48521	kJ/h
Effciency of PTC	0,692078639	
Required Heat for total desorption with respect to the solar collector effciency	55657,67105	kJ/h
Required Solar Collector area	19,75077042	m^2
Water mass sorpted per second	2,804507797	g/s

Table 5.10 Output values for case 5

Classius-Clapeyron Diagram of Case 5

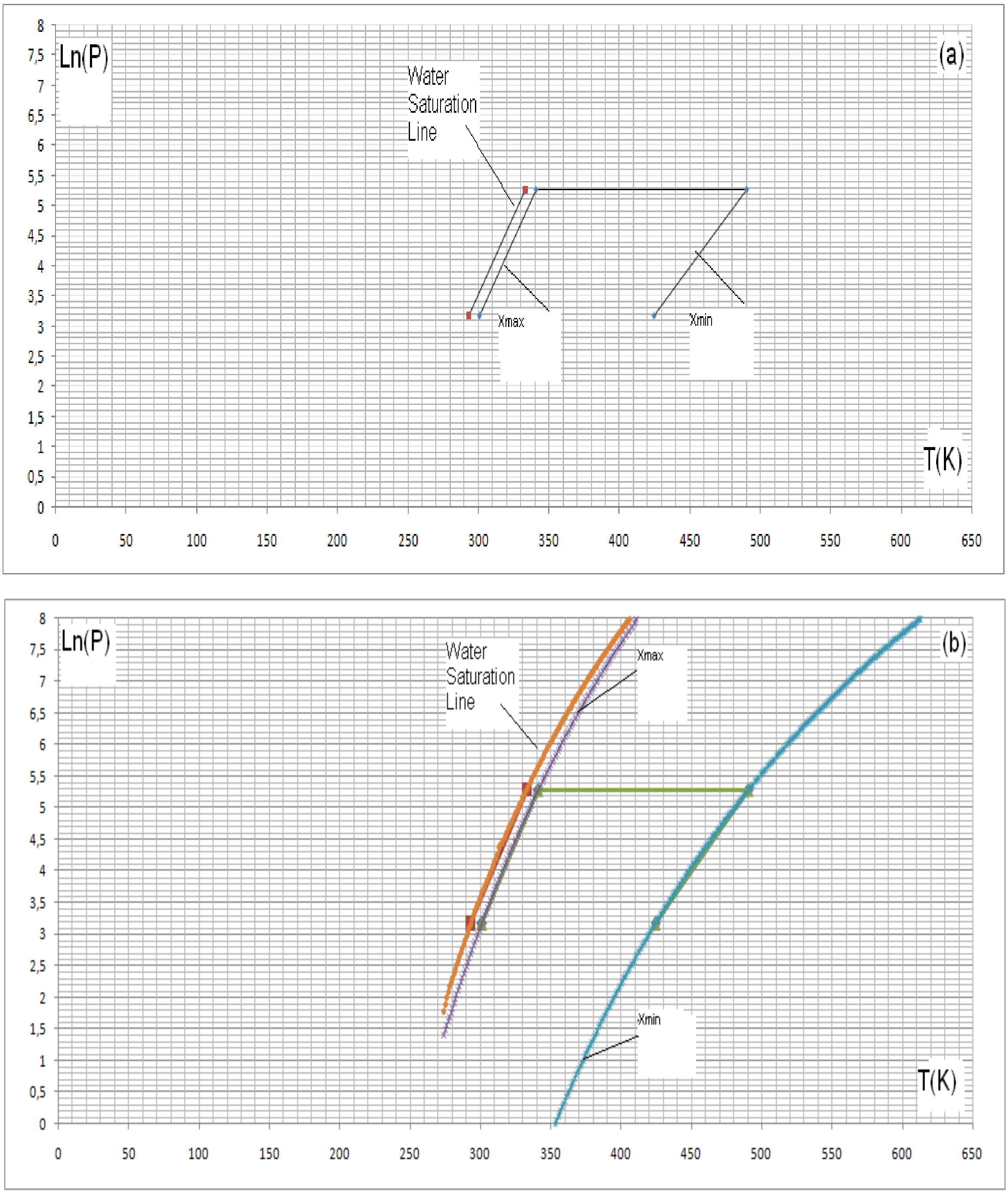


Figure 5.5 Ln(P)-T Diagram for case 5 a) For the important points can be clearly seen b) For representing the important points on the isoster lines, water saturation line

5.6 Case 6 (The Change of $\dot{Q}_{eva}$)

Inputs

T_{eva}	20	°C
T_{con}	60	°C
$x_1=x_{min}$	0,05	
$x_2=x_{max}$	0,25	
$\dot{Q}_{eva}$	24000(25321.34)	btu/h(kJ/h)

Table 5.11 Input values for case 6

Outputs

$\dot{m}_z$	60,57737	kg/h
$\dot{m}_w$	15,14434	kg/h
$\dot{Q}_{ab}$	4889,853	kJ/h
$\dot{Q}_{bc}$	73636,92	kJ/h
COP_{ref}	0,322455	
$\dot{Q}_{cd}$	6149,022	kJ/h
$\dot{Q}_{da}$	64894,02	kJ/h
$\dot{Q}_{con}$	41192,61	kJ/h
T_a	300,8465	K
T_b	341,0061	K
T_c	582,2605	K
T_d	495,5024	K
$Ln(P_{eva})$	3,167776	mbar
$Ln(P_{con})$	5,279959	mbar
Required Heat for total desorption.	78526,77723	kJ/h
Effciency of PTC	0,659553585	
Required Heat for total desorption with respect to the solar collector effciency	119060,4964	kJ/h
Required Solar Collector area	42,24999871	m^2
Water mass sorpted per second	4,206761696	g/s

