Design & Modelling of Absorption Cooling System integrated with Solar Thermal Storage: Utilizing Solar Energy for Cooling of an Educational Building in Pakistan using TRNSYS

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Session 2017-2019

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A Thesis Submitted to the US-Pakistan Center for Advanced Studies in Energy in partial fulfillment of the requirements for the degree of

MASTER of Science in

ENERGY SYSTEM ENGINEERING

US-Pakistan Center for Advanced Studies in Energy (USPCAS-E)

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May 2020

Declaration

I certify that this research work titled "Design & Modelling of Absorption Cooling System integrated with Solar Thermal Storage: Utilizing Solar Energy for Cooling of an Educational Building 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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Acknowledgements

I am grateful to my Creator Allah Subhana-Watala for Your countless blessings upon me throughout my life. Whatever hurdles I faced in my life, you made them fade away for me. You guided me through each and every step of my life. Whoever helped me during my master's thesis, whether my family or friends or any other individual, it was your will. Indeed, none be worthy of praise but you.

I would like to pay special thanks to my supervisor Dr. Adeel Waqas for his continuous support and cooperation and especially for lending me TRNSYS software. This research effort would not have been carried out without his help. Without his cooperation and support, this study would not have been possible. I pay my gratitude for his guidance throughout the whole thesis.

I would also like to pay special regards to my GEC members Dr. Majid Ali & Dr. Adeel Javed for their expert advice, incredible guidance, continuous support and help throughout my thesis. They trusted in my capabilities to carry out this research work and made it possible.

I wish to extend my special thanks to my praiseworthy Co-Supervisor Dr. Jinwang Tan for his help in writing the research article and the acquisition of funding for publication. I must also express my profound gratitude to Shenzhen University China for providing me a wonderful experience to work with international research students via international exchange program.

Finally, I want to thank my parents, spouse and siblings for their prayers, continuous encouragement, inspiration and endless patience and tolerance.

Abstract

Performance of solar absorption cooling system (SACS) is the focus of contemporary studies for decreasing electrical energy consumption of buildings as the conventional cooling system of buildings is the main consumer of electrical energy during summer season in hot-humid climates. In this study, the performance analysis of SACS by manipulating different flow schemes to the heat transfer fluid between different components of the system was performed. TRNSYS model of SACS in an education building located at the city of Peshawar (34.00°N, 71.54°E), Pakistan to encounter the peak cooling load of 108 kW (during operating hours of the building i.e. 09 a.m. to 05 p.m.) is developed and all possible flow schemes of heat transfer fluid between the system's components were compared. In Scheme-1 (S-1), conventional flow pattern is used in which the hot water exiting from the chiller unit flows directly toward stratified thermal storage unit. In Scheme-2 (S-2), the modified flow pattern of hot water exiting from the chiller unit will divert towards the auxiliary unit, if its temperature exceeds the temperature at the hot side outlet of the tank. Another modified flow pattern which is Scheme-3 (S-3) in which the hot water leaving the chiller keep on diverting towards auxiliary unit unless the outlet temperature from hotter side of the tank would reach at the minimum driving temperature (109℃) of the chiller's operation. Simulations in TRNSYS evaluates the SACS's performance of all the schemes (conventional and modified) for the whole summer season and for each month. In General, S-3 with evacuated tube solar collector, results in better primary energy saving with smallest collector area per for kilowatt for achieving 50% primary energy saving for whole summer season.

Keywords:

Solar Absorption cooling system; Educational building; TRNSYS; Flow schemes; Solar cooling; solar energy; Pakistan

Publication:

APPLIED SCIENCES (Switzerland) an Open access journal by MDPI

Appl. Sci. **2020**, 10, 3327; doi:10.3390/app10093327

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Nomenclature

Abbreviations

COP	:	Coefficient of performance
ETC	:	Evacuated tube collector
FPC	:	Flat plate collector
kW	:	kilowatt
PES	:	primary energy saving
SACS	:	Solar absorption cooling system
SCS	:	Solar cooling systems
TR	:	Tons of Refrigeration
TRNSYS	:	Transient system simulation
TMY	:	Typical meteorological year
VCCS	:	Vapor compression cooling system

Nomenclature

List of Symbol

Symbol	unit	Description
$A_{□}$	m^2	Area of solar collectors
Ai	m^2	Surface area for ith tank segment
$a_{□}$		Collector's Optical efficiency
a_1	$kJ\,hr^{-1}m^{-2}K^{-1}$	Efficiency slope
a_2	$kJ\,hr^{-1}m^{-2}K^{-2}$	Efficiency curvature
α_i		a control function defined by αi = 1 if i = Sh; 0 otherwise
a_{season}		Season Offset parameter for cooling load
b_{season}		Season multiplier for cooling load
□		Slope of the collector
$□_i$		a control function defined by βi = 1 if i = SL; 0 otherwise
Cp_f	$kJ\,kg^{-1}\,k^{-1}$	Specific heat of the fluid in the storage tank
Cp_{hw}	$kJ\,kg^{-1}\,k^{-1}$	Specific heat of the hot water
Cp_{chw}	$kJ\,kg^{-1}\,k^{-1}$	Specific heat of chilled water
Cp_{cw}	$kJ\,kg^{-1}\,k^{-1}$	Specific heat of cooling water
$Capacity_{rated}$	$kJ\,hr^{-1}$	rated capacity of the chiller
COP_{rated}		Rated coefficient of performance
dT_i	℃	Change in temperature between adjacent nodes
dt	hr	Change in time interval (time step)
Designload	$kJ\,hr^{-1}$	Designed cooling load
$EndOfSeason$	hr	The hour of year at which specified season ends
$□_{heat}$		Efficiency of the Boiler
$□_{elec}$		Efficiency of thermal power plant
$f_{DesignEnergyInput}$		Fraction of design energy input
G_T	$kJ\,hr^{-1}\,m^{-2}$	Incident global solar radiation
I	$kJ\,hr^{-1}\,m^{-2}$	Solar radiations
I_T	$kJ\,hr^{-1}\,m^{-2}$	Incident radiations on collector surface
Load	$kJ\,hr^{-1}$	cooling load as a function of time
$\dot{m}_f$	kg/s	Mass flow rate of fluid in boiler

$\dot{m}_h$	kg/s	Heat source fluid's mass flow rate
$\dot{m}_L$	kg/s	Mass flow rate of fluid delivered to load
$\dot{M}_i$	kg/s	Mass flow rate of the fluid at ith node of storage tank
$\dot{m}_{cw}$	kg/s	Mass flow rate of cooling water
$\dot{m}_{hw}$	kg/s	mass flow rate of hot water in or out of chiller
$\dot{m}_{chw}$	kg/s	mass flow rate of chilled water in or out of chiller
$\dot{Q}_{boiler}$	kW	Energy rate consumes by the boiler to heat the fluid
$\dot{Q}_{cooling, ref}$	kW	Referenced Cold energy produced by compression cooling system
$\dot{Q}_{fuel}$	kW	Energy rate of fuel consumes to heat up the fluid
$\dot{Q}_{hw}$	kW	Energy rate of hot water entering into chiller
$\dot{Q}_i$	kW	Energy rate of the ith node in the tank
$\dot{Q}_{need}$	kW	Energy rate required by boiler to heat up the fluid
$\dot{Q}_{remove}$	kW	Energy rate of energy removed from chilled water
$\dot{Q}_{solar}$	kW	Energy rate collected from the sun
$\dot{Q}_{cw}$	kW	Energy rate of cooling water
$\dot{Q}_{aux}$	kW	Auxiliary energy consumes by chiller
U	kW/ (m^2.K)	Heat transfer coefficient
$StartOfSeason$	hr	Hour of the year at which season begins
T_a	°C	Ambient temperature
T_h	°C	Temperature of heat source fluid
T_i	°C	Temperature of the fluid at ith node of the tank
T_L	°C	Temperature of the fluid flowing towards load
$T_{hw, in}$	°C	Temperature of the hot water enter in chiller
$T_{hw, out}$	°C	Temperature of the hot water out from chiller
$T_{chw, in}$	°C	Temperature of chilled water entering in the chiller
$T_{chw, set}$	°C	Temperature to chilled water out from the chiller
$T_{cw, out}$	°C	Temperature of cooling water leaving cooling tower
$T_{cw, in}$	°C	Temperature of cooling water entering the chiller
T_{env}	°C	Temperature of the environment

T_i	℃	Temperature of the fluid inlet to the collector
T_{in}	℃	Temperature of fluid incoming to the boiler
T_{i-1}	℃	Temperature of the previous node in the tank
T_{i+1}	℃	Temperature of the next node in the tank
T_{set}	℃	Set point temperature of boiler
γ_i		a control function
time	hr	Simulation time step
η_{boiler}		Efficiency of boiler
X_{day}		Function to be modifying for seasonal variations of loads
X_{hour} (in the loads)		Function to be modifying for hourly variations of loads
$X_{noise,\ hour}$		Function modifying for hourly noise in the calculated loads
$X_{noise,\ day}$		Modifying function for daily noise in the calculated loads
η		Efficiency of Solar collector

CHAPTER 1: INTRODUCTION

1.1 Background

Global Energy demand is increasing rapidly, and scientist's community have serious concerns over difficulties in energy supply, exhaustion of energy resources and their hazardous environmental shocks (like depletion of ozone layer, climate change and global warming). This trend of increasing energy demand is predicted to grow endlessly in future. Energy use of the regions with emerging economies are predicted to grow at a rate of 3.2% annually and will exceed by 2020 [1]. The main energy consumption sectors are industries, transportation, commercial, agriculture and domestic. Commercial and domestic sector includes buildings such as residential, hotels, educational and hospitals. Heating, Ventilation and Air Conditioning is the main end use energy consumption of the buildings with 50% weight, lighting with 15% while 10% for appliances and rest energy consumption end use comprises of food preparation, ventilation etc. [1] which is comparatively less than HVAC, so that it is the noticeable need of time to limit the massive consumption of energy for HVAC purpose.

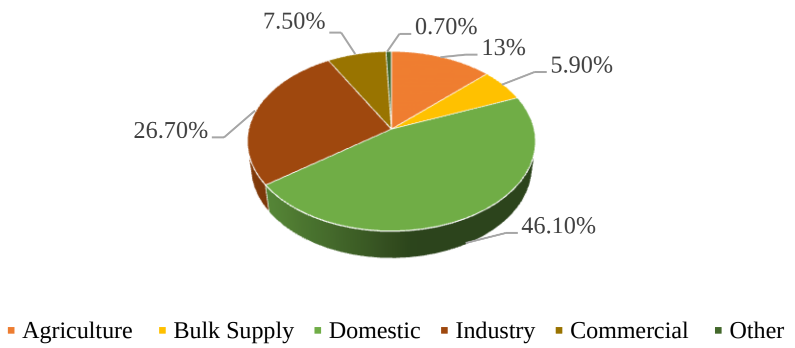

Figure 1-1: Pakistan Energy consumption in various sectors (%) [2]

1.2 Pakistan energy scenario

Pakistan is an evolving country in the region of South-Asia and its growing population and industrial development causing sharp rise in consumption of energy. The energy sector of Pakistan depends heavily on fossil fuels. Oil, gas, coal, nuclear and hydro are the primary energy sources of which gas and oil are dominantly consuming sources of energy in the Pakistan [2]. As the main sectors of energy consumption are commercial and domestic in Pakistan as shown in Figure 1-1 and more than 50% of the electricity is expended by same two sectors. Thus, it is necessary to somehow control the electrical load by employing some alternative resources of energy for HVAC.

1.3 Conventionally used HVAC Systems

The necessity of cooling system for public buildings and industries is rising with increase of energy consumption and also for modern-day standard of living and thermal comfort during summer season. To encounter the cooling load of the buildings, VCCS is frequently used all around the globe, often referred as conventional cooling systems [3].These conventional cooling systems uses intensive electrical energy to operate compressor which is the essential part of these systems as in Figure 1-2.

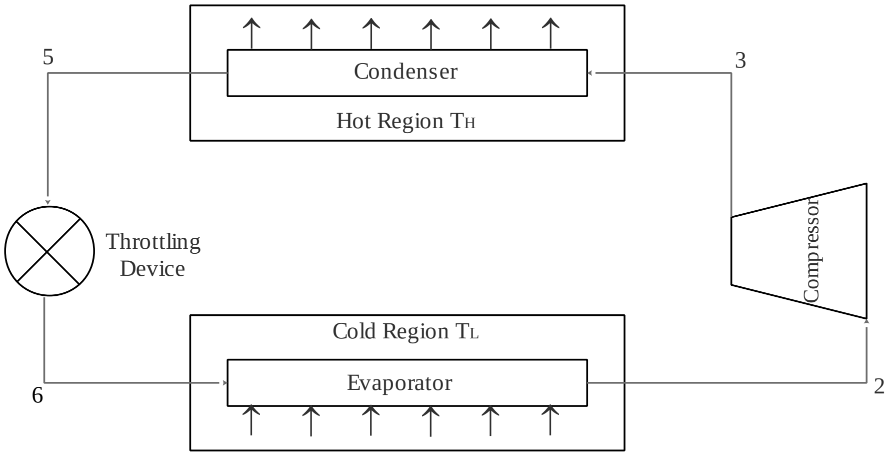

Figure 1-2. Vapor Compression Cooling System [4]

These systems are commonly used because they have high coefficient of performance (COP), small size, and have lower volume and weight than other cooling systems. Common refrigerants used in conventional cooling system are chlorofluorocarbon (CFC), hydro chlorofluorocarbon (HCFC) and hydro-fluorocarbons (HFC) [4] whose emissions are causing depletion of ozone layer and increases the average heat of environment (global warming) [5] which is a serious topic in recent times and researchers are trying their best to eliminate these harmful emissions [6]. One more drawback is that, these systems consumes large amount of electrical energy to fulfill the requirement of thermal comfort for occupant of buildings [4]. Due to this reason, government has to construct new mega projects of power plants to increase the power generation capacity to fulfil the electrical demand of the country which increases exponentially in summer season due to the operation of conventional cooling systems whose downside is high economic and environmental cost [7].

1.4 Solar Cooling

To resolve the problem of high electricity demand in summer season due to conventional cooling systems, Solar cooling technology has a great potential to lessen economic and ecological issues [8]. The deployment of sun’s energy for space cooling purposes is an appealing option, because there is a coincidence between demand of cooling and availability of energy source input [6]. Consequently, solar cooling can be used for saving electricity and are suitable to meet the cooling load of buildings [9]. These solar cooling systems have further advantages like zero potential of ozone layer depletion, energy from sun and excess heat from industries can be effectively utilized for the operation of these cooling systems [5].

Another benefit is that have very small maintenance requirements because there are no moving parts.

Pakistan is facing severe electricity load shedding problem from the last two decades reaching about 10 hours daily average in densely populated areas and about 18 hours in countryside [10] especially during summer season . Domestic and commercial sectors consume huge amount of electricity during summer season as shown in Figure 1-3. The topographical location and hot climatic conditions of the country makes the region ideal for exploiting solar energy. It is estimated that the country experiences more than 300 sunny days annually [11]. The study of weather data acquisition showed that the region of Pakistan receives global irradiance of about 1900–2200 kWh/m^2 annually [12]. Consequently, it is more reasonable option to decrease the electrical load with the deployment of SACS for air-conditioning of buildings in Pakistan. Also, the cost of electricity unit is very high, making these solar cooling systems an economical option in the region [13].

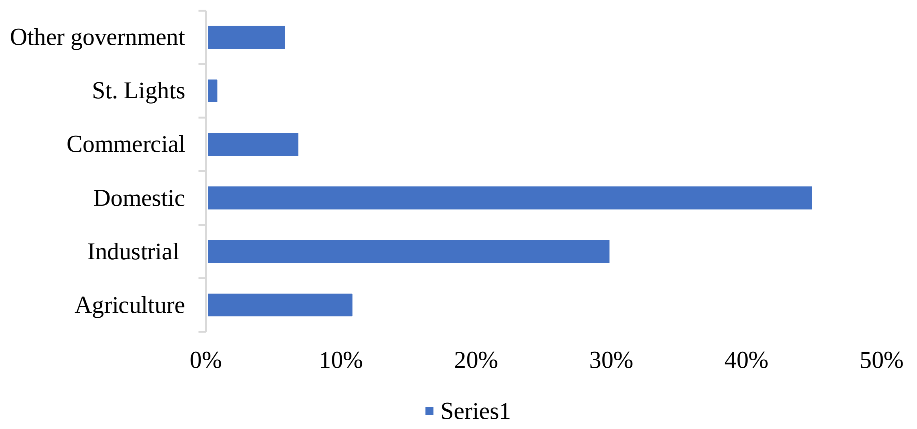

Figure 1-3: Electricity Consumption of different sectors in Pakistan [11]

1.5 Benefits of Solar Cooling

Some benefits of utilizing SACS are as follows:

i. Need of imported fuel can be abated.
ii. Can diversify the energy supply.
iii. Solar cooling systems have capability to conserve natural resources as these resources are depleting quickly.
iv. Lowers CO_2 discharge at minimum price.
v. Control air pollution in urban areas.
vi. Electricity bills of the household can be saved.
vii. Deployment of these solar cooling systems can generate indigenous jobs and accelerate local economy.

Solar air conditioning is not a modern concept. It had been used from the prehistoric times by the name of passive cooling [14]. Solar cooling categorizes as passive cooling and active cooling which are described below:

1.6 Passive Cooling system by using solar energy

When heat from the sun transfer into the building is reduced by architectural design of the building, it is termed as passive cooling. These are eco-friendly and energy-efficient techniques used to improve the human comfort environment inside the buildings with nil or little consumption of electrical energy. These techniques can be employed either by eliminating heat from inside the building by natural flow to a heat sink or by blocking heat from flowing into the dwelling space from heat sources externally. The feasibility of these practices depends mainly on the regional weather [15].

1.7 Active Cooling system by using solar energy

Active cooling technologies are also necessary as only passive cooling cannot provide required human comfort environment because the complete removal of heat (up to the comfort level) is not possible sometimes from required space. These Cooling systems involves the requirement of active energy (usually electricity) to cool required region. Depending on the way how energy is provided, the active solar based cooling systems are usually divided into two main categories which are given below:

i. Systems operated by electricity
   - Vapor compressional cooling system
   - Thermo-electric system
   - Stirling system

ii. Systems operated by heat energy (Solar Thermal Refrigeration)
   - Absorption cooling system
   - Adsorption cooling system
   - Desiccant cooling system
   - Ejector cooling system

1.8 Electrically operated Solar cooling Systems

Solar photovoltaic panels are used for converting sunlight to direct electrical energy and that electrical energy when used to run some refrigeration system, is known as solar electrical refrigeration system. Following are some refrigeration systems that are operated by the use electrical energy of PV panels.

1.8.1 Solar Vapor Compression Cooling System

The combination of PV panels and vapor compression refrigerator is known as solar vapor compression refrigeration system. A PV panel is the arrangement (in series or parallel) of different solar cells made up of semiconductors materials. The net output and cost of solar cells varies widely depending on the fabrication material and their manufacturing methods. Mostly, solar cells which are available commercially in the market are made from silicon. A simple PV panel is depicted in Figure 1-4.

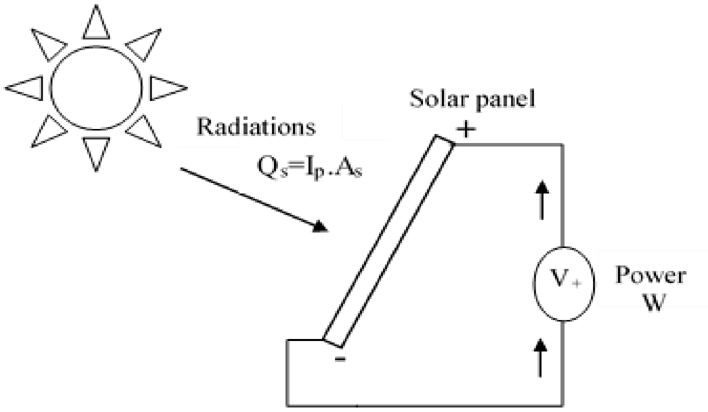

Figure 1-4: Solar Photovoltaic Panel

Efficiency which is reported in laboratories are higher in values than in the field, the highest performance solar panel available in markets only yields 15% efficiency at solar noon during sunny day. [16].

$$\eta_{panel} = \frac{W}{I_p * A_s} = \frac{W}{Q_s} \quad (1.1)$$

The efficiency of convention vapor compressional cooling system is defined as [16]:

$$\eta_{cooling} = \frac{Q_e}{W}$$

(1.2)

The greatest benefit of employing PV panels for space cooling application is its simple structure and higher total efficiency when used with a conventional vapor compression system [16]

$$\eta_{solar,cooling} = \eta_{panel} * \eta_{cooling} = \frac{Q_e}{Q_s} \quad (1.3)$$

A simple VCCS is shown in Figure 1-5, where mechanical compressor run by consuming work W to give the cooling power.

