A COMPARATIVE STUDY OF SMALL SOLAR ABSORPTION AND PV-DRIVEN CHILLER

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A THESIS SUBMITTED AS A PART OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF ENGINEERING IN ENERGY TECHNOLOGY AND MANAGEMENT

THE JOINT GRADUATE SCHOOL OF ENERGY AND ENVIRONMENT AT KING MONGKUT'S UNIVERSITY OF TECHNOLOGY THONBURI

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ABSTRACT

IEA reports that 75% out of total energy consumption in buildings is from the residential subsector, and some studies done in Thailand show that air-conditioning contributes to more than 70% of total electric load from a small house hold. Reducing energy demand is a well researched area due to its impact on environment, to tackle the global issue of scarcity of energy resources, and some countries local issues to reduce energy dependency on other countries. Due to decreasing costs of solar cooling systems, it's believed that they could be used in households and small buildings. This is a comparative study of a solar absorption chiller system and PV-driven chiller system based on the climatic condition of Thailand. This study was done fully by computer simulation using MATLAB with the meteorological data of year 2000. Absorption system was modeled based on the performance curves of a 4.7kW$_{th}$ absorption chiller, while electrical chiller was modeled under the assumption that it operates at constant COP. Life cycle costing for the solar powered absorption chiller, PV-driven electric chiller and a grid operated electric chiller was also done. The results show that, for small capacity systems, grid powered cooling is economically more feasible while for large capacity systems, solar powered absorption chiller is economically more feasible.

Keywords: Simulation, Solar water collector, Absorption chiller, Photovoltaic panel, Sky model, Life cycle cost.

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NOMENCLATURE

β	: Angle of inclination of the plane	
ρ	: Reflectance	
ρ_{ai}	: Density of air in to cooling tower	
ζ	: Angle between the solar direction and given point of sky	
γ	: Azimuth angle of a given point in the sky	
α	: Altitude of a given point in the sky	
ϕ	: Zenith angle of a given point in the sky	
ε	: Clearness index	
Δ	: Brightness index	
ϵ	: Efficiency of Flat Plate Collector	
RH	: Relative Humidity	
A	: Area	
$E_{subscripts}$	: Irradiance	
subscripts	b	: Beam
	eb	: Beam normal to horizontal plane
	bt	: Beam normal to inclined plane
	ed	: Diffuse on the horizontal plane
	ex	: Extra-terrestrial irradiance solar irradiance
	eg	: Global irradiance on the horizontal plane
	$t\theta$	: Total irradiance on inclined plane
$h_{subscripts}$	: Enthalpy	
subscripts	wi	: Water in
	wo	: Water out
	sat	: saturation
	sat,e	: Effective saturation
$m_{subscripts}$	: Mass flow rate	
subscripts	h	: Hot water
	c	: Cooling water
	a	: Air
	w	: Water
p_{vs}	: Saturation vapor pressure	

NOMENCLATURE (Cont')

p_{vs}	: Saturation vapor pressure	
p_v	: Partial vapor pressure	
P_{pv}	: Power output from PV system	
$Q_{subscripts}$	: Heat	
subscripts	*Coll-loss*	: Collector loss
	r	: Refrigeration heat
	u	: Useful output from collector
	g	: Chiller generator
	CL	: Cooling tower
	tank	: Storage tank
	loss	: Loss from pipe & loss from tank
$T_{subscripts}$	: Temperature	
subscripts	*amb*	: Ambient
	wo	: Cooling tower water out
	wi	: Cooling tower water in
	Collector-out	: Solar collector out
	Collector-in	: Solar collector in
	c	: Chilled water
	ai	: Air input to cooling tower
	ao	: Air output to cooling tower
U	: Overall heat transfer coefficient	
w	: Humidity ratio	

CHAPTER 1
INTRODUCTION

1.1. Background

1.1.1. Location and Climatic Conditions of Thailand

Thailand is located at the tropical region of latitude 15°00’ N and 100° 00’E longitude [1]. In the northern part of Thailand, during mid-May to September the weather can be rainy or warm or cloudy, while dry and cool during November to mid-March. However, for the southern part of Thailand, the climate is usually hot and humid. Hence air-conditioning for cooling is common in the southern region.

1.1.2. Thermal Comfort

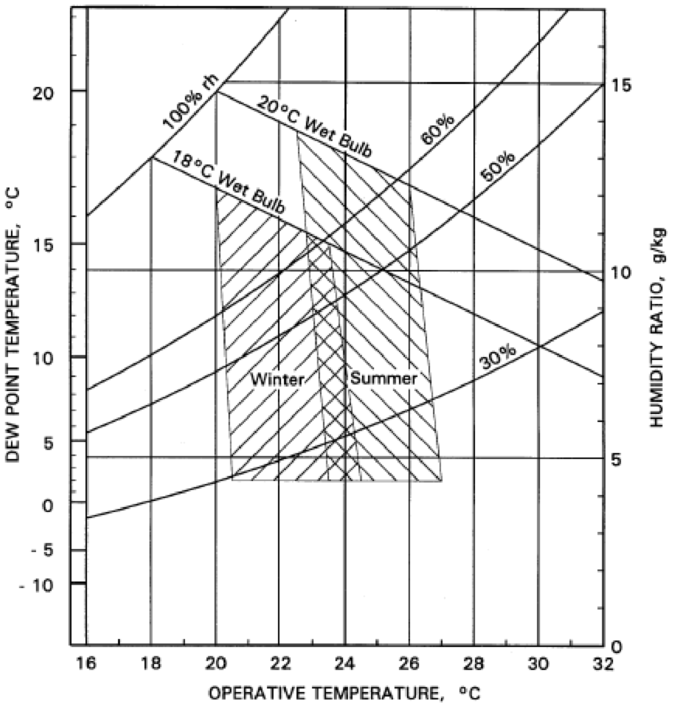


Figure 1-1: ASHRAE Summer and winter comfort zones [2]

