Modeling and Analysis of Solar Assisted Adsorption Cooling System using TRNSYS

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Thesis Supervisor:

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Declaration

I certify that this research work titled *"Modeling and Analysis of Solar Assisted Adsorption Cooling System using TRNSYS"* is my own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources it has been properly acknowledged / referred.

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To my beloved parents, teachers and siblings…

Abstract

Adsorption chillers are a favorable choice for solar based cooling system because of driving lower range of 50-85°C temperatures and therefore simple inexpensive flat plate solar collectors (FPC's) can also furnish the desired energy. In this study three system configurations are analyzed to achieve 13 TR cooling demand load for an Islamabad (33.71° N, 73.06° E) located building for office. In configuration-I (C-I) returning fluid from adsorption chiller moves towards the hot water storage tank (HWST) connected to solar thermal collector whereas in configuration-II (C-II) returning fluid from adsorption chiller may be diverted from collector-HWST loop if temperature of water in HWST is less than the required temperature (i.e. 85°C). In third configuration C-III, HWST is not used. Therefore hot water from the outlet of solar collector directly enters in the auxilary heater (which will be switched on if required else off) and return water from adsorption chiller will become inlet of solar collector. The three system configurations are modeled and simulated in TRNSYS for the whole summer season to investigate the optimum collector tilt, least collector area for maximum solar fraction (SF), fractional primary energy savings (f_{PES}) and monthly solar collector's thermal efficiency. Simulation results showed that C-I gives comparatively higher collector thermal efficiency and f_{PES}. For configurations C-I and C-II, the minimum required collector's area for FPC is estimated to be 250 m² which is reduced to 150 m² for evacuated tube collector (ETC) and corresponding to those areas optimized size of HWST is estimated to be 10000 and 4500 liters for FPC and ETC respectively. For both collectors maximum seasonal solar fraction is obtained at a collector tilt of approximately 13°. C-III gives comparatively higher solar collector's thermal efficiency as compare to C-I and C-II but C-III has comparatively low values of SF and f_{PES} as compare to configuration-II and configuration-I for same FPC and ETC areas of 250 m² and 150 m² respectively.

Key Words: *Adsorption chiller; Flat plate collector; Evacuated tube collector, Solar Fraction; Fractional Primary Energy Savings; Tilt Angle; Collector Efficiency*

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Nomenclature

Abbreviations

C	:	Configuration
COP	:	Coefficient of performance
FPC	:	Flat Plate Collector
PTC	:	Parabolic trough collector
ETC	:	Evacuated Tube Collector
STPP	:	Solar Thermal Power Plant
FPES	:	Fractional primary energy savings
HVAC	:	Heating, ventilation, air conditioning and Refrigeration
SF	:	Solar Fraction

Nomenclature

List of Symbols

A_i	m^2	Surface area of the ith tank
A_c		Area of Evacuated or Flat plate type solar collector
a_1		First order coefficient
a_2	$\mathrm{kJhr^{-1}m^{-2}K^{-1}}$	Second order coefficient
a_o	$\mathrm{kJ\ hr^{-2}m^{-2}K^{-2}}$	Optical efficiency
ε_{heat}		Efficiency of boiler
f_{PES}		Fractional primary energy savings
I_T	$\mathrm{kJhr^{-1}m^{-2}}$	Total radiation on tilted surface
G_T	$\mathrm{kJhr^{-1}m^{-2}}$	Incident global solar radiation
T_{amb}	$^{\circ}\mathrm{C}$	Ambient Temperature
T_{ads}		Temperature in adsorber
T_{gen}		Regeneration temperature
T_{env}		External environment temperature
T_i		i_{th} node temperature
T_{in}		Collector inlet water temperature
$\eta\emptyset$		Latitude
$\eta\emptyset$		Efficiency of solar collector
δ		Declination

CHAPTER 1

Introduction to Solar Based Cooling Technologies

1.1 Overview

Currently most of the research is being emphasized in developing technologies which can contribute to reduction in energy cost, energy consumption and peak energy demand while providing the desired level of thermal comport conditions [1,2].

Major portion of electrical energy is consumed by traditional vapor compression refrigeration system and refrigerants charged in these systems also cause problem greenhouse gas emissions [3].

As compare to above mentioned traditional cooling systems, solar thermal cooling offers several advantages [4].Cooling machines driven by solar thermal heat have no harmful effects on environment (as in the case of conventional vapor compression systems) and moment of highest cooling demand coincides the moment when solar radiation is available. Figure 1.1 shows the l solar radiation (global) (in kWh/m^2/y) at the surface of earth over the year. 1200 kWh/m^2 per year is the average energy received in Europe. For Middle East this value is 1800 kWh/m^2 to 2300 kWh/m^2 per year [5].

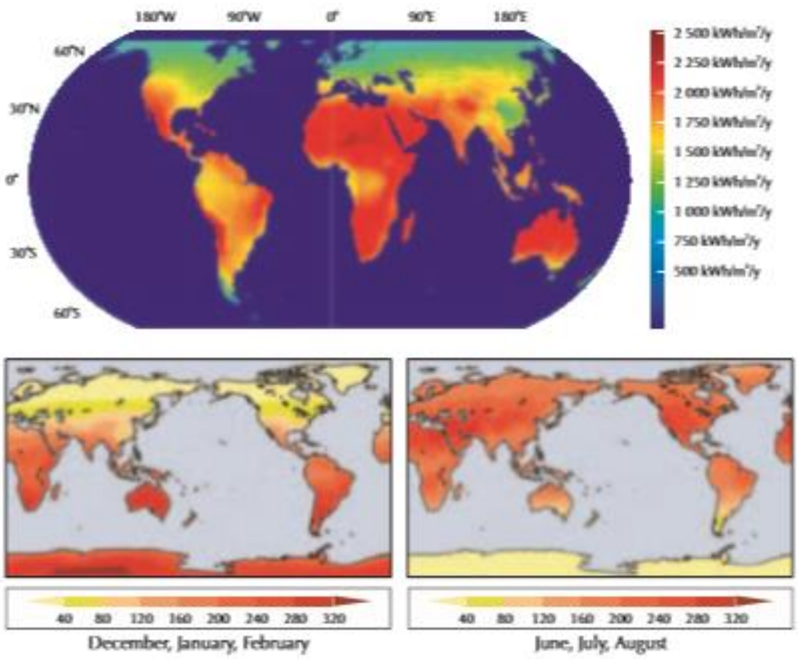


Figure 1.1 solar flux (global) (in kWh/m^2/y) over the year at the surface of earth (bottom) summer and top (winter) [5]

1.1.1 History of Refrigeration and Air Conditioning

Safe and comfortable living conditions are always desired by human race. Development of air conditioning and refrigeration has a vital role in fulfilling this goal. Without year-round control of the indoor environment industrialization of the United States would have never been possible. The history of air conditioning and refrigeration is very older although most of the United States population experienced a conditioned environment in the mid-to-late 20th century. Romans were pioneers of central heating who used double floors through whose cavity the fumes of a fire were passed. They were also the first to use window glazing. There are some evidences that the Jews, Chinese, Persians and Indians understood concepts of air conditioning centuries earlier, but William Cullen [6] brought forward the first published public explanation of phenomenon of refrigeration in 1756. Dr. John Gorrie carried out the initial efforts at building an air conditioner. In 1830s Dr. Gorrie established a refrigeration machine which through air over a ice for producing cooling for hospital rooms. A. R. Wolf established comfort air conditioning in (1859-1909) for more than one hundred buildings. Willis H. Carrier (1876-1950) was an American engineer who developed in 1902 the first modern air conditioning systems. From 1950 to 1970 when the modern HVAC industry was developing, the cost of electricity was very low and the concepts such as global warming and ozone depletion were relatively rarefied. Hence, very little importance was given to other modes of air conditioning. The emphasis on alternative forms of air conditioning is gaining strength due to the two most important challenges facing conventional systems of air conditioning related to environmental and energy.

1.1.2 Environmental Effects

There are two ways by which modern air conditioning machines affect the environment. The first is due to the emissions of hydrochlorofluorocarbons (HCFCs) and chlorofluorocarbons (CFCs) which are used as refrigerants in these machines. Molina and Rowland [10] investigated that chlorine atoms in HFCs and CFCs are causing the breakdown of the ozone layer in the stratosphere which acts as a shield for earth from ultraviolet-B solar radiation which causes cancer. By 1985, scientists noticed a drastic thinning of the ozone layer over Antarctica. On September 16, 1987, leaders from 24 nations signed the Montreal Protocol [11]. Since then, new scientific proofs of the urgency of ozone damage have led all 196 members of the United Nations to ratify the treaty. The CFCs production is already stopped and less active HCFCs is expected to be completely stopped by 2030. Global warming is the second environmental concern due to conventional air conditioning. Most of the refrigerants used in Vapor Compression Refrigeration (VCR) machines have very high Global Warming

Potential (GWP). For instance, HFC-134 a, which is one of the most widely used refrigerant blend, has GWP equivalent to 1320 times of CO_2. Besides releasing greenhouse gases directly into the atmosphere through equipment maintenance and leaks, most of the electricity produced to run these machines also comes from fossil fuels burning (which is a significant source of CO_2 emission). Greenhouse gases are believed to contribute towards an increase in the observed average earth atmosphere temperature [12], which results in higher cooling demand. Manufacturing air conditioning systems based on natural refrigerants which are ozone friendly and have lower GWP would benefit goals for environmental progress.

1.1.3 Crisis of Energy and Peak Load

The conventional refrigeration machines are driven by electricity. Study shows that from total energy consumed by commercial buildings and residences, 45% is used for air-conditioning [1,2]. For the last few years, Pakistan is facing severe energy crisis. In peak summer as the cooling load increases, crisis becomes worse. Ongoing power crisis, availability of abundant solar radiation and absence of solar assisted cooling system in Pakistan indicated that space for research is available to check potential of solar energy to driving cooling machines in Pakistan.

Solar based air-conditioning is one of the most attractive ways to solve both environmental and energy issues as mentioned above. As most of the heat load of the building is due to absorbed solar radiation, the moment of maximum requirement of cooling roughly matches with the moment of day with highest incident solar radiation. Also solar based refrigeration system is capable to meet the requirement for cooling, medical or preservation of food in remote areas. Using solar based cooling systems will reduce the burning of fossil fuel by reducing the required electricity generation hence reducing greenhouse gases in the atmosphere. Solar thermal systems can ozone friendly refrigerants having zero or very small GWP.

1.2 Solar Based Air Conditioning Systems

Researchers have been investigating several solar based refrigeration and air conditioning systems throughout the world [13, 14, 15, 16, 17, 18, 19]. These systems can fall into two different categories i.e. solar electric and solar thermal. The next section provides a brief introduction on solar based thermal and solar based electric refrigeration systems.

1.2.1 Solar Electric Air Conditioning

Air conditioning system based on solar electric mainly consists of PV panels and an air conditioning device powered by electricity. Several kinds of solar electric air conditioning systems have been investigated to-date, for example thermoelectric, photovoltaic powered vapor compression, and Stirling refrigerator [20]. Each of the above mentioned systems has their advantages and disadvantages but the commonly used air conditioning system is the PV based VCRS. The vapor VCRS needs electricity to run the compressor driven by the solar photovoltaic (PV) panels. Figure 1.2 shows a schematic diagram of a solar PV vapor compression system. Manufacturing of solar cells has seen an extraordinary maturity in recent times because of renewable energy increasing demand. Materials currently used for solar cells include poly-crystalline silicon, mono-crystalline silicon, amorphous silicon, cadmium telluride, etc. Silicon is the most widely used solar cell material. The efficiency and cost of solar cells change widely depending on the material from which they are made and manufacturing methods. Solar panel efficiency can be up to 24.7% for mono-crystalline silicon technology and as low as 9.5% for amorphous silicon technology [21]. The biggest benefit of using solar panels for air conditioning is the high overall efficiency and matured technology when coupled with a traditional VCS system.

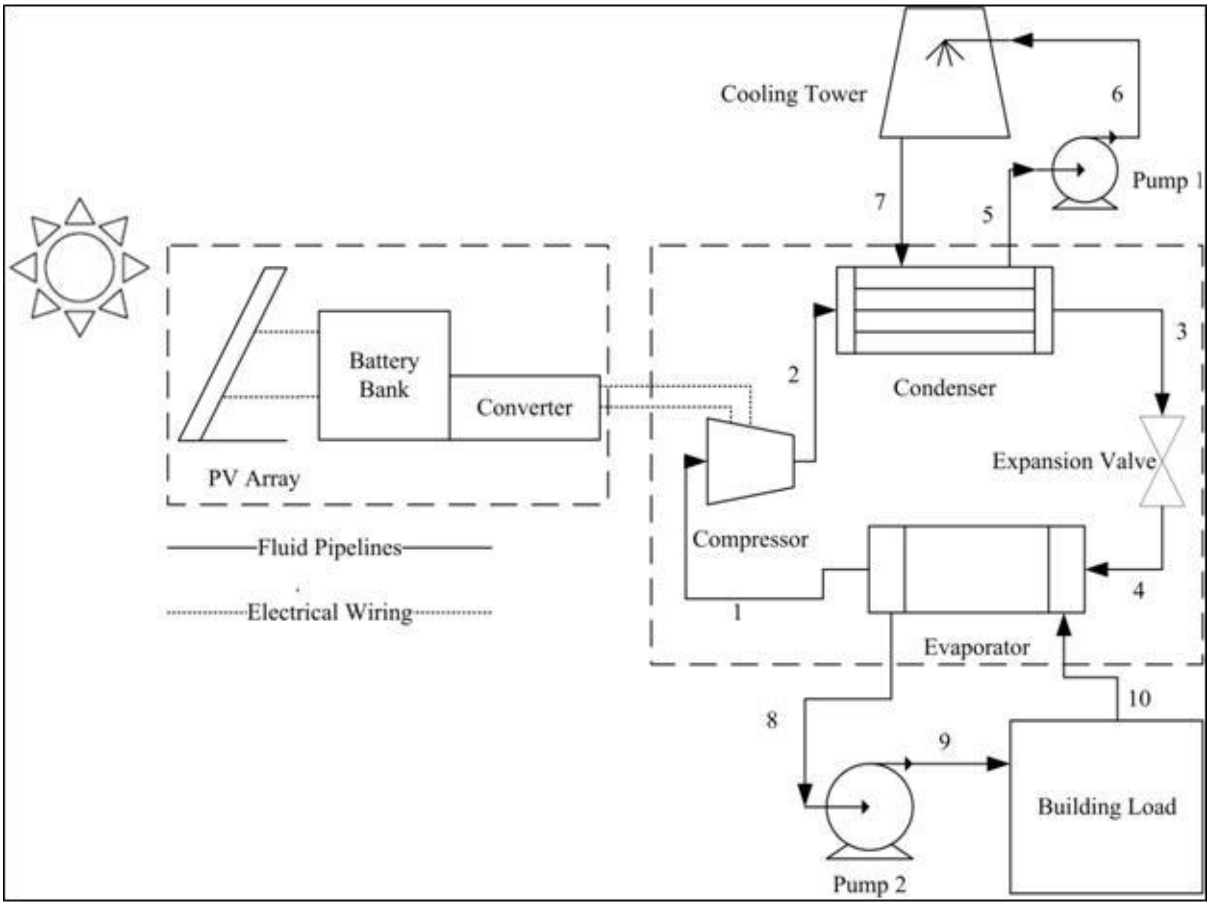


Figure 1.2 Solar PV Based VCRS

The Performance Cofficient (COP) of the vapor compression machines in solar PV vapor compression systems lies in the range of (1 to 3.4) between 45 and 61 °C condenser temperatures .

1.2.2 Solar Thermal Refrigeration Systems

Instead of solar electric energy, solar thermal energy in these systems is employed for effect of cooling.. Generally flat plate collectors are used to provide the required heat. Usually two types of models have been discussed; sorption systems and thermal-mechanical systems (Stirling and Rankine engines). For domestic applications, the only feasible choice is the sorption system because thermo-mechanical systems require very high temperatures to power them. Different solar based sorption systems have been investigated during recent years, including **absorption, adsorption, and desiccant** [2, 22, 23, 24]. Figure 1.3 shows cooling system based on solar thermal. To heat the water which is stored in HWST heat absorbed by solar collector is used. The hot water is then used to drive the chiller (absorption/ adsorption). In case of non sufficient solar radiation is not available (in case of cloudy weather), auxiliary device (which can be electric or gas fired boiler) is switched on to achieve required temperature. The chilled water from chiller is used to provide cooling to load. Hot water can also be used to provide heating in winter as well as domestic applications.

