EFFECT OF OVERALL THERMAL CONDUCTANCE WITH DIFFERENT MASS ALLOCATION ON A TWO STAGE ADSORPTION CHILLER EMPLOYING RE-HEAT SCHEME

by

TAMANNA SULTANA
Registration No.: 040809004P
Roll No.: 040809004P
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Department of Mathematics
Bangladesh University of Engineering and Technology (BUET)
Dhaka-1000, Bangladesh.
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TAMANNA SULTANA
Registration No.: 040809004P
Session: April-2008
Department of Mathematics
Bangladesh University of Engineering and Technology (BUET)
Dhaka-1000

Under the supervision
of
DR. MD. ZAFAR IQBAL KHAN
Professor
Department of mathematics
Bangladesh University of Engineering and Technology (BUET)
Dhaka-1000
The Thesis Entitled

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Submitted by

TAMANNA SULTANA

Registration No.: 040809004P and Session: April 2008 a part time student of M.Phil. (Mathematics) has been accepted as satisfactory in partial fulfillment for the degree of

Master of Philosophy in Mathematics

8th September, 2013

BOARD OF EXAMINERS

1. Dr. Md. Zafar Iqbal Khan
   Professor
   Department of Mathematics
   BUET, Dhaka-1000
   (Supervisor)

2. Head
   Dept. of Mathematics,
   BUET, Dhaka-1000
   (Ex-Officio)

3. Dr. Md. Mustafa Kamal Chowdhury
   Professor
   Department of Mathematics
   BUET, Dhaka-1000

4. Dr. Md. Abdul Hakim Khan
   Professor
   Department of Mathematics
   BUET, Dhaka-1000

5. Dr. Md. Abdus Samad
   Professor
   Department of Mathematics
   University Of Dhaka
   Dhaka-1000.
DEDICATION

This work is dedicated
To
My dearest Parents
CANDIDATE’S DECLARATION

I am hereby declaring that the work in this dissertation is being carried out in accordance with the regulations of Bangladesh University of Engineering and Technology (BUET), Dhaka, Bangladesh. The work is also original except where indicated by and attached with special reference in the context and no part of it has been submitted for any attempt to get for any others degrees or diplomas. All views expressed in the dissertation are those of the author and in no way of by no means represent those of Bangladesh University of Engineering and Technology, Dhaka. This dissertation has not been submitted to any other University for examination either in home or abroad.

____________________
Tamanna Sultana
Date: 8\textsuperscript{th} September, 2013
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ABSTRACT

Now a days, adsorption heat pumps receive considerable attention as they are energy savers and environmentally benign. Silica gel/water based adsorption cycles have a distinct advantage in their ability to be driven by heat of near-ambient temperature so that waste heat below 100 °C can be recovered. One interesting feature of refrigeration cycles driven by waste heat is that they do not use primary energy as driving source. In the present dissertation, an analytic investigation of a two-stage adsorption refrigeration chiller using re-heat with different mass allocation is performed to determine the influence of the overall thermal conductance of sorption elements, condenser and evaporator on the chiller performance. Result shows that cycle performance is strongly influenced by the large thermal conductance values and gives optimum result for low value of cooling water temperature. It is also seen that COP shows better result for large value of time cycle.
NOMENCLATURE

A \hspace{1cm} \text{(area (m$^2$))} \\
E_a \hspace{1cm} \text{(activation energy (Jkg$^{-1}$))} \\
L \hspace{1cm} \text{(latent heat of vaporization (Jkg$^{-1}$))} \\
\textit{m} \hspace{1cm} \text{(mass flow rate (kg s$^{-1}$))} \\
q \hspace{1cm} \text{(concentration (kg refrigerant/kg adsorbent))} \\
q^* \hspace{1cm} \text{(concentration at equilibrium (kg refrigerant/kg adsorbent))} \\
Q_{st} \hspace{1cm} \text{(isosteric heat of adsorption (Jkg$^{-1}$))} \\
P \hspace{1cm} \text{(pressure (Pa))} \\
P_c \hspace{1cm} \text{(condensing pressure (Pa))} \\
P_e \hspace{1cm} \text{(evaporation pressure (Pa))} \\
P_s \hspace{1cm} \text{(saturated vapor pressure (Pa))} \\
R_{\text{gas}} \hspace{1cm} \text{(gas constant (J mol$^{-1}$ K$^{-1}$))} \\
R_p \hspace{1cm} \text{(average radius of a particle (m))} \\
T \hspace{1cm} \text{(temperature ($^0$C, K))} \\
t \hspace{1cm} \text{(time (s))} \\
U \hspace{1cm} \text{(overall heat transfer coefficient (Wm$^{-2}$K$^{-1}$))} \\
W \hspace{1cm} \text{(weight (kg))} \\
C \hspace{1cm} \text{(specific heat (JK$^{-1}$K$^{-1}$))} \\
D_{so} \hspace{1cm} \text{(pre-exponential constant (m$^2$s$^{-1}$))}

Subscripts

ads \hspace{1cm} \text{adsorber or adsorption} \\
\text{cond} \hspace{1cm} \text{condenser} \\
\text{chill} \hspace{1cm} \text{chilled water} \\
\text{cw} \hspace{1cm} \text{cooling water} \\
\text{des} \hspace{1cm} \text{desorber or desorption} \\
\text{eva} \hspace{1cm} \text{evaporator} \\
\text{Hex} \hspace{1cm} \text{heat exchanger} \\
in \hspace{1cm} \text{inlet} \\
\text{out} \hspace{1cm} \text{outlet} \\
s \hspace{1cm} \text{Silica gel} \\
w \hspace{1cm} \text{water} \\
Hw \hspace{1cm} \text{hot water} \\
\textbf{Greek symbol} \\
\delta \hspace{1cm} \text{variance}
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CHAPTER ONE

Introduction

1.1 Background Information

With the increasing trend in energy consumption and worldwide economic growth, the general trend in cooling and air conditioning requirements of industry and buildings is also increasing. To meet the demand in cooling, mechanical vapor-compression systems are commonly used which can be classified as conventional systems; Dhar et al. [1]. These systems are very popular due to their high coefficients of performance, small sizes and low weights. However, they also exhibit some disadvantages such as contributing to global warming and ozone layer depletion and high energy consumptions [2]. One of the disadvantages of conventional vapor-compression systems can be stated that these systems include refrigerants such as chlorofluorocarbon (CFC), hydro-chlorofluorocarbon (HCFC) or hydrofluorocarbons (HFC) which have high global warming potential and ozone depletion potential [3-5]. Since the global warming problem is presently more critical, researchers are studying ways to reduce the emission of these greenhouse gases and overcome this significant disadvantage of conventional cooling systems. Another main disadvantage of conventional cooling systems is their large electricity consumption which causes a need for new investments and new infrastructure, such as new power plants, transmission and distribution lines on the electricity networks; Papadopoulos [5]. However, adsorption cycles have some distinct advances over the other systems in view points of their ability to be driven by relatively low heat source temperature; Kashiwagi et al. [6].

1.2 Overview of Adsorption Cooling Systems

One way to decrease electricity demand due to cooling is to use environmentally benign, thermally-powered cooling systems such as adsorption systems where the term –adsorption‖ in this study refers to solid physical sorption, not chemical sorption. Basically, in an adsorption cooling cycle (or heat pump), the mechanical compressor in the well-known conventional vapor-compression air conditioners that is generally powered by electricity is replaced with a thermal compressor that is driven by low grade thermal energy like solar energy or waste heat, and they do not require electricity, except for circulation pumps. Although adsorption cooling systems are not widely available in the market, these systems are proposed as a promising technology that deserves further research since they do not include rotating compressor parts [7-9].
Some of the advantages of adsorption cooling systems relative to conventional vapor-compression systems can be listed as [7-9]

- Environmentally benign
- Can be operated with low-grade thermal energy such as solar, geothermal and waste heat
- Thermal energy storage possible
- Does not have moving parts, therefore long lifetime without noise and vibration
- Simple control and maintenance

Nevertheless, there are also some disadvantages of adsorption cooling systems compared with the conventional vapor-compression systems that can be listed as [7-9]

- Low coefficient of performance
- Intermittent (not continuous) cooling
- System has to be operated under vacuum conditions and it is hard to maintain this vacuum
- Larger volume and weight

1.3 Objectives of the Present Study

The prime objective of the study is to determine the effect of thermal conductance with different mass allocation on a re-heat two stage adsorption chiller. To get the optimum cooling capacity (CC) and the co-efficient of performance (COP) of the chiller, we have to investigate the following results:

- To investigate the system performance, thermal conductance of the sorption elements were varied by keeping the different mass ratio between upper and bottom beds (3:2 mass ratio) fixed.
- To determine the influence of the thermal conductance on the calculated Cooling Capacity.
- To investigate the effect of the thermal conductance on the calculated coefficient of performance.
- To determine the influence of the thermal conductance on the calculated average chilled water outlet temperature.
- To identify the effect of mass recovery time and total cycle time on system performance.
CHAPTER TWO

Basic Principles of Adsorption

2.1 Introduction

This chapter serves as an essential background before the modeling study. It aims to demonstrate the basic underlying principles of adsorption process. Adsorption of water vapor on the surface of certain solids is important process from the point of view of dehumidifying air for different uses such as air conditioning. The process of adsorption involves separation of a substance from one phase accompanied by its accumulation or concentration at the surface of another. The adsorbing phase is the adsorbent, and the material concentrated or adsorbed at the surface of that phase is the adsorbate. During the adsorption process the first layer of water molecules binds chemically on the surface of the solid desiccant material followed later by physical absorption. A desiccant cycle consists of repeated adsorption and desorption of water vapor from the desiccant material. Usually two air streams take part in this process; one stream loses moisture to the desiccant material while the other removes that vapor due to difference of vapor pressure. The basic desiccant cycle consists of three processes namely desorption, cooling and adsorption processes. The adsorption and desorption processes taking place in the desiccant, usually silica gel, coated matrix of a desiccant wheel.