Thermal comfort varies from person to person and from location to location. It is a complex function of temperature, humidity, air motion, thermal radiation from local surroundings, amount of clothes worn, and the level of activity being done. Thermal comfort can be achieved at temperatures between 20° C to 27° C, and relative humidities between 20% and 70% under varying air velocities and radiant surface temperatures [3]. Figure below shows generalized comfort zone of dry bulb temperature and humidities plotted on a psychrometric chart.

1.2. Rationale

The International Energy Agency (IEA) reported in Energy Technology Perspectives (ETP) for 2012 that 75% out of total energy consumption in buildings was from the residential sub-sector [4]. Chirarattananon and Hien (2011) report that air-conditioning contributes to more than 70% of total electric load in a small household [5]. Reducing this load while not compromising the thermal comfort of the building occupants will reduce the power demand whilst reducing the power generated.

Solar cooling is becoming popular due to environmental problems and energy costs. This study is a comparative study of a solar-powered absorption chiller and PV-driven electric chiller. This section looks into the importance of reducing the energy demand.

1.2.1. Importance of Reducing Cooling Energy Demand

1.2.1.1. Environmental Impact of Air- conditioning

From an environmental point of view, air-conditioning is uniquely catastrophic. Indirectly, large amounts of energy consumed for air-conditioning result in large quantities of CO_2 emissions, while directly contributing to global warming by the leakage of refrigerants (CFCs and HCFCs), which are powerful greenhouse gases. These refrigerants can also deplete the atmospheric ozone layer. Even though new refrigerants are being introduced which are more environmental friendly, the indirect emissions of CO_2 by air-conditioning outweigh the direct impacts of it mentioned above [6].

1.2.1.2. Reducing Energy Demand from Global Point of View

Global warming is one of the major concerns in today's world, and to achieve thermal comfort, the temperature at which a person's body feels comfortable depends on the surrounding temperature for the past two to three months. A study conducted in Michigan in 2013 to see whether people personally experience global warming shows that less than 6% reported that they strongly agreed they had personally experienced global warming, while 21% were less sure even though they believed that they had personally experienced it [7]. NASA Earth Observatory reports in 2010 of a 0.7^{o} C rise in Earth's temperature in the past century alone [8]. It's true that global warming is a slow process, and not all people can feel it personally, but there are other secondary issues related to this such as melting of the poles, and rising of sea level.

Since the existing technologies to extract energy are not so environmentally friendly, green house gas emissions, such as CO_2 and SO_2, are existing major concerns. IEA in 2012 reports that residential services and sub-sectors consumes approximately 32% of global final energy use, which makes them responsible to 15% of total direct energy-related to CO_2 emission from final energy consumers [4]. Even though use of renewable energy is solution for the green house gas emissions, due to the low maturity of renewable energy technologies, and the inability to cater the current energy demand, renewable energy technologies have to be used combined with the conventional energy technologies. One of the scenarios that could result from using renewable energy technologies is "Zero Emissions". According to ETP 2012, zero carbon-emissions energy system appears within range in 2075 with their 2DS scenario. 2DS refers to an energy system with an emissions trajectory which recent climate science research indicates would give an 80% chance of limiting average global temperature increase to 2^oC [9]. So other means have to be explored, and technological breakthroughs will aid to achieve such a scenario mentioned above.

Another predicament of using conventional energy technologies is the scarcity of the existing energy resources. Surveys are been carried out by organization to predict the limited resources available in the world, one such survey was done on world energy resources by World Energy Council in the year 2010. The survey shows that for Coal, a recoverable reserve in Asia is 212 thousand Mtoe while 188 thousand Mtoe for Europe [10]. Similarly for oil, Middle East holds 107 billion tones of reserve while Latin America and the Caribbean holds 16.9 billion tones of oil reserve.

While all of these facts remain, IEA in ETP 2012 reported that space cooling is highly dependent on income while giving China as an example [11]. It was reported that in China, air-condition rates grew from 2.3% to 61% from 1993 to 2003. Hence, due to the limited energy resources and rise in energy demand, exploiting renewable energy technologies and improving the efficiency of existing technologies are important.

1.2.1.3. Reducing Energy Demand from Local Point of View

Not all countries have their own energy resources within their geographic borderline; Thailand is one of the many countries heavily dependent on imported energy. Often energy is taken for granted, while we fail to realize how much we depend up on it in our homes and working lives. Without it, our high tech computer dependant society will cease to function, our productivity will fall and our gross domestic product (GDP) will be greatly reduced. This could be seen by the power cut that brought California to a halt [12]. UK will be another example where a combination of striking coal miners, power workers and crude oil prices in 1970s forced the UK government to restrict the vehicle speed limits, reduce working days to three days per week in order to save energy [13].

