Design of Solar Energy Based Air-Conditioning System for Climatic Conditions of Pakistan

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Dedication

I humbly dedicate this work to my parents and sisters whose endless prayers, continuous motivation and infinite sacrifices equipped me with everything it takes to be an accomplished human being.

Acknowledgment

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Abstract

Cotemporary Energy issues, like depletion of energy resources, rising energy costs and the related environmental issues demand exploration of renewable energy technologies. Solar powered desiccant based air-conditioning systems using solar thermal collector or solar photovoltaic/thermal collector is an emerging renewable energy technology. These systems are very attractive alternative to conventional refrigeration systems for reduction in peak load demand of electricity. This research reveals the feasibility and optimization analysis of a solar desiccant cooling system by using TRNSYS simulation software for Typical Meteorological Year data of Islamabad, Pakistan. TRNSYS is a tool used for modeling of renewable energy systems. A simulation model that contains solar PV/T collector, storage tank, direct evaporative cooler, vapor compression chiller and the building is developed. The results depict that desiccant system of rated capacity 2.5TR is designed for climatic conditions of Islamabad. The study also reveals that to increase the reliability of operation, an optimized system containing 15 m^2 area of flat plat PV/T collector sloped at 15^o is required. Hot water storage tank of 0.5 m^3 and auxiliary heater having set point temperature of 60 ^0C is essential to provide effective cooling to the building. The economic analysis shows that the system has simple and equity payback period of 7.4 and 6.4 yrs. respectively. It could be reduced by using the same system for water and space heating by adding the few controllers and devices for the winter season.

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1. Abdul Samad Farooq, M B Sajid, A W Badar, M Fatima "Simulation of a Solar Energy Driven Desiccant and Vapor Compression Based Air Conditioning System for Climatic Conditions of Pakistan" 7th International Conference on Environmentally Sustainable Development, August 26-28, 2017 Comsats Institute of Information Technology Abbottabad.
2. Hafiza mahreen Fatima, M B Sajid, A S Farooq, "Modeling and Simulation of Solar Water Heater for Islamabad region" 7th International Conference on Environmentally Sustainable Development, August 26-28, 2017 Comsats Institute of Information Technology Abbottabad.
3. Muhammad Mohsin Tanveer, M.B Sajid, Dr. Adeel Javed, Abdul Samad, Nouman Ali, Muhammad Usman "Heat transfer in microchannels using supercritical CO2 for Solar Tower Applications" 2nd International conference on impact of Nanoscience on energy Technologies, 26 OCT 2017
4. Abdul Samad Farooq, M Fatima, M B Sajid, A W Badar "Design of Solar Energy Based Air Conditioning System for Climatic Conditions of Pakistan" National Conference of Green Energy Technologies, May 18, 2017 Pearl Continental Hotel Peshawar.
5. Abdul Samad Farooq, M B Sajid, M Fatima, A W Badar "Design and Simulation of Solar Energy Based Air Conditioning System for Climatic Conditions of Pakistan" International Conference on Sustainable Energy Technologies, September 12-13,2017, Serena Hotels Islamabad.
6. Abdul Samad Farooq, H M Fatima, M B Sajid, A W Badar, "Economic Analysis of Domestic Usage of Solar PV System in Urban Areas of Pakistan" International Conference on Sustainable Energy Technologies, September 12-13,2017, Serena Hotels Islamabad.

Mathematical Symbols

Name	Symbol	Unit
Overall thermal loss coefficient of the collector per unit area	U_L	[kJ/h-m²-K]
Average collector fluid temperature	T_{av}	[°C]
Ambient (air) temperature	T_a	[°C]
Collector slope above the horizontal plane	β	[°]
specific heat of fluid	C_P	[KJ/kg. k]
Mass flow rate	$\dot{m}$	[kg/hr.]
Control function	γ	
Temp of air entering the evaporator side of the coil	$T_{evap, in}$	[°C]
Temp of air exiting the evaporator side of the coil	$T_{evap, out}$	[°C]
Enthalpy of air entering the evaporator side of the coil	$h_{evap, in}$	[KJ/Kg]
Enthalpy of air exiting the evaporator side of the coil	$h_{evap, out}$	[KJ/Kg]
Rate of total energy transferred by the coil	$\dot{Q}_{total}$	[KJ/Hr.]
Isopotential curves fits of wave front propagation through the desiccant matrix	F1, F2	
Humidity Ratio	ω	[kg$H_2$0/kg. Air]
Heat exchanger effectiveness	ε	[0, 1]

List of abbreviation

DWC	Desiccant Wheel Cycle
VCC	Vapor Compression Chiller
COP	Co-efficient of Performance
HCFC	Hydro-chlorofluorocarbon
PV/T	Photovoltaic-thermal
TRNSYS	Transient energy system simulation tool
NPV	Net Present Value
TESS	Thermal energy system specialists
PES	Primary Energy Savings
TSF	Thermal Solar Fraction
ESF	Electrical Solar Fraction
HVAC	Heating, Ventilation and Air Conditioning
CFC	Chloro-fluorocarbon

CHAPTER#1

1 Introduction to Solar Air conditioning

1.1 Overview:

According to the approximation of International Institute of Refrigeration in Paris, out of total electricity produced in the whole world, 15% is consumed for the refrigeration and air-conditioning and out of total energy being consumed by the commercial buildings and residences, 45% is used for air-conditioning [1] [2]. Currently most of research is being emphasized on the development of new technologies which can offer the reduction in energy consumption, peak energy demand and cost while providing the desired comfort level. [3]

Conventional air-conditioning systems consume major portion of electricity and refrigerants used in these systems also contribute to the greenhouse gas emissions. [4]Figure 1.1 shows the global solar flux at the surface of earth. The average solar energy received per year is about 1200 KWh/m^2 in Europe. The united states of America, most of India and many parts of china are also having good solar resource. But the region of northern Europe, Russia and Canada are less favored.

Solar thermal energy is a favorable alternative for air conditioning of buildings, and the interest is growing continuously among engineers and professionals, since the peak cooling is generally required when conditions are most favorable to the use of solar technologies. This can provide substantial economic benefits as a combination of the conventional and solar-cooling systems can be used to meet the peak load requirements. Solar cooling is generally provided either by an electric refrigeration system which mainly consists of photovoltaic panels combined with a conventional vapor compression system or by solar thermal systems, which use heat energy to produced cooling by thermally driven refrigeration cycles.

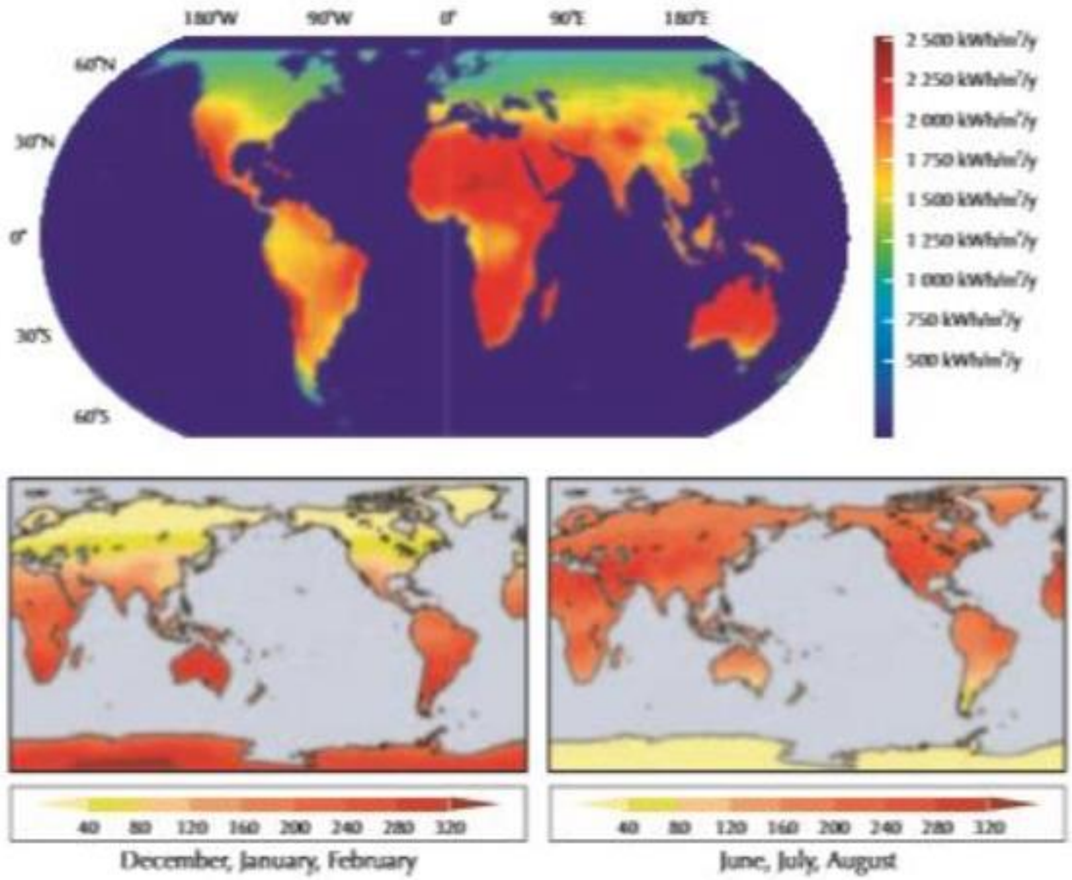

Figure 1-1: The global solar flux (KWh/m2/y) at the earth surface over the year [5]

1.1.1 History of air conditioning and refrigeration

Air conditioning is vital for thermal comfort in indoor atmosphere specially for the hot and humid climatic conditions. In the areas of the high humidity apart from cooling dehumidification also have a significant importance. In tropical regions heating ventilation and air exceed 50% of the total energy used. [6]

The history of air conditioning started to take the intention in 1840s for the very first time when a physician from Florida Dr. John Glorrie proposed the idea of cooling in cities. He believed that malaria and other different diseases might be avoided through cooling. [7] Romans are considered as the pioneers of central heating. There are some signs that Jews, Persians, Chinese and Indians understood the air conditioning concept in earlier centuries. The first public demonstration on refrigeration was given by the Professor from University of Edinburg named William Cullen. William Cullen used a pump to create a partial vacuum in a container containing diethyl ether. The small amount of ice was formed from that process but not as much useful for commercial applications. The very first person who tried to develop a air conditioning system was Dr. John Gorrie. Dr. John was a physician and scientist. He believed that malaria and other different diseases might be avoided through cooling. [7]. Dr. John invented an open cycle refrigeration machine that was used for the cooling of rooms in hospitals.

First patent on air conditioning was by A. Muhl. During 1859-1909 A.R Wolf implemented the air conditioner in more than 100 buildings. An American engineer named Willis H carrier proposed the first modern air conditioner in 1902. Willis H carrier also developed the first psychometric chart and also give the idea to control the humidity in air conditioning devices. Ammonia was used as a coolant at that time which is very toxic but in 1922 ammonia was replaced by the dielene and size of compressor was also reduced which was the great achievement. Window air conditioning unit were emerged after the second world war with the sales over one million in 1953. During 1950 to 1970 as the cost of electricity was low and also the world was not much familiar with the global warming and ozone depletion that's why at that time little importance was given to the other types of air conditioning. Alternative types of air conditioning are obtaining the strength due to environmental issues and high electricity prices.

1.1.2 Environmental effect

Modern air conditioning system effects the environment by the emissions of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). Chlorine atoms in HCFCs and CFCs were investigated by Rowland and Molina and found that they are the cause of breakdown of ozone layer in the stratosphere. [8] Ozone layer as shield against the ultraviolet solar radiations that can cause the cancer. Drastic diminishing of ozone layer over Antarctica was noticed by the scientists by 1985. Montreal protocol was signed by the leaders of 24 nations on September 16,1987. [9] Until now the production of CFCs is stopped and HCFCs are also expected to be stopped until 2030. After the higher cost of electricity global warming is the second key point in the favor of alternative cooling systems because most the refrigerants mostly used in vapor compression cycle contains the components which are contributing in global warming. RA-134 which is the widely used refrigerant has a GWP which is equivalent to 1320 times CO2. Apart from the greenhouse gas that are produced from the air conditioning units if we study deeply then we come to know that most of the energy that they are using also come by burning the fossil fuels which are also the major contributor in the global warming. [10] Average earth temperature is also increasing day by day due to global warming. According to a study the trend of temperature increase in Pakistan shows that up to 208o on average 5 to 60C temperature will be increased due to which cooling demand is also increasing day by day.

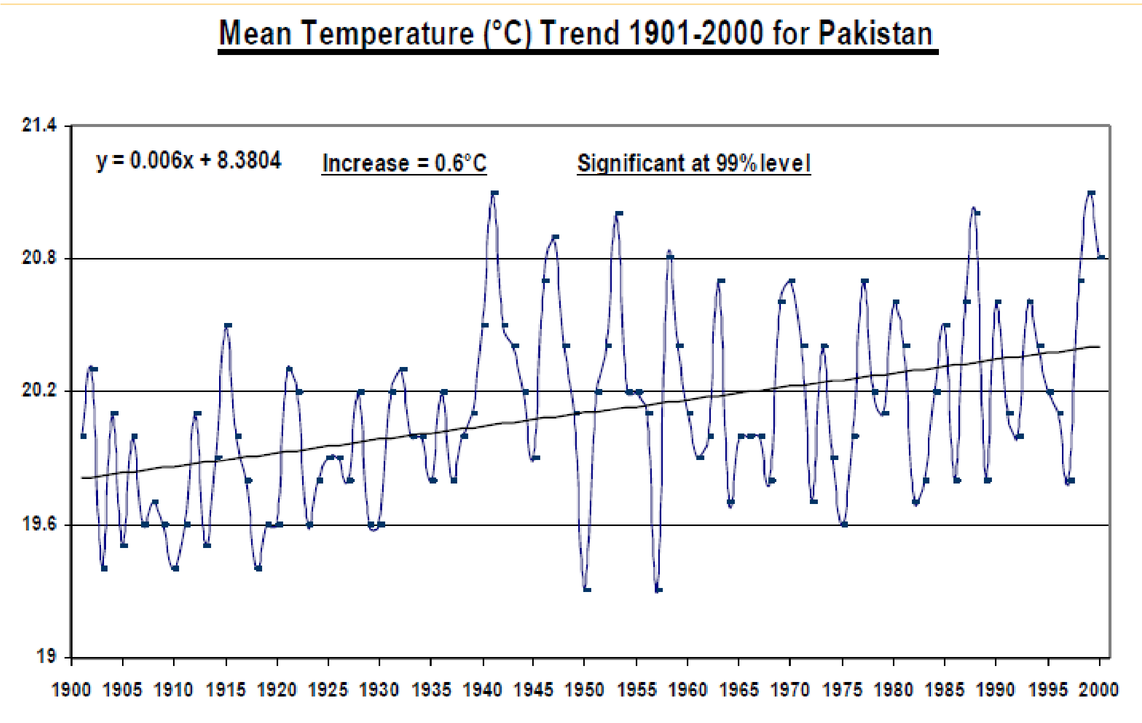

Figure 1-2 Mean Temperature Trends for Pakistan [11]

Due to the increase in temperature it is predicted that all the glaciers in Pakistan will be melt down in next 75 years which may lead to

- Agricultural losses
- Water shortage
- Massive drought
- Food shortage

According to IPCC (Intergovernmental panel on climate change) study for countries most at risk from climate related threats, Pakistan rated as

- 7th in flood
- 12th in agriculture

Figure shows the overall effects of global warming that may occur.

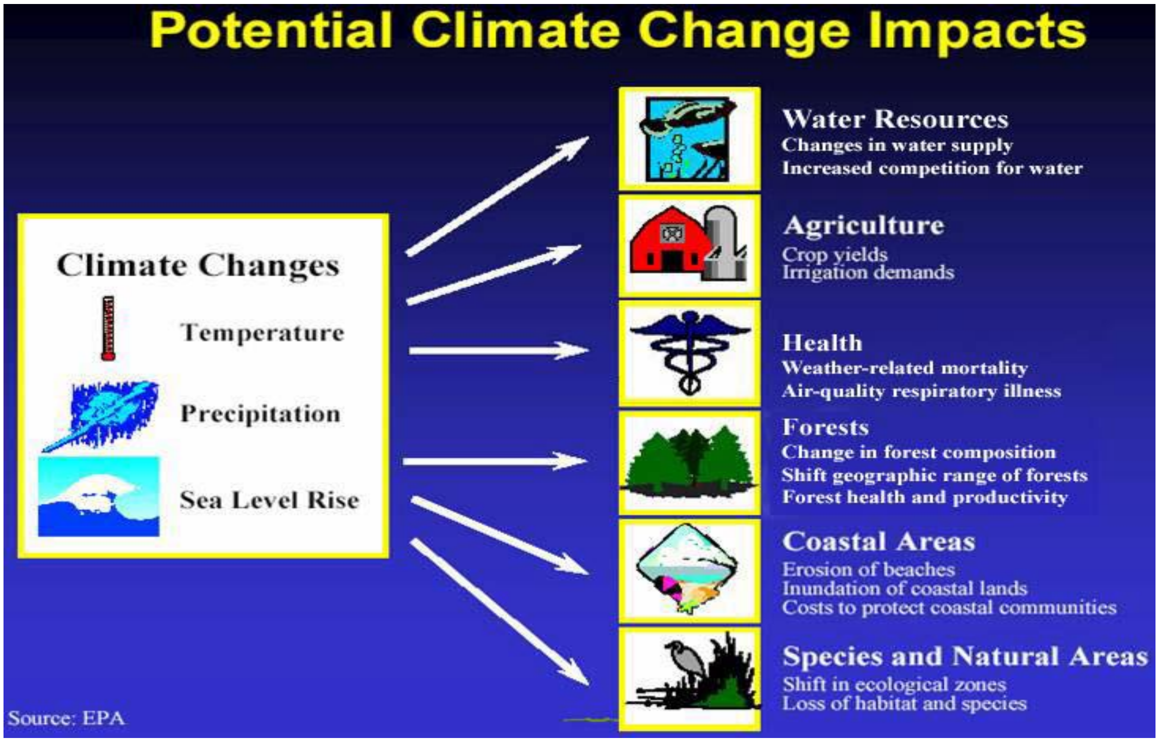

Figure 1-3 Climate Change Impacts [12]

By keeping all these aspects in mind manufacture of air conditioners with natural refrigerant and lower GWP are very crucial to overcome the future climate related threats.

1.1.3 Energy crisis and peak load

Like many of the other developing countries in world, Pakistan is also suffering from energy deficiency. According to the statistics of 2015 the power production was around 15500 MW while the demand was around 21000MW. The main reason of this huge short fall of around 5500MW, there is a huge dependency on the thermal power plants which are operating on high energy terrify, causing the huge circular debut and the source of global warming. [13]

As all the conventional refrigeration units are operated by electricity and according to the estimation out of all the energy produced in the world 15% is consumed only for the air conditioning and refrigeration and if we study the statistics for residential buildings and homes then this percentage increase unto 45%. [2] With the current energy scenario in Pakistan if we consider the summer load then the situation becomes more worse only due to the high demand of cooling and refrigeration. Ongoing energy short fall, presence of large solar radiations and absence of solar cooling systems in Pakistan shows the room for research related to the solar based cooling system for the climatic conditions of Pakistan.

Solar assisted air conditioning systems are one of the most attractive and reliable way to mitigate the environmental and power related issues. On more attractive thing in the solar energy is that our cooling load and solar radiations are in phase which means the time in which maximum cooling demand is required we get the maximum number of incident solar radiations at the same time. If we study the characteristics of the solar thermal collectors, then their efficiency is also increased with the increase in temperature of insolation and atmosphere. Moreover, solar assisted air conditioning systems can also be used for the remote areas which are off grid and there is no access to electricity. By using the solar assisted air conditioning systems, the requirement of fossil fuel will also be decreased which is the good sign for the environment protection because the used natural working fluids and ozone friendly with very insignificant GWP.

1.2 Types of solar air conditioning systems

There are different types of solar air conditioning systems that are introduced or being investigated by the researchers from the whole world. Overall solar air conditioning can be categorized into two main systems i.e. the solar electrical cooling and solar thermal cooling.

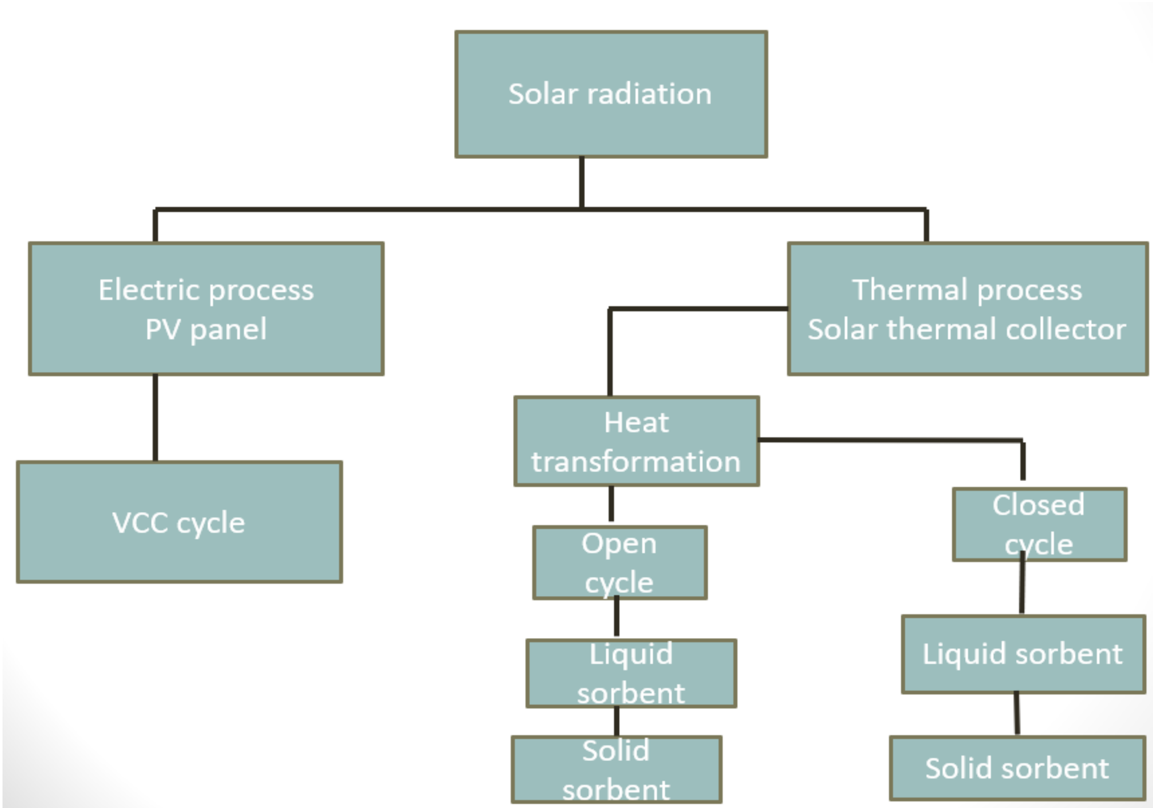

Figure 1-4 Solar Air-Conditioning [14]

Both the processes are discussed in detail in the given section.

1.2.1 Solar electrical air conditioning

Solar electrical air conditioning system consists of mainly two components first is the solar photovoltaic panel and second is the conventional air conditioning device. Apart from this conventional approach there are many other solar electrical systems that are being investigated e.g. Stirling refrigeration and thermoelectric cooling. [15] Each of the system has different pros and cons but usually solar PV based vapor compression is used widely. In the vapor compression cycle electrical power is required to run the compressor which is the most energy extensive part of the machine. That is electrical energy is provided through solar PV panels. PV panel produces the electrical energy by the photovoltaic effect and then couple of regulator and investors are used to regulate the DC output of solar panel and provided that to the vapor compression machine in the form of AC input. A schematic diagram of solar PV system with vapor compression machine is shown in the figure.

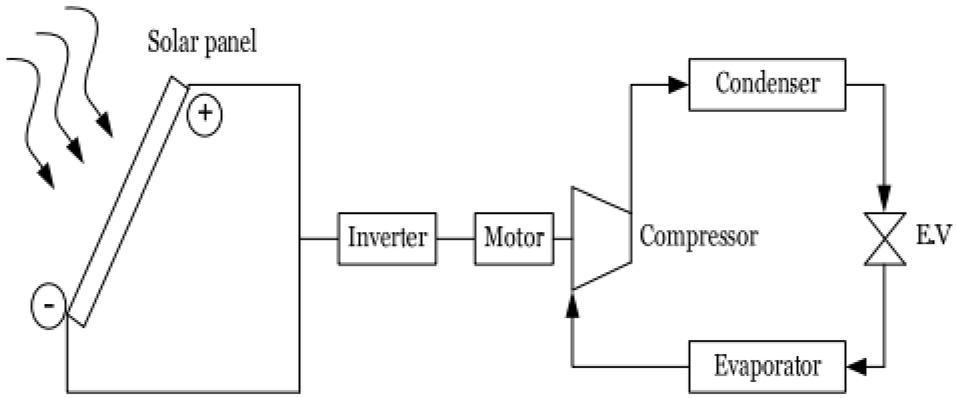

Figure 1-5 Solar PV Based VCC [14]

In these system coefficients of performance depends upon the evaporator temperature which is usually between -5 to 15^{0}C and coefficient of performance varies between 1.1 to 3.3. Condenser temperature is kept normally between 45 to 61^{0}C. [16] In this system battery can also be used for buffer storage to mitigate any transients or for backup in the case of total off gird system.

Pros and cons of this system is given below

Pros

- Low capital cost.
- Higher coefficient of performance.

Cons

- Non-natural working fluid is used which cause the global warming and ozone depletion.

1.2.2 Solar thermal air conditioning

In solar thermal air conditioning system rather than electrical energy, solar heat energy is used for the refrigeration purpose. Depending upon the working principle normally the solar thermal systems can be divided into two types i.e. sorption systems and thermal mechanical systems in which Rankine and Stirling engines are used. In the solar thermal technologies most of the research is taken on sorption technologies in which absorption, adsorption and desiccant system are included. [17] [14]

1.2.2.1 Absorption system

Mostly absorption systems are used in the world for solar based refrigeration or cooling. Schematic diagram of absorption is system is given below.

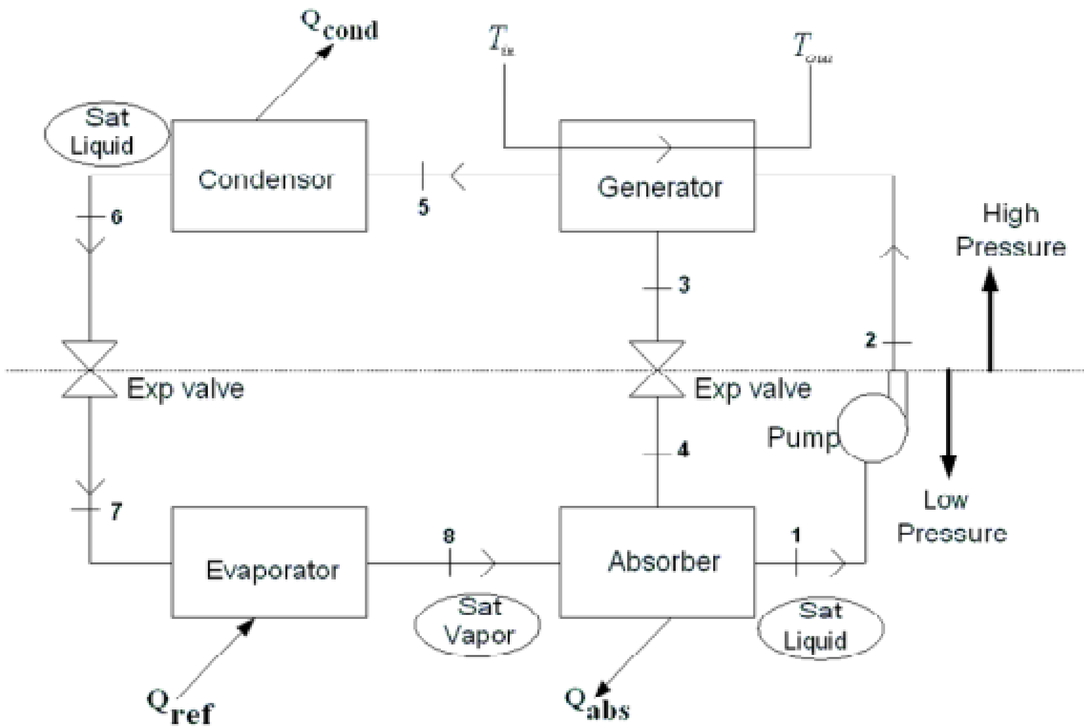

Figure 1-6 Schematic Diagram of Absorption System [1]

Absorption system is a type of closed cycle in which refrigerant is evaporated from a less volatile absorbent, vapors of a refrigerant are condensed in the air or water-cooled condenser. As a result, liquid is moved through the expansion valve and reached to the evaporator. After passing from evaporator refrigerant flows into the absorber. In absorber it is absorbed again in the absorbent and pumped to the generator. In generator refrigerant is evaporated by supplying the heat from heat source. The generator in the absorption cycle works as a thermal compressor and after that need of electrical compressor is eliminated. The energy used by the pump is provided by the auxiliary source but as compared to the compressor it is in a small fraction because the difference in specific density of liquid and vapor medium.

