Modelling and Simulation of Solar Absorption Cooling System using TRNSYS

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August, 2016

Declaration

I certify that this research work titled "*Modelling and simulation of solar absorption cooling system using TRNSYS"* is my own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources it has been properly acknowledged / referred.

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Language Correctness Certificate

This thesis has been read by an English expert and is free of typing, syntax, semantic, grammatical, structural and spelling mistakes. Thesis is also according to the format given by the university.

Signature of Student

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Abstract

In this study, performance analysis of solar assisted single effect absorption cooling system is performed for two system configurations. The analysis is carried out to meet a cooling demand of 298 kW (85TR) for an educational building located in Islamabad (33.71° N, 73.06° E). In configuration-1 (C-1), the return water from absorption chiller always flows towards hot water storage tank which is connected to the solar collector. In configuration-2 (C-2), the return water from absorption chiller may become isolated from the collector-storage tank loop if the water temperature in the storage tank is less than the required temperature i.e. 110℃. Both system configurations are modelled in TRNSYS and dynamic simulations are carried out for the entire summer season. Various performance factors such as solar fraction, collector efficiency and primary energy savings are evaluated to optimize the system components which includes collector tilt, type and size of collector, storage volume. Results demonstrate that C-2 in conjunction with evacuated tube collectors (ETC) results in minimum area per kilowatt of cooling demand and higher monthly collector efficiency than C-1.

Key Words: *Absorption cooling, solar cooling, TRNSYS, solar fraction, primary energy savings*

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Nomenclature

Abbreviations

COP	:	Coefficient of performance
ETC	:	Evacuated tube collector
FPC	:	Flat plate collector
FPES	:	Fractional primary energy saving
SAC	:	Solar absorption cooling
SCS	:	Solar cooling systems
$SPF_{ref.}$	:	Reference seasonal performance factor
SHC	:	Solar assisted heating and cooling system
TEWI	:	Total equivalent warming impact
TR	:	Tons of Refrigeration
TRNSYS	:	Transient system simulation
TMY	:	Typical meteorological year
VCR	:	Vapor compression refrigeration

Nomenclature

List of Symbols

A_c	m^2	Area of Evacuated or Flat plate type solar collector
A_i		Surface area of the ith tank segment
a_o		Optical efficiency
a_1	$\mathrm{kJ\ hr^{-1}\ m^{-2}\ K^{-1}}$	Efficiency slope
a_2	$\mathrm{kJ\ hr^{-1}\ m^{-2}K^{-2}}$	Efficiency curvature
a_{season}		Offset for season
b_{season}		Multiplier for season
β		Collector tilt
C_{pf}		Specific heat of the tank fluid
$Capacity_{Rated}$	$\mathrm{kJ\ hr^{-1}}$	Machin's rated capacity
$DesignLoad$	$\mathrm{kJ\ hr^{-1}}$	The design heating or cooling load
$EndOfSeason$		The hour of year at which specified season ends
ε_{heat}		Boiler efficiency
$f_{sav,shc}$		Fractional primary energy saving for a solar heating cooling system
$f_{DesignEnergyInput}$		Fraction of design energy input
G_T	$\mathrm{kJ\ hr^{-1}\ m^{-2}}$	Incident global solar radiation
$Load$	$\mathrm{kJ\ hr^{-1}}$	The heating or cooling load as a function of time
M_i		Mass fluid in the ith section
$\dot{m}_h$		Fluid mass flow rate to tank from heat source
$\dot{m}_L$		Fluid mass rate to the load
Q_u	$\mathrm{kJ\ hr^{-1}}$	Useful gain energy
$StartOfSeason$		The hour of year at which specified season begins
T_{env}		Temperature of the environment surrounding the tank
T_i		Temperature of the ith segment

X_{day}	Modifying function for seasonal variations in the loads
X_{hour}	Modifying function for hourly variations in the loads
$X_{noise,hour}$	Modifying function for hourly noise in the calculated loads
$X_{noise,day}$	Modifying function for daily noise in the calculated loads
η	Solar collector efficiency
$\emptyset$	Latitude
δ	Declination

CHAPTER 1: INTRODUCTION

1.1 Background

Word energy consumption is rising drastically. In past two decades global primary energy demand has grown by 49 %, with 2% annual increase [1]. It is predicted by researchers that the rising trend will continue [2]. The energy is consumed by three main sectors, industrial, transportation and ‗other', the last named is subdivided as; agriculture, service sector and residential [1]. Service sector includes all commercial and public buildings such as; schools, hotels, restaurants, hospitals, etc. In developed countries buildings utilize about 20-40% of the final energy consumption [3]. The average annual energy consumption growth rate in buildings for developed and underdeveloped countries are 1.1 % and 3.2% respectively [3].

The service sector utilizes energy mostly for HVAC, water heating, domestic use, refrigeration and lighting [1]. As the global population is increasing energy demand for buildings is escalating to provide services and comfort [3].

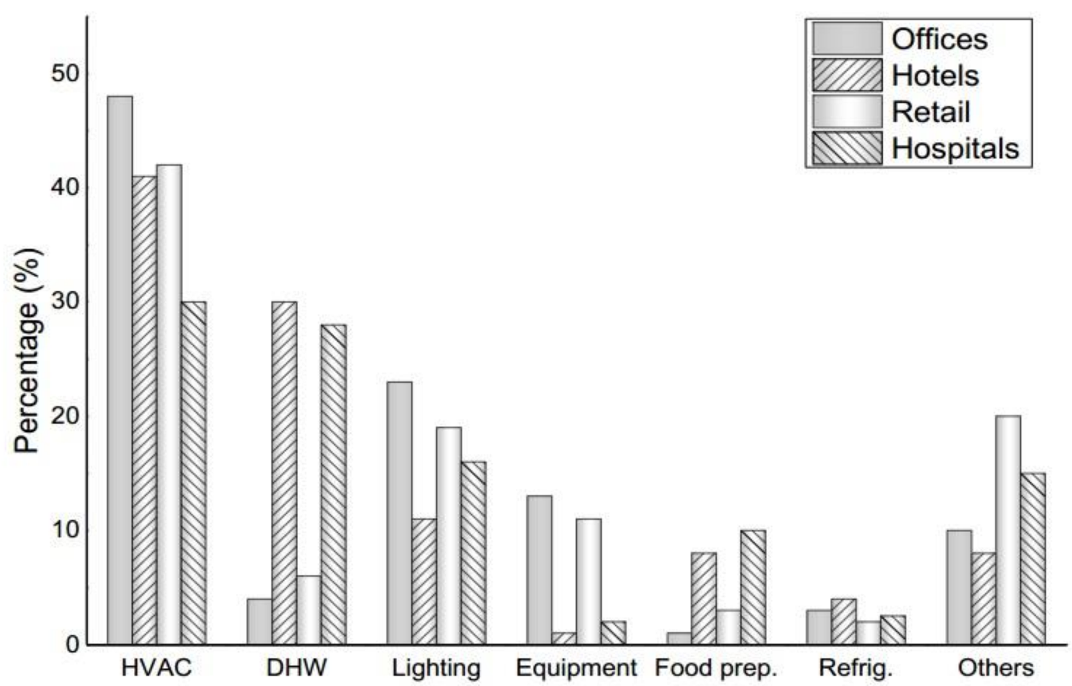

Figure 1-1: Consumption of energy by end use for different building types [1]

Figure 1.1 shows the distribution of energy end use in different types of building. About 30-50% of the total energy need in commercial buildings (offices, hotels, retail and hospitals) is required for HVAC [1,4]. Energy consumed in other applications (ventilation, water heating, lighting etc.) is comparatively less. Therefore, it is requirement of the time to control the huge energy consumption for HVAC purpose.

1.2 Pakistan Energy Scenario

Energy scenario overview indicates that Pakistan is an energy deficient country [5]. Energy infrastructure of the country is under developed and not well managed [6]. And it is facing worse energy crises due to increase in energy demand and reduction in investment for exploring indigenous energy resources [6]. Fossil fuels are contributing 60% in the primary energy mix [5]. Figure 1.2 shows the share of each source in electricity generation and electricity consumption in different sectors [6]. It is clear from the figure that more than half of the electricity is consumed by domestic and commercial sectors [2]. In these sectors electricity is mainly used for HVAC application. Therefore, it is essential to reduce the load from electricity by introducing some alternate renewable energy sources for this purpose.

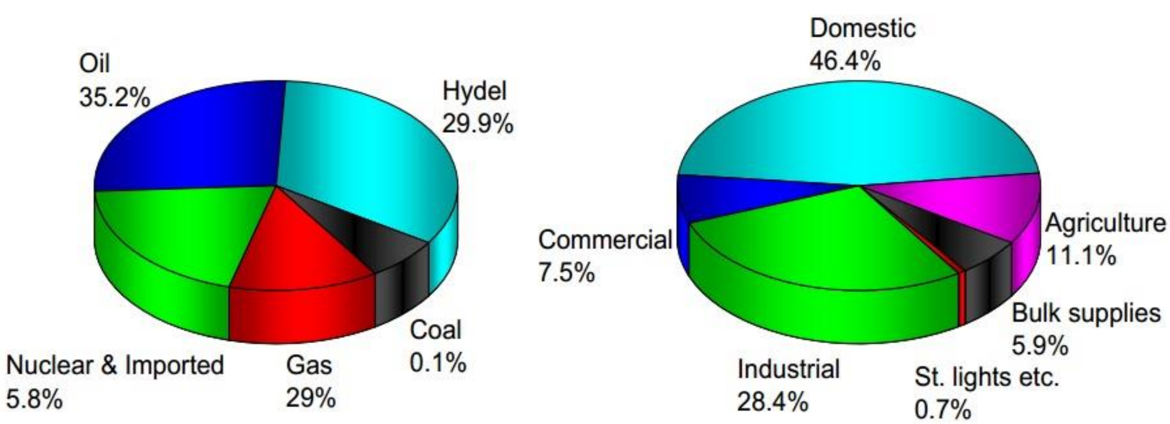

Figure 1-2: Share of diverse sources in electricity generation and consumption of electricity in different sectors of Pakistan

1.3 Conventional HVAC Systems

The demand of air conditioning for domestic and commercial sectors is rising with the boost in consumption of energy. To meet the cooling demand, vapor compression systems (VCS) are commonly used all over the world. These systems are known as conventional systems [7].

These systems are most common because their coefficient of performance (COP) is higher, size is smaller and weight is less as compared to other cooling systems [7]. Meanwhile they use the refrigerants: hydro-chlorofluorocarbon (HCFC), chlorofluorocarbon (CFC) and hydro-fluorocarbons (HFC) [7]. The emissions of these refrigerants are contributing to the depletion of ozone layer and causing heating of environment (global warming) for the last 60 years [4]. Currently global warming is a serious issue and researchers are finding ways to eliminate these emissions [8]. Another central disadvantage is that, these systems are very energy intensive and require large capital investment to fulfill the electricity demand. Therefore, government has to install new power plants and have to modify the distribution and transmission networks [9] .

1.4 Solar Cooling- An attractive alternative

In account of the problems due to conventional systems, solar absorption cooling systems are of great interest as, they have huge potential to mitigate environmental issues [10,11]. These facilities use thermal energy in combination with absorption cycle for refrigeration. The utilization of energy from sun for refrigeration and air conditioning is an attractive alternative because resource availability is apparently coincide with cooling demand [8]. Therefore, these systems can be used for reduction in electricity peak load demand and are appropriate to fulfill the cooling load in buildings [12,13]. Thermally powered absorption refrigeration systems have no ozone depletion potential [7]. Moreover, they utilize natural environmental friendly refrigerants such as; ammonia, water and methanol, etc. Energy from sun and waste heat can be efficiently used to operate these types of systems [7]. As the energy from sun is green energy therefore, these systems have the potential for reduction in emission of greenhouse gases [12]. In addition, the maintenance requirement for these systems is very less because, there are no rotating parts.

Pakistan is the country which is currently facing shortfall of electricity especially during summer season [14]. An extensible amount of electricity is used for air conditioning by domestic and commercial sectors. Therefore, it is more viable for Pakistan to reduce the electricity load by using solar absorption cooling (SAC) systems for air-conditioning [15]. The geographical location of Pakistan is ideal for using SAC systems because summers season is very long and country lies on sunny belt. On average most parts of the country have 300 days of sunshine per year with 1900-2200 kWh/m^2 of global irradiance. It shows that Pakistan is one of the richest country in term of solar potential [16]. The price of electricity is also very high in the country, which make these systems economically more attractive [14].

1.5 Advantages of Solar Cooling Systems

The advantages which can be obtained by using solar absorption cooling systems are given below.

i. They can reduce the dependency on imported fuel.
ii. The energy supply can be diversified by using these systems.
iii. Have potential to save limited natural resources.
iv. Reduces CO_2 emission at very low cost.
v. Control the urban air pollution.
vi. Owner of these systems can save money by reducing their electricity bills.
vii. They have potential to create local jobs and stimulate local economy

1.6 Passive and Active Solar Cooling

Solar cooling is not a new concept. It had been used by the name of passive cooling from the ancient times [17]. Passive cooling systems require no power or very little power for their operation because pumps are not used in these systems. They are energy efficient and provide improved thermal comfort [18]. They provide cooling by removing heat to the atmosphere or preventing heat from entering into the building and mainly depends on the building architecture [18]. There are three main characteristics of building, having influence on thermal conditions of the building [19]:

1) The transmission of the outer atmosphere across the building material. As the building material changes the transmission in accordance with the thermal mass of the building and the insulation provided.
2) The transmission of direct radiations across the glazed and open area of the building (e.g. windows and doors). It can be forbidden by applying blinds and curtains.
3) Outside air Infiltration through opening such as from windows, door and cracks in the building. Infiltration can be reduced by using window control ventilation.

By controlling these parameters thermal heat gain can be reduced to provide passive cooling in the building. However, alone passive cooling is not sufficient to attain required thermal comfort because heat can‘t be removed from required space. Therefore, active cooling technologies are also essential. An active cooling system is one that involves the use of energy to cool required region [18]. Solar absorption cooling (SAC) systems are attractive options to provide active cooling in the buildings [20].

1.7 Active Solar Cooling Systems

The active solar based cooling systems are generally divided in two major groups, depending on the way how energy is delivered. Classifications of these groups are given below [21].

1. Systems driven by electric energy
   a) Vapor compression refrigeration system
   b) Stirling refrigeration system
   c) Thermo-electric refrigeration system
2. Systems driven by thermal energy
   a) Absorption refrigeration system
   b) Adsorption refrigeration system
   c) Chemical reaction refrigeration system
   d) Desiccant cooling system
   e) Ejector refrigeration system

1.8 Solar Electric Refrigeration

1.8.1 Vapor Compression Refrigeration System

This is an electric refrigeration technology. These systems are powered by using photovoltaic panels commonly known as solar cells. These cells are manufactured from large variety of construction materials. The type of material from which solar cells are manufactured determines the efficiency and cost of solar cells. The most common material in the manufacturing of solar cells is silicon (semiconductor material) [22]. Solar panel efficiency is the ratio of output power *"W"* to the product of collector area *"A"* and direct solar radiations "I_p" which is given in eq. (1.1).

$$\eta_{solar} = \frac{W}{A*I_p} = \frac{W}{Q_s} \tag{1.1}$$

Solar panels available in the market have efficiency of approximately 15 %, while building integrated panels have 10% overall efficiency [23]. The attractive advantages of using solar panel in combination with VCS are: easy assembly and high overall efficiency [22]. A simple systematic diagram of solar vapor compression system is presented in Figure 1.3.