2.2 Adsorption Principles

Adsorption is selective binding of a substance by another solid substance; Barrer et al.[10]. Almost all materials have the capacity to adsorb and hold water vapor but commercial desiccants such as silica gel has significant capacity for holding the water. These desiccants adsorb large amount of molecules into pores on their surface. This is made possible by the fact that these pores have a large active surface area in the range of 500 m$^2$/g; Jokisch et al.[11]. Silica gel is a commonly used adsorbent, which is a porous, amorphous form of silica (SiO$_2$). Due to its unique internal structure silica gel is radically different to other SiO$_2$-based materials. It is composed of a vast network of interconnected microscopic pores. Silica gel has great affinity for water vapors and in an unsaturated state it produces a very reduced partial pressure of 0.13 Pa; Willmes [12]. As a reference vapor pressure of water at a temperature of 32.88 °C and a relative humidity 60 % is 3000 Pa. Adsorption of water vapor involves two distinct processes which are

- chemical adsorption (chemisorption) and
- physical adsorption

Chemical adsorption or chemisorption, the initial adsorption process, arises due to binding of water molecules to hydroxyl group on the surface pores of silica gel; Ruthven et al.[13]. Chemical adsorption (chemisorption) occurs when covalent or ionic bonds are formed between the adsorbing molecules and the solid substance. The bonding forces of chemical adsorption are
much greater than that of physical adsorption. Thus, more heat is liberated. This bonding leads to change in the chemical form of the adsorbed compounds and hence, it is irreversible. For this particular reason, most of the adsorption processes applicable to the thermal system or cooling machine mainly involve physical adsorption.

Physical adsorption which follows the chemisorption in additional layers is a reversible process. Physical adsorption occurs when relatively weak intermolecular forces of attraction called Van Der Waal forces hold water molecules on the pore surface of adsorbent. It is physical adsorption which plays vital part in the adsorption of water vapor in solid desiccant wheel. As against the absorption process where the molecules of absorbate penetrate the volume of the absorbent, adsorption is a surface process in which the adsorbed molecules do not diffuse into the lattice of the solid. The molecules of adsorbed substance attach to or detach from the surface of the solid in a dynamic process governed by vapor pressure difference between the adsorbent surface and surrounding air. The molecules that collect on the surface form an interfacial layer that has properties similar to the liquid phase of the adsorbed substance. The process of adsorption is reversible for many substances.

Adsorption occurs in a series of three steps. In the first step, the contaminant is transferred from the bulk gas stream to the external surface of the adsorbent material. In the second step, the contaminant molecule diffuses from the relatively small area of the external surface (a few square meters per gram) into the macropores, transitional pores, and micropores within each adsorbent. Most adsorption occurs in the micropores because the majority of available surface area is there (hundreds of square meters per gram). In the third step, the contaminant molecule adsors to the surface in the pore. Fig. 2.1 illustrates this overall mass transfer, diffusion, and adsorption process.

Fig. 2.1 Adsorption steps.
Steps 1 and 2 occur because of the concentration difference between the bulk gas stream passing through the adsorbent and the gas near the surface of the adsorbent. Step 3 is the actual physical bonding between the molecule and the adsorbent surface. This step normally occurs more rapidly than steps 1 and 2.

2.3 Classification and types of Adsorbents

During adsorption, the gas stream passes through a bed or layer of highly porous material called the adsorbent. The compound or compounds to be removed, termed the adsorbate(s), diffuse to the surface of the adsorbent and are retained because of weak attractive forces, while the carrier gas passes through the bed without being adsorbed.

The adsorption capacity of an adsorbent is a function of the surface area of the pores in its body. Generally the internal surface area of pores is several orders of magnitude larger than the outer surface area of the adsorbents. Consequently the adsorbent can adsorb huge quantities of adsorptive. Traditionally adsorbents are classified according to the average diameter of their inner pores. The pore size determines the type of substance that can be adsorbed. Three classes are given below

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<th>Type of Adsorbent</th>
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<td>2 Å to 20 Å</td>
</tr>
<tr>
<td>Meso porous adsorbents</td>
<td>20 Å to 500 Å</td>
</tr>
<tr>
<td>Macro porous adsorbents</td>
<td>&gt; 500 Å</td>
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Where Å is the angstrom unit of length and is equal to one billionth of a meter, $10^{-10}$ m.

Several types of adsorption materials are available. The most common types of adsorbents for pollution control applications are describe bellow:

2.3.1 Silica gel

Silica gel is a granular, vitreous, porous form of silicon dioxide made synthetically from sodium silicate. It is tough and hard. It is a naturally occurring mineral that is purified and processed into either granular or beaded form. As a desiccant, it has an average pore size of 2.4 nanometers and has a strong affinity for water molecules. Silica gel is most commonly encountered in everyday life as beads in a small paper packet. In this form, it is used as a desiccant to control local humidity in order to avoid spoilage or degradation of some goods. In many items, moisture encourages the growth of mold and spoilage. Condensation may also damage other items like electronics and may speed the decomposition of chemicals, such as those in vitamin pills. Through the inclusion of silica gel packets, these items can be preserved longer. Silica gel may also be used to keep the relative humidity (RH) inside a high frequency radio or satellite transmission system waveguide as low as possible. It is also used to dry the air in industrial compressed air systems.
2.3.2 Activated alumina

Activated alumina is manufactured from aluminium hydroxide by dehydroxylating it in a way that produces a highly porous material; this material can have a surface area significantly over 200 square m/g. The compound is used as a desiccant and as a filter of fluoride, arsenic and selenium in drinking water. It is made of aluminium oxide, the same chemical substance as sapphire and rubies. It has a lot of very small pores, almost like tunnels, that run throughout it. Use as a desiccant, it works by a process called adsorption. The water in the air actually sticks to the alumina itself in between the tiny passages, as the air passes through them. The water molecules become trapped so that the air is dried out as it passes through the filter. This process is reversible, and if the alumina desiccant is heated to about 200°C it will release all of the water stored in it. This process is called regenerating the desiccant. Air always has some water in it; one way to measure the amount of water in the air is the air's dew point.

2.3.3 Zeolites

Zeolites are microporous, aluminosilicate minerals commonly used as commercial adsorbents. The term zeolite was originally coined in 1756 by Swedish mineralogist Axel Fredrik Cronstedt, who observed that upon rapidly heating the material stilbite, it produced large amounts of steam from water that had been adsorbed by the material. Based on this, he called the material zeolite. Zeolites are widely used in industry for water purification, as catalysts, for the preparation of advanced materials and in nuclear reprocessing. They are used to extract nitrogen from air to increase oxygen content for both industrial and medical purposes. Their biggest use is in the production of laundry detergents. They are also used in medicine and in agriculture. Zeolites can also be used as solar thermal collectors and for adsorption refrigeration.

2.3.4 Calcium chloride

Calcium chloride, CaCl₂, is a salt of calcium and chloride. It behaves as a typical ionic halide, and is solid at room temperature. Common applications include brine for refrigeration plants, ice and dust control on roads, and desiccation. Because of its hygroscopic nature, anhydrous calcium chloride must be kept in tightly sealed, air-tight containers. Drying tubes are frequently packed with calcium chloride. Adding solid calcium chloride to liquids can remove dissolved water.

2.3.5 Activated carbon

Activated carbon, also called activated charcoal, activated coal, or carbo activatus, is a form of carbon processed to be riddled with small, low-volume pores that increase the surface area available for adsorption or chemical reactions.

Due to its high degree of microporosity, just one gram of activated carbon has a surface area in excess of 500 m², as determined by adsorption isotherms of carbon dioxide gas at room or 0.0 °C temperature. An activation level sufficient for useful application may be attained solely from high surface area; however, further chemical treatment often enhances adsorption properties.
Activated carbon is used in gas purification, decaffeination, gold purification, metal extraction, water purification, medicine, sewage treatment, air filters in gas masks and respirators, filters in compressed air and many other applications.

2.3.6 Oxide

An oxide is a chemical compound that contains at least one oxygen atom and one other element in its chemical formula. Metal oxides typically contain an anion of oxygen in the oxidation state of −2. Most of the Earth's crust consists of solid oxides, the result of elements being oxidized by the oxygen in air or in water. Hydrocarbon combustion affords the two principal carbon oxides: carbon monoxide and carbon dioxide. Even materials considered pure elements often develop an oxide coating. For example, aluminium foil develops a thin skin of Al$_2$O$_3$ that protects the foil from further corrosion.

2.4 Refrigerants and adsorbents

There are several working pairs for solid adsorption. For the successful operation of a solid adsorption system, careful selection of the working medium is essential. It is because; the performance of the system varies over a wide range using different working pairs at different temperatures. The advantages and disadvantages of different working media and their properties are listed and discussed in this section. For any refrigerating application, the adsorbent must have high adsorptive capacity at ambient temperatures and low pressures but less adsorptive capacity at high temperatures and high pressures. Thus, adsorbents are first characterized by surface properties such as surface area and polarity. A large specific surface area is preferable for providing large adsorption capacity, and hence an increase in internal surface area in a limited volume inevitably gives rise to large number of small sized pores between adsorption surfaces. The size of the micropores determines the effectiveness of adsorptivity and therefore distribution of micropores is yet another important property for characterizing adsorptivity of adsorbents.

Based on the above discussion, the choice of the adsorbent will depend mainly on the following factors:

- high adsorption and desorption capacity, to attain high cooling effect;
- good thermal conductivity, in order to shorten the cycle time;
- low specific heat capacity;
- chemically compatible with the chosen refrigerant;
- low cost and widely available.

The selected adsorbate (working fluid) must have most of the following desirable thermodynamics and heat transfer properties:

- high latent heat per unit volume;
- molecular dimensions should be small enough to allow easy adsorption;
- high thermal conductivity;
• good thermal stability;
• low viscosity;
• low specific heat;
• non-toxic, non-inflammable, non-corrosive; and
• chemically stable in the working temperature range.