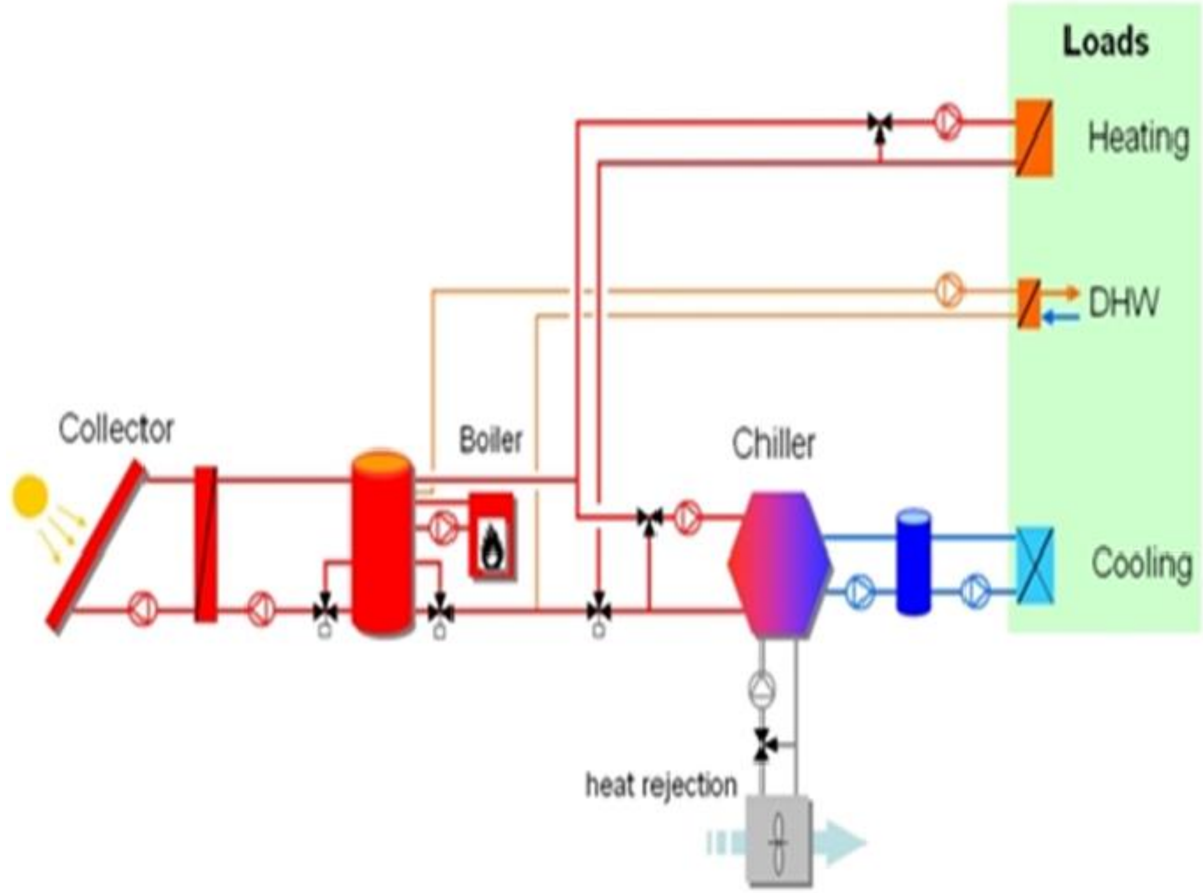


Figure 1.3 Solar Thermal Cooling System [25]

Figure 1.4 shows total number of solar thermal cooling installations in Europe and world till 2013. More than half of total solar thermal cooling systems are installed in Europe where

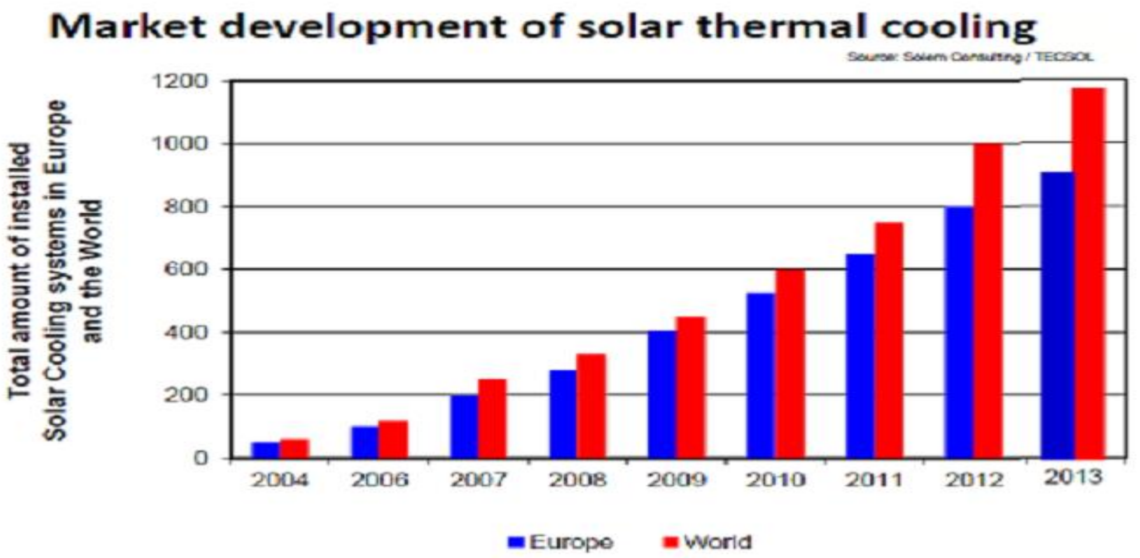


Figure 1.4 solar thermal cooling installations in Europe and world till 2013 [26]

available solar radiation is less then several other countries of world e.g. (Africa, Asia, and Australia etc.).

The main differences between adsorption and absorption is the nature of the sorbent and cycle time, which for adsorption is significantly longer [2]. Figure 1.5 shows market share of different solar thermal cooling technologies across world. 71% of the installed solar cooling systems are absorption based which indicates that a lot of work has been performed in this area. Other technologies include Desiccant solid, Desiccant liquid and adsorption system with share of 14%, 2% and 13% respectively which shows room for research is still available in these technologies as solar thermal cooling systems based on these technologies are still under research and development.

Solar assisted sorption cooling system is an attractive alternative that not only conserves energy but also fulfill the needs for air-conditioning and refrigeration. These technologies are also environment friendly. However, consistent efforts are being carried out for commercial industrial applications and to replace traditional refrigeration machines.

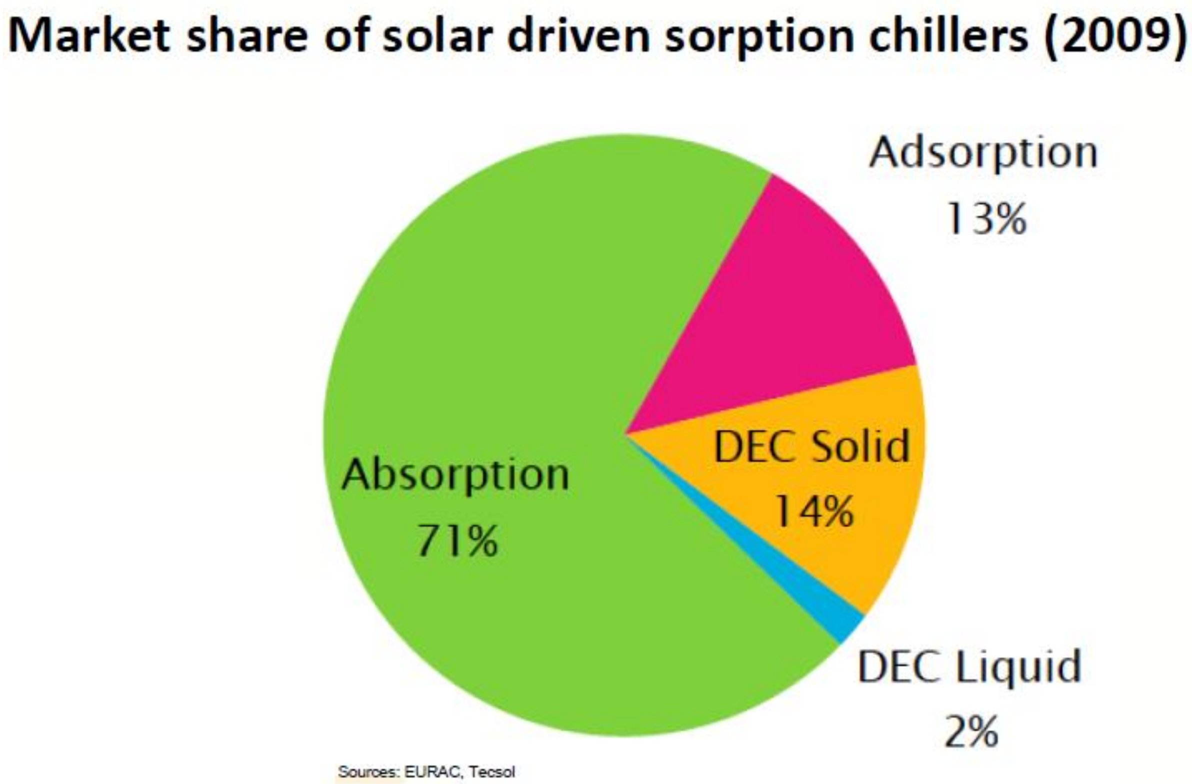


Figure 1.5 Market share of different solar thermal cooling technologies across world [26]

1.2.3 Solar Assisted Absorption Cooling System

One of the oldest cooling technologies is adsorption cooling. In this process the adsorber is to be cooled. The pressure of absorbent/refrigerant mixture increases by solution pump to high pressure level. As compare of compressor of vapor compression system, power consumed by solution pump of vapor absorption system is relatively small. As absorbent is less volatile than the refrigerant therefore when sufficient amount of heat is provided in the generator [28] it is separated from the solution. The absorber-generator-solution pump combination acts as a thermal compressor and hence eliminates the need for an electric driven mechanical compressor [29]. Electrical energy consumption by the liquid pump is a small fraction of the energy consumed by an electric driven mechanical compressor due to specific density difference of vapor and liquid medium. In a solar assisted vapor absorption cycle heat required in the generator is supplied by collector. Figure 1.6 shows absorption chiller connected with solar collector and HWST. As water flows through solar collector, it becomes hot. HWST is used to store this heat. From HWST, circulated through generator of an absorption chiller is hot water to separate the refrigerant from absorbent [30].

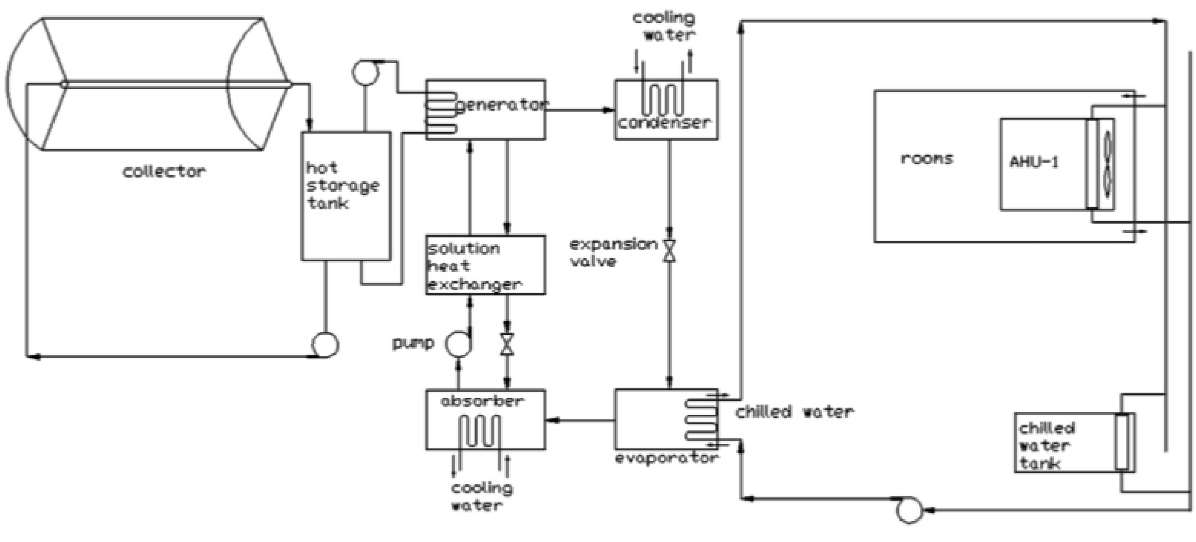


Figure 1.6 Absorption chiller connected with solar collector and hot water storage tank [30]

1.2.4 Drawbacks of Vapor Absorption Cooling System

1. Since ammonia is toxic and harmful, therefore it is very unsafe to use in residential systems.

2. As compared to adsorption and desiccant systems the driving temperature in vapor absorption system is relatively high. Single-effect system requires temperature range of 80 – 100 °C; the double-effect works between 100 – 160 °C; and triple-effect works above 160 °C [31]. Use of double or triple-effect systems is not feasible, since most of the domestic systems are expected to apply flat-plate solar collectors due to the higher price of concentrated collectors.
3. Crystallization problem has always been challenging for refrigeration and air conditioning engineers while working with absorption chillers [32].

1.2.5 Solar Assisted Adsorption Cooling System

The adsorption cooling machine use sorption material in solid form such as zeolite and silica gel for production of cooling effect. This technology is becoming more popular because low regeneration temperatures are required. The advantages of adsorption system are:

1. **Wide Operating Temperatures Range**: 50°C temperature (122°F) is sufficient to activate adsorption systems, whereas at least 90°C (194°F) is required for an absorption system.
2. **No Crystallization Issue:** Since the adsorbent remains in a solid state, in adsorption systems, which means no issues of crystallization.
3. **Suitable for application where serious vibration occurs:** This is the most important application of adsorption.

Detailed discussion on adsorption cooling process along with its combination with solar collectors is presented in chapter 2.

1.2.6 Desiccant System

Desiccant cooling is called as open-cycle based sorption cooling because to dehumidify air it uses sorbent. A desiccant system activated thermally, combines evaporative cooling with dehumidification of air by a desiccant. Both solid or liquid can be used for this purpose. The sorption materials consist of either lithium-chloride or silica gel in the form of desiccant wheel continuously rotating [33]. Figure 1.7 presents a diagram of the desiccant based cooling system. Three basic components in this system are namely the system for cooling, the heat source for regeneration and the dehumidifier. Composition of each of the three components and the possible configurations can change based on type of desiccant used [33].

Desiccant dehumidifier in case of solid desiccant systems is typically a desiccant wheel with slow rotation or a adsorbent bed which is regenerated repeatedly. The desiccant wheel removes the latent load whereas the sensible load is taken by cooling unit. The thermal energy required for removing the moisture adsorbed by desiccant during the sorption phase is supplied by regeneration heat source. A cold coil, an evaporative cooler or evaporator of a traditional VCRS can serve the purpose of cooling system [34].

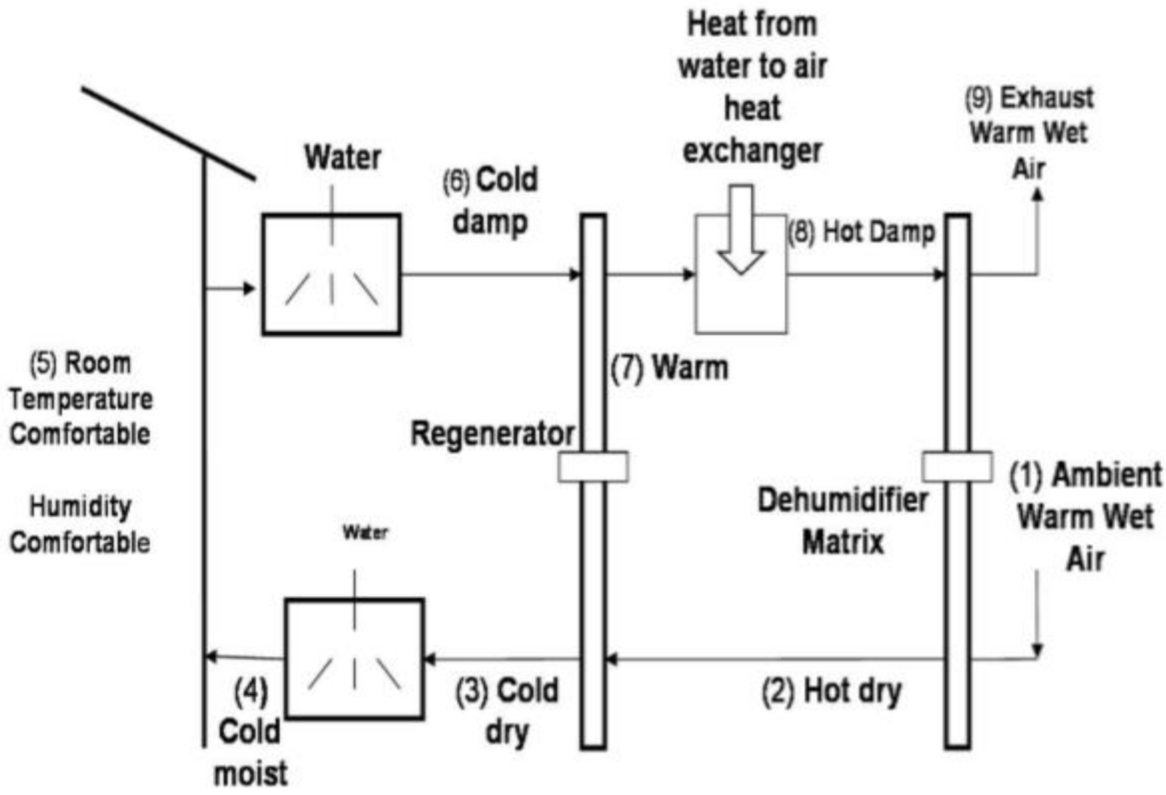


Figure 1.7 Desiccant cooling system [35]

1.2.7 Limitations of Desiccant System

Desiccant cooling systems have a limitation in that they cannot cool supply air below 20 °C) at peak design temperatures especially in hot and humid conditions, therefore a relatively high volume of supply air is required to maintain a design space temperature of 24 °C. Therefore desiccant air systems are relatively bigger and bulkier in size and are also not appropriate where cooling space does not require a large quantity of ventilation air [36] and available space is limited.