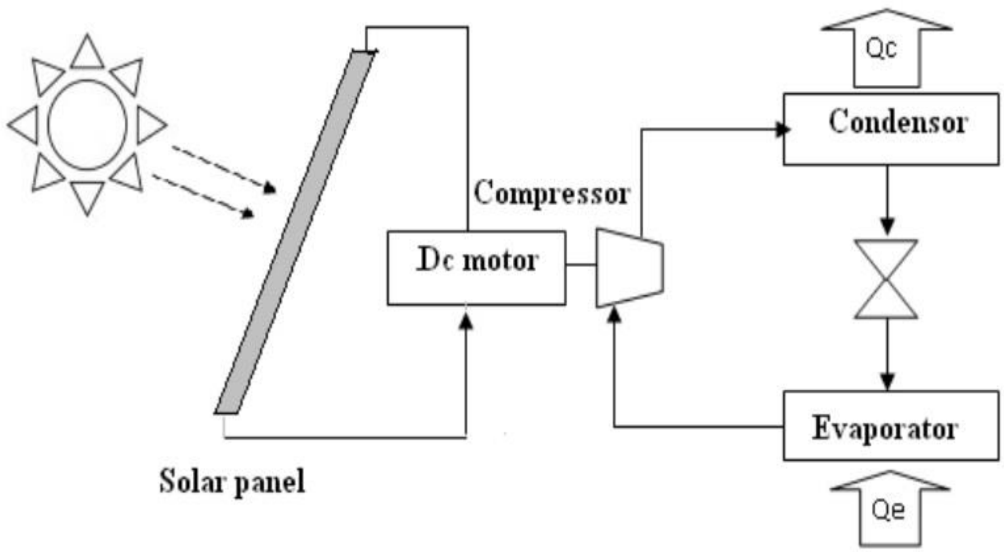

Figure 1-0-3: Solar vapor compression air condition systemic diagram [22]

Efficiency of refrigeration system depends upon cooling capacity"Q_e" and work input *"W"* which is given in eq. (2.2).

$$\eta_{cool} = \frac{Q_e}{W} \tag{1.2}$$

Overall efficiency depends on the product of the individual efficiencies of solar panel and refrigeration system. Overall efficiency of the system is given in Eq. (2.3)

$$\eta_{solar-cool} = \eta_{solar} * \eta_{cool} = \frac{Q_e}{Q_s} \tag{1.3}$$

1.8.2 Thermo-electric Refrigeration

PV solar panels can also be integrated with some other electric refrigeration systems. Thermo-electric technology is one of these technologies. Thermo-electric element is made of semiconductor. Other commonly used construction materials are bismuth telluride (Bi_2Te_3) and antimony telluride (Sb_2Te_3) alloys. These systems are very small including no moving parts or refrigerants. They can be used in transportable refrigerators and electronic chip cooling. Due to their limited size they are very attractive for the applications in satellites and space ships. Currently these systems have low COP ranging from 0.3 to 0.6 and small systems are available in the market having capacity of few hundred watt [22].

1.8.3 Stirling Refrigerator

Solar panel integrated with Stirling refrigerator can also be used to provide space cooling. Ideally sitrling refrigerator works as Carnot cycle. COP value of this system is lower than those of vapor compression systems [24]. The COP of the system ranges from 1.2 to 1.6 that depends upon cold / hot reservoir temperature and also on ambient temperature [25]. To develop an efficient Stirling system, many technical problems emerge. Most common technical problems are: low power density and poor heat transfer in between atmosphere and working fluid [26]. These problems are dominated in larger systems. Therefore, only smaller stirling systems are competitive with conventional VCR systems [22].

1.9 Solar Thermal Refrigeration

In these systems, cooling effect is produced by solar using thermal energy rather than solar electric energy. Solar collectors are used for this purpose. Broadly solar thermal collectors are classified into two types: (a) Concentrating solar collectors (b) Non concentrating collectors. Non concentrating collectors have very low cost than concentrating collectors but provide water which can only be used for single effect absorption refrigeration systems which are not so efficient [27,28]. Types of non-concentrating collectors, which are easily available in Pakistan to harvest solar energy are

1. Flat plate collectors
2. Evacuated tube collectors

Flat plate collector (FPC) is most commonly used type of solar collector in the world [29]. It contains metallic absorber to absorb solar radiations. An anti-reflective coating with glass plate can be adopted to reduce the losses [30]. Evacuated tube collector (ETC) is made from evacuated glass tubes containing metallic absorber inside. The systematic diagram of these two types of collector is given in Figure 1.4 [22].

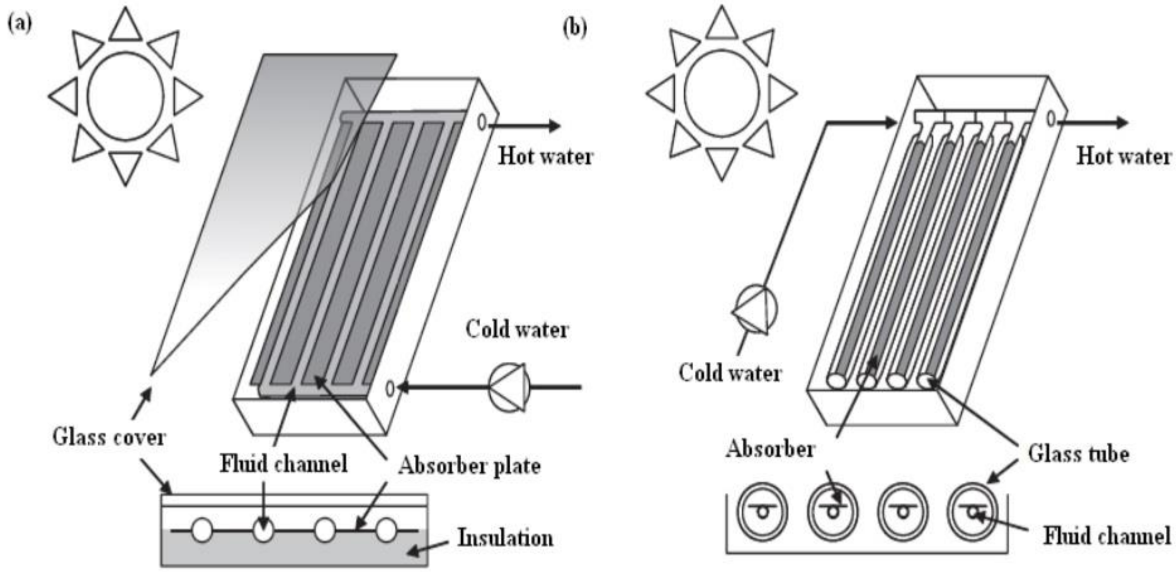

Figure 1-4: Systematic diagram for (a) Flat plate collector (b) Evacuated tube collector

These collectors provide heat to thermally driven refrigeration system. The efficiency of solar thermal collector depends upon its working temperature, ambient temperature and solar radiation falling on the surface of solar thermal collector. As working temperature increases the collector efficiency decreases. While, efficiency of absorption chiller or thermal compressor (heat engine) increases with the increase in working fluid temperature. The overall system should be designed by taking into account these two opposite trends [13].

1.9.1 Absorption Refrigeration system

Absorption cooling system is also known as liquid sorption system because in the absorber liquid phase absorbent is present [31]. It is the most commonly implemented technology for solar air conditioning applications. It needs very less or no electricity to operate. For equal capacity the size of absorption machine is smaller than adsorption because heat transfer coefficient of absorbent is higher [22]. Moreover, as these systems are operated by using heat energy unlike vapor compression refrigeration (VCR) systems, therefore, they do not need compressors for the compression of gas therefore, their maintenance requirement is very less [31]. Absorption refrigeration system requires three thermal reservoirs to operate.

1) Cold reservoir (or the space to be cooled)
2) Reservoir for the temperature of obtainable heat source
3) Surroundings (It is the reservoir in which heat is rejected by using condenser)

The energy required at the generator of the absorption refrigeration system to regenerate the refrigerant is larger as compare to VCR system therefore, COP of VCR system is larger [31].

1.9.1.1 Types of absorption refrigeration systems

There are two types of absorption refrigeration systems which are given below [32].

a. Single Effect

i. It has only one generator to generate the refrigerant.
ii. Low temperature hot water is required 80 to 150℃.
iii. The COP is 0.6 to 0.7
iv. Low cost as compared with double effect

b. Double Effect

i. It has two generators to regenerate the refrigerant at the high temperature generator and low temperature generator.
ii. Water temperature of 155 to 205 ℃ is required.
iii. COP is 0.9 to 1.2
iv. The efficiency of double effect refrigeration system is greater as compared with single effect refrigeration system but meanwhile, it is more expensive [32]

1.9.2 Refrigerant-absorber Pairs

There are different types of refrigerant–absorbent pairs which can be utilized for solar absorption refrigeration systems. The most familiar pairs that provide acceptable thermodynamic performance and are environment friendly, are given below [33].

1) Lithium bromide (LiBr) – Water (H_2O)
2) It is more appropriate for solar applications because temperature required in the generator is about (90–120 ℃)
3) Ammonia (NH_3) – Water (H_2O)

This NH_3–H_2O pair is not appropriate for solar absorption refrigeration applications, as temperature required in the generator is larger (125–170℃) [33].

1.9.3 Single effect solar absorption cooling system

Working principle of single effect solar absorption cooling (SAC) system is given in Figure 1.5. For solar air-conditioning, collectors are used to harvest energy from sun. The heated working fluid obtained from solar thermal collector is kept in the storage tank. The useful energy gain of working fluid is stored in storage tank. This energy is used when the sun is not shining and there is a mismatch between cooling load and energy gain. This working fluid then goes into the generator of absorption chiller. It is utilized to evaporate water vapor from weak solution of LiBr and H_2O. Then, water vapor is passed through the condenser where heat is rejected to the cooling medium from vapor and converted into liquid. The liquid water then is passed through an evaporator and low pressure evaporation takes place by providing cooling effect in the required region.

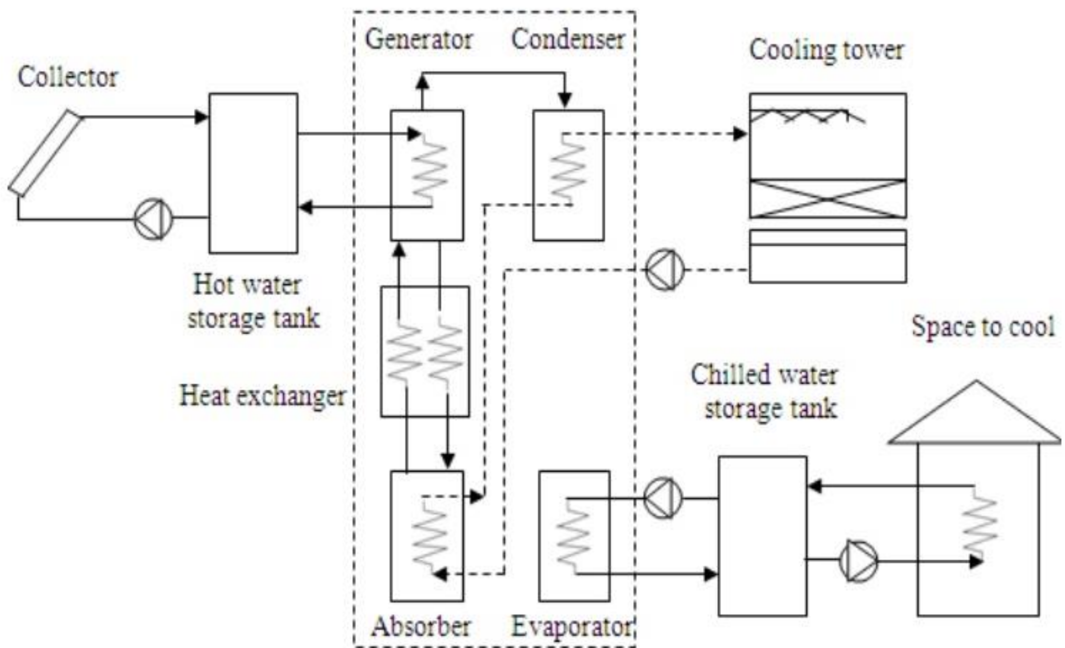

Figure 1-5: Working principle of solar absorption cooling system [34]

The strong solution returns from the generator to the absorber after vaporizing the water in the generator. While coming to the absorber, it passes through the heat exchanger where it pre-heats the weak solution of lithium bromide (LiBr) and water (H_2O), thus increasing the efficiency of absorption chiller. Water is evaporated in the evaporator by absorbing heat from the space being conditioned and is absorbed by strong solution o lithium bromide (LiBr) in the absorber. An auxiliary heater is applied, for continuous supply of hot water into the generator. It works when insufficient solar energy is available to heat water to the required temperature [34].

1.9.4 Process Description

The major steps involved in the absorption cooling process are presented in pressure verses temperature chart as shown in the Figure 1.6 [34].

1) Line 1-7 shows pumping of the weak solution from the absorber to the generator by passing through the heat exchanger. Properties of weak solution at the outlet of heat exchanger are represented by point 7. Whereas concentration remains same for the solution.
2) The heating of the weak solution by hot water in the generator is indicated form route 7-2. It is sensible heating, because it rises the temperature of solution. Line 2- 3 shows the latent heating causing boiling of water from the weak solution at the constant pressure of the condenser Pc. During this practice, the weak solution changes into a strong solution by evaporating water vapor.
3) Path 3-8 indicates that strong solution moves from the generator to the absorber by passing through the heat exchanger. Meanwhile the energy of strong solution is utilized to preheat the weak solution coming from the absorber to the generator. The concentration of the strong solution of lithium bromide (LiBr) remains constant in this process.
4) Water is evaporated in the evaporator by absorbing heat from the conditioned space. The water vapors are then absorbed in the strong solution of LiBr in the absorber and this process is represented by line 8-4-1.
5) The –heat of condenser is rejected in the cooling tower and this process is shown by line 2-5. During this process the pressure of the condenser remains constant (Pc).
6) The water is condensed in the cooling tower by rejecting its heat to the cooling medium usually water. This condensed water then flows towards the evaporator. This process is indicated by line 5-6.
7) Low pressure evaporation takes place in the evaporator by utilizing heat present in the space to be cool. It is represented by line form 6-1, and water vapor is absorbed by the absorber to complete the whole cycle.

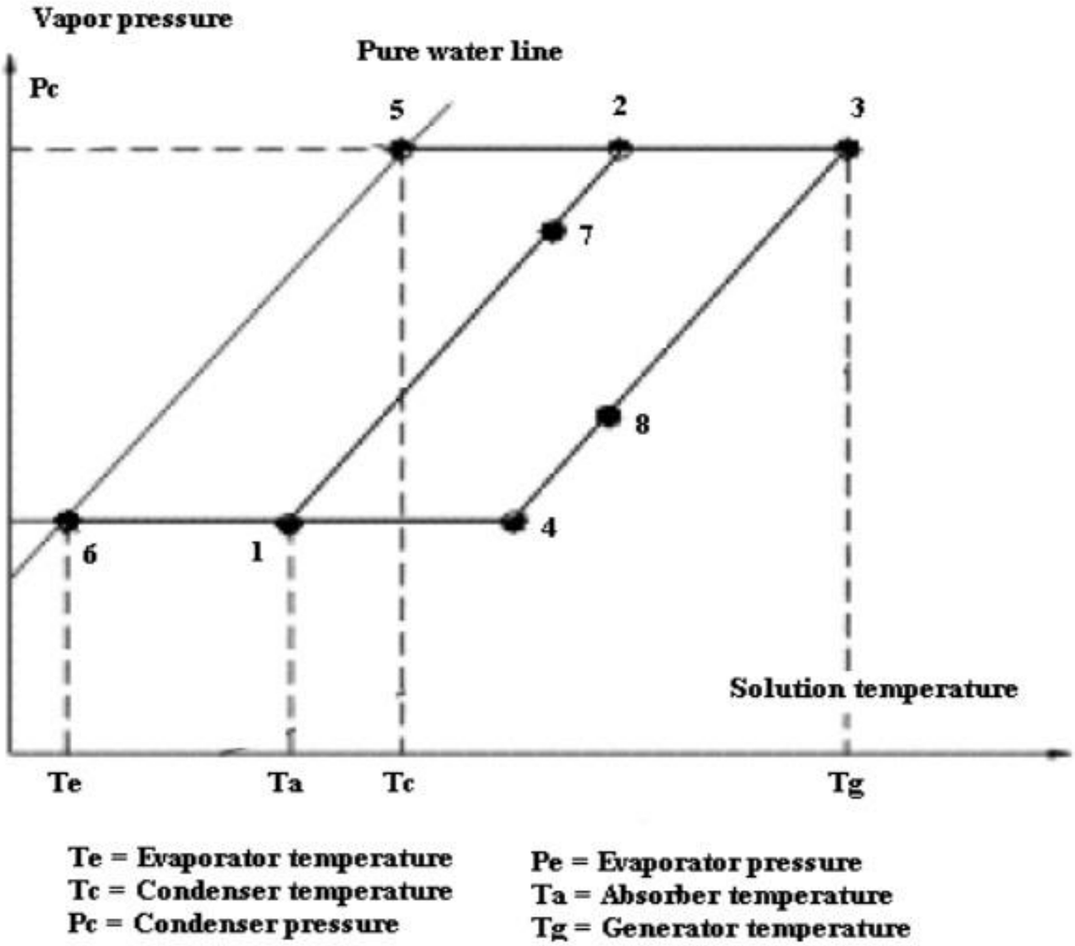

Figure 1-6: Process diagram of single effect absorption air-conditioning system [36]

1.10 Adsorption System

Adsorption process has two types

a) Physical adsorption
b) Chemical adsorption

1.10.1 Physical Adsorption

In physical adsorption process different types of adsorbents can be used. More likely utilized adsorbents are zeolite, alumina, activated carbon and silica gel. These adsorbents are highly spongy and have surface-volume ratio in the range of hundreds. They have ability to catch and hold selective refrigerants. These refrigerants can easily regenerated by heating them, after they became saturated [22]. If single vessel contains adsorbent and refrigerant pair the adsorbent maintains the pressure in that vessel by adsorbing vapor of refrigerant. For continuous operation multiple beds of adsorbent are required [35]. By using solar adsorption system 4-7 kg of ice can be provided daily by using unit square meters of collector area [36]. Silica gel-water pair solar adsorption systems have been used to provide space cooling [37,38]. In comparison with absorption chiller these systems have much lower cooling power density. Moreover, these systems are too massive and costly [39].