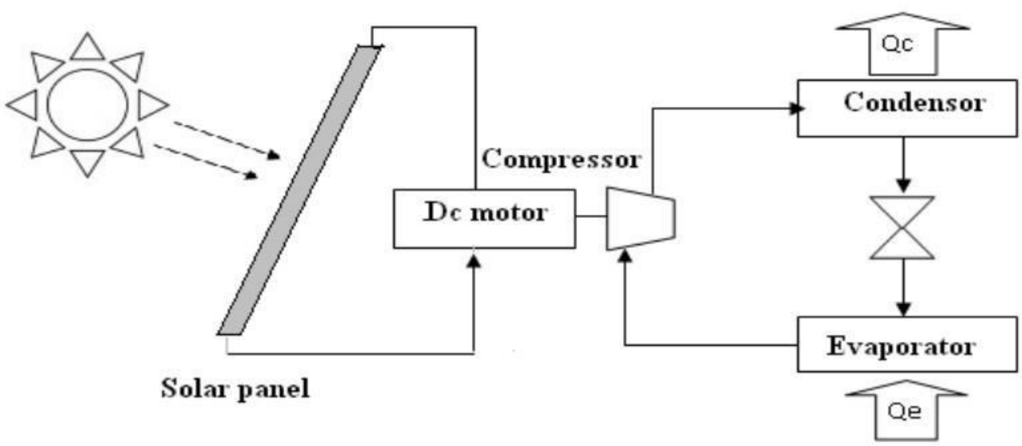

Figure 1-5: Solar Vapor Compression Cooling System

1.8.2 Solar Thermo-Electric Cooling System

This System does not use any refrigerant. Electrons are used as heat carrier. The principal is that, when a direct current will flow across the junction of two different materials, heat will be absorbed or released. This is known as Peltier effect [17]. Solar PV panels can also be coupled with these electrically operated cooling systems. Thermo-electric part of the system consists of semiconductor materials. Different alloys like antimony telluride (Sb_2Te_3) and Bismuth telluride (Bi_2Te_3) are used in these thermo-electric refrigeration systems. These systems have advantage of smaller size and have no moving parts. Absence of refrigerants makes these system pollution free [16]. Their small sizes make them attractive option to be used in satellites and spaceships. The main drawback is their small COP lies between 0.3 to 0.6 [17].

1.8.3 Stirling Refrigeration System

Stirling refrigerator work as Carnot cycle, is one of the vital refrigeration cycle approaches [18]. Stirling refrigerator when coupled with solar panel can be used to realize cooling effect. These refrigerators yields low temperature stream and their COP value is less than vapor compression systems [19]. The COP lies between 1.2 to 1.6 that depends on hotness or coldness of reservoir and also on environment temperature [20]. Their power density is low and heat transfer between the working fluid and ambient is also not appreciable due to which an efficient Stirling system cannot by developed. These constraints are more subjugated in bigger systems [21]. Hence, Stirling systems with small sizes are viable with conventional vapor compression systems [16].

1.9 Solar Thermal Refrigeration

Refrigeration effect is obtained by using solar heat of thermal collectors rather than electric energy of solar PV panels in solar thermal refrigeration systems. Generally, solar collectors have two main types:

- Concentrating solar collectors
- Non-concentrating solar collectors

Non concentrating collector types are cheaper (than concentrating types) and gives hot water that are mostly used to operate single effect absorption systems whose efficiency is also small [22]. Two common types of collectors that are non-concentrating and easily accessible to collect solar heat are, evacuated tube collectors and flat plate collectors The type of solar collector that is most commonly used in the world is flat plate collector (FPC) [23]. The absorber surface of flat plate collectors that is used to absorb solar radiations is made up of some metals with good absorbance property. An anti-reflective coating with glass is often embedded to minimize the heat losses [24]. In Evacuated tube collector (ETC), metallic absorber enclosed in an evacuated glass tubes assembly. Due to vacuum, convectional losses are minimum in this type of collectors. The two types of aforementioned solar collectors are shown in Figure 1-6[16].

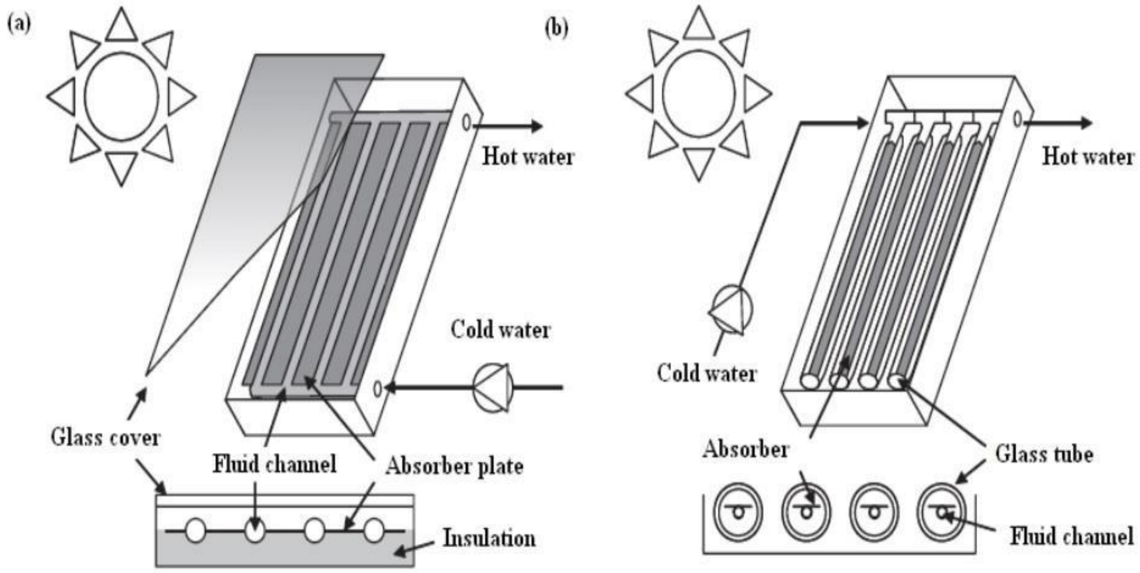

Figure 1-6: (a) Flat plate collector (b) Evacuated tube collector

Collectors in Figure 1-6 are used to provide heat to system that are thermally operated. Their efficiency is the function of its fluid's temperature, environment temperature and global radiation incidence on absorber surface. The collector's efficiency generally has inverse relation with working fluid temperature. Usually, the systems are designed by considering these trends [9].

1.9.1 Absorption Refrigeration System

It is utmost popular applied technology. It requires nil or very little electrical energy to yield cooling effect. These refrigeration systems are driven by the use of thermal energy. The use of compressor for the compression of gas is eliminated. They require very less maintenance because of the absence of moving parts. Absorption refrigeration system needs three thermal reservoirs for their operation.

- The space to be cooled serves as cold reservoir.
- Heat source either from boilers or solar thermal storage acted as hot reservoir.
- Some type of cooling tower acts as the reservoir to reject the heat of condenser.

The reason of their small COP than VCR is that the higher energy needed by the vapor generator unit of the absorption refrigeration system for the regeneration of the refrigerant.

1.9.1.1 Types of Absorption Refrigeration Systems

Absorption refrigeration system usually are of two types that are [25]:

a. Single Effect system

i. Uses a single generator for the generation of refrigerant used.
ii. Requires hot water with temperature ranges between 80 to 150℃.
iii. The value of COP between 0.6 to 0.7
iv. Cheaper than double effect absorption refrigeration system.

b. Double Effect system

i. Uses two generators for the generation of refrigerant and with the higher temperature.

ii. Requires Hot water or heat transfer fluid temperature between 155 to 205 ℃.

iii. COP between 0.9 to 1.2

iv. Efficiency is higher but expensive than single effect refrigeration system [25].

1.9.1.2 Single Effect Solar Absorption Cooling System

In these cooling systems, we usually store hot water in the storage tank with the use of solar collector. This stored heat is used for the continuous supply at the generator and also have the benefit of keeping the system operation uniform if there is a gap between the demand of cooling and solar gain. Water evaporates from the LiBr and H_2O solution in the generator. Water vapors from the generator are passed through the condenser where they condensed and cooled by rejecting heat in cooling tower. This condensed liquid then passes from expansion valve to reduce its pressure and finally this cold water with very low pressure passed through the evaporators and absorbs heat of the living room and in this way, refrigerant vaporizes and attracted towards absorber. Absorber contained lithium bromide solution which has capability to attract the water vapors and serves the purpose of suction of refrigerant same as the compressor and heat due to exothermic reaction in absorber rejected in cooling tower. Then this refrigerant pumped

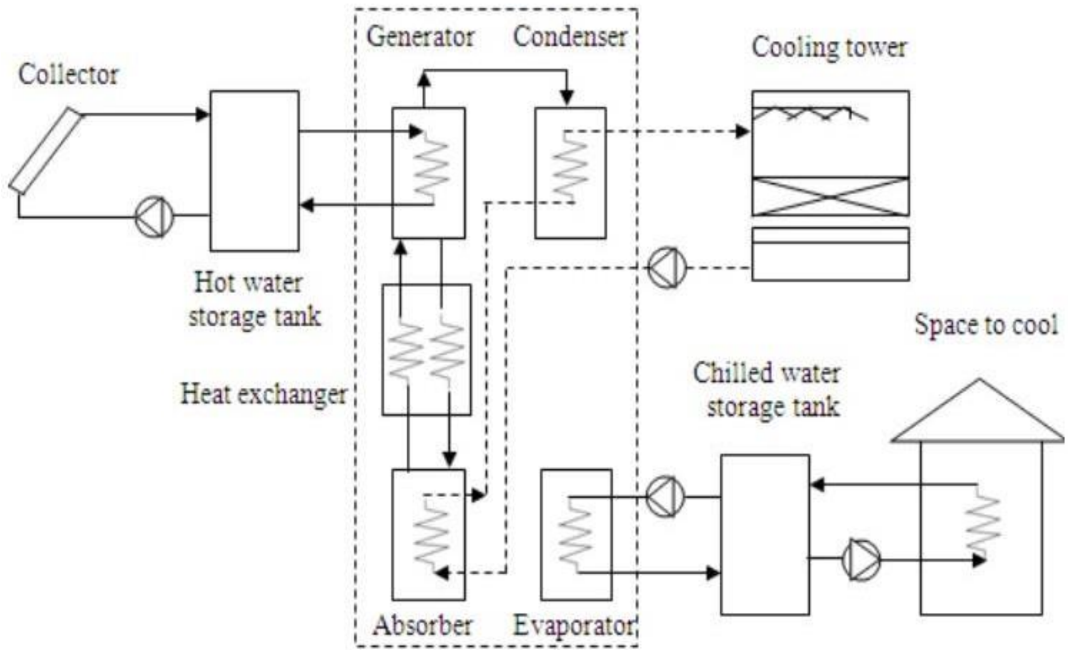

Figure 1-7: Mechanism of solar absorption cooling system

to generator while pump used to increase the pressure to refrigerant and then whole cycle repeats after generator and gives cooling effect (liquid or air stream) at evaporator. The whole working of a single effect SACS is illustrated in Figure 1-7 [26].

1.9.1.3 Process Description

Processes of absorption cooling cycle are represented with pressure-temperature plot as shown in Figure 1-8 [26].

1. Li-Br solution is fed into the generator with the help of pump and heat exchange in between is shown by Line 1-7. Point 7 represents the properties of weak solution at the heat exchanger outlet with same concentration throughout.
2. In generator, the weak solution is heated is shown by route 7-2. This is the sensible heating process as the solution is gaining the temperature only. While Line 2- 3 tells us about the latent heating process which causes boiling of water at the uniform condenser pressure Pc. Due to latent heating, the weak solution becomes strong solution with the evaporation of water vapor in the generator.
3. Line 3-8 shows that strong solution returned from generator. Heat exchanger between absorber and generator utilizes the thermal energy of Li-Br strong solution for heating of the weak solution from the absorber prior to entering into the generator.
4. In the evaporator, water refrigerant evaporated at low pressure after absorbing heat from the space to be cooled. This evaporated refrigerant is then attracted by the absorber is indicated by line 8-4-1.
5. The cooling tower is used as the cold reservoir for condenser where heat is rejected by condenser and is represented by line 2-5. The condenser's pressure (P_c) remains the same at this process.
6. The water flows into the evaporator from condenser is represented by line 5-6.
7. In Evaporator, water evaporates at low pressure by consuming heat of the conditioned space, represented by line 6-1, and then water vapor again absorbed due to attraction of lithium bromide in the absorber and completes one cycle.

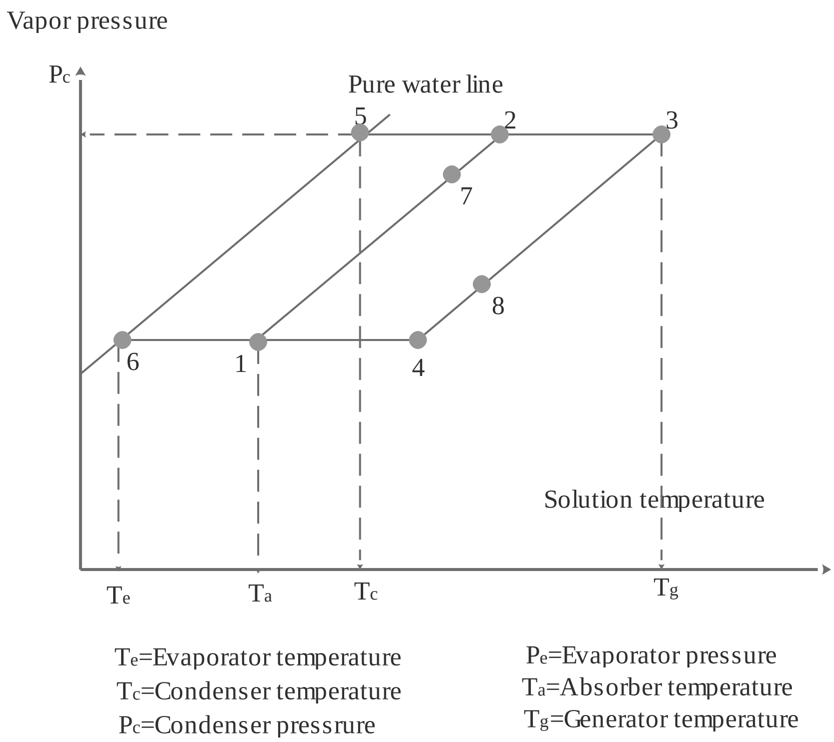

Figure 1-8. Processes of single effect absorption air-conditioning system [27]

1.9.2 Adsorption Cooling System

Adsorption process is divided into two types: Physical and Chemical.

1.9.2.1 Physical Adsorption Cooling

Many kinds of adsorbent materials can be used in the process of physical adsorption. Adsorbents that are most commonly used are activated carbon, alumina, silica gel and zeolite. These adsorbents are very soft and have ratios of surface to volume in the range of hundreds. They have the property of catching and holding the selective refrigerants. By heating we can regenerate refrigerants after which they made saturated [16]. Almost 4-7 kilogram of ice can be produced daily by using specified areas of solar collector by the use of solar adsorption system [27]. In these systems, space cooling can be provided by the use of Silica gel-water pair [28]. They have much lower refrigeration power density than absorption chiller and are heavy and expensive [29].

1.9.2.2 Chemical Adsorption Cooling

There is a solid chemical bonding between adsorbent and adsorbate in this cooling system. It is difficult to regenerate the adsorbent and needs a lot of energy than physical adsorption process. In the application of solar cooling system, common adsorbents material is calcium chloride ($CaCl_2$), which adsorbs water (H_2O) and ammonia (NH_3) to yield $CaCl_2.6H_2O$ and $CaCl_2.8NH_3$ respectively [30].

1.9.3 Desiccant Cooling

Desiccant cooling is used in air dehumidification. Both liquid and solid desiccants are available. Usually all sorbents can be used as desiccants which can absorb water like lithium bromide (Li-Br), activated alumina, lithium chloride (Li-Cl) and silica gel. The liquid desiccant flows between absorber and generator same as in absorption cooling system. The difference between desiccant cooling and absorption cooling is that, in desiccant cooling, the equilibrium temperature is measured by water's partial pressure in the moist air in which the solution is revealed, while in absorption system, equilibrium temperature is the function of total pressure [16].

1.9.4 Ejector Cooling

This system is used to keep cooling comfort in large buildings and trains [31]. The cooling system is not so complicated but due to small COP value, these systems are considered as less attractive option when compared with thermally-operated cooling technologies [32].

1.10 Research Problem

As the performance of the Absorption chiller is mainly affected by the inlet temperature to the generator unit of the chiller. Lack of research related to various flow schemes of heat transfer fluid (HTF) between components of SACS is perceptible and still there have more options of different flow schemes for heat transfer fluid to enhance the system's performance.

1.11 Scope of the Study

A cooling system based on solar absorption cooling cycle is designed, modelled, and simulated. All possible flow schemes of heat transfer fluid are modelled and simulated, and an appropriate flow scheme is chosen which gives the best performance for the solar absorption cooling system. Two types of solar thermal collectors are used to receive the solar energy and stores in a thermally stratified storage tank in the form of hot water. Simulation results of three flow schemes S-1, S-2 and S-3 are compared in terms of saving of primary energy. The best collector slope for evacuated and flat plate collector is decided based on seasonal solar fraction. The hourly cooling load profile of educational building was obtained by using type686 component in TRNSYS. The load is changing throughout the season by specially designed TRNSYS component for varying loads to get more realistic trend. TRNSYS is a very useful and most widely used simulation tool for predicting performance of many renewable energy systems [33]. Components from the TRNSYS v17 library, which are used in solar absorption cooling system are chosen for the modeling of the system. Parameters are designed for two types of solar collectors, FPC and ETC. Weather data of major cities of Pakistan are compared and city of Peshawar is selected on the basis of highest monthly averaged solar radiations. It is revealed that evacuated tube collectors with S-3 are more feasible for SAC systems. This study will also determine optimum parameters like collector tilt, area, and storage size for achieving 50% primary energy saving in comparison with system of vapor compression cycle. Due to huge capital cost, the study is limited to simulation modelling.

1.12 Research Objectives

The overall main objective of the current study is the design and modelling of solar absorption cooling system integrated with thermal storage for cooling of educational building located in the city of Peshawar-Pakistan. The study will explicitly aim to achieve the following goals:

i. Obtain an optimized TRNSYS model of solar absorption cooling system by replacing an installed conventional compression chiller (Dunham-Bush manufacturer) of 108 kW cooling capacity having COP value 3.

ii. Performance analysis of the SACS with the comparison of all possible flow schemes of heat transfer fluid flow between storage tank and chiller loop.

Select best flow scheme of heat transfer fluid which can be employed to achieve 50% primary energy saving than conventional cooling system.

1.13 Summary

Currently, there is a drastic increase in the energy consumption all over the world. According to researcher's community, this trend will continue in the future. Buildings are the main consumer of energy spending about 40% of the total energy in the developed regions and more than 50% of this total energy utilizes only for maintaining thermal comfort inside the building. Vapor compression cooling systems are the most common space cooling technology used for meeting the cooling load of buildings during summer season. They are also named as conventional cooling systems in all over the world. Their main drawback is that, these systems emits refrigerants containing carbon compounds which are harmful for our environment and causing global warming and ozone layer depletion issues. Furthermore, they consume huge amount of energy. Keeping in view these concerns, the deployment of solar absorption cooling systems is getting attention of the researchers. The use of solar energy for cooling applications is also an attractive option because we need more cooling when we have high solar radiations and vice versa. So, we can reduce much electrical consumption of the buildings by employing solar cooling technologies especially in summer season. From the last two decades, Pakistan is facing issue of electricity load shedding and the problem becomes more serious during summer season which also affects the economy of the country. The region of Pakistan is geographically ideal for exploiting solar absorption cooling systems because warm weather stays for longer time of the year. According to weather data reports there are more than 300 full sunny days with more than 2000 kWh/m^2 global radiations in most parts of the Pakistan. This means that Pakistan is having a huge potential for exploitation of solar energy and can be used for operating solar absorption cooling systems.

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Chapter 2: LITERATURE REVIEW

The enactment of solar based cooling technology is an appealing option because the cooling demand coincides with peak solar incident energy. It is also essential to promote its commercialization [34] to overcome the intense electrical load of the buildings due to peak cooling demand in summer because, in many countries, electricity demand reaches to its peak level due to space cooling requirement in the day time of summer season. The solar based cooling technology has been used for numerous years, but their commercial availability is still uncommon. [35]. Until 2007, there were 81 solar assisted cooling systems totally installed on a large scale worldwide. Many installed systems were remained in non-operational condition. Out of those 81 solar cooling systems, 73 were installed in Europe, 1 in America and 6 were located in Asia (mostly in china). 60% installed systems were providing space cooling services to office buildings, for factories the percentage of installed cooling systems was 10, for educational institutes and laboratories 15% installed systems were reported and 6% installed systems were dedicated to hotel buildings, sport centers and canteens etc. Absorption chillers used in 56 installations, 10 installed systems use adsorption cooling mechanism and 17 solar cooling system were operated by desiccant evaporative cooling principal. Out of 17 desiccant evaporative cooling systems, only 2 were uses liquid regenerator as desiccant evaporative cooling liquid. 9 MW was the total cooling capacity of those solar assisted cooling systems out of which 31% of total cooling capacity were installed in Spain, 18% and 12% in Germany and Greece respectively [36]. As the number of installed solar assisted cooling system is increasing globally but still, they could not become a preferable choice. Current challenge is its high capital cost that is still also the problem with other renewable technologies. The initial purchase of solar cooling system (vapor absorption-based) is still more than conventional cooling systems (vapor compression-based) [37]. However, compensation of high initial cost by environmental and operating fuel cost is the strong argument that makes this technology an appealing choice in the current global scenario.