Based on the above criteria, some of the appropriate working pairs are zeolite–water, zeolite–organic refrigerants, silica gel–water, zeolite–water and activated carbon–methanol in solid adsorption systems. Several refrigeration applications have been studied using various adsorbent and adsorbate pairs. The recent new development of activated carbon fibre—ACF, shows the possibility for applications in adsorption refrigeration. One good example is the development in Byelorussia of a refrigerator prototype using ACF–ethanol and ACF–acetone pairs has been reported; Vasiliev et al.[14]. New experiments have also been shown by Vasiliev et al.[15] to use a heat pipe for heating/cooling ACF adsorbers for the ACF–NH3 pair.

In China, several studies had been carried out on solar powered refrigerators using different adsorption pairs such as zeolite–water and activated carbon-methanol. A theoretical analysis was carried out by Oertel and Fischer [16] for a prediction of the achievable COP using methanol/silica gel. Recently, Tamainot and Critoph [17] investigated the thermophysical properties of two types of monolithic activated carbons with an intention to design and fabricate a high performance generator for sorption refrigeration systems and heat pumps using ammonia as refrigerant. It was found that, reduction in volume from granular bed to monolithic bed was up to 50% which could lead to substantial economic gain.

2.5 Steps of adsorption Cycle

The ability of sorption materials such as silica gel to attract and hold large quantities of water on to its internal pore' surface is used to dehumidify air. The process of dehumidifying air with the help of desiccant materials is also called chemical dehumidification. Chemical dehumidification is based on the migration of water vapor from process air towards the surface of the desiccant due to the difference in partial vapor pressure. The pressure gradient is orientated in the direction of desiccant because it is dry and cool. Cargocaire [18] and Pietro et al.[19] give idea about the steps of a typical cycle of the desiccant..

2.5.1 Adsorption Process

Initially the adsorption material (desiccant material) is cool and dry. When air passes through desiccant material adsorption process starts gradually the air becomes drier and desiccant material experiences a rise in its water content. Eventually the desiccant material reaching saturation state where it cannot hold any more vapour and the vapour pressure of the desiccant surface becomes equal to surrounding air partial vapour pressure and the migration of vapour to the desiccant ceases.
2.5.2 Desorption Process

During desorption process vapor is removed from the adsorption material by exposing it to hot regeneration fluid flow. As a result of this heating the gradient changes its direction and the water vapour transfers from the desiccant to the air.

2.5.3 Cooling Process

The material is cooled until the starting temperature is reached consequently the humidity content and vapor pressure is restored; the cycle can now be repeated. It is apparent from above description that desiccant cycle is run by the vapor pressure difference between air and solid desiccant material. When the vapor pressure on desiccant surface is low as compared to surrounding air then water vapor flows toward it and is held by electrostatic forces this is adsorption. On the other hand when air has a lower vapor pressure due to high temperature or dryness then water vapor detach from the solid desiccant surface and are removed by the flowing air in a process called desorption. The regeneration or desorption can be obtained by using hot air or dry air.
CHAPTER THREE

Thermodynamics analysis of adsorption cycles

3.1 Introduction

As a good opportunity to replace CFCs or HCFCs refrigeration, adsorption refrigeration research has got enough attentions during the last few years, specially its potential applications in waste heat recovery, solar energy utilization etc. The research extends not only in refrigeration itself, but also to Thermodynamics, Chemical Engineering, Material Science, New Energy, and specific technologies are involved. Adsorption refrigeration with different cycles have been studied extensively by Wang [20], and the most important application of thermodynamics to adsorption is the calculation of phase equilibrium between a gaseous mixture and a solid adsorbent. Some of the typical adsorption refrigeration cycles are describing bellow:

3.2 Basic adsorption cycle

A basic adsorption cycle consists of four thermodynamic steps, which can be well represented with the aid of the Clapeyron diagram, as shown in Fig. 3.1 The idealized cycle begins at a point (point A in Fig. 3.1) where the adsorbent is at low temperature \( T_A \) and at low-pressure \( P_E \) (evaporating pressure). A–B represents the heating of adsorbent, along with adsorbate. The collector is connected with the condenser and the progressive heating of the adsorbent from B to D causes some adsorbate to be desorbed and its vapor to be condensed (point C). When the adsorbent reaches its maximum temperature \( T_D \); desorption ceases. Then the liquid adsorbate is transferred into the evaporator from C to E and the collector is closed and cooled. The decrease in temperature D to F induces the decrease in pressure from \( P_C \) to \( P_E \); Then the collector is connected to the evaporator and adsorption and evaporation occur while the adsorbent is cooled from F to A. During this cooling period heat is withdrawn to decrease the temperature of the adsorbent.

Fig.3.1 Clapeyron diagram of ideal adsorption
3.3. Heat recovery adsorption refrigeration cycle

The semi-continuous heat recovery cycle is usually operated with two adsorption beds. The adsorber to be cooled will transfer its heat to the adsorber to be heated, which includes sensible heat as well as heat of adsorption. This heat recovery process will lead to a higher system COP. Multi-beds could be also adopted to get more heat recovery and thereby to attain higher COP, but the operation of a practical system will be complicated. A quasi-continuous adsorption refrigeration system with heat recovery was investigated by Wang et al.[21, 22] and the flow path is shown in Fig. 3.2. While adsorber 1 is cooled and connected to the evaporator to realize adsorption refrigeration in evaporator, the adsorber 2 connected to the condenser is heated to obtain heating-desorption-condensation.

![Schematics of heat recovery two-beds adsorption refrigeration system.](image)

The condensed refrigerant liquid flows into evaporator via a flow control valve. The operation phase can be changed, and the go-between will be a short time heat recovery process. Two pumps are used to drive the thermal fluid in the circuit between two adsorbers (the connection to the heater and cooler are blocked during this process). Jones[23] suggested an improvement to the process by installing more than two adsorbers into the system. The operating principle of the cycle remains the same, relying on heat transfer flowing between the adsorbers and the desorbers. Comparing with the basic cycle, heat recovery in this process is only effective if the heat transfer fluid temperature leaving the adsorbers is sufficiently high. Simulation results have shown that the maximum value of the COP depends on the number of adsorbers and desorbers installed. The analysis was further extended to a system containing six adsorbers and six desorbers at the same test temperature conditions (evaporation at 5°C and condensation at 35°C).
3.4. Mass recovery adsorption refrigeration cycle

Apart from the above discussed heat recovery operation, it had been proved that mass recovery is also very effective for heat recovery adsorption heat pump operation. In this process, at the end of each half cycle, one adsorber is cold and the other one is hot. Meanwhile, the former one which is at low pressure, $P_c$ must be pressurized up to the condenser pressure, and similarly, the other one which is at high pressure must be depressurized down to the evaporator pressure. With just one tube between the adsorbers and a vapor valve, part of this pressurization–depressurization can be achieved by transferring vapor from the latter adsorber to the former one. This process can also be called as an internal vapor recovery process', and is reported to enhance the cooling power of the unit without reducing the COP by more than 10%.

The above explained process involves only mass transfer and hence the process is rapid. To obtain a _double effect_, mass recovery could be initiated followed by heat recovery. An ideal heat and mass recovery cycle is shown in Fig.3.3, in which the heat recovery state for a two bed system is shown by the state points $e - e'$. The mass recovery cycle ($a_2 - a_3 - g_1' - g_1 - g_2 - g_3 - a_1' - a_1 - a_2$) is an extended form of a two bed basic cycle or two bed heat recovery cycle ($a_2 - g_1 - g_2 - a_1 - a_2$) shown in Fig. 3.3, and the cycled mass is increased from $\Delta x$ to $\Delta x + \delta x$; which causes the refrigeration effect to increase. The principle of these cycles can be described using Fig. 3.3. The very first part of each half cycle is the mass recovery process (path $g_2 - g_3$ and $a_2 - a_3$). Then the heat recovery process proceeds: heat is transferred from the hot adsorber to the cold one (path $g_3 - e'$). As a consequence, the hot adsorber is first depressurized (path $g_3 - a_1'$), it then adsorbs vapor from the evaporator (path). Meanwhile, the cold adsorber is first pressurized (path $a_3 - a_1'$), and then vapor that is desorbed passes into the condenser (path $g_1' - e$). Theoretically, the heat recovery process develops until the adsorbers reach the same temperature. Actually, there still remains a temperature difference between the adsorbers at the end of this period. Then, for closing each half cycle, the adsorbers are, respectively, connected to the heat source and heat sink (path $e - g_2$ and $e' - a_2$). The second half-cycle is performed the same way except that the adsorbers now exchange their roles. Due to this process, about 35% of the total energy transmitted to each adsorber can be internally recovered, including part of the latent heat of sorption.

![Fig. 3.3. Diagram of heat and mass recovery cycle.](image-url)
3.5. Thermal wave cycle

To further improve the heat regenerative ratio, Shelton[24] had proposed an attractive cycle called ‘thermal wave cycle’. In this process, it is assumed that a large temperature gradient exists along an adsorption bed. Heating and cooling of the adsorbent beds is achieved via a heat transfer fluid such as high temperature oil. The system consists of two adsorber beds and two heat exchangers connected in series (Fig. 3.4) to effect semicontinuous process. The function of the bed and heat exchanger is to combine a large area of heat transfer surface with a low oil flow rate.