1.10.2 Chemical Adsorption

A strong chemical bond is formed between adsorbent and adsorbate in chemical adsorption. The regeneration of adsorbent is difficult and requires excessive energy as compare to physical process. Adsorbent which is commonly used for solar cooling application is calcium chloride ($CaCl_2$). This adsorbent adsorbs ammonia (NH3) and water (H2O) to produce $CaCl_2$.8NH3 and $CaCl_2$.6H2O respectively as products [35].

1.11 Desiccant Cooling

Desiccant cooling is also known as sorption cooling as sorbent is utilized in air dehumidification. Desiccants are present both in liquid and solid states. Generally all sorbents which have ability to absorb water can be used as desiccants. Such as: activated alumina, silica gel, lithium bromide (LiBr) and lithium chloride (LiCl). In desiccant cooling, just like absorption system, the liquid desiccant circulates in between absorber and generator. The central difference is that in liquid desiccant equilibrium temperature is assessed by the partial pressure of water in the humid air in which the solution is exposed. While in absorption system it depends upon total pressure. The working principle of desiccant cooling is presented by Kim [22].

1.12 Ejector Refrigeration

This technology is utilized to provide air conditioning in large buildings and trains [40]. Construction of ejector cooling systems is not complex. Meanwhile, low COP value makes these systems less competitive then other thermal-driven technologies [41].

1.13 Introduction to TRNSYS

TRNSYS (TRaNsient SYstem Simulation) is a comprehensive and extensible simulation tool for the dynamic simulation of systems, which also includes multi-zone buildings. New energy concepts can be validated in TRNSYS. From simple domestic use of hot water systems to the complete design and simulation of zero-energy buildings, occupants behavior and their schedule, control strategies, equipment in the building, renewable energy systems like, solar thermal, photovoltaic, wind turbines, Hydrogen systems, etc.

Third party developers and users can easily add custom made components model due to the dynamic link library (DLL) based architecture in TRNSYS. All common programming languages like PASCAL, C++, C, FORTRAN etc. can be used for adding new components.

Typically, components are inter-connected graphically in Simulation Studio to setup a TRNSYS project. Every component has its own mathematical model.

TRNSYS components are often denoted as Types (e.g. Type1 is the solar collector). The Multi-zone building model is known as Type56. These types are divided into groups; each one has number of types that represent a specific application. These groups include the following application:

i. Solar systems (solar thermal and PV)
ii. Zero energy buildings and HVAC systems with advanced design features.
iii. Renewable energy systems
iv. Fuel cells, cogeneration

TRNSYS consists of a number of suite of programs. In this thesis, only Simulation Studio has been used. TRNSYS is quite a mature simulation tool which is based on well-established analytical and empirical correlations. It is being used for simulating systems to predict their long term performance which is otherwise very difficult to guess in the planning phase. The purpose of current study is to propose a solar based cooling system for an office building and TRNSYS is used to model and simulate various design concepts which were not previously compared and analyzed.

1.14 Summary

World energy consumption is rising drastically. It is predicted by researchers that the rising trend will continue. In developed countries buildings utilize about 20-40% of the entire final energy consumption. About 30-50% of the total energy need in commercial buildings is required for the heating, ventilation and air conditioning (HVAC). To meet the cooling demand, vapor-compression systems (VCS) are most commonly used all over the world. These systems are also known as conventional systems. The emissions of refrigerants from conventional systems are contributing to the depletion of ozone layer. Moreover, these systems are very energy intensive. In account of these problems, solar absorption cooling systems are of great interest. As the use of energy from sun for cooling is an attractive alternative because resource availability is apparently coincide with cooling demand. Therefore, these systems can be used for reduction in electricity peak load demand and are appropriate to fulfill cooling load in buildings. Pakistan is the country which is currently facing shortfall of electricity especially during summer season. The geographical location of Pakistan is ideal for using these systems because summers season is very long and country lies on sunny belt. On average most parts of the country have 300 days of sunshine per year with 1900-2200 kWh/m^2 of global irradiance. It shows that Pakistan is one of the richest countries in term of solar potential. These solar radiations can be used effectively for solar thermal applications such as solar absorption cooling systems.

CHAPTER 2

LITERATURE REVIEW AND RESEARCH OBJECTIVES

2.1 Literature Review

About 81 large scale solar cooling systems (SCS) were installed till 2007. It included the systems which were not in operation. 73 solar cooling systems are located in Europe, 7 are working in Asia, most of them are in China, and 1 is located in America. 60% of these systems provide cooling to office buildings, 10% are dedicated to factories, 15% for cooling laboratories, 6% provide comfort in hotels and the remaining percentage is used for buildings with diverse final use like hospitals, sports centers, canteen, etc. The total cooling capacity of the chillers driven by solar energy amounts to 9 MW. 31% of total capacity is located in Spain, Germany contributes about 18% and Greece provides 12% [42].

Numerous studies have been reported including absorption refrigeration systems using TRNSYS. Tao He et. al. [43] simulated and a varied a solar absorption cooling system in Beijing. China using TRNSYS. Cooling load of the building is 50 TR and total aperture area of evacuated tube collectors is 358 m^2. Solar collectors are tilted at 10° facing south. Rated capacity of absorption chiller is 175.8 kW and rated COP is 0.7. A biomass boiler is used as backup auxiliary device. Size of thermal storage tank and cooling storage tank is optimized and the optimum sizes are 15 m^3 and 8 m^3 respectively. Average annual efficiency of solar collectors is found to be 37.6 % and solar fractions for summer and winter are 0.76 and 0.38. COP of the system for entire summer season is 0.32. Primary energy consumption can be decreased by 66 %, CO_2 emission by 35.4 ton and SO_2 by 114.8 kg per year. Optimization of the SAC system with internal energy storage was conducted using TRNSYS by Sanjuan and Soutullo. The system stored solar energy in the crystallized salts during day time and discharged energy to provide heating or cooling to the building. It was investigated that by applying controlled strategies the value of solar fraction could enhance from 50% to 90% [12]. Tsoutsos et al [44] simulated a HVAC system of a hospital in Crete using TRNSYS. The optimum configuration was selected on the basis of solar fraction. Different parameters like collector area, size of storage tank, collector slope, back up heater, cooling tower and nominal capacity of absorption chiller were optimized. Four scenarios were discussed and it was concluded that optimum area is 500 m^2 and number of collectors was 179. The solar fraction for cooling season is 74.23% and solar fraction for heating season was 70.78%.

Gomri et al. [45] examined the overall performance of the solar air-conditioning system for a building situated in Algeria. It was observed that COP varied by changing the temperature of condenser and generator at constant evaporator temperature and maximum COP of 0.82 was achieved. Florides et al. [32] presented a model for solar absorption cooling for the weather of Nicosia, Cyprus. He used TRNSYS for simulation of the cooling system. Optimization of type, area and tilt angle of solar collector, volume of thermal storage, thermostat setting of the auxiliary boiler is carried out. The optimum slope of the parabolic trough collectors is 30° from horizontal and optimum size of hot water thermal storage is 600 liter.

A complete solar based cooling system which includes solar thermal collectors and an absorption chiller unit for a lifetime of 20 years, will have a total life cycle cost of the order of C£ 13,380. Only absorption chiller unit costs about C£ 4800. Economic analysis was also carried out for this system. It was concluded that the system will only be competitive economically as compared with conventional cooling systems if its capital cost is less than C£ 2000. However, the TEWI of the system is 1.2 times smaller as compared with vapor compression systems [33].

Mateus and Oliveria [46] analyzed the performance of solar absorption chillers for different building types including residential, hotel and office. By apply diverse local energy costs for gas and electricity economic analysis was performed and reduction in CO2 emissions was calculated (367-588 tons per 20 years for hotel). Assilzadeh et. al. [47] designed a solar cooling system for the climate of Malaysia. They used TRNSYS program to model and simulate the system. They designed the system using lithium bromide (LiBr) absorption chiller and evacuated tube collectors (ETC). The results demonstrate that the system is in phase with the weather i.e. the cooling load is high when global radiation is high. The cooling load is 3.5 kW. The optimum configuration comprises of 0.8m^3, 35m^2 evacuated tube collectors tilted at 20°. Umberto gave different ways for technical installations of solar absorption systems. He also analyzed the techno–economical viability of SAC systems, made for two dissimilar applications: one was for refrigeration in industry and other was to provide air conditioning in a hotel [48].

Barhane H. [49] offered a decision-making tool working on mathematical equations for the modeling of SAC system. The model was developed as bi-criteria mixed-integer nonlinear programming (MINLP) optimization problem that evaluates the reduction of the total investment cost of the SAC system and the related environmental effects calculated for the whole life time of the system by using weather data of Barcelona (Spain). Lecuona et al. gave an explicit equation for the hot water temperature optimization which is delivered to the generator of SAC system. The study is based on the postulate that as the temperature of the hot water rises, the COP value of the absorption chiller increases, meanwhile the collector efficiency decreases. They determined that, there is an optimized temperature that gives the greatest combined efficiency of SAC system [50].

Molero et al. [51] compared different configurations for solar based absorption cooling system for a residential building in Spain. Effects of hot storage and cold storage were analyzed. Collector area, efficiency curve of solar collector, COP of absorption chiller, size of thermal storage, COP of absorption chiller and temperature set points of chiller were optimized. Typical capacity of thermal storage is 40 L/m2. The main purpose of this research work is to determine the advantages of hot and cold storages. It is clear from the results that the system performs better when cold storage is employed in the system, especially when the storage size is large and collector area is small.

Mazloumi simulated a solar absorption cooling system. The system employs lithium bromide single effect absorption chiller. The system is located in Ahwaz (Iran). Parabolic trough collectors were employed in this system. The system fulfills the cooling demand of a typical house having peak cooling load of 17.5 kW (5 tons of refrigeration). The peak cooling load occurs in July. The required collector area was 57.6 m^2. Water is used as working fluid which is fed to the collectors directly. The results show that mass flow rate has significant effect on the optimal efficiency of thermal storage [52].

2.2 Research Objectives

Overall the broad objective of the current study is to model the absorption cooling system for an office building located in Islamabad (33.7167° N, 73.0667° E) powered by the solar thermal energy.

Specifically the study will focus on the following objectives:

i. Modeling the SAC system by using advance renewable energy software Transient System Simulation tool (TRNSYS)
ii. Thermal performance analysis of the SAC system
iii. Optimization of system components by using flat plate collector (FPC) and evacuated tube collector (ETC) for weather data of Islamabad.

2.3 Scope of the Work

In this study, a solar based absorption cooling system is modelled and simulated. The objective of the simulations is to choose the suitable configuration for the systems. Two configuration C-1 and C-2 are simulated and the results are compared on the basis of solar fraction and fractional primary energy saving (FPES). The optimum tilt angle for ETC and FPC is determined on the basis of solar fraction for the whole summer season. An arbitrary cooling demand of the building is used in simulations. The load is varied throughout the season to make it more realistic.

Thermal performance of SAC system is observed for the actual building by using TRNSYS 17 software. It is a comprehensive and extensible simulation environment used for renewable energy projects [47]. For the modeling of the system suitable components are selected and placed in the deck file. Then, the optimization of the SAC system components is performed. For the optimization two types of collectors, FPC and ETC are used for the weather data of Islamabad. It is investigated that evacuated tube collectors when used with C-2 combination are more viable to use in combination with SAC systems. Fractional primary energy saving gives us the idea how much energy can be saved if the system were operating on vapor compression cycle.

Experimental setup of the system is not developed because of the high initial cost of the equipment and installation of the system.

CHAPTER 3

Research Methodology

This chapter provides the methodology which is applied to carry out this research. Configurations of the system and all components are discussed in this chapter.

3.1 Approach for study

The steps which are followed for the modelling and simulation of this system are outlined as follows:

1. Climate data of the location is collected from meteorological department for generating typical meteorological year (TMY) file. The file encloses the hourly variation data of ambient temperature, relative humidity, beam solar radiation, diffused solar radiation, total solar radiation and wind speed.
2. The cooling load of the building is specified by a component type682. This component simply imposed a user defined constant load (positive load = cooling, negative load = heating) on a flow stream and calculates the subsequent outlet fluid conditions. Boiling and freezing effects are ignored in this component. Often in simulating an HVAC system, cooling and heating loads of the building are known through calculations or through another simulation program and yet there is a need to simulate the effect of these loads upon the system. This component allows the user to define a load upon a fluid stream through a device. It acts as an interaction between such pre-calculated loads and HVAC system in TRNSYS.
3. Suitable components are selected and connected to each other. Hot water fired single effect LiBr-H_2O chiller is used to provide chilled water to the load. Type107 serves this purpose. Other main components are
   i. Solar thermal collector
   ii. Stratified hot water storage tank
   iii. Boiler as auxiliary backup
   iv. Cooling load profile component
   v. Pumps and controllers
   vi. Printer and integrator

4. All the components, weather data and absorption chiller are placed and interlinked in TRNSYS simulation studio
5. Simulations are run and a parametric study is performed to get the results in the form of data then plots are drawn.
6. Comparison of two configurations is performed with two solar thermal collectors commercially offered in the market; flat plate collectors and evacuated tube collectors.
7. System is optimized for optimum tilt angle of the solar thermal collectors and selection of optimum thermal storage.

3.2 System Description

There are two configurations compared and analyzed. Both are discussed below.

3.2.1 Configuration -1

In configuration-1 (C-1), water enters from solar collectors into the thermal storage. Then water flows in the auxiliary backup boiler. If the temperature of water coming out of thermal storage is less than 110℃, then auxiliary back up boiler turns on and raises the water temperature to desired level. A thermostat continuously monitors the temperature of water coming out of thermal storage. Waters enters the absorption chiller at a constant temperature of 110℃. After delivering heat to the absorption chiller, water goes back to the thermal storage tank and then to the solar collector and the cycle continues. There is a controller which monitors the temperature of water into the collectors and out of the collectors. If the water temperature coming out of collectors is greater than the inlet temperature, the pump turns on otherwise it turns off.

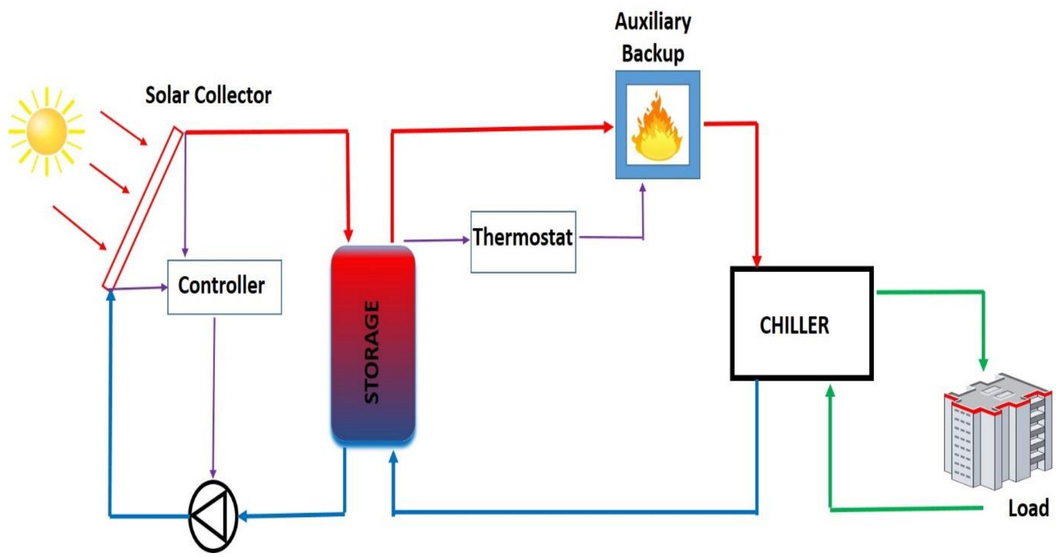

Figure 3-1: Configuration-1 for absorption cooling system (C-1)

3.2.2 Configuration-2

In configuration-2 (C-2), water enters from solar collectors to thermal storage tank. If the temperature of hot water leaving thermal storage tank is less than 110℃, water does not flow into the auxiliary backup boiler, it continues to circle in the solar loop. Water coming out of absorption chiller is diverted to the mixer and is heated in auxiliary boiler to the desired level. A tee is used for this purpose. In this configuration, the system is running on either complete solar energy or complete auxiliary energy.