Table 1 shows comparison of different parameters of absorption and adsorption cooling system. From table it is clear that temperature of hot water required to drive this system is low. This temperature range can be achieved by simple flat plate collectors.

Table 1 Comparison of Absorption and Adsorption Cooling System [37]

Parameter	**Absorption**			**Adsorption**
	Single Effect	**Single Effect**	**Double Effect**	**Single Effect**
Refrigerant	Water	Ammonia	Water	Water
Sorbent	LiBr	Water	LiBr	Silica Gel
Cooling Medium	Water	Water-glycol	Water	Water
Temperature °C Hot Water	80-95	75-100	140-190	55-95
Temperature °C Cooling Water	20-40	20-40	20-40	20-45
Temperature °C Chilled Water	20-Jun	-50	20-Jun	20-Jun
COP	0.6-0.8	0.5-0.7	1.1-1.35	0.5-0.65

Chapter 2

Review of Adsorption Based Cooling System

2.1 Introduction

Process of adsorption between a solid adsorbent and a refrigerant is the cause of adsorption air conditioning system as shown in Figure 2.1.As heat supplied to the adsorber increases the pressure inside the adsorber, the refrigerant vapor is released. Vapor of refrigerant flows in to the condenser and is condensed, as soon as the adsorber pressure becomes equal to the condenser pressure. In a vessel which is called as "adsorber" the vapor coming out from the evaporator gets adsorbed by the adsorbent. Between condenser and evaporator same process undergoes.

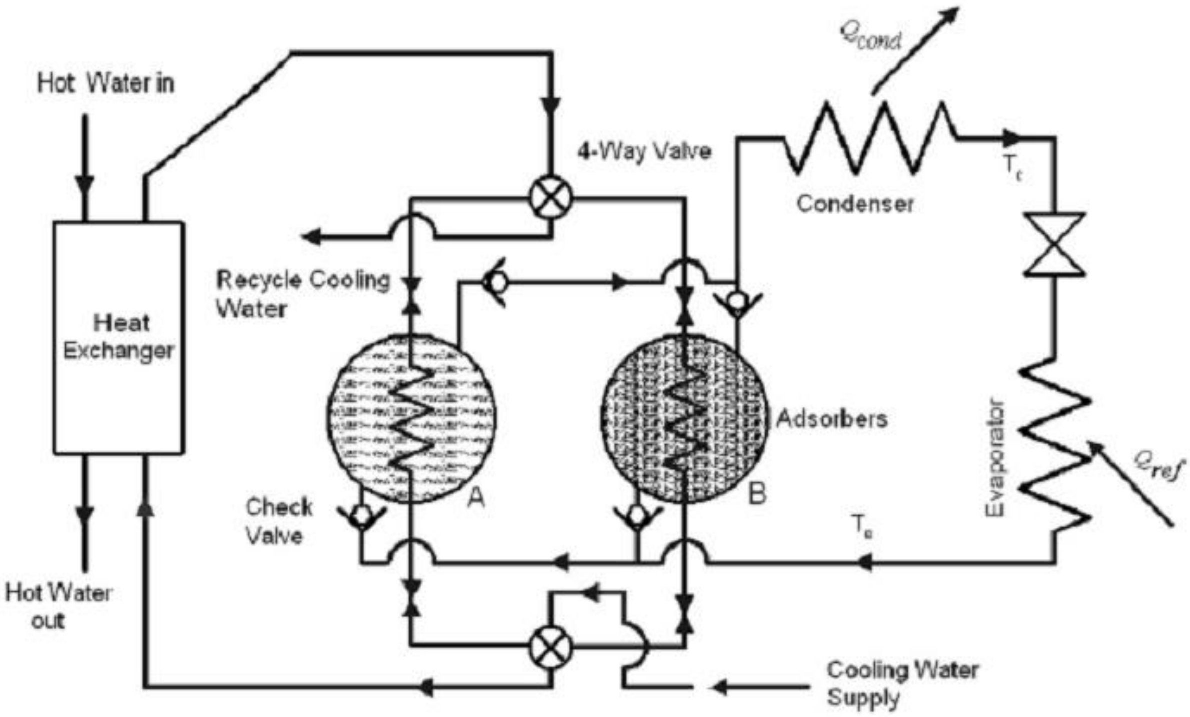


Figure 2.1 Schematic of Adsorption Cooling System [38]

2.1.1 Adsorption Process

Solid porous medium (the adsorbent) is the requirement of process of adsorption and gas/liquid molecules (the adsorbate) so as to occupy the volume of micropore [39]. The attraction forces from a solid can be of two main kinds, chemical and physical, and they give rise to chemical adsorption or "chemisorption" or physical (or "van der Waals") adsorption or ''physisorption'' respectively. As compared to physisorption, the bonding forces of chemisorption are much greater, hence more heat is required to desorb the molecules. The chemical bonding is irreversible as the adsorbed compound undergoes change in their chemical form. By applying heat, as shown in Figure 2.2, therefore the process is reversible. Because of this particular reason, physisorption is involved in most thermal systems which are cyclic in nature [40].

2.1.2 Heat of adsorption

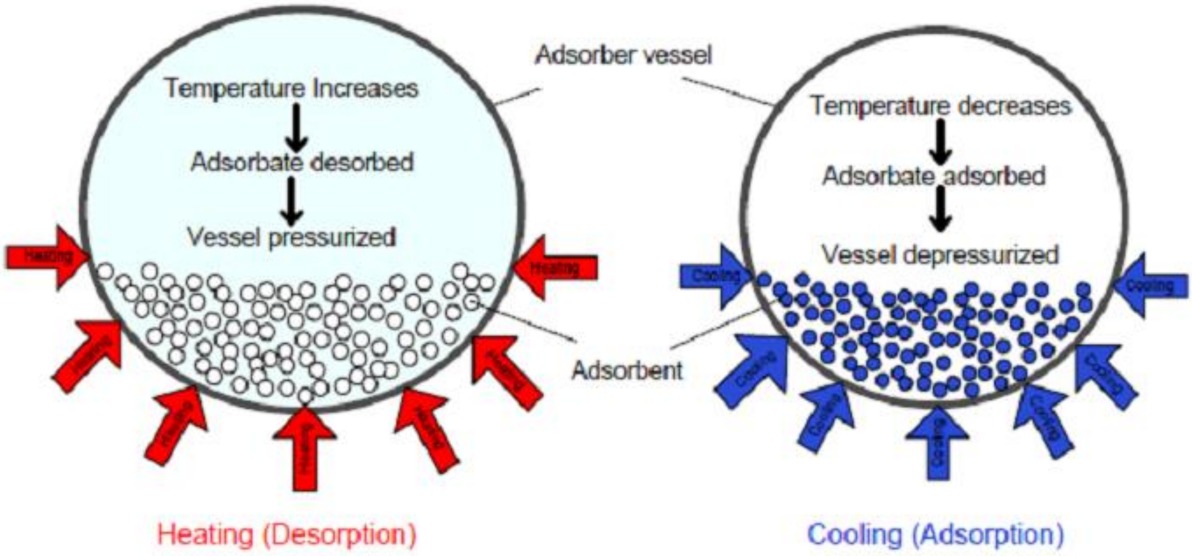


Figure 2.2 Processes of Adsorption and Desorption [40]

As compare to condensation to the liquid phase, adsorption is stronger. Ratio of the infinitesimal variation in the enthalpy of adsorbate to that in the amount adsorbed is termed as Isosteric adsorption heat. Adsorbate latent heat and the strength of force between the adsorbate and adsorbent are causes on which heat quantity which is released depends. Process of adsorption heat is liberated (an exothermic process), can also be termed as 'Isosteric adsorption heat'. Refrigeration and air conditioning involves this law to have a cooling effect. As compare to the adsorbate condensation ,the adsorption heat is normally higher [41.42].

2.1.3 Adsorbate and Adsorbent Working Pairs

The selection of a proper adsorbate (referred to as the refrigerant in cooling systems) and adsorbent pair for an adsorption refrigeration system is a complex process and depends on several factors. There are several working pairs to choose from like water/ silica gel, water/ activated carbon, , zeolite/water [43].

The adsorbent must have following properties:,

- Chemical compatibility to the refrigerant selected
- Higher thermal conductivity
- increases density of cooling

Desirable heat transfer and the thermodynamic properties in the adsorbate (refrigerant) are as follows:

- High thermal conductivity and good thermal stability
- Non-inflammable, chemically stable, non-corrosive and non-toxic
- At normal operating temperatures, lower saturation pressure
- Dimensions of molecules must be small to undergo easy adsorption
- Low specific heat, low viscosity

2.1.4 Adsorption Cycle

The main parts of this system are labeled as A and B in Figure 2.2, at a time one bottle is online and other bottle is on desorption or we can say that at any certain instant of time, one bottle is being cooled, while the other is being heated. The refrigerant flow between the evaporator and condenser is similar to the traditional VCRS. The basic adsorption cooling is shown in Figure 2.1. Figure 2.3 represents fundamental adsorption process Clapeyron diagram, showing the change of temperature and pressure in the adsorber.

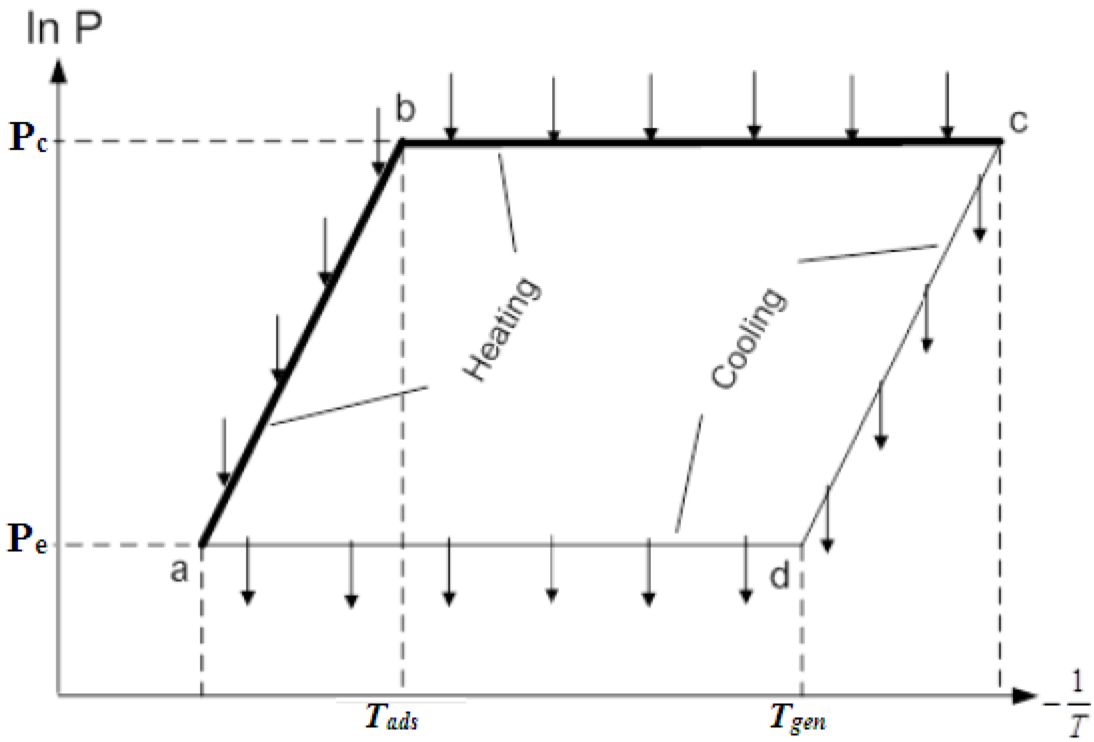


Figure 2.3 Adsorption cooling cycle clapeyron diagram

T-s diagram for the ideal cycle is shown in figure 2.4 for standard adsorption cycle on. The thermodynamic cycle consists of following seven processes:

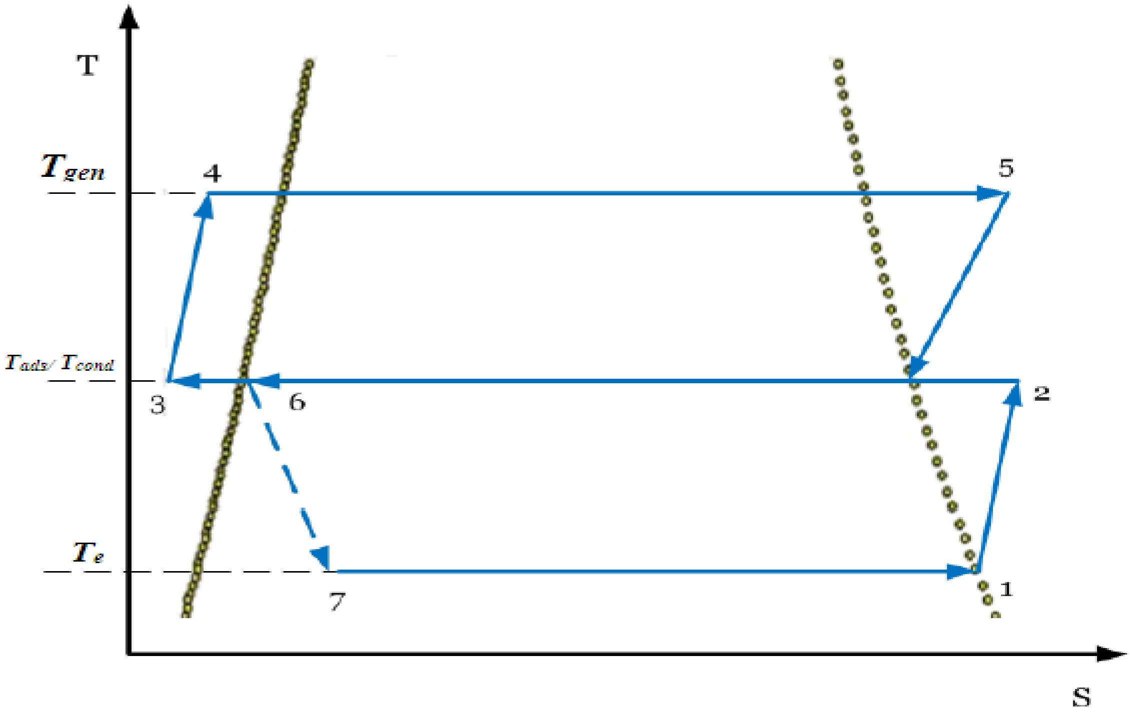


Figure 2.4 T-s diagram for standard adsorption cooling cycle

2.2 Solar Based Adsorption Cooling System

Traditional fundamental single stage adsorption system has two bottles, an evaporator, a condenser, heater and a cooling tower. Douss *et al.* [44] first described two bed basic cycle. Chiller in solar based adsorption cooling system is connected with a series of(flat plate/evacuated tube) as a heat source. In this system water is used as refrigerant. Fig 2.5 shows solar based adsorption cooling system.