Absorption cooling system is the most commonly applied technology among other cooling systems [16]. By using TRNSYS, a number of studies on absorption cooling system has been published. The core purpose of all studies is to improve the performance of the system in terms of solar fraction, efficiency etc. As, the experimental study of solar absorption cooling system requires huge cost, so researchers usually use simulation tools for the prediction of long-term performance of solar absorption cooling system. Tsoutsos et al. [38] designed a simulation model of cooling and heating system for a hospital in Crete using TRNSYS, where he focused on improving the solar fraction of the system. The solar air conditioning system comprises of an absorption chiller of 70 kW with a working fluid of LiBr–H_2O, and a compression chiller of 50 kW was used in addition in order to provide required energy for cooling. The system includes an auxiliary pre-heater (fossil fuel) of 87 kW. Different parameters like collector area, size of storage tank, collector slope, back up heater, cooling tower and nominal capacity of absorption chiller were optimized and have achieved peak solar fraction cooling of 74.23%. Shirazi et al. [39] developed simulation model of single, double and triple effect absorption chiller for both heating and cooling systems. Their results suggest that for fraction of normal incidence solar radiation less than 50%, single and double effect chillers with ETCs require less collector area than parabolic and linear Fresnel collectors. Double effect chillers with evacuated tube collectors resulted in better thermal and economic performance for various climatic conditions. A. Lecuona et al. [40] provided explicit relation for the optimizing temperature of hot water which is supplied to the generator of absorption cooling system. With the rise of hot water temperature, COP of the system increases and the efficiency decreases. There should be an optimized temperature at which higher overall efficiency of the system can be achieved. Bellos et al. [41] worked on examination of two working pairs (LiCl-H_2O and LiBr-H_2O) in a solar absorption chiller. The examined single effect absorption chiller is driven by flat plate collectors. The system is analyzed energetically for 3 ambient temperatures and concluded that LiCl-H_2O performs better than LiBr-H_2O in all the examined cases. He also observed that the optimum operating temperature is lower for the case of pair LiCl-H_2O. M. Mazloumi et al. [42] simulated solar absorption system located in Iran. The system uses Li-Br single effect absorption chiller located in Ahwaz (Iran). In July, peak cooling demand of 5 TR met by using parabolic trough collectors to provide the thermal comfort in a house. Heat transfer fluid was water which is fed to the collector of 57.6 m^2 directly. Mass flow rate has great impact on the optimal efficiency of storage. Optimal efficiency of the hot water storage tank was mainly affected by mass flow rate. Y. Agrouaz et al. [43] presented a detailed performance analysis of a solar air-conditioning system operating under Moroccan climate. His simulation results showed that the climatic conditions significantly influence the performance of the solar cooling system. During peak-load months, Errachidia presented the highest performance. The corresponding COP reached a value of 0.3 while the solar fraction was 45%. Assilzadeh et al. [44] used evacuated tube collector (ETC) and absorption chiller for the simulation in TRNSYS for the climate of Malaysia. He concluded that ETCs tilted at 20° and 35 m^2 area of solar collectors fulfilled the requirement of 1-ton cooling capacity along with the volume of 0.8 m^3 of a hot storage tank. Hamideh Sheikhani et al. [45] summarized his research as the high-temperature SHC combined with multiple-effect absorption with PTCs may be considered as one of the most beneficial renewable energy technologies for cooling purpose in public buildings (such as universities, hospitals and educational buildings). However, there are some limitations of using PTCs such as commercial unavailability or high initial cost. On the other hand, the single effect absorption cooling system using LiBr/water as working couple can be properly selected for domestic solar purposes. Applying such system or a high temperature SHC system integrated with FPCs and ETCs can be more reliable and economical.

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Chapter 3: RESEARCH METHODOLOGY

This chapter contains complete description of the research work. All system components and their arrangement are briefly explained:

3.1 Methodology flow chart:

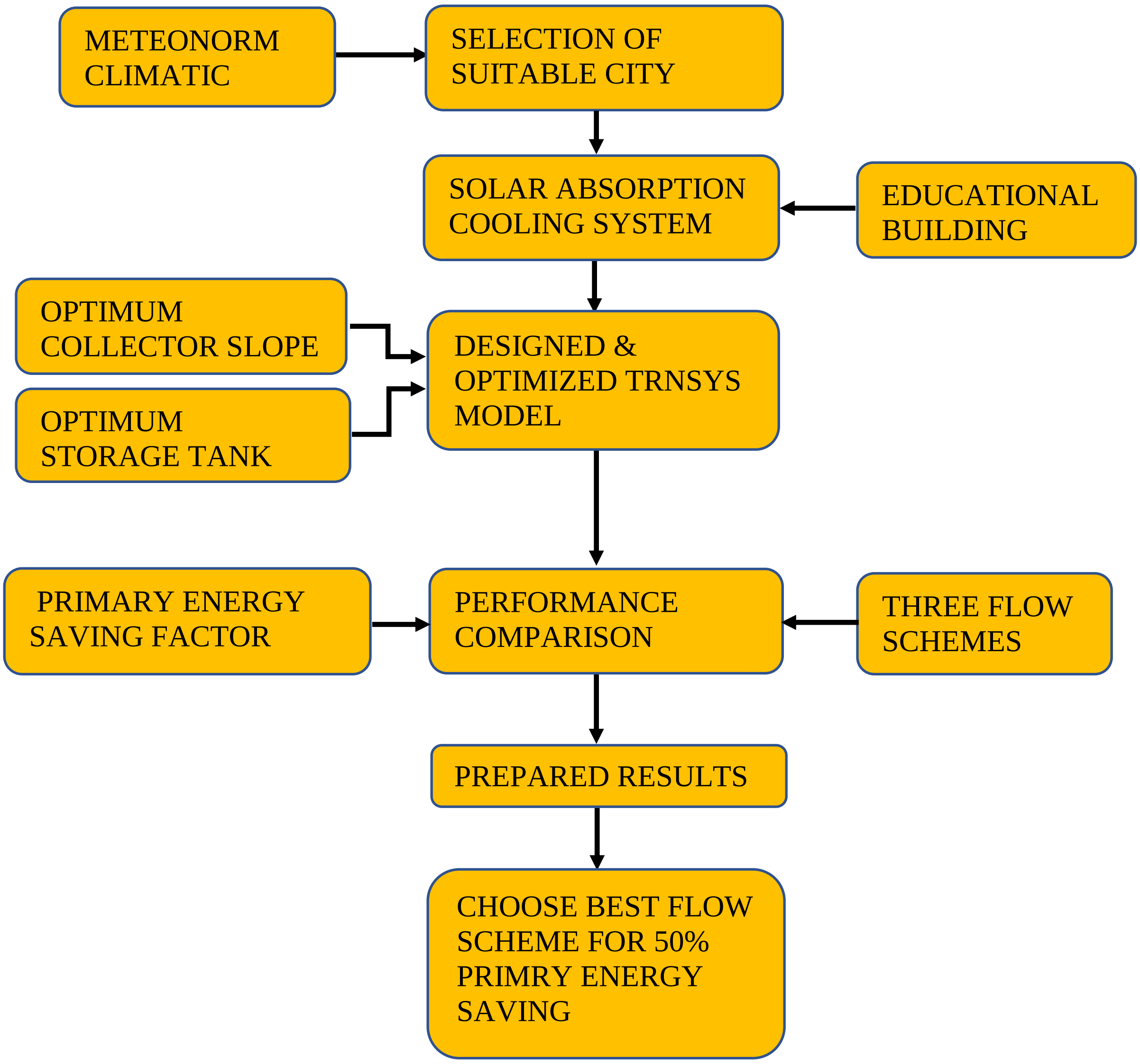


Figure 3-1. Flow chart for research Methodology

Complete flow chart representation of the study is shown in Figure 3-6 and all steps that involved in the design and modelling of the system are explained as follows:

1. The weather data of major cities (Lahore, Multan, Peshawar, Islamabad and Karachi) of

Pakistan is compared and the city of Peshawar is selected for current simulation study because of its highest global irradiance among other aforementioned cities as presented by meteorological department for typical meteorological year (TMY) in Figure 3-1. The TMY file includes hourly detail of ambient temperature, wind speed, direct & diffused solar radiations, global radiations, and relative humidity.

2. A solar absorption cooling system is designed for an Educational building with three proposed schemes of heat transfer fluid flow between storage tank-chiller loop of the cooling system. Type682 component in the TRNSYS v17 library is used to specify the hourly cooling load profile of an educational building. Cooling system is designed for the peak cooling demand of 108 kW which is the cooling capacity of one of the installed conventional compressional chillers (Dunham-Bush manufacturer) in educational building. The distinct feature of educational building is its operating schedule from 9 a.m. to 5 p.m. and least cooling load on weekend. The component type682 uses a user defined positive peak load and evaluates the subsequent conditions of outlet flow of fluid. Boiling and freezing effects of the heat transfer fluid are neglected in this component. Generally, cooling or heating load of the building are calculated by tedious calculations or by the use of different others simulation programs but in this study we use type682 component to get almost same trend like real building loads as our principal purpose is to simulate the effect of cooling load on the designed cooling system.
3. Components from the TRNSYS v17 are selected and logically connected to each other just like real connected components in cooling systems. Type107 in TRNSYS library serves the purpose of hot water fired absorption chiller. Other components are solar collectors, pumps, controllers, thermally stratified storage tank for hot water, auxiliary heater, cooling load generator component, synthetic building component which handles the chilled water stream, output components like printers and integrators. Optimization of the system is done for best slope of the collectors at maximum solar fraction and for optimum thermal storage size.
4. Temperature of 109℃ is chosen as minimum driving temperature for the chiller operation because the sample catalogue data file available with TRNSYS v17 have the inlet hot water temperature range is 108℃-116℃. Otherwise, the calculated results would not be correctly justified by TRNSYS. Simulations are run by using two types of solar collector that are commercially available in the market i.e. flat plate collector (type1b) and evacuated tube collector (type71) for all the possible flow schemes of heat transfer fluid in the system and their performance is compared by using different factors like solar fraction, efficiency of collector and primary energy saving.
5. After modelling a complete TRNSYS model and assign the designed parameters to each components of the cooling system, simulations are run and get the output files in excel

format and prepared the required results.

6. Performance of cooling system with all three schemes were analyzed and 50% Primary Energy saving was the main factor for comparison of performance of all three proposed flow schemes of the study.

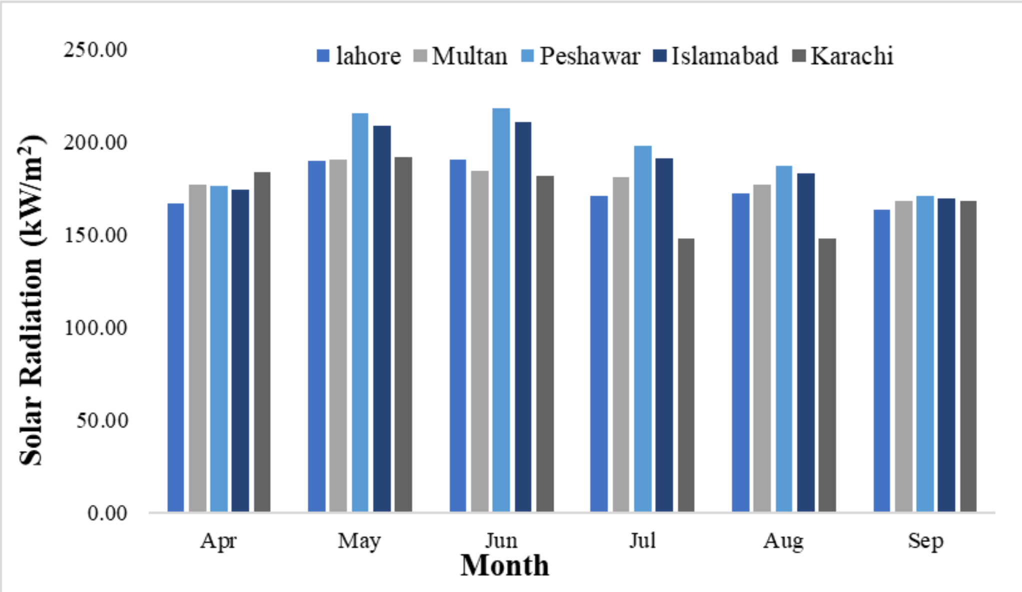


Figure 3-2: Monthly available radiations in Pakistan

3.2 Introduction to TRNSYS

Now a days, most of the researchers are using TRNSYS (TRaNsient SYstem Simulation) tool for predicting the dynamic behavior of many engineering designs. Multi-zone buildings with thermal properties can also be modelled by using this tool. Many people are using this tool to validate lot of new energy concepts like usage of hot water systems in domestic and commercial buildings, designs of zero energy buildings, behavior of occupants in the building and their schedule of activities, cooling and heating demands of buildings etc. Also, many systems of renewable energy like, solar photovoltaic, thermal, hydrogen systems and wind energy etc. can be designed and modelled with TRNSYS.

There are lot of components with their complete design parameters are available in TRNSYS library. However, third party developers and any user can certainly add his own desired components model because of the dynamic link library (DLL) based structure in TRNSYS. For adding new component, common programming languages can be used like FORTRAN, C++, PASCAL, C etc.

Usually, while setting up a TRNSYS project, components in the main simulation studio are graphically inter-connected. All TRNSYS components have their own mathematical model with different conditions.

Components in the TRNSYS library are named as Types number e.g. Type4 is thermal storage tank while Type1 is solar thermal collector etc. The model for multi- zone building is named as Type56. Following applications are included in these groups:

i. HVAC systems and zero energy buildings with progressive design features
ii. Solar thermal and PV systems
iii. Renewable systems
iv. Fuel cells and cogeneration

TRNSYS has many collections of programs. However, only simulation studio program is used in this thesis. TRNSYS is relatively a better tool whose formulation is based on sound-conventional empirical and analytical correlations. In the planning phase of any project it is difficult to predict the performance for long span of time, so TRNSYS is helpful to guess the performance because of the dynamic simulations system which are very close to real system calculations [33].

The idea of the present study is to suggest a SACS for an educational building with the help of TRNSYS tool to model and simulate a variety of designs which were not analyzed and assessed before.

3.3 Description of the Proposed Cooling System

The schematic of the main components used in solar absorption cooling system for educational building is shown in Figure 3-2. Heat transfer fluid (water in current study) flows between different components of the cooling system to exchange its heat during flow. For analyzing the controlled flow of HTF, three loops are presented and analyzed between the components of the cooling system in the current study i.e.

- Collector-tank loop
- Storage-chiller loop
- Chiller-building loop

These loops of HTF are indicated by yellow, red and green colors respectively in Figure 3-3. The current study focused on the loop in which HTF feeding the generator of the chiller (i.e. Storage-chiller loop). To the best of our knowledge, HTF can be circulated between the components of storage-chiller loop by 3 different ways which are known as three proposed flow schemes of the current study.

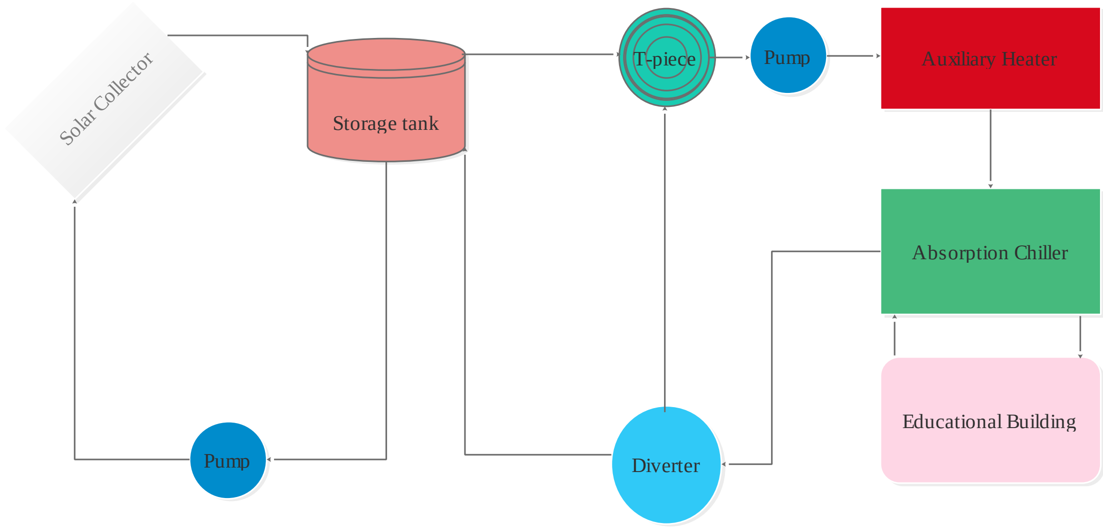


Figure 3-3: Solar absorption cooling system

3.3.1 Flow Scheme-1

In flow scheme 1, best recognized heat transfer fluid i.e. water circulates between solar collector and thermal storage tank by receiving solar radiations via solar collector and exchanges heat with the fluid in the tank. A pump is used to set the required mass flow rate of solar collector-tank loop which can be turned off by the controller signal when the temperature of the fluid inlet to the collector exceeds the outlet temperature of the collector. From the storage tank, water flows towards auxiliary heater. Boiler is used as auxiliary heating device that turns ON when fluid coming from the storage tank have temperature less than 110℃ which is the required temperature for the chiller operation in this study. ON or OFF condition of boiler is controlled by using a thermostat controller component which continuously monitors the temperature of fluid inlet to the boiler. Absorption chiller yield chilled water with its mechanism explained in first chapter and hot water exits after transporting heat to the generator of the chiller, returned to the storage tank and continues the cycle by the flow of fluid from storage tank to solar collector. In Figure 3-2, the removal of tee and diverter component and direct links (of storage to pump 2 and chiller to storage) in storage-chiller loop shows conventional flow pattern of heat transfer fluid which is named as scheme 1 of the current study.

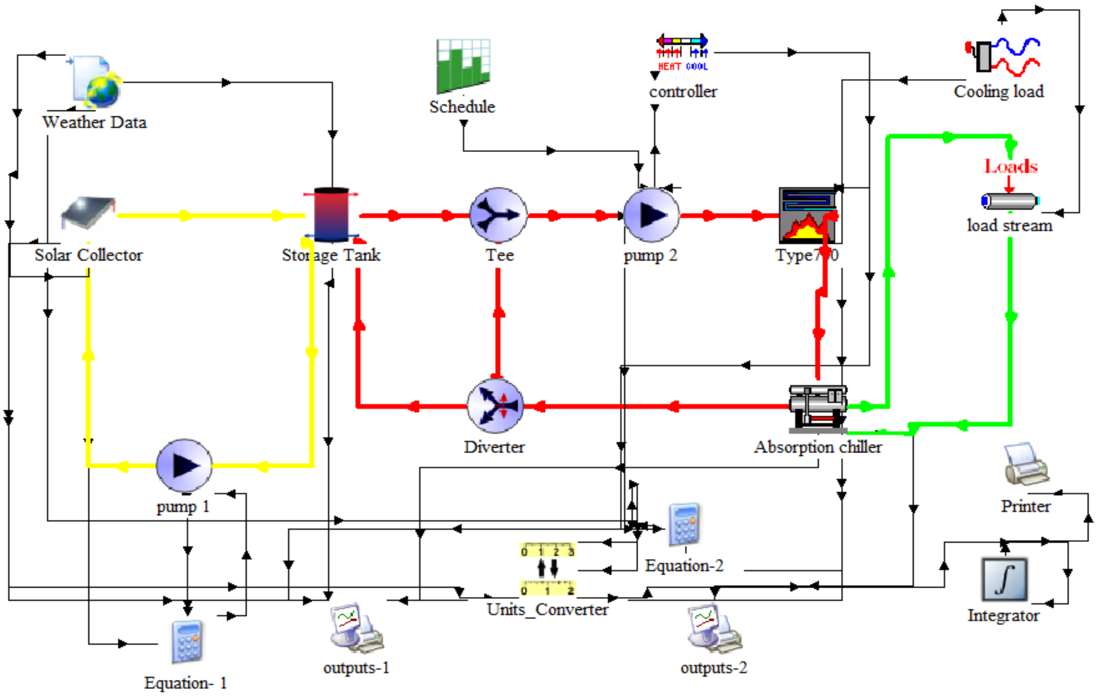


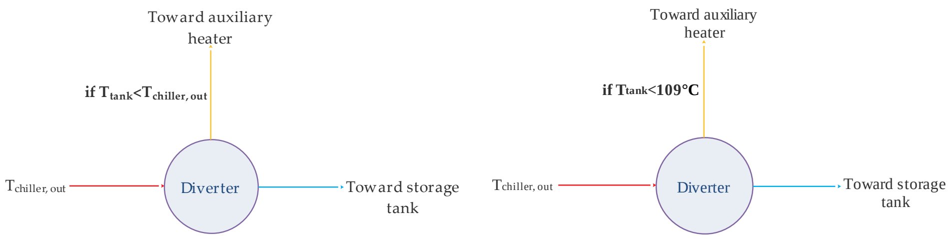
Figure 3-4: TRNSYS Model of SACS for modified flow schemes

3.3.2 Flow Scheme-2

In this scheme, heat transfer fluid, on returning from the absorption chiller, will flow towards the auxiliary heater and feed the chiller rather than directly towards the tank if its temperature is greater than the outlet temperature of the storage tank. A flow diverter is used for diversion of chiller's out hot water to auxiliary heater and a tee piece component to pass the hot water either from storage tank or from diverter. A tee piece component in TRNSYS library is used to flow either storage tank outlet fluid or diverted fluid to the boiler. This diverted fluid is then again examined by thermostat controller component same as scheme-1 and decides the state of boiler as ON or OFF. There is no change in the solar collector-tank loop of water as in the flow scheme-1 shown in Figure 3-4(a).