There are three types of absorption system. Single, double and triple effect absorption system. Single effect process contains one heating level while the double and triple effect absorption system have two and stages for the process of vapor generation respectively. Basically, vapor generation is used to separate the refrigerant from the absorbent. As the system is enhanced from single effect to double and triple stage the heat transfer is also enhanced on high temperatures.

Usually in absorption air conditioning system waste heat or heat from co-generation system are used as a heat source. Based on the working fluid there are two types of absorption system i.e. water ammonia and lithium bromide-water.

There are certain pros and cons of the absorption system which are given below.

Pros

- Environment friendly.
- Can run with low grade energy.
- Low maintenance required

Cons

- In the absorption system the required range of the driving temperature is higher as compared to the desiccant or adsorption based cooling systems. [18]The temperature range for single effect absorption cooling varies between 80-100^{0}C, 100-160^{0}C for the double effect and triple effect operates on the temperature above 160^{0}C. For the single homes usually evacuated or flat plat solar thermal collector are used are by using them the concept of triple effect absorption system is not feasible.
- For the absorption cooling ammonia is toxic so it cannot be used for the residential buildings.
- In lithium bromide based cooling there could be a possibility for the crystallization of LiBr which may result in the machine failure. [19]
- High capital cost.
- Low Co-efficient of performance.

1.2.2.2 Adsorption System

Adsorption system is also a type of closed cycle and the difference between absorption and adsorption refrigeration is type of absorbent used. In adsorption cooling there is solid adsorbent used. Schematic of adsorption cycle is shown in the figure.

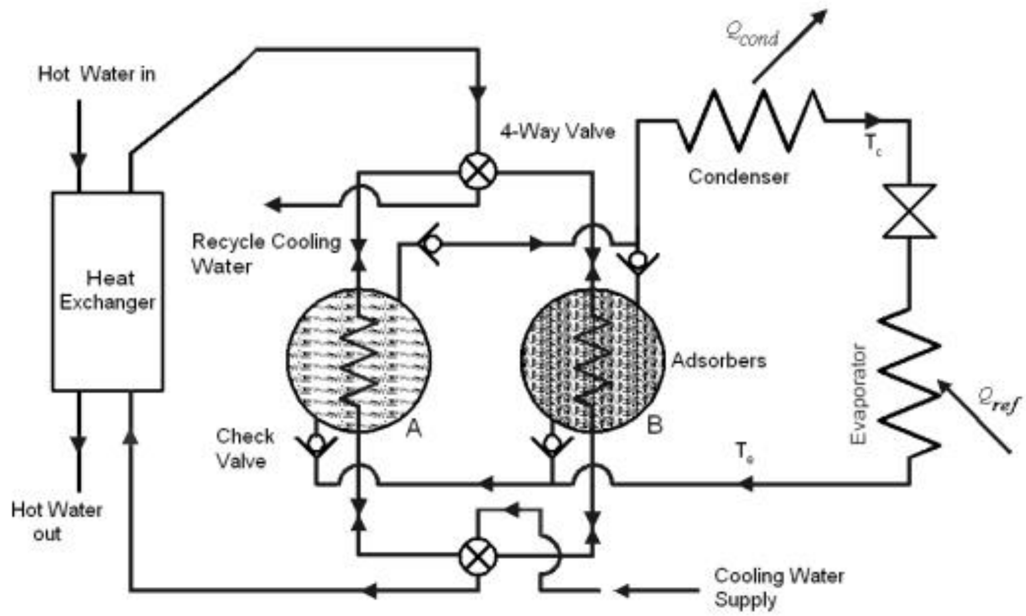

Figure 1-7 Schematic Diagram of Adsorption system [17]

In the adsorption cooling solid adsorbent is used to adsorb the refrigerant exiting from the evaporator in the vapor form. This whole process takes place in a chamber called "adsorber". Heat is supplied to the adsorber, the vapor refrigerant released as a result pressure is increased inside the adsorber. When vapor pressure in the adsorber increased increase than the condenser pressure then refrigerant starts to flow into the condenser and condensed. The working principle between condenser and evaporator is same as the absorption cooling. Usually two or more than two vessels are used to make this process functional.

As compared to the desiccant and absorption based cooling systems there is no major disadvantage of adsorption cycle. It can be functional at the driving temperature of 50^0C which is much lower as compared to the driving temperature required for the absorption cycle. [20] There is a issue of crystallization in the absorption cycle which may lead to the failure of the machine but here in this case there is no crystallization formation because liquid is not used here for sorption process. One of the main advantage of the adsorption cooling is that it is totally environmentally friendly. But the main issue in the adsorption cycle is it low COP which is even less than 0.5 as compared to this single effect LiBr absorption cycle have COP around 0.8.

1.2.2.3 Desiccant based cooling system

Desiccant based cooling system use the open cycle and sorption is used for the dehumidification of the air. A conventional desiccant cycle consists of a rotatory desiccant wheel and couple of direct or indirect evaporative coolers depending upon the climatic conditions where it installed. Desiccant wheel consists of hygroscopic material which may be solid or liquid. Mostly silica gel is used in desiccant wheels but apart from silica gel lithium chloride is also used. [21] [22] [23]

On the basis of working there are three main components in the desiccant wheel system which are, heat source for the process of regeneration of silica gel, rotatory desiccant wheel and evaporative cooler. But according to the composition and configuration of each component there are different possibilities depending upon the requirements. Figure shows the layout diagram of the desiccant wheel system

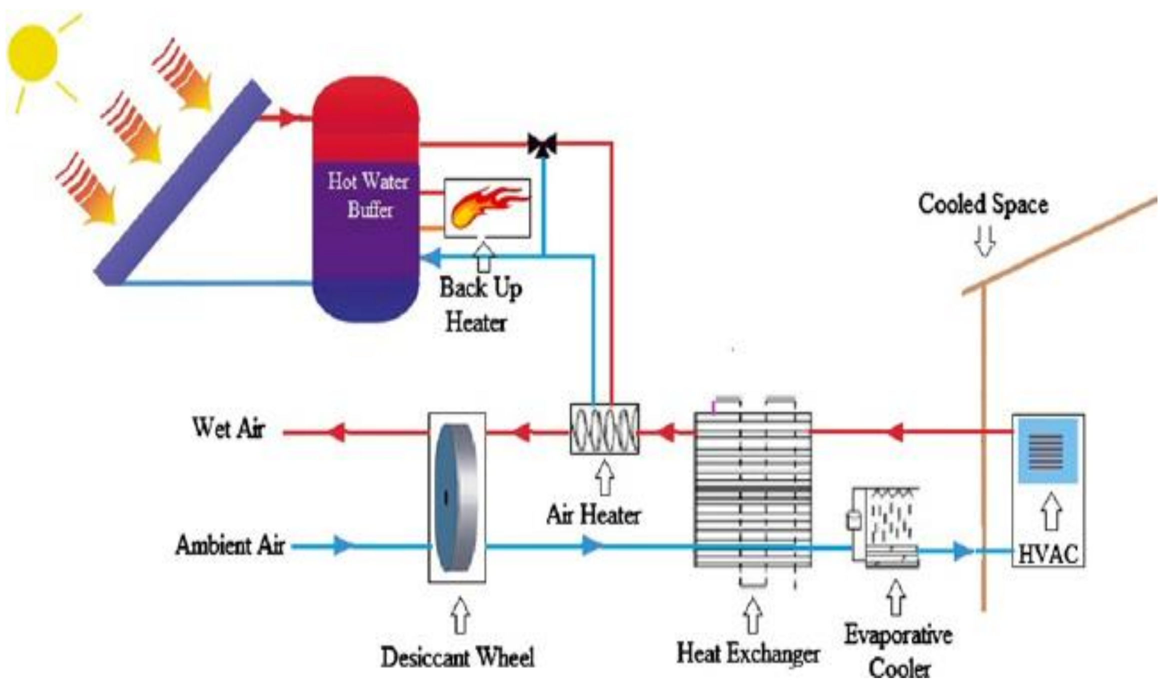

Figure 1-8 Schematic Diagram of Desiccant cooling system [24]

Firstly, there is a rotating desiccant wheel which acts as a dehumidifier and separate the latent load from the sensible load and absorbs the humidity present, in this way cooling system don't have to work on the humidity mitigation. After passing through the desiccant wheel air passed through the heat exchanger which is the air to air heat exchanger in which cool air coming from the room and process air from the desiccant wheel exchange their sensible energies and after passing through it air pass though the evaporative cooler which is may be direct or indirect evaporative cooler depending upon the required conditions. Sometime more than one evaporative coolers are also used and after that air is passed to the conditioned space. On the other hand, after passing through the room exhaust air firstly pass through the air to air heat exchanger and after that passed through the water to air heat exchanger in which this air is heated by using hot water coming from the solar thermal collector or any other heat source. And then this hot air passed through the desiccant wheel by doing the regeneration of silica gel. Remember, the process of regeneration is very important part of this cycle because otherwise desiccant wheel may go to the saturation state. There are certain pros and cons of this system which are given below.

Pros

- Accommodate the sensible and latent loads separately.
- Required low medium temperature for the regeneration process.
- No non-natural working fluid is used
- Environment friendly.

Cons

- Desiccant system cannot cool the supply air below 20^0C in the case of hot climates like most of the cities of Pakistan. For that purpose, auxiliary cooling unit is also used to meet the desired comfort level.
- Bigger and bulkier in size.
- Not viable where the required cooling space don't need a high fraction of ventilation air.
- Low COP as compared as compared to the conventional system.

CHAPTER#2

2 Review of Desiccant Based Cooling

2.1 Introduction:

Cooling load can be divided into two parts. One is related to humidity which is known as latent load and second one is related to temperature and called sensible load. To get the desired comfort conditions air conditioner have to accommodate both of the loads. Conventionally air is cool down by the vapor compression system or the vapor sorption system. Air is cool down below the dew point, in that way water vapors from the air condensed and then we need to reheat the heat to get the desired comfort conditions. In this whole process lot of energy is wasted to cool the air below the dew point. So, it strengthens the idea to accommodate the latent and sensible loads separately.

Almost every material in the world have the capacity to adsorb the water. In commercial use descant based materials are used for this purpose. There are different materials that can be used in the desiccant wheel but mostly silica gel is used for this purpose. Depending on the type of material desiccant wheel can adsorb the humidity from 10 percent to 1100 percent of its weight in dry state. Regeneration of this desiccant material is also important for that purpose we have to heat it from the temperature range 50^0C to 260^0C. This technology can be used for the latent load accommodation and then sensible load can be accommodated separately. This type of process is very useful for the climatic conditions with large humidity level. Because in that type of conditions we have to spend much energy on dehumidification which can be saved by separating the loads and accommodating them separately.

The desiccant materials absorb and adsorb the water vapors due different between the vapor pressure of water between the desiccant surface and the surrounding air. They can be in liquid or in solid states and each type have its own pros and cons. Liquid desiccant materials have low pressure drop across the side while solid desiccants are more compact and less effected by corrosion.

Diverse arrangement technology is used for the desiccant materials. Usually desiccant wheel rotates with 8-10 revolutions/h. From one side air is entered and other side hot air blows out after regenerating the wheel.

2.1.1 Principles of desiccant cooling

In the desiccant cooling air stream passed through desiccant wheel. Humidity is absorbed by the desiccant material and then air is dried to the desired indoor conditions. To keep the whole system in proper working state it is important to regenerate the desiccant material so that it is dried out to adsorb/absorb water in next cycle. This process of regeneration is carried out by heating the material to the specific value of regeneration temperature. The value of reaeration temperature varies from material to material.

Therefore, a desiccant cooling system consist of three main components, dehumidifier, and cooling unit which is usually evaporative coolers and a heat source for regeneration. [25]

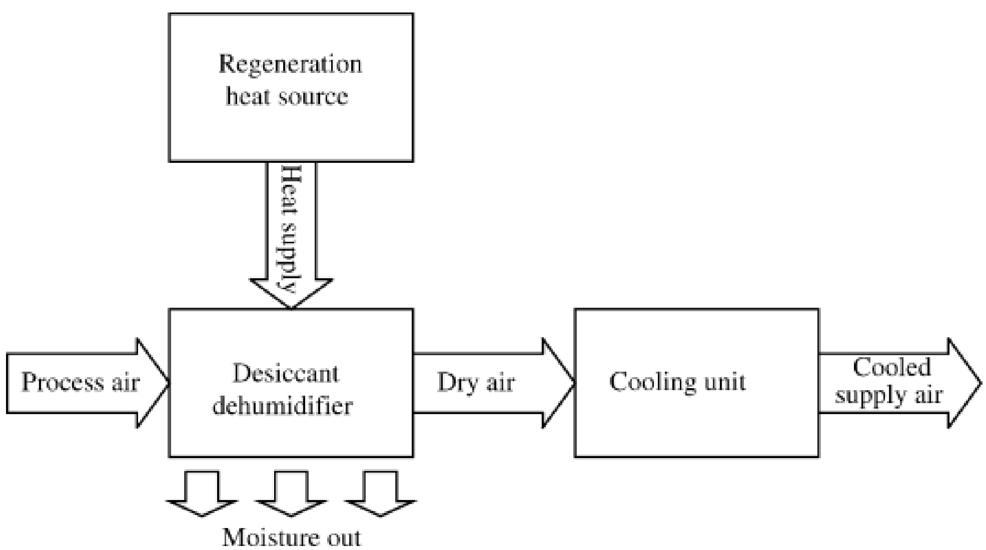

Figure 2-1 Desiccant Cooling [25]

Sensible heat ratio has a strong influence while calculating the efficiency of desiccant cooling system. Sensible heat ratio is defined by the ratio of sensible heat gain to the latent heat gain for the conditioned space. If the value of SHR is low, then it shows that the latent load is more as compared to the sensible load while if the value is high then sensible load has a higher value as compared latent load. The detail of these three components is described in below section.

2.1.2 Desiccant dehumidifier unit

Desiccant dehumidifier usually consists of a slow rotating wheel in which desiccant material is placed. Desiccant bed is regenerated periodically. In the case if liquid desiccant is used then air comes to contact with inside absorber. There are different possible types e.g. packed tower, coil absorber, spray tower and finned tube surface.

2.1.3 Regeneration heat source

To mitigate the moisture absorbed by the desiccant wheel or you can say to dry the absorbent heat is provided by some thermal energy. There are different possible sources for the thermal energy e.g. solar energy, heat from heat recovery units, by natural gas or any other possible source.

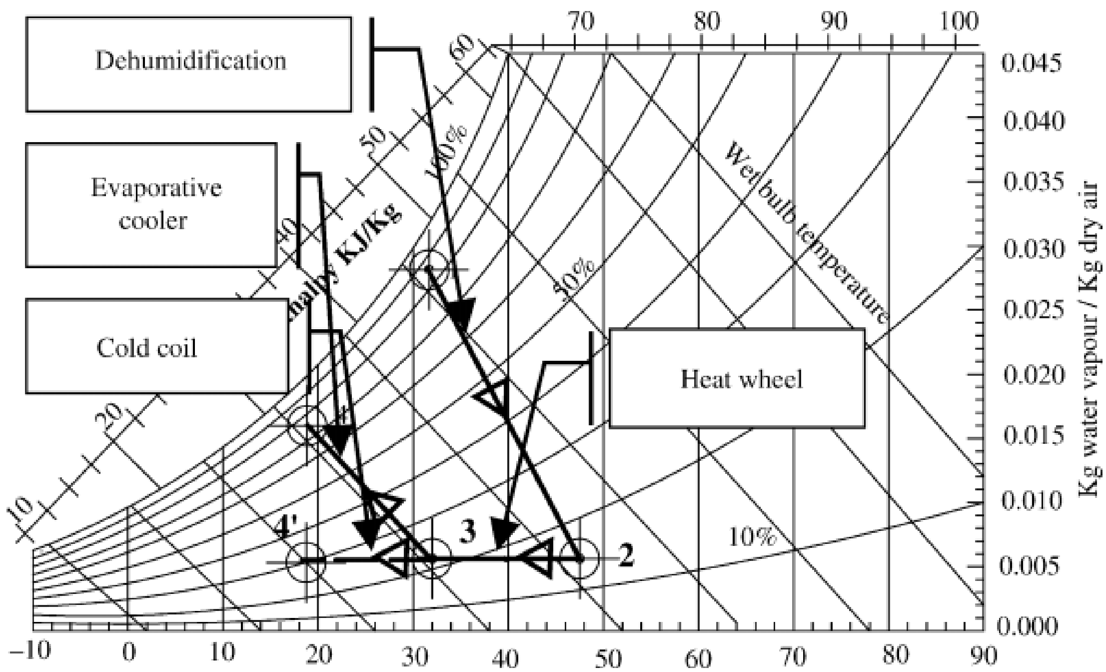

Figure 2-2 Psychometric Chart for Desiccant Cooling [25]

If liquid desiccant material is used, then the regeneration heat energy is provided to the desiccant solution. Hot air is blown through the regenerator and it carry out the moisture which is absorbed already.

2.1.4 Cooling unit

In the conventional desiccant cooling systems evaporative coolers are used as cooling unit but sometimes cooling coil may also be used. Direct evaporative coolers can be further divided into two parts one is the direct evaporative coolers and others are indirect evaporative coolers. Humidity level is increased in the direct evaporative cooler while the humidity level remains same in indirect evaporative coolers because there is no direct contact between the water and air stream. Basic function of the cooling unit in the desiccant based air conditioning is to accommodate the sensible load while the latent load is accommodated by the desiccant wheel. In desiccant air conditioning system usually, air to air heat exchanger is used before the evaporative coolers for the precooling. Air to air heat exchanger and evaporative cooler when combined they are called cooling unit.

2.2 Literature Review

Kern and Russell invented the idea for combining the photovoltaic and thermal system named as PV/T. [26] In 1970s for the very first time flat plate PV/T collector was introduced by the Martin Wolf due to the issue if decrease in efficiency of PV module with the increase in temperature. In that study air and water based PV/T systems were analyzed and found a electrical efficiency around 6.8% and thermal efficiency around 40.4%. During the time span of 1980s there was a huge development takes place on the PV/T technology and photovoltaic thermal technology grew rapidly. [27] [28]

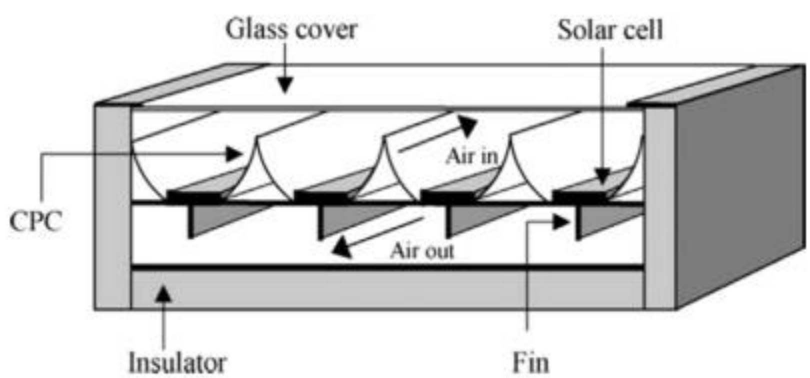

Figure 2-3 Double pass PV/T collector with CPC and fins [28]

Depending upon the types photovoltaic thermal systems are analyzed as amorphous, monocrystalline, polycrystalline and thin film cells. Each of the type have different characteristics and features. Flat plat PV/T collector took more intention from researchers as compared to the concentrated PV/T collectors.

Depending upon the types of PV cells, covered panels type of fluid and type of concentrator PV/T collectors have different types and categories which are explained a figure below.

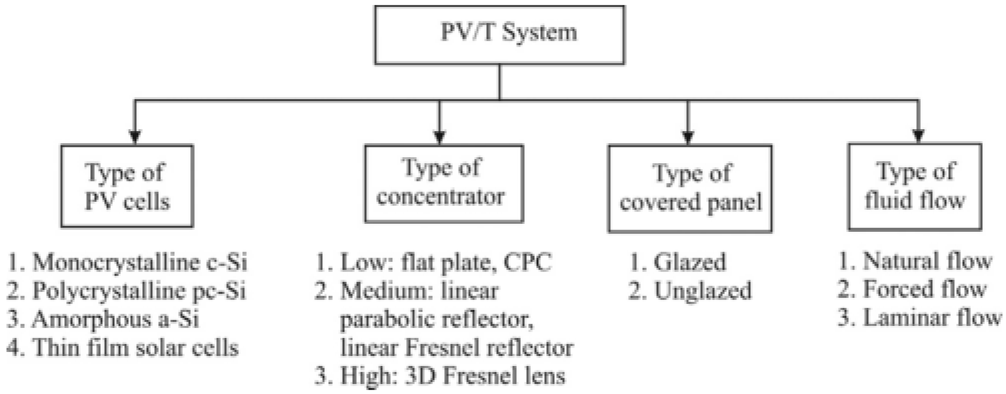

Figure 2-4 Classification of PV/T system [28]

PV/T collectors are divided into three classes. First class of PV/T collectors are in more interest by the researchers. Compound parabolic concentrators and flat plate collectors are categorized in the class one. Linear Fresnel reflectors and linear parabolic reflectors are present in second class while 3D Fresnel lens are present in third class of PV-T collectors. Glazed PV/T collectors have more thermal efficiency than unglazed PV/T collectors. In the PV/T collectors with air as working fluid if air flow rate is increased then temperature of panels decreased, and electrical efficiency increased. [29] [30]

Alizadeh et al. [31] designed a system in which solar collector is used for the regeneration of the desiccant wheel. Calcium chloride was used as a desiccant medium for dehumidification of the air. Performance of desiccant wheel was investigated by changing the air flow rate. They concluded that the performance of regenerator was improved by increasing the air flow rate. Yadav [32] simulated the desiccant wheel based air conditioning system and compared its results with the vapor compression cycle. Liquid desiccant was used as a dehumidifier in that system. He concluded that if the latent load comprises 90% of the total load then 80% of the energy saving can be obtained from the desiccant based air conditioning system as compared to the conventional vapor compression cycle. Dai et al. [33] did a comparative study of desiccant cooling by using the direct and indirect evaporative coolers. Mazzi et al. [34] did a simulation based study for the comparison of the desiccant wheel based cooling as compared to the conventional vapor compression based cooling. Result obtained from that study was, there is 35% energy saving in desiccant based cooling as compared to the vapor compression system. He also concluded that if the regeneration of the desiccant wheel is done by the waste heat recovery then the energy saving can increase up to 87%. They found that the cooling power requirement can also be decreased by using the evaporative coolers with the desiccant wheel but the cost saving depends upon the cost of electricity in that particular region where the study is carried out. That's why percentage of cost saving varies from place to place and country to country. Hennig [35] studied a desiccant system. According to evaluation of performance of different parameters he found that desiccant system save energy up to 50%. Shen et al. designed a desiccant system and separated the carbon dioxide and water vapors from the air after process from desiccant wheel. They studied the optimization of performance of system by varying different parameters. Techajunta et al. [36] designed a desiccant based system and used silica gel as a dehumidifier. They used the solar energy for the regeneration of the silica gel. As a result of this study they found that the regeneration of desiccant material is strongly dependent upon the solar radiations and the dependence on the air flow rate is comparatively weak. Sanjev et al. studied the liquid desiccant cooling system and consider the impact of wetness in dehumidifier and regenerator. That study was found to be useful for designing the view point and by using that study more accurate sized contactors can be made. Kadoma et al. studied the impact of the speed of desiccant wheel, regeneration temperature and the mass flow rate of the air on the coefficient of performance of the cooling system. They found that if the mass flow rate of the air is increased then the COP of the system decreased. Shyi-Min designed a standalone solar desiccant based air conditioning system. He used the concept of nocturnal natural cycle. Fathalah et al. investigated the heat recovery system. It was a solar based absorption cooling system with the lithium bromide and water as a working fluid. Heat was recovered from the condenser. By the installation of the heat recovery system the co efficient of performance of the system was improved up to 1.2 times. Evaporative temperature of the cycle was also increased to 19.3^{o}C from 11.5^{0}C. Arshad et al. studied the dehumidifier with the liquid absorber and modeled the mathematical model. The found that if NTU of the heat transfer increased between the desiccant solution and process air then the performance of the system is improved. Adam designed a simulation based study for the desiccant assisted cooling system. $CaCl_2$ was used a desiccant. Performance of the system was studied under the consideration of different parameters. In that study sensible heat ratio of space and effectiveness of the heat exchanger was studied. Ratio of the flow rates of the desiccant liquid and air were also studied. He found that the ratio of both flow rates has minor effect on the performance of the system. If the inlet temperature of the desiccant is increased up to certain level, then the performance of the system increased. The coefficient of performance of the system also increased with the decrease in SHR. If the effectiveness of the heat exchanger decreased, then the performance of the system also decreased. Khalid et al. [37] developed a desiccant based simulation model by using the TRNSYS for the climatic conditions of Pakistan. He designed the system with different configurations and found that the regeneration temperature can be decreased up to 15% if indirect evaporative coolers are used as compared to the direct evaporative coolers and study found that a huge amount of energy is saved by the desiccant cooling system as compared to the conventional vapor compression cycle. They found that the payback period of the system was around 14years. Al Alili et al [38] simulated the solar assisted air conditioning system for the climatic conditions of Dubai. They used a PV/T collector for the desiccant system. Instead of the evaporative coolers he used a vapor compression chiller for the accommodation of the sensible load while the latent load was accommodated by the desiccant wheel. This study found that more level of comfort can be obtained with the high COP by this system for the area of high humidity as compared to the any other solar based air conditioning system.

Researchers also took a keen interest in the search of the new desiccant materials with the better absorption capacity. Aristov developed the materials that are hybrid in nature and formed by the process of impregnation of silica gel or vermiculite with calcium chloride or lithium chloride that are basically the hygroscopic salts. He developed a material with this process whose capacity against the sorption is found to be triple as compared to the host material. Liu et al. also developed a material by the impregnation of the silica gel with calcium chloride and used to absorb the humidity from the air. William studied the ageing phenomena of the desiccant based materials.

Apart from the use of the desiccant based materials for the comfort, desiccant cooling is also used for the super markets and cold stores. Desiccant cooling is the most attractive option where the fresh air is needed in every cycle like for the conditions of hospitals.

Thorpe et al. designed a desiccant based air conditioning system. Solar energy is used to regenerate that system. They used this system for the storage of the grains. It was a very energy efficient system and give a cooling 50times as compared to the electrical energy used. Dai et al. developed the hybrid desiccant based air conditioning system in which rotatory desiccant wheel was used. That system was used to store the grain and co efficient of performance was found to be 0.4.

2.3 Research objectives

As cleared from the literature review, desiccant wheel air conditioning system are in use all over the world. But very few studies regarding PV/T based desiccant wheel air conditioning system are available. The design parameters for desiccant system using the photovoltaic thermal panels are not yet standardized and documented properly. Moreover, there is no such a study available for the climatic conditions of Pakistan in which PV/T based air conditioning system is discussed. Therefore, there is a need to investigate the dynamic parameters of such system under the local climatic conditions. In this study, various design parameters of PV/T based desiccant air conditioning system are discussed and optimized. Simulations are done by using the TRNSYS 17.0 and calculations done for the whole summer season. Performance parameters are solar thermal and electrical fractions, solar collector's thermal and electrical efficiencies, primary energy savings and optimized size of the storage tank.

CHAPTER#3

3 System Configuration and Modeling in TRNSYS

3.1 Introduction

In this study three different configurations are discussed for desiccant based solar air conditioning. Each of the configuration is shown below.

3.1.1 System's discussion and configurations

Cooling load can be divided into two main parts. First is the sensible load which is directly related to the temperature of the air while the second portion is related to the latent load. Latent load is more related to humidity and moisture removal to the desired comfort level. For the more humid climatic conditions a huge amount of electrical power is consumed to entertain the latent load in the conventional air conditioner. Because in the conventional vapor compression based air conditioning systems moisture is removed through condensation by cooling the air up to low temperatures and in some cases even we have to reheat the air to get the desired temperature range. For power saving the concept of hybrid air conditioners is introduced. Hybrid air conditioners are capable to deal the sensible and latent loads separately. In this research desiccant wheel based air conditioning system is discussed and analyzed.

In the designing of the required there are two main loops. First loop consists of solar photovoltaic-thermal based collector, stratified storage tank, couple of pumps, invertor, regulator and a battery.

While in the second loop a desiccant wheel, couple of heat exchangers, direct evaporative cooler, vapor compression based air conditioner and load is presented. The given system is designed for the load of 2.5Tons that is equal to 8.79KW.