Table 5.12 Output values for case 6

Classius-Clapeyron Diagram of Case 6

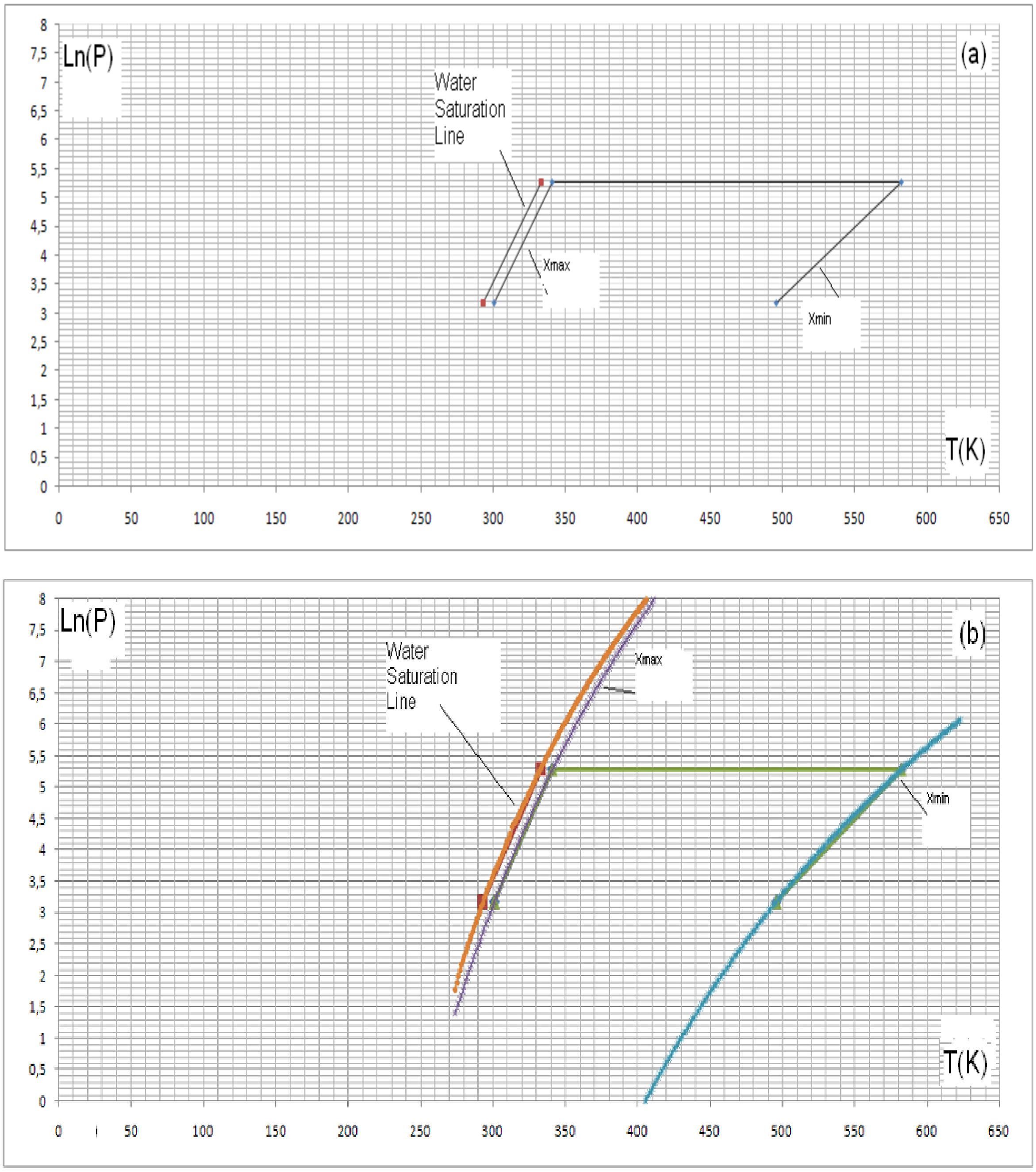


Figure 5.6 Ln(P)-T Diagram for case 6 a) For the important points can be clearly seen b) For representing the important points on the isoster lines, water saturation line

CHAPTER 6

RESULTS AND DISCUSSIONS

The calculations of the study has been given in Chapter 5. In the first case, some optimal values had been used as initial parameters. Then in the following cases some other calculations were done but only one parameter has been entered numerically different and physically different with respect to the case 1. The outputs are changed with respect to the case 1. These different output values are going to be criticized in this chapter.

In Table 5.2 the T_c value occurs as 582,2605 K. T_c is the solar source driven temperature of the system. With respect to the property tables of the solar collectors that are given in chapter 4, we need a CPC or a LFR or a PTC or a CTC for an optimal driven solar source temperature for the system as seen in the Table 5.2 and 5.1. The required amount of the solar collector changes with respect to the required heat amount, solar collector temperature and the selected collectors’ efficiency. A PTC collector is used in the calculations for an optimal approach.

In case 2 the condensation temperature was respectively entered different. With decreasing the condensation temperature which also reduces the condensation pressure, the COP of the refrigeration has been increased. That also means COP of refrigeration decreases while the condensation temperature increases. That means when the outer temperature of the surroundings decreased the system will perform better. These two values show counter relation between each other. The working pairs’ usage amounts are also decreased with respect to the case 1 when the condensation temperature is decreased. It is also clearly be seen the amount of heat required for the source of the desorption processes and isosteric heating processes are getting decreased. However the

T_a and T_d doesn't change respectively, the T_b and T_c (solar source driven temperature) decreased and as mentioned before the required heat amount is decreased. So it means the solar collector harvesting area can be reduced respectively. It is also can be seen in Table 5.4 respectively in Table 5.2. That finally means when surrounding temperature reduced, a smaller size system is going to work sufficiently respectively. Figure 6.1 shows the change of the COP with respect to the condensation temperature while the other parameters are same with the given value in case 1 (Table 5.1) Changing the condensation temperature means shifting the points b and c in Clapeyron Diagram of the system. However the locations of the x_{max} and x_{min} don't replaced on the diagram the point a and b will shift down on the x_{max} and xmin line while the condensation temperature is shifting down on the saturation line. This can be seen in Figure 5.2 with respect to Figure 5.1.