3.3.3 Flow Scheme-3

In flow scheme-3, diverter and tee piece components are used for diversion and required fluid stream flow respectively like scheme-2 as shown in Figure 3-4 (b). The difference between the flow scheme-2 and flow scheme-3 is the range of comparison temperature for diverter to perform its function. Unlike scheme-2, the diversion of the hot water returning from chiller only stops when the outlet temperature of storage unit becomes equal to or greater than 109℃, otherwise diverter remains ON and solar collector-tank loop remain in isolated circulation of fluid.



(a) Scheme 2 (b) Scheme 3

Figure 3-5: Mechanism of modified flow schemes

3.4 Modelling in TRNSYS

TRNSYS is employed for the modelling and simulating all schemes of the heat transfer fluid flow between storage tank to chiller in SACS. The monthly averaged solar radiations of hot climate cities of Pakistan (Lahore, Islamabad, Multan, Peshawar and Karachi) are compared as shown in Figure 3-1 and a cooling system in an educational building of Peshawar (highest radiations city of Pakistan) is chosen for modelling of solar absorption cooling system, having 108 kW peak cooling load during summer season and daily operating schedule from 9 a.m. to 5 p.m. TRNSYS model for modified flow schemes exhibited in Figure 3-3.

Thick red lines shown in Figure 3-3 depict hot water loop from storage to chiller unit, yellow lines depict the collector-tank loop while green lines indicated loop of chilled water from chiller to building. Scheme 2 and Scheme 3 are having same components except temperature range for diverter. Meteonorm weather data available with the TRNSYS v17 is used in the current model of SACS.

3.4.1 Assumptions of the Study

To accomplish the objective of comparing performance of three proposed flow schemes of the study, following assumptions are considered with the design of cooling system used:

- Heat losses from pipes are neglected.
- Energy balance technique is used for cooling tower specifications.
- Boiling effects of heat transfer fluid are ignored.
- Electricity consumption of pumps is not included as auxiliary energy consumption record.

Connections shown in Figure 3-3 are just logical connections and not represents pipes. For simplicity of the model, pipes or valves connections are not included in current model by which heat losses from pipes can be measured as it can be assumed that they are perfectly insulated. The second assumption of the study reflects theoretical calculations of cooling water energy by energy balance equation (equation 3.5) which tells us energy from absorber and condenser rejected by cooling water by keeping the temperature difference of 5℃ between inlet and outlet cooling water for design consideration in equation (3.4) of the current study and no separate TRNSYS component of cooling tower is used. The third assumption depicts the pressurized water system by which normal b oiling point of water can be varied [46], however, pressure calculations are not included in the study. All flow schemes of the current study use two flow pumps and would consumes considerable but same amount of electricity as the designed flow rates are same in all schemes, so that this electricity consumption of the pumps is not included as auxiliary energy consumption calculation.

3.5 Components of solar absorption cooling system

Detail of all the TRNSYS components used in the modelling of solar absorption cooling system are explained below:

3.5.1 Hot water fired single effect Absorption chiller

This is the most crucial component in designing of an absorption cooling system. Parameters are designed to meet the peak cooling load of 108 kW for educational building during summertime. TRNSYS uses certain equations as provided in the mathematical reference of the TRNSYS v17 by which we are able to design parameters of this component [47].

$$\dot{Q}_{hw} = \dot{m}_{hw} Cp_{hw} (T_{hw,in} - T_{hw,out}) \tag{3.1}$$

Where $\dot{Q}_{hw}$ denotes the energy rate for hot water that is used for operating the chiller, mass flow rate of hot water is denoted by $\dot{m}hw$, heat capacity of the fluid is Cphw, the standard value is taken as 10°C for the difference between hot water inlet and outlet denoted by ($T_{hw, in}$ - $T_{hw, out}$) in the above equation. This equation is used for calculating the mass flow rate of hot water from storage tank to chiller loop. However, $\dot{Q}_{hw}$ is calculated by the following equation:

$$\dot{Q}_{hw} = \frac{Capacity_{rated}}{COP_{rated}} f_{DesignEnergyInput} \tag{3.2}$$

Where $Capacity_{rated}$ and COP_{rated} denotes the rated capacity and rated COP of the chiller respectively, $f_{DesignEnergyInput}$ denotes fraction of design energy input which is taken from performance data file for type107 component for designed input conditions. This equation calculates the energy rate that must be provided in order to perform the operation of the chiller. Mass flow rate of chilled water can be found by using following equation:

$$\dot{Q}_{chw} = \dot{m}_{chw} Cp_{chw} (T_{chw,in} - T_{chw,set}) \tag{3.3}$$

Where $\dot{Q}_{chw}$ denotes the energy rate which is required to remove from chilled water stream. Initially at designing phase, it is considered as cooling demand, Cp_{chw} is the heat capacity of chilled water while ($T_{chw, in}$-$T_{chw, set}$) is the difference between the chilled water inlet to the chiller and chilled water outlet from the chiller. $T_{chw, set}$ must be set point temperature of chiller which is 6.667□ in the current model. Mass flow rate of cooling water $\dot{m}_{cw}$ is derived from the following equation:

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

Cp_{cw} is heat capacity and ($T_{cw, out}$ - $T_{cw, in}$) is temperature difference of cooling water exiting and enter the chiller. However, $\dot{Q}_{cw}$ is calculated by using energy balance approach as:

$$\dot{Q}_{cw} = \dot{Q}_{hw} + \dot{Q}_{chw} + \dot{Q}_{aux} \tag{3.5}$$

Where $\dot{Q}_{aux}$ is the auxiliary energy consumed by the solution pump and refrigerant pump etc.

of the chiller.

3.5.2 Stratified Thermal Storage Tank

An energy balance approach is used to determine the temperature at each outlet of all nodes. A thermal storage tank type4a with uniform losses is used in current model. The default value of heat transfer coefficient is 0.83 $W/m^2 °C$ is used. This component of the solar absorption system minimizes intermittent nature of solar energy. By using stratification effects, hot water enters or leaves the storage tank at the top sides while the replacement fuel enters or leaves from the bottom sides of the storage tank. The following energy balance equation is used to calculate the temperature at each node:

If $g_i > 0$

If $g_i < 0$

$$
\begin{aligned}
M_i C_{pf} \frac{dT_i}{dt} &= \alpha_i \dot{m}_h C_{pf} (T_h - T_i) + \beta_i \dot{m}_L C_{pf} (T_L - T_i) + UA_i (T_{env} - T_i) \\
&+ \gamma_i (T_{i-1} - T_i) C_{pf} \\
&+ \gamma_i (T_i - T_{i+1}) C_{pf} \\
&+ \dot{Q}_i
\end{aligned} \tag{3.6}
$$

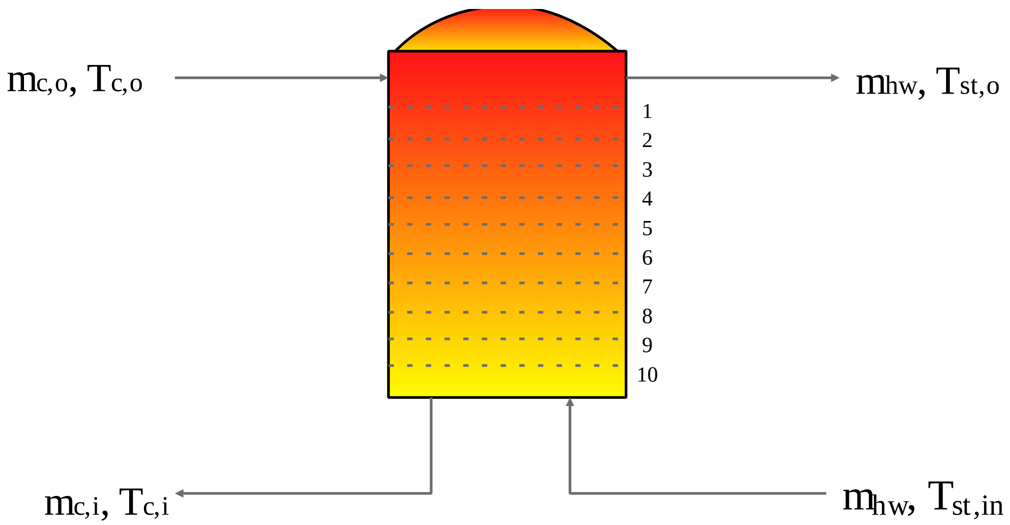


Figure 3-6: Thermally stratified storage tank

3.5.3 Weather Data Component

From TRNSYS v17 library, type 15-6 component is used which processes climatic data of Meteonorm in standard format. Climatic conditions of Lahore, Multan, Peshawar & Karachi are available with this component while weather of Islamabad is used from the generated file from Meteonorm software.

3.5.4 Outputs

We obtained excel files of simulation results in TRNSYS by using different output components like type25b for printing unformatted values of user supplied units, type

65b for user supplied unit to outputs and type 65c for without units. The forcing function 14h is used to operate boiler supply pump only during schedule time of educational building.

3.5.5 Pumps, Tee, and Diverter

There are two pumps, one Tee and one diverter used in the current model shown in Figure 3-3**. These hydronic components used for the controlled flow of fluid between the other system's components. Pump in the solar collector-tank loop used for desired mass flow rate in the loop and operated by the signal generated by an equation component as required. The second pump is used in the tank-chiller loop before auxiliary unit for the controlled flow of fluid to make sure that fluid will only flow during the operating hour of the educational building. A tee piece is used for allowing the one fluid stream at a time either from storage tank or diverter depending upon the conditions specified in scheme-2 and scheme-3. A diverter is used to divert the hot water flow when controller gives the signal for diversion in scheme-2 and scheme-3. There is no tee and diverter used in scheme 1.**

Table 3-1: Parameters used in TRNSYS model simulation

DESIGNED PARAMETERS	**VALUES**
Type 700 (Auxiliary Boiler)	
Time step (minutes)	7.5
Boiler Efficiency (%)	85
Combustion Efficiency (%)	78
Type 107 (Absorption Chiller)	
Specific heat of heat transfer fluid ($kJ.kg^{-1} K^{-1}$)	4.19
Flow rate of Hot water from tank to Chiller ($kg.h^{-1}$)	17507
Flow rate of Chilled Water from Chiller to Building ($kg.h^{-1}$)	16710
Flow rate of Cooling Water for condenser and absorber heat rejection ($kg.h^{-1}$)	54527
Minimum Temperature for chiller operation (℃)	109
Cooling water inlet temperature (℃)	29.4
Chiller Water set point (℃)	6.667
Peak Cooling Demand (kW)	108
Type 4a (Thermal Storage Tank)	
loss Coefficient for tank ($W.m^{-2} K^{-1}$)	0.83
Number of nodes	10
Storage water density ($kg.m^{-3}$)	1000

3.5.6 Synthetic building and Flow stream Loads

Flow stream loads component (type682) act as a load handler which consumes chilled fluid stream in TRNSYS model and returned fluid with little higher temperature depending upon the load. This outlet fluid temperature from type682 component is the input for chiller unit as chilled water inlet temperature. TRNSYS v17 have an updated option to keep the maximum value of this temperature as user defined input for convenient designing purposes. Synthetic building component named as type686 in the TRNSYS v17 library is used to generate the heating and cooling load imposed on the fluid stream. Nowadays, mostly researchers are predicting the hourly loads of the buildings by the use of this component of synthetic building as.

$$\text{Load}=\text{designload}*X_{day}*X_{hour}*X_{noise,hour}*X_{noise,day} \tag{3.7}$$

Where *design load* is the peak load input from the user, X_{day} is the function for seasonal variations in the load, X_{hour} is the hourly variations in the load, $X_{noise,\ hour}$ is the hourly noise in the calculated load while $X_{noise,\ day}$ is the daily noise in the calculated load. The user just needs to put the designed peak load and this component returns hourly load trending just like the load of real building as shown in Figure 4-2*:* . This component also allows the user to set the seasonal and daily variations in the hourly generated load by selecting different offsets and multipliers that includes daily and seasonal variations in the load as:

$$X_{day}=a_{season}+b_{season}\sin\left(180\left(\frac{\text{time-StartOfSeason}}{\text{EndOfSeason-StartOfSeason}}\right)\right) \tag{3.8}$$

Where a_{season} is the offset for the season, b_{season} is multiplier for season, time is the current simulation time step, *StartOfSeason* is the hour of the year at which specified season begins and *EndOfSeason* is the hour of the year at which specified season ends.

3.5.7 Solar Thermal Collector

There are two types of solar collectors whose performance is compared in terms of different performance factors. The efficiency η of these collectors can be find out by the following relation:

$$\eta=a_0-a_1\frac{(T_i-T_a)}{I_T}-a_2\frac{(T_i-T_a)^2}{I_T} \tag{3.9}$$

Where a_0 denotes the optical efficiency or intercept efficiency of the collectors and a_1 and a_2 indicates the negative of the first and second order coefficient, respectively. Global radiations falling on the surface of the collectors are represented as I_T, the temperature receives by the collector at its inlet is T_i and the temperature from the ambient is denoted by T_a. All default values of flat plate collector (type 1b) parameters [47], were used while values for the parameters of evacuated tube collector (type71) (including their IAMs and efficiency curve) were specified by the manufacturer [48], were used in the current study. Coefficient and parameters values of both collectors are given in table 1:

Table 3-2: Solar Collector Specifications

Type of Collector	**ETC**	**FPC**
a_0	0.804	0.8
a_1 (kJ h^{-1} m^{-2} K^{-1})	1.56	13
a_2 (kJ h^{-1} m^{-2} K^{-2})	0.0054	0.05
Testing Flow rate (kg/h.m^2)	100	40

IAM Transversal at 50º	1	1-0.2(1/cosθ-1)
IAM Longitudinal 50º	0.93	

3.5.8 Auxiliary heater

A TRNSYS component type700 is used as the auxiliary heating device in the model. A thermostat controller is used to control the boiler function ON or OFF when required. The fluid is heated by this component by consuming energy until its temperature reaches 109℃. If the temperature of incoming fluid is equal to or more than 109□ then this component turned off and allows the fluid to flow without consuming any energy. This component used following equation to evaluate the required auxiliary energy $\dot{Q}_{need}$:

$$\dot{Q}_{need} = \dot{m}_f C_{pf} (T_{set} - T_{in}) \tag{3.10}$$

Where $\dot{m}_f$ is the mass flow rate of the incoming fluid to the boiler, Cp_f is the heat capacity and (T_{set}-T_{in}) is the difference between set point temperature and inlet to the boiler. The quantity of fuel consumption can be found by using this component with the efficiency of the boiler as:

$$\dot{Q}_{fuel} = \frac{\dot{Q}_{need}}{\eta_{boiler}} \tag{3.11}$$

3.6 Performance factors

For evaluating the performance of the designed cooling system, three performance factors are used in this study:

3.6.1 Efficiency

Collector's efficiency η for each month and whole season is estimated by the following relation:

$$\eta = \frac{\dot{Q}_{solar}}{A_c . I} \tag{3.12}$$

Where $\dot{Q}_{solar}$ is the energy rate collected from sun, area of the collector is A_c and I denotes all the radiations incident on surface of the collector. Integrated values are used for tracing monthly or seasonal performance evaluation.

3.6.2 Solar Fraction

Solar fraction (SF) is used as the main criteria for comparison of solar based systems and used in most of the research related to the utilization of solar energy. The equation for this performance factor is:

$$SF = \frac{\dot{Q}_{solar}}{\dot{Q}_{solar} + \dot{Q}_{boiler,aux}} \tag{3.13}$$

Where $\dot{Q}_{boiler,\ aux}$ is the rate of additional heat energy that we need to supply by auxiliary heater to run the cooling system and $\dot{Q}_{solar}$ is the same term that used in efficiency equation. This factor is calculated in post-simulation study.

3.6.3 Saving in Primary Energy

For the comparison of solar based cooling system with conventional compression cooling system in terms of primary energy saving (PES), a general equation is used [49]:

$$PES=1-\left[\frac{\frac{\int \dot{Q}_{boiler}}{\varepsilon_{heat}}}{\frac{\int \dot{Q}_{cooling,ref}}{COP_{ref}.\varepsilon_{elec}}}\right] \tag{3.14}$$

Where $\dot{Q}_{boiler}$ is the auxiliary energy from boiler whose value is provided by the component at its output, the efficiency of the boiler is ε_{heat} which is 78% in the current study and is defined as “number of units of heat for doing useful work per unit of same type of primary energy from fossils”, $Q_{cooling, ref}$ is the same cooling effect realized in terms of cold energy if using the system with vapor compressional cooling cycle, COP_{ref} is the term used for output of compressional cooling systems having value of 3 and $\square_{elec}$ representing the conventional efficiency of a thermal power generating station having 40% in the current model and is defined as “number of units of electricity consumed per unit of same type of primary energy from fossils”. Thus, the above relation can simply define as the ratio of consumption of auxiliary primary energy consumed in solar cooling system to the consumption of primary energy in vapor compression cooling system operating for meeting the equal cooling demand.

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Chapter 4: RESULTS AND DISCUSSION

Simulations are run using TRNSYS tool to get the optimized factors for performance evaluation of the system for the time span of summer season (April to September). Ambient temperature for all months is plotted in Figure 4-1. Only few numbers of Rainy days are being recorded in Peshawar during summer season. The city of Peshawar is considered as a steppe climate region where June is the hottest month with average temperature of 40.4℃. Subsequent sections contain simulation results and their detailed discussion.

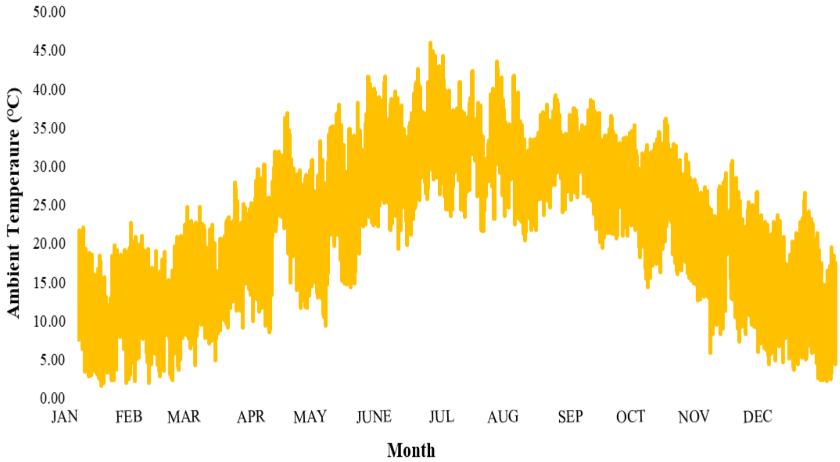


Figure 4-1: Ambient temperature of Peshawar

4.1 Cooling Load

Cooling demand variation generated by using TRNSYS component type 686 as shown in Figure 4-2. The variations are set by using seasonal and daily offset and multipliers as in equation (3.7) to get the trend similar to the real building loads, so time consuming calculations and the use of other simulation tools for the determination of cooling load are avoided in this study. Parameters of type686 are set to acquire the designed peak cooling demand of 108 kW over the prescribed simulation time step.

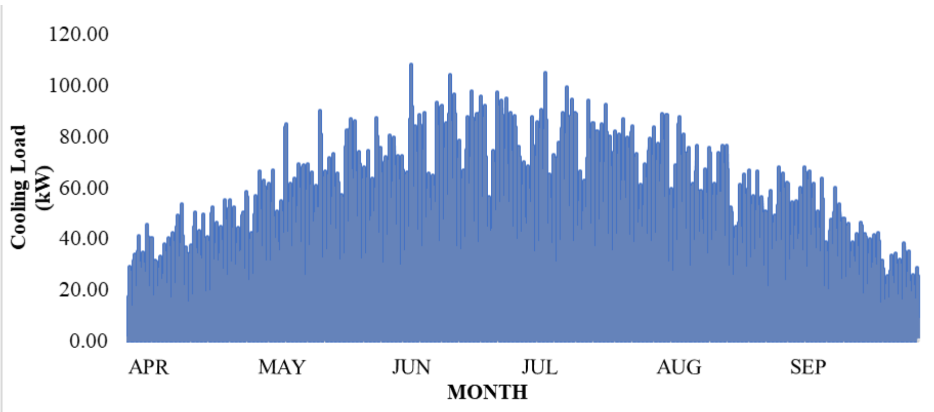


Figure 4-2: Cooling Load of an Educational building

4.2 Optimum slope of collector

The best tilt angle for the collectors based on maximum solar fraction is shown in Figure 4-

3. In our model, the greatest seasonal SF is acquired for the area at which 50% PES is achieved with an optimum collector slope of approximately 9° for ETC and 15° for FPC as shown in Figure 4-3.