Energy efficient technologies are increasing and becoming more affordable, but this is not enough to bring down the overall energy demand of the country. Increasing population and increasing use of technologies overtake the effort to lower the demand of energy with efficient technologies. According to EPPO (Energy Policy and Planning office of Thailand) 2011 report, total primary energy consumption of 2011 increased by 3.5% from that of 2010, while the total energy import of 2011 was worth THB 1,237.34 billion which is 30.2 % higher than that of 2010 [14]. Another thing to take into consideration is the rise of energy cost. In 2012, EGAT (Energy Generating Authority of Thailand) reported an increase of average electricity sales price from 2.57 Baht/kWh to 2.8 Baht/kWh from 2011 to 2012 [15]. Hence, dependency to other countries for Energy demand needs to be reduced, to reduce the effect on the country in cases like Middle East oil crisis in 1970s [16].

1.2.1.4. Reducing Energy Demand from the Consumer Side

From the demand side, or the consumer point of view, energy tariffs keep on increasing, while the amount of electrical appliances we rely on also keeps increasing. Furthermore, living standard is also something that remains constant or gets higher. Hence, energy bill is always on an increasing trend and it’s important from the consumer side to reduce their energy consumption by increasing the efficiency of the systems used while not compromising there comfort and living standard.

One way of reducing the cooling load is by reducing the heat gain into the cooled space. A study conducted in 2004 reports that the heat gain through building envelopes accounts for up to 60% of the cooling load [17]. Hence, improving wall performance of the air-conditioned building is one approach to tackle the issue. Another way is to use solar energy from a PV system to supply the energy required for the air-conditioning system. Air-conditioning systems that uses solar energy already exist which either uses solar energy in the form of electrical energy from Photovoltaic systems, or which uses solar energy as thermal energy. This paper will be doing a comparative study of PV driven electric chiller and solar powered absorption system.

1.3. Problem Statement

As discussed later in the literature review, the price of photovoltaic and chiller systems is decreasing. Due to this, there is a growing concern that a solar cooling is economically feasible. Therefore, the aim of this study is to do a comparative study of solar powered absorption chiller and PV-driven electric chiller in terms of performance and economic feasibility.

1.4. Literature Review

Bilgili divides solar cooling into to three categories: solar electric refrigeration, solar thermal refrigeration, and solar thermal air-conditioning [18]. First category uses photovoltaic panels to power the conventional refrigeration system. In the second category, refrigeration effect is brought by solar thermal gain and systems such as solar mechanical compression refrigeration. Solar absorption refrigeration and solar adsorption refrigeration fall into this category. Third category is where solar thermal gain is used for air-conditioning by means of desiccant cooling.

The literature review of this paper focuses on the prospects of solar cooling, and also presents studies done on solar-powered absorption and vapor compression systems along with environmental and economical comparisons of solar cooling technologies.

1.1.1. Review of Solar Refrigeration and Cooling Systems

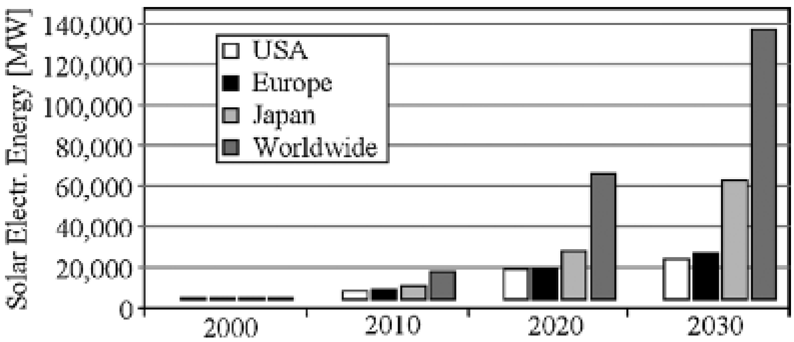


Figure 1-2: Global PV-based solar electricity production for four decades [19]

A study done by Sarbu and Sebarchievici (2013) reviews solar refrigeration and cooling systems [19]. In this study, before the comparison of the existing technologies, the rapidly increasing PV-based electricity generation alongside the conventional power plants in the past two decades is highlighted. The Figure **1-2** shows the PV-based electricity production for four decades

Sarbu and Sebarchievici conclude that the next few years will be the most decisive for the success of solar cooling systems. It depends on the policymakers and the manufacturers. Policymakers' role will be to offer promotional schemes for the consumers, while the manufacturers' role will be to improve cost efficiency and search for environment friendly working fluids which operate in lower temperatures.

1.1.2. Solar Refrigeration for Rural Applications

Enibe (1997) reviews alternative refrigeration technologies for remote locations where grid electricity is unavailable [20]. In this paper, the author reports from [21] that over 800 photovoltaic powered refrigerators were reportedly built between the year 1981 and 1986 specifically for storage of medicines and vaccines for demonstrational purposes. Enibe also reports from [21] of fully tested nine installations in Mali with a percentage operational time of 27 to 79% with 4 to 20 months of average mean time between failures. Furthermore, the system reliability was reported to be 59%, and the system cost being pointed to be the major problem with Photovoltaic powered refrigeration system.