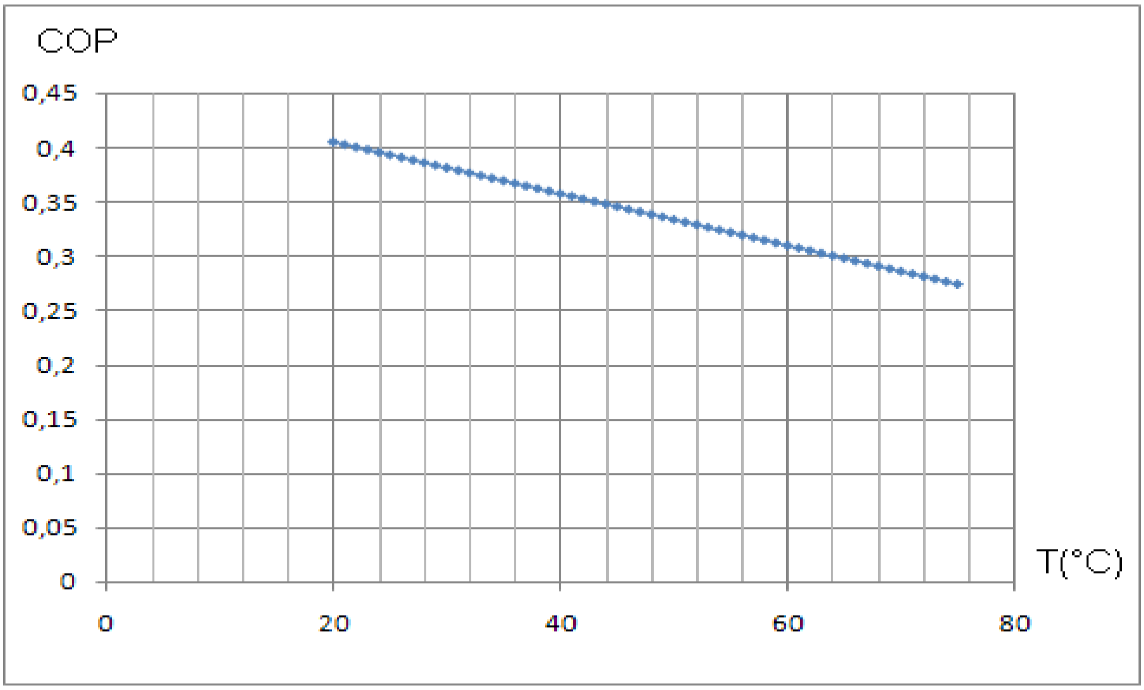


Figure 6.1 The change of COP with respect to the $T_{con.}$

In the case 3 the evaporation temperature parameter has been change with respect to the case 1. Changing the evaporation temperature means, changing the cooled zone's temperature. We can clearly see the COP of refrigeration decreased in Table 5.6 with respect to the Table 5.2. The required total heat is increased while evaporation temperature decreased. That means it is needed a bigger size system for a sufficient cooling while evaporation temperature is dropping. The heat requirement are increasing and also the required working pair amount are increasing. But the T_c (solar source driven temperature) doesn't change. That means it isn't required to use a better solar collector type. It is just required a larger solar sorption area for the collection of the heat at the same time period. Changing the evaporation temperature means changing the location of point a and d in system's Clapeyron Diagram. Decreasing of evaporation temperature, shifts down the point a and d on respectively stabile x_{max} and x_{min} curve. Increasing of evaporation temperature, shifts up the point a and d on respectively stabile x_{max} and x_{min} curve. And also evaporation point on the saturation line moves similarly. In Figure 6.2, it can be seen the change of COP with respect to the evaporation temperature change.

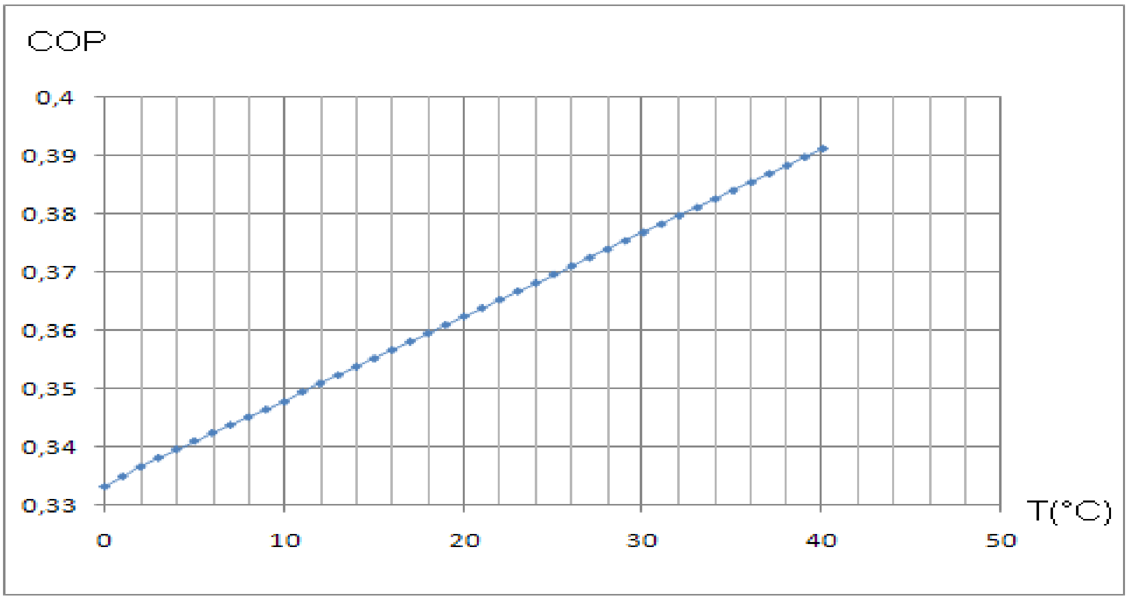


Figure 6.2 The change of COP with respect to the T_{eva}

In cases 4 and 5 the maximum and minimum concentration ratios (x_{max} and x_{min}) are changed with respect to the case 1. In case 4, the x_{max} had been decreased. It means the x_{max} curve in the Clapeyron diagram shifted to the right. In the system to obtain a increased or decreased x_{max} value, we must cut the isosteric adsorption period in the time that is required for the target x_{max} value. That means for a decreased x_{max} value, the valve between the adsorption bed and evaporator must be closed earlier and for an increased value of x_{max} the valve between the adsorption bed and evaporator must be on for a longer time period to obtain the required concentration value. The COP is increased when the x_{max} decreased as shown in Figure 6.3. That is because it going to be needed less desorption heat for the system and it is much simple to acquire maximum value of the working pairs' concentrations when they are smaller. But the cycle time is going to be decreased. But for this it is required much working pair amount. So it is required much expensive system however the T_c doesn't change and the required solar collector area decreased because of the COP's increased as seen in Table 5.8 with respect to the Table 5.2.