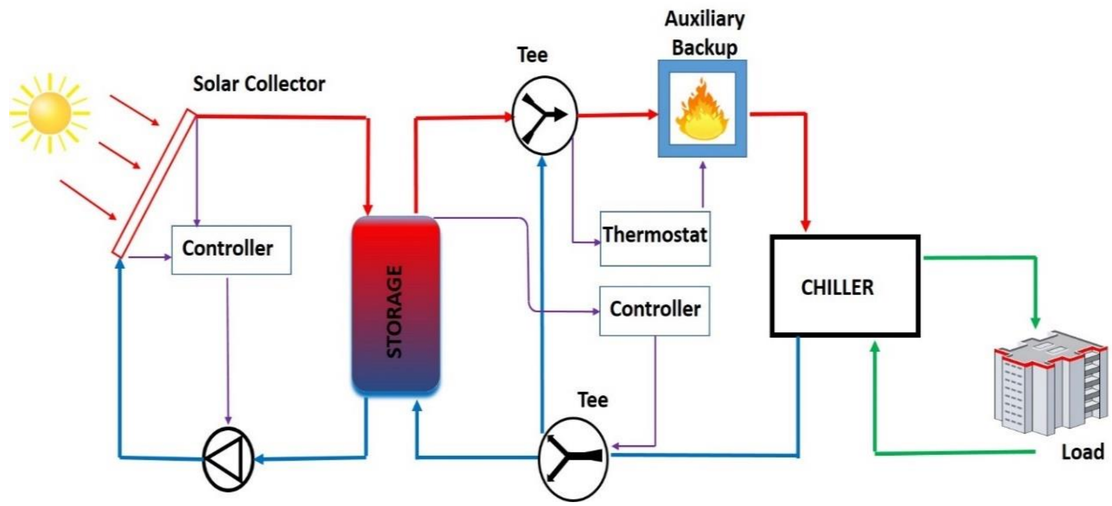

Figure 3-2: Configuration-2 for absorption cooling system (C-2)

3.3 TRNSYS Model

Both configurations (C-1 and C-2) are modelled in TRNSYS by connecting various system components, as shown in figure 3.3 and figure 3.4.

In this model it is supposed that the working fluid i.e. water does not boil in the system. The only losses considered are in the solar thermal collectors and thermal storage. The mass flow rate for solar loop is obtained by multiplying the solar collector area and tested flow rate of collector. In the hot water loop of chiller, the temperature difference between entering the chiller and leaving the chiller is restricted to 10℃. This gives us hot water flow rate for chiller loop. The solar fraction is taken to be the part of generator load that is covered by useful energy gain from the solar collectors. Power consumed in other equipment such as circulating pumps and controllers is not considered. To minimize the thermal energy losses from the storage tank, it is often kept outdoors.

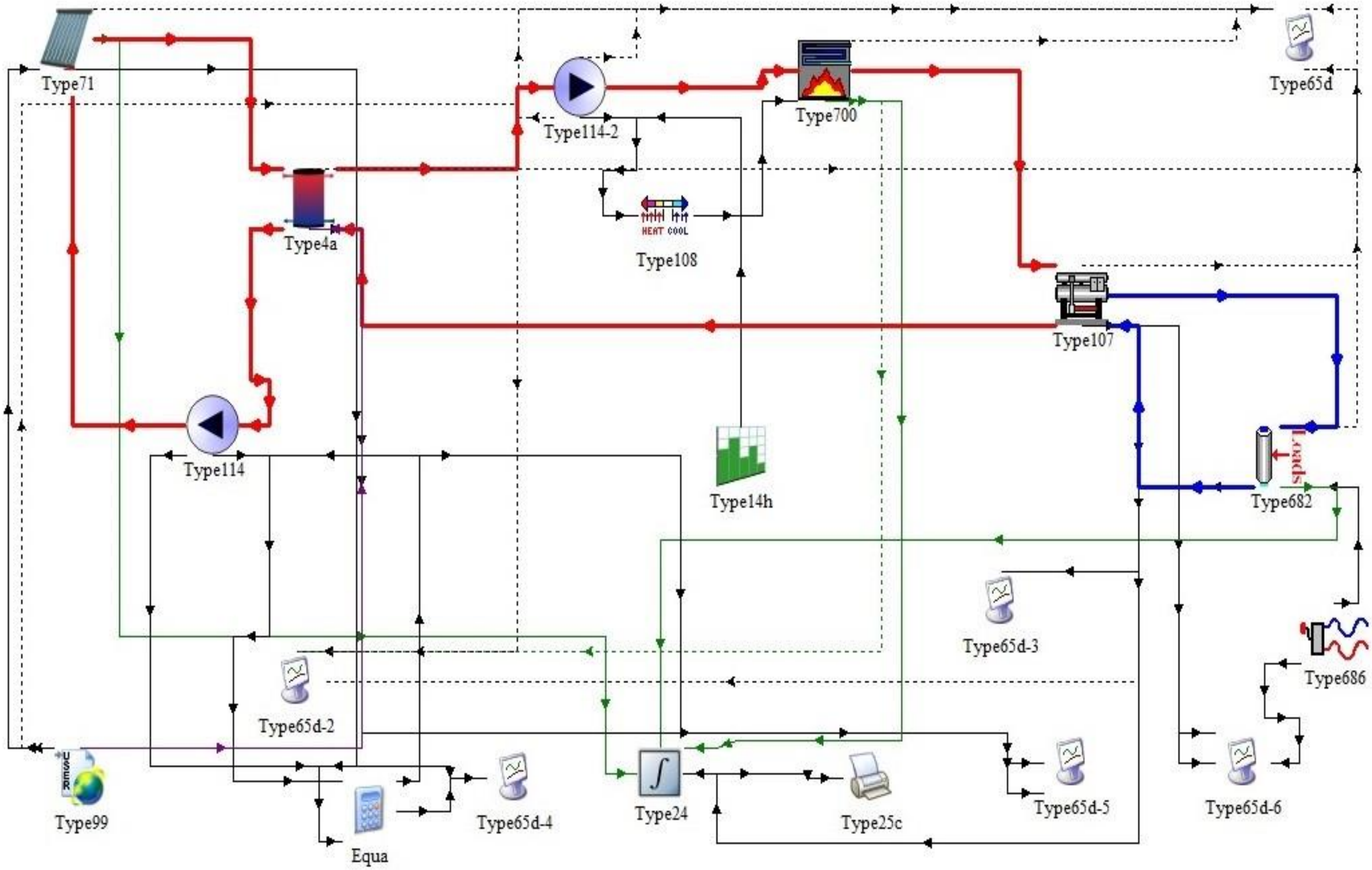

Figure 3-3: Pictorial view of the TRNSYS model for C-1

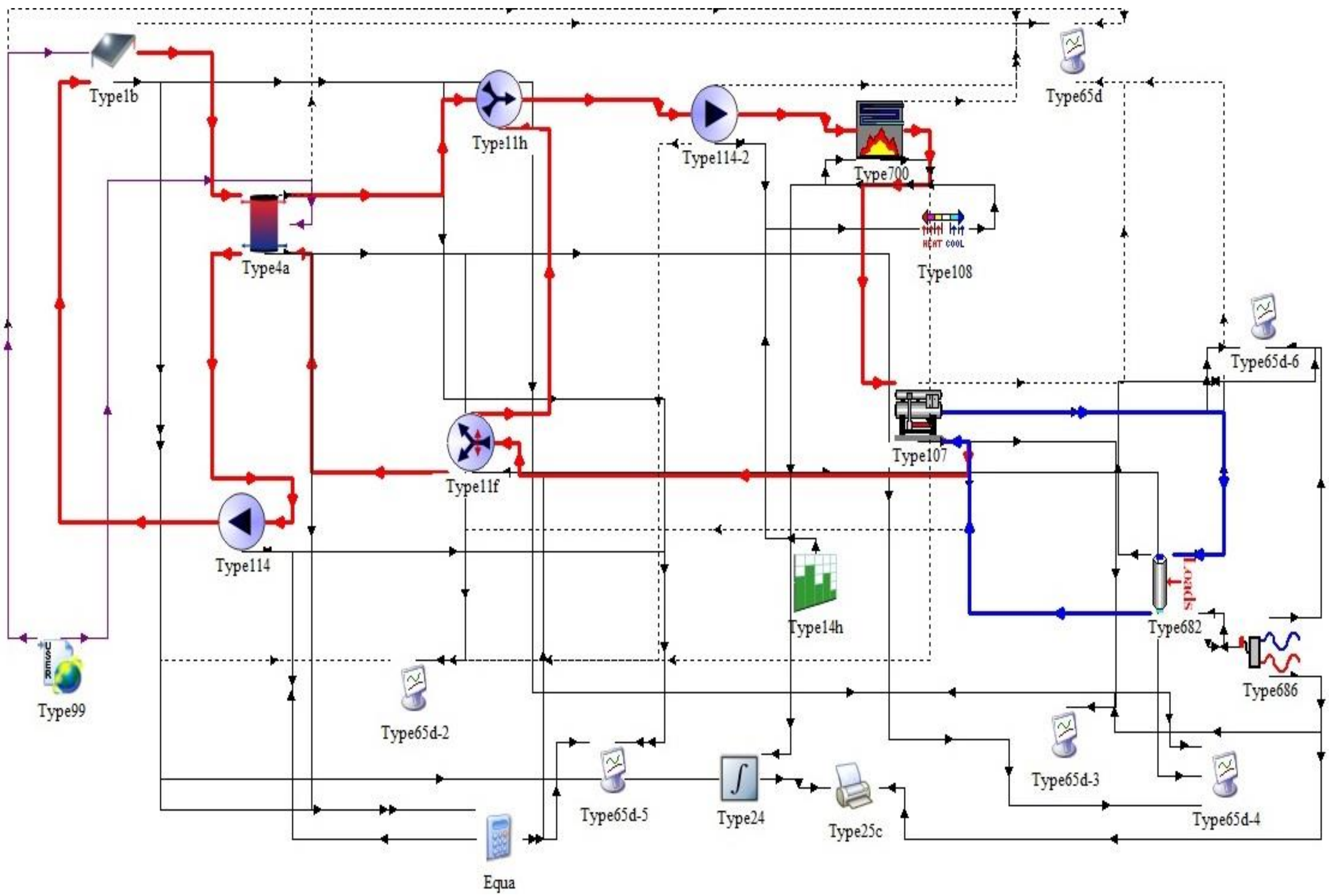

Figure 3-4: Pictorial view of the TRNSYS model for C-2

3.4 System Components

3.4.1 Solar collectors

Type1b and type71 are used as flat plate collectors and evacuated tube collector.

The efficiency of the solar collectors is given by

$$\eta = a_o - a_1 \frac{(\Delta T)}{I_T} - a_2 \frac{(\Delta T)^2}{I_T} \qquad (3.1)$$

a_o is optical efficiency and a_1 and a_2 are the heat loss coefficients, ΔT $(T_{in} - T_{amb})$ is the difference between water inlet temperature in the solar thermal collectors and the ambient temperature. I_T is incident solar radiation.

The values of these coefficients are provided by the manufacturer of solar thermal collectors (FPC and ETC) and are listed in tables 3.1 and 3.2.

Table 3-1: Parameters of Flat plate collectors

Parameter	Description	Value
Collector type	Flat plate collector	
C_p	Specific heat of fluid (kJ/kg. K)	4.19
η	Efficiency mode	$(T_{in} - T_{amb})$
a_o	Intercept efficiency	0.80
a_1	Efficiency slope (kJ/hr. $m^2.K$)	13
a_2	Efficiency curvature (kJ/hr. $m^2.K^2$)	0.05
Optical mode		Incidence modifiers

Table 3-2: Parameters of evacuated tube collectors

Parameter	Description	Value
Collector type	Evacuated tube collector	
C_p	Specific heat of fluid (kJ/kg. K)	4.19
η	Efficiency mode	$(T_{in} - T_{amb})$
a_o	Intercept efficiency	0.802
a_1	Efficiency slope (kJ/hr. $m^2.K$)	5.4
a_2	Efficiency curvature (kJ/hr. $m^2.K^2$)	0.00576
Optical mode		Incidence modifiers

3.4.2 Single Effect absorption chiller (Type 107)

The performance of the absorption chiller is measured in terms of its COP. The coefficient of performance of the absorption chiller is given by

$$COP = \frac{\dot{Q}_{chw}}{\dot{Q}_{hw}} \tag{3.2}$$

where $\dot{Q}_{chw}$ is the cold energy delivered by the absorption chiller and $\dot{Q}_{hw}$ is the hot water energy supplied to the generator of the absorption chiller.

The quantity of hot water energy required to yield desired cooling demand is given by

$$\dot{Q}_{hw} = \frac{Capacity_{Rated}}{COP_{Rated}} f_{DesignEnergyInput} \tag{3.3}$$

The mass flow rate of the hot water entering in the absorption chiller is calculated by the equation

$$\dot{Q}_{hw} = \dot{m}_{hw} Cp_{hw} (T_{hw,in} - T_{hw,out}) \tag{3.4}$$

$\dot{Q}_{hw}$ is the amount of heat exchanged at the generator of absorption chiller and Cp_{hw} is the specific heat of hot water. The temperature difference between inlet water temperature to the absorption chiller and exit water from the absorption chiller is restricted to 10℃ which is standard practice.

Fraction of design load at which machine is required to operate is calculated by

$$f_{DesignLoad} = \frac{\dot{Q}_{remove}}{Capacity_{Rated}} \tag{3.5}$$

$\dot{Q}_{remove}$ is the amount of energy that has to be removed by the absorption chiller i.e cooling load.

3.4.3 Auxiliary Backup Device (Type700)

If there is a flow through the boiler and the control signal to the boiler is ON then the model calculates the quantity of required energy necessary to elevate the temperature of inlet water to the set temperature using equation:

$$\dot{Q}_{need} = \dot{m}_{fluid} Cp_{fluid} (T_{set} - T_{in}) \tag{3.6}$$

The device will not calculate a negative value of $\dot{Q}_{need}$ if the inlet temperature is higher than the set temperature and control signal is ON. The required energy is limited by the capacity of the boiler.

Once the required energy is calculated, the amount of fuel consumed by the boiler can be calculated by equation:

$$\dot{Q}_{fuel} = \frac{\dot{Q}_{need}}{\eta_{boiler}} \tag{3.7}$$

3.4.4 Storage Tank (Type4a)

A multinode approach is used in this component in which the tank is divided into N segments or nodes or sections and each node is governed by energy balance. Writing the energy balances results in N differential equations which are then solved for the temperature of each segment as a function of time. The flow towards the generator of absorption chiller always leaves from the top node and return flow towards collectors always leaves from bottom node.

An energy balance written about ith tank node can be expressed as

$$M_i C_{pf} \frac{dT_i}{dt} = \alpha_i \dot{m}_h C_{pf} (T_h - T_i) + \beta_i \dot{m}_L C_{pf} (T_L - T_i) + UA_i (T_{env} - T_i) + \begin{cases} \gamma_i \, (T_{i-1} - T_i) C_{pf} & if \ g_i > 0 \\ \gamma_i (T_i - T_{i+1}) C_{pf} & if \ g_i < 0 \end{cases} \tag{3.8}$$

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3.4.5 Cooling Load (Type682 and Type686)

In most cases, the cooling load and heating load of the building are already known with the help of a simulation program or by calculations, yet we have to investigate the effect of load on the system. This component imposes a user-defined load upon a liquid stream through a device. The convention of using this component is that the positive values indicate the cooling load and negative values impose a heating load. Freezing and boiling effects are ignored in this component. Mathematically, this model is very simple, the user provides the flow rate, specific heat, and temperature of liquid at a point in the system loop. The building loads are added to, or subtracted from that liquid, resulting in an outlet temperature just past the interaction point.

$$T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}\, C_p} \tag{3.9}$$

According to the sign convention, the outlet temperature will be greater than the inlet temperature in case of positive load and vice versa; a negative load results in an outlet temperature lower than the inlet temperature.

Type686 generates heating load and cooling load for a synthetic building on hourly basis. The user defines peak cooling and heating loads and modifying sine-wave functions which accounts for time of day variations, seasonal variations and weekday/weekend differences. Random noise can also be generated on both daily basis and hourly basis. It makes the building loads more realistic. The researchers use this component as their first excellent choice for dynamic simulations of residential, industrial and commercial buildings. This component eliminates the time intensive model which is required for a real building. It is quick and time saving.

The hourly heating load and cooling load for a building are in this method as

$$Load = DesignLoad * X_{day} * X_{hour} * X_{noise,hour} * X_{noise,day} \tag{3.10}$$

The model allows the heating and cooling loads to vary throughout the heating and cooling season by use of a modifying SIN wave function. The user must enter the start and stop times of the heating and cooling seasons and offset and multiplier parameters for the SIN function. The formulation of the function depends on whether the season crosses over the first day of the year or whether the season is completely contained within one year. For the case where the season crosses the annual boundary, two functions are needed; 1 for the time of the year from the start of the year to the end of the season and 1 for the time of the year from the start of the season to the end of the year. There are no heating loads during the non-heating season and no cooling loads during the non-cooling season.