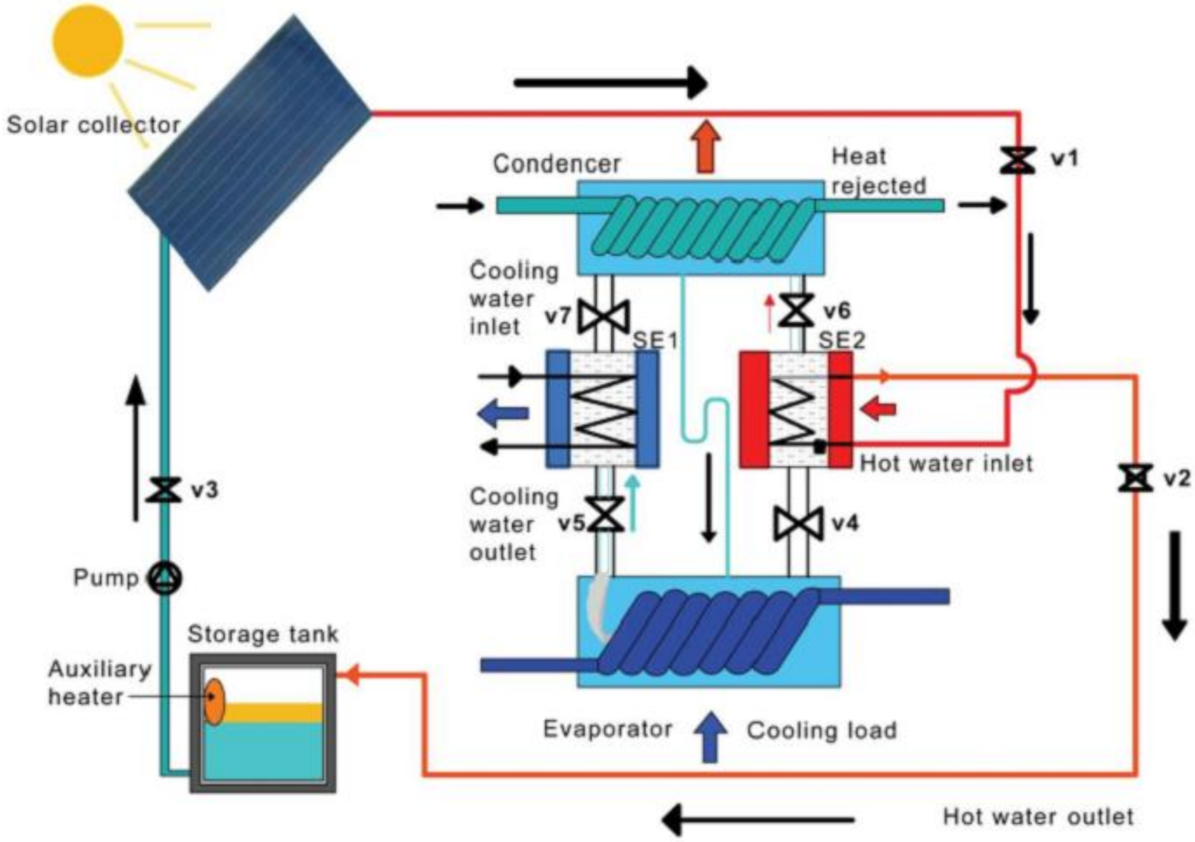


Figure 2.5 solar coupled adsorption refrigeration system [45]

Table 2 Different modes of SE2 and SE1 during a complete adsorption cycle

Cycle	Mode	Pre-Cool	Evaporation/ Adsorption	Advance-heat	Regeneration/ Condensation
1stCycle	1	ES1	------	ES2	------
1st Cycle	1	------	ES1	------	ES2
2nd Cycle	2	ES2	-----	ES1	-----
2nd Cycle	1	-----	ES2	-----	ES1

2.3 Literature Review of Solar Based Adsorption Cooling

Energy system design engineers are now more attracted towards the alternate energy resources due to increase in demand for heating and air conditioning,. In the current solar thermal based systems sorption cooling and more specifically absorption is the prevailing technology in the market. However, adsorption cooling system on the other hand can be driven at working temperatures (50 -85 °C). As compare to absorption refrigeration system this temperature range is quite lower. The above mentioned driving temperatures requirement meet by flat plate solar collectors, Choudhury et al. [46].No need of moving parts such as solution pump, hence no vibration, simple control and less maintenance required are the other advantages. Janusevicius at al. [47] analyzed the variation of performance cofficient (COP), solar fraction, and seasonal performance factor with collector tilt, and absorber area for 8.0 kW solar assisted adsorption cooling system installed at Vilnius, Lithuania. Evacuated Tube Collector (ETC) with slope of 30°, 0.5 m^3 hot water storage tank, 16m^2 absorber area gives maximum values of SF, COP and seasonal performance factor.

Zhai and Wang [48] experimentally analyzed the solar assisted adsorption chiller performance without/with heat storage for a Shanghai Research Institute of Building Science which was a green building. System without hot water storage tank was more suitable for areas having abundant solar energy. By increasing collector's area system without heat storage tank can achieve same refrigeration effect which was achieved by system in presence of hot water storage tank.

Nunez and Mehling [49] reported the results of an adsorption system of a 5.5kW capacity. In summer, adsorption chiller driven by FPC of 20 m^2 area, 2 m^3 storage tank and combined heat and power (CHP) provides cooling whereas in winter chiller served as thermal driven heat pump. Highest collector efficiency was 36.2% in the month of June whereas share of solar heat was 99.7% in the month of June.

He et al. [50] presented work on solar thermal cooling with absorption chillers of 175kW capacity driven by Chinese manufactured evacuated tube collectors for an office building in Beijing, China. With 358 m^2 of total aperture area, slope of 10° due south and using biomass boiler as an auxiliary device, 37.6% annual average collector's efficiency is achieved. 66% of consumption of fractional primary energy was reduced, similar to saving of 13.5 tons of standard coal when solar based cooling system was used compare to conventional heating and cooling system (which usually adopts boiler for district heating and chiller for cooling in China), Solar fraction achieved in winter and summer is 0.38 and 0.76, respectively.

Alam et al. [51] investigated application of solar assisted cycle possibility for weather of Tokyo, Japan. Compound parabolic collectors having total area of 36.22 m^2 were used in order to get temperature of heat source around 85°C. Results showed that by optimizing adsorption cycle time, solar collector size can be reduced.

Alam et al. [52] investigated adsorption chiller performance which was regenerated by thermal heat from solar collectors for weather conditions of Dhaka, Bangladesh. Results showed hot water storage was more efficient as compare to system in which chiller is coupled to FPC without HWST.

Hassan [53] evaluated working of continuous operation solar thermal based adsorption refrigeration system. The seasonal average solar collector thermal efficiency was found to be 50.91 %.Simulation results revealed that chiller acquired a COP of 0.402.

For a solar adsorption refrigeration system Luo et al. [54] optimized heat source temperature for maximum COP and investigated that its coefficient of performance can be greater than 0.25 under moderate solar radiation and with the regeneration temperature (optimized) by using evacuated tube solar thermal collector. Results also showed for a range of heat source temperatures, high efficiency of an evacuated tube collector is achieved if radiation was more than 700 W/m^2.

Wang et al. [55] presented achievements in solid sorption refrigeration technology for ice making and space cooling applications. The chiller could be driven by (CCHP) or solar energy. By using latest solar powered adsorption ice making machines COP of 0.10 to 0.15. Usually COP is in between 0.2 to 0.6 but COP of 0.7 is achieved in some commercially produced adsorption machines.

Wang et al. [56] experimentally analyzed adsorption refrigeration chiller's performance. Its application includes solar powered air conditioning and a chiller coupled with CCHP. Similarly the system gives cooling effect of 1kW and coefficient of performance of 0.3 driven by hot water of 80°C. An adsorption refrigeration chiller based on water-silica gel pair with 10 kW capacity, can be driven 60-100°C regeneration fluid, COP of 0.4 can be achieved.

Jakob et al. [57] presented developments and investigated performance of an ACS 08 thermally driven, single effect chiller based on adsorption technology with water-silica gel pair having 7.5 kW capacity. In 2008 a domestic scale adsorption cooling system ACS 08 was launched having 7.5 kW cooling capacity and in 2008, another larger system with 15 kW was launched.

Alili et al. [58] reviewed different solar thermal cooling technologies. Other than absorption and desiccant cooling technologies, they also presented summary of different adsorption cooling systems installed worldwide. Literature review showed that normalized thermal storage, average normalized collector area and system COP for the studies considered were

0.36 m^3 kW_c^{-1}, 11.17 m^2 kW_c^{-1}, and 0.4 respectively. Recent activities concentrate on the development of novel refrigerant-adsorbent working pairs, improving components of cycle and increasing efficiency of system.

2.4 Research Objectives

As cleared from literature review, very few installations of solar assisted adsorption cooling system are available all over the world which highlights the reality that more research has to be done in this field. The parameters of design for solar based adsorption cooling system are not yet standardized and properly documented. Moreover analysis of solar assisted adsorption system for different configurations is not available in the literature. Therefore there is still a need to further investigate the dynamic parameters of such systems under varying operating conditions, system configurations and control schemes. In present study, various system design parameters are optimized for a solar based adsorption system by employing three different system configurations. Dynamic simulations for the whole summer season are carried out in TRNSYS for Islamabad (33.71° N, 73.06° E) office building. The performance parameters of interest are solar fraction, solar collector thermal efficiency (η), fractional primary energy saving, f_{PES} and optimized size of storage tank.

Chapter 3

System Configurations and Modelling in TRNSYS

3.1 Introduction

In this study three system configurations of solar based adsoption cooling system are investigated. First two configurations are based on flow path from chiller to hot water storage tank. In third configuration system is modelled and analysed without using hot water storage tank. These three configurations are as follows:

3.1.1 Configuration-I

As illustrated in Figure 3.1, in this configuration water returning from adsorption system moves towards hot water storage tank connected to solar thermal collector. The auxilary boiler is connected in series with the stoarage tank to fulfill the remaining heat demand (if any) before the working fluid is fed to an adsorption chiller.

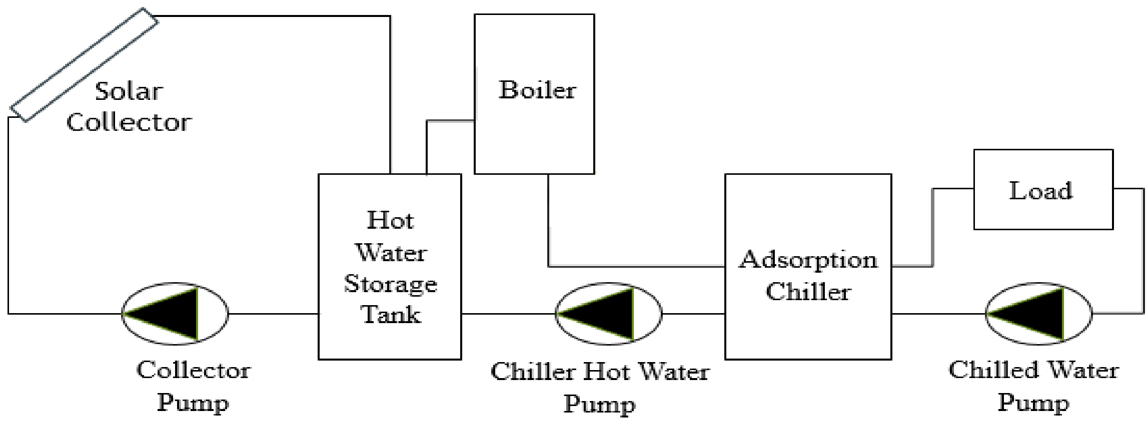


Figure 3.1 Schematic of Configuration-I

3.1.2 Configuration-II

In this configuration water returning from adsorption system can be diverted from storage tank-collector loop if temperature of water in HWST is less than 85°C. In that case system energy will be entirely furnished by the auxiliary boiler. Flow diverter is used to serve this purpose given in figure 3.2.

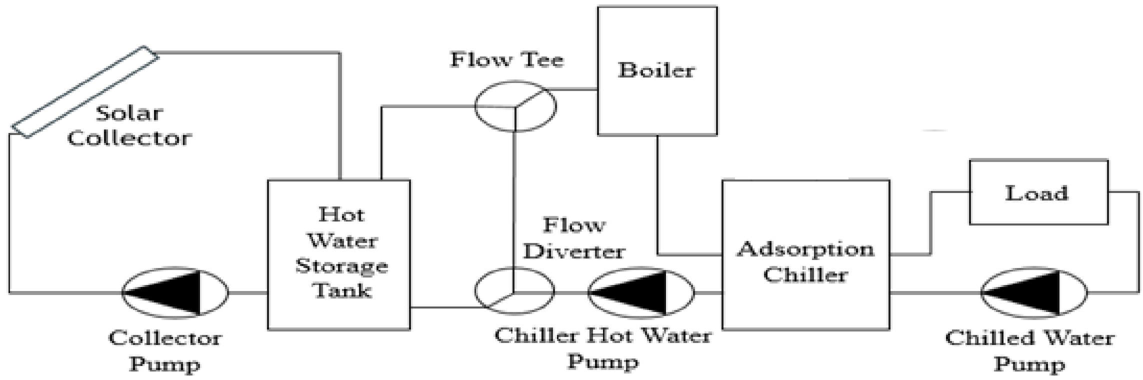


Figure 3.2 Configuration-II

3.1.3 Configuration-III

In this configuration storage tank for hot water is not used. Therefore water from the outlet of solar collector directly enters in the auxilary heater (which will be switched on if required else off) and return water from adsorption chiller will become inlet of collector as presented in figure 3.3.

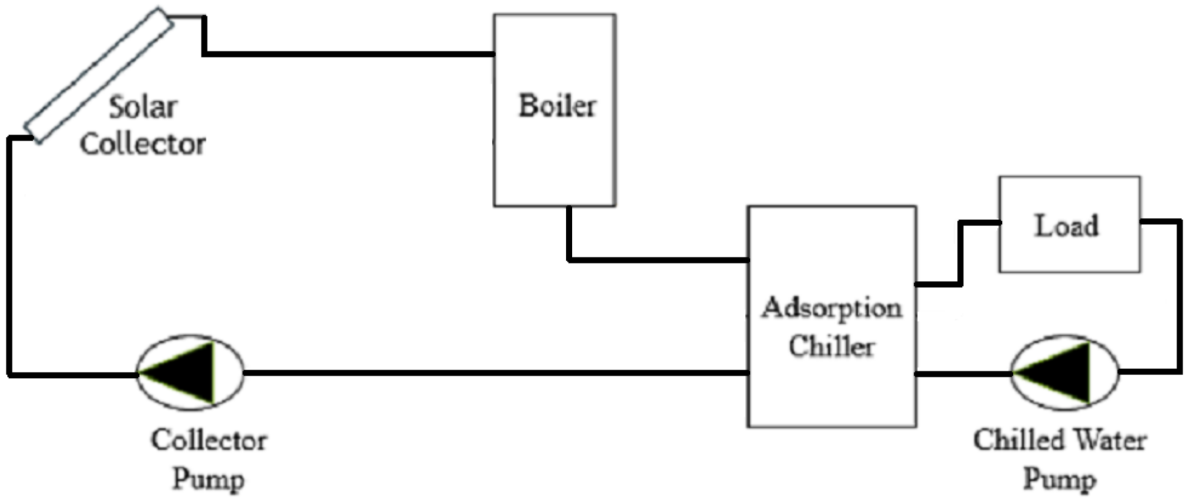


Figure 3.3 Configuration-III

3.2 Modeling in TRNSYS

To model and simulate the two system configurations, TRNSYS 17 is used. TRNSYS is a dynamic simulation tool capable of simulating the complete system performance over long period of time based on predefined characteristic equations of individual system components. Figures 3.4, 3.5 and 3.6 show the pictorial view of TRNSYS model for C-I, C-II and C-III respectively.