In the case 5 the x_{min} had been increased. It means the x_{min} curve in the Clapeyron Diagram shifted to left. For this it is required the isosteric desorption period be less than before which means the valve between the condenser and adsorption bed must be opened sooner for an increased value of x_{min}. When the opening of the valve between the condenser and adsorption bed is delayed, we acquire a smaller value of x_{min}. The COP is increased when the x_{min} increased as shown in Figure 6.4. That is because it going to be needed less desorption heat for the system. The required solar collector area is decreased with respect to the case 1. But again it is needed a bigger size of system with respect to the case 1. Because the required working pair amounts are increased and the cycle time reduced. That means in a smaller time it is needed a better system to gather the same sufficient of cooling. It is same in case 4 (the x_{max} decreased) . In these cases (cases 4 and 5), it is seen that the concentration curves are getting closer in the Clapeyron Diagram and the COP is increasing. But the cycle time is decreasing. If we don't want to decrease the cycle time we need a better system. Like much adsorbent and much adsorbate (when the water sorpted per second is same in the case 1). It is seen that the water sorpted per second is getting increased to balance the system in two chases. But the main problem occurs when the rate of the water sorption by Zeolite reaches and overcomes the water sorption capacity of the Zeolite. Than like it is mentioned before it going to be required much working pair amount to balance (for same sorption rate per second). x_{max} cannot be over 0,27 with respect to the Table 3.4, This type of molecular-sieve (Zeolite) doesn't able to adsorbe over the ratio 0.27 and that also means the system won't work after the ratio 0.27. The other parameters are same with the given value in Table 5.1 the temperatures T_a and T_b are going to change with the change of the x_{max}.

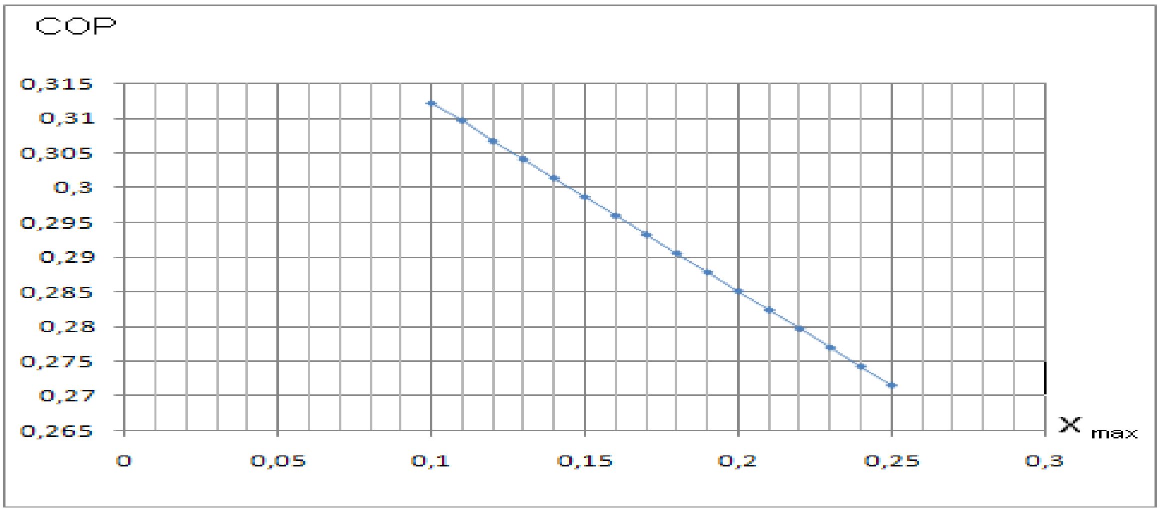


Figure 6.3 The change of COP with respect to the x_{max}

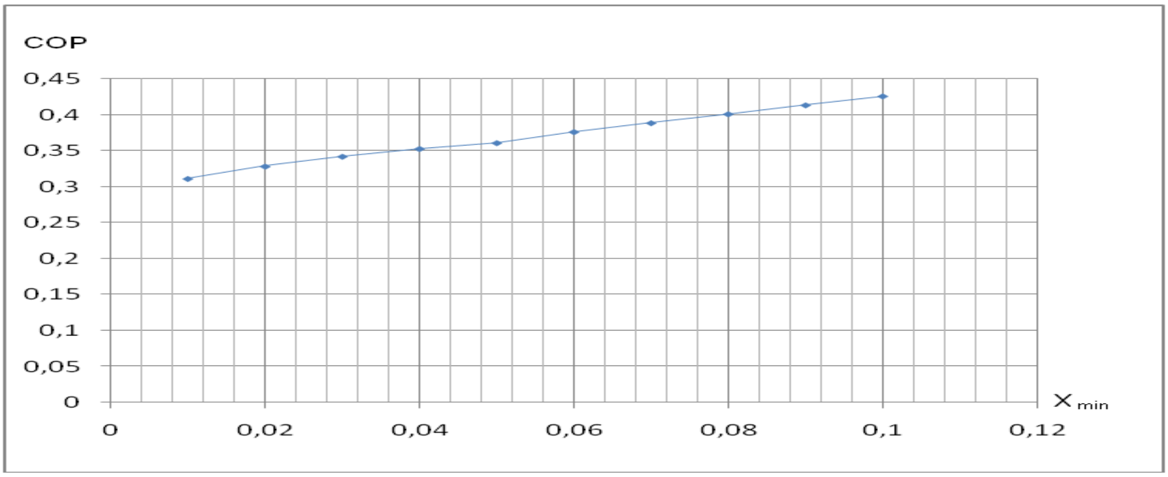


Figure 6.4 The change of COP with respect to the x_{min}

In case 6, when the other parameters remain same as given in Table 5.1 the COP is going to remain constant with respect to the change in the heat amount of evaporation. But the amounts of heat inlets and heat outlets are going to increase or decrease with perpendicular to heat amount of evaporation. It also means that the required solar energy is going to increase or decrease with perpendicular to the heat amount of evaporation. Increasing in the cooling capacity means increasing system sizes in every way. As it is clearly seen the in Table 5.12, the amount of working pair increased. The Zeolite 13X amount and the water amount used were increased to obtain the same COP with respect to the case 1. But the temperature on point a, b, c and d didn’t change. The pressure values didn’t change as seen in Table 5.12 with respect to the case 1 (Table 5.2). It is clearly be seen, only the inlet heat amounts, required collector surface area for the collection of the solar heat and mass parameters of the working pair are increased or decreased perpendicularly to the required heat of cooling. But this is in theoretical. That means if we don’t enlarge the system’s capacity with respect to the cooling capacity, we don’t harvest the same COP for cooling. In realistic, when the cooling demand increased while using the same capacity of the system; the COP decreases. For a better satisfied efficient operating system, it can be used a multi bed system like it is mentioned before and this is valid for all the cases discussed above. A multi bed system reduces the operating time and the values of concentration ratios can be kept in better accuracy parameters for a better efficiency. That means the x_{max} and x_{min} values that are seen in the Clapeyron Diagram getting closer to each other and efficiency is getting increased. Meanwhile the same amount of adsorption rate can be obtained by using multi bed system which also means much amount of working pair is needed.