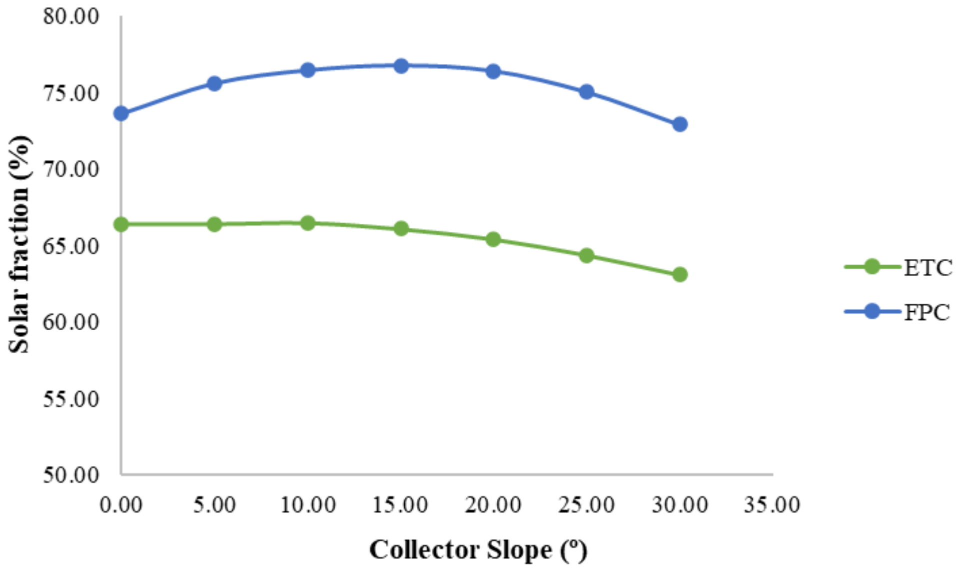


Figure 4-3: Solar fraction of ETC and FPC

4.3 Optimum size of storage tank

Storage sizes are plotted for collector areas of FPC in Figure 4-4 at which flow schemes are realizing 50% seasonal primary energy savings. It is essential to note that the legends in Figure 4-4 & Figure 4-5 are representing the area of collector in m^2 used for one of the flow schemes of the study. For instance, legend 510 S1 means that 510 m^2 collector area used for flow scheme 1. Similarly, all other legends in Figure 4-4 & Figure 4-5 have same syntax.

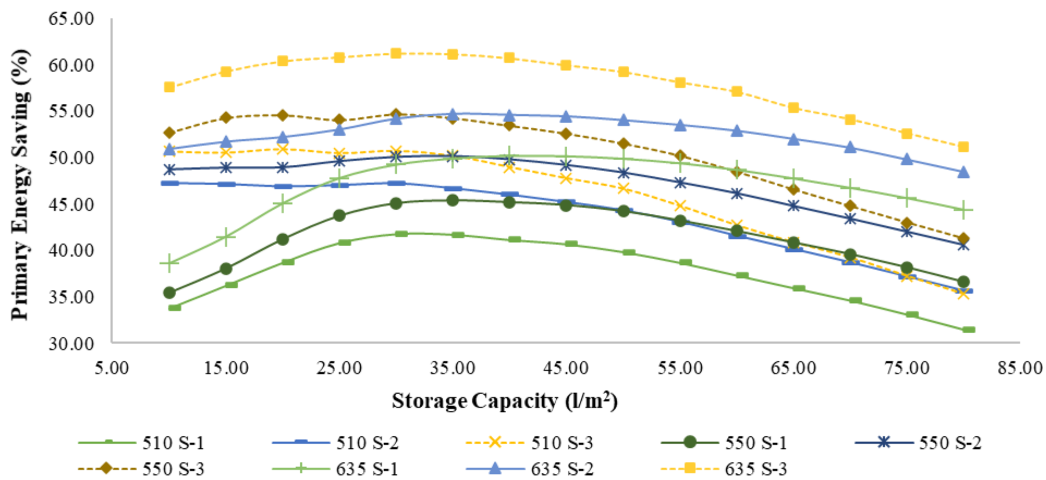


Figure 4-4: Trend of Storage sizes for FPC areas of 50% PES for all flow schemes

It can be observed from Figure 4-4 that with increasing storage capacities, primary energy saving first increases, then start decreasing afterwards. For FPC, collector area of 635 m^2 with 30 m^3 storage tank volume is required for achieving 50% PES with S-1 while collector area and storage tank volume is reduced to 550 m^2 and 20 m^3 respectively, is used for saving 50% of primary energy.

In S-3, optimized volume (10.2 m^3) with comparatively small collector area (than S-1 & S-

2) is used to achieve the target of 50% primary energy saving. Similar trends are obtained for ETC as shown in Figure, however, the requirement of collector area and storage tank size is greatly reduced because of the higher efficiency of the ETC [50]. With S-1, 128 m^2 collector area and 3 m^3 storage tank, while S-2 simulated with reduced storage size of 2 m^3 and 112 m^2 collector area to get 50% saving of the primary energy. Collector area of 93 m^2 is required to achieve 50% primary energy saving with scheme 3 of the current study with optimum storage capacity of 15 l/m^2 (1.4 m^3). The reason of decreasing PES with increasing storage sizes is that the higher tank sizes requires more time to increase the tank's average temperature so that more auxiliary energy will be consumed and for longer span of time to heat up the colder intake of tank's outlet. This is also the reason that S-2 would perform better than S-1 because S-2 diverts chiller's outlet flow to auxiliary unit when it is hotter than the tank's outlet so that less auxiliary energy consumption as compared to S-1.

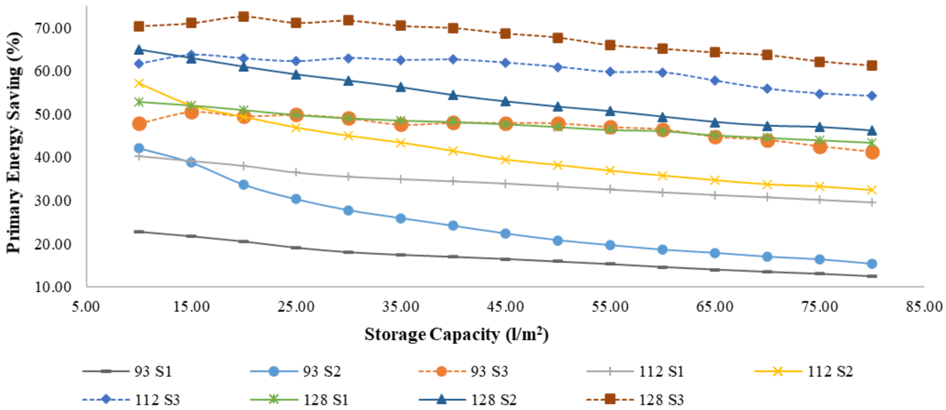


Figure 4-5: Trend of Storage sizes for ETC areas of 50% PES for all flow schemes

4.4 Primary Energy Saving trend with Collector Area

In Figure 4-6 & Figure 4-7, the seasonal variation of fraction of energy saving with collector areas of FPCs and ETCs to fulfil the cooling demand for all flow schemes is demonstrated at their optimum conditions. S-3 returns higher energy saving for ETC and FPC and S-1 yields less energy saving.

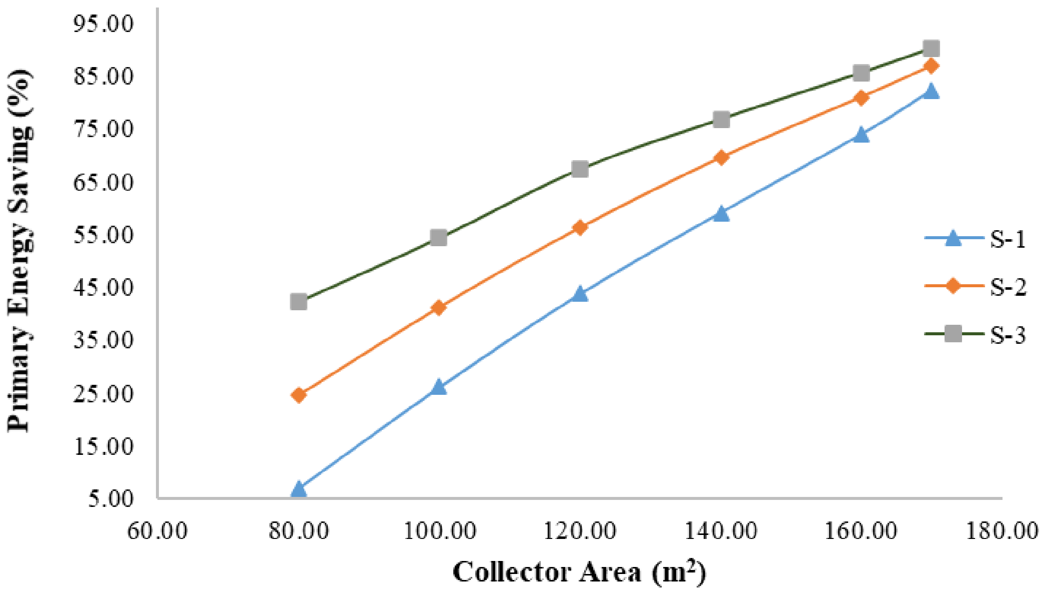


Figure 4-6: PES vs ETC area

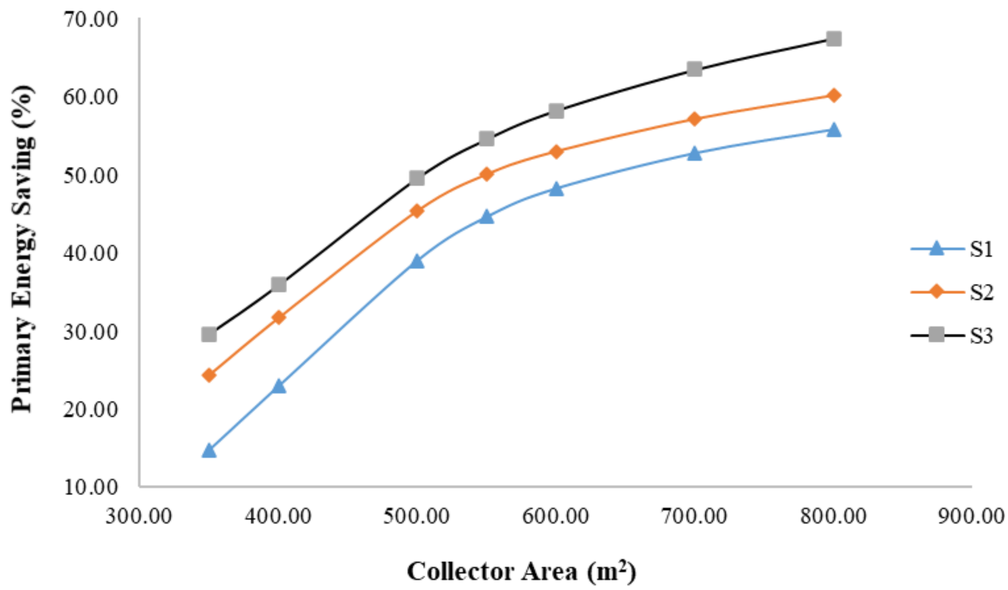


Figure 4-7: PES vs FPC area

The trend in Figure 4-6 & Figure is explained by Figure 4-8 & Figure 4-9, in which an area of 93 m^2 is selected with their optimum storage sizes and collector's tilt of each scheme and simulated for a day of 15th June (3960-3984 hrs.), in order to comprehend the fact that how schemes S1, S2 & S3 are acting in descending order of their primary energy saving trend.

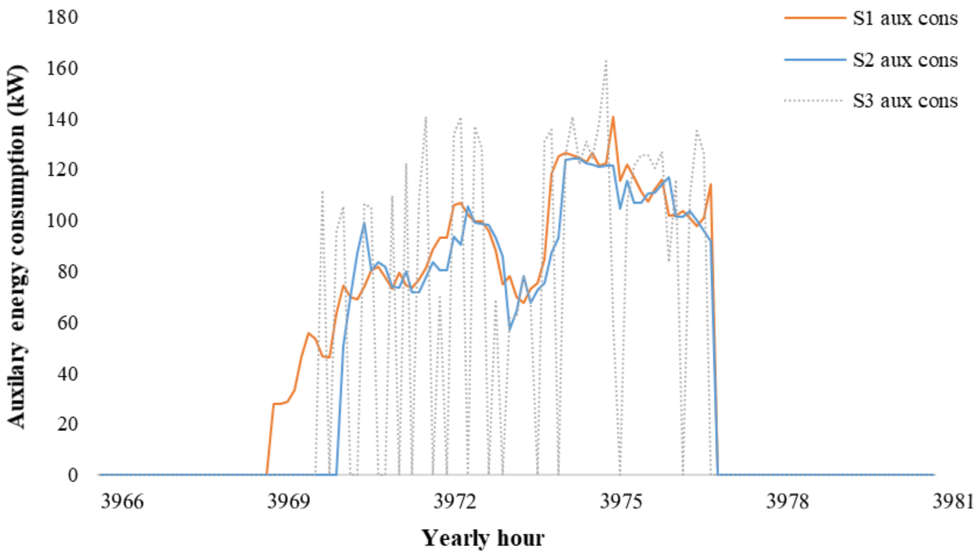


Figure 4-8: Auxiliary energy consumption for the day of mid of June

In Figure 4-8, it can be seen that auxiliary energy consumption of S-1 starts rising at earliest of the day than S-2 & S-3 while auxiliary energy consumption of S-3 mostly fluctuates from zero to its maximum point many times a day. As, it is obvious from the equation (3.14) that primary energy saving primarily depends on auxiliary energy consumption by the boiler in the system, so during operating hours of the building, the lowest (zero) value of S-3 auxiliary energy consumption (legend S3 aux cons) shown in Figure gives the clear indication of OFF condition of the boiler many times during the day and reflects higher PES than S-1 & S-2. However, in order to have a more clear & logical understanding that how boiler consumes energy to heat up the hot water up to the required temperature for chiller, Figure 4-9 explains the behavior of boiler & chiller component for all three schemes on operating hours of 15th June (i.e. 09 a.m. to 05 p.m.). At the start of cooling operation, inlet temperature to the boiler was already higher than the minimum driving temperature of the chiller due to stored thermal energy in all three schemes. As the time pass by, the hot water coming from chiller combines and transfers heat with the water in the tank whose effect is to decrease the temperature of the fluid flowing towards the chiller and declining behavior of temperatures at the inlet of auxiliary boiler was observed. Mechanism of modified flow schemes (S-2 & S-3) serves the purpose here for saving of auxiliary energy by allowing the hot water to feed the boiler with higher temperatures than conventional scheme S-1. Figure illustrates the transient behavior of inlet temperature to boiler for all schemes throughout the operating hours of the day. It is also understood that boiler's outlet temperatures will be equal to minimum required temperature of the chiller at the times when boiler's inlet is less than 109℃ for S-1 & S-2. COP of optimized scheme is shown as the secondary axis of Figure. Dynamic profile of chiller's generator outlet temperatures is displayed in Figure 4-10 with minor differences in their average values stayed under 103℃. Dotted lines are the average temperatures for boiler inlet for the day in the month of June shown in Figure 4-9 are the clear & strong clue that why & how boiler consumes energy for heating up the hot water

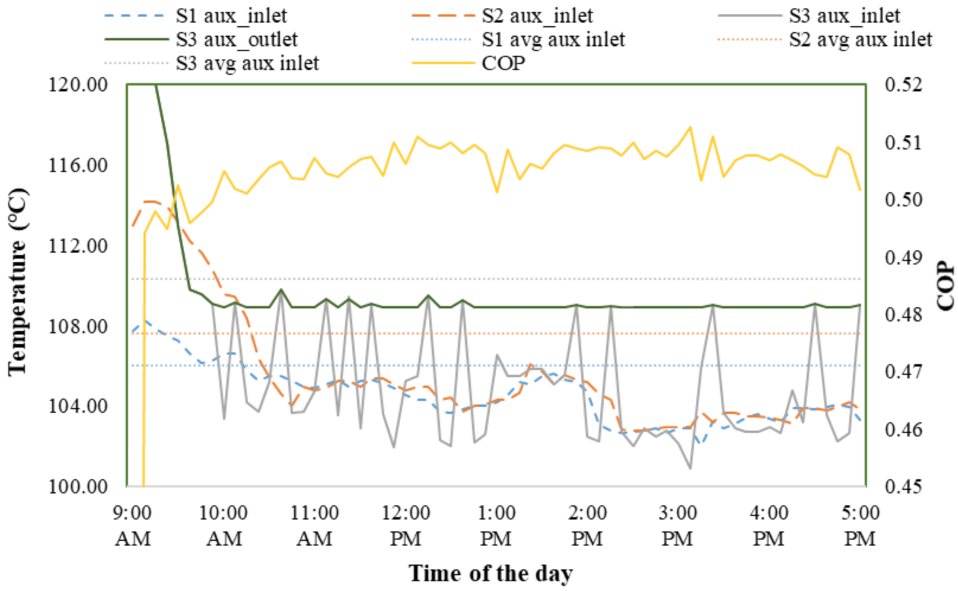


Figure 4-9: Chiller COP and inlet/outlet temperature profile for auxiliary heater for mid-day of June

up to the minimum driving temperature (109℃) for the chiller in descending order of S-1, S-2 & S-3. In Figure 4-6, the probable reason of narrowing a gap between flow schemes curves is that the boiler's inlet HTF attains 109°C so early and operates the chiller with higher generator inlet temperatures more than the minimum required temperature.

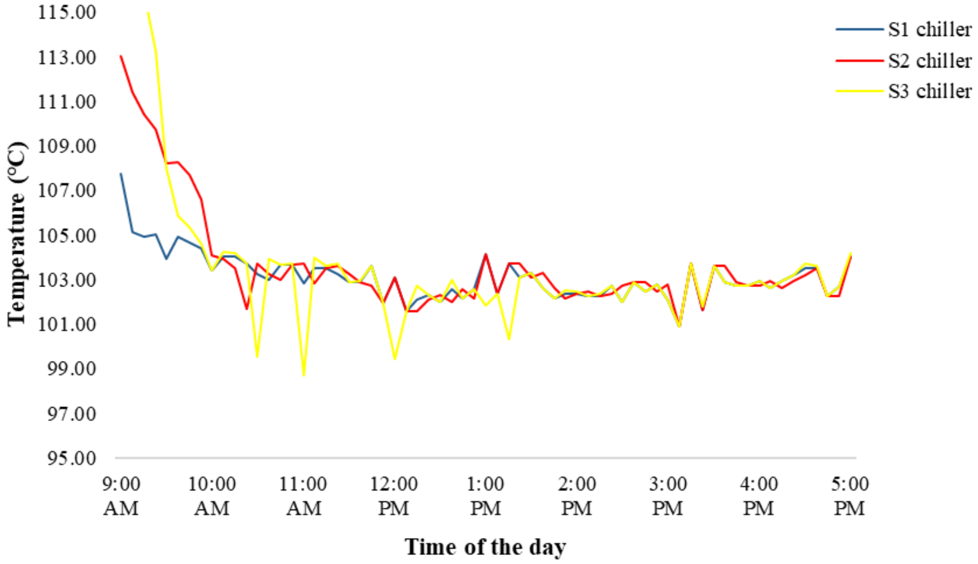


Figure 4-10: Outlet temperature profiles for all flow schemes at chiller's generator

Returning fluid from chiller in case of S-2 & S-3 would remain above 109°C for higher collector areas, so no matter if it is diverted or not because the inlet to the chiller will remain more than 109°C that results in auxiliary unit remain in OFF condition. Trend of primary energy saving with higher areas of FPC for all schemes is shown in Figure 4-7, where optimized flow scheme S-3 performed always better than the other two schemes.

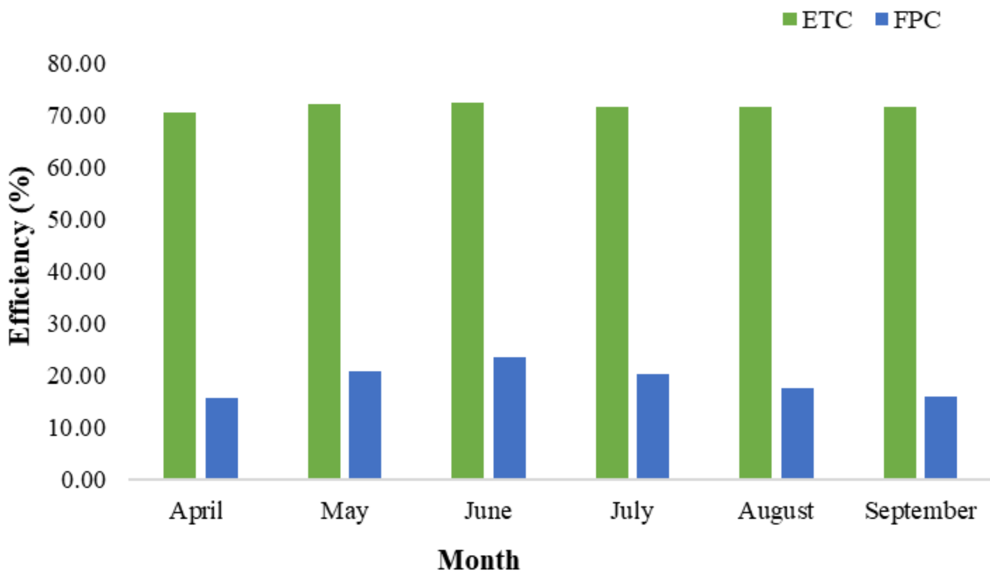


Figure 4-11: Monthly Efficiencies of ETC & FPC for optimized flow scheme (S-3)

4.5 Comparison of Monthly Averaged Efficiencies of Collectors

Monthly efficiencies in Figure 4-11 are plotted at optimal thermal storage and collector slope and for the areas of ETC (93 m^2) and FPC (510 m^2) by which 50% primary energy saving would achieve by using flow scheme-3 of the current study i.e. monthly efficiencies of ETC are plotted for 93 m^2 area, optimal thermal storage size of 15 l/m^2 and 9º collector tilt. Similarly, monthly efficiencies of FPC are plotted for 510 m^2, optimal storage size of 20 l/m^2 and collector tilt of 15º.