The author concludes the paper referring back to the successful installations of solar cooling technologies proving its technical feasibility. Author also mentions the prospect of solar energy technologies becoming competitive with the conventional technologies in the future either by the increase in system performances and decrease in solar technology cost, or by the rise of energy prices.

1.1.3. Performance Assessment of an Integrated Free Cooling and Solar Powered Single-effect Lithium Bromide-water Absorption Chiller

Ali *et al* (2008), report on the performance assessment done on a solar powered Lithium Bromide-water (LiBr/H20) absorption chiller system, which has been in operation since August 2002 in Oberhausen, Germany [22]. The study was done on a 35.7kW$_{th}$ chiller which uses 108m^2 vacuum tube collectors, 6.8m^3 hot water storage tank, 1.5m^3 cold water storage tank and a 134kW cooling tower. The system was integrated to the heating system to utilize the available hot water as a supplementary source. The cooling system was used for cooling meeting rooms and three offices of the Faunhofer Institute. Here the office rooms required cooling only during working hours while the laboratories required 24-hour cooling.

The assessment was carried out for 5 years (2002-2007), and the assessment shows that the system was able to provide 31,365 kWh_{th} during the 5 years, and the integrated system was able to provide the required cooling demand during the cooling season. In addition to the performance assessment, author concludes that based on the obtained results 4.23 m^2/kW_{th} collector area was required to harvest the maximum possible solar energy available.

1.1.4. Performance Evaluation of Solar Electric-Vapor Compression Refrigeration System

A study conducted by Bilgili in 2011 evaluates the performance of a solar electric-vapor compression refrigeration system [18]. The study was conducted for the climatic conditions of Adana city located in the southern region of Turkey. Hourly average solar radiations and atmospheric temperatures from meteorological data were used in this simulation, and 23rd days of May, June, July August and September months were used.

Bilgili conclude the paper saying that in a Solar Electric Vapor Compression Refrigeration system, required photovoltaic panel area increases as the evaporating temperature decreases. The author's simulation shows that 18.691m^2 photovoltaic panel surface areas is required for an evaporating temperature of 10^oC, while 38.65m^2 surface area is required for an evaporating temperature of -10^oC.

Furthermore, the author adds that by using a solar track system, maximum possible power from the photovoltaic can be obtained. Author also adds that his proposed system can be successfully operated at the tested location, while further study can be done to find a more suitable refrigerant.

1.1.5. Performance Analysis of a Solar Photovoltaic Operated Domestic Refrigerator

Modi *et al*. (2009) evaluated the performance of a solar photovoltaic operated domestic refrigerator [23]. The study was done by modifying a domestic refrigerator by adding a battery bank, inverter and transformer, and solar photovoltaic panels to power the system.

A brief description of the system used by Modi *et al*. (2009) is as follows. Refrigerant used in this system was R-134a. Rated power, rated running current, acceptable voltage and electricity frequency are 110 W, 0.95 A, 160-250 V, 50 Hz respectively. Four 35 W_p solar photovoltaic panels were used with two 12 V- 135 A lead-acid batteries for back-up during times of low solar intensity.

The authors concluded from this study that it was technically feasible to convert an existing refrigerator to a photovoltaic refrigerator. They also conclude that performance of the system under normal conditions is similar to a conventional domestic refrigerator, and they add that the photovoltaic capacity and the battery bank capacity used in this study was the minimum requirement for the system under normal ambient conditions. However, for a sustainable system, larger photovoltaic and battery bank capacity will be required increasing the initial cost and making it economically unviable.

1.1.6. Economical Comparison between Solar-Powered Vapor Absorption Air-Conditioning System and Vapor Compression System

A study done by Elsafty and Al-Daini (2002) in a city in Egypt was to economically compare three systems: solar-powered vapor compression system, and single and double-effect solar vapor absorption systems [24]. The study was focused to choose an optimum system for a hospital's requirement.

The study includes a theoretical analysis of cooling and heating loads, the required conditions for the site, building survey followed by the costs. The costs were further divided into initial cost, operating cost, and maintenance cost. To carry out the economical comparison, in addition to the above mentioned factors, present value cost, estimated annual cost were also considered. Furthermore, global warming impact was also considered since USA Economists estimated a cost for reducing annual carbon emissions [25].

The study concludes that the present worth cost of a vapor compression system is lower than a single effect vapor absorption system while it is higher than that of a double-effect system. Similarly, in terms of equivalent annual cost, cost of vapor compression system is lower than single-effect absorption system but higher than double effect system.

1.1.7. Prospects of Solar Cooling – an Economic and Environmental Assessment

An economic and environmental assessment was done by Otanicar *et al.* (2012) on the prospects of solar cooling [26]. This assessment was done for typical single-family residences in southwestern USA climatic conditions. In this assessment, the authors present a current and projected component and storage costs of solar technologies as shown in Figure **1-3**.