It can clearly be seen that the given references’ following notifications are similar with respect to this study’s calculations and results. The COP is going to be increase when the generation temperature is increased [31] like maintained in this study. “For three temperature reservoir adsorption-cooling system, the range of variation for generation temperature is much greater than the other operating temperature.” [31] “The effects of generation temperature on SCP with different driven temperatures (the temperature of heat exchange fluid during the heating process), the SCP gets a maximum value for every different driven temperature. The reason is that the cycle's adsorbate mass is proportional to the increasing cycle time with an increase in generation temperature at low temperature." [34] However, when the generation temperature approaches the driven temperature, the heat transfer rate between the heat exchanger fluid and adsorber bed decreases [34]. So the cycle adsorbate mass will increase very little and the cycle time becomes longer, resulting in decrease of SCP [34]. When the condensation temperature increases, the condensation pressure also increases and both COP and SCP decrease linearly with increasing condensation temperature [34] like maintained in this study. The cycled adsorbate mass decreases for a lower condensation pressure with a fixed generation temperature and the cooling energy is proportional to the cycled adsorbate mass [34]. The cycle time also decreases with respect to the increase in the condensation temperature [34] like maintained in this study. When the evaporation temperature increases the both COP and SCP also increase the heat input also increase with the increasing of the evaporation temperature [34]. With the increase of the adsorption temperature, the COP is going to be decreased and the cycle time also increases with reducing the adsorption temperature and the cycle adsorbate mass will increase and the heat input will need to be increased when the adsorption temperature decreased [34] like maintained in this study. "The velocity of heat exchange fluid is also an important parameter."[34] The COP value changes very little with respect to the velocity of heat exchange fluid. For fluid velocities smaller than 0.1m/s, the cycle time will increase very quickly with an increase in fluid velocity. This can also be seen in the given cases in Chapters 5 and 6. "Hence the SCP increases significantly with an increase in fluid velocity. However, for the velocities larger than 0.5m/s, the cycle time does not change with the velocity leading to very little change in SCP. To reduce operating energy cost, the optimal velocity of the heat exchange fluid should be in the range of 0.1m/s-0.5m/s "[34]

The solar collectors must be so efficient to answer the required heat and maximum reachable temperature amount for an efficient adsorption heat pump cycle. We see that the most required solar collectors types for an moderate adsorption heat pump cycle are most generally ETC,CPC,LFR,PTC and CTC with respect to the Table 3.5. The flat plates collectors don’t absorb enough source power and temperature for driving an adsorption heat pump cycle. A PDR or a HFC collector which are two-axes tracking collectors with respect to the given property tables in chapter 3, carries much unnecessary heat with respect to the other basic collectors. They are going to be so much undesirable and over unnecessary initial payment for the system. A flat collector type is going to be a pointless or semi pointless initial payment for the known system. If the required freezing power is too much for a commercial purpose, the system can be powered with an PDR or an HFC with respect to the tables given in chapter 3.

CHAPTER 7

CONCLUSIONS AND RECOMANDATIONS

The main energy consumption are made of from industry, from residences and from transportation areas. The use of energy in an economical and efficient way is getting so much importance in our days with decreasing energy sources and increasing air pollution. The solar houses are designed in our days. As a way of these conditions the renewable energy sources are getting so much importance. The solar energy is one of the most important energy sources of all. A cooling and also a heating unit which is driven by solar power in residences or in industry further more in transportation vehicles within a more applicable way is going to be so economical and environmental after all investment.

The initial investment of these types of systems may occur much expensive than classical ways of cooling’s initial investments. But in progressive with the effect of a very little dependence of electricity (can also be neglected) the owner will get profit. The performance values are also so small with respect to the classical ways of cooling. But again the nearly independence from electricity provides so much profit to the owner in further time.

In the study, an adsorption heat pump’s thermodynamic analyses have been done. In the calculations it is obviously seen that the required solar power gathering area (solar collector area) is huge and expendable. We can say that the adsorption cooling units driven by solar power are going to be much sufficient and applicable in industrial applications. Because the system which is issued in chapter five and chapter six has a high investment cost for a residential life. The chosen solar collector type is parabolic through collector in the study and a parabolic through collector will also lead us to a structural and architectural disfiguring in a city life. In industry the system can be used for keeping the desired products cold and also for keeping stuff cool in the working areas. Parabolic dish collector can also be used for a sufficient study because these types of solar collectors are able to reach high values of temperature for feeding the inlet temperature of the adsorber. Especially a sun tracking mechanism gathers so much solar energy from the irradiation amount of the sun in specified the location. An evacuated tube type of solar collector may less sufficiently use in the application but with respect to the limited temperature of the collector for the purpose is going to lead us a way of decreasing the cooling amount or increasing the solar collector area. The solar irradiation value is taken from the Cyprus region that is because the study takes place in Cyprus. The huge amount of irradiation in Cyprus is also an advantage for the study.

In further studies a experimental set up can be formed for the study. In the experimental setup the evaporator, condenser and adsorbers must carefully be selected for a sufficient experiment and sufficient output values. It is recommended that the experimental setup is designed for a small capacity of cooling power for a much disposable and economical setup. Furthermore the experiments can be expandable. The numerical studies major reason is to lead the study in experimental ways.