Electrical and thermal energy obtained from the solar PV-T collector are stored in battery and stratified storage tank. Fresh air passed through the desiccant wheel which removes the moisture from the air and temperature of air is increased with a small amount due to heat of adsorption. After passing through the desiccant wheel air is passed from the heat exchanger while on the other side cool air coming from the room is supplied. Both air streams will exchange the sensible energy over here and the temperature of the fresh air will be decrease to the certain amount. After passing through heat exchanger air will pass through the direct evaporative cooler for cooling. Room thermostat will check the temperature that if we get the desired room temperature of 23^{o}C after the DEC then the vapor compression unit which is used as the auxiliary air conditioner will not turn on otherwise the auxiliary air conditioner will be turned on to achieve the desired comfort conditions.

Desiccant wheel after absorbing the moisture from the air may go to the saturation level and if this happens then no more moisture can be absorbed through it. To avoid this problem air coming out from the room after passing through the air to air heat exchanger where it cools down the fresh air will go to the water to air heat exchanger. In this heat exchanger hot water from storage tank will flow and heat the air. There is the thermostat that will check the temperature of the heated air. If the temperature of 60^{o}C is achieved after passing through the heat exchanger, then the auxiliary heater will not turn on. Otherwise auxiliary heater will be turned on and help to get the desired temperature level of air. This heated air will pass through the rotary desiccant wheel and used as the regeneration air.

There are three different configurations discussed over here. In C-1 we check the temperature of regeneration air and if it is not 60^{o}C then the auxiliary heater is turned on. While in C-2 the temperature of hot water is checked. We set the temperature of 65^{o}C for the hot water and if it is not achieved then then the boiler is turned on. While C-3 is similar to the C-2 but there is no direct evaporative cooler in this configuration. C-3 is based on the research paper published in the Applied Energy [38]

All the block diagrams are shown below.

3.2

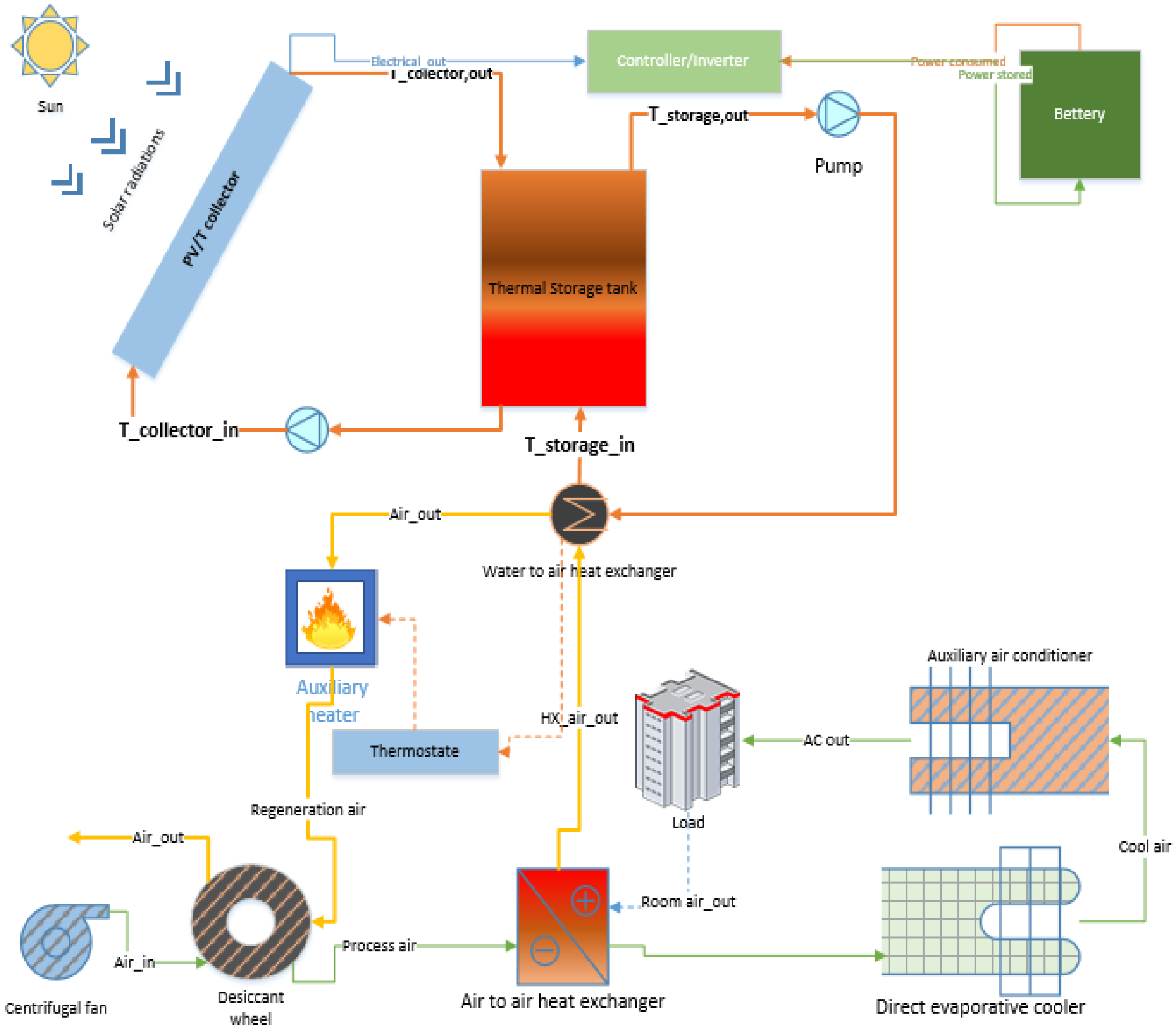

Figure 3-1 Schematic of Configuration-1

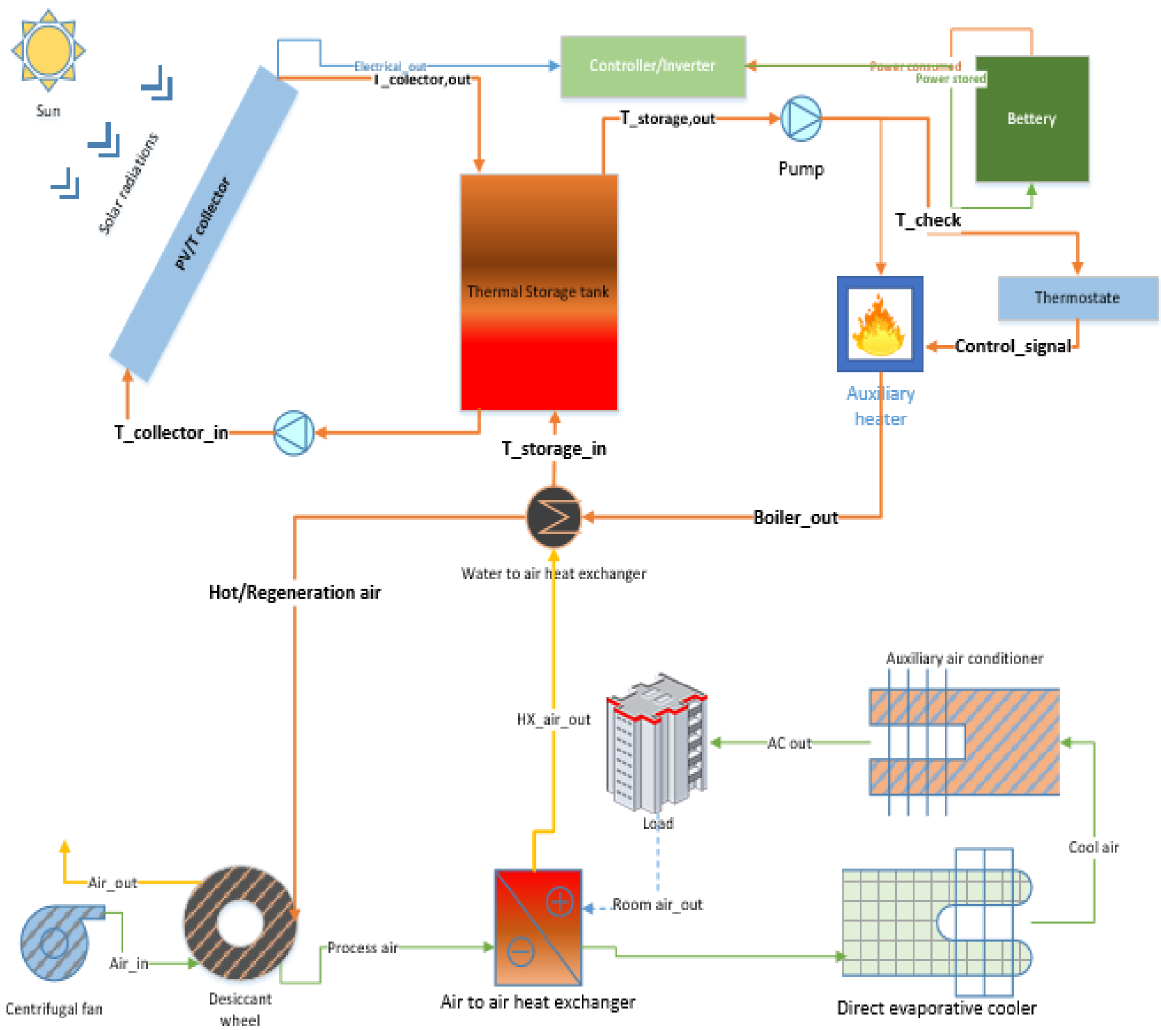

Figure 3-2 Configuration-2

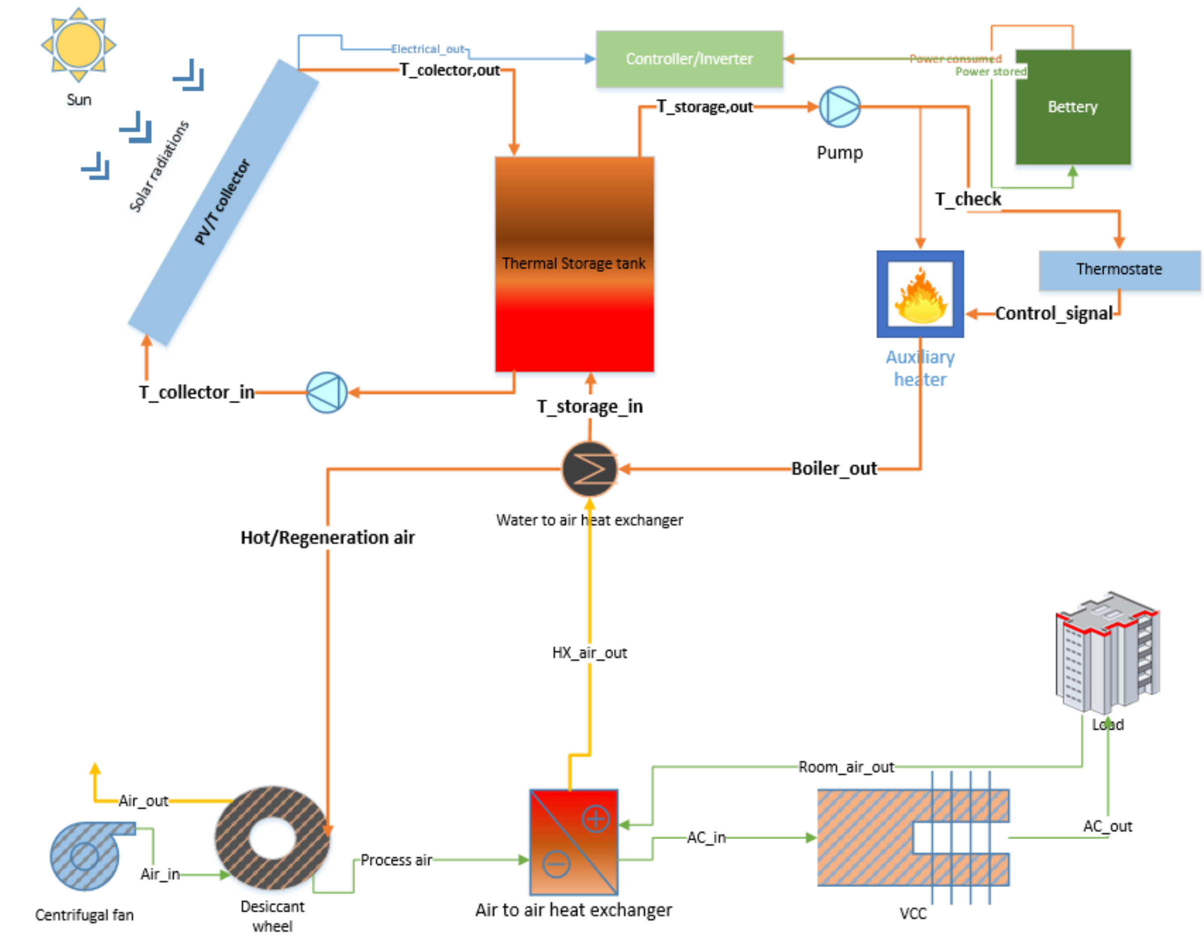

Figure 3-3 Configuration3

3.2.1 Modeling in TRNSYS

Simulation of the given system is done by using the TRNSYS 17.0. TRNSYS is a very dynamic tool for simulations and whole system performance can be simulate for complete one year. Each component used in TRSYS have its own predefined characteristics equations. Figure shows the pictorial view of TRNSYS model.

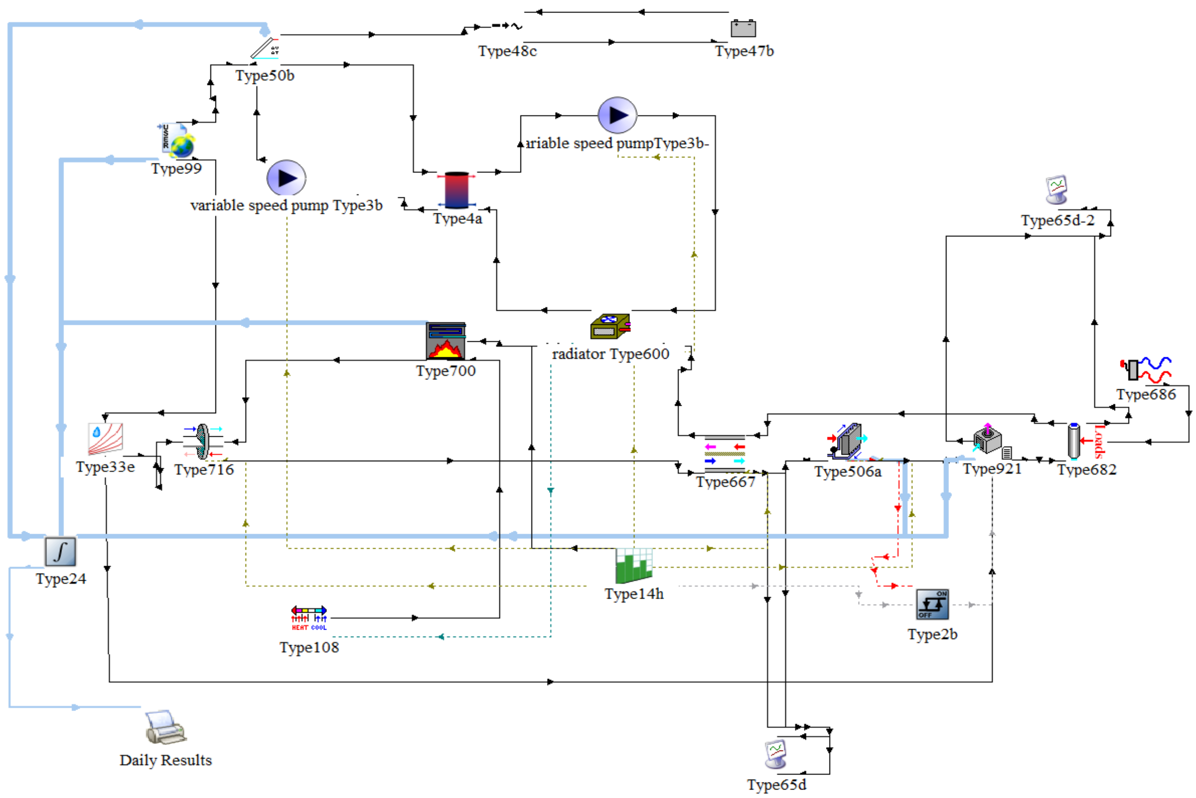

Figure 3-4 Pictorial View of TRNSYS Model C-1

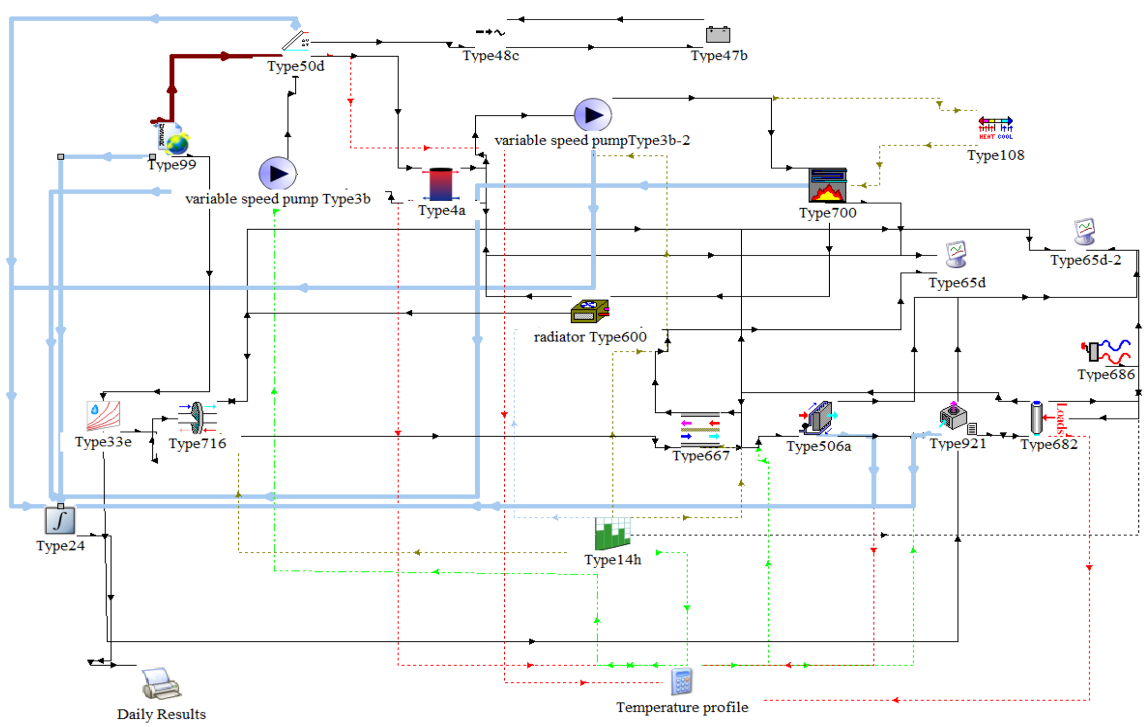

Figure 3-5: Pictorial View of TRNSYS Model C-2

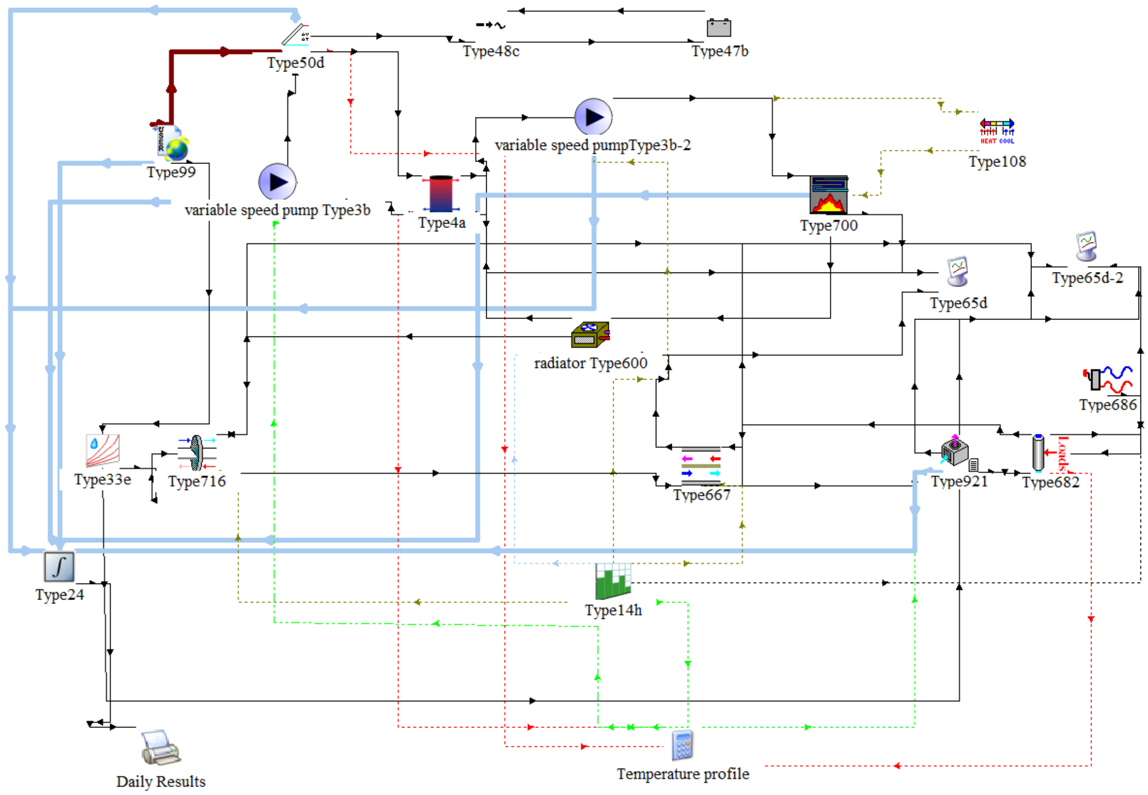

Figure 3-6: Pictorial View of TRNSYS Model C-3

3.3 System component

To develop complete a solar assisted desiccant cooling system model in TRNSYS, following components are used.

3.3.1 Solar photovoltaic/thermal collector (Type 50)

In this type standard PV module is added in a standard flat plat collector. The principle of working of type 50 PV/T collector is based on the modified version of combined collector which use the working principle of Florschuetz Type50 have four different modes in TRNSYS. Mode 1 to 4 are used for the simulations of the flat plate collectors while 5 to 8 are used for the simulations of the concentrating collectors.

In our case we are using mode 4 which is the combination of the mode 2 and 3. U_L is calculated from this following equation for this mode.

$$U_{L,j} = \frac{3.6}{\frac{N_G}{\frac{C}{T_{P,j}}\left(\frac{T_{avg}-T_a}{N_G+f}\right)^{.33}}+\frac{1}{h_w}} + \frac{3.6\sigma({T_{avg,j}}^2+{T_a}^2)(T_{av,j}+T_a)}{\frac{1}{\varepsilon_p+\frac{1}{0.05N_G(1-\varepsilon_p)}+\frac{2N_G+f-1}{\varepsilon_g}-N_G}} + U_{be} \qquad \text{Equation 3-1}$$

The parameters that are used in PV/T panels are given below.

Parameter	Value	Unit
Mode	4	---
Collector area	15	m^2
Number of glass covers	1	---
Collector plate emittance	0.09	---
Collector slope	45	degrees
Temperature of cell reference efficiency	25	^oC
Packing factor	0.8	---
Cell efficiency	0.2	----

Table 1 (PV/T parameters)

3.3.2 Weather data reading and processing:

In the TRNSYS type99 is used for reading the weather data. Weather data file is downloaded from the metreonorm software and this is a link in type 99 for the external file where it is placed. It is used to convert the data in desired format for the solar radiations that are obtained on the surface of tilted collector. The ground reflectance is taken as 0.2 for the ground while if the ground is covered by the snow then is value of reflectance is taken as 0.7.

3.3.3 Lead storage battery

As the desired system run online during the office timings from 9 to 5 but the solar radiations during summer are even available before 9 and after 5. The extra power generated during those hours can be stored by using the battery and then can be using during operational time to mitigate the transients and in the of unavailability of sun in cloudy conditions.

In the TRNSYS library type 47 is used for the modeling of the lead storage battery. There are different modes that are available for the different conditions. If the battery voltage, state of charge and current is to investigate then type 47c is used which is basically operate with the conjunction with power conditioning and solar cell. In Type47 a, b and c power is used as the input. While in d and e current is used as current.

The parameters used in type 47c are given below.

Name	Value	Unit
Mode	2	---
Cells in series	12	---
Charging efficiency	0.9	---

Table 2 (Battery parameters used)

Number of cells in series shows the voltages of the battery. 12volt battery contains 6 cells in series while the 24volt battery contains 12 cells connected in series.

3.3.4 Inverter/ Regulator:

In the closed circuit of PV panel couple of regulator and inverters are required to use the electrical output in the smooth way. PV/T collector have the imbedded PV panel in it which gives the electrical output as a DC current. Inverter is used to convert the DC output of the PV panel to the AC format. The alternating current can be used to run the home appliances. Excess energy is stored in the battery again in the form of the direct current. While to convert from the AC to DC a regulator is used. In TRNSYS library inverter and regulator is embedded in a single component shown by the type47.

There are different parameters that can be controlled and changed in the library of type47 which are given below

Name	Parameter	Unit
Regulator efficiency	0.78	---
DC to AC efficiency	0.96	---
Power output limit	3	KW
AC to DC efficiency	0.8	---

Table 3 Inverter/regulator

3.3.5 Variable Pump:

Two variable pumps are used for the mass flow rate of water. One pump is used in the solar collector loop while second pump is used in the loop on the load side. In the TRNSYS library type3 is used for the variable load pump.

The is a control function which generates the value of 0 or 1 depending upon the requirement of operation. Flow rate of the variable pump can be changed depending upon the requirement of the user. The parameters for the variable pump are

Name	Value	Parameters
Maximum flow rate	200 (can be vary)	Kg/hr
Fluid specific heat	4.190	kJ/kg.k
Maximum power	745	W
Conversion coefficient	0.05	---
Power coefficient	0.5	---

Table 4(Variable Pump)

The fraction of the pump power that is converted to fluid thermal energy is called the conversion coefficient.

$$T_{out} = T_{in} + (f * Power)(m^{\cdot} * C_p) \qquad \text{Equation 3-2}$$

3.3.6 Stratified Thermal Storage Tank

TRNSYS component Type 4a is used to model hot water storage tank which is thermally stratified. For this purpose, multi-node approach is used in this component in which the tank is divided into N sections or nodes and energy balances for each node are written. Consequently, N differential equations are obtained that can be solved for the temperatures of N nodes as function of time. The flow toward collectors always leaves from bottom node and flow towards load always leaves from top node. In this simulation tank is divided into ten number of nodes and constant heat loss coefficient 3 KJ/hr m^2 K is used in all simulation.

An energy balance of ith tank segment can be described as

$$\dot{m}_i C_{pf}\frac{dT_i}{d_t} = \alpha_i \dot{m}_h C_{pf}(T_h - T_i) + \beta_i \dot{m}_L C_{pf}(T_L - T_i) + UA_i(T_{env} - T_i) \quad \text{Equation 3-3}$$

$$\begin{cases} \gamma_i(T_{i-1} - T_i)C_{pf} & \text{if } g_i > 0 \\ \gamma_i(T_i - T_{i+1})C_{pf} & \text{if } g_i < 0 \end{cases}$$

3.3.7 Water to air heat exchanger:

Type600 is used as a water to air heat exchanger in the TRNSYS library. It is basically a two-pipe fan coil system which can mix the two air streams and then passed them through the coil in which hot or cold water is flowing depending upon the requirement. Type600 operates with the free-floating coil system which means there is no control on the outlet temperatures of air or water.

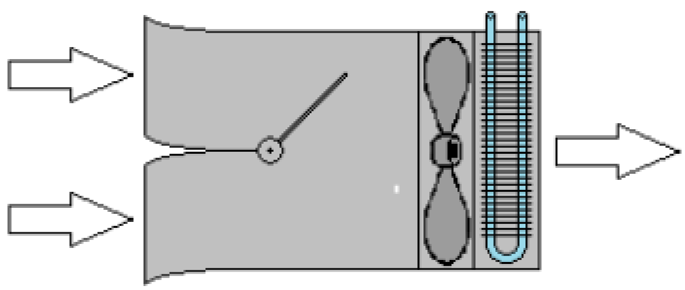

Figure 3-7 (Two-pipe fan coil schematic)

The is a control function in the type600 which generates the on/off signal to control the fan. In our scenario a dedicated controller is used to provide the control signal during 9am to 5pm. There is an option of specific heat of the liquid stream in the parameters of the type600 is present which can be can be vary depending upon the liquid used. Humidity mode can also be changed from 1 or 2. Mode 1 is used if the input humidity ratio is used while if the % relative humidity is considered then mode2 is used.