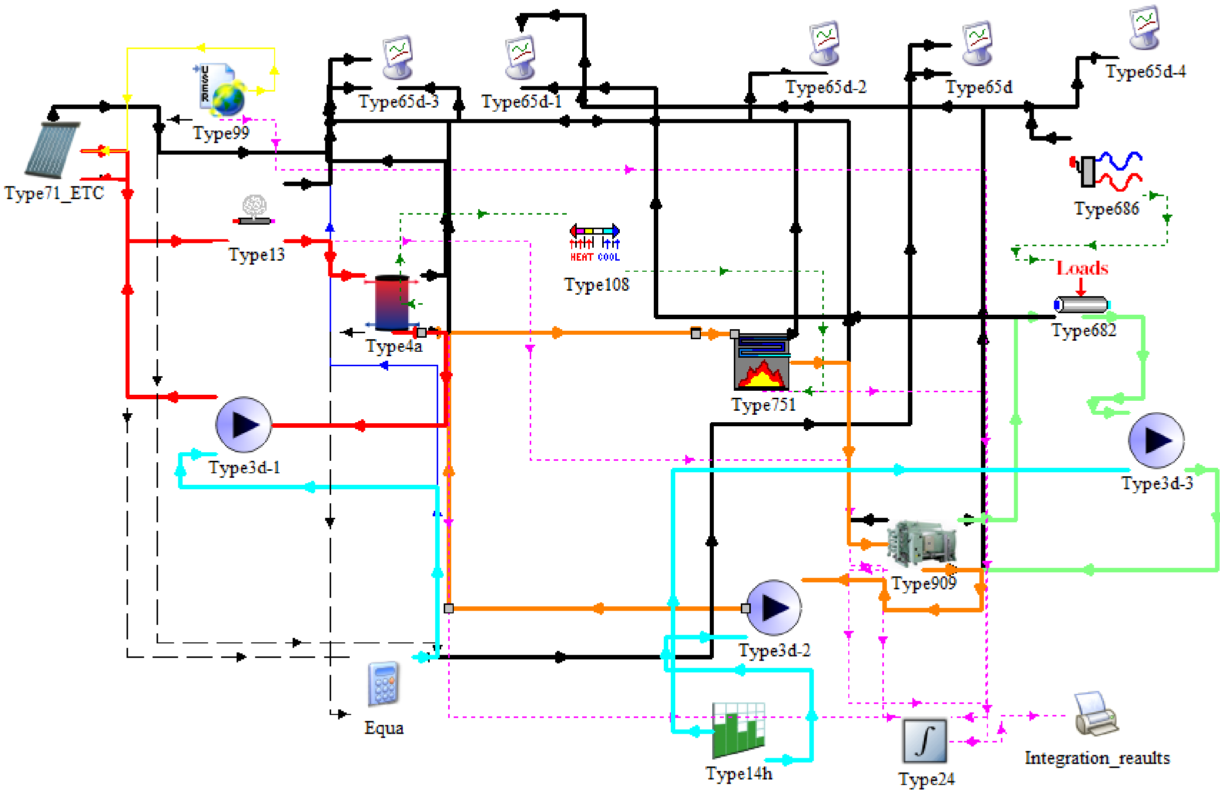


Figure 3.4 Configuration-I TRNSYS Model

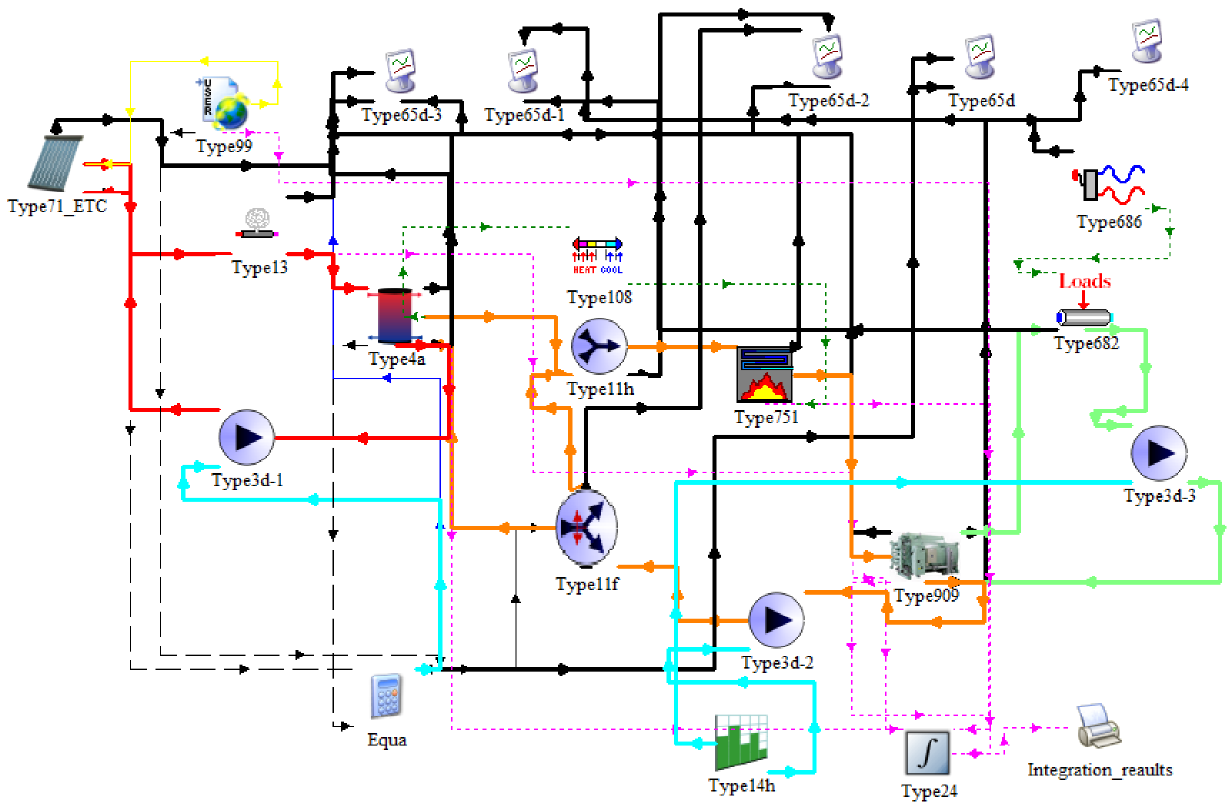


Figure 3.5 Configuration-II TRNSYS Model

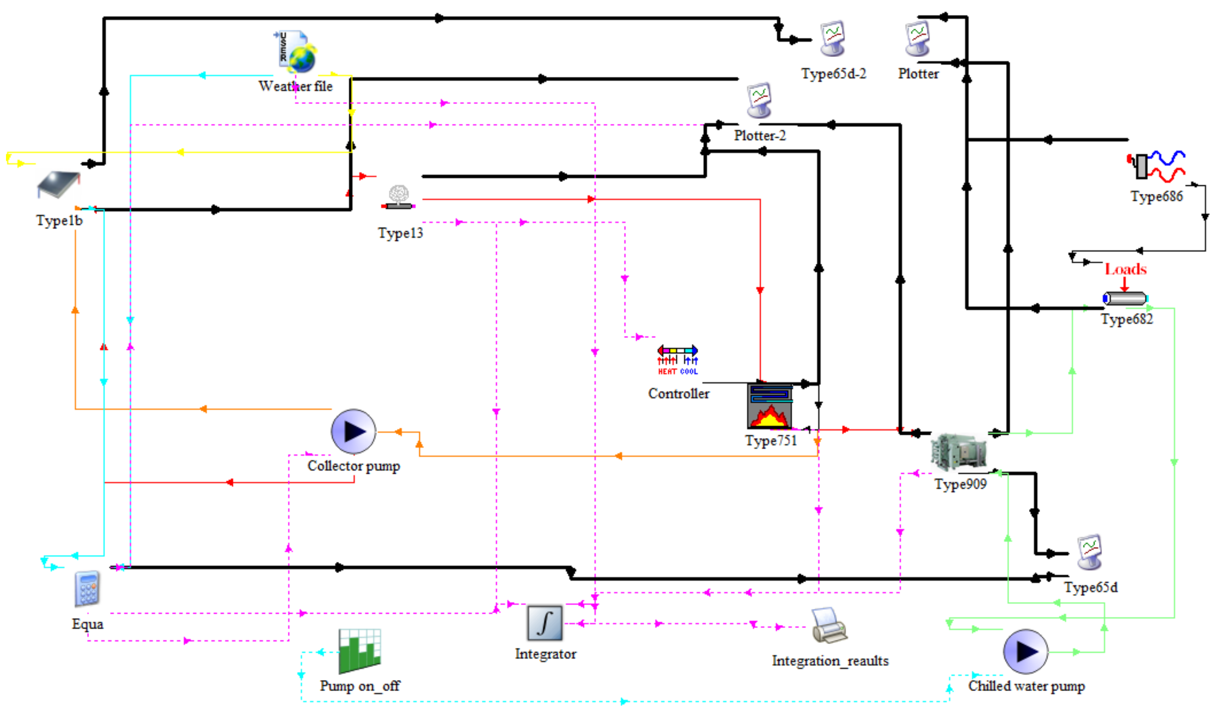


Figure 3.6 Configuration-III TRNSYS Model

3.2.1 System Components

To develop a complete solar assisted adsorption cooling system model in TRNSYS, the following components are used:

3.2.2 Solar Thermal Collectors (Type 1b and Type 71)

Flat plate collector thermal performance is analyzed by using, TRNSYS component Type 1b. Similarly, evacuated tube collector thermal performance is analyzed by using, TRNSYS component Type 71.In this regard data provided by manufacturer Apricus is used. Manufacturer also provide experimental based curves which have delta T (difference between collector enering and ambient temperature on x-axis and collector efficiency on y-axis) . Solar collector thermal efficiency is written as:

$$\eta = a_o\text{-}a_1 \frac{(T_{in}-T_{amb})}{I_T} - a_2 \frac{(T_{in}-T_{amb})^2}{I_T} \qquad (3.1)$$

where a_O is the optical efficiency, a_2 and a_1 are the negative of second and first order energy loss coefficients, T_{in} is temperature of water at inlet of solar collector and T_{amb} is the ambient air temperature.In this study values of a_O, a_1 (W/m^2.K) and a_2 (W/m^2.K^2) are taken as 0.845, 1.47 and 0.01 for ETC and 0.749, 2.770 and 0.023 for FPC, respectively. These values are taken from a manufacturer “Apricus” collector’s catalogue.

3.2.3 Hot water storage tank (Type 4a)

To model hot water storage tank this is thermally stratified TRNSYS component Type 4a is used. For this purpose multi-node method is applied in this model in which the tank is meshed into N sections and energy balances for each section are written. To express ith tank segment energy balance, following equation is used .

3.2.4 Auxiliary Heater (Type 751)

In TRNSYS, Type 751 models an auxiliary heater (gas boiler). Amount of energy required to raise the inlet water temperature to the required temperature (provided that control signal to the boiler is ON and there is a flow through the boiler) is calculated by using the following equation:

$$\dot{Q}_{need} = \dot{m}_{fluid} Cp_{fluid} (T_{st} - T_{in}) \qquad (3.3)$$

If control signal is ON and set temperature is less than inlet temperature, heater will not find a negative value of $\dot{Q}_{need}$. The energy required is limited by capacity of the boiler.

Equation used to determine the amount of fuel utilized by the boiler after calculating the required energy is given as:

$$\dot{Q}_{fuel} = \frac{\dot{Q}_{need}}{\eta_{boiler}} \quad (3.4)$$

3.2.5 Adsorption Chiller (Type 909)

Like most performance map components, Type 909 depends on TRNSYS standard external data file format. The file provides values of normalized COP ratio and normalized capacity ratio for a number of cooling water, hot water temperatures, and chilled water inlet temperatures. The capacity ratio can be defined as the device's cooling capacity at current inlet water conditions divided by the cooling capacity at rated inlet water conditions. The user provides rated cooling capacity as a parameter to the model. Similarly the COP ratio is defined as the device's COP at current operating conditions divided by the COP at rated inlet water stream conditions. The rated COP is again provided as a parameter. The structure of a data file containing rated COP multipliers and rated capacity makes the resizing of equipment simpler; in order to model a different capacity device it is sufficient to change the value of two parameters instead of generating a new performance map.

The COP and normalized capacity values are provided as a parameter of inlet chilled, hot water temperature and inlet cooling water temperature.

Type 909 linearly interpolates between cooling performance points which are based on the current inputs values.It is important to note that the component does not extrapolate beyond the data range provided. Type 909 cools the chilled water stream when it's on/off control signal input is set to a value of 1. The device operates in flow-through mode having the monitoring signal is fixed to 0 and the temperatures of the three outlet streams are set to the corresponding inlet temperatures.

The chilled water energy is calculated as:

$$\dot{q}_{chw} = MIN(\dot{q}_{capacity}, (\dot{m}_{chw} Cp_{chw}(T_{chw,in} - T_{chw,set}))) \quad (3.5)$$

In which $T_{chw,set}$ is the temperature required for the coldwater stream outlet and is provided as an input. The hot water energy required to regenerate the desiccant matrix is:

$$\dot{q}_{hw} = \frac{\dot{q}_{chw}}{COP} \tag{3.6}$$

The energy that has to be absorbed by the cooling water is then:

$$\dot{q}_{cw} = \dot{q}_{chw} + \dot{q}_{hw} + \dot{q}_{aux} \tag{3.7}$$

Where $\dot{q}_{aux}$ is the required additional heat to drive the controllers and pumps. With all of the energy terms computed, Type 909 next calculates the outlet water stream conditions:

$$T_{cw,out} = T_{cw,in} + \frac{\dot{q}_{cw}}{\dot{m}_{cw} Cp_{cw}} \tag{3.8}$$

$$T_{hw,out} = T_{hw,in} - \frac{\dot{q}_{hw}}{\dot{m}_{hw} Cp_{hw}} \tag{3.9}$$

Finally, Type 909 calculates the chiller's current COP based on:

$$COP = \frac{\dot{q}_{chw}}{\dot{q}_{aux} + \dot{q}_{hw}} \tag{3.91}$$

3.2.6 Heating and Cooling Load (Type 682)

Normally in modeling an HVAC system, the cooling and heating loads have been calculated and the task of simulation is to investigate the impacts of these loads on the system. Type 682 can be considered as an interaction point between a building load and the liquid working fluid in an HVAC system outlet temperature is calculated as:

$$T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}\, C_p} \tag{3.92}$$

3.2.7 Synthetic Building Load Generator (Type 686)

The hourly load defined in this method as:

$$\boldsymbol{Load = DesignLoad * X_{day} * X_{hour} * X_{noise,hour} * X_{noise,day}} \tag{3.93}$$

3.3 Methodology

Among the above mentioned three configurations, the first two configurations consist of 3 loops. In first loop, pump circulates hot water from solar collector (Type 1b or Type 71) to hot water storage tank (Type 4a).

In second loop of C-I, water from hot water storage tank always flow to adsorption system (Type 909) by chiller hot water circulation pump via auxiliary heater Type 751 (gas boiler), whenever pump receives an on signal from controller. In case of C-II, water from hot water storage tank will flow to adsorption system only water temperature at storage tank outlet is at least 85 °C. Set point temperature of auxiliary heater was 85 °C. Controller Type 108 switches on the boiler whenever storage outlet temperature is less than 85 °C.

In third loop cold water coming from adsorption system flows through the load (Type 682) by chilled water circulation pump. Chilled water set point temperature is 5 °C. Type 682 imposes a cooling load on a chilled water flow stream flowing through it. In order to have a load variation like in an actual air conditioning systems and to avoid the conventional method of modeling an actual building, a synthetic building Type 682 is coupled with Type 686 which provided realistic loads.

The third configuration incorporates two loops. Hot water from solar collector in first loop flows directly to auxiliary heater i.e. in this configuration there is no storage tank for hot water is used. The heater will be switched on if solar collector outlet water temperature is less than 85 °C else off. Similarly hot water coming out from adsorption chiller directly enters into the solar thermal collector. Second loop of this configuration is same as third loop of first two configurations. In second loop chilled water from adsorption chiller flows through the load (Type 682) by chilled water circulation pump as already discussed above.

3.4 System Performance Indicators

Following performance indicators based on integrated values of various energies over the whole summer season, are used to optimize the system:

1. Solar Fraction (SF) is the quantity of energy contributed by solar thermal collector divided by total energy needed to meet the required heating and/or cooling demand.

$$\mathrm{SF} = \frac{\int Q_u}{\int Q_u + \int Q_{aux}} \tag{3.94}$$

where Q_{aux} is the heat energy added from auxiliary boiler and Q_u is the energy gain by the solar collector which is useful . Its value is zero when system is entirely running on auxiliary heater and 1 when all the required energy is supplied by the solar system. High quantity of solar fraction is always desired to impose less load on auxiliary device.

2. Solar collector's long term performance is accessed by estimating collector efficiency which is defined by the following equation:

$$\eta = \frac{\int Q_u dt}{A \int G dt} \tag{3.95}$$

where *G* is the solar radiation (global) on a horizontal plane and A is the collector's area.

3. Equation for fractional primary energy savings (f_{PES}) is defined as [26]

$$f_{\mathrm{PES}} = 1 - \left[\frac{\frac{Q_{bioler}}{\epsilon_{heat}}}{\frac{Q_{cooling,ref}}{SPF_{ref} \cdot \epsilon_{elec}}} \right] \tag{3.96}$$

where $Q_{cooling,ref}$ is the cold energy provided by a traditional vapor VCRS and Q_{boiler} is the auxiliary heat provided by the boiler, ϵ_{heat} is the heat conversion efficiency in the boiler , SPF_{ref} is the typical efficiency or COP of compression chiller, and ϵ_{elec} represents the efficiency of a thermal power plant. These fractional energy conversion factors for primary energy for electricity (ϵ_{elec}) and heat (ϵ_{heat}) from fossil fuels and SPF_{ref} are taken as:

ϵ_{heat}= 0.84

ϵ_{elec}= 0.39

SPF_{ref}= 2.8

The term in brackets in equation 3.96 is the ratio of auxiliary energy consumed in solar based adsorption cooling system to the energy consumed by an electrically operated compression chiller to meet the same cooling demand.

Chapter 4

Results and Discussion

Simulations were performed in TRNSYS for the whole cooling season (i.e. from May to September). Simulation start and end time was set to 2880 hr and 6552 hr respectively with a time step of 0.125 hr.

4.1 Variation of solar fraction with collector area and collector tilt

Figure 4.1 exhibits the effect of varying ETC area and tilt angle on SF. Maximum value of SF is achieved at 13° tilt angle for each value of solar collector area. Increasing tilt angle further causes SF to decrease. Similar trend is also obtained in case of FPC.