In the light of these types of studies, the governments must consider using the renewable energies as energy policies and make use of these types of renewable energy driven systems for the health of the environment and for the health of the energy economics. It is going to be a salvation for the governments form the energy bottleneck of the earth. The bottleneck is going to increase with the consumption of disposable energy sources.

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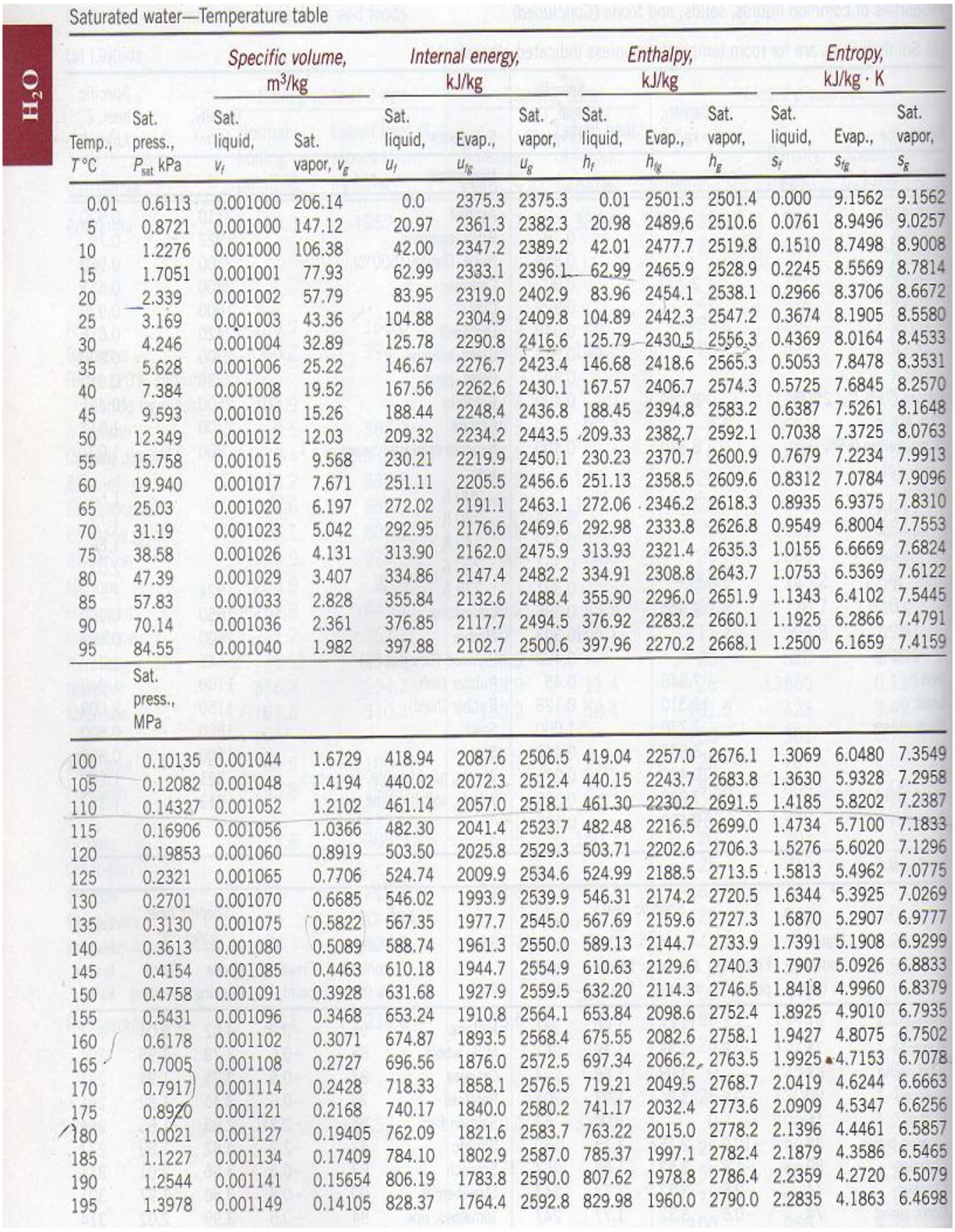
APPENDIX