Two different modes of the two-pipe fan coil system are given below.

3.3.8 Heating mode:

If the liquid temperature entering the coil is more than the air temperature which is leaving from the fan then the mode is considered as the heating mode. Different iterations are used to get the maximum heat transfer between two working fluids.

The energy that is transferred from the air to liquid stream can be calculated from the following equation.

$$\dot{q_{liq}} = \dot{m_{air}}(1 - f_{bypass})(h_{air.cool} - h_{air.fan}) \qquad \textit{Equation 3-4}$$

While the T_{liq} can be calculated from the following equation

$$T_{liq.new} = T_{liquid.in} + \frac{q_{liquid}}{m_{liquid}CP_{liquid}} \qquad \textit{Equation 3-5}$$

The calculated temperature from this equation and supposed outlet temperatures are compared. If the difference between the two temperatures are in the range of 0.01C then model is assumed to be converging and the part of air streams that come to saturation with the average temperature of coil is mixed with the air that came out by bypassing the coil. The final enthalpy of the given stream is given as

$$h_{air,out} = h_{air,coil}(1 - f_{bypass}) + h_{air,fan}f_{bypass} \qquad \textit{Equation 3-6}$$

3.3.9 Cooling mode

If the temperature of the liquid flowing inside the coil is less than the temperature of air stream coming from the fan, then the fan coil system is called in the cooling mode. The working principle of the cooling and heating mode are the same. The temperature that is supposed for this case is given as

$$T_{liquid,out} = T_{liquid,in} + \frac{T_{liquid,in} - T_{air,fan}}{CP_{liquid}} \qquad \textit{Equation 3-7}$$

In the cooling mode same as heating mode iterations are continued and new outlet temperature and supposed temperatures are compared until the difference between the values of the both streams reach to the 0.01C.

3.3.10 Auxiliary boiler

Air after heating from the water to air heat exchanger goes into the auxiliary boiler. In our model TYPE700 is used from the TRNSYS library for fulfilling the energy demand from the auxiliary source. Controller is attached which check the temperature of the incoming air and if this temperature is below 60°C then controller turns the signal on which turned on the boiler to reach the stream on desired temperature. Fluid specific heat can be changed depending upon the working fluid running through the boiler. The parameters used are

Name	Value	Unit
Rated Capacity	20000	KJ/hr
Fluid specific heat	1.008	KJ/kg.k
Minimum turn down ratio	0.2	---

Table 5: Auxiliary Boiler Rating

Here the rated capacity can be changed depending upon the requirement of heat. But fluid specific heat is the important parameter which should be changed depending upon the type of the used fluid stream. The minimum turns down ratio is explained as "The minimum operating part-load ratio for the modulating boiler. If the calculated part load ratio for the boiler (load/capacity) is less than this parameter, the boiler will operate at the minimum turn-down ratio and the outlet temperature will exit the boiler greater than the desired setpoint."

The energy consumed in the TYPE700 depends upon the efficiency of the combustion and boiler that are defined. When the boiler is turned on then the heat energy required to reach the temperature to the desired level is calculated from the following equation.

$$\dot{Q}_{boiler} = \dot{m_f} C_{Pf}(T_o - T_i) \quad \textit{Equation 3-8}$$

m_f is the mass flow rate wate the Cp is the specific heat respectively.

3.3.11 Psychometrics

In the TRNSYS library type33 is used for the psychometric conditions. There are different types of type33 depending upon the known parameters of dry bulb, dew point, enthalpy, humidity ratio, relative humidity or the wet bulb temperature.

In our study type33e is used. TYPE33e use the dry bulb temperature and the relative humidity. From the weather file type99 these two values are obtained as the output which are provided as input to the type33 respectively.

3.3.12 Dehumidifier

In current study rotating desiccant wheel is used as the dehumidifier. This type of desiccant system uses the silica gel for the absorption/adsorption of water vapor from the air stream. In TRNSYS library type716 is used as a rotatory desiccant wheel. Psychometric conditions are used as the input in the desiccant wheel. Control function is used to control the running time of desiccant wheel. From 9AM to 5PM control function set the input signal 1 which turns on the desiccant wheel in rotatory position. Mode is selected on the bases of the operational requirements. Mode 1 is used if humidity ratio is used as a input while if the percent relative humidity is used as the input from the psychometric conditions then mode2 is used. Air flow rate effects the process air temperature and humidity ratio. By doing couple of iterations optimum flow rate is find.

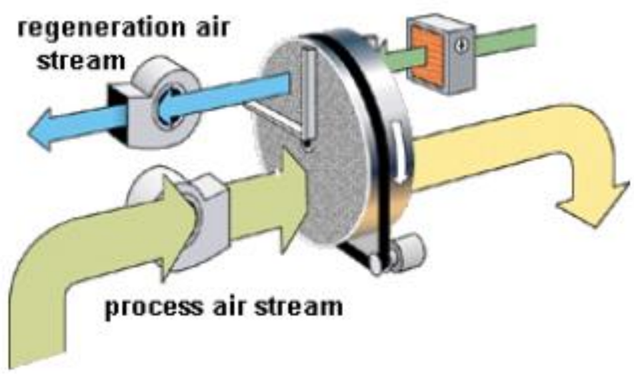

Figure 3-8 desiccant wheel

The performance of this rotatory desiccant wheel is evaluated based on the equations for F1-F2 which was developed by Jurinak. The equations developed for the silica gel based desiccant wheel by the Jurinak are given as

$$F1 = \frac{-2865}{T^{1.490}} + 4.344\omega^{0.8624} \quad \text{Equation 3-9}$$

$$F2 = \frac{T^{1.490}}{6360} - 1.127\omega^{0.0969} \quad \text{Equation 3-10}$$

T is measured in K while $\boldsymbol{\omega}$ is measured in $kgH_20/kgAir$.

3.3.13 Air-air heat recovery wheel

Heat recovery wheel is used to transfer the sensible energy between two air streams i.e. two air streams at different temperatures pass through it and they exchange their temperatures between each other. In TRNSYS library type667 is used as air to air heat recovery wheel. TYPE667 use the constant effectiveness method for modeling this system with the minimum capacitance. In this model both air streams can be flow in any configuration i.e. parallel, counter or cross flow. With the sensible aspects latent energy aspects can also be studied in type667.

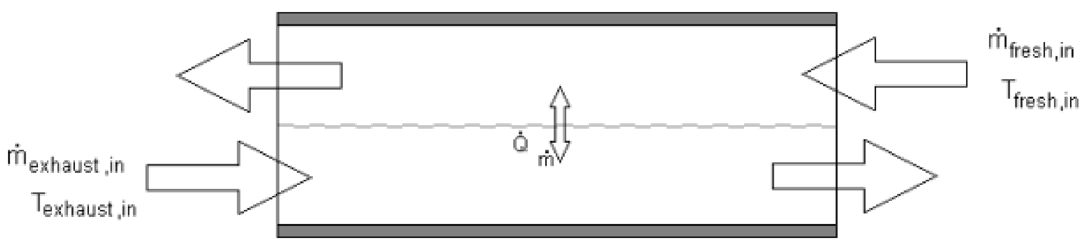

Figure 3-9 Air-Air heat recovery wheel

Sensible energy transferred through type667 can be calculated from the following equations

$$\dot{Q}_{sensible} = \text{€}_{sensible}\, C_{min}(T_{exhaust,in} - T_{fresh,in}) \qquad \textit{Equation 3-11}$$

C_{min} is obtained from the product of specific heat and mass flow rate of the two streams.

While the latent energy transfer is obtained from the following equation.

$$\dot{m}_{transfer} = \text{€}_{latent}\dot{m}_{min}(\omega_{exhaust,in} - \omega_{fresh,in}) \qquad \textit{Equation 3-12}$$

m_{min} is the minimum from the two air flow rates. If the resulted mass flow rate is in negative, then moisture or the latent energy is transferred from the fresh air stream to the exhaust air stream.

Humidity mode is to be selected same as type716 as discuss earlier.

3.3.14 Direct evaporative cooler

Direct evaporative cooling is a process of cooling or removing heat by the evaporation of the water. Even the same process occurs in our body to reduce the temperature by sweating. As compared to the HVAC systems direct evaporative coolers use very small amount of electrical energy.

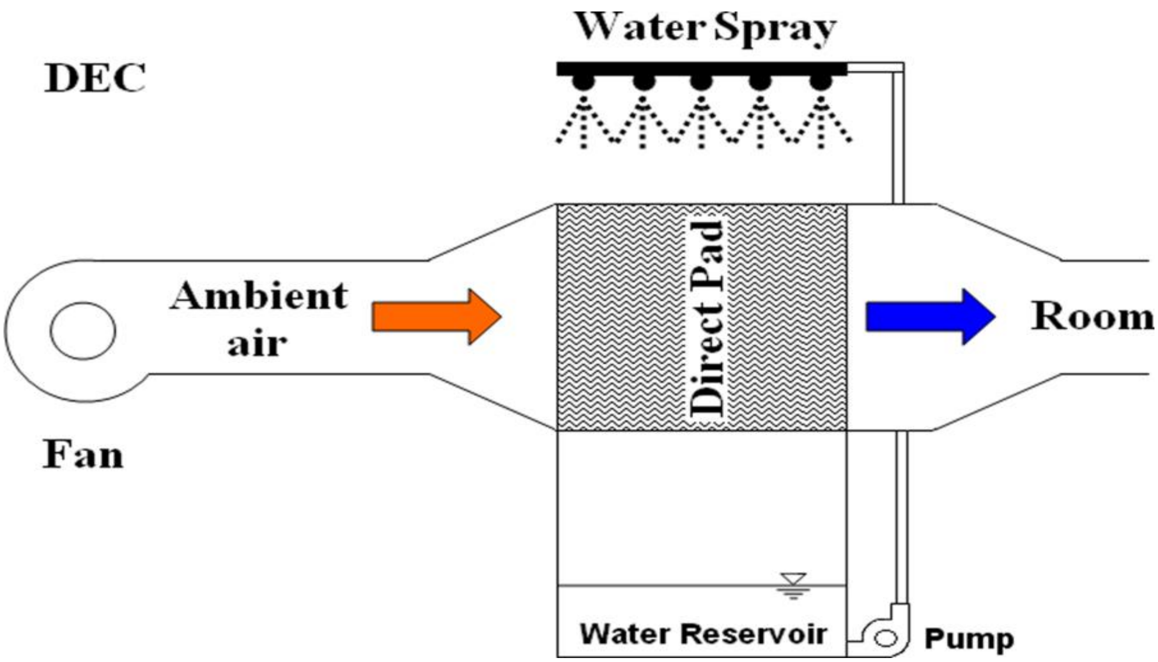

Figure 3-10 Direct evaporative cooler

In TRNSYS library type506a is used for the direct evaporative cooling. In this device it is assumed that whole cooling process take place at constant wet bulb temperature. There is no built-in controller in this device to control the parameters of the outlet air. There is a control function in this device whose signal is set to be 1 when device is required to be on.

There are some limitations for the achieved outlet temperature which can be shown from the given.

$$T_{min,air\ out} = T_{wet\ bulb,air\ in} \qquad \text{Equation 3-13}$$

This equation not only shows that minimum outlet temperature is dependent on the incoming value of the wet bulb temperature but also that the direct evaporative cooler with no controlled device have limited performance by the dryness of the incoming air. The more the dry air gets in the evaporative cooler more cooling can be achieved. That's the reason due to which direct evaporative coolers are used with the desiccant wheel cycle because in that process humidity is already absorbed by the silica gel present in the desiccant wheel.

3.3.15 Air conditioning unit:

Conventional air conditioning units operate on the basis of the vapor compression cycle. In TRNSYS library type921 is used for the residential air conditioning unit. Performance of this device is control by the catalog file which is already provided. Control signal is provided to control it on/off operations.

If the device is on then outlet conditions of the air coming from evaporator are calculated from the following equation.

$$h_{evap,out} = h_{evap,in} - \frac{\dot{Q}_{total}}{\dot{m}_{evap}} \quad \text{Equation 3-14}$$

$$T_{evap,out} = T_{evap,in} - \frac{\dot{Q}_{sensible}}{\dot{m}_{evap} * CP_{air}} \quad \text{Equation 3-15}$$

$$P_{evap.out} = P_{evap,in} - \Delta P_{evap} \quad \text{Equation 3-16}$$

The power drawn by the air conditioning unit depends upon the power of the controller, blower and compressor while the compressor is the most energy extensive unit among the all parts of air conditioner.

In our system there is no direct link of the power input in air conditioner, so we simply calculate the power used by the type921 and the power produced by the PV/T collector. There difference shows that how much power is obtained from the PV/T panel while how much auxiliary is used.

The heat of rejection from this device can be calculated as

$$\dot{Q}_{rejected} = \dot{Q}_{total} + P\dot{W}I_{total} \quad \text{Equation 3-17}$$

While the coefficient of performance (COP) and ERR are calculated as

$$COP = \frac{\dot{Q}_{total}}{P\dot{W}I_{total}} \quad \text{Equation 3-18}$$

$$ERR = COP * 3.413 \quad \text{Equation 3-19}$$

The moisture removed by the air conditioner through the process of condensation can be calculated as

$$\dot{m}_{condensate} = \dot{m}_{evap,in}(\omega_{evap,in} - \omega_{evap,out}) \qquad \text{Equation 3-20}$$

3.3.16 Heating and cooling load

In HVAC systems sometimes, the load in the form of heating or cooling is already calculated and known. In TRNSYS library type682 is used for the built-in heating and cooling load where you can directly put the value of your cooling load or the heating load. Cooling load is shown by positive while the heating load is shown by the negative.

Usually the type682 and type686 are used in the form of the combination and both are used to produce the desired load profile.

Fluid specific heat is also changed depending upon the working fluid is water or air. The outlet temperature can be calculated from the following equation

$$T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m} * Cp} \qquad \text{Equation 3-21}$$

For the positive load outlet temperature will obtained higher as compared to the inlet temperature while for negative loads condition is reverse.

3.3.17 Synthetic building load generator

In the TRNSYS library load generator for synthetic buildings is shown by the type686. This component can generate the load profile of the building on hourly basis. Type686 operates by changing the sine-wave function to account the seasonal variations e.g. day variations with respect to time or difference between the week days and weekends.

There are some parameters that are changed in the case of the heating and cooling load respectively. These parameters are shown in the given tables.

Name	Value	Unit
Start of cooling season	2160	hr.
End of cooling season	6552	hr.

Start of heating season	6552	hr.
End of heating season	2160	Hr.
Seasonal offset for cooling	0.0	
Seasonal multiplier for cooling	1	
Seasonal offset for heating	0.0	
Seasonal multiplier for heating	1.0	

Table 6 synthetic building generator load parameters.

This is the case 1 which is normally used for the residential buildings. Here the heating season ends on 1st April while the cooling season ends on the 1st October while in our case same values of parameters are used for the calculations.

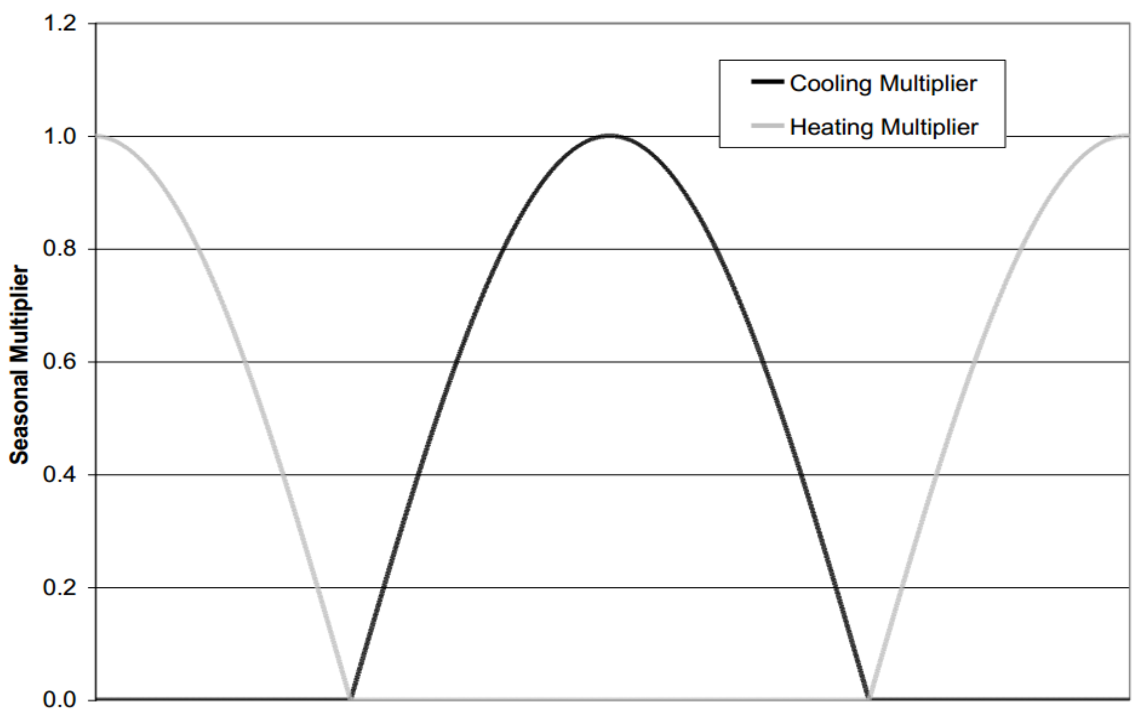

Figure 3-11 Cooling/heating multipliers

3.3.18 Thermostat

Thermostat is the type of controller which is used to control the temperature in specific device or the room temperature. In the TRNSYS library type108 is used as a thermostat. It has different modes of cooling and heating. In our case we are using it for the control of auxiliary heater. Its function is to monitor the temperature of air stream coming from the water to air heat exchanger. If the temperature is above 60 it didn't generate the on signal while if the temperature is below 60^{o}C then it turned on the auxiliary heater. There are five different stages of set point temperature. First three are for heating mode while the last two are for the cooling mode. In our case we set the first stage point temperature 60^{o}C. While the exit air stream from water to air heat exchanger is attached with the monitoring temperature. Below figure shows the set point definition.

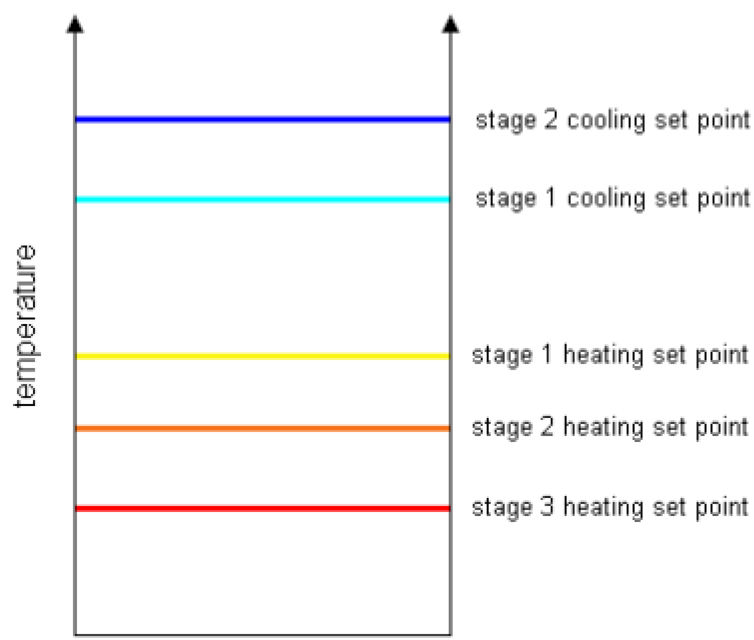

Figure 3-12 Set point definition

Here some furnaces ae designed with lower and high burner settings. If the monitored temperature falls from the specific point, then the low burner set. If the temperature continues to fell, then the burner with highest power turns on.

3.3.19 On/Off differential controller

On/off differential controller is used to turn on the air conditioning device. In the TRNSYS library it is shown by the type2b. Firstly an input signal is attached with this controller from the controller 14h which allows to work this controller only during 9am to 5pm. During this time interval on/off controller regulates the temperature of the air coming from the direct evaporative cooler. If the temperature is above 20 then it turned on the air conditioner to get the desired comfort level.

3.3.20 Time dependent forcing function

Time depending forcing function is used to generate the control signal to turn the device on during our required time interval. In TRNSYS library it is shown by the type 14h. In our case as we are designing a air conditioning system for the office building that works during 9am to 5pm. So, we turn the time dependent function during this time interval.

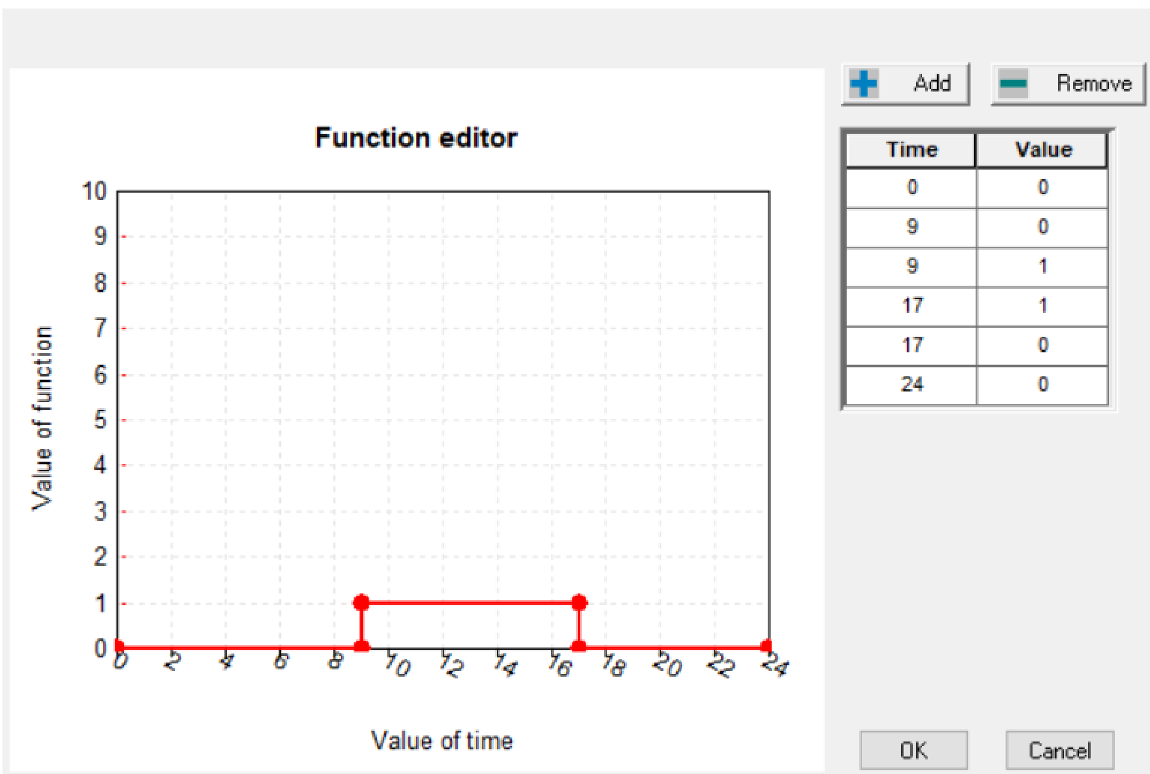

Figure 3-13 Time dependent forcing function

Given figure shows the configuration of function editor of the forcing function for our case.

CHAPTER#4

4 Results and Discussion

Simulations are performed in TRNSYS 17.0 for the whole summer season that starts from May and ends in September. In the TRNSYS it starts from 2880hr and ends with 6552hr.

4.1 Space cooling load profile

Figure shows the cooling load profile for the cooling load of 2.5tons. Load profile is generated by the type686 which is used to generate the heating and cooling load for synthetic buildings. Type 682 and type686 are combined to generate the load profile. The designer has to set some parameters which are discussed in the previous chapter and then give the value of the cooling load. The load profile represents the daily cooling demands of the system for the whole summer season.

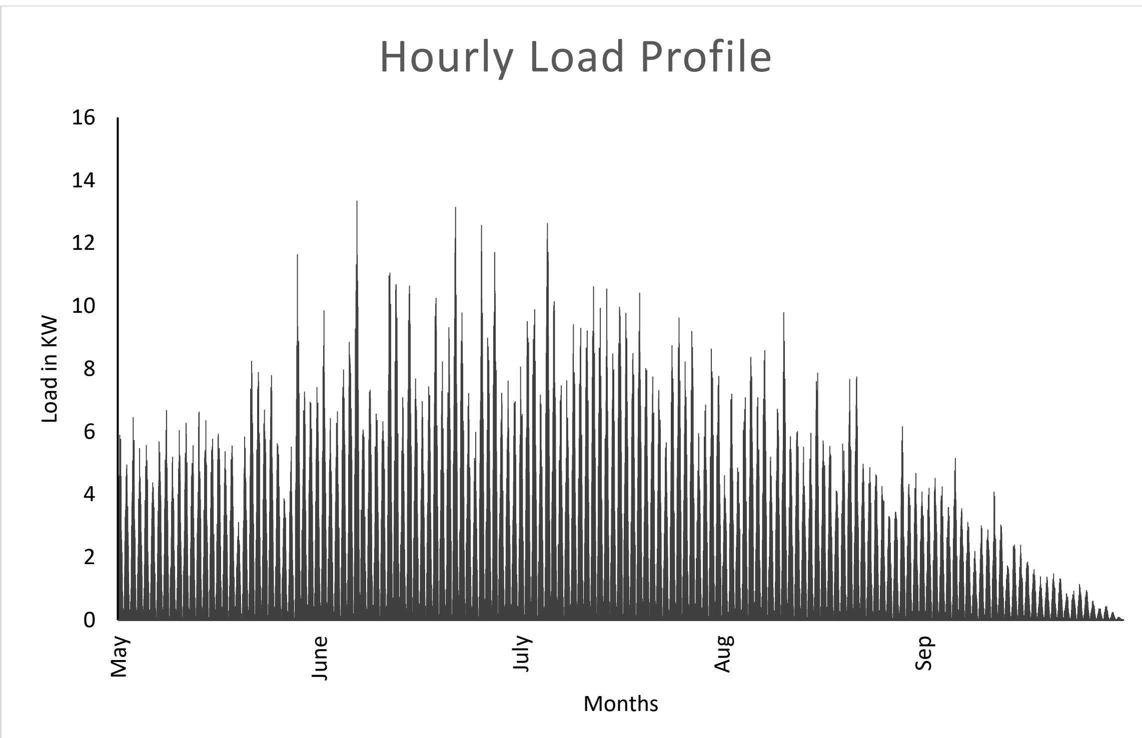

Figure 4-1 Cooling load profile

4.2 Results for configuration-1

4.2.1 Storage Tank optimization

For the optimization of the storage tank, solar fraction and solar primary energy saving is calculated by varying the volume of the storage tank by keeping constant the solar collector area. Couple of simulations are done in the same way for the different size of collectors.

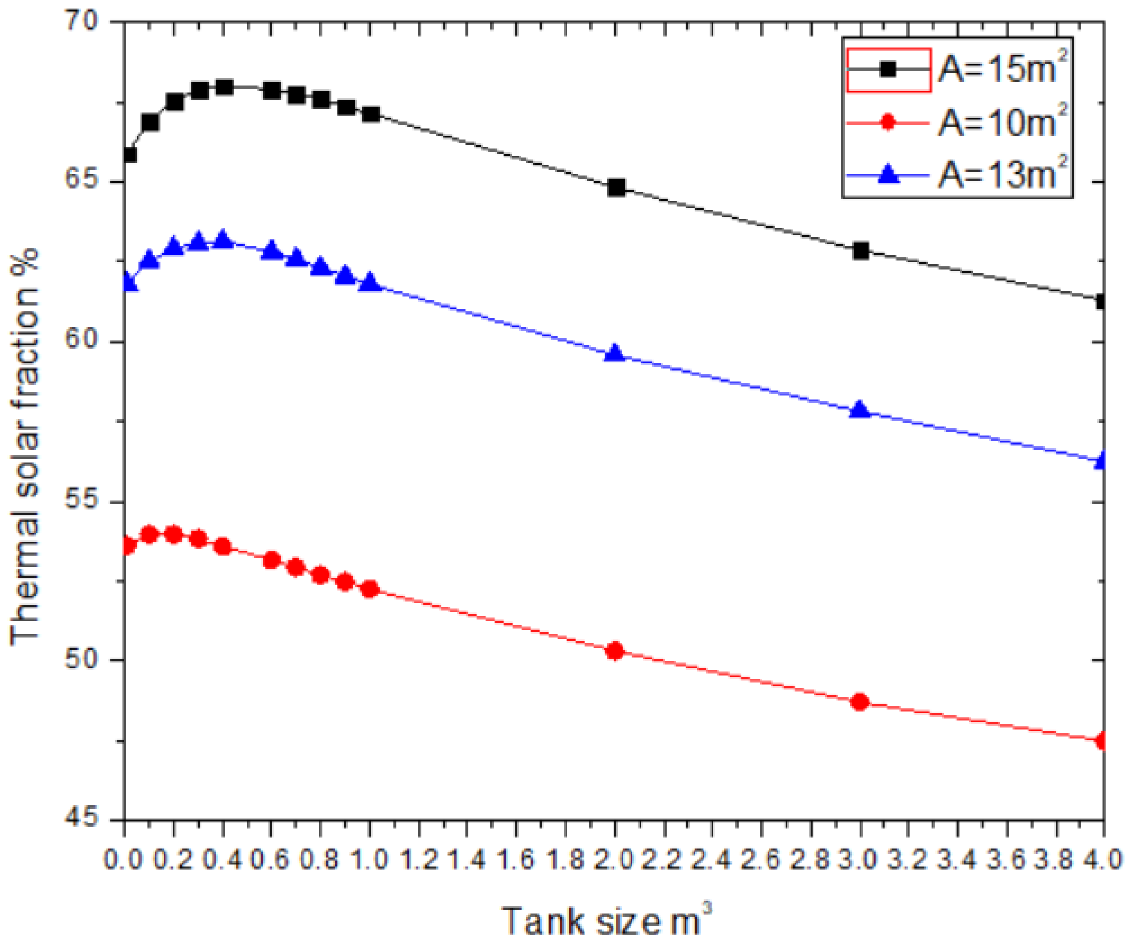

Figure 4-2: Storage Tank VS Solar Fraction for C-1

Solar fraction is maximum at certain size of tank and after that it decreased because the thermal energy obtained from the tank is not enough to heat the water present in the storage tank.