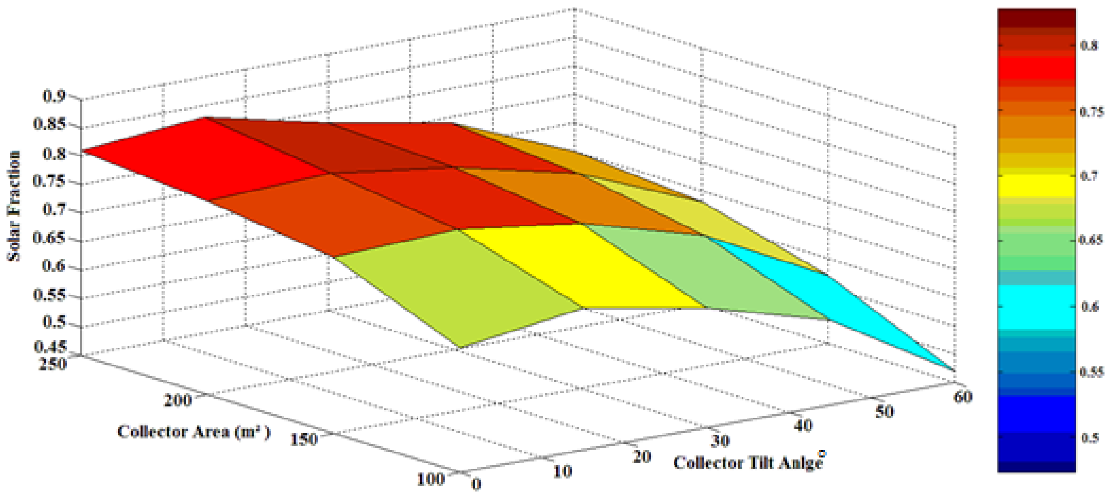


Figure 4.1 Effect of Variation of ETC area and collector tilt on solar fraction

4.2 Fractional primary energy savings

Estimation of optimum size of storage value is critical with regard to both thermal performance of the whole system and its overall cost. Fractional primary energy savings, f_{PES} and collector size are used as key parameters to estimate optimum storage volume. Figures 4.2 and 4.2.1 shows variation of f_{PES} by changing storage tank size for different collector areas of ETC and FPC. It can be seen that if collector area of both FPC and ETC is less than a certain minimum value, f_{PES} results in negative values and keeps on decreasing with increase in storage size. This minimum value of collector area is found to be 200 m² and 130 m² for FPC and ETC, respectively. For collector areas larger than these, f_{PES} increases with increase in storage volume, reaches a maximum and then starts to decrease. For a fixed value of storage volume, increasing collector's area directly increases f_{PES} but cost also increases. For this study collector area for FPC and ETC is taken as 250 m² and 150 m² respectively as these areas's gives quite reasonable values of f_{PES}. It can be seen that for positive values of f_{PES}, maximum f_{PES} in case of FPC and ETC is obtained for an approximate storage volume of 40 l/m^2 and 30 l/m^2 respectively. Results showed that by increasing storage size upto a certain value, auxilary energy demand decreases and takes its minimum value. Further increasing the storage size results in increase in Q_{aux}. Q_{aux} and f_{PES}. are related to each other by equation 3.96 which clearly shows that decrease in Q_{aux} results in increase in f_{PES}. Therefore for an appropriate collector area, as storage size increases, f_{PES} takes its maximum value and then decreases (as shown in figure 4.2 and 4.2.1).

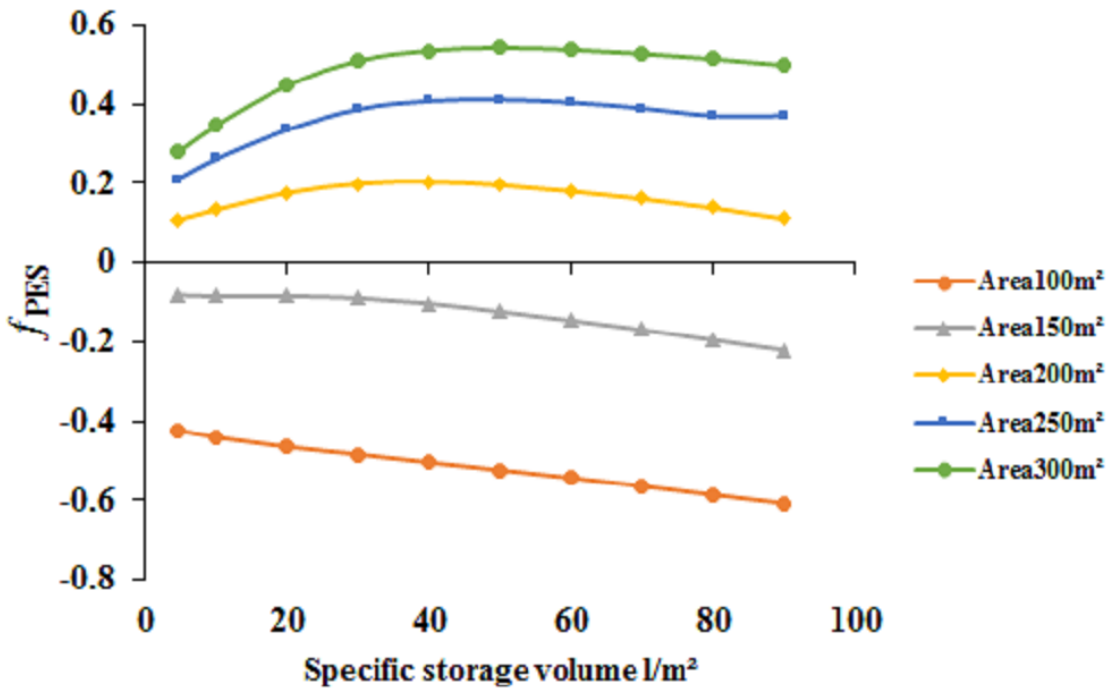


Figure 4.2 Graph between storage volume and f_{PES} for different values of FPC areas

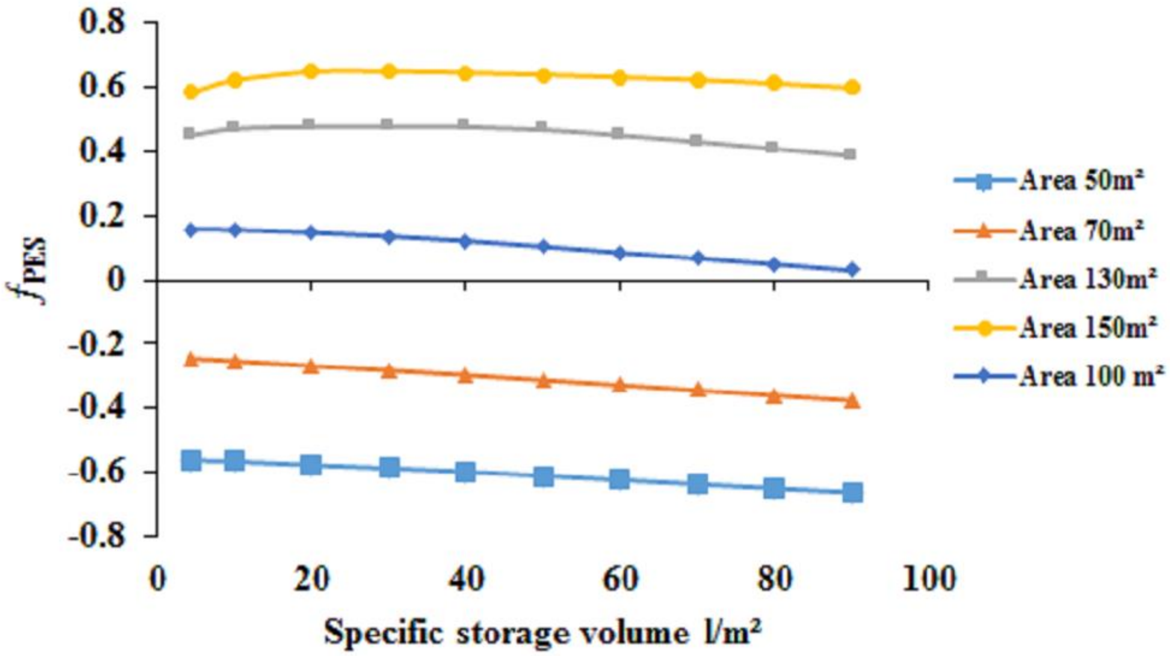


Figure 4.2.1 Graph between storage volume and f_{PES} for different values of ETC areas

4.3 Storage tank size Optimization

Figure 4.3 illustrates the effect of changing storage size on seasonal f_{PES} and seasonal SF. Based on FPC area of 250 m² and ETC area of 150 m² obtained from figure 4.2 and figure 4.2.1, it can be concluded from figure 4.3 that for both FPC and ETC (for C-I), highest values of f_{PES} and SF occur simultaneously and are obtained at a storage size of 40 l/m^2 and 30 l/m^2 respectively. Similar trend is observed in case of C-II.

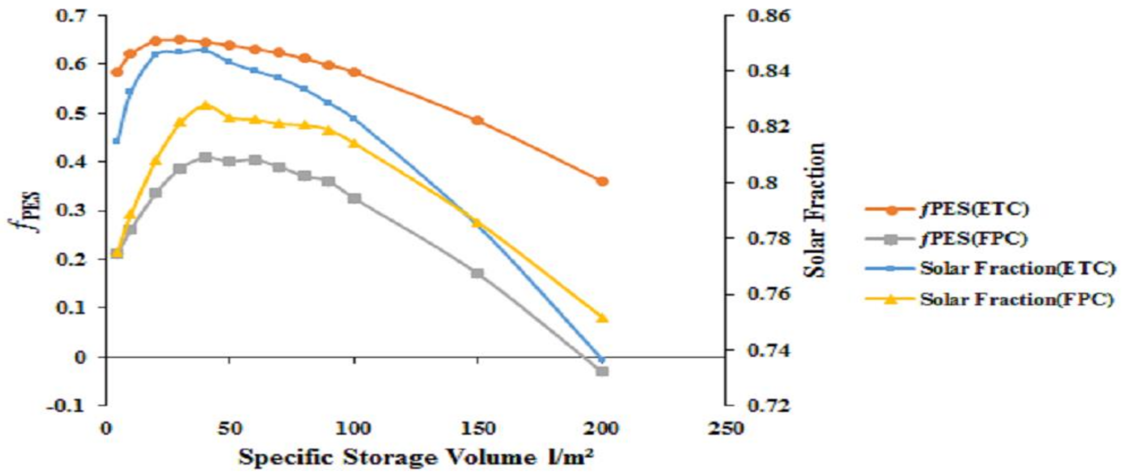


Figure 4.3 Variation of seasonal fractional primary energy savings and seasonal SF by varying storage size for both FPC and ETC

4.4 Seasonal variation of solar fraction

Figure 4.4 shows variation of monthly SF across various months of cooling season for three configurations corresponding to FPC and ETC areas of 250 m^2 and 150 m^2 respectively. For both FPC and ETC, C-II has slightly higher SF than C-I. Moreover ETC gives slightly higher SF as compare to FPC which implies that compared to expensive ETC's, low cost FPC's can also furnish the desired energy demand. In case of C-III it is observed that for same areas of 250 m^2 and 150 m^2, 50% of total energy required by system can be furnished from solar collectors even in the absence of hot water storage tank. In figure 4.4 it can be seen that SF is less in July and August as compare to May and June because of availability of less solar radiation in July and August. Decrease in cooling load and availability of high SF in September results in higher value of SF in September as compare to July and August.

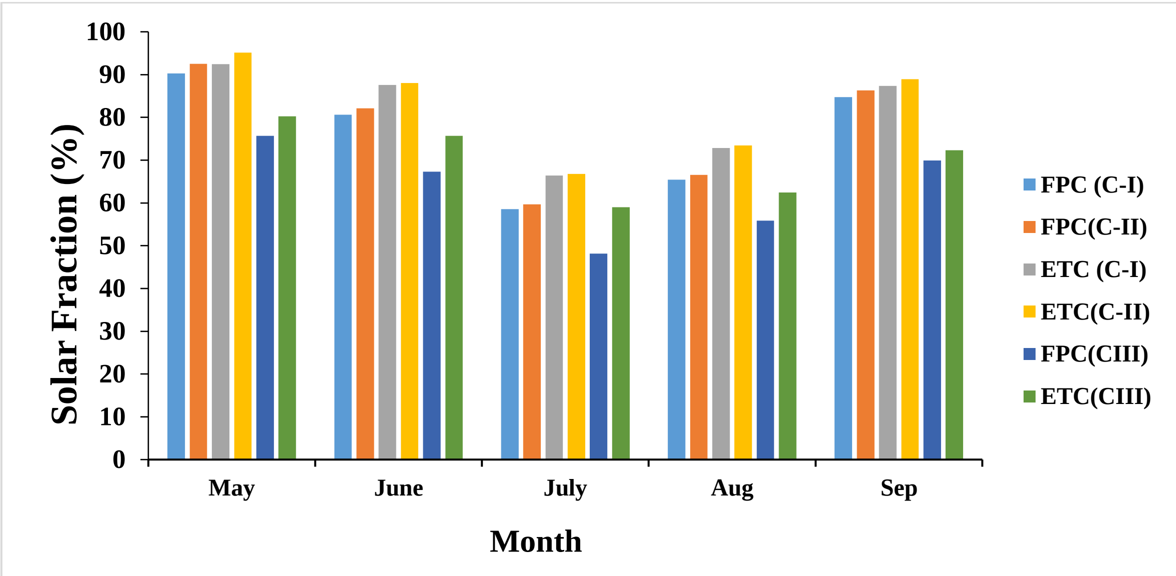


Figure 4.4 Variation of solar fraction across various months of cooling season

4.5 Seasonal variation of collector efficiency

Figure 4.5 presents monthly thermal efficiency of FPC and ETC for all three configurations corresponding to FPC and ETC areas of 250 m^2 and 150 m^2 respectively. It is evident that ETC yields higher monthly efficiency than FPC, however the effect of system configuration (C-I or C-II) on collector efficiency is marginal. It is also observed that for fixed collector areas of 250 m^2 and 150 m^2, system without hot water storage tank has higher values of collector thermal efficiency. It is due to the fact collector array in a system without hot water that temperature of the water just like in than FPC entering the solar storage C-I and C-II, It is also evident in C-III that ETC yields higher monthly efficiency tank is comparatively lower.

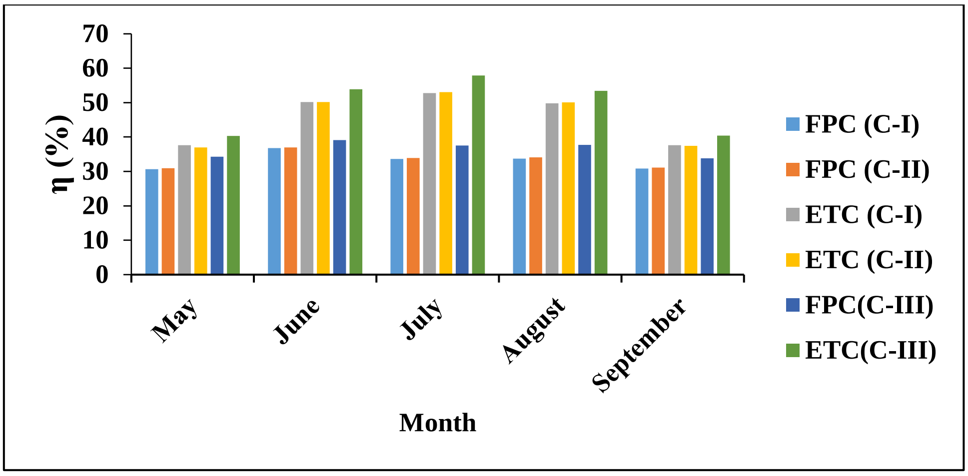


Figure 4.5 Monthly efficiency of solar collectors for various months of cooling season

4.6 Variation of FPES with collector area for different configurations

Figure 4.6 shows variation of f_{PES} with increase in collector's area for all three configurations. For both FPC and ETC, C-II has slightly higher f_{PES} than C-I but the difference of f_{PES} between C-I and C-II decreases as collector's area increases. As expected from trends of figure 4.2 and 4.2.1 for C-I and C-II, system without storage tank has lower values of f_{PES} as compare to that for system with hot water storage tank. Moreover as compare to FPC, system with ETC gives comparatively higher values of f_{PES}. Contrary to C-1 and C-II in which f_{PES} value increases by increasing hot water storage tank size up to a certain value and then decreases for a fixed value of collector area (provided that collector area must be greater than a certain value to avoid negative increase in f_{PES} value), in C-III f_{PES} keeps on increasing with increase in collector area.