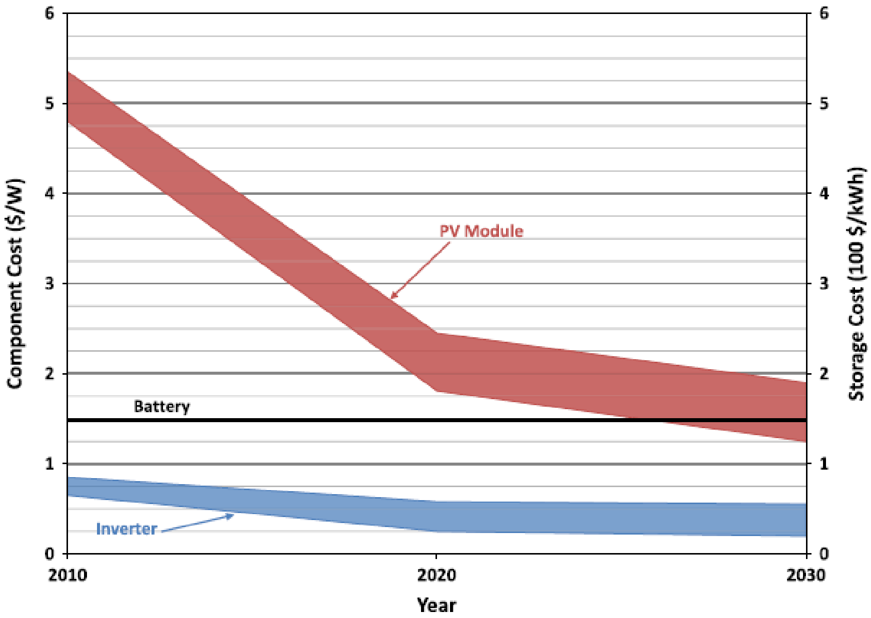


Figure 1-3: Current and projected component and storage costs of solar electric technologies [26]

Conclusions drawn from this paper are that in the case of solar electric cooling, system costs highly depend on system COP when the prices remain at the current levels, but when the prices become lower, as projected in Figure **1-3**, the impact of COP is diminished. This is because PV cost represents 50% to 60% of the overall system cost. Secondly, for the case solar thermal cooling, solar collection represents a minor portion of the overall system cost and that cost is not projected to decrease. Therefore, if costs of refrigeration were to come down and thermal refrigeration COP increases, then it could be expected for solar thermal cooling cost to be competitive with solar electric cooling. Authors also conclude that from an environmental view, solar electric cooling has a lower Carbon dioxide emission compared to any of the thermal technologies despite the fact that they use refrigerants with global warming potential. The authors also claim that another advantage of solar electric cooling over solar thermal cooling is the collector area footprint. While solar electric systems expected sizes in 2010 were between 24 and 48 m^2 – depending on the system COP, for solar thermal, the size ranges from 78 to 106 m^2. Hence, even if the cost of solar thermal systems is competitive by 2030, huge improvements on COP and/or thermal collector cost needs to be brought to have favorable shifts.

1.5. Objectives

The main objective of this study is to conduct simulations on a small solar powered absorption chiller system and economically compare it with a PV-driven electrical chiller. Specific objectives which come under the main objective are

- To find a suitable mathematical model for a single effect absorption chiller system and electrical chiller.
- To carry out simulations to see the difference in performance between a direct loop (leave water in the collector at night time) system and an open loop (drain the water from the collector to an insulated drain tank).
- To find a suitable mathematical model for PV system
- To carry out simulations for PV driven electric chiller
- To find life cycle cost single effect absorption system, PV-driven electric chiller and electric chiller operated using grid power.

1.6. Scope of Research Work

To conduct simulations, the work scope includes sizing and configuring the system used in the study. MATLAB programs were developed for the mathematical models to carry out the simulation. Use yearly meteorological data for the simulation program to predict the yearly performance and economic feasibility compared to solar thermal cooling systems.

CHAPTER 2
THEORIES

2.1. Sky Models

2.1.1. Uniform Sky Model

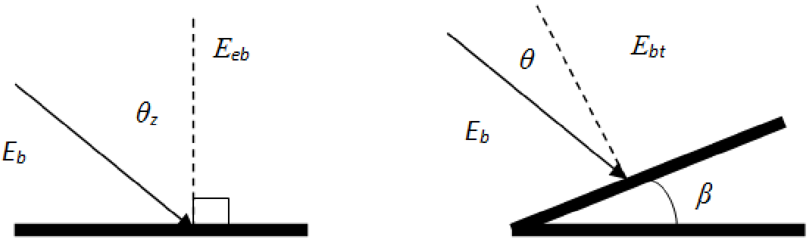


Figure 2-1 : R_b factor

$$R_b = \frac{E_{bt}}{E_{eb}} = \frac{E_{eb}\cos\theta}{E_{eb}\cos\theta_z}, \tag{1}$$

With R_b defined in Equation (1), the total solar radiation on an inclined plane can be calculated by Equation (2). The first term in the Equation (2) gives the beam component of the radiation on the tilted surface, second component gives the diffuse component, and the third term gives the contribution from the ground reflected radiation [27].

$$E_{t\theta} = E_{eb}R_b + \left(\frac{1+\cos\beta}{2}\right)E_{ed} + \rho\left(\frac{1-\cos\beta}{2}\right)E_{eg}, \tag{2}$$

Where,

$E_{t\theta}$	:	Total radiation on tilted plane,
E_{eb}	:	Beam radiation falling on horizontal plane,
E_{ed}	:	Diffuse radiation falling on horizontal plane,
E_{eg}	:	Global radiation falling on the horizontal plane.