As the collector area is changed optimized tank size also changed. For the collector of $10m^2$,$13\ m^2$ and $15m^2$ the optimized tank size is $0.2\ m^3$,$0.3\ m^3$ and $0.4m^3$ is found respectively.

4.2.2 Optimization of the flow rate of pump on source sides

If we draw the profile of source side pump's flow rate VS electrical and thermal energy obtained from the collector, we see that at very low flow rates this value is small then at the certain flow rate value will be maximum and after that it increase slightly with the flow rate. But for higher flow rates pump of higher power is required. We chose the optimum flow rates where maximum energy can be obtained by keeping some tradeoff between the flow rate and useful energy gain.

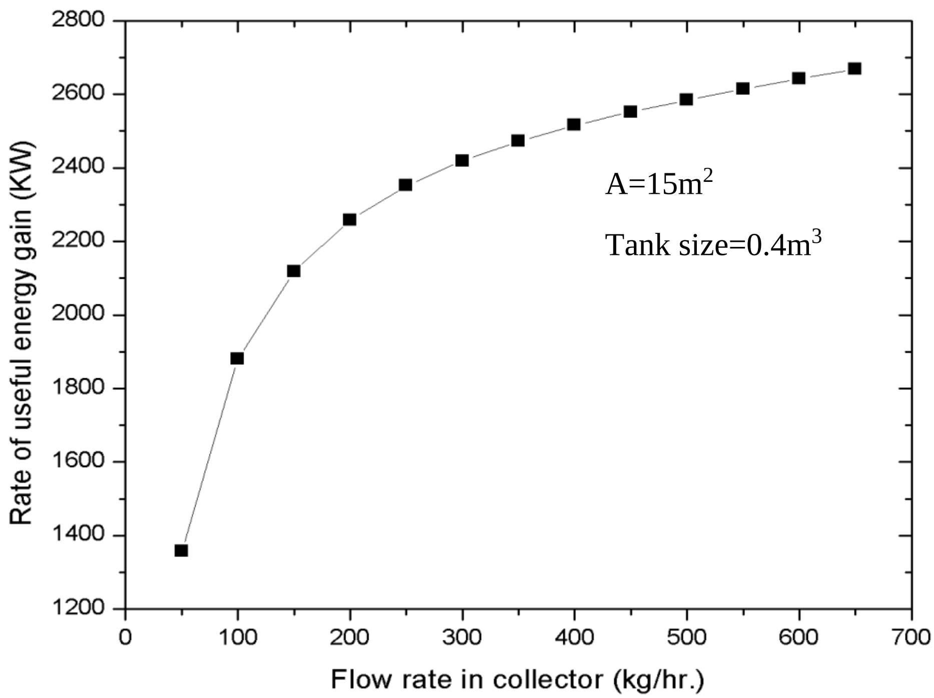

Figure 4-3 Pump Flow Rate VS Rate of Useful Energy Gain for C-1

From the figure 29 it can be shown that after the flow rate of 450kg/hr there is small change of useful energy gain against the flow rate. So, in our simulation we take the optimized value of 450kg/hr.

4.2.3 Relation of Collector slope vs solar fraction:

Solar fraction is the important parameter during the consideration of solar technology. The solar fraction tells us that how much energy is provided by the solar collector and how much by the auxiliary source. It is described by the following formula,

$$Solar\ fraction = \frac{Q_u}{Q_u + Q_{aux}}$$ *Equation 4-1*

In the PV/T collectors there are two types of solar fractions, one is for the thermal energy and second is for the electrical energy. So, both the solar fractions with respect to area are discussed. Current simulations are done for the whole summer season.

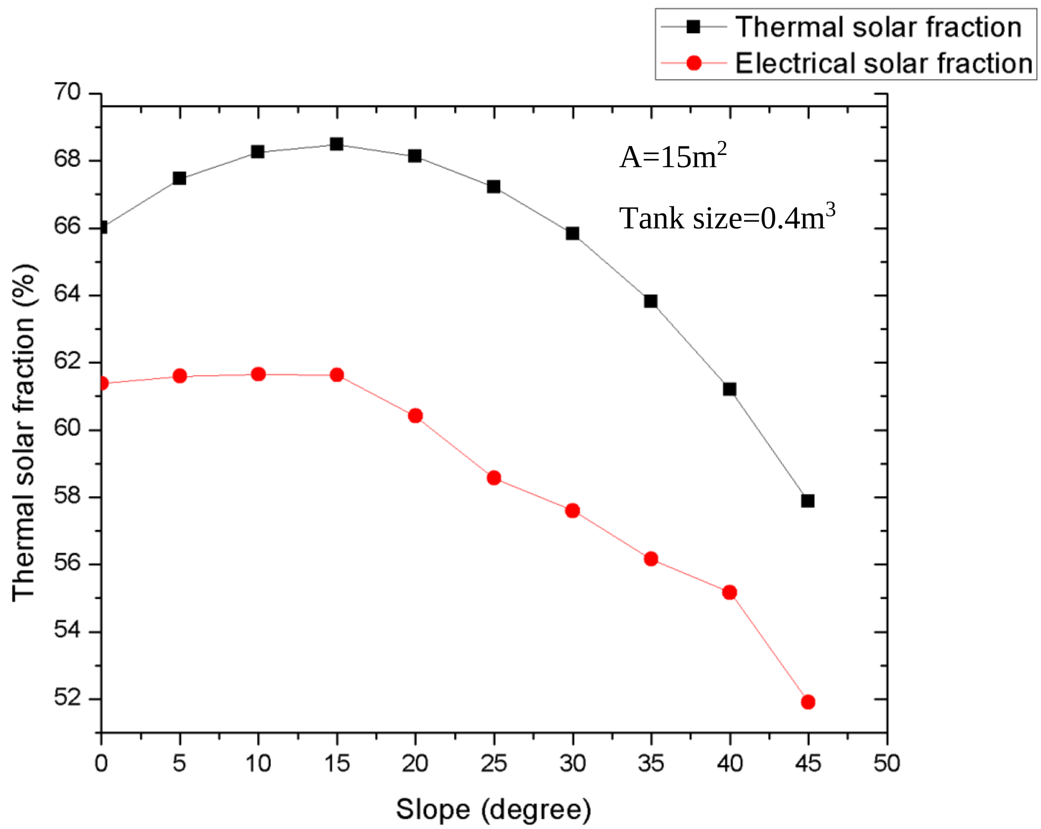

Figure 4-4: Collector Slope VS Solar Fraction for C-1

From the figure24 it is shown that maximum solar fraction is obtained between the slope of 10^{o}-20^{o} for the region of Islamabad during the summer season.

4.2.4 Regeneration temperature VS PES

As the temperature of the regeneration air increased more energy is required to heat the air to desired point. Which cause to decrease in primary energy savings. But the dehumidification rate increase with the increase in temperature. There is the certain range of temperature for the regeneration of the desiccant wheel depending upon the application.

Graph shows the relation between the temperature of regeneration air and primary energy savings.

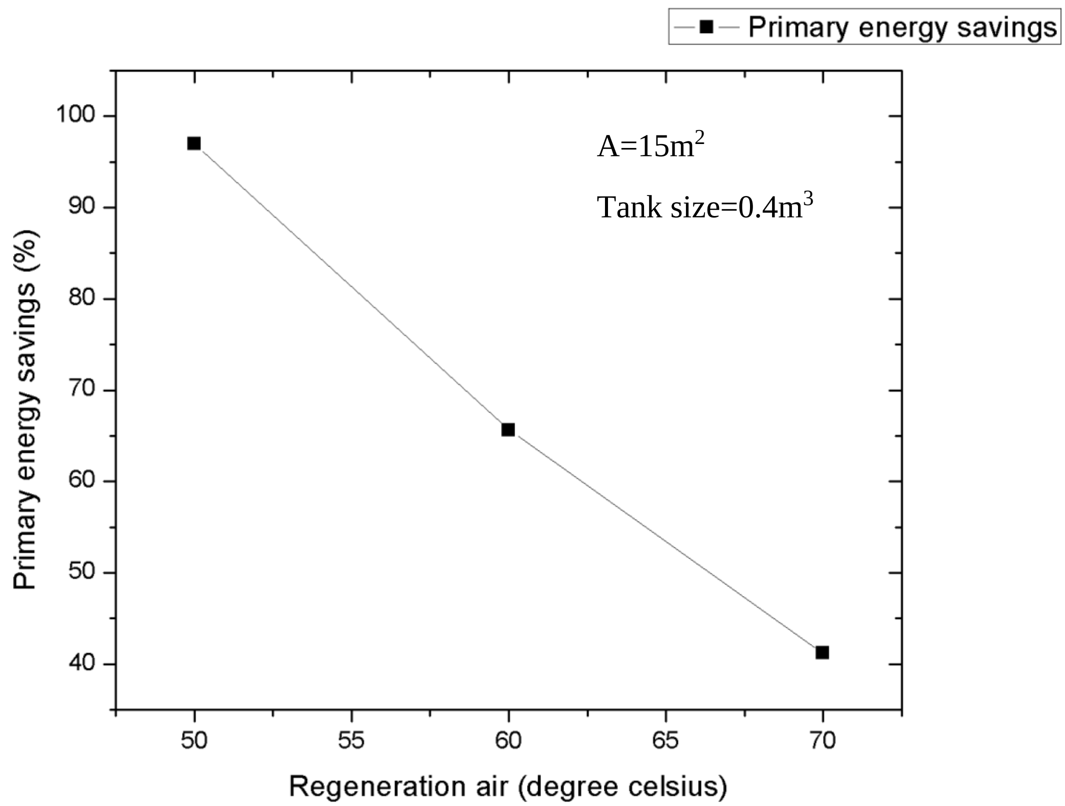

Figure 4-5: Regeneration temp VS PES for C-1

4.2.5 Collector Area VS Solar Fraction

Figure25 shows the variation of electrical and thermal solar fraction of flat plate photovoltaic-thermal collector with respect to area. As we increase the area of the solar collector the amount of solar radiations collected by the collector also increase. As the solar radiations increase the energy of utilization also increased.

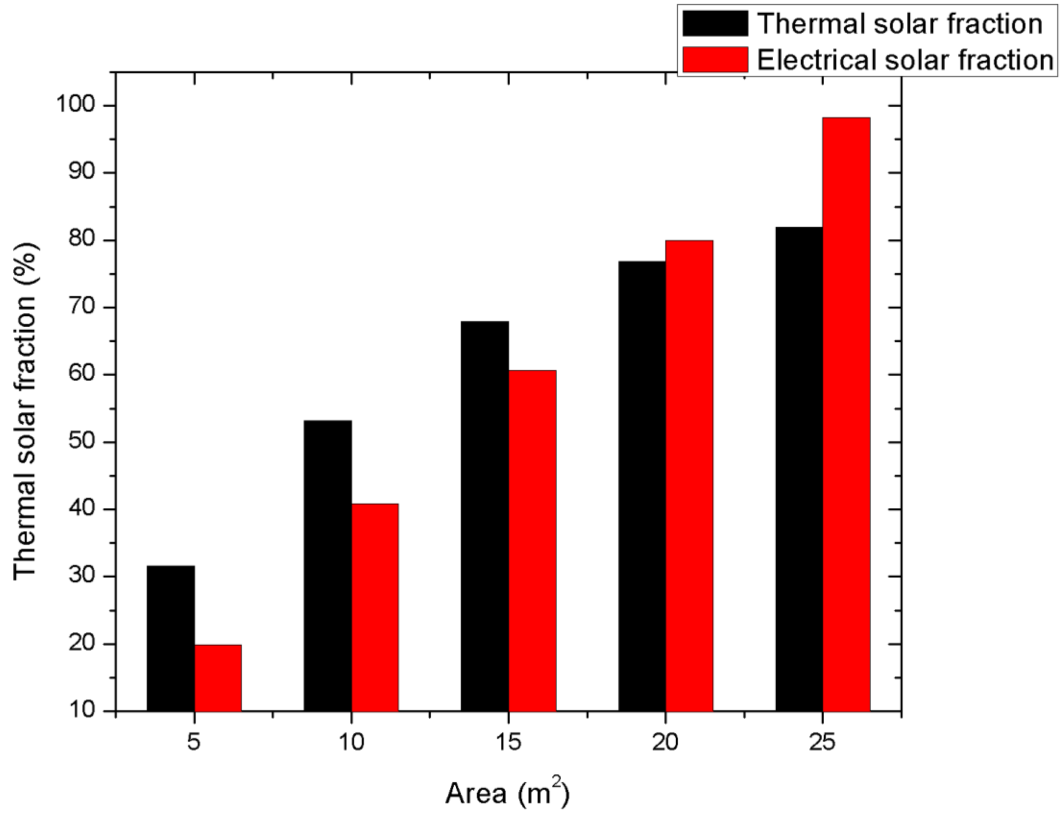

Figure 4-6 Area VS Solar Fraction for C-1

It depends upon the designer that how much solar fraction he chose for the desired designed system. Usually 50% or more than 50% solar fraction is kept for the designed systems.

4.2.6 Collector efficiency VS Collector Area

Collector efficiency can be defined as ratio of useful thermal and electrical energy producing from the PV-T collector to the useable solar irradiance that are falling on the aperture area of the collector.

$$\eta_{thermal} = \frac{Q_u}{G*A} \quad \text{Equation 4-2}$$

$$\eta_{electrical} = \frac{P_{produced}}{G*A} \quad \text{Equation 4-3}$$

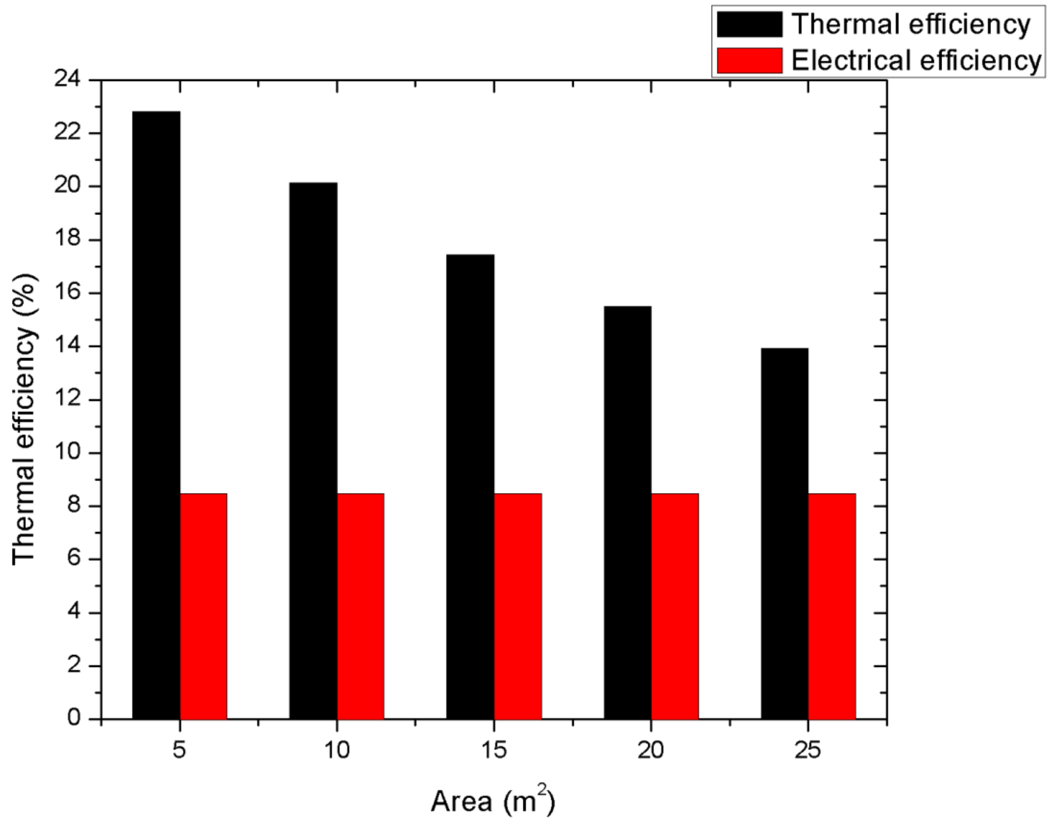

Figure 4-7 Area VS Efficiency for C-1

Figure 26 shows that when we increase the area of the collector then the efficiency decreased which is also can be seen from the equation for calculating the efficiency. Area is also inversely proportional over there.

4.2.7 Collector Area VS Primary Energy Savings

Primary energy saving tells about the energy that we saved by using the solar collector.

Equation for the primary energy savings is defined as

$$f_{sav} = 1 - \frac{\frac{Q_{boiler}}{\varepsilon_{fossil}} + \frac{E_{elec}}{\varepsilon_{elec}} + \frac{Q_{cooling,missed}}{SPF.\varepsilon_{elec}}}{+\frac{Q_{cooling,ref}}{SPF_{ref} * \varepsilon_{elec}}} \quad \text{Equation 4-4}$$

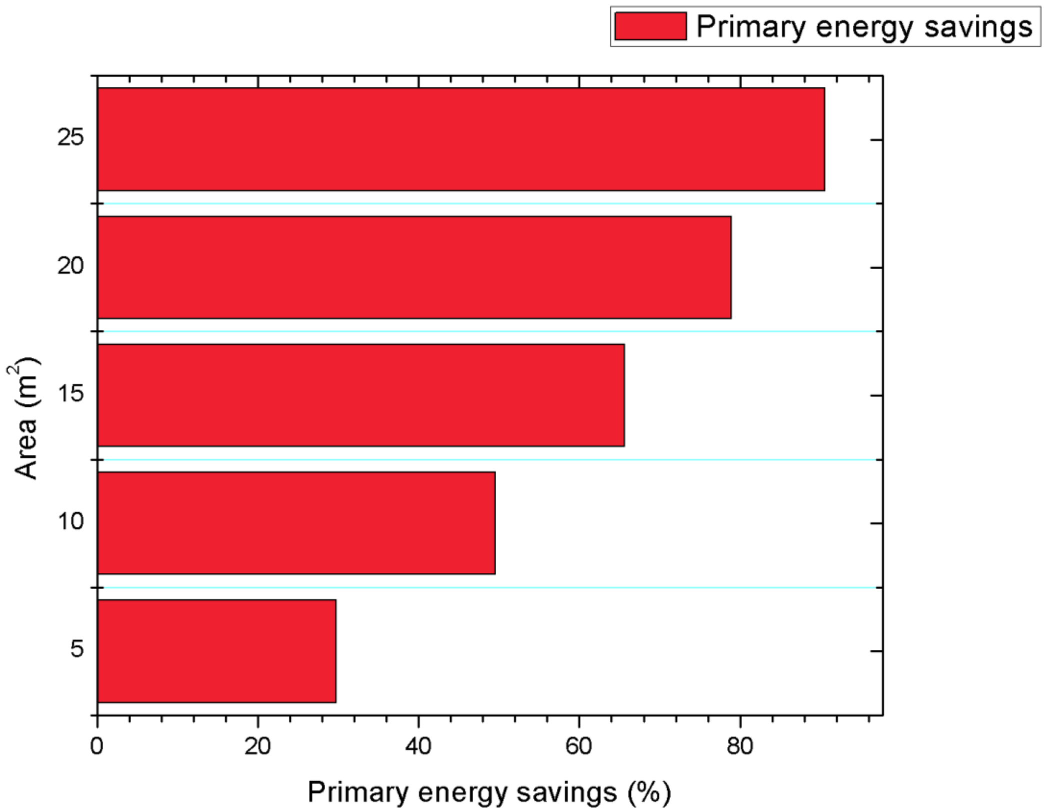

Figure 4-8 Area VS Primary Energy Savings for C-1

It can be seen from the figure 27 that as the area of solar collector increase primary energy savings also increased.

4.2.8 Monthly Results

Simulations are done for the specified collector area and the pumps flow rates. The collector is set as $15m^2$. All the simulations are done for the summer season i.e. April to September. Primary energy savings, collector efficiency, solar fractions are calculated for all months.

4.2.8.1 Primary Energy Savings

Simulations are done for flat plate photovoltaic thermal collector for the constant area and pump flow rates.

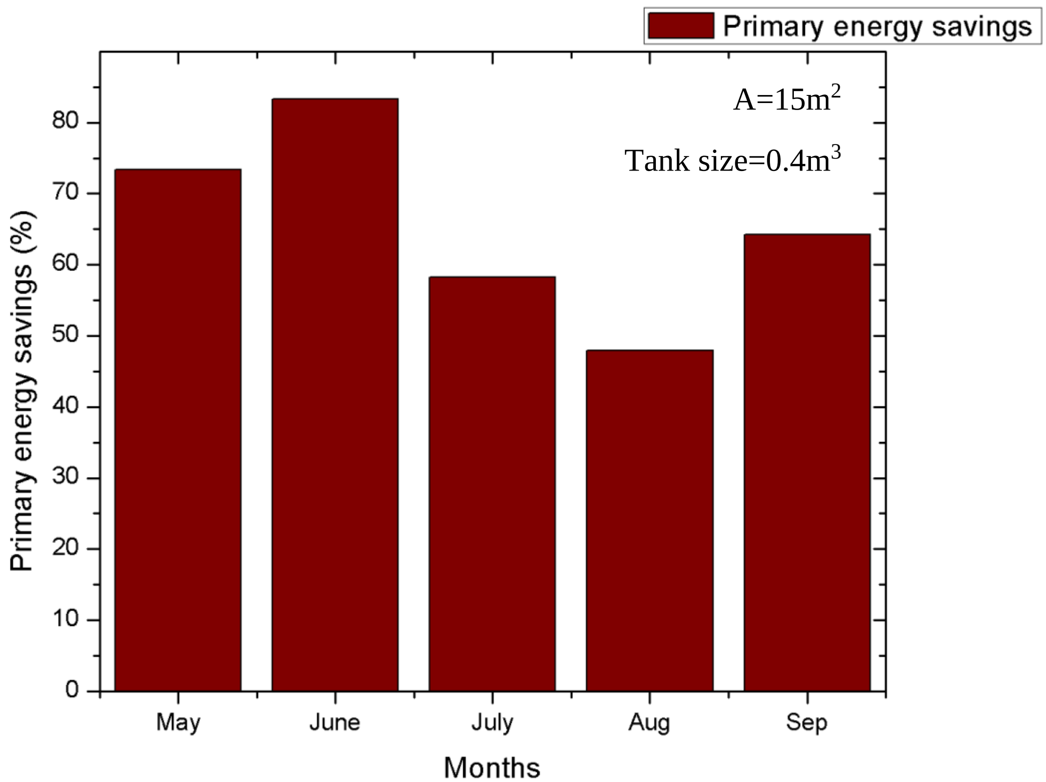

Figure 4-9 Months VS Primary Energy Savings for C-1

Figure 31 shows that the maximum primary energy saving is obtained during the month of June while the minimum primary energy saving is obtained during the august.

4.2.8.2 Collector Efficiency

Flat plate solar PV-T collector efficiency is shown in the figure 32. Thermal Efficiency lies between 19% to 15%. Maximum thermal and electrical efficiency is obtained in September while the minimum thermal efficiency is obtained in June.

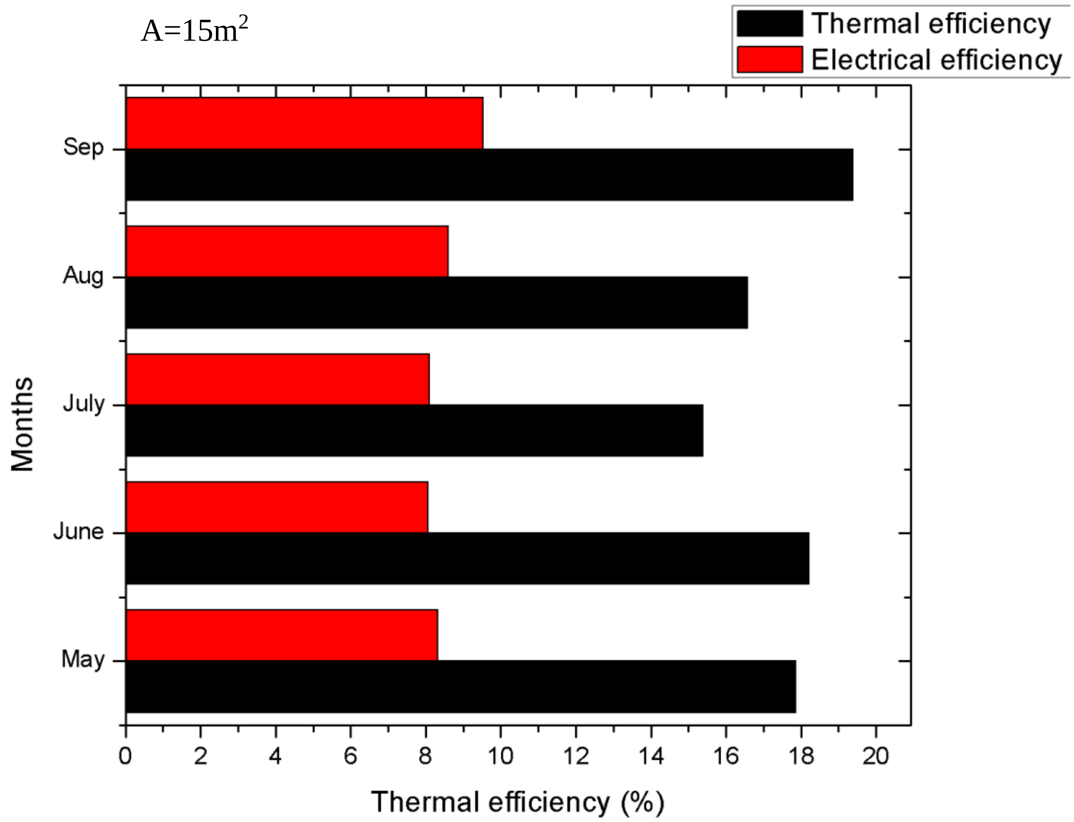

Figure 4-10 Months VS Efficiency for C-1

4.2.8.3 Solar Fraction

Figure 33 shows the monthly solar fraction of the collector at constant area and flow rates.