A typical thermal wave cycle is shown in Fig. 3.4. The cycle consists of two phases: In the first phase, the oil recovers heat from bed 2 (hot), has a further heat addition from the heat exchanger and then proceeds to heat bed 1 (cold). As the heating of the bed proceeds, bed 1 desorbs refrigerant which passes to the condenser (giving a useful heat output in the case of a heat pump) and bed 2 adsorbs gas from the evaporator which provides cooling. In the following phase (second phase) of the cycle the pump is reversed, and hence, bed 1 is cooled (adsorbing) and bed 2 is heated (desorbing) in a similar fashion until the original conditions are reached and the pump can again be reversed. Though the procedure is simple, significant heat recovery can be achieved. Further, the system would achieve much better performance due to the combination of the special nature of the internal bed heat exchangers and the low flow rate. Although many researchers have studied the cycle, up to now, there is no report of a successful prototype adopting thermal wave cycle. Also, some experimental reports had shown that the performance of the thermal wave cycle is not very good. The efficiency of the thermal wave regenerative system depends on a relatively large number of parameters: for example, rates of various heat transfer processes, the flow rate of the circulating fluid, the cycle time, the adsorber configuration, etc. A numerical analysis of adsorptive heat pumps with thermal wave heat regeneration had been presented by Sun et al. [25]. They had derived two time constants which can be used directly to quantify the relative importance of the two heat transfer processes. This allows ready determination of which of the two processes is rate limiting and needs to be improved. The work has also confirmed that the performance of an adsorptive heat pump system using a traditional packed-bed would be too low, even with a heat regeneration, and therefore a significant enhancement of heat transfer properties inside the adsorber is necessary.

Similar to the above numerical study, the effect of various operating parameters on the performance of an adsorptive thermal wave regenerative heat pump had been studied by Ben Amar et al.[26], theoretically. They had developed a two-dimensional model which
simultaneously considers heat and mass transfer in the bed. The results have shown that under ideal conditions, the performance of a thermal wave regenerative heat pump is considerably better than that of a basic ‘uniform temperature’ heat pump. The study showed that a COP greater than 1 and a power of cold production of near 200 W per kg of adsorbent could be obtained. For air-conditioning applications, these figures are slightly higher than those obtained with other single-stage solid-gas systems as chemical heat pumps; Spinner et al. [27].

3.6 Convective thermal wave cycle

Thermal wave cycles normally suffer from low power density because of poor heat transfer through the adsorbent bed. Rather than attempting to heat the bed directly, it is possible to heat the refrigerant gas outside the bed and to circulate it through the bed in order to heat the sorbent. The high surface area of the grains leads to very effective heat transfer with only low levels of parasitic power needed for pumping. Hence, Critoph [28, 29] has presented a modified version of a thermal wave cycle, known as ‘convective thermal wave cycle’. The concept is the same as thermal wave cycle, however, the thermal fluid for heating and cooling to the beds is initiated by the refrigerant itself, thus the heat transfer between thermal fluid and adsorption bed is a direct contact heat transfer, which is incorporated with mass transfer in the system. A practical schematic of the proposed system is shown in Fig. 3.5. The two ‘active’ beds are packed with activated carbon and the two ‘inert’ beds are packed with non-reactive particles such as steel balls. The diagram shows the first half of the cycle, during which Active bed 1 is heated and desorbs ammonia while Active bed 2 is cooled, adsorbing ammonia. In the fluid circulation loop shown on the left, a low power pump circulates ammonia steam through inert bed 1 which is initially hot. The gas stream is heated by the bed and a ‘cold’ wave passes through the bed from right to left. Having been preheated by the inert bed, the ammonia stream is heated to the maximum cycle temperature (150°C–200°C) in a heat exchanger. The ammonia gas then passes to active bed 1 where it heats the carbon. A ‘hot’ thermal wave passes from left to right through the active bed. As the temperature of the active bed rises it desorbs ammonia which first increases the pressure in the left hand loop and then condenses in the condenser, rejecting heat to the environment. The mass flow rate of circulating ammonia is typically ten times that of the condensing stream of ammonia and it may take about ten minutes for the two thermal waves to travel the length of their respective beds. In similar fashion to the left hand loop the circulating flow might be ten times the adsorption flow from the evaporator. The advantages of this system are:

- The four packed beds are in effect heat exchangers of very high surface areas but at minimal cost. They are not only cheap but very compact.
- There are only four conventional heat exchangers and this is the minimum number allowed by thermodynamics. These are the evaporator and condenser, a gas heater whereby high-grade heat is input and a gas cooler whereby the low grade heat of adsorption is rejected to the environment.
- The cycle is highly regenerative since the packed beds act like large counter flow heat exchangers. This results in good energy efficiency (i.e. high COP).
3.7 Multi-stage and cascading cycle

The adsorption cycles discussed in previous sections are applicable only to a single stage cycle. The single stage cycle systems have certain limitations, that is, they cannot effectively utilize high temperature heat source, as well as do not perform well at very low temperatures. Hence, to improve the system performance under such situations, adsorptive processes may be adapted for advanced cycles, such as, multi-stage and cascading cycle. The basic idea of a multi-stage cycle is to perform the desorption–condensation processes and evaporation–adsorption processes at different temperature/pressure levels by using the same working pair. The internal re-use of heat of condensation or adsorption can increase the system performance significantly. Another practical cycle that can make good use of high temperature heat source is the cascading cycle, which operates with different working pairs (either liquid/liquid or solid/liquid), such as zeolite–water/activated carbon–methanol, or zeolite–water/silica gel–water, etc. These cascading cycles are applied to situations especially, when there exists a large temperature difference between the heat source/ambient and the temperature in the evaporator/refrigeration space. For such situations, it may not be practical to use single stage cycle. Hence, one way of dealing with such situations is to perform the evaporation/refrigeration process in stages, that is, to have two or more cycles that operate in series at different temperature levels (cascading). A high temperature heat source (e.g. boiler) is used to drive the high temperature stage adsorption refrigeration cycle. The low temperature stage adsorption refrigeration is driven by sensible heat and heat of adsorption obtained from high temperature stage. To minimize the contribution of sensible heat, special care has been attached to the heat management of the adsorbers; n-adsorber cycles operating with a single evaporator and a single condenser have been proposed with sequences of heat recovery between adsorbers. Such cycles offer some advantages: for example, a single condenser is used and pressure in the n-adsorber unit is not higher than that in the unit operating an intermittent cycle; moreover, adsorption heat at high temperature is used as desorption heat at low temperature. Counteracting heat transfer fluid circuits between adsorbers.
reduces entropy generation in comparison with what happens in intermittent cycles. The driving heat supplied to the cycle is at high temperature level (Fig. 3.6) so that the entropy generation—due to the inadaptation between the temperature levels of the source and of the adsorber is much less in an n-adsorber cycle than in an intermittent cycle. The same thing happens for the rejected heat: the rejection temperature is much closer from the utility temperature with an n-adsorber cycle than with an intermittent cycle. Very similar conclusions to that drawn by Scharfe et al. [30] have been presented by Meunier [31] in the case of heat recovery between adsorbers. In a particular case, Meunier has shown that using an infinite number of adsorbers with ideal heat recovery between adsorbers, the maximum achievable—with given conditions of operating temperatures—would be a cooling COP equal to 1.85 corresponding to 68% of ideal Carnot COP. Another cascading cycle which includes a triple effect machine operating a cascade between a water zeolite heat pump and LiBr–H₂O refrigerator has been tested by Ziegler et al. [32]. A lot of theoretical studies exists on the possibilities of cascading cycles while few experimental data are available.

Fig. 3.6. n-Adsorber cascading cycle.

3.8 Conclusions

The development of adsorption system for refrigeration is promising. An overall thermodynamics-based comparison of sorption systems shows that the performance of adsorption systems depends highly on both the adsorption pairs and processes; Pons et al. [33]. The technology continues to develop, and the cost of producing power with solar thermal adsorption refrigeration is falling. If the costs of fossil fuels, transportation, energy conversion, electricity transmission and system maintenance are taken into account, the cost of energy produced by solar thermal adsorption systems would be much lower than that for conventional refrigeration systems. This work presents an overall review on the fundamental understanding on the various adsorption refrigeration cycles and the applicability of solar adsorption both in air conditioning and refrigeration, with the improvement of the COP. Of the several kinds of adsorption systems analyzed in this paper, the intermittent system has been extensively studied both theoretically and experimentally, owing to its simplicity and cost effectiveness. However, the main disadvantages such as long adsorption/desorption time have become obstacles for commercial production of the system. Hence, to compete with conventional absorption and vapor-compression technologies, more efforts should be made in enhancing the COP and SCP (A
summarization on the different systems along with their respective COP cooling and SCPs is listed in Table 3). Heat recovery, mass recovery, multi-bed and multi-stage technologies are promising technologies in improving the COP and SCP. Also, recent developments [34, 35] of activated carbon series—activated carbon fiber (ACF)—has called attention to the possibility for application in adsorption refrigeration. For instance, with a specially treated ACF, the measured adsorption capacity on methanol is 2–3 times that of normal activated carbon, and the estimated adsorption time of ACF on methanol is about 1/5 to 1/10 of that of normal activated carbon. Besides, activated carbon can be made with properties to suit particular applications by varying the activation time and temperature. Hence, with such technologies the specific refrigeration capacity as well as the COP of the adsorption refrigeration cycle can be improved. Adsorption technology combined with other technologies for multi-purpose application seems to be a new trend in the research. This will widen the area of applications of adsorption technologies and make the adsorption refrigeration more cost effective. Any method that improves the efficiency even marginally would improve the economic viability of operating such devices. Thus, further studies need to be carried out to validate the potential for possible application in household refrigerators.
CHAPTER FOUR

Multi-stage adsorption chiller using silica gel/water as adsorbent-adsorbate pair

Different types of adsorption chiller with their operating principals are described in this Chapter. The main goal is to work with multi-stage adsorption chiller without re-heat using silica gel/water as adsorbent-adsorbate pair.