Season contained within a year:

$$X_{day} = a_{season} + b_{season} \, sin\left(180\left(\frac{TIME-StartOfSeason}{EndOfSeason-StartOfSeason}\right)\right) \tag{3.11}$$

3.5 System Performance Factors

The performance factors for evaluation of system are defined as follows:

i. Solar Fraction

Solar fraction is given by [53]

$$SF_{thermal} = \frac{\int Q_{solar}}{\int Q_{solar} + \int Q_{Aux}} \tag{3.12}$$

where $SF_{thermal}$ is the solar fraction of solar thermal cooling system, Q_{solar} is the useful energy gain from solar thermal collectors and Q_{Aux} is heat provided by auxiliary backup.

In C-1, the water coming out of collectors is constantly supplying hot water to the absorption chiller but in C-2, the hot water only flows to the absorption chiller when its temperature is reached upto the required level. In that case solar fraction is only calculated when hot water flows towards the absorption chiller, otherwise solar fraction is zero.

ii. Collector efficiency

A measure of monthly or seasonal collector thermal performance is estimated by collector efficiency, which is defined as

$$\eta = \frac{\int \dot{Q}_u \, dt}{A_c \int G_T \, dt} \tag{3.13}$$

where Q_u is the useful energy gain, A_c is the aperture area of solar collectors and G_T is the global solar radiation.

iii. Primary energy savings

Based on general definition of primary energy saving ($f_{sav,shc}$) [54], $f_{sav,shc}$ is defined here with reference to an electrically operated compression chiller system as:

$$f_{sav,shc} = 1 - \left[\frac{\frac{\int Q_{boiler}}{\varepsilon_{heat}}}{\frac{\int Q_{cooling,ref}}{SPF_{ref.}\varepsilon_{elec}}} \right] \tag{3.14}$$

where Q_{boiler} represents the auxiliary heat provided by a boiler, ε_{heat} is the boiler efficiency, $Q_{cooling,\, ref}$ is the energy for cold delivered by a conventional vapor compression refrigeration system and *SPF* is the typical compression chiller efficiency.

The term in brackets represents the ratio of total primary energy consumed by auxiliary device in a solar based system to the total primary energy consumption of a reference or conventional system to meet the same cooling load.

Typical values of conversion factors in Eq. (4) are assumed as follows:

$\varepsilon_{elec} = 0.4$ (kWh of electricity per kWh of fossil primary energy)

$\varepsilon_{heat} = 0.78$ (kWh of useful heat work per kWh of fossil primary energy)

$SPF_{ref} = 2.8$ (typical COP of a compression chiller)

3.6 Collector Tilt

The useful energy gain by solar thermal collectors depends strongly on the tilt angle of collectors. As declination angle changes every single day, it effects the optimum tilt for surfaces. Figure 3.5 demonstrates the geometric analysis to select the optimum tilt of surface for both hemispheres for the entire year. The solar thermal collectors with this kind of slopes, sun's rays are normal to the surface at solar noon so the surface incidence angle θ is zero. The slopes for achieving maximum solar thermal energy gain for Northern Hemisphere latitudes when ($\emptyset - \delta$) > 0 are as follows.

$$\beta(\gamma_s = 0°) = \begin{cases} \emptyset - \delta & \text{for } \delta > 0 \\ \emptyset & \text{for } \delta = 0 \\ \emptyset + |\delta| & \text{for } \delta < 0 \end{cases} \tag{3.15}$$

The declination is given by

$$\delta = 23.45 \sin\left(\frac{360}{365}\,(284 + n)\right) \tag{3.16}$$

where n is number of the day of year.

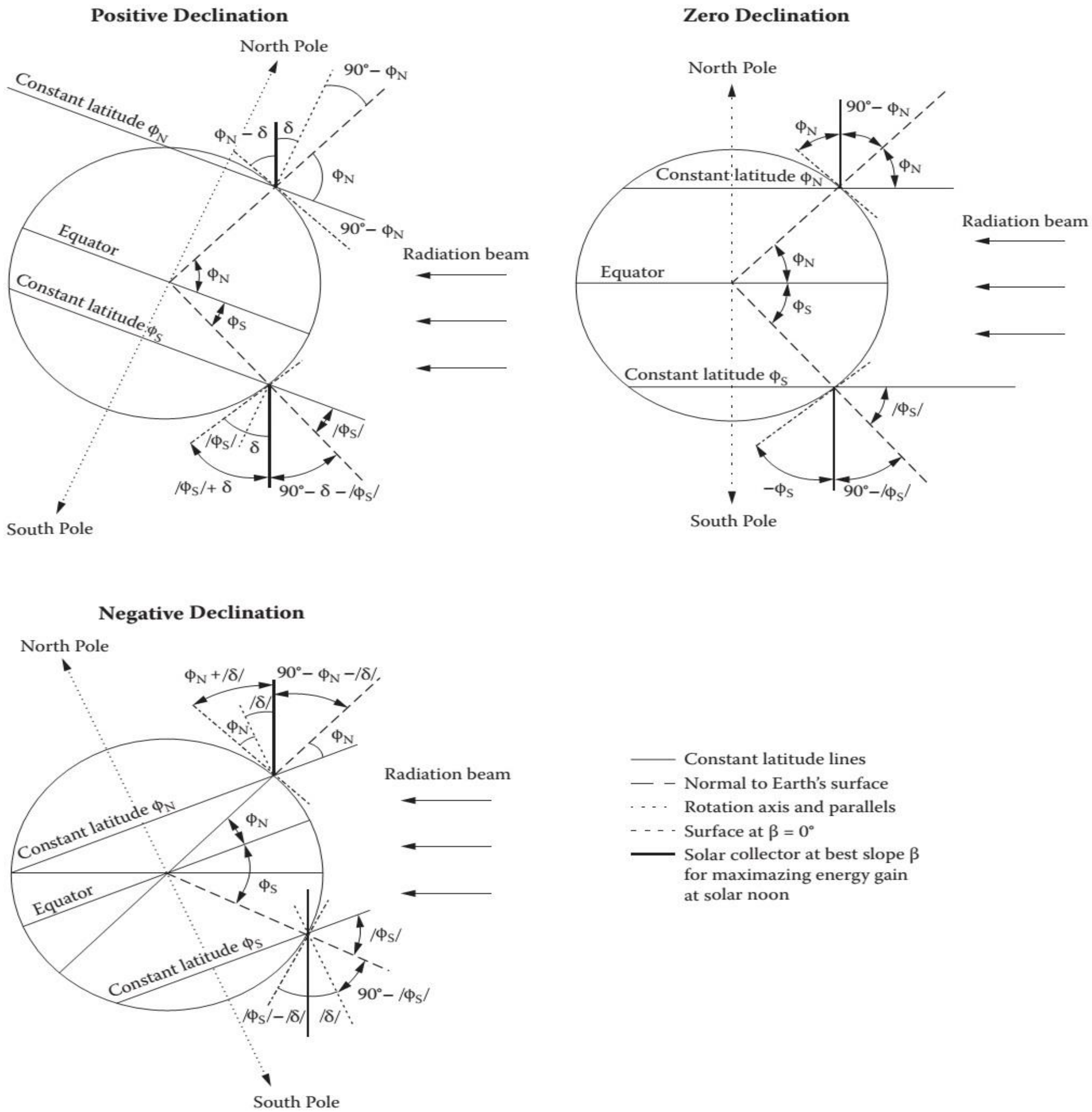

Figure 3-5: Geometric relationship for solar collectors perpendicular to the solar radiation beam at solar noon when δ=+23.45°, 0°,-23.45°

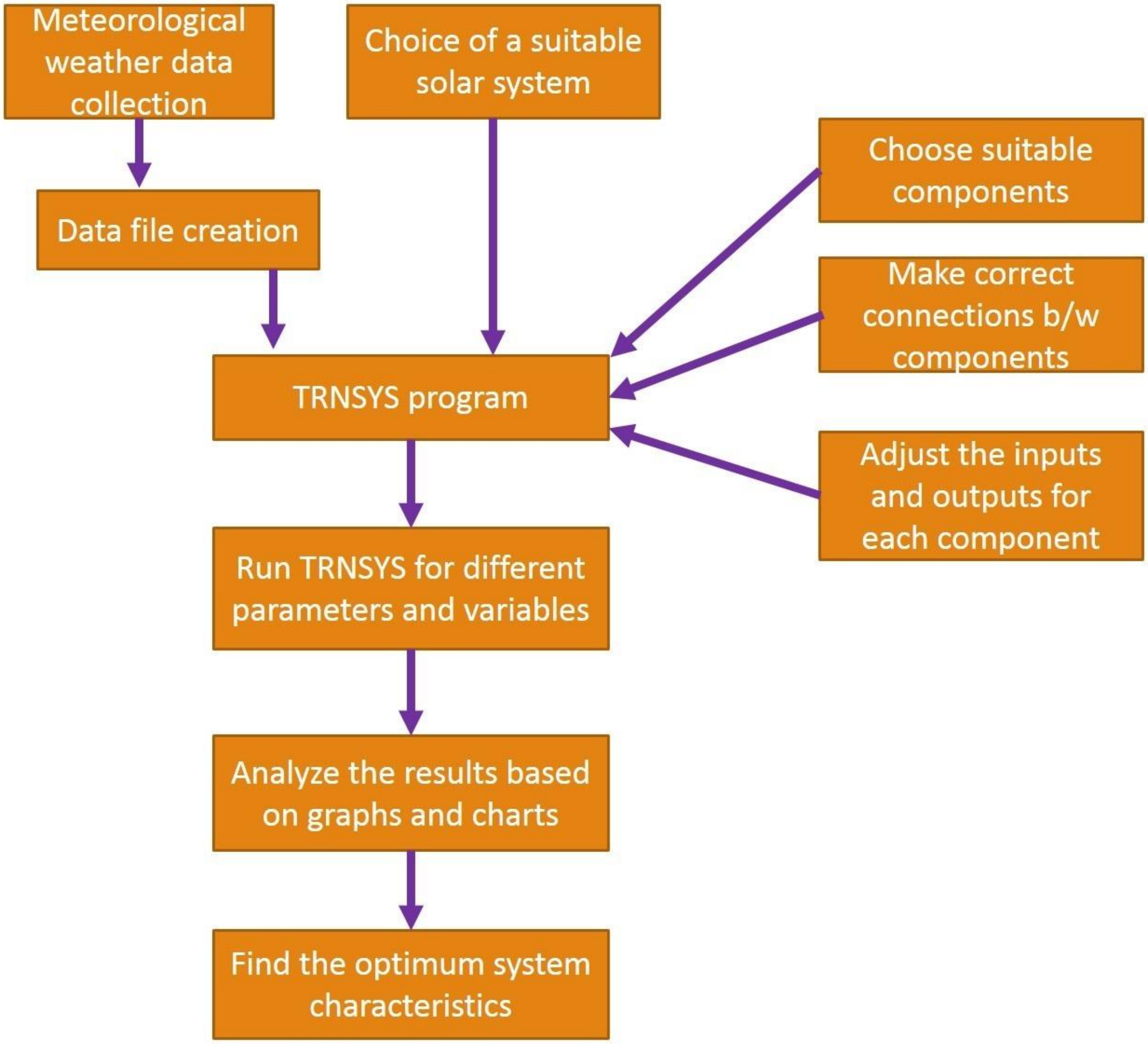

Figure 3-6: Flow chart for methodology

Chapter 4

Results and Discussion

A number of simulations are performed in order to optimize various factors affecting the performance of the system for the entire summer season i.e. from May to September. Islamabad has a humid subtropical climate with five seasons: Winter (Nov-Feb), spring (March-April), summer (May-June), rainy monsoon (July-Aug) and autumn (Sep-Oct). The hottest month is June with average temperature of 38℃. The wettest month is July with heavy rainfall where average relative humidity remains above 65%. In the following sections, simulation results are presented and discussed in detail.

4.1 Available solar radiation and ambient temperature

Figure 4.1 displays the total global hourly solar radiation and diffuse radiation of Islamabad for all months. The maximum solar radiation reaches upto 1000 W/m^2. The variation of ambient temperature for the whole year is presented in figure 4.2. The maximum temperatures can be seen in the months of May and June reaching upto 42℃.

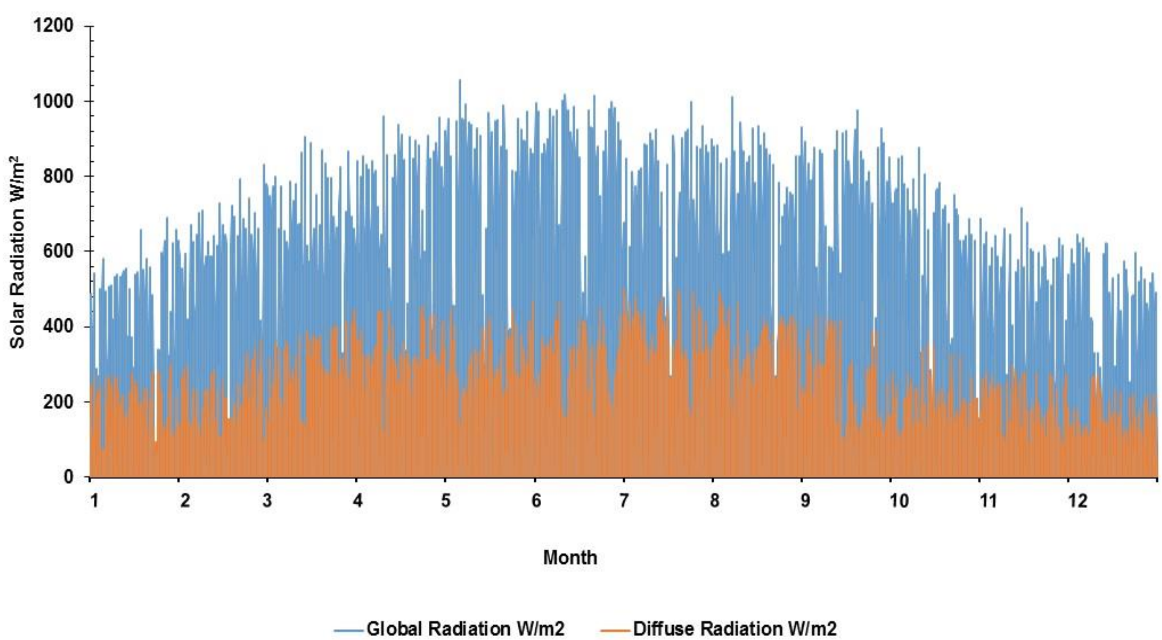

Figure 4-1: Hourly global radiation for Islamabad

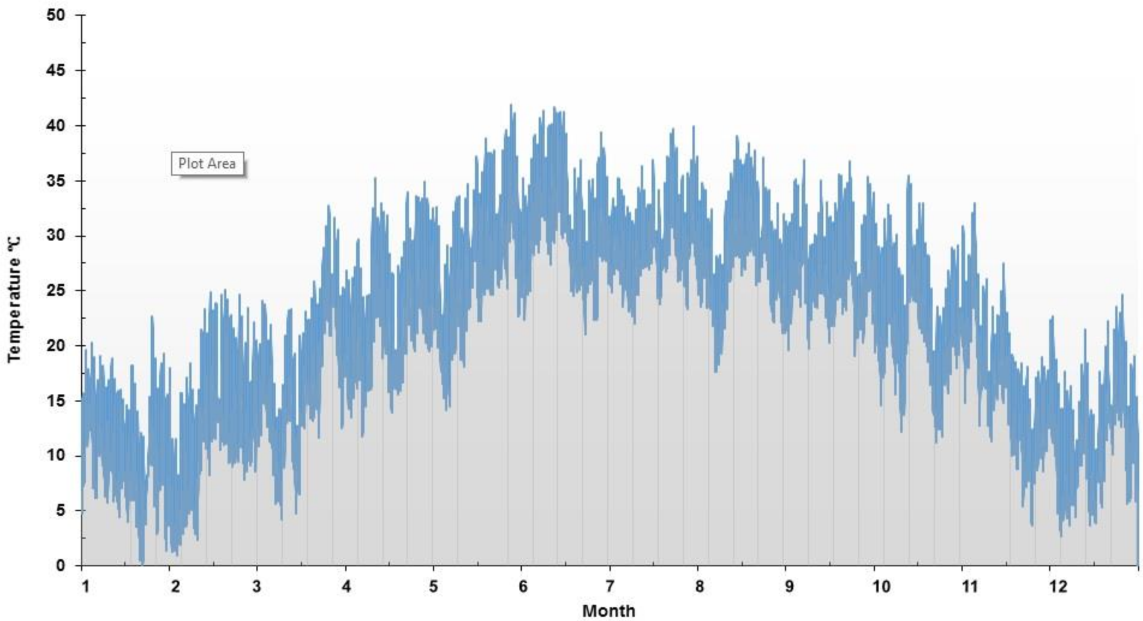

Figure 4-2: Hourly ambient temperature for Islamabad

4.2 Cooling Load Variation

Figure 4.3 shows the cooling load profile developed by type 686 component in TRNSYS. It provides realistic loads instead of time consuming calculations for real buildings. The peak cooling load is 298 kW in the month of July. The average cooling demand of the building is in the months of May and September which is around 150 kW.