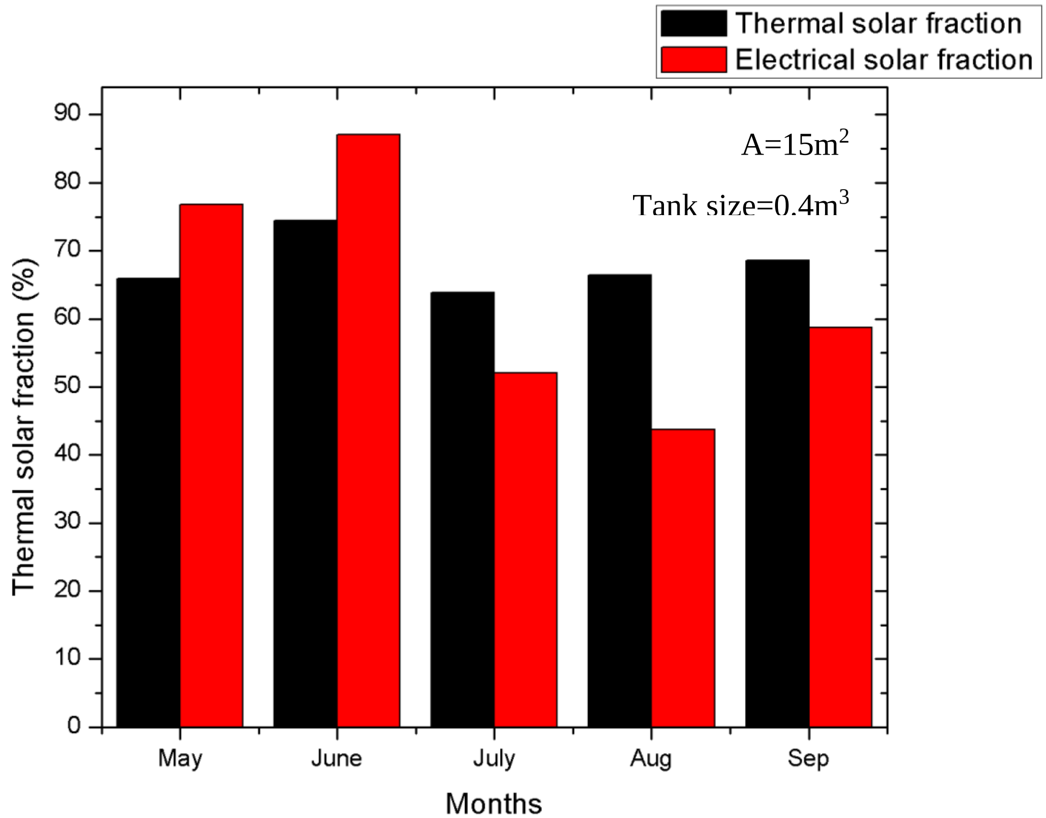

Figure 4-11 Months VS Solar Fraction for C-1

From the graph it can be shown that the solar and electrical solar fractions vary from month to month. While the maximum solar fraction is obtained during the month of June.

4.3 Results for configuration-2

4.3.1 Source Side Pump's Flow Rate VS PES

When the flow rate of pump increased then the value of Qu and P obtained from the collector increased to the certain amount. After some specific value of flow rate Qu increase with a very small amount while the power consumption of motor for higher flow rate is greater comparatively the increase in energy obtained from the collector. To set that desired point graph is plotted between the flow rate of the pump on the source side and primary energy saving for the whole season. For these calculations values for area, size of tank and flow rate for the second pump remains constant. The optimum flow rate obtained from this graph is 500kg/hr. where we get the maximum value of primary energy savings as shown in the figure.

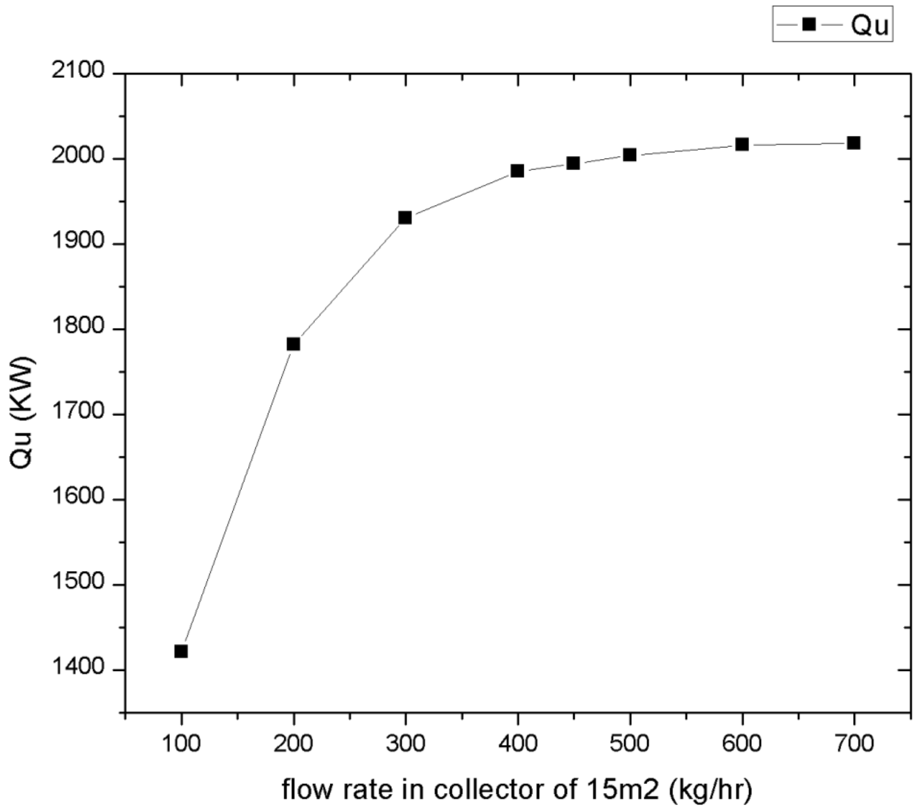

Figure 4-12: Pump1 flow rate VS PES for C-2

4.3.2 Storage Tank Size VS Thermal Solar Fraction

After setting the flow rates for both pumps the size of storage tank is determined. As we increase the size of storage tank the value of thermal solar fraction increases up to certain amount and after that its value starts decreasing because the amount of energy obtained from the solar collector is not enough to heat that water present in the storage tank. Graph for storage tank size against the TSF is plotted to obtain the point where maximum energy is obtained

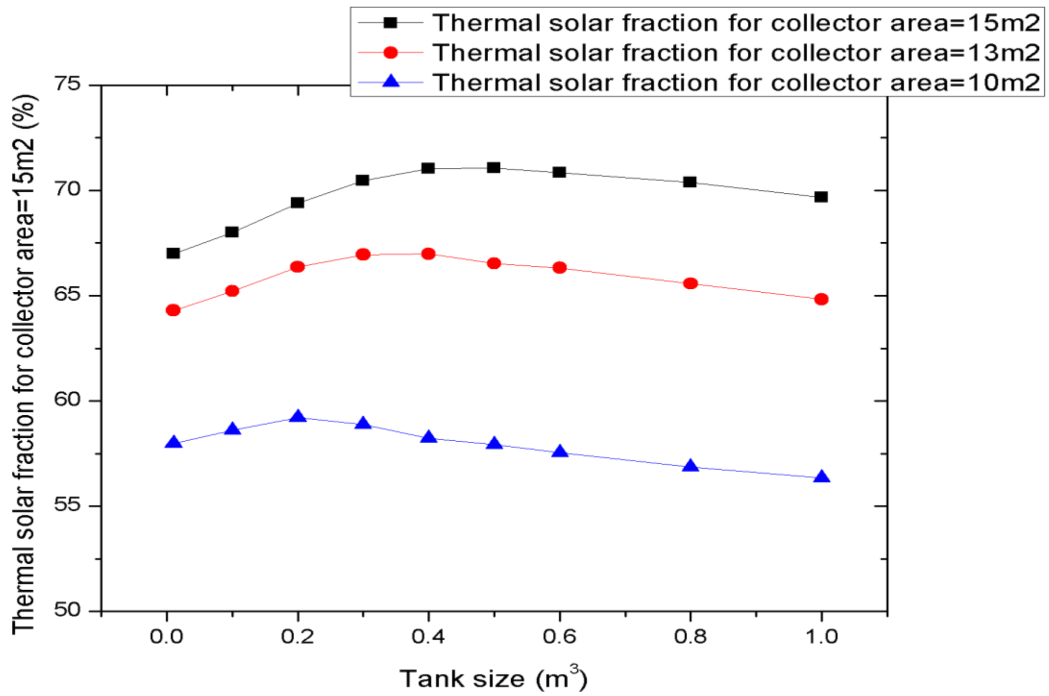

Figure 4-13: Tank Size VS TSF for C-2

4.3.3 Collector Slope VS Solar Fraction

TSF and ESF of the system is calculated by varying the slope of collector from 0 to 45°. Maximum solar fraction is obtained between the 10° to 15°.

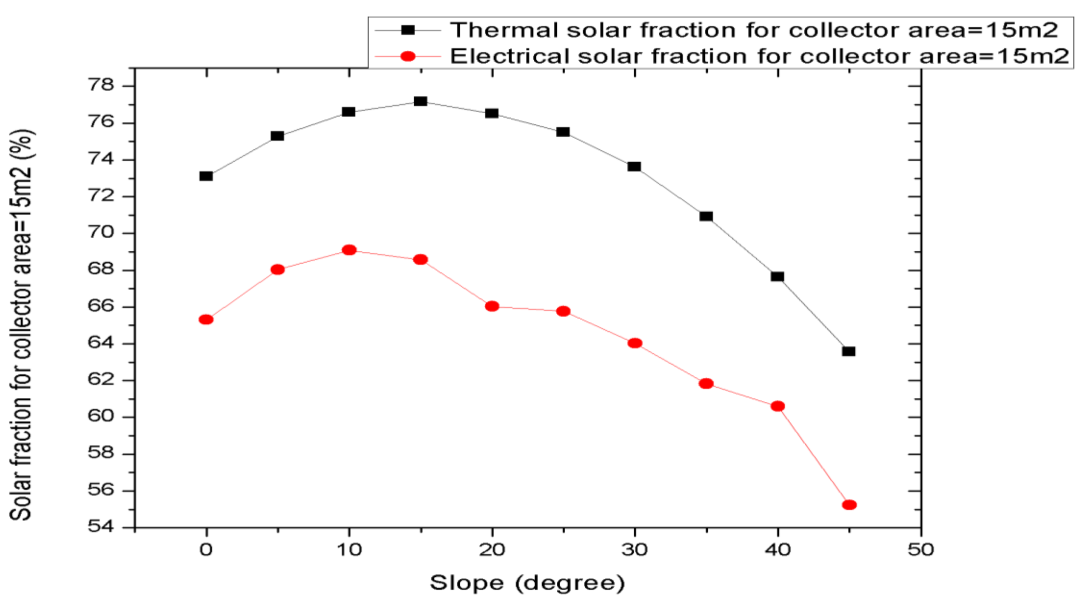

Figure 4-14: Collector slope VS Solar fraction for C-2

All the calculations are done for the slope of 21° for the weather of Islamabad.

4.3.4 Regeneration Temperature VS PES

Regeneration temperature is the temperature that requires to regenerate the silica gel present in the desiccant wheel otherwise desiccant wheel may go to the saturation state. As the regeneration temperature increase more flow rates of incoming air can be achieved. But to achieve the higher regeneration temperature more thermal energy is required due to which primary energy savings decreased. Graph shows the relation between the regeneration temperature and PES.

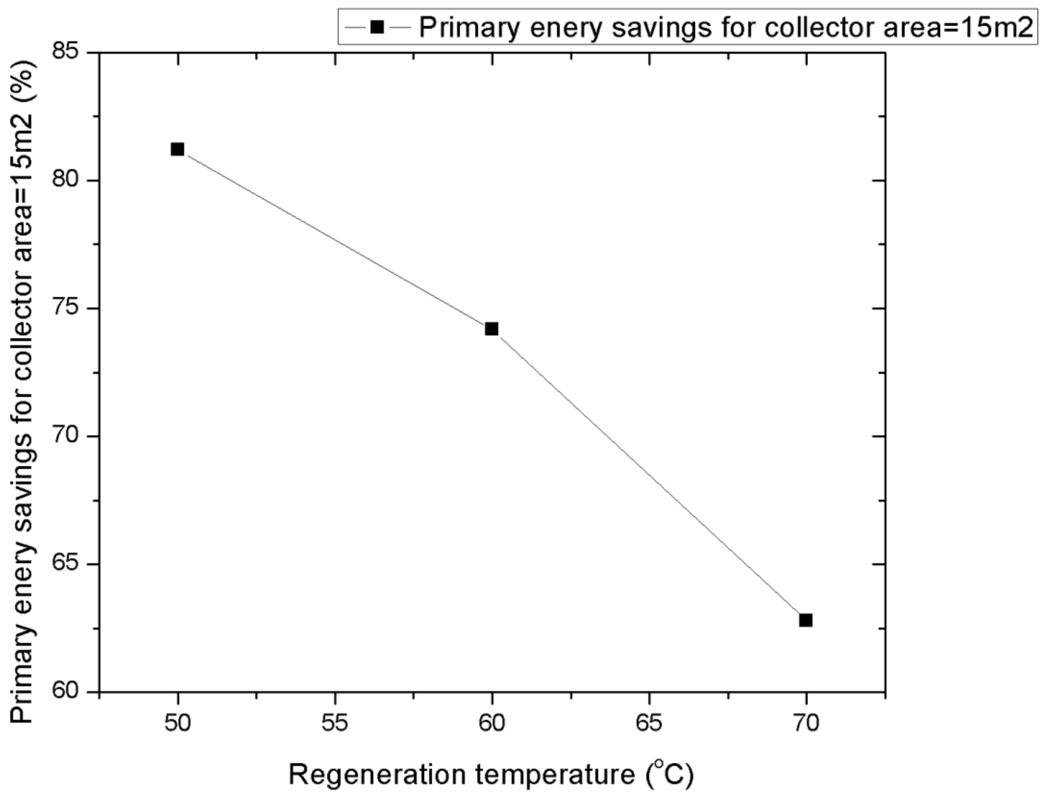

Figure 4-15: Regeneration temperature VS PES for C-2

4.3.5 Area VS Solar Fraction

For the whole season starts from May to September the graph for the thermal and electrical solar fraction is plotted against the different areas after calculating the optimum values for the tank size and pumps flow rates. Graphs shows as the area for the collector increased both the electrical and thermal solar fractions also increased because more energy is obtained with the increase in size.

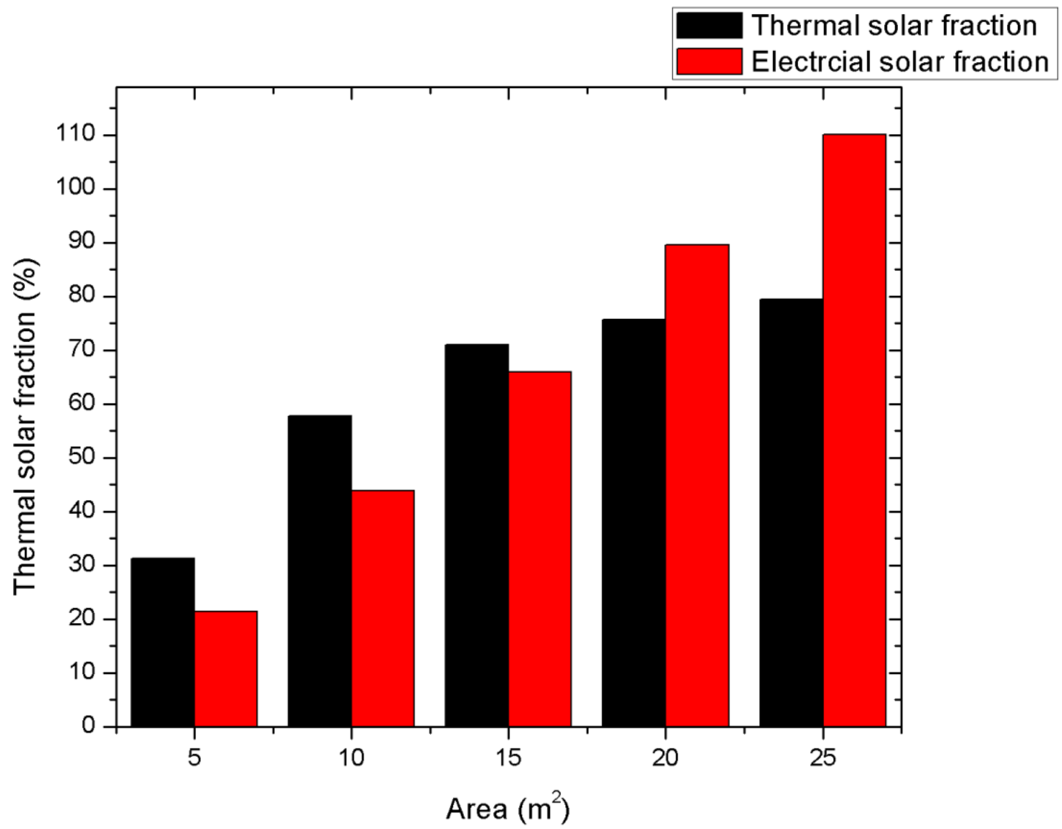

Figure 4-16: Area VS Solar Fraction for C-2

4.3.6 Area VS Primary Energy Savings

For the whole summer season, the results are calculated for the primary energy savings by varying the different area of collectors. Graphs shows as the area increased the PES also increased.

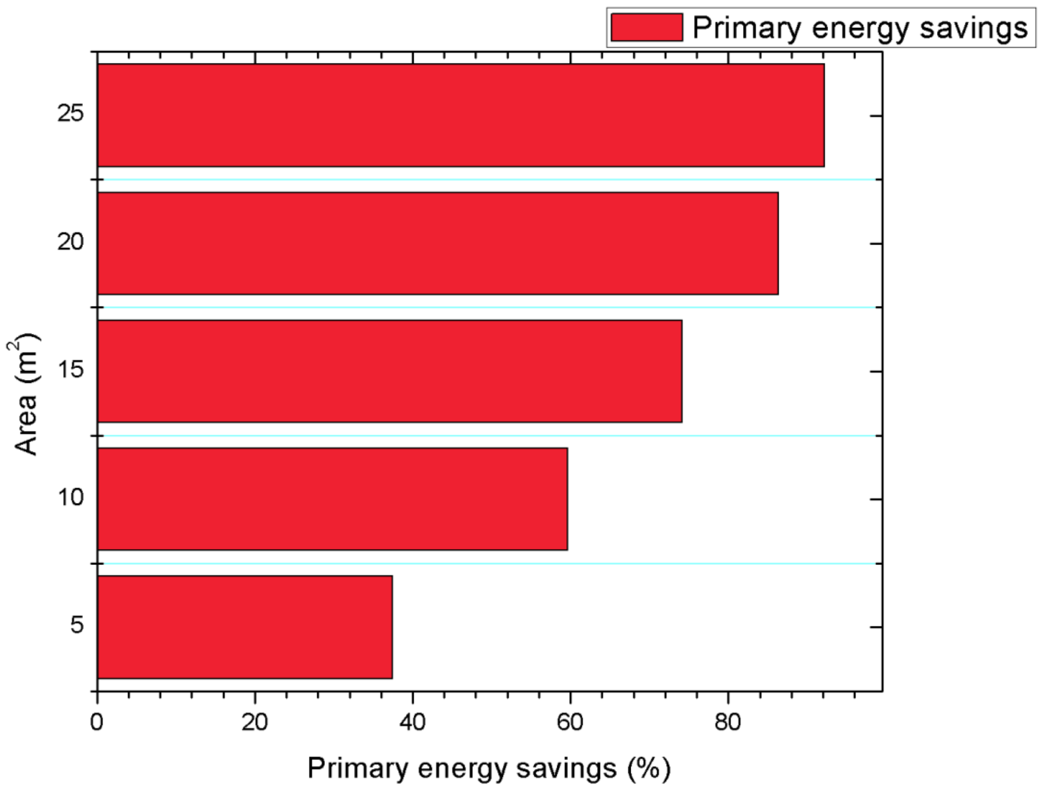

Figure 4-17: Area VS PES for C-2

4.3.7 Area VS Efficiency

For the whole season both the electrical and thermal efficiencies are plotted against area. As the area for the collector increase the thermal efficiency decreased which can be illustrated from the following equation

$$\eta_{thermal} = \frac{Q_u}{G*A} \qquad \textit{Equation 4-5}$$

While the electrical efficiency almost remains constant.

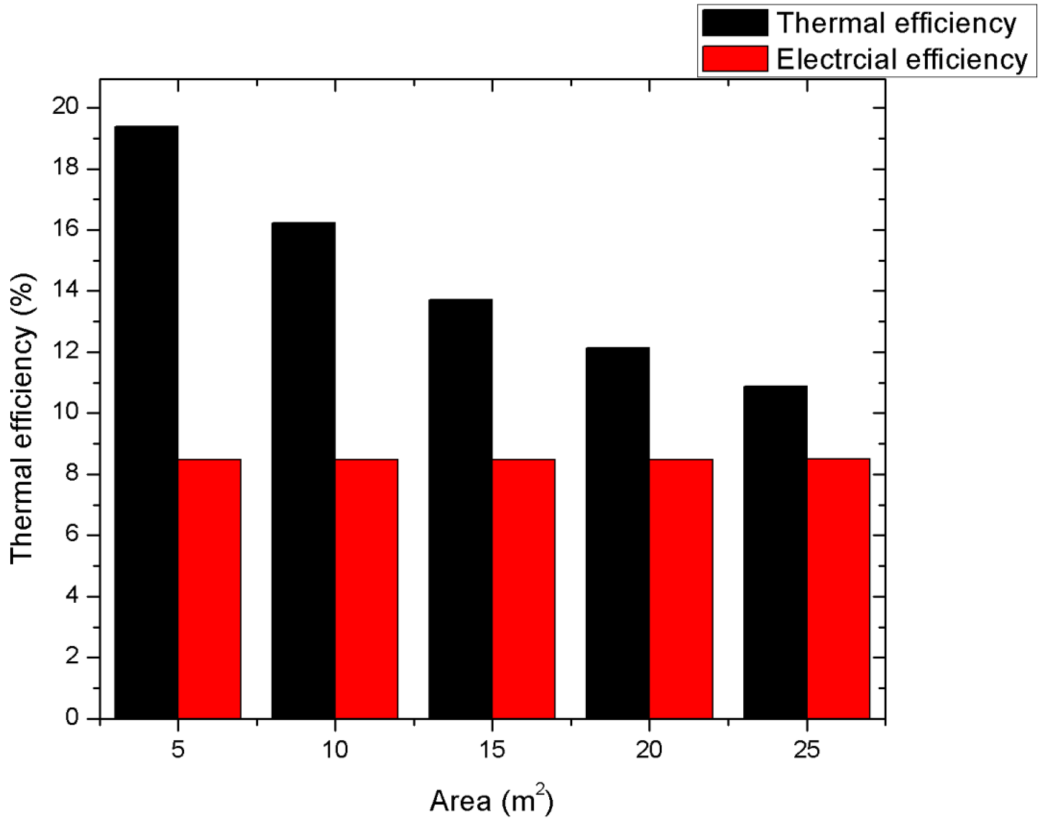

Figure 4-18: Area VS Efficiency for C-2

4.3.8 Monthly Results

4.3.8.1 Months VS Solar Fraction

By setting the area, flow rates, tank size and regeneration temperature the monthly solar fractions are also calculated. Both the ESF and TSF vary from month to month because of the different weather conditions. Graph shows the monthly solar fractions.

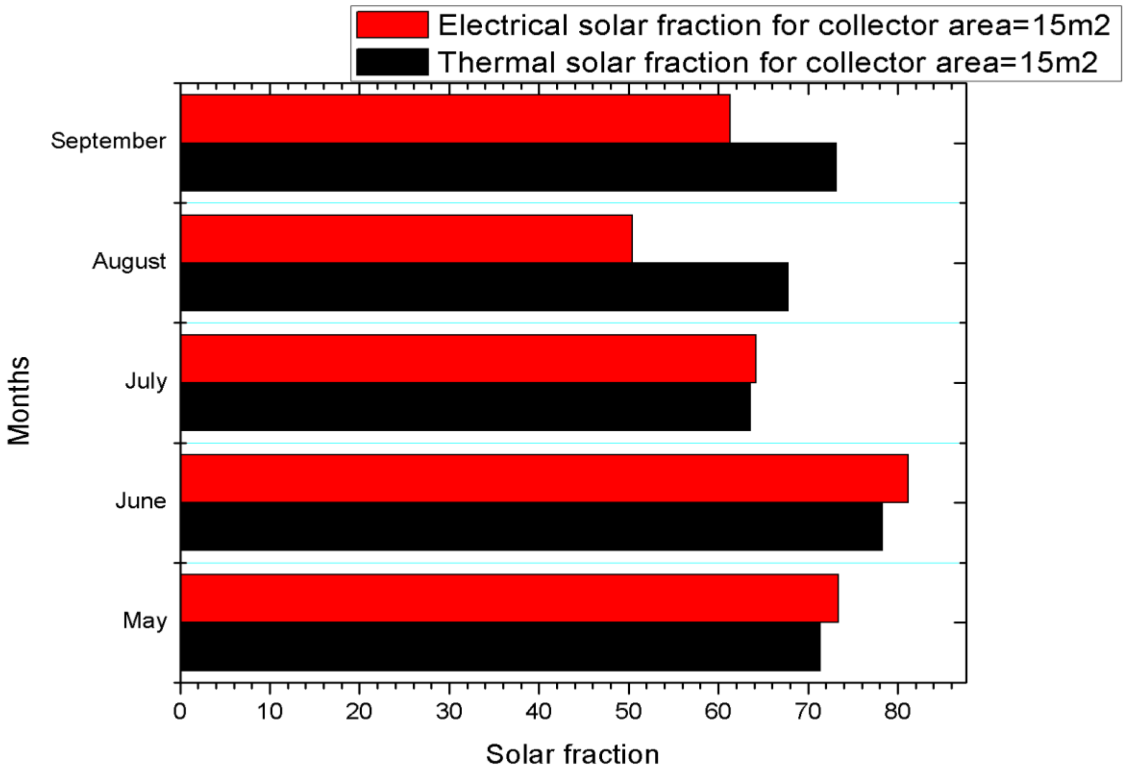

Figure 4-19: Months VS Solar Fraction for C-2

4.3.8.2 Months VS Primary Energy Savings

Primary energy savings are also calculated on the monthly basis. The values for PES vary because from month to month due to changes in weather conditions energy obtained from the collector also change.

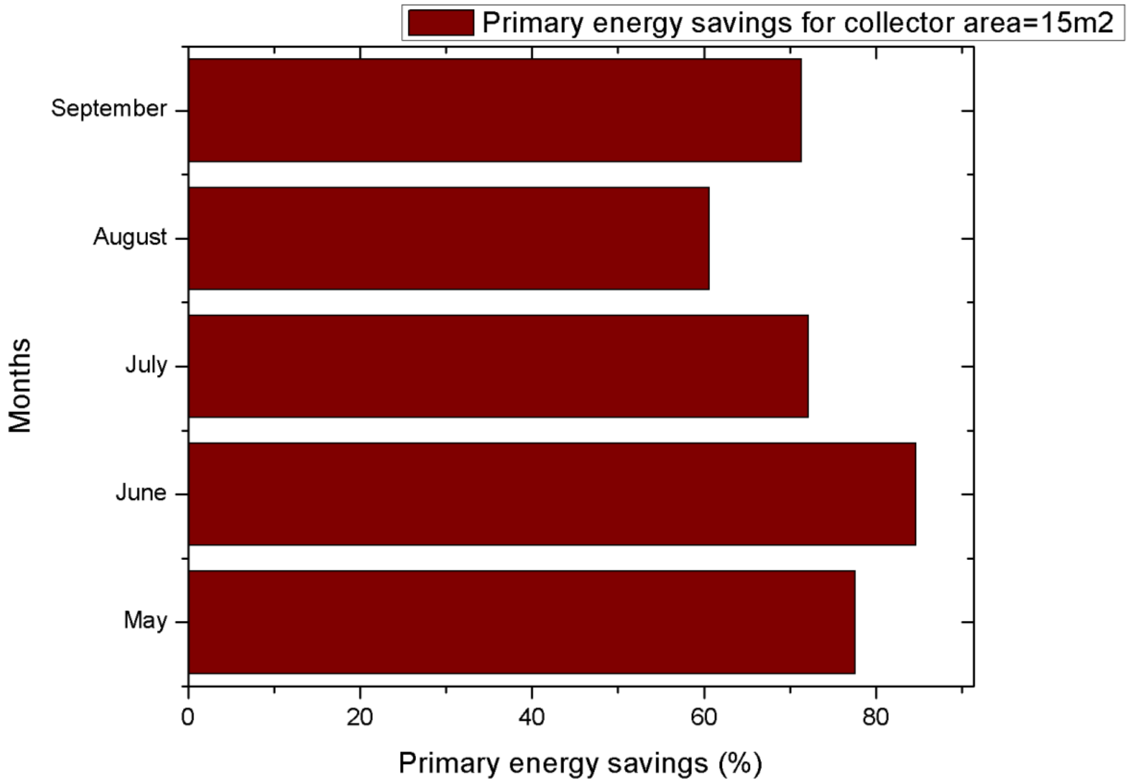

Figure 4-20: Months VS PES for C-2

4.3.8.3 Months VS Collector Efficiency

Thermal and electrical efficiencies are also calculated on the monthly basis which are shown in the figure below. Due to change in weather conditions, total solar radiations changed from month to month and due to this reason, the collector efficiency also varies slightly.