4.1 Introduction

The quest to accomplish a safe and comfortable environment has always been one of the main preoccupations of the sustainability of human life. Widespread efforts are currently underway to utilize available energy resources efficiently by minimizing waste energy and develop replacements for the traditionally refrigerants (CFCs and HCFCs), which contribute to ozone depletion and greenhouse warming. Accordingly during the last few decades research aimed at the development of thermally powered adsorption cooling technologies has been intensified. They offer double benefits of reductions in energy consumption, peak electrical demand in tandem with adoption of environmentally benign adsorbent/refrigerant pairs such as zeolite/water[36], activated carbon/ammonia[37], activated carbon/methanol[38] and silica gel/water[39] without compromising the desired level of comfort conditions. It is well known that the performance of adsorption cooling/heating system is lower than that other heat driven heating/cooling systems specially, absorption system provided that the available heat source temperature is at 75°C or higher. From this context, many authors proposed and/or investigated the adsorption cooling and heating system to improve the performance. Meunier [40] investigated the system performance of cascading cycle in which an active/methanol cycle is topped by zeolite/water cycle. To improve the value of Coefficient of Performance (COP), Shelton et al.[41] proposed thermal wave regenerative adsorption heat pump. In a similar effort, Critoph [28] proposed a forced convection adsorption cycles. Pons and Poyelle [42] investigated the effect of mass recovery process in convectional two bed adsorption cycle to improve the cooling power. Later, Wang [22] showed the mass recovery process for COP and SCP values. Few cycles, however, are proposed to utilize relatively low temperature heat source. To utilized low temperature waste heat source between 40°C and 60°C, Saha et al.[43] proposed and examined experimentally a three-stage adsorption chiller with silica gel/water pair. Saha et al.[44] also introduce a two-stage adsorption chiller and the required driving heat source temperature is validated experimentally. In the two stage chiller, it is possible to get effective
cooling with driving heat source temperature bellow 60°C. A novel adsorption chiller, namely, ‘re-heat two stage’ chiller is introduced by Alam et al.[45] and he has shown that the chiller is able to exploit the heat source of temperature between 50°C and 90°C and can produce effective cooling. Khan et al.[46] studied the effect of thermal conductance and adsorbent mass on a two-stage adsorption chiller using re-heat scheme. Later, Khan et al.[47] investigated the performance evaluation of multi-stage, multi-bed adsorption chiller employing re-heat scheme. Saha, et al.[48] investigated the influence of the thermal conductance of sorption elements (adsorber/desorber, evaporator and condenser) on the performance of a silica-gel-water advanced adsorption chiller. They considered the thermal capacitance ratio of the adsorbent and metal of the adsorber/desorber heat exchanger. Recently, Farid et al.[49] studied numerically the performance investigation of a silica gel/water based on two-stage, four-bed adsorption chiller with different mass allocation. In the present study, the adsorption chiller investigated the effect of thermal conductance on cooling capacity (CC), the co-efficient of performance (COP), chilled water outlet and their improvement ratio by keeping the different mass ratio between upper and bottom beds (3:2 mass ratio) fixed.

4.2 Operating Principles

4.2.1 Single stage chiller without re-heat scheme

A single stage adsorption chiller consist of four heat exchangers, namely, a condenser, an evaporator and one pair adsorbent bed and metallic tubes for heat transfer fluid and refrigerant flows as shown in Fig. 4.1. In an adsorption refrigeration system, adsorbent beds are operated in a cycle through the four thermodynamic states, namely, pre-heating, desorption, pre-cooling and adsorption period. The chiller can be operated in different strategies. only one strategy has been described in the present study; the chiller has four modes, A, B, C, D (Table 4.1). To describe the cycle of the system mode A, it is assumed that Hex1-Evaporator is in adsorption process while Hex 2 – Condenser is in desorption process. In mode B, all valves are closed. Hex 1 is heated by hot water, this is called pre-heating position and Hex 2 is cooled down by cooling water, this is called pre-cooling position. These positions are happened only for a short intermediate period of time (30s for this system). In this process no adsorption/desorption occurs. After this short period of time, valves V_1 and V_4 are opened to allow refrigerant to flow from Hex 1 to condenser, from evaporator to Hex 2 (mode C). In mode D, all valves are closed and Hex 1 are in pre-cooling and Hex 2 are in pre-heating position respectively.
Fig. 4.1 Schematic of without re-heat single stage chiller Mode A

Fig. 4.1 Schematic of without re-heat single stage chiller Mode B
Fig. 4.1 Schematic of without re-heat single stage chiller Mode C

Fig. 4.1 Schematic of without re-heat single stage chiller Mode D
Fig 4.2 Conceptual P-T-X diagram for a single-stage chiller

Table 4.1 Operational strategies of a single-stage adsorption chiller

<table>
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<th>Mode</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
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<td>Pre-cooling</td>
<td>Pre-heating</td>
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<tr>
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<td>Adsorption</td>
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<td>Pre-heating</td>
<td>Adsorption</td>
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</table>
4.2.2 Two-stage chiller without re-heat scheme

Before going to describe working principle of a two-stage chiller employing re-heat scheme it is better to know about basic of two-stage chiller without re-heat and its operational strategy a conventional silica gel water adsorption cycle cannot be operational with the driving heat source temperature 50°C, if the heat sink is at 30°C or more. For practical utilization of these temperatures (between 50°C and 75°C) to adsorption chiller operation, an advanced two-stage adsorption cycle can be introduced. As can be seen from Fig. 4.3. the cycle allows reducing regeneration temperature (pressure) lift of the adsorbent (T\text{des} - T\text{ads}) by diving the evaporating temperature (pressure) lift (T\text{cond} - T\text{eva}) into two smaller lifts. Thus refrigerants (water-vapour) pressure rises into two consecutive steps from evaporation to condensation level; to achieve this objective, an additional pair of adsorber/desorber heat exchangers is necessary to include to the conventional two bed adsorption chiller. An advanced two-stage adsorption chiller consists of six heat exchangers, namely, a condenser, an evaporator and two pairs adsorbent bed heat exchangers as shown in Fig.4.3. In an adsorption refrigeration system, adsorbent beds are operated in a cycle through the four thermodynamics states, namely

(i) pre-heating
(ii) desorption
(iii) pre-cooling and
(iv) adsorption period.

To describe the cycle of the system, it is assumed that Hex-1 and Hex-4 are in cooling position while Hex-2 and Hex-3 are in heating position. At the beginning of the cycle all valves are closed. The desorbers (Hex-1 and Hex-4) are heated by hot water while adsorbers (Hex-2 and Hex-3) are cooled by cooling water. During a short intermediate process (30s for this system) no adsorption/desorption occurs. After this short period, valves 2, 4 and 6 are opened to allow refrigerant to flow from Hex-1 to condenser, from evaporator to Hex-2 and from Hex-4 to Hex-3. When refrigerant concentrations in the adsorbers and desorbers are at neat their equilibrium level, the flows of hot and cooling water are redirected by switching the valves so that the desorber can change its mode into adsorber and adsorber into desorber. The adsorption/desorption process can be continued by changing the direction of hot and cooling water flow. The standard operational strategy of two stages without re-heat is shown in Table-4.2
Fig. 4.3 Schematic of without re-heat two stage chiller Mode A

Fig. 4.3 Schematic of without re-heat two stage chiller Mode B
Fig. 4.3 Schematic of without re-heat two stage chiller Mode C

Fig. 4.3 Schematic of without re-heat two stage chiller Mode D
Fig 4.4 Conceptual P-T-X diagram for a two-stage chiller without re-heat

Table 4.2 Operational strategies of a two-stage adsorption chiller without re-heat

<table>
<thead>
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- Adsorption
- Mass recovery with cooling
- Pre-heating
- Desorption
- Mass recovery with heating
- Pre-cooling
4.2.3 Two-stage chiller with re-heat scheme

The design criteria of the two-stage adsorption chiller using re-heat is almost similar to that of a two-stage adsorption chiller without re-heat which is developed by Alam et al [50]. Operational strategy of a conventional two-stage chiller (Table 4.3), however, is completely different from operational strategy of a conventional two-stage chiller (Table 4.2). In two-stage adsorption chiller, the evaporating pressure lift is divided into two consecutive pressure lifts to exploit low heat source temperature by introducing four adsorbent beds. In the two-stage adsorption chiller using re-heat, the evaporating pressure (temperature) lift (Fig. 4.6), however, can be divided into different ways from the conventional two-stage chiller (Fig. 4.4). If one bed is in the end position of adsorption-evaporation process (Fig. 4.6) and another bed is in the end position of desorption-condensation process (Fig. 4.6), those two beds can be connected with each other by continuing cooling and heating and that can be done in the two bed conventional adsorption chiller which is known as the mass recovery process presented by Akahira et al. [51]. In this time, another two beds are connected with condenser and evaporator to continue cooling effects. to complete one full cycle in re-heat scheme, all adsorbent beds pass through six consecutive steps:

i) desorption     ii) mass recovery process with heating     iii) pre-cooling
iv) adsorption    v) mass recovery process with cooling     vi) pre-heating.

The two-stage adsorption chiller using re-heat comprises with four adsorbent beds, one condenser, one evaporator, and metallic tubes for hot, cooling and chilled water flows as shown in Fig. 4.5. In a conventional two stage chiller, lower two beds never interact with the condenser and upper two beds never interact with the evaporator. However, in the two-stage adsorption chiller using re-heat, all beds undergo through all process and interact with the condenser and evaporator. The chiller can be operated in different strategies. However, only one strategy has been considered in the present study. the chiller has 10 modes, mode-A, B, C, D, E, F, G, H, I and J (Table 4.3). Mode A, B, C and eva-Hex-2 is in adsorption process and cond-Hex-1 is in desorption process. In the adsorption-evaporation process, refrigerant (water) in evaporator is evaporated at evaporation temperature, $T_{eva}$ and seized heat, $Q_{eva}$ from the chilled water. The evaporated vapor is adsorbed by adsorbent (silica gel), at which cooling water removes the adsorption heat, $Q_{ads}$, the desorber (Hex-1) is heated up to the temperature ($T_{des}$) by heat $Q_{des}$, provided by the driving heat source. The resulting refrigerant is cooled down by temperature ($T_{cond}$) in the condenser by the cooling water, which removes heat, $Q_{cond}$. In Mode A, adsorber (Hex-4) is connected with desorber (Hex-3) through pipe with continuing cooling water in Hex-4 and hot water in Hex-3. It is noted that, at the beginning of mode A, Hex-4 was in the end position of evaporation-adsorption process and Hex-3 was in the end position of desorption-condensation process. Due to higher pressure difference at Hex-3 and Hex-4 in mode A, the refrigerant mass circulation will be higher than that of conventional two stages. This will lead the chiller to provide better performance. In mode B, adsorber (Hex-4) is heated up by hot water and the desorber (Hex-3) is cooled down by cooling water. Mode B is warm up process for Hex-4 and Hex-3.
When the pressure of adsorber (Hex-3) and desorber (Hex-4) are nearly equal to the pressure of evaporator and condenser respectively, then Hex-3 and Hex-4 are connected to evaporator and condenser respectively to flow the refrigerant.