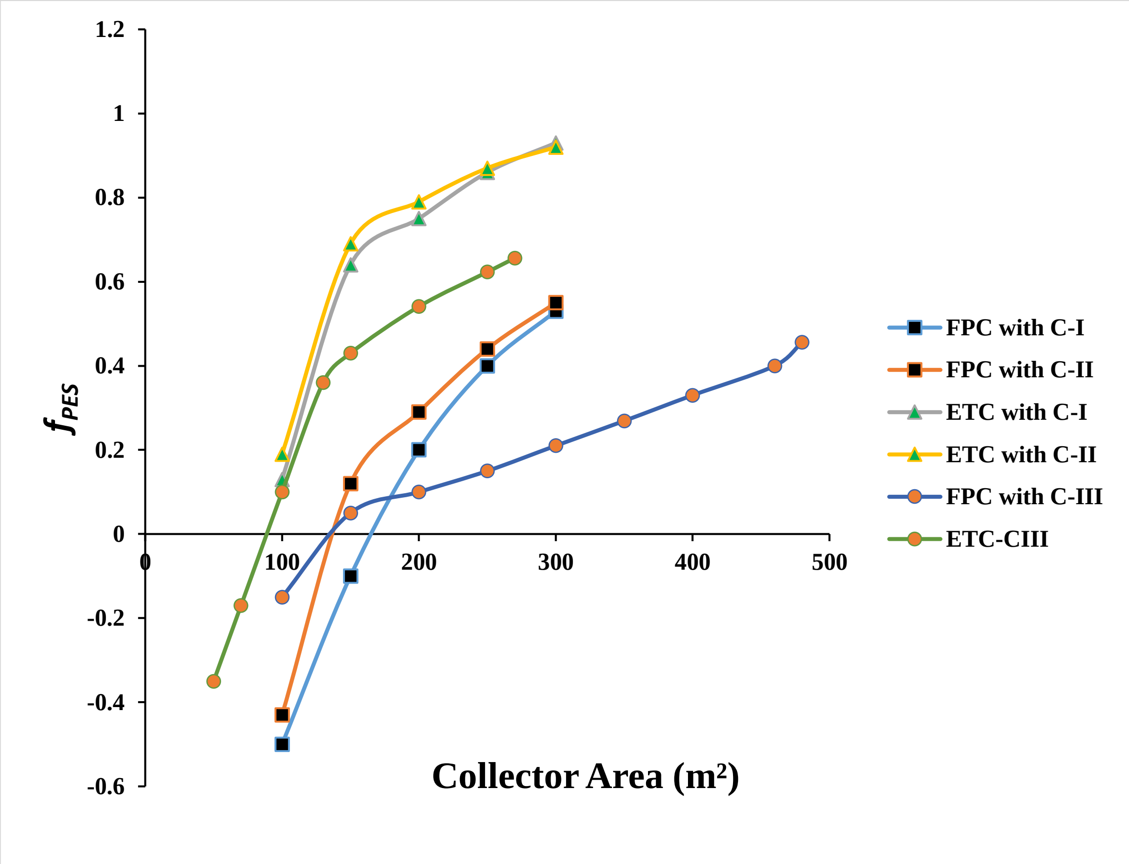


Figure 4.6 Comparison of f_{PES} For Different Configurations

4.7 Variation of solar fraction with chiller hot water inlet temperatures

Figure 4.7 and 4.7.1 shows effect of variation of chiller hot water inlet temperatures on solar fraction for FPC (Area 250 m^2) and ETC (Area 150 m^2) respectively for all three configurations. Lower the chiller hot water inlet temperature higher is the solar fraction for any configuration. For both FPC and ETC, C-I and C-II have higher values of SF as compare to C-III. At lower chiller hot water driving temperatures, C-II has higher values of SF as compare to C-I. This is due to the fact that at low driving temperatures in C-I, as the returning cold water form chiller mixes with the cold water in the bottom of storage tank, the stratification in the tank disturbs which imposes load on auxilary devices. Collector needs more time to attain the required temperature at the outlet of hot water storage tank. The difference in SF between C-I and C-II decreases at higher chiller driving temperatures.

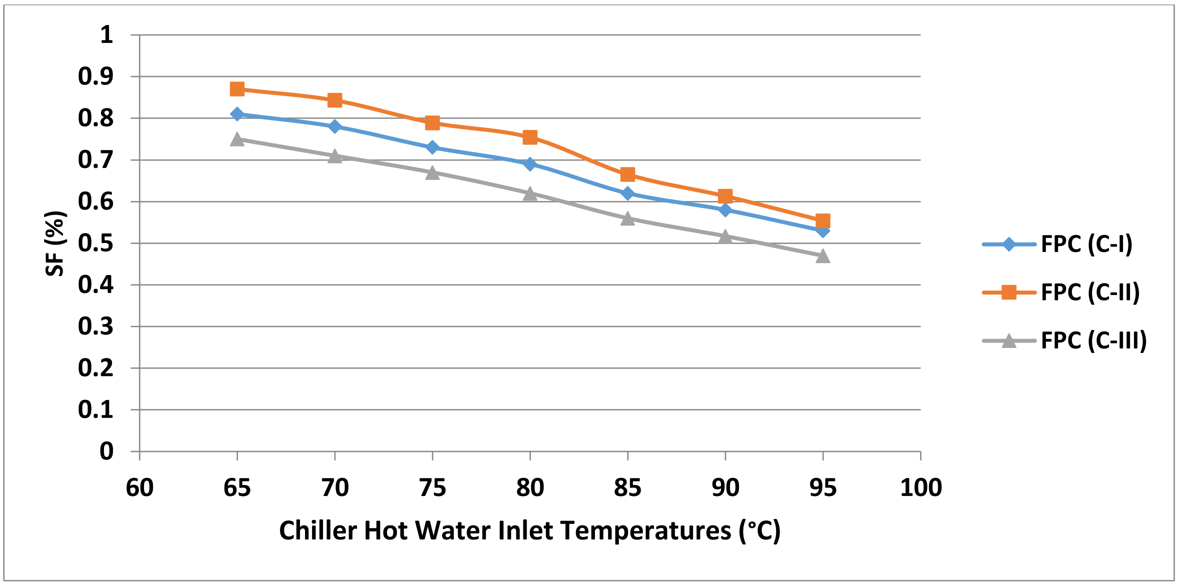


Figure 4.7 Effect of chiller hot water inlet temperature on solar fraction for FPC

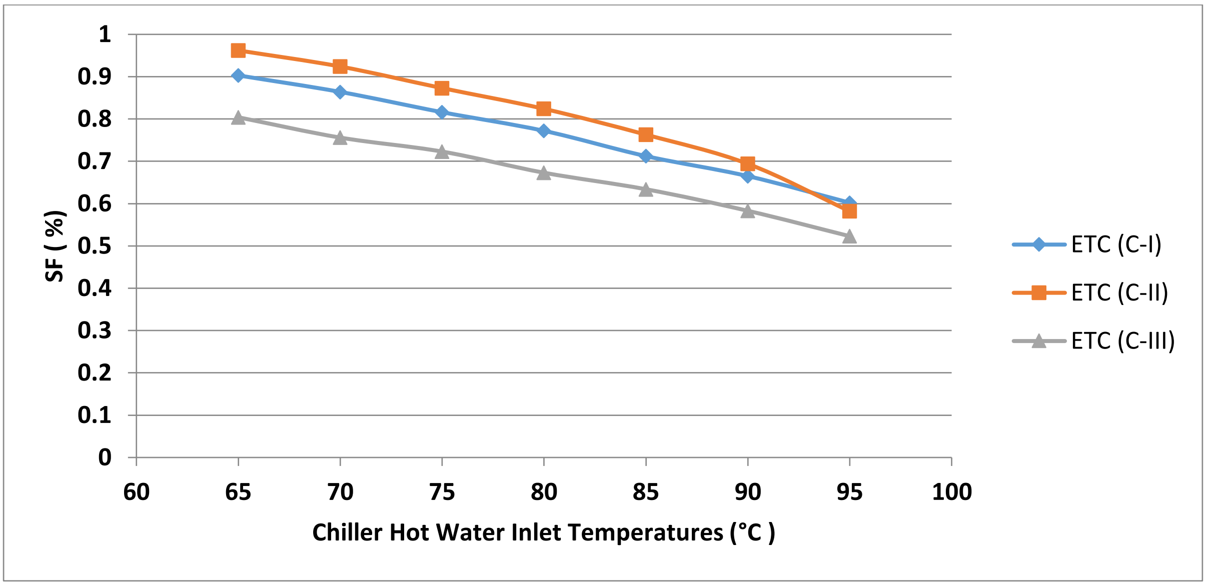


Figure 4.7.1 Effect of chiller hot water inlet temperature on solar fraction for ETC

4.8 Variation of FPES with chiller hot water inlet temperature

Figure 4.8 and figure 4.8.1 shows effect of variation of chiller hot water inlet temperatures on FPES for all three configurations for both FPC (Area 250 m²) and ETC (Area 150 m²) respectively. Lower the chiller hot water inlet temperature higher is the FPES for any configuration. For both FPC and ETC, C-II has comparatively high values of FPES as compare to C-I and C-III. The reason behind this is same as discussed in variation of SF with chiller hot water inlet temperatures. Figures also shows that quite lower FPES is obtained in case of C-III which suggest that at these selected areas of 250 m² and 150 m² for FPC and ETC, C-III is not feasible.

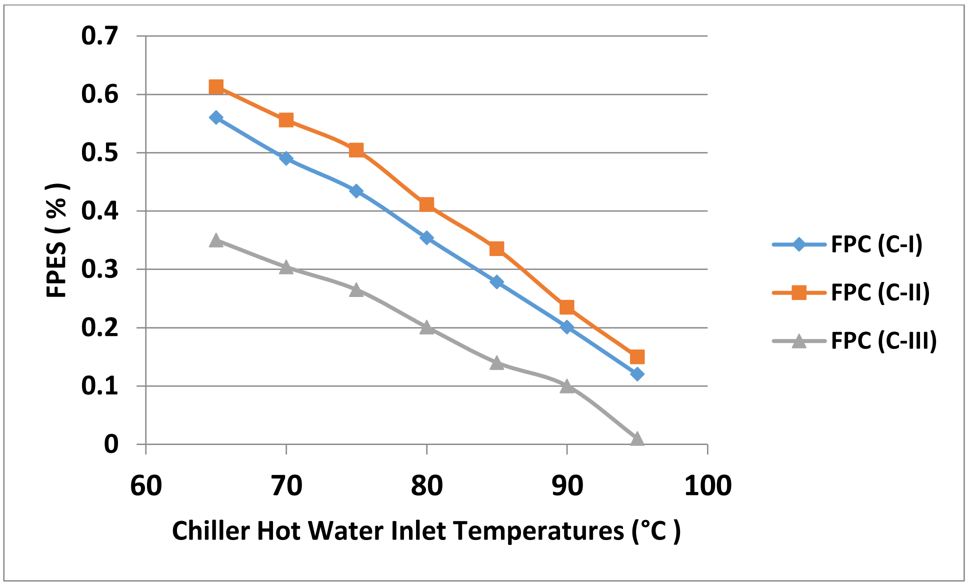


Figure 4.8 Effect of chiller hot water inlet temperature on FPES for FPC

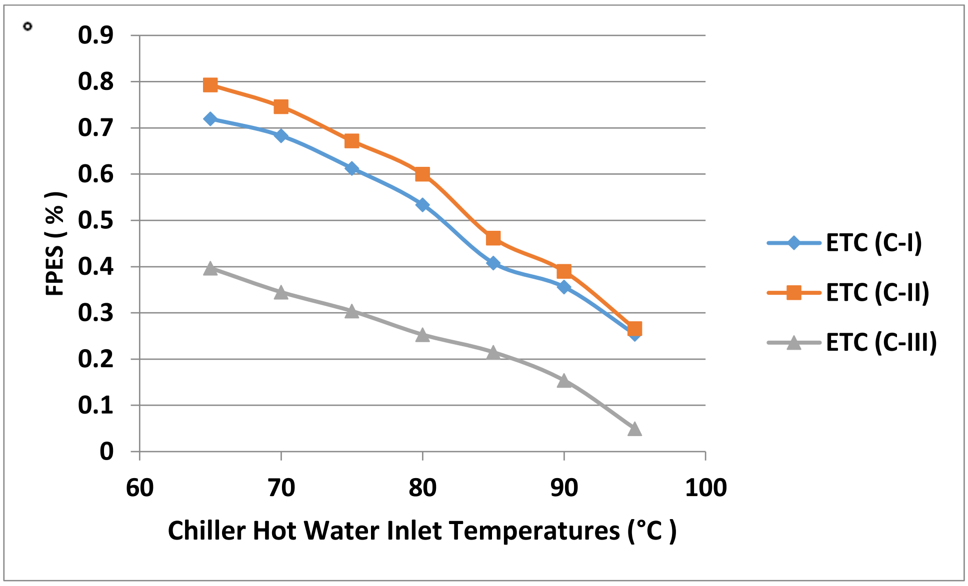


Figure 4.8.1 Effect of chiller hot water inlet temperature on FPES for ETC

4.9 Variation of collector efficiency with chiller hot water inlet temperature

Figure 4.9 and figure 4.9.1 shows effect of variation of chiller hot water inlet temperature on collector efficiency. As compare to CI tank is C-II, C-III has slightly higher collector efficiency. It is due to the collector array in a system without fact that temperature of the comparatively lower as and water entering the solar hot water storage water entering in the solar collector array is coming directly from adsorption chiller.

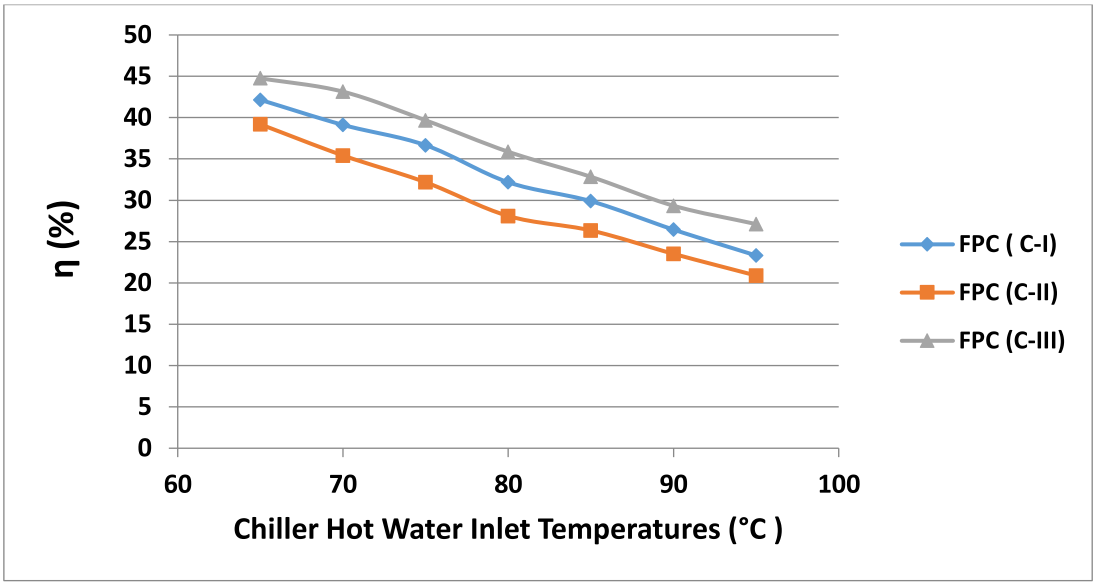


Figure 4.9 Effect of chiller hot water inlet temperature on collector efficiency for FPC