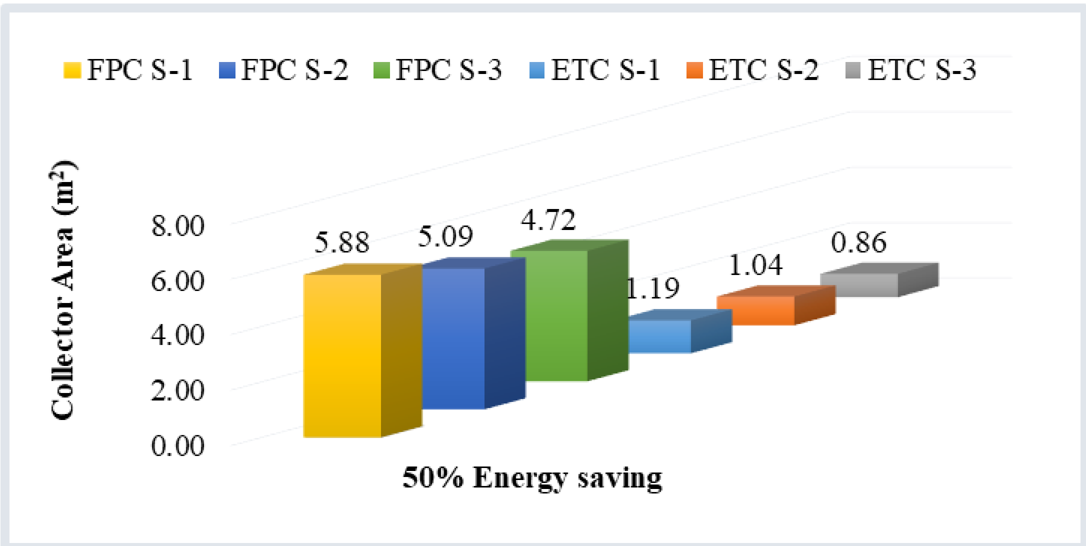
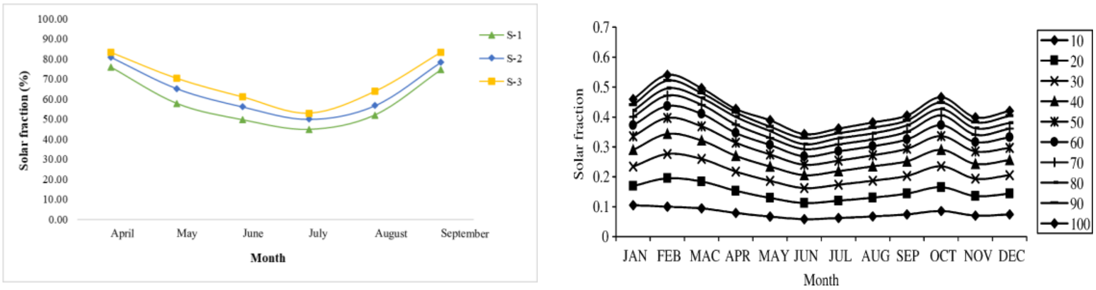


Figure 4-12: Collector area per kW of cooling capacity

4.6 Validation of Results:

Monthly solar fractions curves obtained from three schemes of the current study as shown in Figure 4-13(a) was compared with Assilzadeh et. al [44] study shown in Figure 4-13(b) and found similarity of the trend for the months of summer season. It can be seen from Figure (b) that the solar fraction is plotted for entire year in previous published article while in current study, results of solar fraction from April to September were evaluated

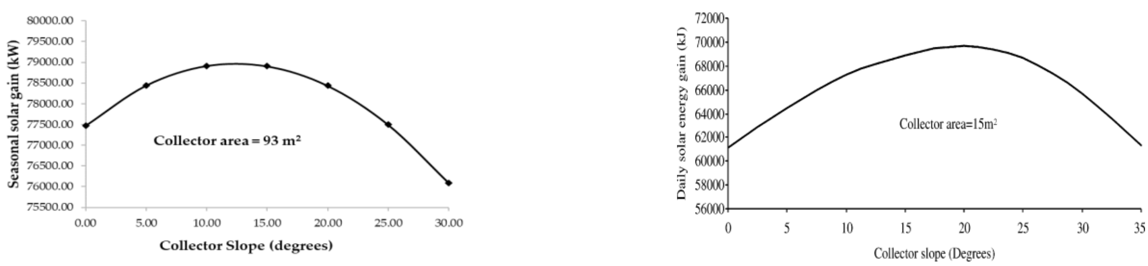


(a) (b)

Figure 4-13: Validation of monthly solar fractions

for the parameters of 50% PES and plotted in Figure 4-13(a). Legends (10 to 100) in the already published paper representing different collector areas used. Lower solar fractions at the mid of the season (July) is an indication of higher auxiliary energy consumption and lowest primary energy saving month of the season justifying equation (3.13) and equation (3.14) of the current study.

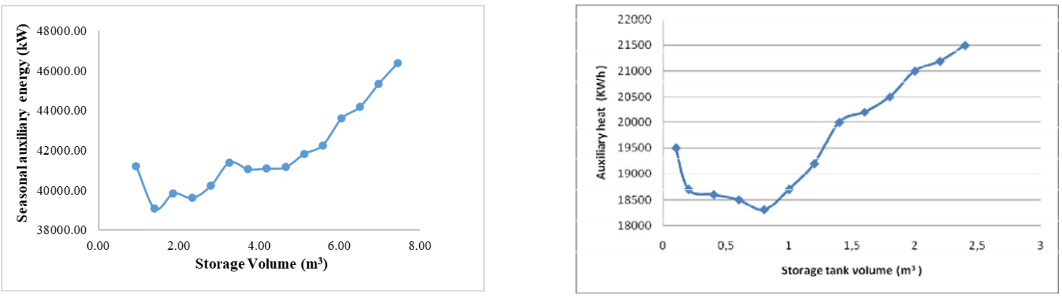
Tilt angles of the collector of S-3 by using optimum storage size were plotted against seasonal collector gain obtained for ETC in the current study .As Figure 4-14(a) & Figure 4-14(b) exhibited fairly similar trend for optimum tilt angle of the collector. Figure 4-14(a) matching with Assilzadeh et. al [44] as shown in Figure 4-14(b). However, the optimum value of the tilt angle of the



(a) (b)

Figure 4-14: Validation of optimum slope

solar collector depends upon the specific location; therefore, plots may differ in values (because of different collector areas used) and units, but trends are found to follow similar paths in both (current and previous) studies.



(a) (b)

Figure 4-15. Validation of optimum storage size

Another similarity of trend for storage size with auxiliary energy consumption for cooling system is shown in Figure 4-15, as minimum value of auxiliary energy consumption corresponds to the optimum tank volume. It can be clearly seen that current study Figure 4-15(a) is quite similar to the Djelloul et. al [51] study shown in Figure 4-15(b) for relation between storage size and auxiliary energy.

Table 4-1: Summary of results

Type of collector	Optimum slope of collector	Flow Schemes	Collector Area m^2	Tank size m^3	Area per kW of Refrigeration
FPC	15°	S-1	635	30	5.88
		S-2	550	20	5.09
		S-3	510	10.20	4.72
ETC	9°	S-1	128	3	1.19
		S-2	112	2	1.04
		S-3	93	1.4	0.86

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Chapter 5: CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

i. Best collector slope based on highest SF was found to be a 15° for FPC and 9° for ETC.

ii. Results indicated a perceptible difference between maximum monthly averaged efficiencies for both collector types. Hence, it can be decided that Flat Plate Collector is not an ideal option of the presented study because of requirement of larger area than ETC and the losses associated with operating temperature of above 100°C.

iii. Evaluation of 50% Primary energy saving is superior for Scheme 3 compared to Schemes 1 and 2 with matching storage at much smaller collector areas for ETC. The discrepancy in fraction of energy saving of all schemes shrinkages for higher collector area, but for real scenarios, economics consideration of the system also involves while deciding to use higher collector areas.

iv. Overall, Scheme-3 with evacuated tube collector is considered as the best selection for the current study to use with smallest ratio of collector area to kW of cooling capacity.

v. Because of insufficient experimental records that can be used for direct validation of current results, trends in previous publications of simulation studies were found to be matched with current simulation trends.

5.2 Limitations of the study

i. Cost analysis of the Absorption cooling system was not performed in this study.

ii. Energy consumed by two pumps used for circulation of heat transfer fluid in the system was not considered, so the performance indicators would have lower values for practically installed system than evaluated in current study.

iii. As the working temperature of the fluid is more than the normal boiling point in this study, which indicates pressurized water system in storage unit, however, study did not explain pressure variation conditions of the storage system.

iv. The current study is limited to sensible storage system.

5.3 Suggestions for future work

i. We can extend this study with modified design and model for solar heating system during winters using TRNSYS.

ii. The arrangement of solar collectors by series and parallel combination can enhance the performance of the system.

iii. This TRNSYS model can be compared with other installed systems and their performance may enhanced by employing best flow scheme specified in current design.

iv. The technical data of different manufacturers of solar collectors and absorption chillers can be used in TRNSYS modelling to get higher performances than the current system.

v. Different types of collectors like parabolic trough or other concentrated types can be coupled with double or triple effect chillers can be utilized to encounter higher cooling demand.

vi. Performance can be analyzed by using efficient auxiliary heating device other than boiler.

vii. Current System can be employed for cooling other commercial buildings like shopping malls which operates during daytime.

viii. The performance of the system can be improved by using phase change materials in thermal storage tank.

ix. The deployment of absorption cooling system for air conditioning of vehicle can be an interested application to study and research in future by using the combined heat of sun and vehicle engine.

APPENDIX: PUBLISHED RESEARCH ARTICLE

applied sciences

MDPI

Article

Effect of Modified Flow Schemes of Heat Transfer Fluid on the Performance of a Solar Absorption–Cooling System for an Educational Building in Pakistan

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Received: 31 March 2020; Accepted: 5 May 2020; Published: 11 May 2020

Abstract: Performance of solar absorption cooling systems (SACS) is the focus of contemporary studies for decreasing the electrical energy consumption of buildings as the conventional cooling system of buildings is the main consumer of electrical energy during the summer season in hot–humid climates. In this study, the performance analysis of SACS by manipulating different flow schemes to the heat transfer fluid between different components of the system was performed. TRNSYS model of SACS in an education building located at the city of Peshawar (34.00 N, 71.54 E), Pakistan to encounter the peak cooling load of 108 kW (during operating hours of the building i.e., 09 a.m. to 05 p.m.) is developed and all possible flow schemes of heat transfer fluid between the system's components were compared. In Scheme-1 (S-1), a conventional flow pattern is used in which the hot water exiting from the chiller unit flows directly toward the stratified thermal storage unit. In Scheme-2 (S-2), the modified flow pattern of hot water exiting from the chiller unit will divert towards the auxiliary unit, if its temperature exceeds the temperature at the hot side outlet of the tank. Another modified flow pattern is Scheme-3 (S-3) in which the hot water leaving the chiller to keep diverting towards the auxiliary unit unless the outlet temperature from the hotter side of the tank would reach the minimum driving temperature (109 C) of the chiller's operation. Simulations in TRNSYS evaluates the SACS's performance of all the schemes (conventional and modified) for the whole summer season and for each month. In general, S-3 with evacuated tube solar collector results in better primary energy saving with the smallest collector area per kilowatt for achieving 50% primary energy saving for the whole summer season.

Keywords: solar absorption cooling system; educational building; TRNSYS; modified flow schemes; 50% primary energy saving

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1. Introduction

Cooling systems installed in most of the educational buildings use vapor-compression cycles that result in a substantial increase of electrical load and hence, a rise in the operating cost of the education sector, especially during the summer season. One of the best ways to reduce the cooling related electricity load of the educational building is to replace the conventional vapor compression cooling systems with absorption cooling systems without compromising the thermal comfort of the occupants. The absorption cooling relying on solar or low-grade waste heat is one of the clean cooling technique [1]. An absorption cooling cycle driven by geothermal energy is also an economically viable option [2]. However, the use of geothermal sources is entirely location-specific. Utilizing solar heat for cooling of buildings can be attractive because the demand for cooling coincides with the availability of peak solar heat [3]. Pakistan receives about 1900–2200 kWh/m2 annual solar insolation that may be effectively used for solar thermal applications where low-grade heat requires a special absorption cooling system [4]. To minimize the intermittent nature of solar energy, the thermal storage unit is one of the integral parts of the absorption cooling system [5]. Simulation and modelling of the absorption cooling systems is vital and less cost-effective for the optimization of its different components. TRNSYS is an extensible and comprehensive simulation tool for the transient simulation of systems and is used by researchers around the world for modeling of new energy ideas, renewable energy simulations. It is observed from the reviewed literature that a lot of research work on Solar Absorption Cooling Systems (SACS) has been conducted with this simulation tool i.e., references [6–10] optimized the values of key components of the cooling system like collector area, tilt angle and the volume of the storage tank, and some performance indicators, like solar fraction and COP. M. Shoaib et al. [11] performed configuration based modeling to improve the primary energy saving from single effect solar absorption cooling system, some studies [12–14] compared performance based on economic evaluation of solar absorption cooling systems while some [15–18] studies analyzed the optimization effects of storage system on the performance of SACS. Most of the above-referenced studies indicate that the temperature of the heat transfer fluid flowing from solar collectors to the absorption cooling system chiller greatly affects the overall performance of the absorption cooling system. Schematics of proposed absorption cooling systems in most of the above-referenced studies clearly indicating that only one flow scheme for circulation of heat transfer fluid was used between storage tank to chiller (i.e., flow of HTF directly form the storage tank to the chiller's generator and then back to storage tank). As it is revealed from preceding studies [19,20] that the performance of the chiller was mainly affected by the inlet temperature to the generator unit of the chiller. Therefore, the main idea of the current study is to examine different ways of feeding heat transfer fluid to the generator of the chiller and analyzed their effects on the performance of the cooling system. In the light of the above-reviewed literature, the lack of research related to various flow schemes of heat transfer fluid (HTF) between components of SACS is perceptible and still there have more options of different flow schemes for heat transfer fluid to enhance the system's performance. The general significance and core purpose of the current study is to compare all possible flow patterns of heat transfer fluid that can be exercised between the storage–chiller loop of SACS and analyze their effects on the performance of SACS and in doing so, obtain an optimized TRNSYS model by replacing an installed conventional compression chiller of 108 kW cooling capacity having COP value 3 (by Dunham–Bush manufacturer) in an educational building located at the city (having the highest monthly averaged radiations) of Pakistan and finally decide the flow scheme of best performance, with an efficient solar collector.

2. Description of Flow Schemes for HTF

The schematic of the main components used in the solar absorption cooling system for educational building is shown in Figure 1. Heat transfer fluid (water in the current study) flows between different components of the cooling system to exchange its heat during flow. For analyzing the controlled flow of HTF, three loops are presented between the components of the cooling system in the current study i.e.,

Collector–tank loop
Storage–chiller loop
Chiller–building loop

These loops of HTF are indicated by yellow, red and green colors respectively in Figure 2. The current study focused on the loop in which HTF feeding the generator of the chiller (i.e., storage–chiller loop). To the best of our knowledge, HTF can be circulated between the components of a storage–chiller loop by three different ways which are known as three proposed flow schemes of the current study.

In scheme 1, solar collector receives the radiations from the sun and and heat up the water that flows toward thermal storage which then circulates towards solar collector after increasing the temperature of stored hot water.

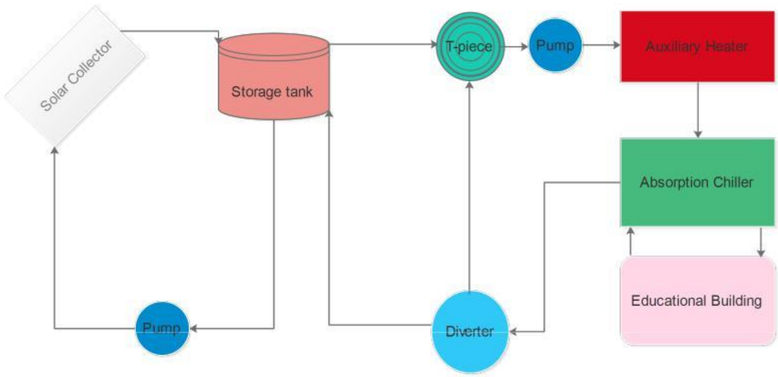


Figure 1. Solar absorption cooling system.

The flow of the collector pump is controlled in such a way that when the hot water inlet to the solar collector exceeds its outlet temperature, it will stop pumping. The minimum required input to the chiller unit is the 109 C hot water (neglecting the boiling point as the system is pressurized) either at storage temperature (if it is equal to greater than 109 C) or after heated by auxiliary unit fulfilling the requirement of refrigerant evaporation in the chiller's generator and returns to the storage tank.

A pump is used to control the flow of hot water from the storage tank to the chiller only during the operating hours of the building. During this time period an auxiliary heating unit with the thermostat is used that turns on the pump when the temperature from the hotter side of the storage outlet is less than the required operating temperature to run the chiller.

In Figure 1, the removal of tee, diverter component and direct links (of storage to pump 2 and chiller to storage) in storage–chiller loop shows conventional flow pattern of heat transfer fluid which is named as scheme 1 of the current study.

In scheme 2, the conventional flow pattern is modified in such a way that when hot water ($T_{chiller,out}$) coming out from absorption chiller, it will divert towards the auxiliary heating unit instead of flowing directly towards the storage tank if storage's outlet is colder compared to the water returning from chiller as shown in Figure 3a. A flow diverter was used for diversion of the chiller's output hot water the to auxiliary heater and a tee piece component to pass the hot water either from storage tank or from diverter. The flow of water in the collector–tank and chiller–building loop is the same as in scheme 1 shown in Figure 2.

In scheme 3, the flow of heat transfer fluid is modified with one of the most advanced techniques by diverting the flow after coming out from the absorption chiller unit. Figure 3b illustrates the mechanism of scheme 3 in which hot water ($T_{chiller,out}$) will continuously divert towards the auxiliary heater unless the temperature of the storage tank's hot side outlet would reach up to the minimum required temperature i.e., 109 C for the chiller so that it will operate without consuming auxiliary energy. A controller signal has been introduced to generate the signal for diverter to start or stop the diversion of HTF, the remaining components are the same as scheme 2 as shown in Figure 2 with the only difference in the temperatures values for managing the diversion flow of chiller's hot water outlet.

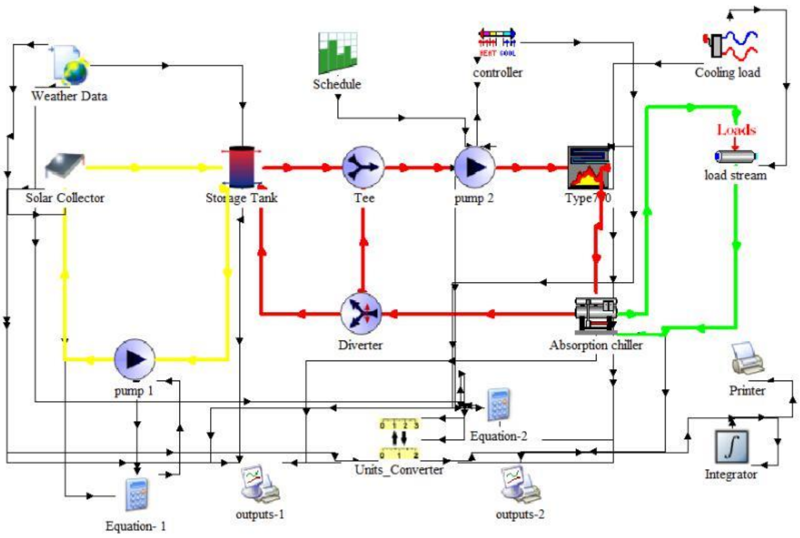


Figure 2. TRNSYS model of solar absorption cooling system (SACS) for modified flow schemes.

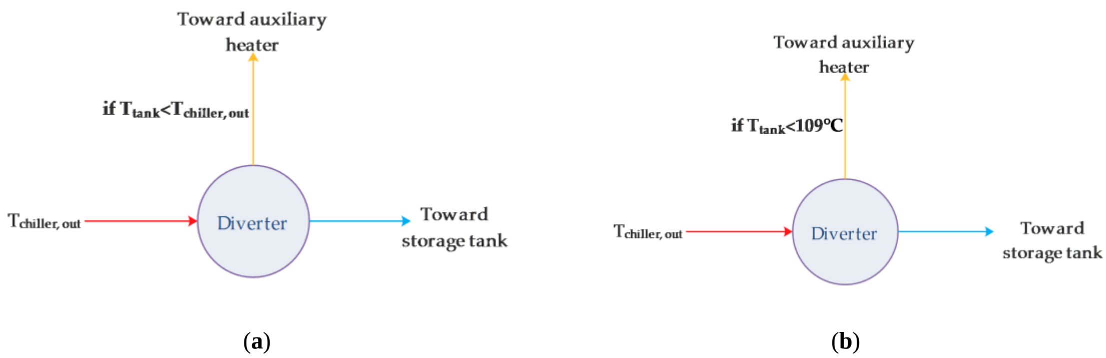


Figure 3. Mechanism of modified flow schemes. (**a**) Scheme 2. (**b**) Scheme 3.