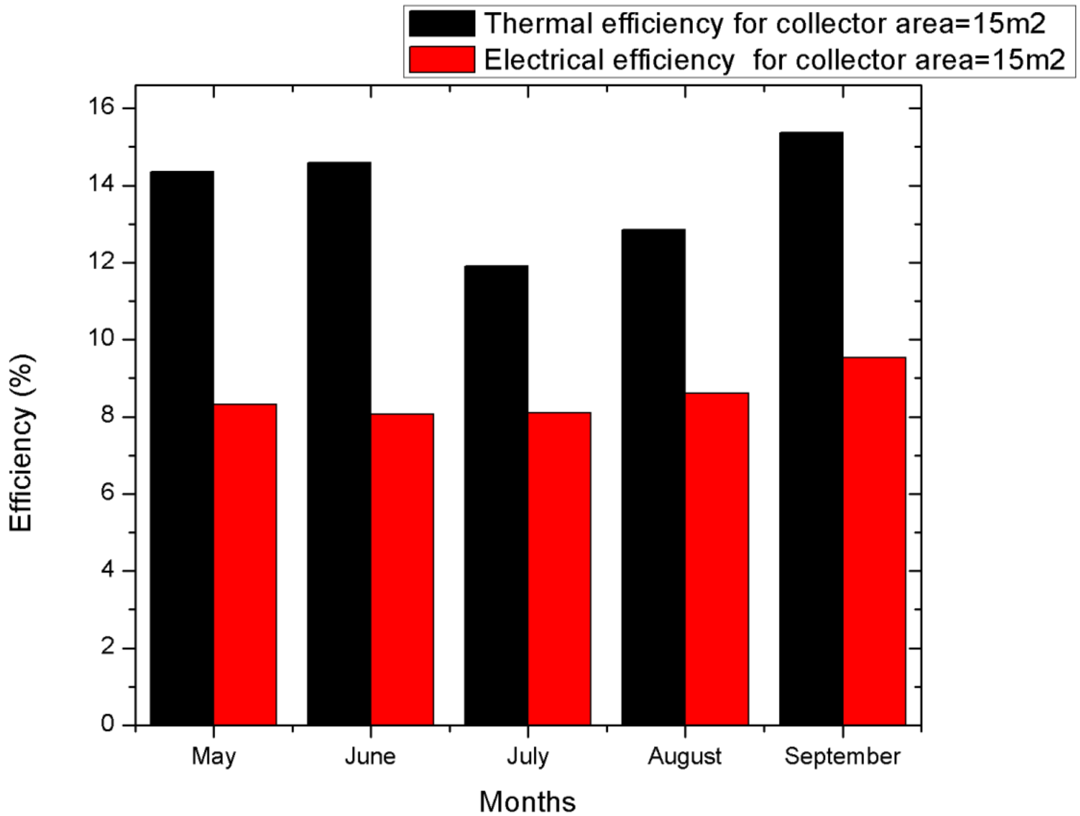

Figure 4-21: Months VS Efficiency for C-2

4.4 Results for Configuration-3

As it can be seen from the figure14 and figure15 that the only difference between configuration 2 and configuration 3 is that there is no direct evaporative cooler on the cooling side. As there is no difference in the first loop in both configurations that's why the flow rates for pumps and storage tank size is same for this configuration as like configuration2. While the other parameters are calculated which are shown below one by one.

4.4.1 Area VS Solar Fraction

Solar fraction for the whole season is calculated by setting the parameters based on different size of collector which is shown in the figure below. As the area increased solar fraction also increased. So, it upon the designer that how much area he selected depending upon the portion of energy he wants to obtain from solar energy.

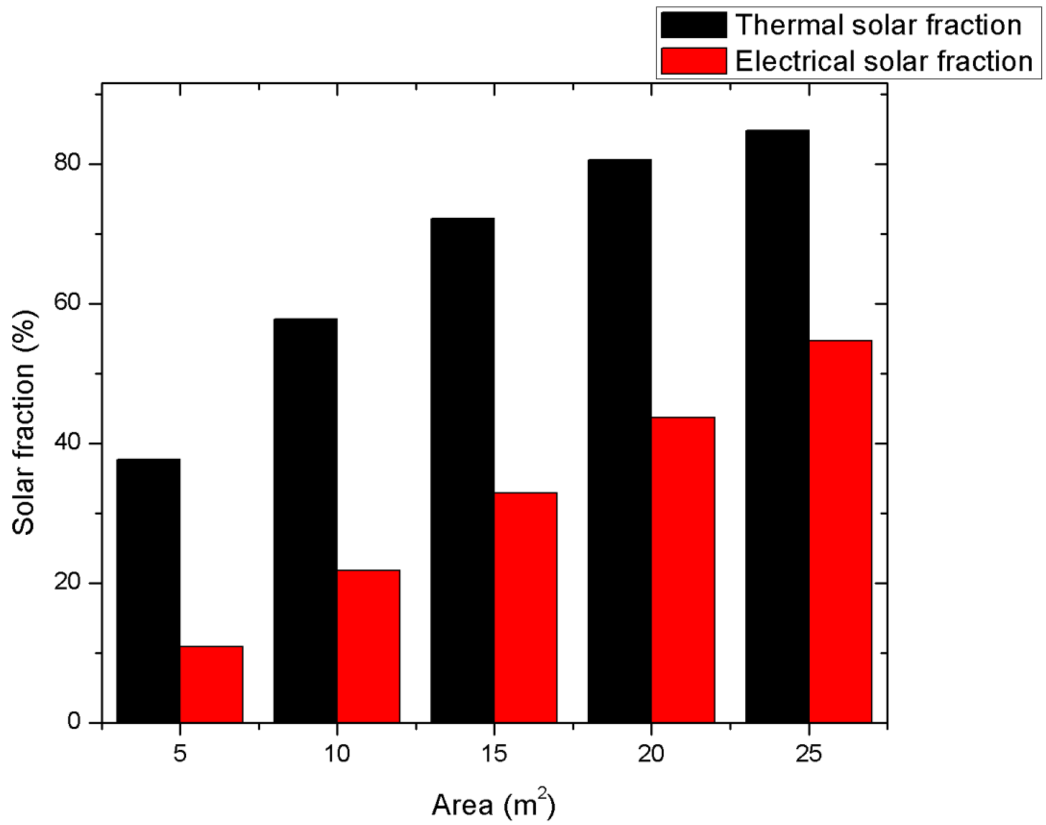

Figure 4-22 Area VS Solar Fraction for C-3

4.4.2 Area VS Primary Energy Savings

The calculated primary energy savings for the whole season are shown in the figure below. As the area increased PES also increased. So same as solar fraction it depends upon the designer that how much energy savings he wants through solar energy. In our case the size of collector is taken as 15m^2

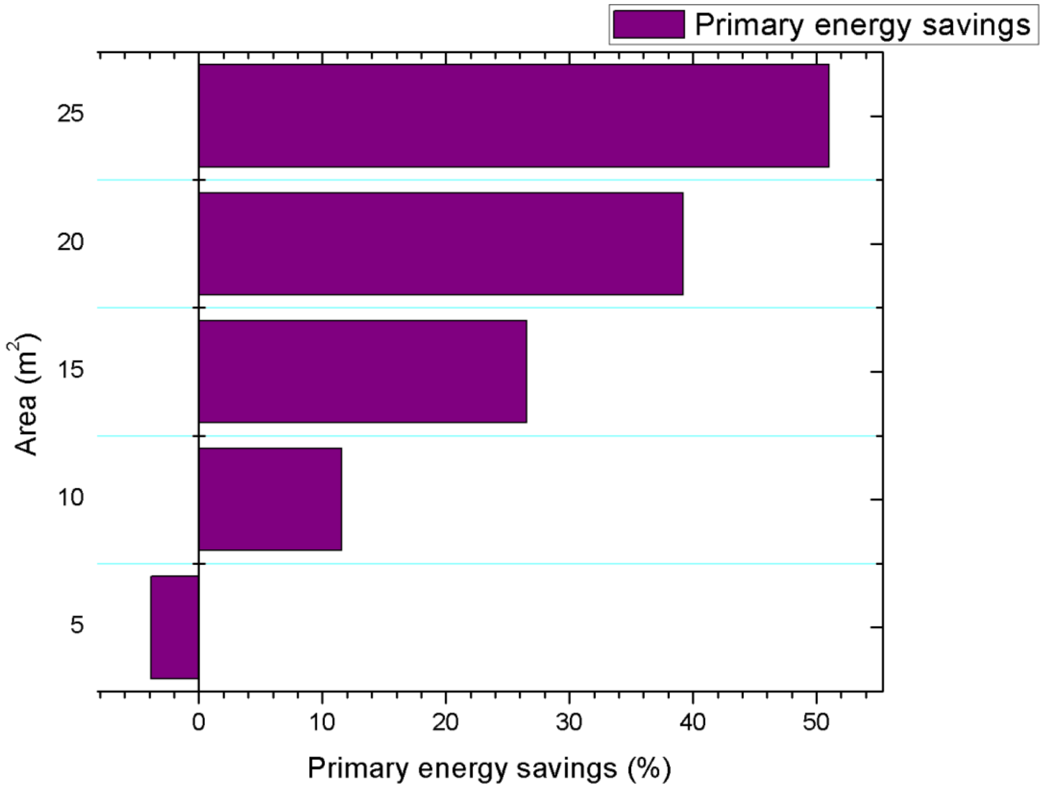

Figure 4-23: Area VS PES for C-3

4.4.3 Area VS Efficiency

The calculated amount of electrical and thermal efficiency of solar collector is shown in figure. These calculations are done for the whole season by varying the value of collector area. Electrical efficiency (I_L) depends upon the linearity of solar radiations that's why is remains constant while in the formula of thermal efficiency is inversely proportional to the area that's why with the increase in area thermal efficiency decreased.

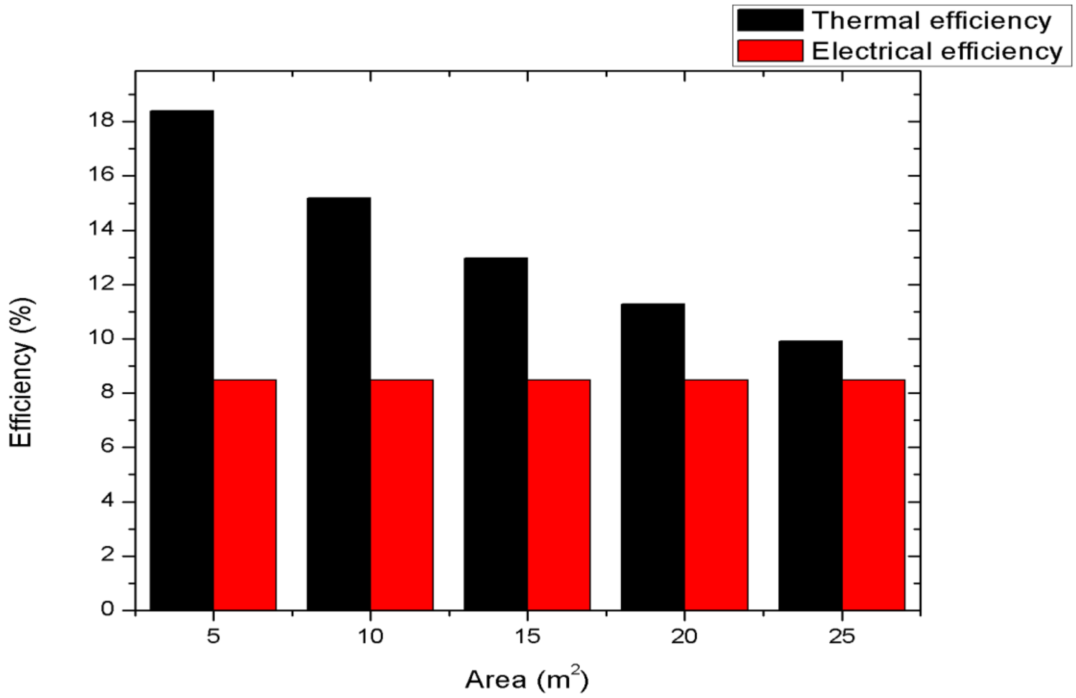

Figure 4-24: Area VS Efficiency for C-3

4.4.4 Monthly Results

4.4.4.1 Months VS Solar Fraction

Below graph shows the monthly solar fraction for the third configuration. These values are calculated for certain values of flow rates, collector and storage tank size.

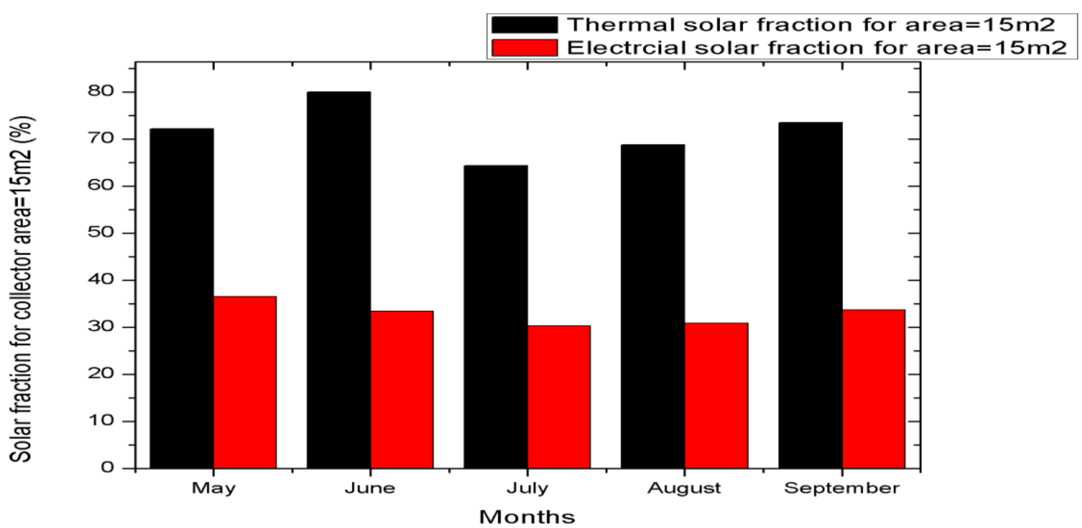

Figure 4-25 Months VS Solar Fraction for collector area=15m^2 in C-3

Similar to the C-2 and C-3 Solar fraction varies from month to month due to change in solar radiations and weather conditions on monthly basis.

4.4.4.2 Months VS PES

The primary energy savings for the given configuration are shown in the figure below. Similar to solar fraction PES also varies from month to month due to change in weather conditions

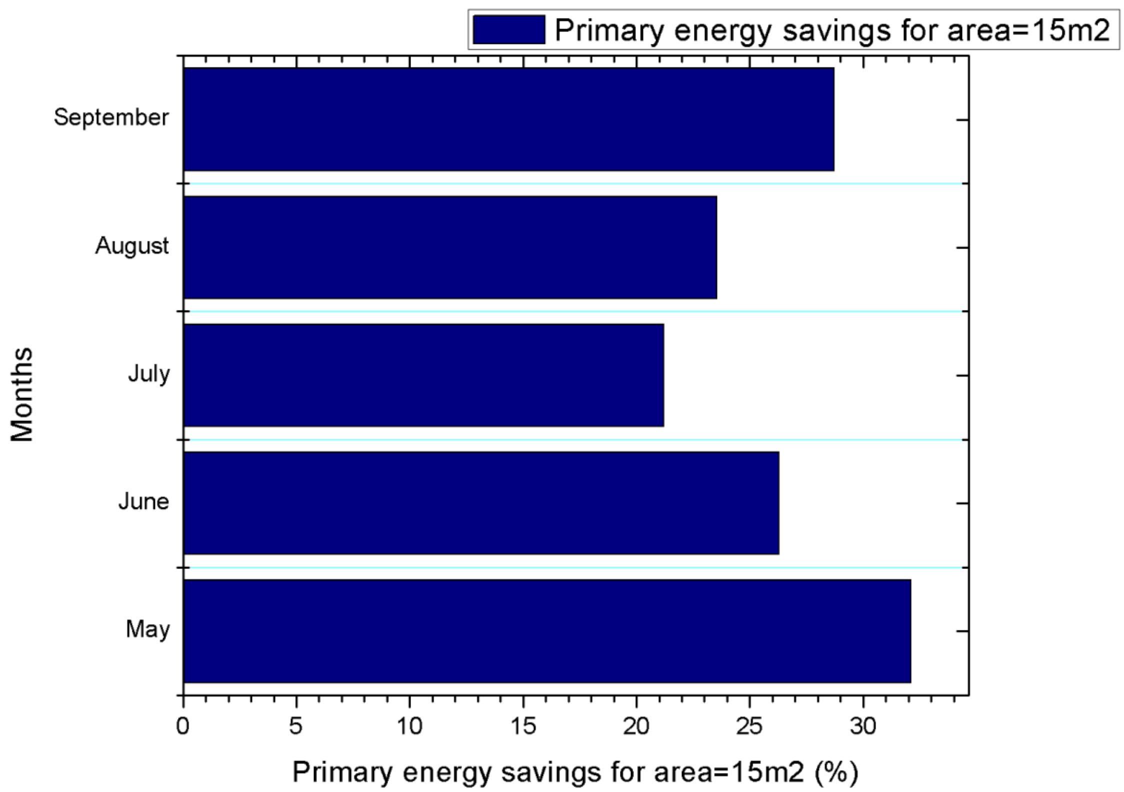

Figure 4-26: Months VS PES for collector area=15m² in C-3

4.4.4.3 Months VS Efficiency

The monthly efficiencies for the given configuration are shown in the figure. Similar to the C-2 and C-1 monthly efficiencies vary from month to month.

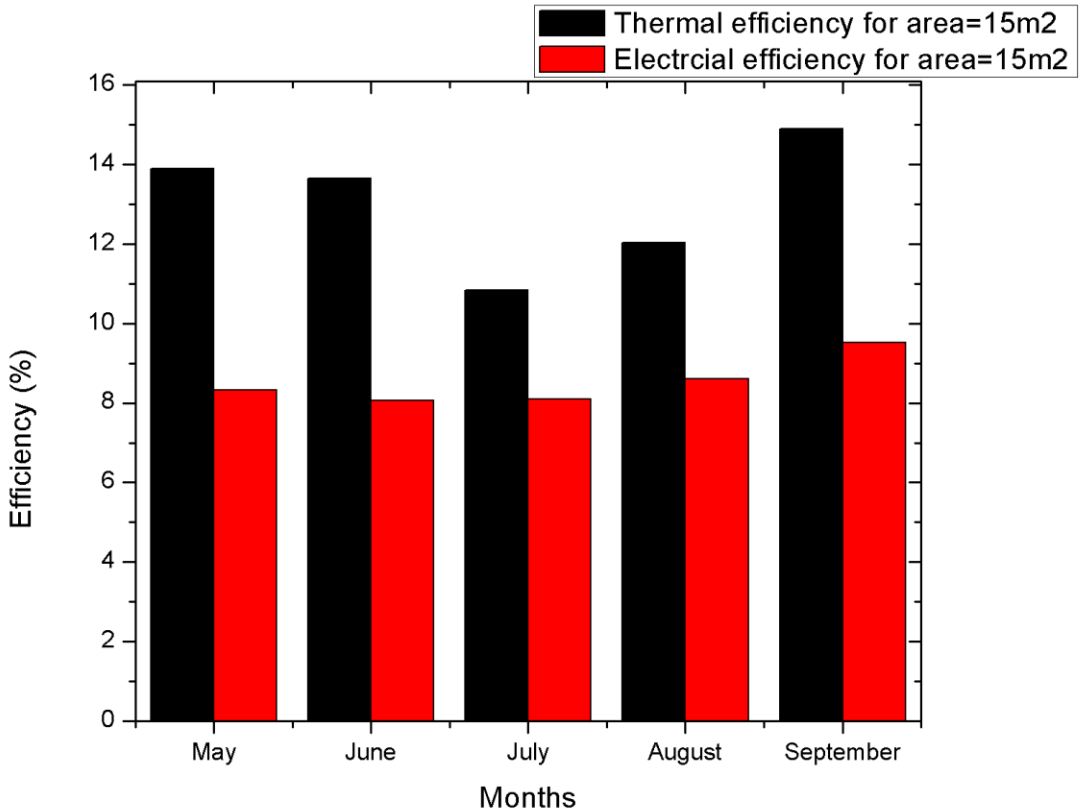

Figure 4-27 Months VS Efficiency for collector area=15m^2 in C-3

4.5 Performance Comparison

Performance of all the three configurations is compared based on thermal solar fraction, electrical solar fraction and primary energy savings. All the calculations are done for the whole season as well as on the monthly basis.

4.5.1 Area VS Thermal Solar Fraction

Thermal solar fraction for all the three configurations are calculated and compared. There is not much difference for the value of thermal solar fraction because the required amount of temperature to achieve is same and the rate of useful energy gain from the collector is also same.

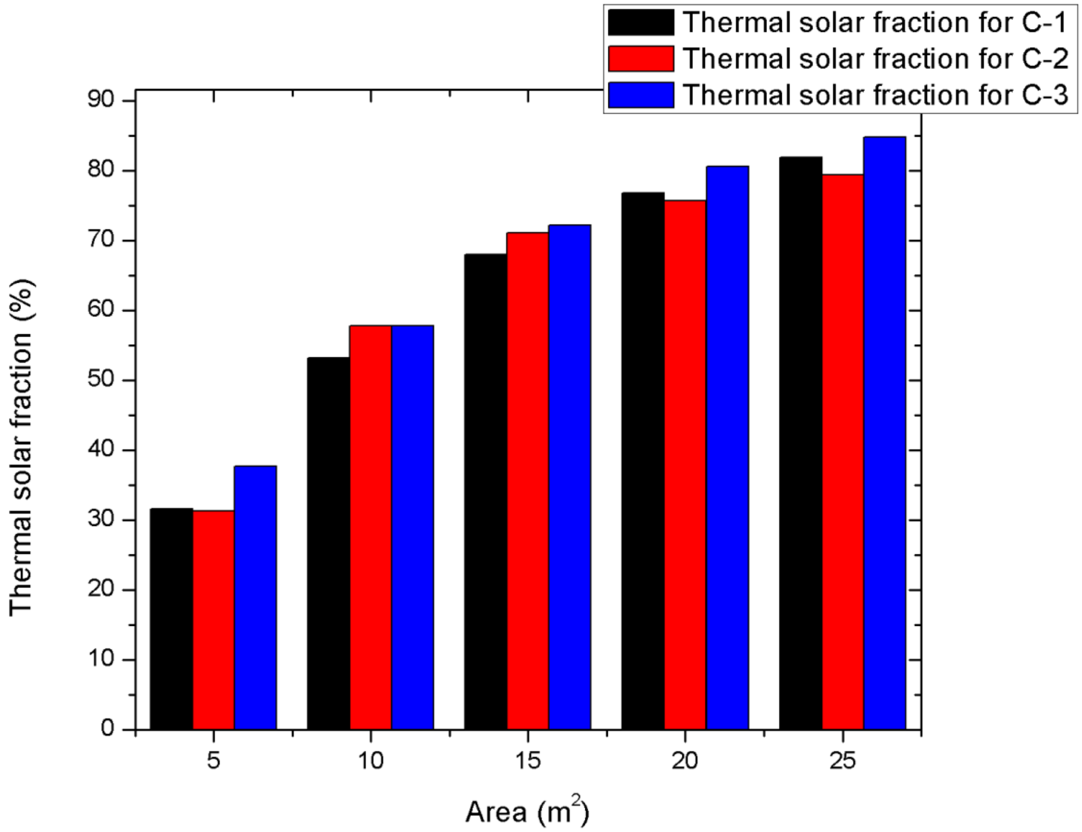

Figure 4-28 Comparison of Area VS TSF for all three configurations

4.5.2 Area VS ESF

The ESF for the configuration 1 and 2 is similar to some extent while the ESF for the configuration 3 is very low as compared to the above 2 configurations because there is no direct evaporative cooler and all the cooling is done by the vapor compression chiller which is the more energy extensive unit.

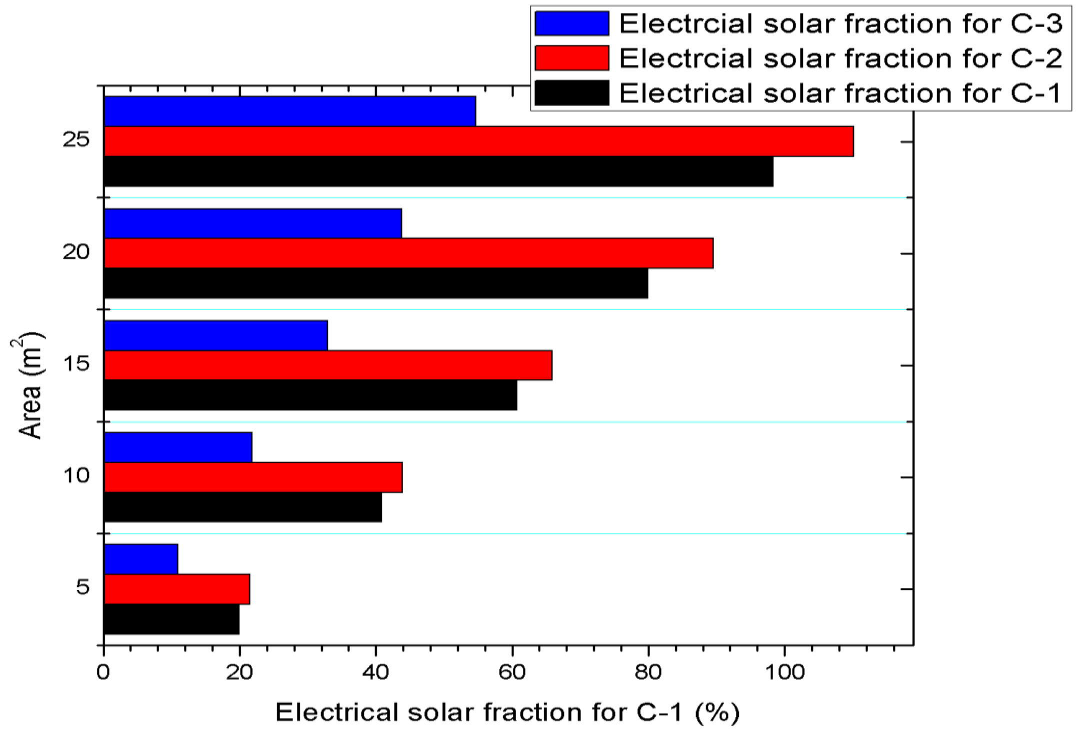

Figure 4-29 Comparison of Area VS ESF for all three configurations

4.5.3 Area VS Primary Energy Savings

Primary energy savings are calculated for the whole season for all the three configurations. The values of primary energy savings of configuration 3 is very low as compared to the configuration 1 and 2 due to the VCC. While the configuration 2 is better in the sense of primary energy savings from all the three configurations because in configuration 2 we check the temperature of water and heat it if required. That heated water remains in the loop through the tank while in configuration 1 we heat the air directly which goes out into the environment and in next cycle we need the same amount of energy for reheating.

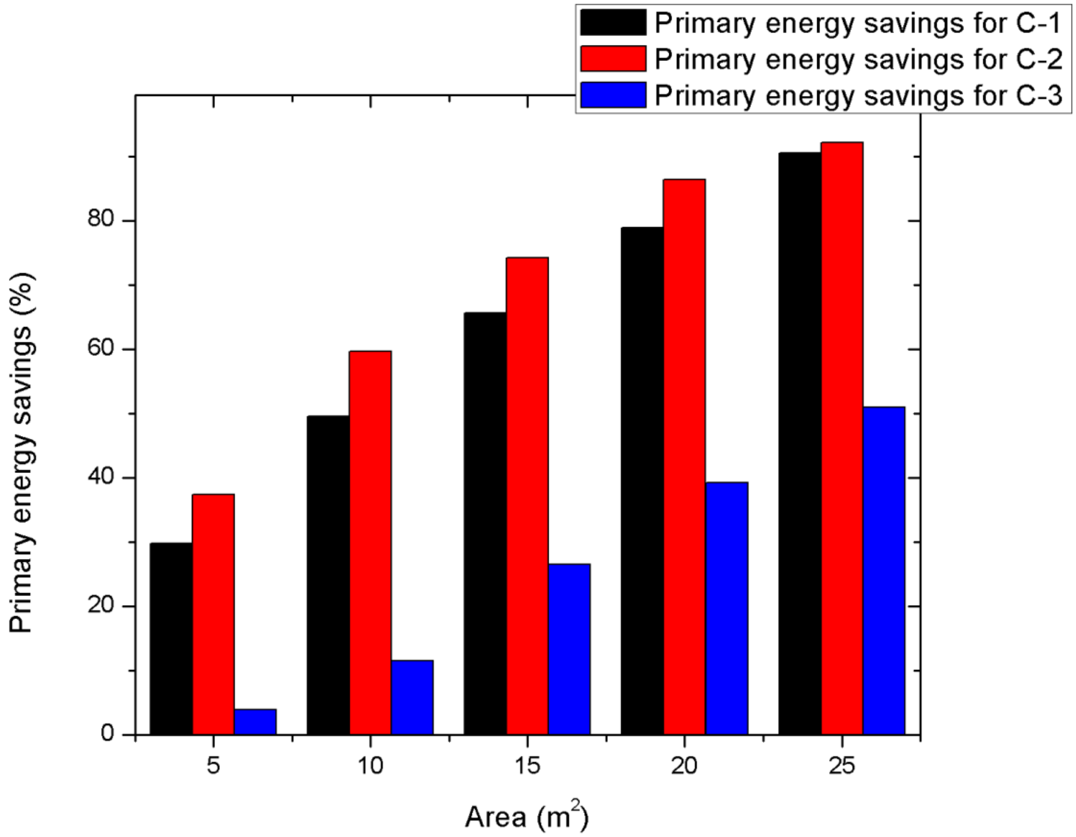

Figure 4-30 Comparison of Area VS PES for all three configurations

4.5.4 Months VS Thermal Solar Fraction

Thermal solar fractions are calculated on the monthly bases for the specific size of collector. Again, like figure 51 there is no major difference between the TSF of these three configurations.