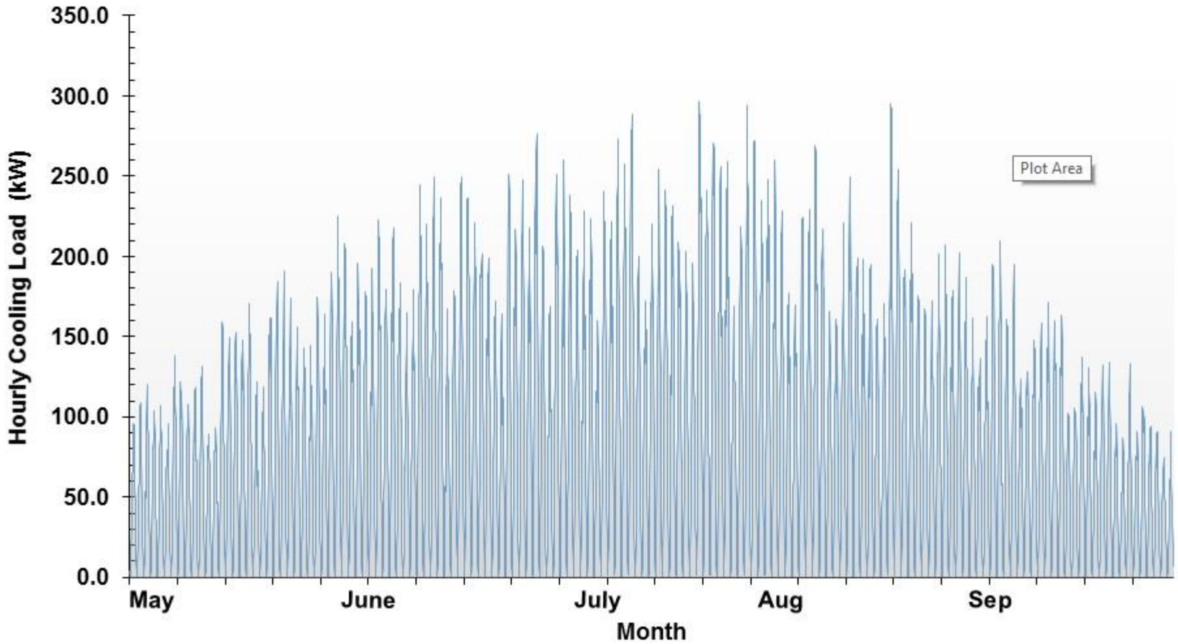

Figure 4-3: Profile of hourly cooling load variation

4.3 Optimum Collector Tilt

Figure. 4.4 shows the effect of collector tilt on solar fraction. It is clear from the figure that maximum solar fraction is obtained for a collector tilt in the range of 5°-10° for ETC and 15°-20° for FPC while mass flow rate, collector area and storage size are kept constant.

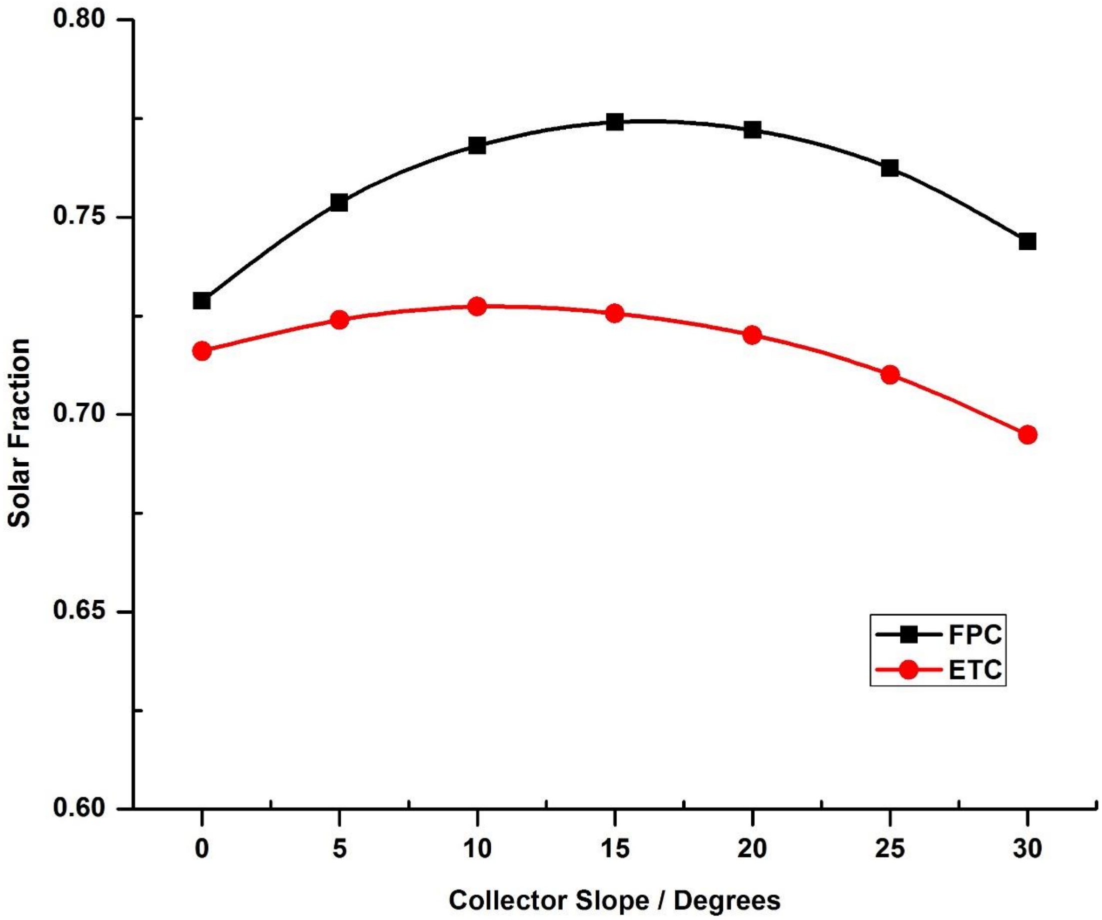

Figure 4-4: Variation of solar fraction with collector slope for FPC and ETC having areas of 2050 m^2 and 400 m^2, respectively

The average monthly declinations and optimum tilt angles for FPC are also analytically (Eq. 3.16) calculated and presented in table 4-1.

Table 4-1: Monthly average declination and optimum tilt angle for FPC

Month	Average Declination δ (Degrees)	Optimum tilt angle $\emptyset - \delta$ (Degrees)
May	18.57	15.14
June	23.03	10.68
July	21.26	12.45
August	13.62	20.09
September	3.49	30.22

The average declination for entire summer season i.e. from May to September comes out to be 15.70°. The corresponding average optimum collector tilt of FPC is estimated to be 18°, which is in good agreement with that estimated from TRNSYS simulation.

4.4 Optimum Storage Size

Figure 4.5 demonstrates the effect of size of thermal storage on fractional primary energy savings ($f_{sav,\ shc}$) for both system configurations and for different flat plate collector areas. For the both configurations, fractional primary energy savings ($f_{sav,\ shc}$) first increases with increase in tank size and then decreases subsequently. For FPC, area of 2050 m^2 in configuration-1, 0.50 $f_{sav,\ shc}$ is achieved with 60 m^3 (29 Lm^{-2}). Similar trend is obtained for configuration-2 however now thermal storage is reduced to 30 m^3 (18 Lm^{-2}) for area 1650 m^2 for the same 0.50 $f_{\mathrm{sav,shc}}$.

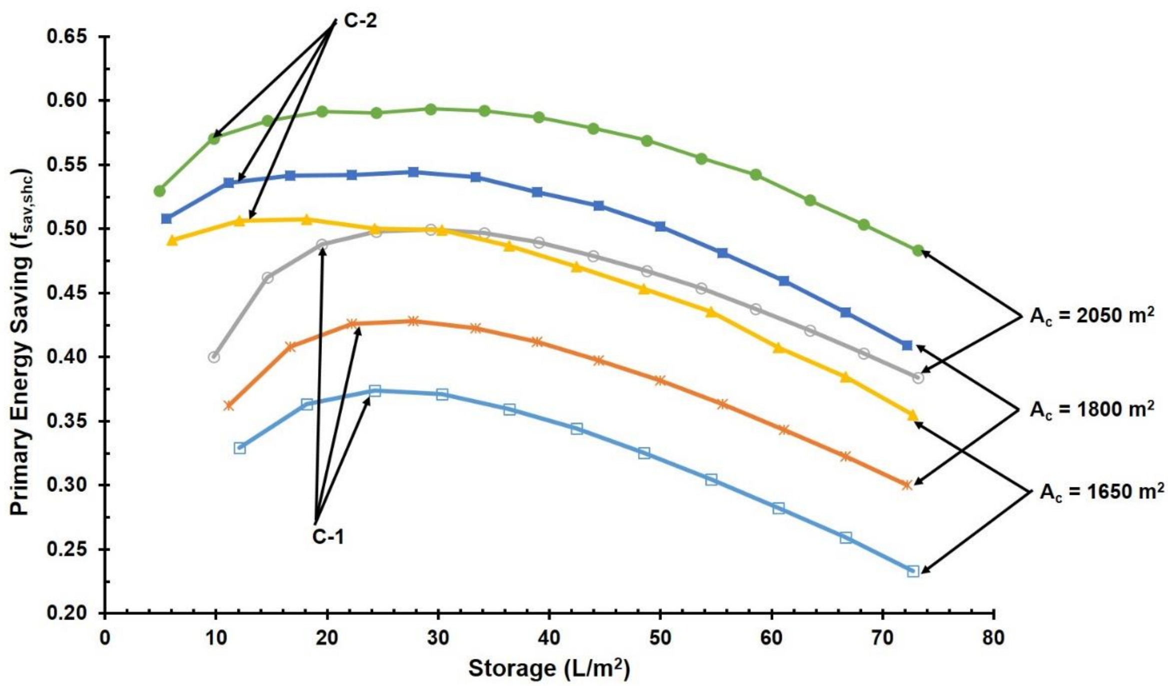

Figure 4-5: Variation of primary energy savings of C-1 and C-2 versus thermal storage size for different flat plate collector areas

Figure 4.6 shows the variation of thermal storage volume with primary energy savings ($f_{\mathrm{sav,\ shc}}$) for different evacuated tube collector areas and for both system configurations. For 0.50 primary energy saving in configuration-1, collector area of 560 m^2 area and storage volume of 15 m^3 (26 Lm^{-2}) storage volume is required. In configuration-2, both solar collector area and storage size are reduced, i.e. collector are of 400 m^2 and storage volume of 10 m^3 (25 Lm^{-2}) result in 0.50 fractional primary energy. It is important to observe that for collector area 400 m^2 and Configuration-1, $f_{sav,\ shc}$ is throughout decreasing with increase in thermal storage tank size. This implies that storage tank can be excluded from the system for such peculiar case. It is interesting to note that as thermal storage size is increased, then after a certain point, C-1 yields more primary energy savings than C-2. It is because of the fact that volume of tank is so large that it takes longer time to reach the required temperature of water. In that case, auxiliary backup remains active for a longer period and there is less contribution from solar collectors.

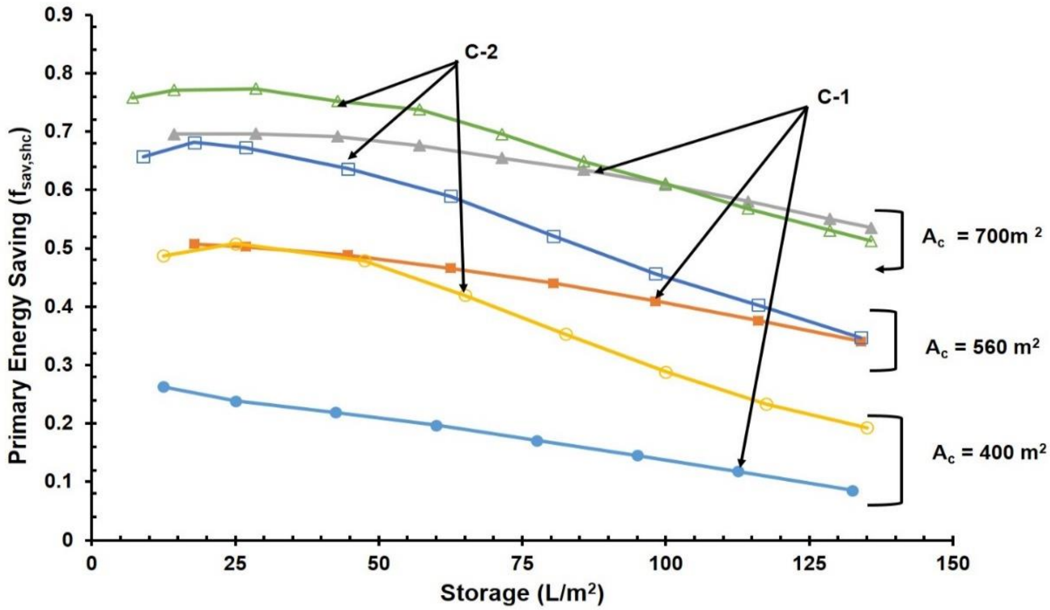

Figure 4-6: Variation of primary energy savings of C-1 and C-2 versus thermal storage size for different evacuated tube collector areas

4.5 Collector Efficiency

Monthly collector efficiencies of FPC and ETC for C-1 and C-2 are shown in Figure 4.7. The efficiencies are calculated for the collector areas at which $f_{sav,\ shc}$ is at least 0.50. Maximum efficiencies are achieved in the month of June for FPC. FPC efficiency for C-1 lies in the range of 0.12 - 0.19 with an average seasonal efficiency of 0.16. ETC's average seasonal efficiency is significantly higher than FPC i.e. 0.48 in C-2 and 0.47 in C-1. This significant difference of monthly efficiency between FPC and ETC can be attributed to high driving temperature (110℃) for absorption chiller as FPC is not recommended for high temperatures. However, it can be seen that the relative difference between collector efficiencies (FPC and ETC) with respect to configurations (C-1 and C-2) is marginal.

In July and August, the average monthly efficiency of ETC for C-1 is higher than C-2 while in May and September C-2 gives better performance. Efficiency depends upon the fluid inlet temperature, ambient temperature and the solar radiation on tilted surface (eq. 3.1). As ambient temperature and solar radiation are same for both configurations, it is then fluid temperature which is responsible for efficiency variation in C-1 and C-2. As in C-2, the fluid keeps on circulating in the solar collector loop, inlet fluid temperature to the solar collector increases which marginally decreases the collector efficiency.

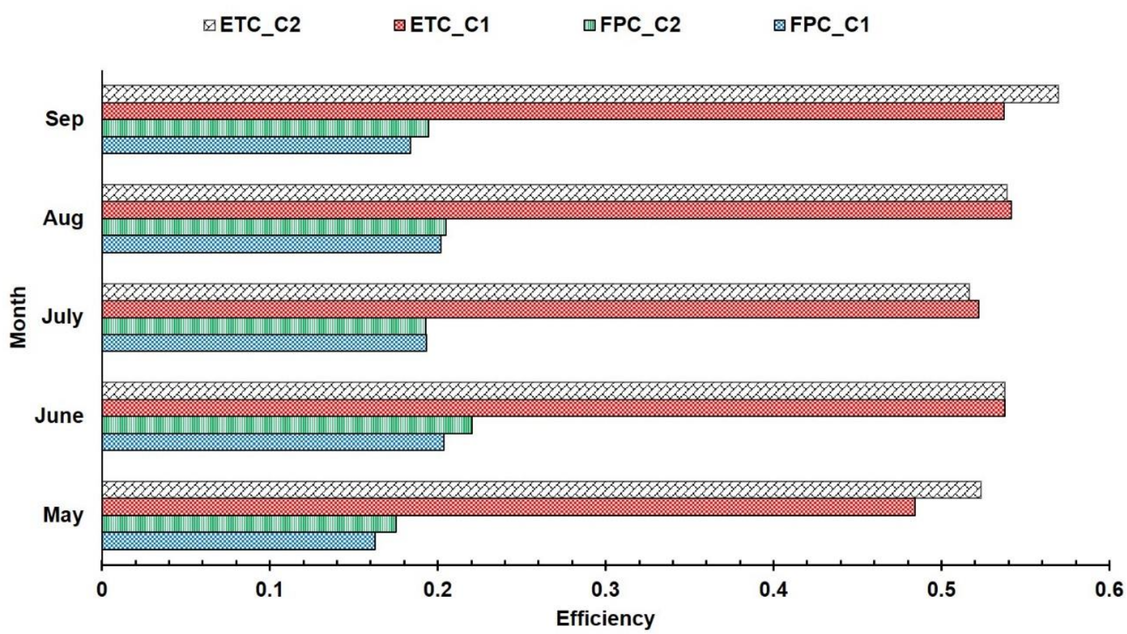

Figure 4-7: Monthly efficiencies of FPC and ETC for C-1 & C-2

4.6 Primary Energy Savings

In Figure 4.8 and 4.9, the variation of $f_{sav,shc}$ with respect to the required collector area of ETC and FPC to meet the cooling demand for both configurations is illustrated. It can be clearly seen that for both collectors, C-2 always yields higher $f_{sav,\ shc}$ than C-1and this difference decreases with increasing collector area. The difference in $f_{sav,shc}$ is more prominent in the case of ETC (see figure. 4.11). Above 900 m^2 of ETC, both configurations yield same fractional primary energy savings because of high temperatures which lead to increase in thermal losses.