3. TRNSYS Modelling

TRNSYS is employed for the modeling and simulating all schemes of the heat transfer fluid flow between storage tank to the chiller in SACS. The monthly averaged solar radiations of hot climate cities of Pakistan (Lahore, Islamabad, Multan, Peshawar and Karachi) are compared as shown in Figure 4 and a cooling system in an educational building of Peshawar (highest radiations city of Pakistan) is chosen for modeling of the solar absorption cooling system, having 108 kW peak cooling load during the summer season and daily operating schedule from 9 a.m. to 5 p.m. TRNSYS model for modified flow schemes exhibited in Figure 2. Thick red lines shown in Figure 2 depict hot water loop from the storage to the chiller unit, yellow lines depict the collector–tank loop while green lines indicated loop of chilled water from the chiller to building. Scheme 2 and Scheme 3 are having the same components except for the temperature range for the diverter. Meteonorm weather data available with the TRNSYS v17 was used in the current model of SACS.

To accomplish the objective of comparing the performance of three proposed flow schemes of the study, the following assumptions are considered with the design of cooling system used:

- Heat losses from pipes are neglected.
- Energy balance technique is used for cooling tower specifications.
- Boiling effects of the heat transfer fluid are ignored.
- Electricity consumption of pumps is not included as the auxiliary energy consumption record.

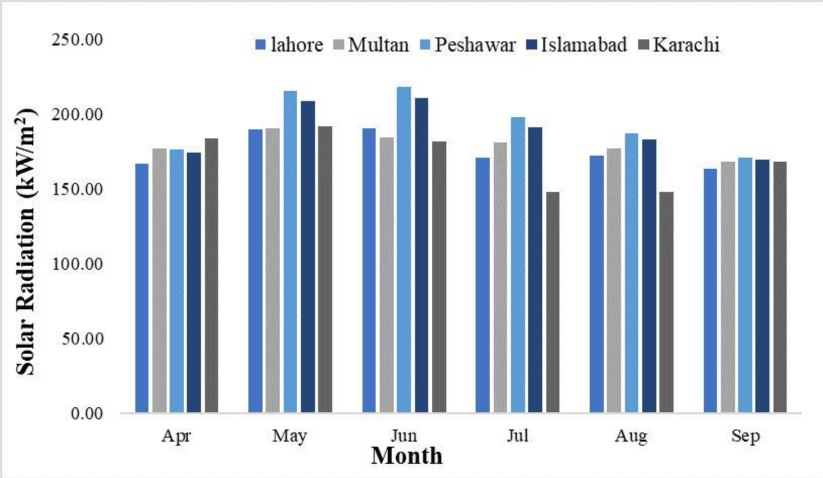


Figure 4. Monthly solar radiations in Pakistan.

Connections shown in Figure 2 are just logical connections and not represents pipes. For simplicity of the model, pipes or valves connections are not included in the current model by which heat losses from pipes can be measured as it can be assumed that they are perfectly insulated. The second assumption of the study reflects theoretical calculations of cooling water energy by energy balance equation i.e., Equation (6) which tells us energy from absorber and condenser rejected by cooling water by keeping the temperature difference of 5 C between the inlet and outlet cooling water for design consideration in Equation (6) of the current study and no separate TRNSYS component of the cooling tower is used. The third assumption depicts the pressurized water system by which the normal boiling point of water can be varied, however, pressure calculations are not included in the study. All flow schemes of the current study use two flow pumps and would consume considerable but same amount of electricity as the designed flow rates are the same in all schemes, so that this electricity consumption of the pumps is not included as auxiliary energy consumption calculation.

With the aforementioned assumptions, it is suggested that this simulation study provides a somewhat higher evaluation of performance indicators than practically installed system but still, our sole purpose from this study of analyzing the effects on the performance of the solar cooling system due to modified flow schemes can be fulfilled with these assumptions.

3.1. Weather Data Processing

Weather data for the cities of Pakistan was taken from the built-in Meteonorm files provided with TRNSYS v17. Type 15-6 component used standard format data while Type 99 component used user format weather data file of Islamabad from Meteonorm Software because of the absence in TRNSYS v17 package.

3.2. Solar Thermal Collectors

An evacuated tube solar collector and a flat plate solar collector has been used in the current study to utilize solar energy for water heating. The second order incidence angle modifier efficiency equation [21] is described below:

$$\eta = a_0 - a_1 \frac{(T_i - T_a)}{I_T} - a_2 \frac{(T_i - T_a)^2}{I_T} \tag{1}$$

In Equation (1) a_0 is the intercept efficiency and a_1 and a_2 are the efficiency slope and efficiency curvature respectively. I_T is the global solar radiations on the surface of the collector, T_i is the temperature of the heat transfer fluid at collector inlet and T_a is the temperature of ambient air. The mass flow rate of the Solar loop is set by considering the flow rate at the testing condition and area used. Technical data includes default values of Type1b for Flat plate collector from TRNSYS library and Evacuated tube collector (model number: Enertech Enersol HP 70-8). Further details of these collectors are provided in the Table 1.

Table 1. Technical specifications of collectors.

Type of Controller	ETC	FPC
a_0	0.804	0.800
a_1 (kJ/h m^2 K)	1.56	13
a_2 (kJ/h m^2 K)	0.0054	40
Testing flow rate (kg/h m^2)	100	40
IAM longitudinal at 50	0.93	1 0.2(1/cos*q* 1)
IAM Transversal at 50	1	1 0.2(1/cos*q* 1)

3.3. Thermal Storage

Uninterrupted supply of solar energy input is provided by the use of thermal Storage tank of stratification type with all having 10 nodes and an energy balance approach is used to get the outlet temperature from each node over the prescribed time step [21]. The value of the designed flow rate from tank outlet to the chiller is given in Table 2. The losses from the tank are uniform and the value is 0.83 W/m^2 C is considered. Hot water enters and exits from top sides and cold water replaces from bottom sides of the stratified tank as shown in Figure 5.

Table 2. Design parameters for solar absorption cooling system.

Designed Parameters	Values
Type 700 (Auxiliary Boiler)	
Time step (minutes)	7.5
Boiler Efficiency (%)	85
Combustion Efficiency (%)	78
Type 107 (Single effect hot water fired Absorption Chiller)	
Specific heat of heat transfer fluid (kJ/kg K)	4.19
Chiller’s rate COP (default)	0.53
Hot water’s flow rate from tank to Chiller (kg/h)	17,507
Chilled Water’s flow rate from Chiller to Building (kg/h)	16,710
Cooling Water’s flow rate for condenser and absorber heat rejection (kg/h)	54,527
Minimum Temperature for chiller operation (°C)	109
Cooling water inlet temperature (°C)	29.4
Chiller Water Setpoint (°C)	6.667
Peak Cooling Demand (kW)	108
Auxiliary electrical energy for solution pumps etc. (kW)	5.55
Type 4a (Thermal Storage Tank)	
Heat transfer coefficient for tank (W/m^2 K)	0.83
Number of total nodes	10
Density of water stored in tank (kg/m^3)	1000

3.4. Auxiliary Unit

To raise the temperature of HTF up to 109 C for generator operation of the chiller, a boiler is used as an auxiliary heating unit named type700 in TRNSYS library. This component gives the amount of energy consumed to raise the temperature of the water up to the desired temperature by considering input combustion and boiler efficiencies. When the heat transfer fluid enters the unit with a temperature less than the designed temperature, auxiliary heating unit turns on by the controller signal and heat up the fluid to the designed temperature. The boiler part load ratio is given by the following equation [21]:

$$PLR = \frac{Q_{need}}{Q_{max}}. \quad (2)$$

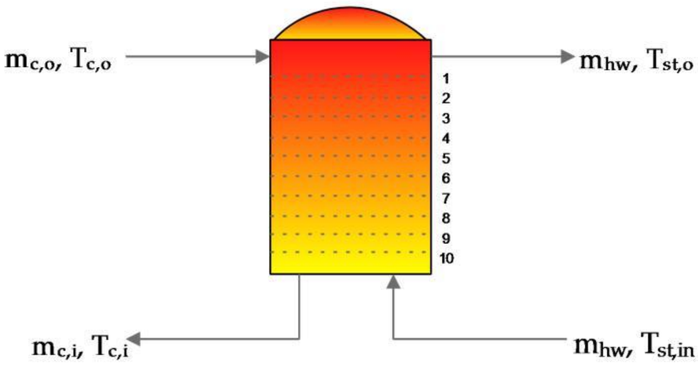


Figure 5. Hot water storage tank.

In this equation, Q_{max} is the maximum capacity of the auxiliary unit and Q_{need} is the amount of energy consumed by the auxiliary heating unit to keep the hot water temperature at the designed point. In order to raise the temperature of the incoming fluid, the value of Q_{max} is selected in such a way that the part load ratio of the boiler should remain equal or less than one, for raising the temperature of incoming fluid.

3.5. Absorption Chiller

Single effect hot water fired absorption chiller (type107) is used to encounter the cooling load of the building. This component uses a catalog data lookup approach to predict the performance and gives the correct output by interpolation if given input are in the range of performance catalog file. In TRNSYS v17 catalog data file of type107, the operating range of inlet hot water temperature for the chiller's generator is between 108 C to 116 C. Therefore, in current design, the minimum operating temperature of 109 C is chosen to operate the absorption chiller to encounter the peak cooling load of 108 kW (31 TR). The inputs of this section is crucial for designing of the cooling system as the chiller's capacity decides the amount of cooling demand that it can withstand. TRNSYS referenced formulation [21] is used for designing the parameters:

$$Q_{hw} = \frac{Capacity_{rated}}{COP_{rated}} f_{Designedenergyinput}. \quad (3)$$

In Equation (3), Q_{hw} is hot water energy that rated capacity must be available for the chiller operation, and COP are given in Table 2 while catalogue data file of $f_{Designedenergyinput}$ is taken from performance type107 at designed conditions.

$$Q_{coolingcapacity} = m_{chw} Cp_{chw} (T_{chw,in} \quad T_{chw,set}). \quad (4)$$

In Equation (4) $Q_{coolingcapacity}$ is the energy rate which must be removed by chiller (sometimes denoted by Q_{remove} or $Q_{chilled}$) that must be removed by chiller, m_{chw} is the flow rate of water from the chiller, m_{chw} is flow rate of water out from the chiller, C_{pchw} is the fluid's heat capacity from the chiller and ($T_{chw,in}$ $T_{chw,set}$) is the difference between the temperature of the chilled fluid entering and leaving at the chiller which is taken by considering default values of inputs of type107 i.e., default values of $T_{chw,in}$ and $T_{chw,set}$ are 12.22 C and 6.667 C respectively, in current study.

$$Q_{hw} = m_{hw} Cp_{hw} (T_{hw,in} \quad T_{hw,out}). \quad (5)$$

In Equation (5) Q_{hw} is hot water energy that must be available for the chiller operation, m_{hw} is the flow rate of hot water and is calculated from this equation by choosing the difference of hot water inlet and hot water outlet ($T_{hw,in}$ $T_{hw,out}$) value as 10 C while C_{phw} is the heat capacity of hot water. Qcw is cooling water energy rate is found out by the energy balance approach in the current study i.e.,

$$Q_{cw} = Q_{chw} + Q_{hw} + Q_{aux}. \tag{6}$$

In Equation (6), Q_{aux} is the auxiliary electricity consumed by the chiller for solution pumps and refrigerant pump etc. Equation (6) is used to balance the total energy of the chiller. This is the total energy rejected from chiller to environment and is used to find Q_{cw}. The Q_{cw} will be further used in Equation (7) for design consideration. The design parameters are already tabulated in Table 2.

$$T_{cw,out} = T_{cw,in} + \frac{Q_{cw}}{m_{cw} C_{pcw}}. \tag{7}$$

where, Q_{cw} is the energy required by the cooling water, m_{cw} is the flow rate of cooling water and is found from the Equation (7) by fixing the temperature difference of cooling water outlet and cooling water inlet (T_{cw}, out $T_{cw,in}$) as 5 C while C_{pcw} is the heat capacity of cooling water.

3.6. Hydronic Components

Type 114, Type 11h and Type 11f are the hydronic components as pump, tee-piece and the flow diverter respectively. These components are used for controlling the flow of heat transfer fluid with the aid of external control signals from the temperature controllers or conditional equations used in the model and with the designed mass flow rate. Flow diverter and tee-piece are used in flow schemes 2 and 3 where we need to divert the flow of hot water leaving the chiller, either towards the auxiliary or to the thermal storage by comparing the temperatures.

3.7. Controllers and Outputs

A five-stage thermostat component type108 for turning on the auxiliary heater and equations components for temperature comparison for stopping or starting the flow of the solar collector loop pump was used in all schemes. When the inlet temperature of the boiler is colder than the minimum required temperature of the chiller then the type108 causes the boiler to on, while the collector pump stops the flow by using the signal from the equation output value when inlet temperature exceeds the outlet temperature of the heat transfer fluid. Type65b, Type65c and Type25b are the output components used for obtaining output files in excel formats. Type14h is the forcing function that is used for generating a control signal at the input of the pump during the operating hours of educational building in this model.

3.8. Building Load Generator

Type686 is the heating and cooling load generator component for the building synthetically. This component has been extensively used in previous studies [22–24] to save their time from tedious calculations. Type686 takes the input as peak load and gives hourly cooling load demand which is quite like real building load and also uses daily and seasonal variations in the load as shown in Figure 6. Type682 component takes its input of cooling load from type686 and inlet fluid’s temperature from the chiller’s output and returns water stream (i.e., chilled water inlet for chiller) after handling the cooling load of building.

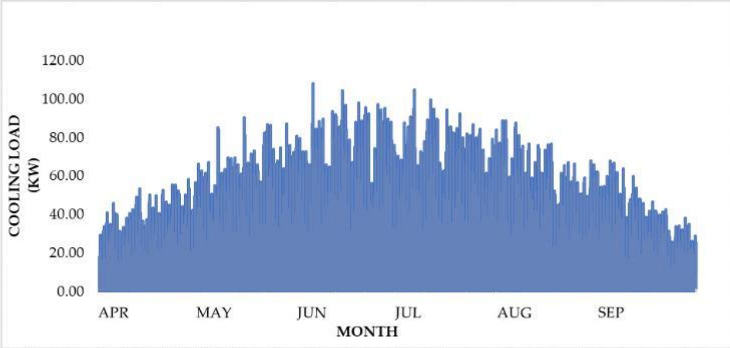


Figure 6. Cooling load profile.

4. Performance Indicators

Performance indicators for analyzing effects by comparing the performances of all schemes are given below:

4.1. Solar Fraction

Many studies used solar fraction as the main performance indicator in their study [3–5]. The solar fraction is defined [3] as:

$$SF = \frac{Q_{solar}}{Q_{solar} + Q_{boiler,aux}} \quad (8)$$

where Qsolar is the useful energy gain by the collector and $Q_{boiler,aux}$ is the auxiliary energy consumed by the boiler. Its value always remained between 0 and 1 having inverse relation as auxiliary energy consumed.

4.2. Primary Energy Saving

M. Shoaib et al. [11] used a generalized equation for fractional energy saving by which performance of the designed cooling system can compared with conventional VCCS for same cooling demand.

$$PES = 1 - \left[\frac{\dfrac{\int \dot{Q}_{boiler}}{\varepsilon_{heat}}}{\dfrac{\int \dot{Q}_{cooling,ref}}{COP_{ref} \cdot \varepsilon_{elec}}} \right] \quad (9)$$

The numerators inside the square brackets represent the total auxiliary energy expenditure of the boiler in our system while the denominator represents the electrical energy consumed by the referenced conventional vapor compression system unit installed in the building. In the Equation (9), $Q_{boiler,aux}$ is the integrated value of auxiliary heat from boiler, e_{heat} is the efficiency of boiler having Rparametric value 0.78 in our system, $Q_{cooling,ref}$ is the integrated value of the energy for cooling effect realized from referenced conventional cooling system (i.e., 108 kW), COP value of the installed chiller (model number WCPS 380 BP R/H) installed in the building from the manufacturer is 3, while e_{elec} denotes the conventional efficiency of a thermal power plant (ratio of electrical energy generation to the primary energy consumption) whose value is considered as 0.4. The electricity consumption of the hydronic components is not considered in this study which may cause the investigated results to slightly deviate from the actual fraction of primary energy saving and saving of at least 50% primary energy is considered as worthwhile as economical point of view in this study.

5. Results and Discussion

TRNSYS model runs for the time span of summer months i.e., from April to September. As not enough exploratory information exists from a real framework, the current simulations are compared with the already published results for validation of the current model. M. Azmi and A.Q. Malik [25] found in their study that the best tilt angle plays an extensive role to expand the output of a solar collector. The determination of the optimum collector slope is influenced by using many limiting factors, such as deficiency of manpower body and inaccessibility of the region of solar collector subject due to which normal tuning of the slope is mostly not feasible. So, in our model, the greatest seasonal SF is acquired for the area at which 50% PES is achieved with an optimum collector slope of 9 for evacuated tube collector (ETC) and 15 for a flat plate collector (FPC) as shown in Figure 7.

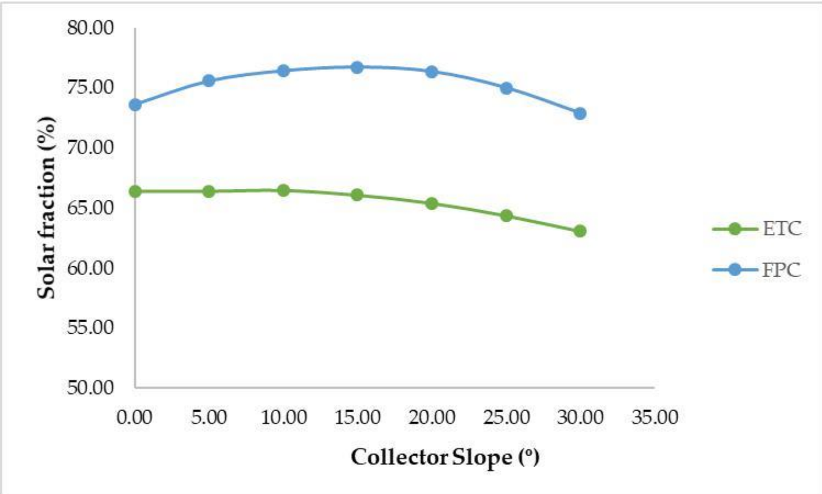


Figure 7. Optimum slope of solar collectors.

Storage sizes are plotted for collector areas of FPC in Figure 8 at which flow schemes are realizing 50% seasonal primary energy savings. It is essential to note that the legends in Figures 8 and 9 are representing the area of collector in m^2 used for one of the flow schemes of the study. For instance, legend 510 S1 means that 510 m^2 collector area used for flow scheme 1. Similarly, all other legends in Figures 8 and 9 have same syntax.

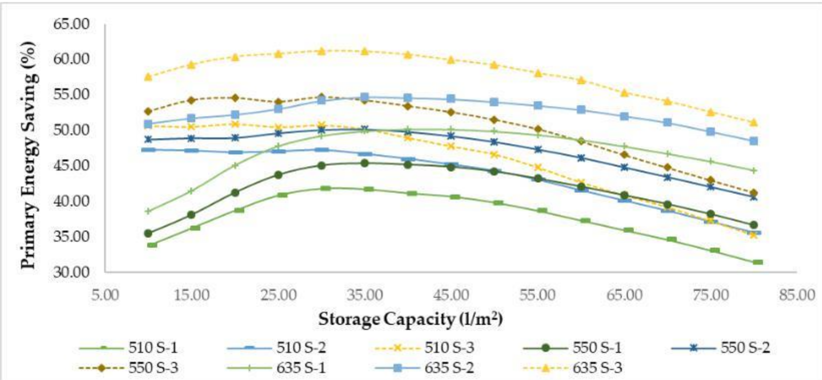


Figure 8. The trend of storage sizes for flat plate collector (FPC) areas of 50% PES for all flow schemes.

It can be observed from Figure 8 that with increasing storage capacities, primary energy saving first increases, then start decreasing afterwards. For FPC, collector area of 635 m^2 with 30 m^3 storage tank volume is required for achieving 50% PES with S-1 while collector area and storage tank volume is reduced to 550 m^2 and 20 m^3 respectively, is used for saving 50% of primary energy.
In S-3, an optimized volume (10.2 m^3) with a comparatively small collector area (than S-1 and S-2) is used to achieve the target of 50% primary energy saving. Similar trends are obtained for ETC as shown in Figure 9, however, the requirement of the collector area and storage tank size was greatly reduced because of the higher efficiency of the ETC [26]. With S-1, 128 m^2 collector area and 3 m^3 storage tank, while S-2 simulated with reduced storage size of 2 m^3 and 112 m^2 collector area to get 50%

saving of the primary energy. A collector area of 93 m^2 was required to achieve 50% primary energy saving with scheme 3 of the current study with an optimum storage capacity of 15 L/m^2 (1.4 m^3). The reason for decreasing PES with increasing storage sizes is that the higher tank sizes require more time to increase the tank's average temperature so that more auxiliary energy will be consumed and for a longer span of time to heat up the colder intake of tank's outlet. This is also the reason that S-2 would perform better than S-1 because S-2 diverts chiller's outlet flow to the auxiliary unit when it is hotter than the tank's outlet so that less auxiliary energy consumption as compared to S-1.