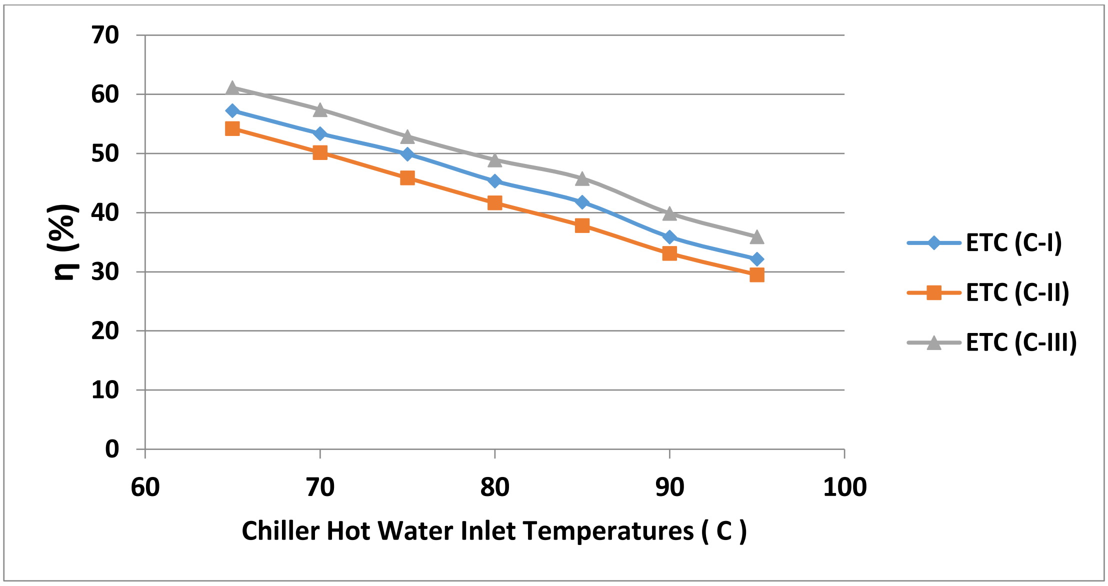


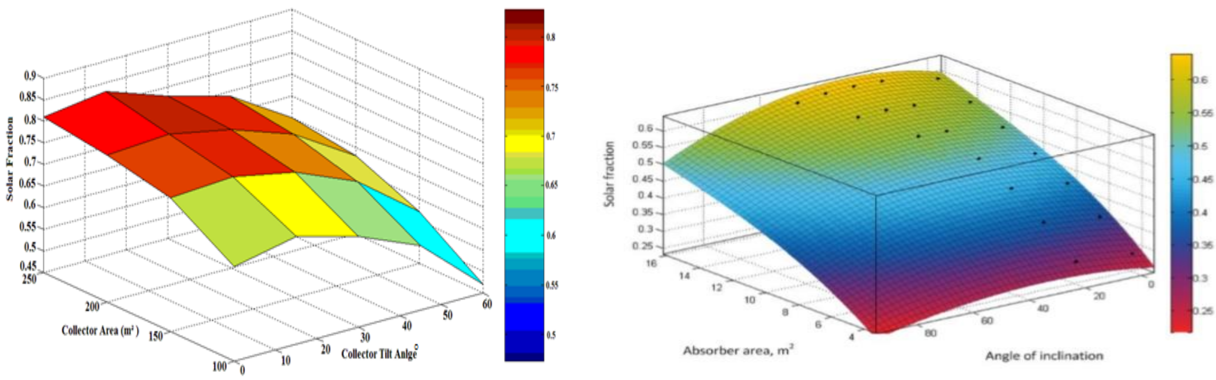
Figure 4.9.1 Effect of chiller hot water inlet temperature on collector efficiency for ETC

Chapter 5

Model Validation

5.1 Variation of solar fraction with collector area and collector tilt

Figure 4.1 in chapter 4 exhibits the effect of varying ETC area and tilt angle on SF. Maximum value of SF is achieved at 13° tilt angle for each value of solar collector area. Increasing tilt angle further causes SF to decrease. Similar trend is also obtained in case of FPC. Trend shown in figure 4.1 is supported by Janusevicius et al. [47] for a solar based adsorption chiller. The highest value of SF for is obtained if the collector tilt angle is approximately 30° for Vilnius, Lithuania. Papoutsis et al. [57] presented similar trend for a solar based adsorption chiller. Their results showed that maximum SF is obtained at a collector tilt of 14° for Athens, Greece. Similarly simulation results of Assilzadeh et al. [1] for solar assisted absorption cooling system also show that the optimum angle for Malaysia environment is 20° for maximum solar heat gain as shown in figure 5.1.



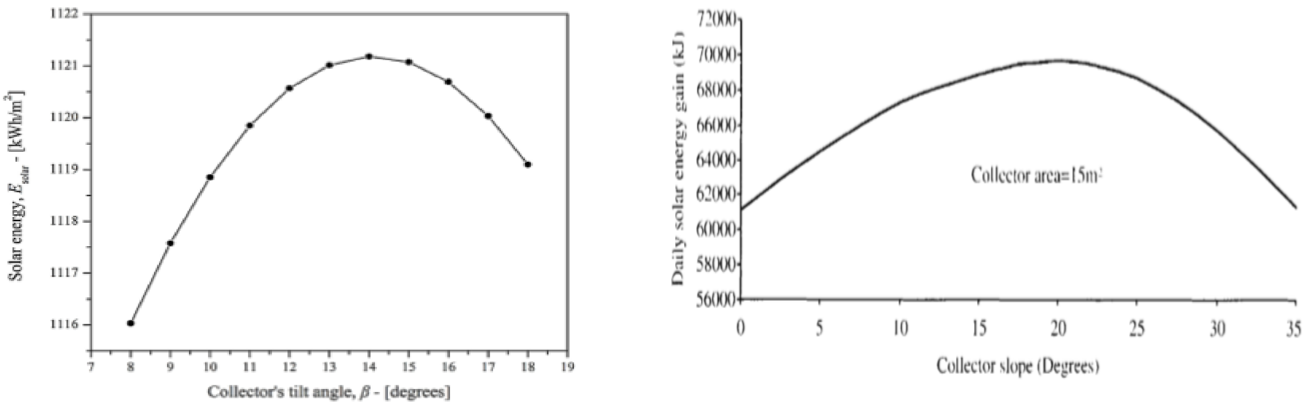
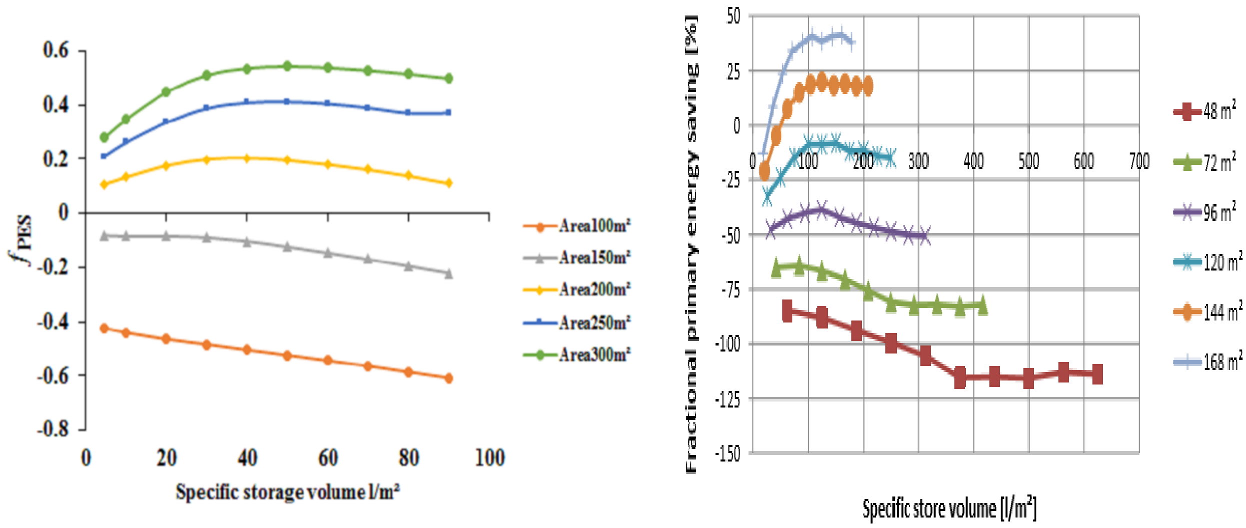


Figure 5.1 Comparison of effect of variation of collector area and collector tilt on SF with literature

5.2 Fractional primary energy savings and solar fraction

Figure 5.2 shows comparison of results presented in figure 4.2 and figure 4.2.1 with literature. He et al. [54] and Assilzadeh et al.[1] analyzed the variation of auxilary energy demand (Q_{aux}) with storage tank size. Although their findings are for a solar based absorption cooling system but the trend shown between Q_{aux} and storage size is quite similar to that for an adsorption system. Ghaghazanian [56] examined the PV/T solar collector performance for an office building in Cape Town, Los Angeles, New Dehli and Dubai and presented similar trend as shown in figure 4.2 and 4.2.1. Results showed that by increasing storage size upto a certain value, auxilary energy demand decreases and takes its minimum value. Further increasing the storage size results in increase in Q_{aux}.



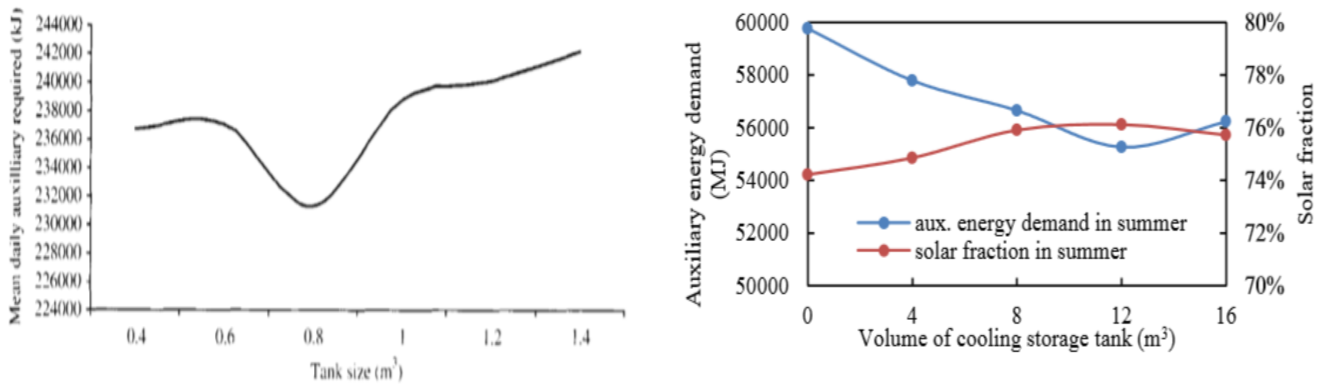


Figure 5.2 Comparison of effect on f_{PES} by varying size of hot water storage tank with literature

Figure 5.2.1 shows comparison of variation of SF by varying size of hot water storage tank with literature. As in case of f_{PES}, by increasing tank size, SF increases up to a certain value. Further increase in tank size results in decrease in SF.

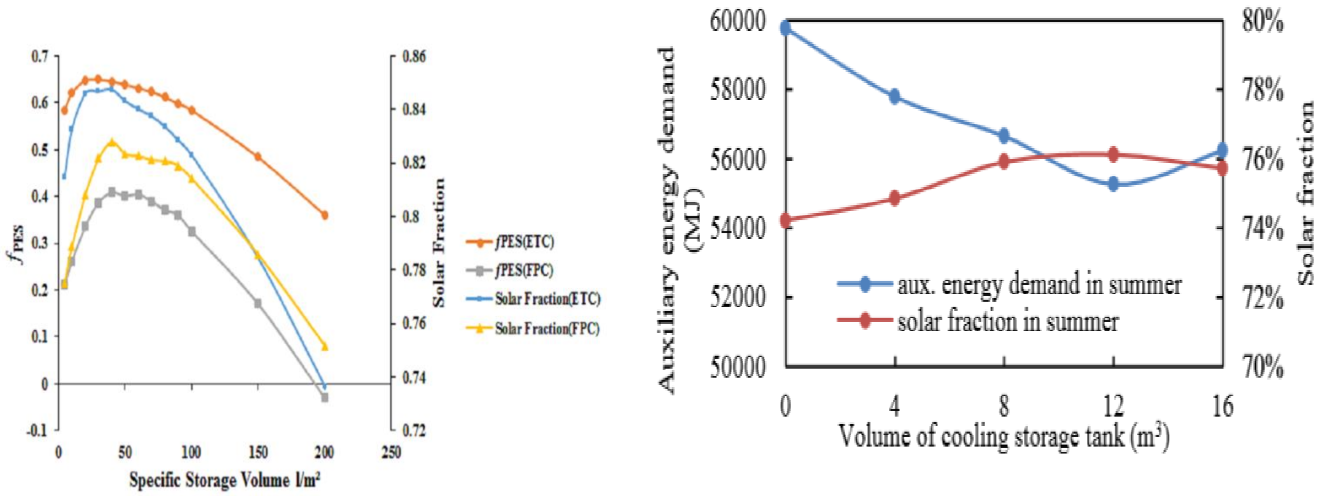


Figure 5.2.1 Comparison of effect on SF by varying size of hot water storage tank with literature

5.3 Collector Efficiency

Figure 5.3 shows comparison of collector efficiency of all three configurations with literature. Trend shown in figure 4.5 is supported by Wang et al. [48]. Their analysis shows that entering water in a system without the system without storage tank the solar collector hot water has greater values of collector thermal efficiency. It is because of temperature of storage tank is comparatively lower as entering water in the solar collector array is coming directly from adsorption chiller.

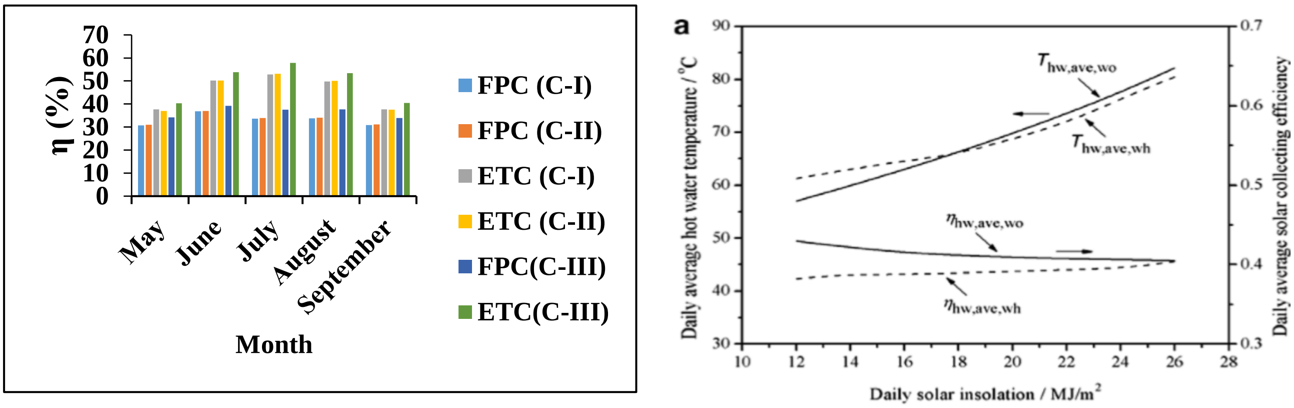


Figure 5.3 Comparison of collector efficiency with literature

Conclusions

This thesis presents the modeling and analysis of three system configurations of solar assisted adsorption chiller system for Islamabad (33.71° N, 73.06° E) located office building located having a peak cooling load of 46 kW. Simulations of the system are done in TRNSYS for the whole summer season to investigate the possibilities of driving adsorption chiller with low grade heat, i.e. from an evacuated tube or flat plate solar collector. Results of the analysis showed that for a given collector area, highest solar fraction is achieved at collector tilt of 13°. With FPC area of 250 m^2, up to 92% monthly SF and maximum of 0.20 monthly f_{PES} can be obtained at hot water storage size of 10000 liters. In case of ETC, up to 95% monthly SF and maximum of 0.64 monthly f_{PES} can be obtained at a reduced solar collector area of 150 m^2 and having 4500 liters hot water storage size. For both FPC and ETC, the difference in SF for C-I and C-II is marginal but as compare to FPC, higher SF is obtained from ETC irrespective of configuration. For both FPC and ETC, C-II has slightly higher f_{PES} then C-I but the difference reduces to zero as collector area increases.. ETC also offers higher thermal efficiencies as compare to FPC, but no prominent variation in efficiencies is seen as we switch from C-I to C-II. In case of C-III, up to 75% monthly SF and maximum of 0.15 monthly f_{PES} is obtained with FPC area of 250 m^2 whereas up to 80% monthly SF, maximum of 0.43 monthly f_{PES} are obtained with ETC area of 150 m^2. However thermal efficiency of FPC and ETC is higher in case of C-III. It is because water temperature entering the solar thermal collector in without hot water storage tank system is comparatively lower as water entering in the solar collector is coming directly from adsorption chiller. Though lower than C-I and C-II, C-III still gives quite reasonable values of SF. Therefore if it is not feasible or economical to install a storage tank for hot water in solar assisted adsorption cooling system design, it is recommended that model without hot water storage tank can work by increasing solar thermal collector area to obtain SF and f_{PES} equal to that of model with hot water storage tank.

Future Work

Simulation results presented in this study have shown that there is good potential for solar based adsorption cooling system in Pakistan. Till date there is not even a single solar based adsorption cooling system installed in Pakistan. Therefore model presented in this study can be taken as a reference for design and installation of solar assisted adsorption cooling system. However the following ideas should be addressed in future work:

1. Performance analysis of system by addition of PV panels in the proposed solar based adsorption cooling system so that electric load of different devices e.g. pumps, adsorption chiller, AHU/Fan coil can be shifted on PV panels. In this way the system will be less dependent on electricity from grid hence results in more energy savings.
2. Due to absence of any solar based adsorption cooling system in Pakistan, results presented in this study for proposed configurations have been validated with literature and need experimental validation.
3. By addition of some components, the proposed model in this study can be used to carry out year around simulations for space heating and cooling.
4. The proposed model can be used to investigate the configuration based study of solar assisted adsorption cooling system coupled with solar assisted desiccant cooling system.

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CERTIFICATE OF COMPLETENESS

It is hereby certified that the dissertation submitted by NS Muhammad Wajahat Khan, Reg No.NUST201362449MCEME35113F,Titled:Modeling and Analysis of Solar Assisted Adsorption Cooling System using TRNSYS has been checked/reviewed and its contents are complete in all respects.

Supervisor's Name: Dr. Tariq Talha

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Date: ___________________