Appendix 1

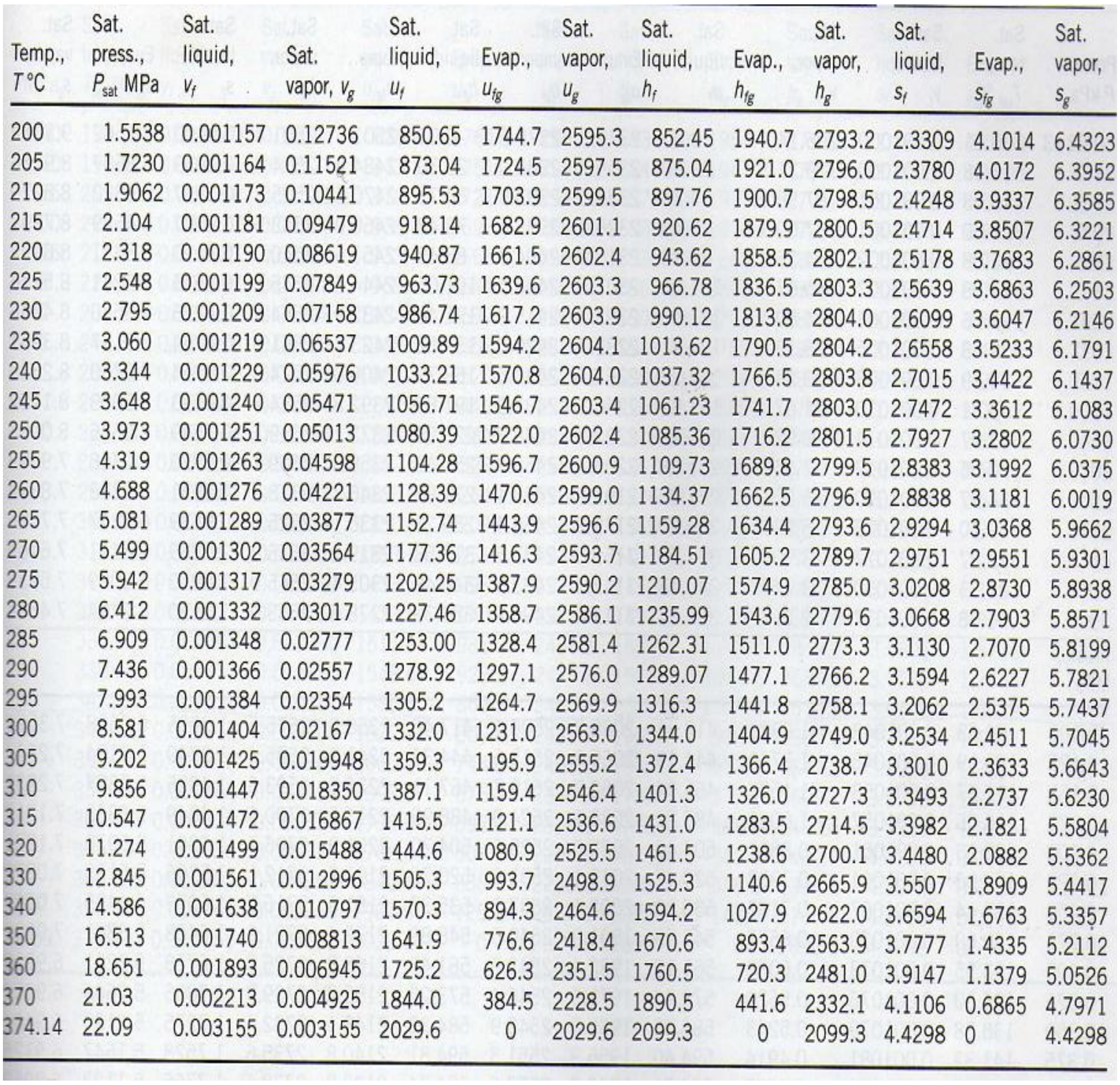


H_2O

Saturated water—Temperature table

		Specific volume, m³/kg		*Internal energy,* kJ/kg			*Enthalpy,* kJ/kg			*Entropy,* kJ/kg · K		
Temp., T °C	Sat. press., P_{sat} kPa	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
0.01	0.6113	0.001000	206.14	0.0	2375.3	2375.3	0.01	2501.3	2501.4	0.000	9.1562	9.1562
5	0.8721	0.001000	147.12	20.97	2361.3	2382.3	20.98	2489.6	2510.6	0.0761	8.9496	9.0257
10	1.2276	0.001000	106.38	42.00	2347.2	2389.2	42.01	2477.7	2519.8	0.1510	8.7498	8.9008
15	1.7051	0.001001	77.93	62.99	2333.1	2396.1	62.99	2465.9	2528.9	0.2245	8.5569	8.7814
20	2.339	0.001002	57.79	83.95	2319.0	2402.9	83.96	2454.1	2538.1	0.2966	8.3706	8.6672
25	3.169	0.001003	43.36	104.88	2304.9	2409.8	104.89	2442.3	2547.2	0.3674	8.1905	8.5580
30	4.246	0.001004	32.89	125.78	2290.8	2416.6	125.79	2430.5	2556.3	0.4369	8.0164	8.4533
35	5.628	0.001006	25.22	146.67	2276.7	2423.4	146.68	2418.6	2565.3	0.5053	7.8478	8.3531
40	7.384	0.001008	19.52	167.56	2262.6	2430.1	167.57	2406.7	2574.3	0.5725	7.6845	8.2570
45	9.593	0.001010	15.26	188.44	2248.4	2436.8	188.45	2394.8	2583.2	0.6387	7.5261	8.1648
50	12.349	0.001012	12.03	209.32	2234.2	2443.5	209.33	2382.7	2592.1	0.7038	7.3725	8.0763
55	15.758	0.001015	9.568	230.21	2219.9	2450.1	230.23	2370.7	2600.9	0.7679	7.2234	7.9913
60	19.940	0.001017	7.671	251.11	2205.5	2456.6	251.13	2358.5	2609.6	0.8312	7.0784	7.9096
65	25.03	0.001020	6.197	272.02	2191.1	2463.1	272.06	2346.2	2618.3	0.8935	6.9375	7.8310
70	31.19	0.001023	5.042	292.95	2176.6	2469.6	292.98	2333.8	2626.8	0.9549	6.8004	7.7553
75	38.58	0.001026	4.131	313.90	2162.0	2475.9	313.93	2321.4	2635.3	1.0155	6.6669	7.6824
80	47.39	0.001029	3.407	334.86	2147.4	2482.2	334.91	2308.8	2643.7	1.0753	6.5369	7.6122
85	57.83	0.001033	2.828	355.84	2132.6	2488.4	355.90	2296.0	2651.9	1.1343	6.4102	7.5445
90	70.14	0.001036	2.361	376.85	2117.7	2494.5	376.92	2283.2	2660.1	1.1925	6.2866	7.4791
95	84.55	0.001040	1.982	397.88	2102.7	2500.6	397.96	2270.2	2668.1	1.2500	6.1659	7.4159
	Sat. press., MPa											
100	0.10135	0.001044	1.6729	418.94	2087.6	2506.5	419.04	2257.0	2676.1	1.3069	6.0480	7.3549
105	0.12082	0.001048	1.4194	440.02	2072.3	2512.4	440.15	2243.7	2683.8	1.3630	5.9328	7.2958
110	0.14327	0.001052	1.2102	461.14	2057.0	2518.1	461.30	2230.2	2691.5	1.4185	5.8202	7.2387
115	0.16906	0.001056	1.0366	482.30	2041.4	2523.7	482.48	2216.5	2699.0	1.4734	5.7100	7.1833
120	0.19853	0.001060	0.8919	503.50	2025.8	2529.3	503.71	2202.6	2706.3	1.5276	5.6020	7.1296
125	0.2321	0.001065	0.7706	524.74	2009.9	2534.6	524.99	2188.5	2713.5	1.5813	5.4962	7.0775
130	0.2701	0.001070	0.6685	546.02	1993.9	2539.9	546.31	2174.2	2720.5	1.6344	5.3925	7.0269
135	0.3130	0.001075	0.5822	567.35	1977.7	2545.0	567.69	2159.6	2727.3	1.6870	5.2907	6.9777
140	0.3613	0.001080	0.5089	588.74	1961.3	2550.0	589.13	2144.7	2733.9	1.7391	5.1908	6.9299
145	0.4154	0.001085	0.4463	610.18	1944.7	2554.9	610.63	2129.6	2740.3	1.7907	5.0926	6.8833
150	0.4758	0.001091	0.3928	631.68	1927.9	2559.5	632.20	2114.3	2746.5	1.8418	4.9960	6.8379
155	0.5431	0.001096	0.3468	653.24	1910.8	2564.1	653.84	2098.6	2752.4	1.8925	4.9010	6.7935
160	0.6178	0.001102	0.3071	674.87	1893.5	2568.4	675.55	2082.6	2758.1	1.9427	4.8075	6.7502
165	0.7005	0.001108	0.2727	696.56	1876.0	2572.5	697.34	2066.2	2763.5	1.9925	4.7153	6.7078
170	0.7917	0.001114	0.2428	718.33	1858.1	2576.5	719.21	2049.5	2768.7	2.0419	4.6244	6.6663
175	0.8920	0.001121	0.2168	740.17	1840.0	2580.2	741.17	2032.4	2773.6	2.0909	4.5347	6.6256
180	1.0021	0.001127	0.19405	762.09	1821.6	2583.7	763.22	2015.0	2778.2	2.1396	4.4461	6.5857
185	1.1227	0.001134	0.17409	784.10	1802.9	2587.0	785.37	1997.1	2782.4	2.1879	4.3586	6.5465
190	1.2544	0.001141	0.15654	806.19	1783.8	2590.0	807.62	1978.8	2786.4	2.2359	4.2720	6.5079
195	1.3978	0.001149	0.14105	828.37	1764.4	2592.8	829.98	1960.0	2790.0	2.2835	4.1863	6.4698