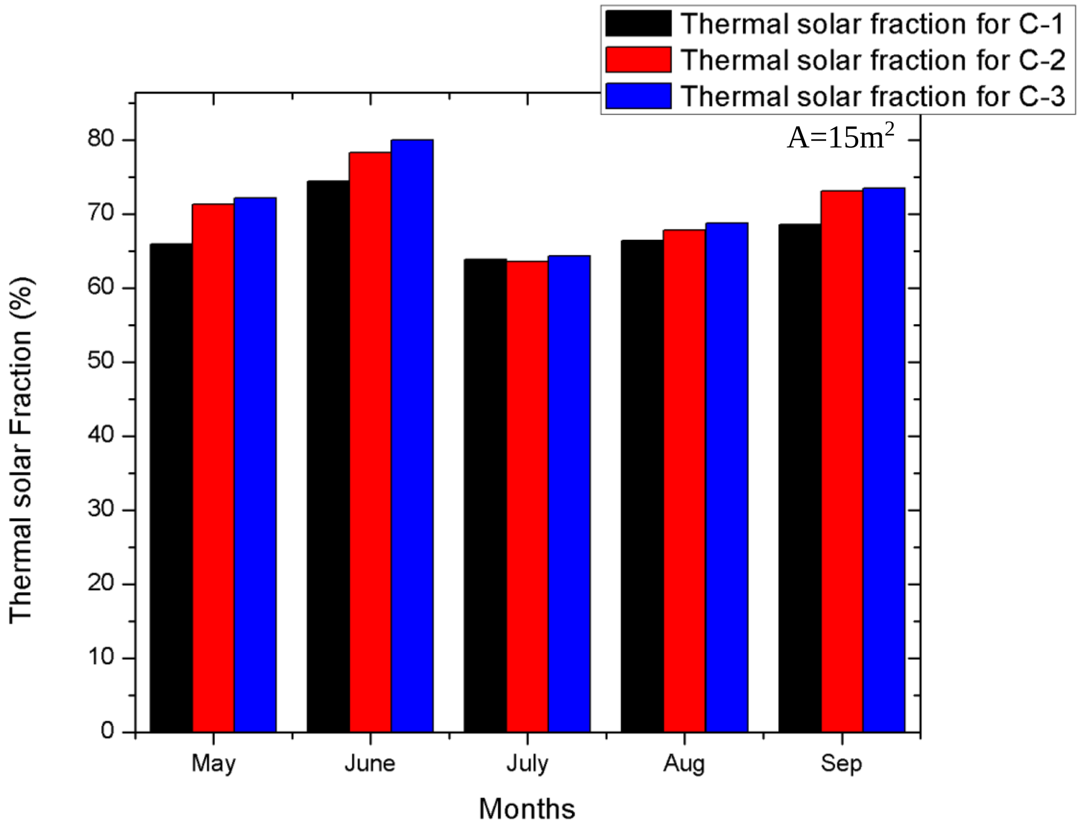

Figure 4-31 Comparison of Months VS TSF for all three configurations

4.5.5 Months VS ESF

The calculated amount of ESF VS Months for all the three configurations is shown in figure below. C-1 have the highest ESF while the ESF for the C-3 have the least. The reason behind the lowest ESF of C-3 is that, in the C-3 all the cooling is done by VCC which is more energy intensive unit while in C-1 and C-2 most of the cooling is done is DEC while VCC is used as auxiliary.

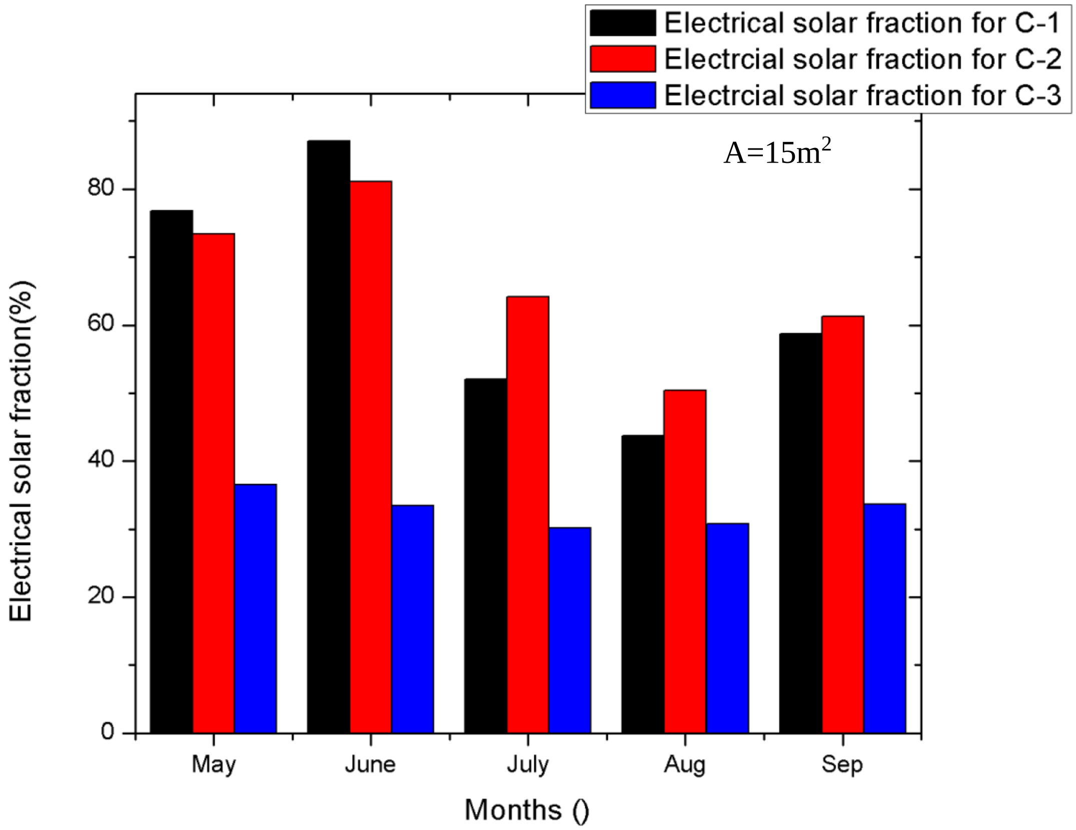

Figure 4-32 Comparison of Months VS ESF for all three configurations

4.5.6 Months VS PES

Graph for the PES for all the three configurations is shown below. C-2 have maximum PES because in C-1 we are heating the air which goes out after regeneration while C-2 we heat water which is used to heat the air by water to air heat exchanger and we circulate this water again. In the way Qboiler is more in the case of C-1 while less in C-2. While in C-3 the ESF is much lower as compared to the other configurations. That's the reason due to which PES are maximum in C-2. PES varies from month to month due to change in weather conditions.

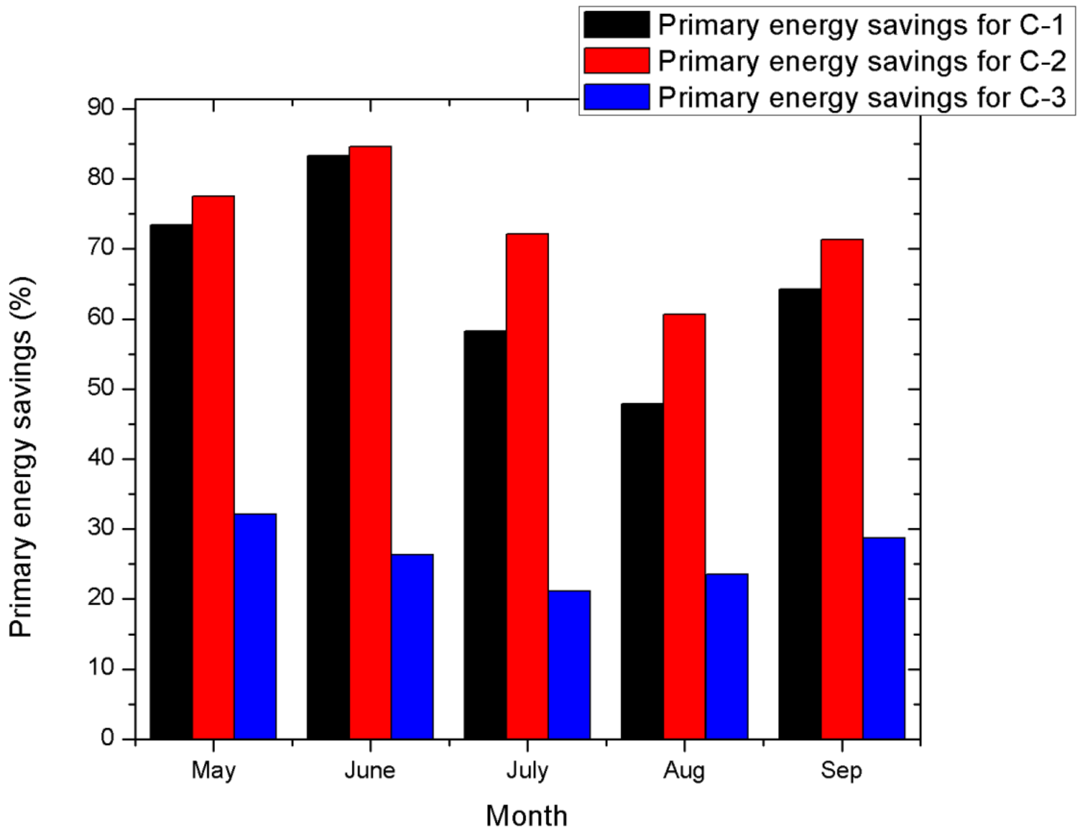

Figure 4-33 Comparison of Months VS PES for all three configurations

4.6 Day Profile for energy consumption:

System is simulated for a single day of July 2nd. As in each day system is simulated only for the 8 hours so the simulations for that 8 hours of july 2nd are shown in the figures 67, 68 and 69 for C-1, C-2 and C-3 respectively. Solar radiations, rate of useful energy gain of collector, power out from the collector, boiler's energy, energy consumed by direct evaporative cooler and vapor compression chiller are studied and analyzed.

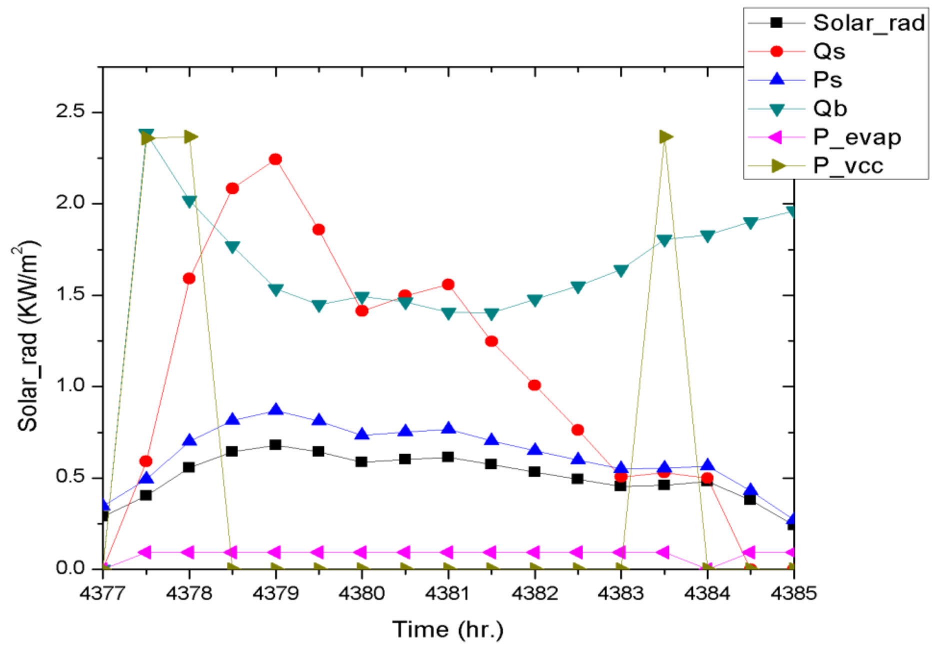

Figure 4-34 Energy profile for a single day C-1

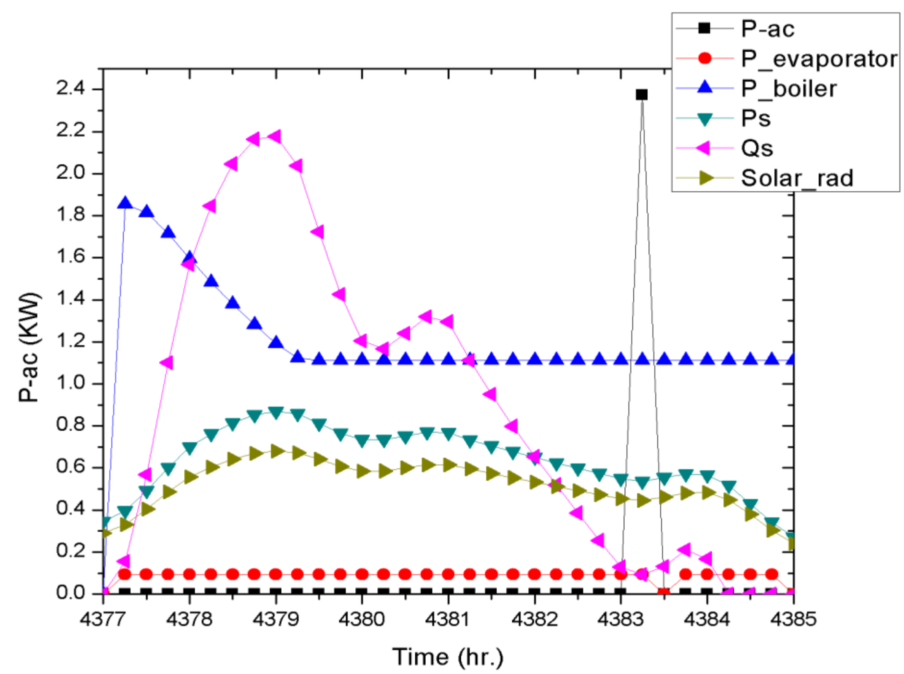

Figure 4-35 Energy profile for a single day C-2

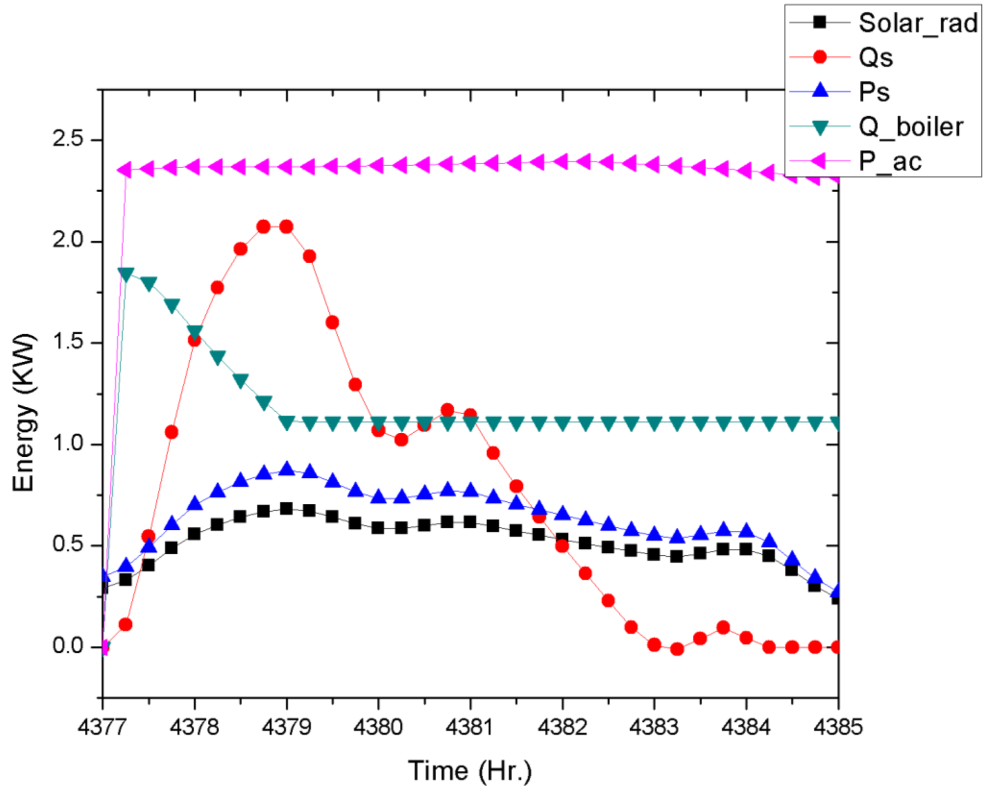

Figure 4-36 Energy Profile for a single day C-3

In C-1 and C-2 direct evaporative cooler is used to do the cooling and VCC chiller is just turned on in that time when the required temperature range is not achieved. VCC is much more energy intensive unit as compared to the DEC. In configuration1 Q_{boiler} is more as compared to the configuratin2 and configuration3 because in C_1 we are heating the regeneration air and in each cycle, we must heat the regeneration air with the same intensity and it goes out to the atmosphere after regenerating the desiccant wheel. While in C-2 and C-3 we are heating the water, which is used to heat the regeneration air via water to air heat exchanger. The water which we heat is recirculated via storage tank and in that way energy provided by the boiler is not wasted. This is the reason due to which we get more thermal solar fraction in these configurations as compared to the configuration1 while C-2 have maximum PES because in C-3 all the cooling is done by the VCC which is more energy intensive unit as compared to the DEC.

4.7 Temperature Profile for a single day

System is simulated for the single day of July 2nd and temperature profile is analyzed for the 8 hours of the July 2nd when the system is turned on during this day.

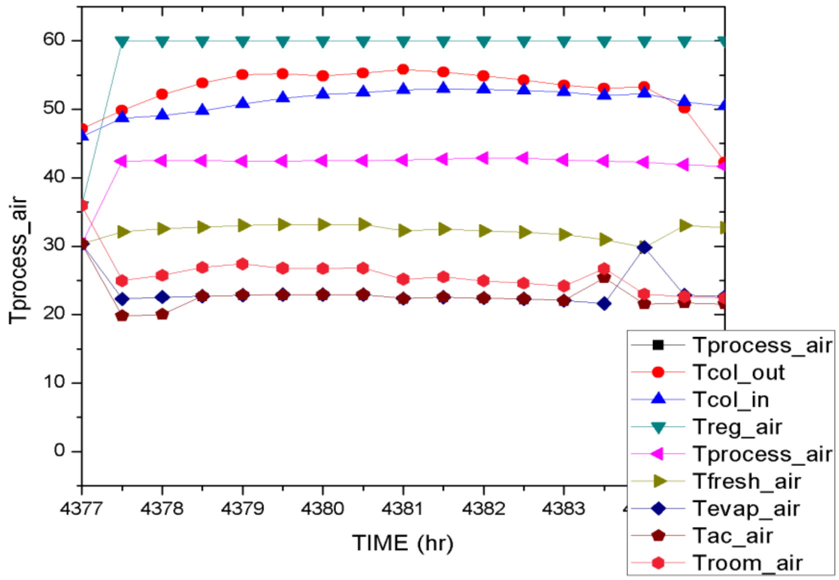

Figure 4-37 Temperature Profile for Single Day C-1

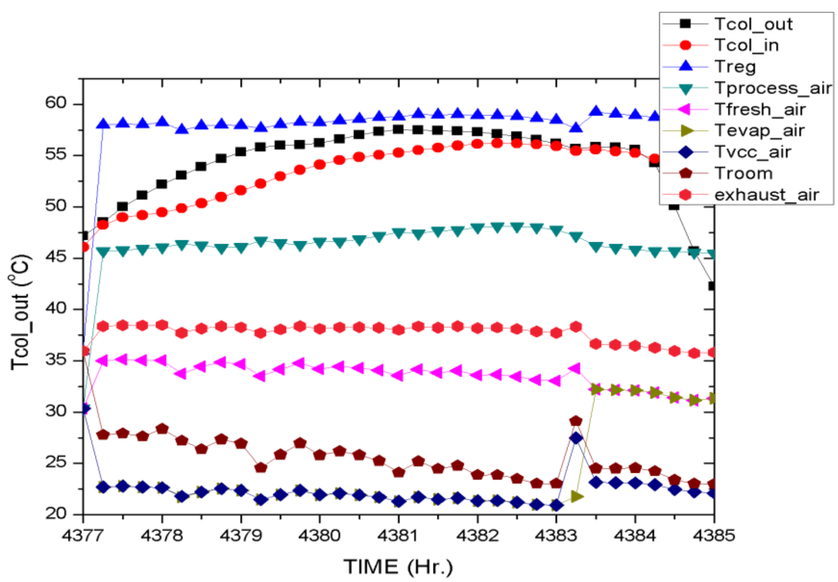

Figure 4-38 Temperature Profile for Single Day C-2

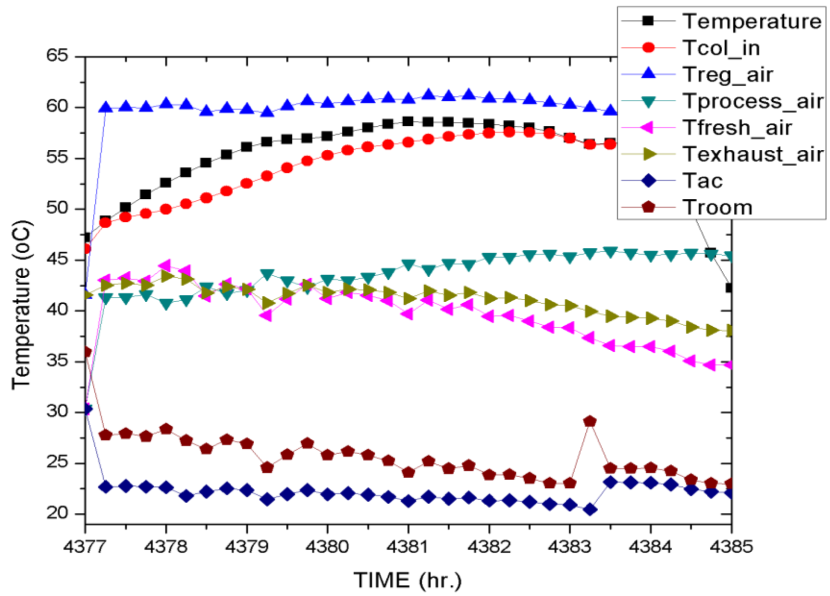

Figure 4-39 Temperature Profile for Single Day C-3

Regeneration air is the air which is used to regenerate the desiccant wheel. Normally desiccant wheel is regenerated between the temperature range of 50ºC to 80ºC. In our case temperature of regeneration air is kept as 60ºC. In C-1 smooth value of 60ºC is achieved because we placed the auxiliary heater in the loop of air. While in C-2 and C-3 boiler is placed to heat the water that is used to heat air via water to air heat exchanger that's why a range of temperature between 58ºC to 63ºC is achieved in that case. Temperature of the water entering and exiting the solar thermal collector are also witnessed while the room temperature shows that when it increases from 23ºC and required cooling is not achieved by the DAC then VCC is turned on and tripped after achieving the desired level.

CHAPTER#5

5 Economic Analysis

Economic analysis of the desiccant based cooling is carried out on RET Screen by considering the 74% energy delivered by the solar system while the 26% energy is delivered by the auxiliary energy source. Monthly primary energy savings are given in the table below.

Month	Percentage Primary Energy Savings
May	78%
June	85%
July	72%
August	61%
September	71%

Table 7 Monthly Savings for Economic Analysis

Parameters used for the economic analysis are given below

Parameters	Base Case	Proposed Case
Indoor temperature	23	23
Operating days per week	5	5
Operating hours per day	8	8
Slope of collector	21	21

Table 8: Parameters for Economic Analysis

Cooling capacity of the air conditioning system is 13KW while the COP of the system varies from 2.5 to 3. Cost of electricity used in 10 PKR/KWh. GHG emission credit is 8.20€/tCO2. Initial cost calculated is 500,000 PKR while the annual cost is 122,996PKR. The savings calculated per year are 190,658PKR.

Inflation rate used is taken as 5% while the project life is considered as 20 years. After the calculations the simple payback period calculated is 7.4years while the equity payback period is 6.4years which is shown in graph below.

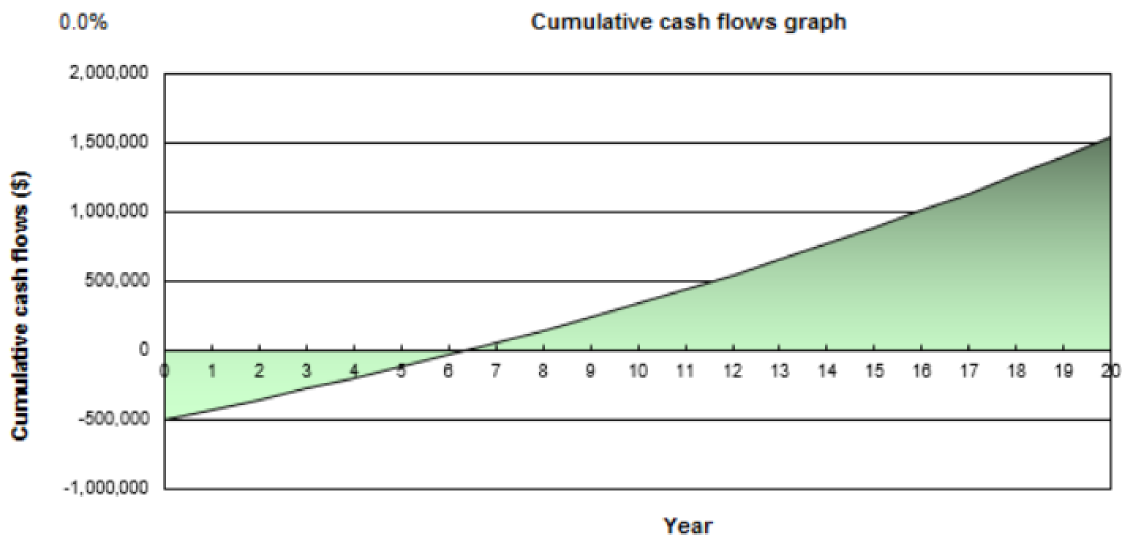

Figure 5-1 Cash Flow graph

CHAPTER#6

6 Conclusion and Future Work

6.1 Conclusion

In this study, three different configurations for desiccant based air conditioning system are modeled and simulated in TRNSYS 17.0. The main idea is to evaluate the thermal and electrical performance of the given system for the peak load of 13KW. Weather data file is for location of Islamabad (33.71 ̊N, 73.06 ̊E) is used to simulate the system for entire winter seasons. Collector slope is estimated from 15^o to 20^o. From all the three configurations maximum primary energy savings are obtained from the second configuration because as compared to the first configuration there is the savings in auxiliary heating system. We heat the water in C-2 which remains the loop and recirculates which reduce our energy demand while in C-1 we heat the air exiting from the desiccant wheel and in next cycle we need to heat it again with the same intensity. For the flat plate PV-T collector in this configuration, thermal efficiency is around 14% while the electrical efficiency is 8% for the collector size of $15m^2$. Thermal solar fraction obtained is 71% while the electrical solar fraction is 66% while primary energy savings obtained is 72%. Optimized size of storage tank is $0.5m^3$. Simulations results shows that configuration-3 is not feasible because due to absence of direct evaporative cooler all the cooling is done by the vapor compression chiller which is more energy extensive unit. The proposed system has very low GHG emissions as compared to the conventional air conditioning unit. No experimental data was available for model validation of simulation results, but overall trends of results were determined using already published data.

6.2 Future Work

In this study simulation results shows that there is a good potential for solar based air conditioning system in Pakistan.

However, the following ideas should be addressed in future work:

1. The proposed system can be simulated for different locations of Pakistan i.e. Lahore, Karachi, Abbottabad.
2. By addition of some components, the proposed model can be used to carry out for simulation of space heating.
3. The proposed system can be suggested for off grid areas by using thermosyphon collector and simulations can also be performed in TRNSYS.
4. Components used in TRNSYS can be designed in ANSYS/ Comsol
5. Experimental setup can be installed to validate the simulation based results.

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