This mode is denoted as mode C. In mode C, Hex-4 works as desorber and hex-3 works as adsorber and this process will continuing up to mode E. Om Mode D, Hex-2 is at the end position of adsorption-evaporation process and Hex-1 is at the end position of desorption-condensation process. And are connected worn each other continuing cooling water and hot water respectively.

When the pressure (temperature) of both Hex-1 and Hex-2 are nearly equal, the warm up process will start, called mode E. In Mode E, Hex-2 is heated up by hot water, and cooling water cools down the Hex-1. When the pressure of Hex-1 and Hex-2 are nearly equal to the pressure of evaporator and condenser respectively, then Hex-1 and Hex-2 are connected to evaporator and condenser respectively to flow the refrigerant. This connection will continue up to mode F, G and H for Hex-1 and Hex-2. In mode F, Hex-3 (at the end position of adsorption-evaporation process) and Hex-4 is at the end position of desorption-condensation process) are connected worth each other continuing cooling water and hot water respectively. When the pressure (temperature) of both Hex-3 and Hex-4 are nearly equal, the warm up process will start, called mode G. In Mode G, Hex-3 is heated up by hot water, and cooling water cools down the Hex-4. When the pressure of Hex-3 and Hex-4 are nearly equal to the pressure of evaporator and condenser respectively, then Hex-3 and Hex-4 are connected to evaporator and condenser respectively to flow the refrigerant. This connection will continue up to mode H, I and J for Hex-3 and Hex-4. In mode I, Hex-1 (at the end position of adsorption-evaporation process) and Hex-2 is at the end position of desorption-condensation process) are connected worth each other continuing cooling water and hot water respectively. When the pressure (temperature) of both Hex-1 and Hex-2 are nearly equal, the warm up process will start, called mode J. In Mode J, Hex-1 is heated up by hot water, and cooling water cools down the Hex-2. The mode J is the last process and after this mode, all return to mode A (Table-1). The values adapted in simulation are presented in Table 4.4 and Table 4.5.
Fig. 4.5 Schematic of a re-heat two stage chiller Mode A

Fig. 4.5 Schematic of a re-heat two stage chiller Mode B
Fig. 4.5 Schematic of a re-heat two stage chiller Mode C

Fig. 4.5 Schematic of a re-heat two stage chiller Mode D
Fig. 4.5 Schematic of a re-heat two stage chiller Mode E
Fig. 4.5  Schematic of a re-heat two stage chiller Mode G

Fig. 4.5 Schematic of a re-heat two stage chiller Mode H

Hex : Heat Exchanger
X : Valve-closed
X : Valve-open

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Fig. 4.5 Schematic of a re-heat two stage chiller Mode I

Fig. 4.5 Schematic of a re-heat two stage chiller Mode J
Fig. 4.6 Conceptual P-T-X diagram for a two-stage chiller with re-heat

Table 4.3 Operational strategies of a two-stage adsorption chiller with re-heat

<table>
<thead>
<tr>
<th>Mode</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>H</th>
<th>I</th>
<th>J</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hex 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Hex 2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hex 3</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hex 4</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Legend:
- Adsorption
- Mass recovery with cooling
- Pre-heating
- Desorption
- Mass recovery with heating
- Pre-cooling
CHAPTER FIVE

Mathematical model

The mathematical models presented in this Chapter are used in the simulation procedure.

5.1. Energy balance for the adsorber/desorber

Adsorption and desorption heat balances [43] are expressed by identical equations, where heat transfer fluid (water) temperature terms $T_{in}$ and $T_{out}$ indicate cooling water upon adsorption and hot water upon desorption. $T$ stands for bed temperature. In addition, the adsorbent bed temperature, pressure and concentration are assumed to be uniform all over the adsorbent bed. The specific of refrigerant (water) in liquid phase is taken as our system works in low the concentration range. The heat transfer and energy balance equations for the adsorbent beds can be described as follows:

$$T_{out} = T + (T_{in} - T) \exp \left(-\frac{U_{hex} A_{hex}}{m_{w} c_{w}}\right)$$  \hspace{1cm} (5.1)

$$\frac{d}{dt}\left\{ (w_{s} c_{s} + w_{s} c_{w} q + w_{hex} c_{hex}) T \right\} = w_{s} Q_{st} \frac{dq}{dt} - \delta w_{s} c_{w} \{\gamma (T - T_{eva}) + (1 - \gamma)(T - T_{ww})\} \times \frac{dq}{dt} + m_{w} c_{w} (T_{in} - T_{out})$$  \hspace{1cm} (5.2)

where $\delta = \begin{cases} 0, & \text{desorption process} \\ 1, & \text{adsorption process} \end{cases}$ and $\gamma = \begin{cases} 1, & \text{bed is connected with evaporator} \\ 0, & \text{bed is connected with another bed} \end{cases}$

The importance of heat transfer parameters is described in Eq. (5.1). These parameters are heat transfer area $A_{hex}$ and heat transfer coefficient $U_{hex}$. The left hand side of the adsorber/desorber energy balance equation (Eq. 5.2) provides the amount of sensible heat required to cool or heat the silica–gel (s), the water (w) as well as metallic (hex) parts of the heat exchanger during adsorption or desorption. In addition, the first term on the right hand side of Eq. (5.2) stands for the release of adsorption heat or the input of desorption heat, while the second and third terms represent for the sensible heat of the adsorbed vapor. And, the last term on the right hand side of Eq. (5.2) gives the total amount of heat released to the cooling water upon adsorption or provided by the hot water for desorption. As all beds are assumed to be well insulated, Eq. (5.2) does not account for external heat losses to the environment.
5.2 Energy balance for the evaporator

In this study, it is considered that the tube bank surface is able to hold a certain maximum amount of condensate and the condensate would flow into the evaporator easily. The heat transfer and energy balance equation (5.3) for the evaporator can be written as

\[ T_{eva, out} = T_{eva} + (T_{chill, in} - T_{eva}) \exp \left( -\frac{U_{hex} A_{hex}}{m_{chill} c_{chill}} \right) \]  

\[ \frac{d}{dt} \{ (w_{eva,w} c_w + w_{eva,hex} c_{eva,hex}) T_{eva} \} = -L w_s \frac{dq_{ads}}{dt} - w_s c_w (T_{cond} - T_{eva}) \frac{dq_{des}}{dt} \]

\[ + \dot{m}_{chill} c_{chill} (T_{chill,in} - T_{chill,out}) \]  

(5.4)

where the suffixes chill and eva indicate chilled and evaporator respectively.

The sensible heat required by the liquid refrigerant \( w \) and the metal of heat exchanger tubes in the evaporator can be found from the left hand side of Eq. (5.4). The first term on the right hand side of Eq. (5.4) represents the latent heat of evaporation \( L \) for the amount of refrigerants adsorbed \( \frac{dq_{ads}}{dt} \), while the second term gives the sensible heat required to cool down the incoming condensate from the condensation temperature \( T_{cond} \) to evaporation temperature \( T_{eva} \), and the last term indicates the total amount of heat given away by the chilled water.

5.3 Energy balance for the condenser

The heat transfer and energy balance equation (5.3) for the condenser can be written as

\[ T_{cond, out} = T_{cond} + (T_{cw, in} - T_{cond}) \exp \left( -\frac{U_{cond} A_{cond}}{m_{cw} c_w} \right) \]  

\[ \frac{d}{dt} \{ (w_{cw,w} c_w + w_{cond,hex} c_{cond,hex}) T_{cond} \} = -L w_s \frac{dq_{ads}}{dt} - w_s c_w (T_{des} - T_{cond}) \frac{dq_{des}}{dt} \]

\[ + \dot{m}_{cw} c_w (T_{cw,in} - T_{cw,out}) \]  

(5.6)

5.4 Mass balance

Mass and heat balances are based on the assumption that both the temperature and the amount of refrigerant adsorbed are uniform in the adsorbent beds. As the temperatures in an adsorption cycle are unsteady, the energy balance equations (Eqs. 5.2, 5.4, and 5.6) must account for sensible heat input and/or output during the cycle period. The mass balance for the refrigerant can be written as

\[ \frac{dq_{eva,w}}{dt} = -W_s \left( \frac{dq_{des-cond}}{dt} + \frac{dq_{eva-ads}}{dt} \right) \]  

(5.7)
where the subscripts des-cond and eva-ads stand for the refrigerant vapor flows from desorber to condenser and evaporator to adsorber, respectively.