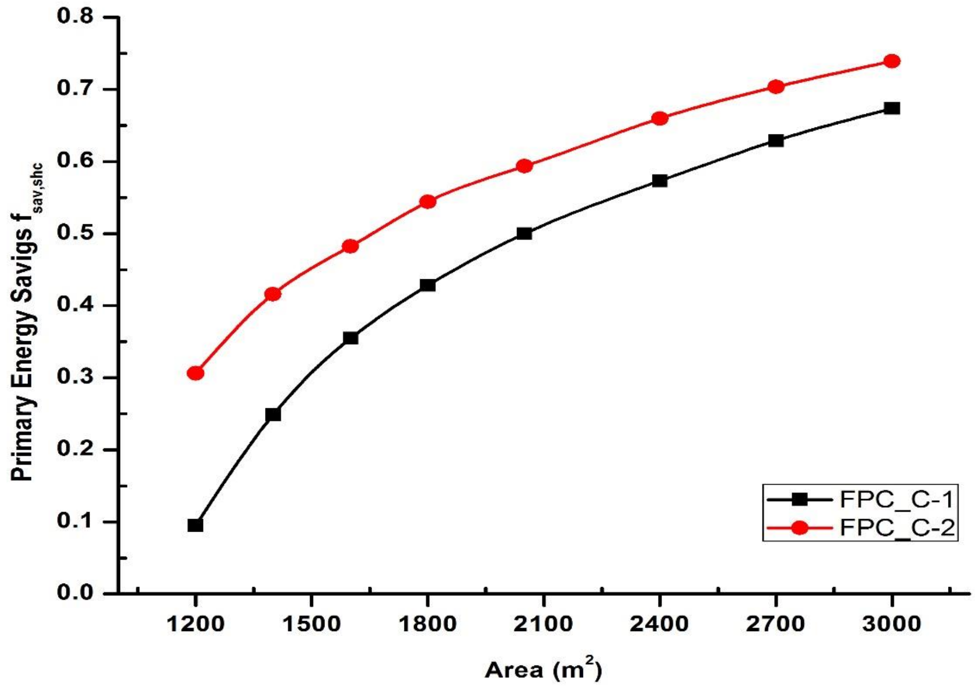

Figure 4-8: Comparison of primary energy savings between two configurations for flat plate collectors

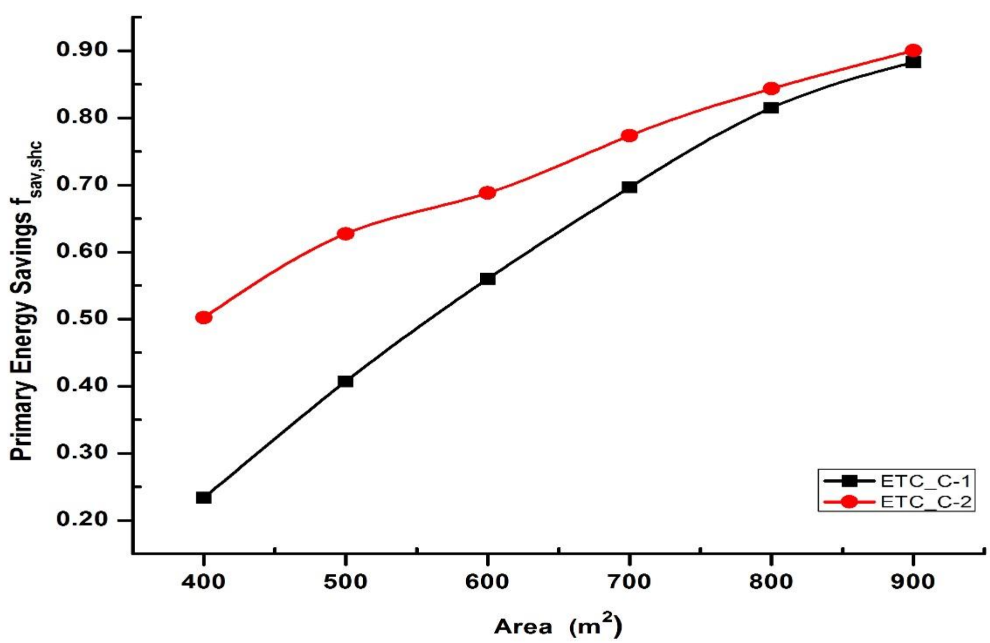

Figure 4-9: Comparison of primary energy savings between two configurations for evacuated tube collectors

4.7 Monthly solar fraction for FPC for both configurations

Figure 4.10 represents the monthly variation of solar fraction for the entire summer season i.e. from May to September for both configurations with flat plate collectors. The area is kept constant for both configurations i.e. 1650 m^2. The overall trend of solar fraction which is higher in May and September is due to the low cooling load of the building in these months as compared in July in which cooling load is maximum. C-2 configuration yields higher SF in the months of May and September while C-1 gives higher SF in the month of July when there is peak cooling load. Average solar fraction for entire season is almost equal to 0.71 for both configurations. This implies that on the basis of solar fraction both system configurations give similar results.

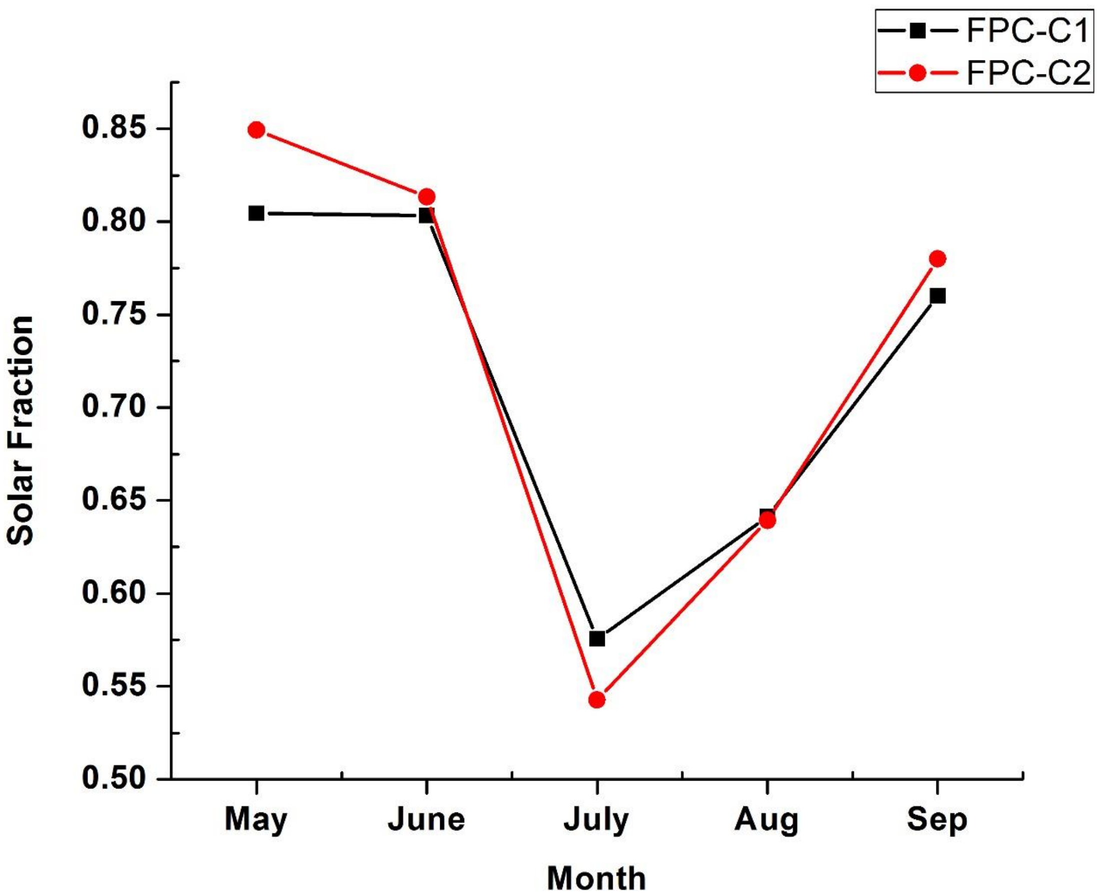

Figure 4-10: Monthly variation of solar fraction for FPC for same area of 1650 m^2 for both configurations

4.8 Monthly solar fraction for ETC for both configurations

Figure 4.11 demonstrates the monthly variation of SF for evacuated tube collectors. Same collector area 400 m^2 is used for both configurations C-1 and C-2. It is evident from the figure that C-2 configuration yields more solar fraction than C-1 configuration in the months of May, June and September. Solar fraction is higher in July and August for C-1. The overall solar fraction for C-2 for entire season is 0.57 while for C-1 is 0.56. For both configurations, solar fraction is minimum in July when cooling load is maximum (figure 4.3). Again the overall difference of solar fraction for both configurations is negligibly small as it is for FPC too.

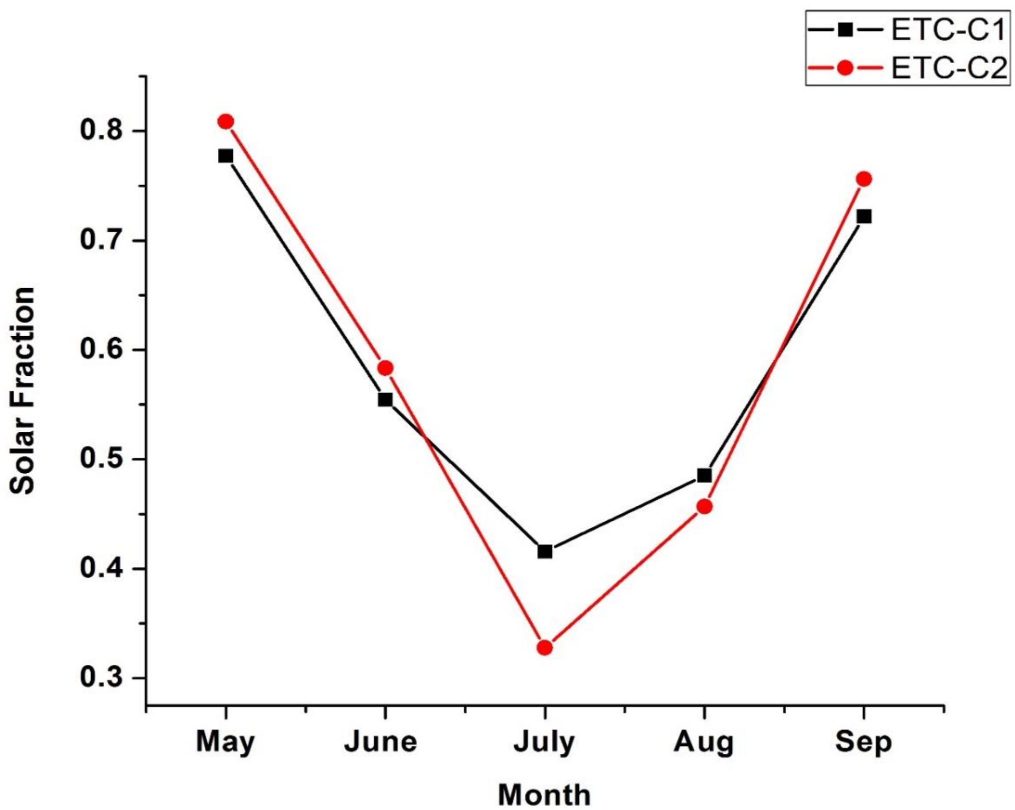

Figure 4-11: Monthly variation of solar fraction for ETC for same area of 560 m^2 for both configurations

4.9 Monthly primary energy savings for both configurations with FPC and ETC

Figure 4.12 demonstrates the monthly variation of primary energy savings for different flat plate collector areas for both configurations. It is evident that for same collector areas for both configurations, C-2 is always yielding more primary energy savings ($f_{sav, shc}$) than C-1 for all months. Average $f_{sav,\,shc}$ of C-2 for whole season is 0.12 and 0.08 higher than C-1 for 1800 m^2 and 2050 m^2 respectively.

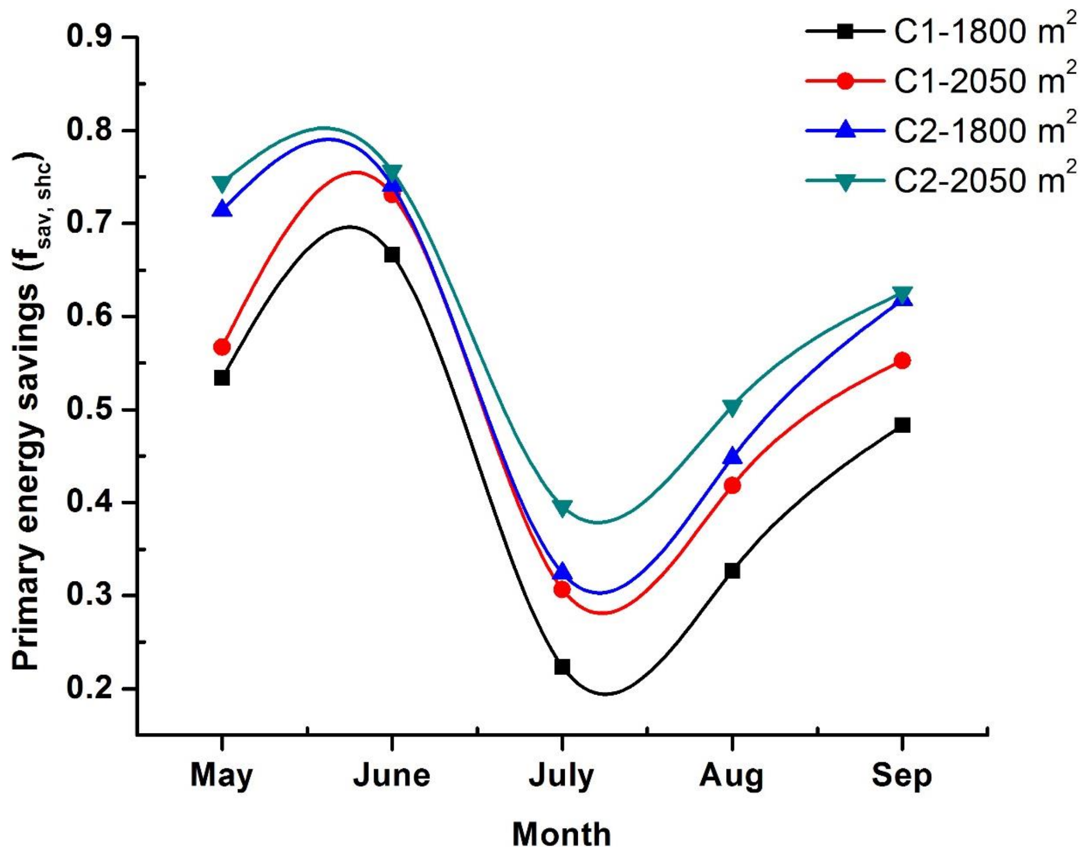

Figure 4-12: Monthly primary energy savings for different flat plate collector areas for both configurations

Figure 4.13 shows primary energy savings for different collector areas of evacuated tube collector areas for different months. $f_{sav,\,shc}$ is again higher than C-1 and the relative difference is even larger than C-2. Variation of solar radiation and cooling load causes the increase and decrease in primary energy savings for different months. Collector area 560 m^2 for C-1 and 400 m^2 for C-2 is essential to achieve 50 % primary energy savings ($f_{sav,\,shc}$).