Appendix 1 continued



Temp., T °C	Sat. press., P_{sat} MPa	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
200	1.5538	0.001157	0.12736	850.65	1744.7	2595.3	852.45	1940.7	2793.2	2.3309	4.1014	6.4323
205	1.7230	0.001164	0.11521	873.04	1724.5	2597.5	875.04	1921.0	2796.0	2.3780	4.0172	6.3952
210	1.9062	0.001173	0.10441	895.53	1703.9	2599.5	897.76	1900.7	2798.5	2.4248	3.9337	6.3585
215	2.104	0.001181	0.09479	918.14	1682.9	2601.1	920.62	1879.9	2800.5	2.4714	3.8507	6.3221
220	2.318	0.001190	0.08619	940.87	1661.5	2602.4	943.62	1858.5	2802.1	2.5178	3.7683	6.2861
225	2.548	0.001199	0.07849	963.73	1639.6	2603.3	966.78	1836.5	2803.3	2.5639	3.6863	6.2503
230	2.795	0.001209	0.07158	986.74	1617.2	2603.9	990.12	1813.8	2804.0	2.6099	3.6047	6.2146
235	3.060	0.001219	0.06537	1009.89	1594.2	2604.1	1013.62	1790.5	2804.2	2.6558	3.5233	6.1791
240	3.344	0.001229	0.05976	1033.21	1570.8	2604.0	1037.32	1766.5	2803.8	2.7015	3.4422	6.1437
245	3.648	0.001240	0.05471	1056.71	1546.7	2603.4	1061.23	1741.7	2803.0	2.7472	3.3612	6.1083
250	3.973	0.001251	0.05013	1080.39	1522.0	2602.4	1085.36	1716.2	2801.5	2.7927	3.2802	6.0730
255	4.319	0.001263	0.04598	1104.28	1596.7	2600.9	1109.73	1689.8	2799.5	2.8383	3.1992	6.0375
260	4.688	0.001276	0.04221	1128.39	1470.6	2599.0	1134.37	1662.5	2796.9	2.8838	3.1181	6.0019
265	5.081	0.001289	0.03877	1152.74	1443.9	2596.6	1159.28	1634.4	2793.6	2.9294	3.0368	5.9662
270	5.499	0.001302	0.03564	1177.36	1416.3	2593.7	1184.51	1605.2	2789.7	2.9751	2.9551	5.9301
275	5.942	0.001317	0.03279	1202.25	1387.9	2590.2	1210.07	1574.9	2785.0	3.0208	2.8730	5.8938
280	6.412	0.001332	0.03017	1227.46	1358.7	2586.1	1235.99	1543.6	2779.6	3.0668	2.7903	5.8571
285	6.909	0.001348	0.02777	1253.00	1328.4	2581.4	1262.31	1511.0	2773.3	3.1130	2.7070	5.8199
290	7.436	0.001366	0.02557	1278.92	1297.1	2576.0	1289.07	1477.1	2766.2	3.1594	2.6227	5.7821
295	7.993	0.001384	0.02354	1305.2	1264.7	2569.9	1316.3	1441.8	2758.1	3.2062	2.5375	5.7437
300	8.581	0.001404	0.02167	1332.0	1231.0	2563.0	1344.0	1404.9	2749.0	3.2534	2.4511	5.7045
305	9.202	0.001425	0.019948	1359.3	1195.9	2555.2	1372.4	1366.4	2738.7	3.3010	2.3633	5.6643
310	9.856	0.001447	0.018350	1387.1	1159.4	2546.4	1401.3	1326.0	2727.3	3.3493	2.2737	5.6230
315	10.547	0.001472	0.016867	1415.5	1121.1	2536.6	1431.0	1283.5	2714.5	3.3982	2.1821	5.5804
320	11.274	0.001499	0.015488	1444.6	1080.9	2525.5	1461.5	1238.6	2700.1	3.4480	2.0882	5.5362
330	12.845	0.001561	0.012996	1505.3	993.7	2498.9	1525.3	1140.6	2665.9	3.5507	1.8909	5.4417
340	14.586	0.001638	0.010797	1570.3	894.3	2464.6	1594.2	1027.9	2622.0	3.6594	1.6763	5.3357
350	16.513	0.001740	0.008813	1641.9	776.6	2418.4	1670.6	893.4	2563.9	3.7777	1.4335	5.2112
360	18.651	0.001893	0.006945	1725.2	626.3	2351.5	1760.5	720.3	2481.0	3.9147	1.1379	5.0526
370	21.03	0.002213	0.004925	1844.0	384.5	2228.5	1890.5	441.6	2332.1	4.1106	0.6865	4.7971
374.14	22.09	0.003155	0.003155	2029.6	0	2029.6	2099.3	0	2099.3	4.4298	0	4.4298