2.1.2. ASRC-CIE Sky Model

The ASRC-CIE sky model is used in this study[28]. The model was designed by Perez *et al*. (1990) [28], and later modified (1992) [29]. This model is linear combination of four other sky models which is described in this section. Sky clearness and sky brightness factors are used to determine the linear combination coefficients. Originally the model is for sky illuminance, but the author claims that with certain approximation, the model could be used for sky irradiance. Description of the models were referred from [30] and [31].

Below is some of the angle variables used in describing the models.

ζ is the angle between direction to the solar zenith and direction to the given point

γ is the azimuth angle of a given point in the sky

α is the altitude angle of a given point in the sky

ϕ is the zenith angle of a given point in the sky

ϕ_s is the solar zenith angle

γ_s is the solar azimuth angle

α_s is the solar altitude angle

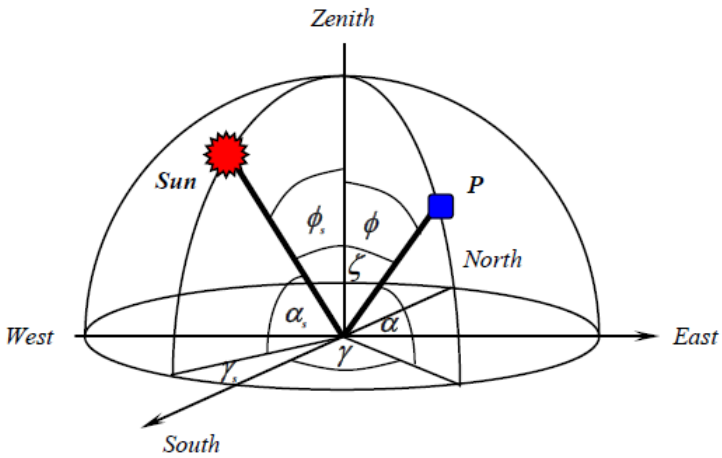


Figure 2-2: Angles defining the position of a sky element [31]

The model is described as follows.

$$E_{ed} = b_{cl}E_{cl} + b_{tcl}E_{tcl} + b_{in}E_{in} + b_{oc}E_{oc} \tag{3}$$

The terms b_{cl}, b_{tcl}, b_{in}, and b_{oc} are coefficients that depend on Perez's clearness index (ε) and brightness index (Δ), defined as:

$$\varepsilon = \frac{(E_{ed} + E_{eb})/E_{ed} + 1.041\phi^3}{1 + 1.041\phi^3} \tag{4}$$

$$\Delta = \frac{m.E_{ed}}{E_{ex}} \tag{5}$$

where E_{ed} is the diffuse horizontal irradiance,

E_{eb} is the beam normal irradiance,

for non-zero solar altitude angle,

$$m = \frac{1}{\sin\alpha_s},$$

E_{ex} is the extra-terrestrial solar radiation, 1353 W.m^{-2}.

In using Equation (3), the following are the conditions of Perez's clearness index.

If ε < 1.4 then

b_{in} is the positive and larger value between $\frac{\varepsilon - 1}{0.4}$, and $\frac{\Delta - 0.15}{0.6}$ where b_{in} is limited to 1.0,

$$b_{oc} = 1 - b_{in}, and$$
$$b_{cl} = b_{tcl} = 0$$

If 1.4 < ε < 3, then

$$b_{tcl} = \frac{\varepsilon - 1.4}{1.6},$$
$$b_{in} = 1 - b_{tcl}, and$$
$$b_{cl} = b_{oc} = 0$$

If ε > 3, then

$b_{cl} = \frac{\varepsilon - 3}{3}$, and *if* ε > 6, b_{cl} =1 , and

$$b_{tcl} = 1 - b_{cl},$$
$$b_{in} = b_{oc} = 0,$$

The subscripts *cl, tcl, in,* and *oc* refer to *Clear Sky, Turbid Clear Sky, Intermediate Sky,* and *Overcast Sky* models, respectively. These standard sky models are given below.

Clear Sky Model

$$\frac{E_{cl}}{E_z} = \frac{\left[1 - e^{\left(-0.32/\cos\phi\right)}\right]\left[0.91 + 10e^{(-3\zeta)} + 0.45\cos^2\zeta\right]}{\left[1 - e^{-0.32}\right]\left[0.91 + 10e^{-3\phi_s} + 0.45\cos^2\phi_s\right]} \tag{6}$$

where E_z is the irradiance at the zenithal point of the sky.

The following equations define the angular distance A_z between the sky element and the Sun.

$$\zeta = \arccos(\cos\phi_s.\cos\phi + \sin\phi_s.\sin\phi.\cos A_z) \tag{7}$$

$$A_z = |\gamma - \gamma_s| \tag{8}$$

Turbid Clear Sky Model

$$\frac{E_{tcl}}{E_z} = \frac{\left[1 - e^{\left(-0.32/\cos\phi\right)}\right]\left[0.856 + 16e^{(-3\zeta)} + 0.45\cos^2\zeta\right]}{\left[1 - e^{-0.32}\right]\left[0.856 + 16e^{-3\phi_s} + 0.45\cos^2\phi_s\right]} \tag{9}$$