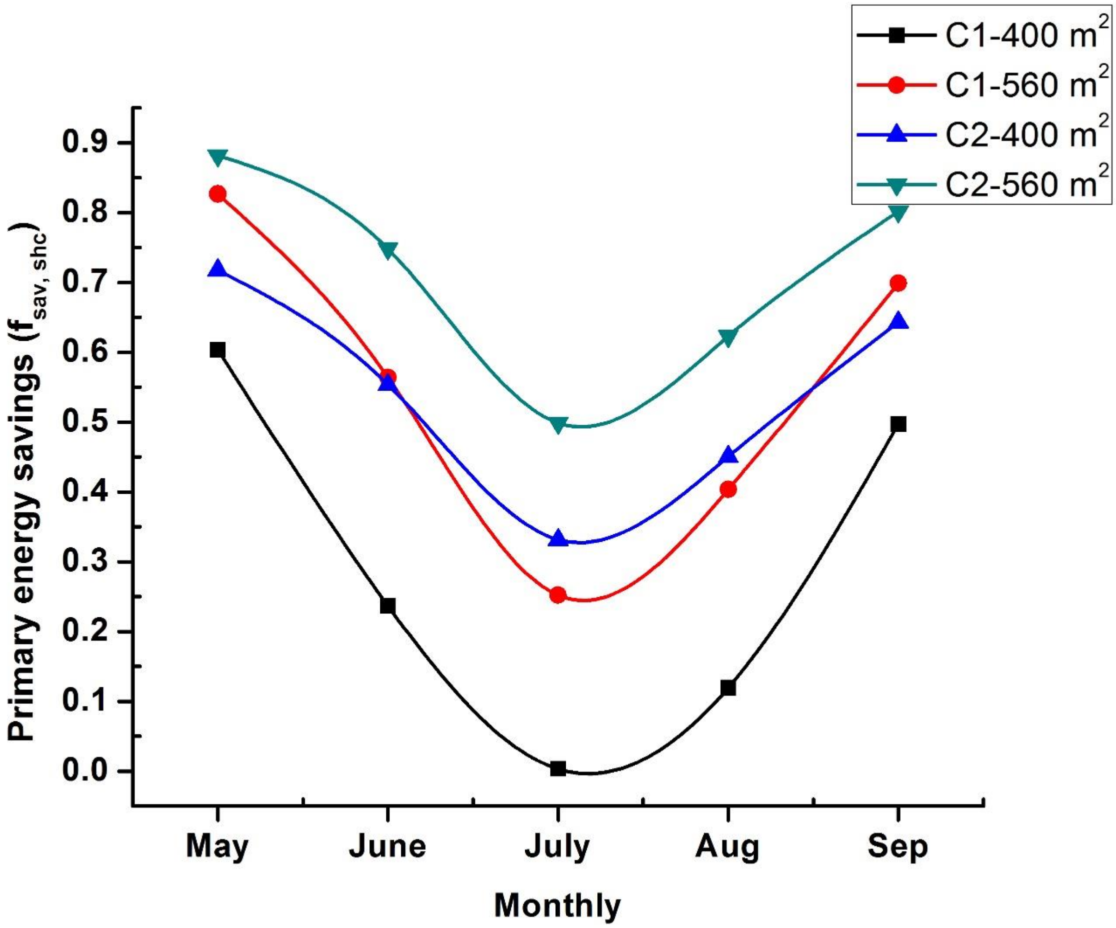

Figure 4-13: Monthly variation of primary energy savings for different evacuated tube collector areas for both configurations

4.10 Collector area per kilowatt of refrigeration

Figure 4.14 shows the collector areas per kilowatt of cooling capacity for 30 % and 50 % $f_{\text{sav, shc}}$ for both configurations. It is can be seen that for 30% $f_{sav,shc}$, difference between solar collector areas is higher as compared with 50 % $f_{sav,shc}$. The minimum required flat plate collector area to achieve $f_{sav,\, shc}$ of 0.5, is more than 4 times larger than evacuated tube collectors. This huge difference is attributed to high driving temperature (110℃) required at the generator of the absorption chiller.

As collector area increases, solar fraction also increases but solar fraction does not increase much when the collector area is too high. The high temperatures tend to decrease the efficiency of solar collectors which decreases the useful energy gain by solar collectors. At higher collector areas, the cost per kilowatt of cooling is high so there should be a balance between useful energy gain from solar collectors and auxiliary energy. The optimum solar collector area is decided on the basis of economic analysis.

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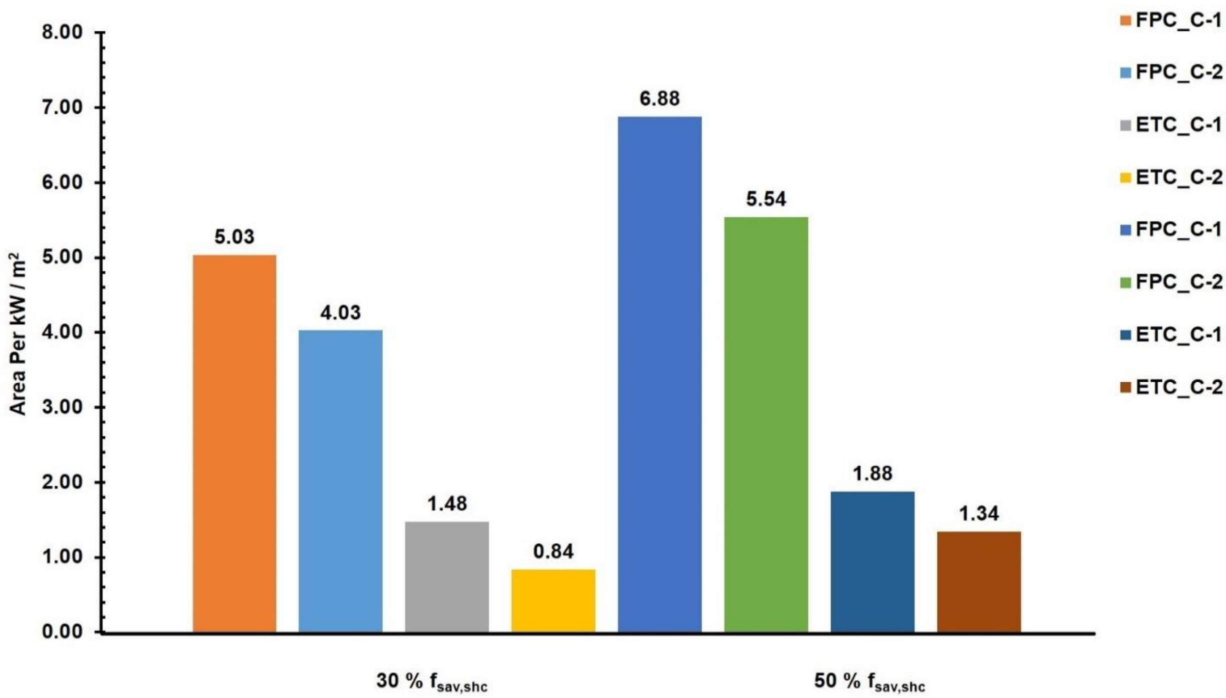


Figure 4-14: Comparison of required collector areas per kW of cold generation

4.11 Summary

The specifications of optimum system for both configurations and both collectors are described in the following table 4.2.

Table 4-2: Summary of optimum parameters of the solar absorption air conditioning system of 85 TR cooling demand for Islamabad

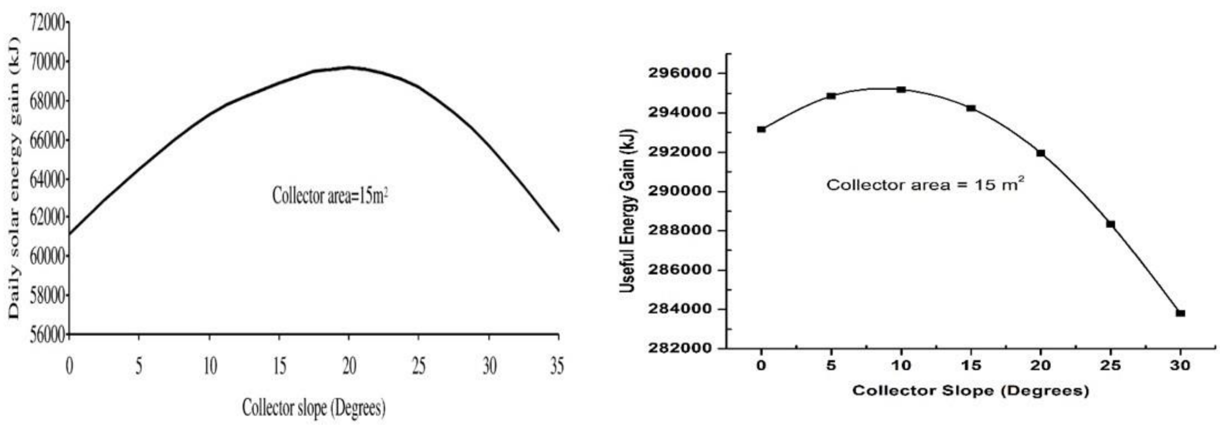
Collector type	Optimum Tilt Angle	Configuration	Collector Area m^2	Tank size m^3	Area per Kilowatt m^2
FPC	15°-20°	C-1	2050	60	6.88
		C-2	1650	30	5.54
ETC	5°-10°	C-1	560	15	1.88
		C-2	400	10	1.34

4.12 Validation Study

As real data from an actual installed system is not available, therefore we have to rely on comparison of some of our simulation results (i.e., their overall trend or pattern) with already relevant published data in journal paper (s).

Assilzadeh et. al. [47] modelled and simulated a system for the climate of Kuala Lumpur, Malaysia using TRNSYS. The system was designed for cooling demand of 1 TR (3.51 kW). From the economic analysis, it was found that 35 m^2 of ETC was the optimum collector area for this cooling demand. The optimum collector tilt was found to be 20° for evacuated tube collectors for Kuala Lumpur, Malaysia as shown in figure 4.15 (a).

For validation purpose, simulations are also carried out for our system with cooling capacity of 1TR (3.51 kW) and 15 m^2 of ETC. It can be observed from figure 4.15 (b) that a similar trend is observed in current study where optimum tilt for ETC is 5°-10° for Islamabad, Pakistan.



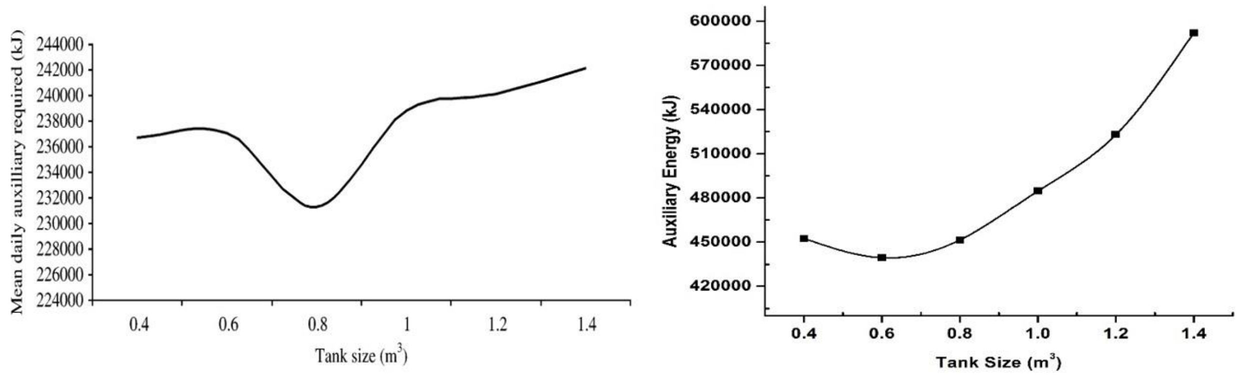
(a) Variation of daily solar energy gain with collector slope for Kuala Lumpur [47]

(b) Variation of useful energy gain with collector slope for Islamabad

Figure 4-15: Comparison of collector slope variation with literature

For the purpose of model validation, variation of auxiliary energy with tank size for 1 TR and 15 m^2 of ETC is compared with Assilzadeh et al. [47]. Simulation is carried out for our system with 1TR (3.51 kW) and 15 m^2 of ETC and for the location of Islamabad. Figure 4.16(b) shows that the auxiliary energy first decreases, at 0.6 m^3 it is minimum and then it tends to increase with increase in tank size. In Assilzadeh et al. [47] minimum auxiliary energy occurs at 0.8 m^3 for 1TR with 15m^2 of ETC for the climate of Kuala Lumpur, Malaysia (see figure 4.16(a)).

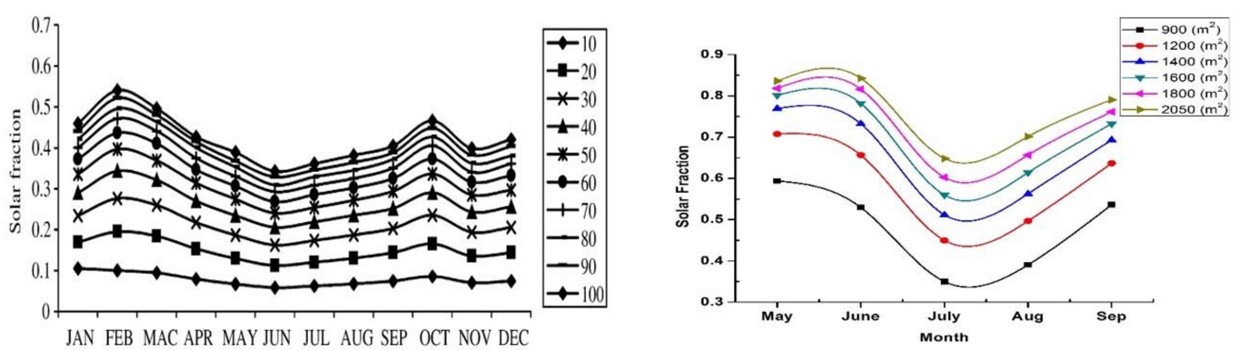
The overall trends of curves in figure 4.16(a) and (b) do not exactly match with each other; which is may be due to the differences in location, relevant weather data and cooling load pattern. Also the results of Assilzadeh et al. [47] are based on the cooling system which is operating for the whole year, while our system results are for summer season only.



(a) for Kuala Lumpur, Malaysia (b) for current system in Islamabad

Figure 4-16: Comparison of variation of auxiliary energy with tank size

Figure 4.17 represents the solar fraction for different collector areas. In the published paper, the solar fraction is determined for entire year while in present study the summer season is spanned over five months. The weather of Kuala Lumpur is humid and the ambient temperature is above 26℃ throughout the year so cooling is required for the entire year. The average global radiation on a horizontal surface is about 700 W/m^2 year around.



(a) System proposed for Kuala Lumpur (b) System proposed for Islamabad

Figure 4-17: Variation of solar fraction for different collector areas of FPC

CHAPTER 5

Conclusions and Recommendations

5.1 Conclusions

TRNSYS simulations are performed for two configurations of a solar based single effect absorption cooling system having a peak cooling capacity of 85 TR (298 kW).Weather data file for the location of Islamabad (33.71° N, 73.06° E) is used to simulate the system for the entire summer season.

a) Optimum collector tilt based on optimum SF was estimated to be 5°-10° for ETC and 15°-20° for FPC.
b) Optimum storage volume for C-1 and C-2 is estimated to be 25 L/m^2 for ETC.
c) Optimum storage volume for C-1 and C-2 is estimated to be 30 L/m^2 for FPC.
d) Configuration-2 with isolated collector-storage loop from auxiliary-chiller loop results in better $f_{sav,shc}$ compared to configuration-1 with common storage.
e) With increase of collector area, the difference between primary energy savings for both configurations diminishes.
f) Difference between solar fraction (SF) of C-1 and C-2 for same collector areas is marginal when compared with primary energy savings. This implies that primary energy savings is a better criteria for comparison of various system configurations than solar fraction.
g) With increase of collector area, the difference between primary energy savings for both configurations diminishes.
h) The trend of results (i.e. useful energy gain vs collector slope, auxiliary energy vs storage tank, monthly solar fraction for different collector areas) are found to be in good agreement with published data.
i) There is a marked difference of upto 30% between monthly collector efficiencies of ETC and FPC, so FPC is not a preferred choice for current system and Location.
j) Finally, configuration-2 with ETC is found to be the optimum selection for achieving maximum primary energy savings and minimum collector area per kilowatt of cooling capacity.

5.2 Future Work Suggestions

a) The system can be modified to be used for heating purpose in winter too.

b) The performance of the system may be assessed and improved by arranging the collectors in series and/or parallel arrangement.

c) The thermal storage may be divided into more than one tanks. It may have a positive influence on the system performance.

d) The system can be integrated with a real building and a control strategy to control to the indoor temperature and relative humidity.

e) Vapor compression system can be used as auxiliary device for producing cold.

f) The performance curves of absorption chillers of various manufacturers with different energizing temperatures can be implemented in TRNSYS.

g) Double and triple effect absorption chillers along with CPCs can be studied for meeting cooling loads.

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CERTIFICATE OF COMPLETENESS

It is hereby certified that the dissertation submitted by **NS Muhammad Shoaib Ahmed Khan**, Reg No. **NUST201362438MCEME35113F**, Titled: ***Modelling and simulation of solar absorption cooling system using TRNSYS*** has been checked/reviewed and its contents are complete in all respects.

Supervisor's Name: **Dr. Tariq Talha**

Signature: ______________________

Date: ______________________