5.5 Adsorption rate

The adsorption rate is expressed as

\[
\frac{dq}{dt} = \kappa_s a_p (\dot{q} - q) \tag{5.8}
\]

where the overall mass transfer coefficient (\(\kappa_s a_p\)) for adsorption is considered as

\[
\kappa_s a_p = \frac{15D_s}{(R_p)^2} \tag{5.9}
\]

The adsorption rate is assumed to be controlled by surface diffusion inside a gel particle and surface diffusivity (\(D_s\)) is expressed by Sakoda and Suzuki [54] as a function of temperature by

\[
D_s = D_{so} \exp[-(E_a)/(RT)] \tag{5.10}
\]

and \(q^*\) is the amount adsorbed in equilibrium with pressure \(P_s(T_w)\) and is derived from the manufacturer property data by the following equation

\[
q^* = \frac{0.8[P_s(T_w)/P_s(T_s)]}{1+0.5[P_s(T_w)/P_s(T_s)]} \tag{5.11}
\]

where \(P_s(T_w)\) and \(P_s(T_s)\) are the saturation vapor pressure at temperatures \(T_w\) (water vapor) and \(T_s\) (silica gel), respectively. The saturation vapor pressure and temperature are correlated by Antoine's equation, which can be written as

\[
P_s = 133.32 \exp\left(18.3 - \frac{3820}{T-46.1}\right) \tag{5.12}
\]

5.6 Measurement of the system performance

The performance of a two-stage adsorption chiller using re-heat is mainly characterized by cooling capacity (CC), coefficient of performance (COP) and can be measured by the following equations

\[
\text{Cooling Capacity (CC)} = \dot{m}_{chill} c_w \int_0^{t_{\text{cycle}}} (T_{\text{chill,in}} - T_{\text{chill,out}}) \, dt / t_{\text{cycle}} \tag{5.13}
\]

\[
\text{Coefficient of performance (COP)} = \frac{\dot{m}_{chill} c_w \int_0^{t_{\text{cycle}}} (T_{\text{chill,in}} - T_{\text{chill,out}}) \, dt}{\dot{m}_{hot} c_w \int_0^{t_{\text{cycle}}} (T_{\text{hot,in}} - T_{\text{hot,out}}) \, dt} \tag{5.14}
\]
Table 4.4 Baseline Parameters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_{\text{bed}}$</td>
<td>1.45</td>
<td>m²</td>
</tr>
<tr>
<td>$A_{\text{eva}}$</td>
<td>0.665</td>
<td>m²</td>
</tr>
<tr>
<td>$A_{\text{con}}$</td>
<td>0.998</td>
<td>m²</td>
</tr>
<tr>
<td>$C_s$</td>
<td>924</td>
<td>J/kgK</td>
</tr>
<tr>
<td>$C_w$</td>
<td>4.18E+3</td>
<td>J/kgK</td>
</tr>
<tr>
<td>$C_{\text{chill}}$</td>
<td>4.20E+3</td>
<td>J/kgK</td>
</tr>
<tr>
<td>$D_{so}$</td>
<td>2.54E-4</td>
<td>m²/s</td>
</tr>
<tr>
<td>$E_a$</td>
<td>2.33E+3</td>
<td>J/kg</td>
</tr>
<tr>
<td>$L$</td>
<td>2.50E+6</td>
<td>J/kg</td>
</tr>
<tr>
<td>$Q_{st}$</td>
<td>2.80E+6</td>
<td>J/kg</td>
</tr>
<tr>
<td>$R$</td>
<td>4.62E+2</td>
<td>J/kg</td>
</tr>
<tr>
<td>$R_p$</td>
<td>0.35E-3</td>
<td>M</td>
</tr>
<tr>
<td>$UA_{\text{des}}$</td>
<td>2532.5</td>
<td>Wk⁻¹</td>
</tr>
<tr>
<td>$W_s$</td>
<td>16</td>
<td>kg</td>
</tr>
<tr>
<td>$W_{cw}$</td>
<td>5</td>
<td>kg</td>
</tr>
<tr>
<td>$W_{\text{eva, w}}$</td>
<td>25</td>
<td>kg</td>
</tr>
</tbody>
</table>

Table 4.5 Standard operating conditions

<table>
<thead>
<tr>
<th></th>
<th>Temperature [°C]</th>
<th>Flow rate [kg/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Hot water</strong></td>
<td>80</td>
<td>0.5</td>
</tr>
<tr>
<td><strong>Cooling water</strong></td>
<td>30</td>
<td>0.3(ads) + 0.3(cond)</td>
</tr>
<tr>
<td><strong>Chilled water</strong></td>
<td>14</td>
<td>0.3</td>
</tr>
<tr>
<td><strong>Cycle Time</strong></td>
<td>1800s=(630 ads/des + 240mr + 30 ph/pc) s×2s</td>
<td></td>
</tr>
</tbody>
</table>

Ads/des = adsorption/desorption, mr = mass recovery, ph/pc = pre-heat/pre-cool
CHAPTER SIX

Solution Procedure

6.1 Solution Procedure (Finite difference technique)

In the present study, a cyclic simulation computer program is developed to predict the performance of the innovative two-stage chiller using the re-heat scheme. The system of differential Eqs. (5.1)–(5.12) (Adsorber/desorber energy balance equations, Evaporator energy balance, Condenser energy balance, Mass balance, Adsorption rate) are solved by employing finite difference approximation with a time step 1 sec. The results taken in the study are from the cyclic steady state conditions. A real chiller starts its function with unbalanced conditions. However, after a few cycles it reaches its cyclical steady state condition. Thus, an iteration process has been used in solution technique to fix all the initial values for the cyclic steady state conditions. From the beginning of the solution process, the initial values are considered and lastly those are adjusted by this iteration process.

When two beds are connected with evaporator or condenser, the vapor pressure is unknown that are calculated through the Antonie's equations as the vapor temperature is calculated from the energy balance equation of evaporator or condenser. It is, however, difficult to calculate the saturated vapor pressure when two beds are connected with each other, which are essential for the calculation of adsorption/desorption rate inside the adsorbent beds. In the state, the pressure is assumed and the amounts of vapor adsorbed/desorbed beds are calculated. Conceptually, the desorbed vapor is equal to the amount of adsorbed vapor by the other beds. If these amounts are not equal then vapor pressure is adjusted for next iteration. Once the satisfactory convergence criterion is achieved, then process goes for the next time step. The convergence criterion for all cases of present study has been taken as $10^{-3}$.

Elaborate calculations for heat transfer equation (4.1) and energy balance equation (4.2) for desorber:

For heat transfer equation:

We know the heat transfer equation, $Q_{water} = \dot{m} c_p (T_{wi} - T_{wo})$

Again, we know from thermodynamics, Logarithms Mean Temperature difference,

$$LMTD = \frac{(T_{wo} - T_{ad}) - (T_{wi} - T_{ad})}{\ln \left( \frac{(T_{wo} - T_{ad})}{(T_{wi} - T_{ad})} \right)}$$
The heat equation for water is,

\[ Q_{water} = LMTD \times UA \]

or,

\[ \dot{m}c_p (T_{wi} - T_{wo}) = \frac{(T_{wo} - T_{ad}) - (T_{wi} - T_{ad})}{\ln \left( \frac{T_{wo} - T_{ad}}{T_{wi} - T_{ad}} \right)} \times UA \]

or,

\[ -\dot{m}c_p = \frac{UA}{\ln \left( \frac{T_{wo} - T_{ad}}{T_{wi} - T_{ad}} \right)} \]

or,

\[ -\frac{UA}{\dot{m}c_p} = \ln \left( \frac{T_{wo} - T_{ad}}{T_{wi} - T_{ad}} \right) \]

or,

\[ \exp \left( -\frac{UA}{\dot{m}c_p} \right) = \frac{T_{wo} - T_{ad}}{T_{wi} - T_{ad}} \]

or,

\[ (T_{wo} - T_{ad}) = (T_{wi} - T_{ad}) \exp \left( -\frac{UA}{\dot{m}c_p} \right) \]

or,

\[ T_{wo} = T_{ad} + (T_{wi} - T_{ad}) \exp \left( -\frac{UA}{\dot{m}c_p} \right) \]

For energy balance equation:

\[
\frac{d}{dt} \left\{ \left( w_s c_s + w_s c_w q_{des} + w_{des} c_{des} \right) T_{des} \right\} = w_s Q_{st} \frac{dq_{des}}{dt} + \dot{m}_w c_w (T_{in} - T_{out})
\]

or,

\[
(w_s c_s + w_s c_w q_{des} + w_{des} c_{des}) \frac{dT_{des}}{dt} + w_s c_w \frac{dq_{des}}{dt} T_{des} = w_s Q_{st} \frac{dq_{des}}{dt}
\]

\[
+ \dot{m}_w c_w \left\{ T_{in} - T_{des} - (T_{in} - T_{des}) \exp \left( -\frac{UA}{\dot{m}_w c_w} \right) \right\}
\]

(5.15)

Let, \( w_s c_s + w_s c_w q_{des} + w_{des} c_{des} = \gamma \)

Then equation (5.15) becomes,

\[
\gamma \frac{T_i - T_{i-1}}{\Delta t} + w_s c_w \frac{q_i - q_{i-1}}{\Delta t} T_i = w_s Q_{st} \frac{q_i - q_{i-1}}{\Delta t}
\]

\[
+ \dot{m}_w c_w \left\{ T_{in} - T_{des} - (T_{in} - T_{des}) \exp \left( -\frac{UA}{\dot{m}_w c_w} \right) \right\}
\]

(5.16)
Let, \[ NTU = \frac{UA}{m_w c_w} \]

Then equation (5.16) becomes,

\[
\gamma \frac{T_i - T_{i-1}}{\Delta t} + w_s c_w \frac{q_i - q_{i-1}}{\Delta t} T_i
= w_s Q_{st} \frac{q_i - q_{i-1}}{\Delta t} + \dot{m}_w c_w \{T_{in} - T_{des} - (T_{in} - T_{des}) \exp(-NTU)}
\]

or, \[
\gamma \frac{T_i - T_{i-1}}{\Delta t} + w_s c_w \frac{q_i - q_{i-1}}{\Delta t} T_i
= w_s Q_{st} \frac{q_i - q_{i-1}}{\Delta t} + \dot{m}_w c_w (T_{in} - T_{des}) \{1 - \exp(-NTU)}.
\]

or, \[
\gamma (T_i - T_{i-1}) + w_s c_w (q_i - q_{i-1}) T_i
= w_s Q_{st} (q_i - q_{i-1}) + \Delta \dot{m}_w c_w (T_{in} - T_{des}) \{1 - \exp(-NTU)}.
\]  

(5.16)

Let, \[ EXPN = 1 - \exp(-NTU) \]

Then equation (5.16) becomes,

\[
T_i \{\gamma + w_s c_w (q_i - q_{i-1}) + \Delta \dot{m}_w c_w EXPN\} = \gamma T_{i-1} + w_s Q_{st} (q_i - q_{i-1}) + \Delta \dot{m}_w c_w \dot{T}_{in} EXPN
\]

Again let, \[ EXPND = \dot{m}_w c_w EXPN \]

Then (5.16) takes the form,

\[
T_i \{\gamma + w_s c_w (q_i - q_{i-1}) + \Delta EXPND\} = \gamma T_{i-1} + w_s Q_{st} (q_i - q_{i-1}) + \Delta \dot{T}_{in} EXPND
\]

which implies,

\[
T_i = \frac{\gamma T_{i-1} + w_s Q_{st} (q_i - q_{i-1}) + \Delta \dot{T}_{in} EXPND}{\gamma + w_s c_w (q_i - q_{i-1}) + \Delta EXPND}.
\]
CHAPTER SEVEN

Results and Discussion

In the present treatment, the effect of thermal conductance of sorption element, evaporator and condenser on the performance of the chiller, chilled water outlet and their improvement ratio are discussed. It should be noted that the total silica gel mass of the four beds are considered to be the same. However, the mass ratio of upper bed and lower bed are kept 3:2 fixed.