Intermediate Sky model

$$\frac{E_{in}}{E_z} = \frac{(num1)(num2)}{(deno)} \tag{10}$$

Where:

$$num1 = 0.4399\left[\alpha + 4.799 + 1.35\{\sin(3.59\alpha - 0.009) + 2.31\}\sin(2.6\alpha_s + 0.316)\right]$$

$$num2 = e^{\left[-0.563\{(\alpha+1.059)(\alpha_s-0.008)+0.912\}\zeta\right]}$$

$$den = 0.988\{\sin(2.6\alpha_s + 0.316) + 2.7358\}e^{\{-1.48058(\alpha_s+0.300769)\alpha_s\}}$$

Overcast Sky Model

$$\frac{E_{oc}}{E_z} = \frac{1}{3}(1.2\cos\phi) \tag{11}$$

2.2. Vapor Compression System

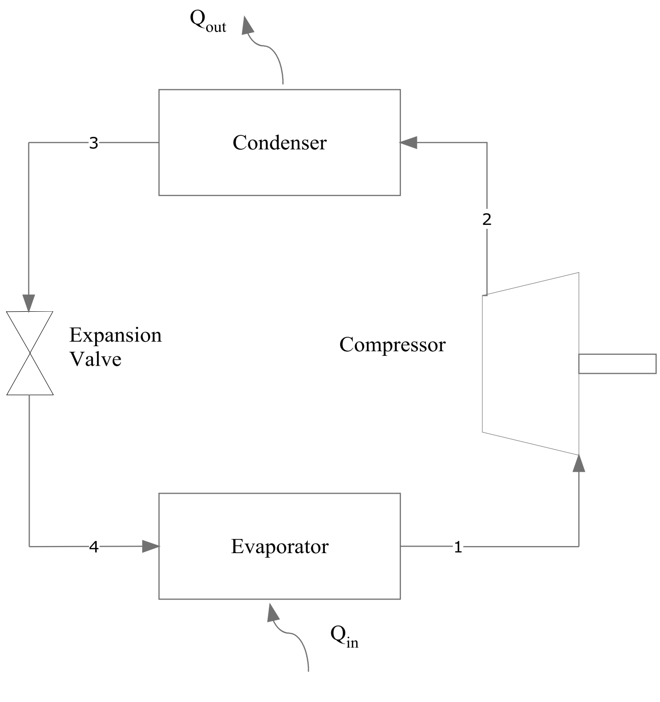


Figure 2-3: Vapor compression system

A vapor compression system is comprised of four major components: evaporator, compressor, condenser, and expansion valve. The functions of the components and the process carried out within the component are as follows.

2.2.1. Evaporator

This is the point where heat is captured in a refrigeration system, providing the required cooling effect. Evaporators are divided into two categories: direct cooler evaporators that cools air (air and gas coolers) which then cools the product, and indirect cooler evaporators that cool a liquid (liquid coolers) which then cools the product [32]. Generally, Higher COP results from a higher evaporating temperature, and it's estimated that a rise in 1^{o}C of evaporating temperature yields an operating cost reduction of 2 to 4% [33].

2.2.1.1. Evaporation Process

Evaporation, unlike freezing or melting, can take place at any temperature. After a considerable amount of heat is absorbed, without any change in temperature, evaporation may occur, and refrigerants at all temperatures can have increased rates of evaporation at higher temperatures. The refrigerant absorbs the heat from the medium or matter to be cooled changing phase from liquid to vapor, eventually boiling and forming low-pressure saturated vapor.

2.2.2. Compressors

In a refrigeration cycle, compressors have two main functions. One is to maintain the desired pressure and temperature at the evaporator by pumping the refrigerant from the evaporator. The other is to increase the refrigerant vapor pressure through compression and simultaneously increase its temperature. Compressors are referred as the heart of the vapor compression cycle [32].

2.2.2.1. Compression Process

A compressor raises the pressure of the refrigerant vapor obtained from the evaporator, by means of shaft work and the addition of heat. This increase in pressure raises the boiling and condensing temperature of the refrigerant and upon sufficient compression, raising the boiling point temperature higher than the heat sink temperature.

2.2.3. Condenser

There are different types of condensers, such as air-cooled, water-cooled, shell and tube, shell and oil, tube within a tube, and evaporative condensers. Each type of condenser has its own application. Low condensing temperature results in lower operating costs, and it's estimated that drop of 1^{o}C in condensing temperature reduces operating cost by 2 to 4% [33].

2.2.3.1. Condensation Process

Condensation is the process where vapor changes to liquid by extracting heat. When there is high pressure, high temperature refrigerant from the compressor comes to the condenser, heat is transferred from the refrigerant to the heat sink due to the temperature gradient. The heat transfer condenses the vapor to high pressure saturated liquid, and the heat source will be cooled by dumping the heat to the heat sink.

2.2.4. Throttling Valve or Expansion Valve

This is a device used to reduce the refrigerant condensing pressure to evaporating pressure by a throttling operation and to regulate the liquid-refrigerant flow to the evaporator to match equipment load characteristics. In the throttling process, the refrigerant pressure is reduced, together with the boiling temperature of the refrigerant to below the temperature of the heat source.