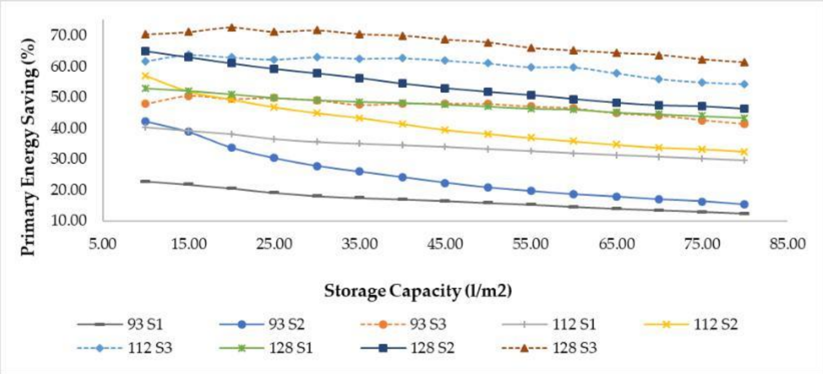


Figure 9. The trend of storage sizes for evacuated tube collector (ETC) areas of 50% PES for all flow schemes.

In Figures 10 and 11, the seasonal variation of the fraction of energy-saving with collector areas of ETCs and FPCs to fulfil the cooling demand for all flow schemes is demonstrated at their optimum conditions. S-3 returns higher energy saving for ETC and FPC and S-1 yields less energy saving.

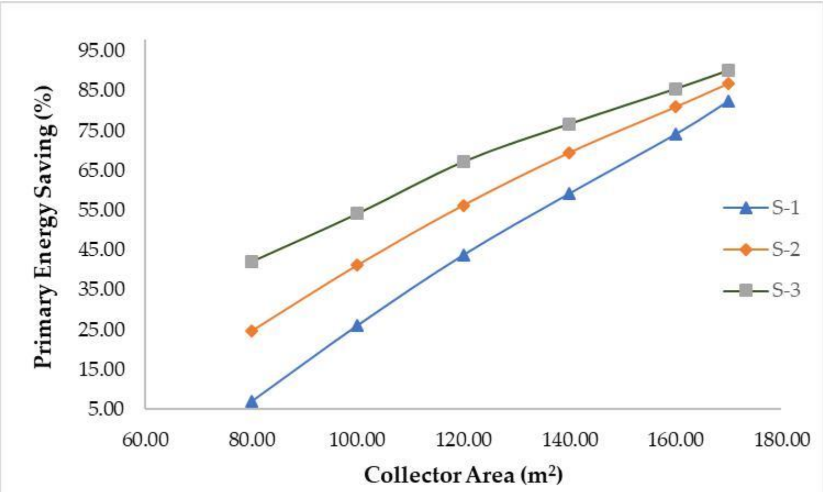


Figure 10. PES vs. ETC area.

The trend in Figures 10 and 11 is explained by Figures 12 and 13, in which an area of 93 m^2 is selected with their optimum storage sizes and collector's tilt of each scheme and simulated for the month of June. The results of a typical day of the summer season is plotted (3960–3984 h), in order to comprehend the fact that how schemes S1, S2 and S3 are acting in descending order of their primary energy-saving trend.

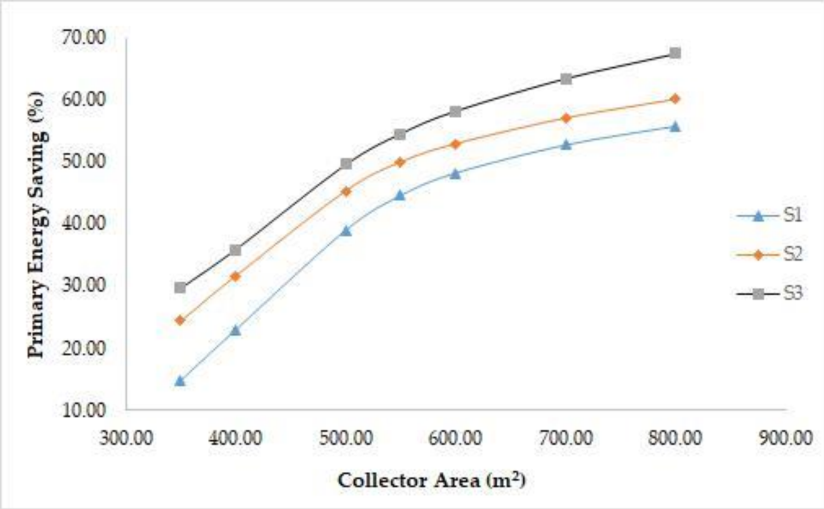


Figure 11. PES vs. FPC area.

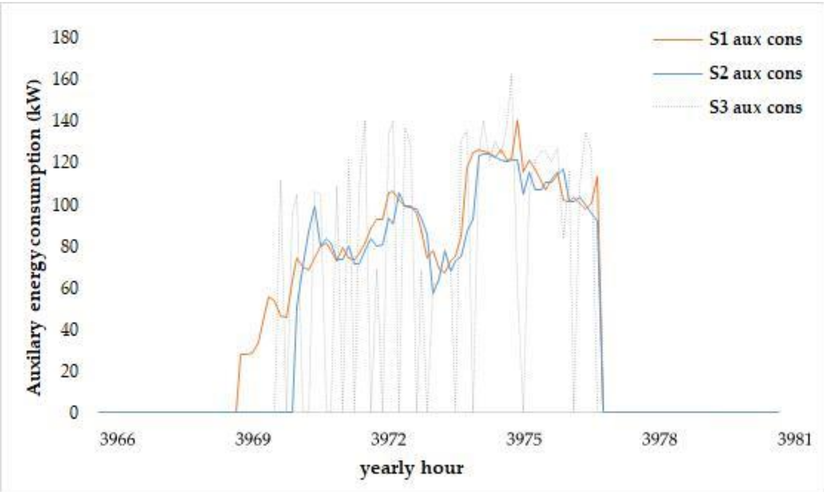


Figure 12. Auxilary energy consumption for the day of mid of June.

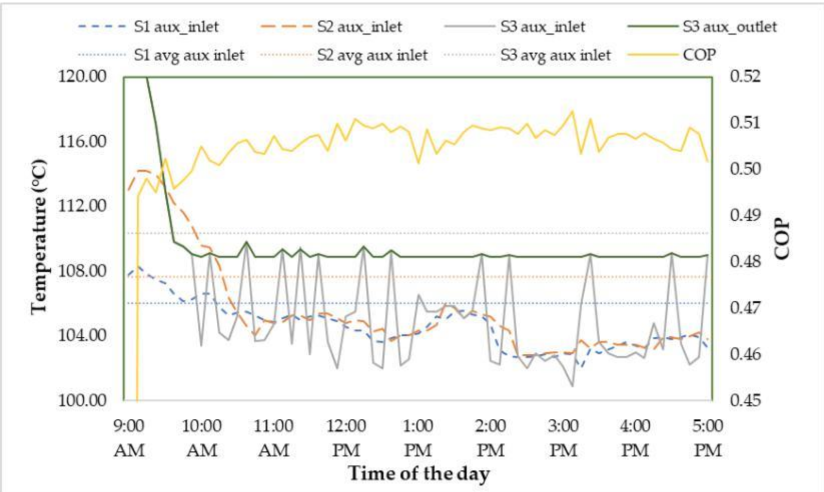


Figure 13. Chiller COP and inlet/outlet temperature profile for auxiliary heater for the mid-day of June.

In Figure 12, it can be seen that auxiliary energy consumption of S-1 starts rising earlier in the day than S-2 and S-3 while auxiliary energy consumption of S-3 mostly fluctuates from zero to its maximum point many times a day. As is obvious from Equation (8), primary energy saving primarily depends on auxiliary energy consumption by the boiler in the system, so during operating hours of the building, the lowest (zero) value of S-3 auxiliary energy consumption (legend S3 aux cons) shown in Figure 12 gives a clear indication of the OFF condition of the boiler many times during the day and reflects higher PES than S-1 and S-2. However, in order to have a clearer and logical understanding that how boiler consumes energy to heat up the hot water up to the required temperature for chiller, Figure 13 explains the behaviour of boiler and chiller component for all three schemes on operating hours of 15 June (i.e., 9 a.m. to 5 p.m). At the start of cooling operation, inlet temperature to the boiler was already higher than the minimum driving temperature of the chiller due to stored thermal energy in all three schemes. As time passes by, the hot water coming from the chiller combines and transfers heat with the water in the tank whose effect is to decrease the temperature of the fluid flowing towards the chiller and declining behaviour of temperatures at the inlet of the auxiliary boiler was observed. Mechanism of modified flow schemes (S-2 and S-3) serves the purpose here for saving auxiliary energy by allowing the hot water to feed the boiler with higher temperatures than conventional scheme S-1. Figure 13 illustrates the transient behaviour of inlet temperature to the boiler for all schemes throughout the operating hours of the day. It is also understood that the boiler's outlet temperatures will be equal to the minimum required temperature of the chiller at the times when the boiler's inlet is less than 109 ℃ for S-1 and S-2. COP for an optimized scheme is shown as the secondary axis of Figure 13. The dynamic profile of chiller's generator outlet temperatures is displayed in Figure 14 with minor differences in their average values stayed under 103 C. Dotted lines are the average temperatures for boiler inlet for the day in the month of June shown in Figure 13 is the clear and strong clue that why and how boiler consumes energy for heating up the hot water up to the minimum driving temperature (109 ℃) for the chiller in descending order of S-1, S-2 and S-3.

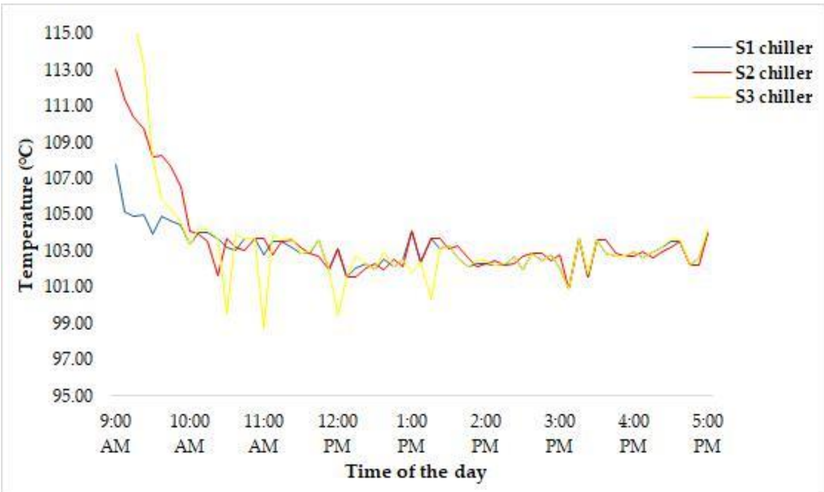


Figure 14. Outlet temperature profiles for all flow schemes at chiller's generator.

In Figure 10, the probable reason for narrowing a gap between flow schemes curves (at higher collector areas) is that the boiler's inlet HTF attains 109 C so early and operates the chiller with higher generator inlet temperatures more than the minimum required temperature.

Returning fluid from chiller in case of S-2 and S-3 would remain above 109 C for higher collector areas, so it does not matter if it is diverted or not because the inlet to the chiller will remain more than 109 C that results in auxiliary unit remain in the OFF condition. The trend of primary energy saving with higher areas of FPC for all schemes is shown in Figure 11, where optimized flow scheme S-3 always performed better than the other two schemes.

Monthly efficiencies in Figure 15 are plotted at optimal thermal storage and collector slope and for the areas of ETC (93 m^2) and FPC (510 m^2) by which 50% primary energy saving would achieve by using flow scheme-3 of the current study i.e., monthly efficiencies of ETC are plotted for 93 m^2 area, the optimal thermal storage size of 15 L/m^2 and 9º collector tilt. Similarly, monthly efficiencies of FPC are plotted for 510 m^2, optimal storage size of 20 L/m^2 and collector tilt of 15º. In Figure 16 results of all proposed flow schemes of the study are summarized in term of collector area per kW of cooling.

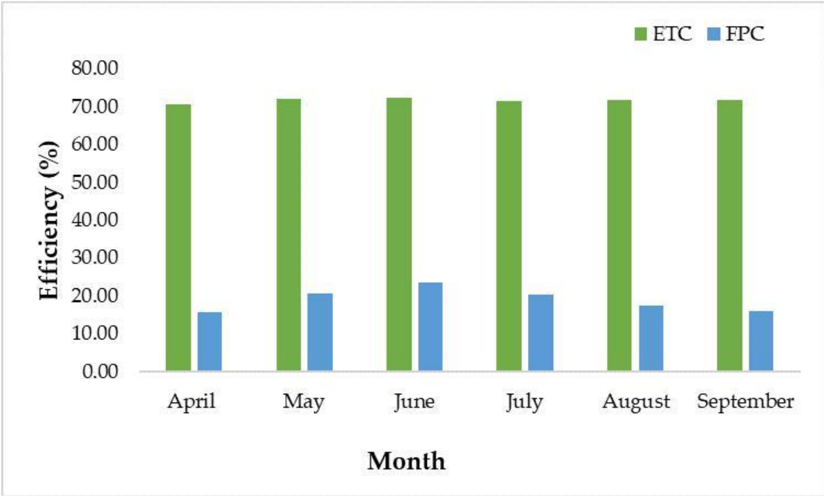


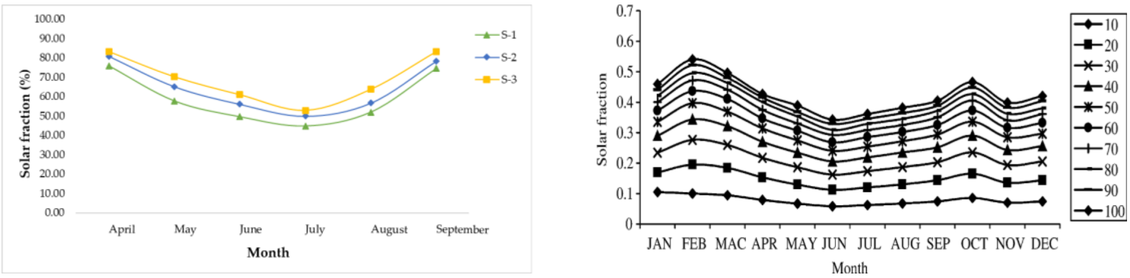
Figure 15. Monthly efficiencies of ETC and FPC for optimized flow scheme (S-3).

Figure 16. Collector area per kW of cooling capacity.

6. Validation of Results

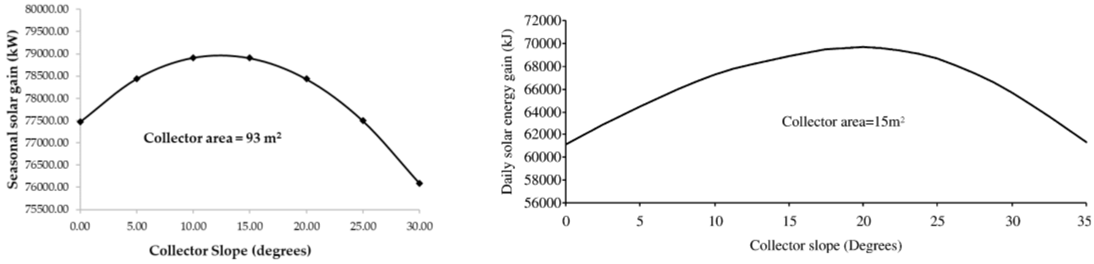
Monthly solar fraction curves obtained from three schemes of the current study as shown in Figure 17a was compared with a study by Assilzadeh et al. [7], shown in Figure 17b and we found similarity of the trends for the months of summer season. It can be seen from Figure 17b that the solar fraction is plotted for entire year in previous published article while in current study, results of solar fraction from April to September were evaluated for the parameters of 50% PES and plotted in Figure 17a. Legends (10 to 100) in the already published paper representing different collector areas used. Lower solar fractions at the mid of the season (July) is an indication of higher auxiliary energy consumption and lowest primary energy saving month of the season justifying Equations (8) and (9) of the current study.

Tilt angles of the collector of S-3, using optimum storage size, were plotted against seasonal collector gain obtained for ETC in the current study. As Figure 18a,b exhibited a fairly similar trend for the optimum tilt angle of the collector. Figure 18a matching with Assilzadeh et al. [7]. However, the optimum value of the tilt angle of the solar collector depends upon the specific location, therefore, plots may differ in values (because of different collector areas used) and units, but trends are found to follow similar paths in both current and previous studies).



(**a**) (**b**)

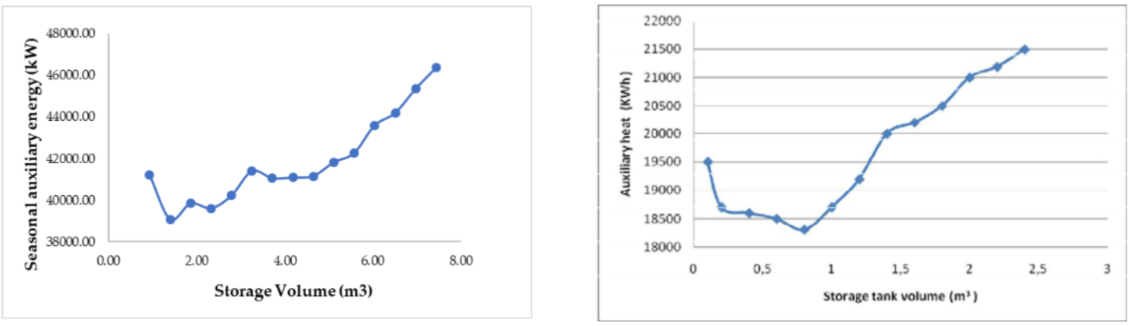
Figure 17. Validation of monthly solar fractions. (**a**) Seasonal solar fraction trend for city of Pakistan. (**b**) Yearly solar fraction trend for city of Malaysia.



(**a**) (**b**)

Figure 18. Validation of optimum slope. (**a**) Useful Collector Gain Vs Collector Slope for city of Pakistan. (**b**) Useful Collector Gain Vs Collector slope for city of Malaysia.

Another similarity of the trend for storage size with auxiliary energy consumption for cooling system is shown in Figure 19, as minimum value of auxiliary energy consumption corresponds to the optimum tank volume. It can be clearly seen that the current study (Figure 19a) is quite similar to the Djelloul et al. [27] study shown in Figure 19b for relation between storage size and auxiliary energy.



(**a**) (**b**)

Figure 19. Validation of optimum storage size. (**a**) Variation of Auxiliary energy consumption with tank volume for climate of Pakistan. (**b**) Variation of Auxiliary energy consumption with tank volume for climate of Algeria.

7. Conclusions

Solar absorption cooling system designed for a peak cooling demand of 108 kW was modeled and simulated by using TRNSYS. The components in the TRNSYS standard library are employed and linked in all possible flow schemes between the storage–chiller loop with different control strategies of the working fluid. An educational building located at Peshawar (34 N, 71.54 E), where a vapor compression cooling system of 108 kW cooling capacity was replaced by SACS, was selected to simulate the system for the whole summertime. The best collector slope was found to be around 9 for ETC and 15 for FPC at the highest solar fraction. Results also indicated a perceptible difference between maximum monthly averaged efficiencies for both collector types as shown in Figure 13. Hence, it can be decided that the flat plate collector is not an ideal option for the presented study because of the requirement of more than five times larger area than ETC. Primary energy saving is much better for Scheme 3 compared to Schemes 1 and 2 with matching storage at somewhat smaller collector areas for ETC. The discrepancy in the fraction of primary energy saving of all schemes shrinkages for higher collector area as shown in Figures 9 and 10, but for real scenarios, economics consideration of the system is also involved while deciding to use higher collector areas. Monthly SF curves for ETC, as shown in Figure 15, in current simulation results also indicated S-3 is far better than the other two schemes to be exploited. In Figure 16, collector area per kW of refrigeration is plotted. Scheme 3 has the smallest area/kW for both ETC and FPC for achieving 50% energy savings. Overall, flow scheme-3 with evacuated tube collector is considered as the best selection for the current study to use with the smallest ratio of collector area to kW of cooling capacity. Because of insufficient experimental records that can be used for direct validation of current results, trends in previous publications of simulation studies were found to be matched with current simulation trends.

Author Contributions: Conceptualization by I.B.B., J.T. and A.W. Methodology, modeling and the discussing of the whole system by I.B.B., J.T. and A.W. Writing- original draft preparation and editing by I.B.B., J.T. and A Y.A. Writing-review and editing by I.B.B., M.A. and A.J. Investigation and general discussion by I.B.B., J.T. and A.W. Overall English and grammar editing by M.A. and A.J. and A.Y.A. Overall Supervision by J.T. and A.W. Funding Acquisition by J.T. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the Guangdong Basic and Applied Basic Research Foundation grant number (2019A1515111189) and Natural Science Foundation of SZU grant number (2110270).

Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

The following abbreviations are used in this manuscript:

VCCS	vapor compression cooling system
HTF	heat transfer fluid
ETC	evacuated tube collector
FPC	flat plate collector
IAM	incident angle modifier
S-1	Scheme-1
S-2	Scheme-2
S-3	Scheme-3
SACS	solar absorption cooling system
PES	Primary energy saving
SF	Solar fraction
COP	Coefficient of performance

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CERTIFICATE OF COMPLETENESS

It is hereby certified that the dissertation submitted by **Iftikhar Bashir Butt**, Reg No. 00000204920, Titled***: "Design & Modelling of Absorption Cooling System integrated with Solar Thermal Storage: Utilizing Solar Energy for Cooling of an Educational Building using TRNSYS"*** has been checked/reviewed and its contents are complete in all respects.

Supervisor's Name: Dr. Adeel Waqas

Signature: ______________

Date: __________________