In Figs. 7.1(a) and 7.1(b), numerical values of Cooling Capacity (CC) and Coefficient of Performance (COP) are depicted against the diving heat source inlet temperatures which vary from 50°C to 90°C. Four lines are drawn for four different thermal conductance (UA) values for adsorber, namely c1 = 497.6 W/K, c2 = 2497.6 W/K, c3 = 4497.6 W/K, c4 = 6497.6 W/K respectively. It is observed from these figures that both CC and COP increase with the increase of heat source temperature in all cases. It is noticed that, CC and COP increase as UA of adsorber increases. This is because large UA values mean large heat transfer area or high heat transfer coefficient or both, this leads to increase refrigerant inside adsorbent elements, so high performance. Another important observation is that, if COP is in concern, then UA of adsorber gives optimal result for the value 2497.6 W/K. As the value of UA of adsorber greater than 2497.6 W/K, COP shows insignificant effect. From Fig. 7.1(b), it is seen that, in 50°C to approximate 60°C heat source temperatures, CC shows maximum result for the thermal conductance (UA) value of adsorber 2497.6 W/K. After the heat source temperature 60°C, 4497.6 W/K gives optimum result.

In Figs. 7.2(a) and 7.2(b), it is observed that UA of condenser has increasing effect on both CC and COP as heat source temperature increases. It can be also noticed that maximum CC and corresponding COP are no longer beneficial for UA of condenser greater than 2404.3 W/K. However, for low heat source temperature (50°C to 60°C), COP is maximum for the value of UA of condenser 1904.3 W/K which is observed from fig. 7.3(b).

From Figs. 7.3(a) and 7.3(b), it is seen that CC and COP increase when both UA of evaporator and heat source temperature increase. It is observed that both CC and COP increase up to its maximum value where the value of UA of evaporator is equal to 2989.9 W/K. So, UA value of evaporator greater than 2989.9 W/K is no longer effective for the present baseline condition.

In Figs. 7.4(a) and 7.4(b), the improvement ratios of Cooling Capacity (CC) and Coefficient of Performance (COP) are depicted against the diving heat source inlet temperatures which vary from 50°C to 90°C. These two figures show the effect of thermal conductance of evaporator on the improvement ratios of CC and COP. It is observed that the CC improvement ratios are:
12.54%, 12%, 13.04%, 13.8%, 14.61% and COP improvement ratios are: 11.07%, 8.56%, 8.06%, 7.24%, 6.5%, when the thermal conductance of evaporator ratio is c1 to c2. Again when the UA_e ratio is c1 to c3, the CC improvement ratios are: 18.58%, 16.27%, 17.15%, 18.55%, 19.27% and COP improvement ratios are: 16.74%, 11.59%, 10.43%, 9.67%, and 8.17%. It is also seen that the CC improvement ratios are: 21.17%, 18.30%, 18.272%, 19.71%, 21.6% and COP improvement ratios are: 19.06%, 13.14%, 10.76%, 9.82% and 9.36%, when the thermal conductance of condenser ratio is c1 to c4.

Fig. 7.1(a) Effect of Thermal conductance [UA] of adsorber on cooling capacity

Fig. 7.1(b) Effect of Thermal conductance [UA] of adsorber on coefficient of performance
Fig. 7.2(a) Effect of Thermal conductance [UA] of condenser on cooling capacity

Fig. 7.2 (b) Effect of Thermal conductance [UA] of condenser on coefficient of performance
Fig. 7.3(a) Effect of Thermal conductance [UA] of evaporator on cooling capacity

Fig. 7.3(b) Effect of Thermal conductance [UA] of evaporator on coefficient of performance
Fig. 7.4(a) Effect of Thermal conductance [UA] of evaporator on CC improvement

Fig. 7.4(b) Effect of Thermal conductance [UA] of evaporator on COP improvement
In an adsorption chiller, the chilled water for air-condition purposes is obtained from the outlet of the chilled water. Generally, less chilled water outlet temperature is expected, while the requirement of cooling capacity is high. The chilled water outlet temperature, however, affects cooling demand of the demand side. Therefore the requirement of chilled water outlet temperature is very important. From this context, the effect of heat source temperature as well as UA of adsorber, evaporator and condenser are presented in Figs. 7.5 (a), 7.5 (b) and 7.5 (c). It may be seen that the chiller with different level heat source temperature provides different level chilled water outlet temperature. It can be also seen that the higher is the heat source temperature; the lower is the chilled water outlet temperature.

From Fig. 7.6(a), it is seen that CC increases with increase of heat source temperature from 50°C to 90°C in all cases. One interesting observation is that, CC increases as total time cycle increases, but it happens only for heat source temperature from 50°C to approximate 55°C. After that temperature the phenomena is totally reverse; that is, CC decreases when cycle time increases from 1100 to 3300. But opposite tendency is observed for COP (Fig. 7.6(b)). Because of an excessive long cycle time, the outlet temperature of hot water approaches its inlet temperature value. However there is still some cold production. Hence COP increases as cycle time increases for all heat source temperature.

The effect of heat source temperature on cooling capacity and coefficient of performance for different mass recovery times are shown in figure 7.7 (a) and 7.7 (b) respectively. It is observed that as the mass recovery time increases the CC also increases but the opposite phenomena is appeared for the coefficient of performance (COP). It is noticed that COP is high for low value of mass recovery time. It should be mentioned that in this case total cycle time is considered 1800 sec.

Fig. 7.5(a) Effect of Thermal conductance [UA] of adsorber on chilled water outlet
Fig. 7.5(b) Effect of Thermal conductance [UA] of condenser on chilled water outlet performance

Fig. 7.5(c) Effect of thermal conductance of evaporator on chilled water outlet performance
Fig. 7.6(a) Effect of total time cycle on cooling capacity

Fig. 7.6(b) Effect of total time cycle on coefficient of performance
Fig. 7.7(a) Effect of mass recovery time on cooling capacity

Fig. 7.7(b) Effect of mass recovery time on coefficient of performance
In figs. 7.8(a) and 7.8(b), the numerical values of CC and COP are depicted against the cooling water temperature which varies from 20°C to 40°C. From these two figures, it is observed that, CC and COP are maximum for low value of cooling water temperature. As the cooling water temperature increases, the performance of the machine is decreased. It should be mentioned that, in this case, heat source temperature 80°C is fixed and the time cycle and mass recovery time are considered 1800 sec and 240 sec respectively. The values of thermal conductance for which the system performance is optimum, are considered to plot the graphs of CC and COP shown in figs. 7.8(a) and 7.8(b).

The present study improves the performance of the chiller relative to the earlier study undertaken by Farid et al. [49]. In this study, the total cycle time, mass recovery time and thermal conductance of evaporator are taken different from the previous study and the fig.7.9 shows the effect of cooling water temperature on cooling capacity. From the figure, it is clear that the present study gives a better performance of the chiller than the earlier work for the changes undertaken in the study.

Fig. 7.8(a) Effect of cooling water temperature on cooling capacity
Fig. 7.8(b) Effect of cooling water temperature on coefficient of performance

Fig. 7.9 Effect of Cooling water temperature on cooling capacity
CHAPTER EIGHT

Conclusions

In this study, the effect of thermal conductance on the performance of a two-stage adsorption chiller with re-heat scheme has been investigated numerically. A finite difference iterative technique is used in the analysis for the solution procedure and mass ratio of upper and lower bed is kept 3:2 fixed. Through analysis, calculations can be drawn as follows.

- As the heat source temperature increases, CC and COP increase. However, chilled water outlet is decreased.

- For large values of thermal conductance of sorption element, condenser and evaporator, the system gives better performance.

- COP increases as the total time cycle increases. One interesting investigation is, CC increases as time cycle increases. But it happens only for comparatively low heat source temperature. As the heat source temperature increases the trends become opposite.

- COP and CC give optimum result as the cooling water temperature is very low (approximately 20°C).

- The improvement ratio in cooling capacity however increases with the increase of heat source temperature as well as the thermal conductance of evaporator.

- Comparing with the performance of the study by Farid S.K. [49] the cooling capacity has improved by 29% while the cooling water temperature is at 30 °C.
CHAPTER NINE

Suggestions for future investigations

In this analysis, the system performance is investigated for different phase of parameters. The present studies may be extended further. Some suggestions for future investigations are listed below:

- In the present study, the effect of thermal conductance of two-stage adsorption chiller with re-heat scheme with different mass allocation have been investigated numerically. It may be extended as the number of bed and stages are enhanced.

- In this analysis, silica gel-water are considered as adsorbent-refrigerant pairs. New adsorbent – refrigerant pairs such as SiO2 / CaCl2 can be added so that the problem may be extended.

- In this investigation, the flow rate of hot water, cool water and chilled water are kept fixed. If the flow rate is changed then further extension of this work is possible.

- The work may be extended by changing both silica gel and bed sizes.
REFERENCES