2.3. Vapor Absorption Chiller

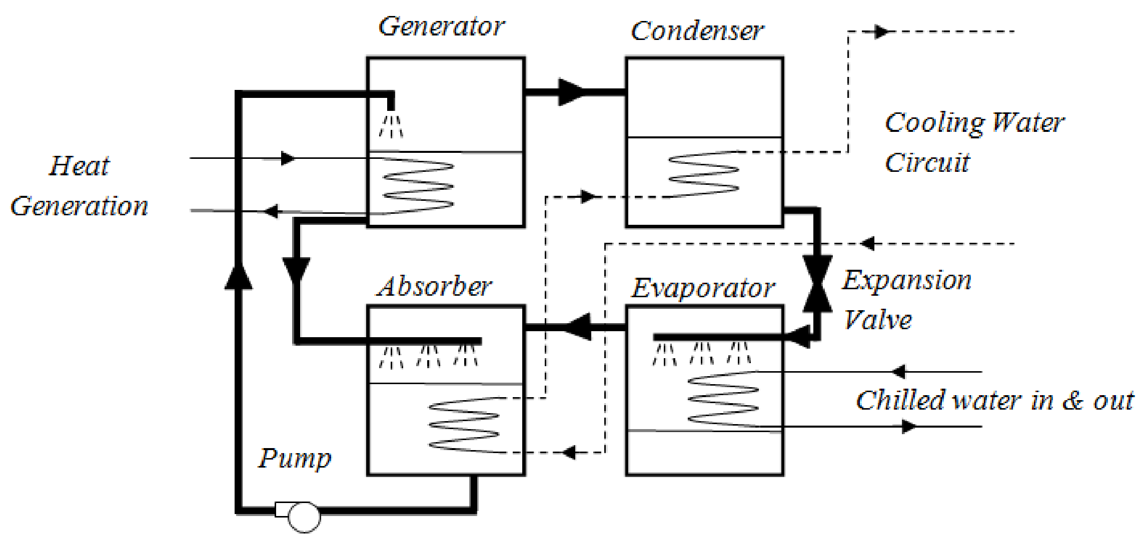


Figure 2-4: Absorption chiller diagram

A vapor Absorption Chiller system can be divided in five major components. They are the heat generator, condenser, expansion valve, evaporator and the absorber. The processes taking place in condenser, expansion valve and evaporator is the same as that of in a vapor compression cycle. Generator and absorber together takes the place of compressor of a vapor compression cycle [34].

At the generator, when the absorbent-refrigerant comes in, the refrigerant is driven off to the condenser and the absorbent returns back to the absorber to absorb the refrigerant again for the next cycle. In absorption system, water can be used as a refrigerant and hygroscopic material such as Lithium bromide (LiBr) can be used as the absorbent.

Unlike a vapor compression system, in an absorption system, cooling water is required for both condenser and absorber. The energy balance of the sub-systems could be given as:

At the absorber,

$$\begin{bmatrix} heat\ of \\ entering \\ refrigerant \\ vapor \end{bmatrix} + \begin{bmatrix} heat\ of \\ entering \\ absorbent \end{bmatrix} - \begin{bmatrix} heat\ of \\ leaving \\ absorbent + \\ refrigerant \end{bmatrix} = \begin{bmatrix} heat\ to\ be \\ removed\ at \\ absorber \end{bmatrix}$$

At the generator:

$$\begin{bmatrix} heat\ of \\ leaving \\ refrigerant \\ vapor \end{bmatrix} + \begin{bmatrix} heat\ of \\ leaving \\ absorbent \end{bmatrix} - \begin{bmatrix} heat\ of \\ entering \\ absorbent + \\ refrigerant \end{bmatrix} = \begin{bmatrix} heat\ to\ be \\ supplied\ to \\ generator \end{bmatrix}$$

At the condenser:

$$\begin{bmatrix} heat\ of \\ entering \\ refrigerant \\ vapor \end{bmatrix} - \begin{bmatrix} heat\ of \\ leaving \\ liquid \end{bmatrix} = \begin{bmatrix} heat\ to\ be \\ removed\ at \\ condenser \end{bmatrix}$$

2.4. Life Cycle Costs

In order to calculate the life cycle costs, the Series Present Worth Factor (SPWF) and Capital Return Factor (CRF) are calculated by Equations (12) and (13). Annual cost and present value of the systems will then be calculated by the following equations.

$$SPWF(i,n) = \frac{(1+ {}^{i}\!/_{100})^{n} - 1}{({}^{i}\!/_{100})(1+ {}^{i}\!/_{100})^{n}} \tag{12}$$

$$CRF(i,n) = \frac{1}{SPWF} \tag{13}$$

$$Annual\ Cost = (investment \times CRF) + Annual\ Electricity\ Cost \tag{14}$$

$$Present\ Value = SPWF \times Annual\ Cost \tag{15}$$

CHAPTER 3

METHODOLOGY AND WORKING SCHEDULE

3.1. Solar Radiation on an Inclined Plane

Using the ASRC-CIE sky model and the measured station data, a sky dome was created with 145 points, such that the summation of irradiance of these 145 points at an instant are equivalent to the measured diffuse sky irradiance at that instant.

Once these 145 points and their contributions to a horizontal plane are known, these points are used to obtain the total diffuse irradiance on the inclined plane. The inclined plane was set at 13.6 ° facing south. As shown in Figure **3-1**, the angle between the vector of respective sky point and the normal vector of the inclined plane can be evaluated by Equation (16) and the normal component of sky point to the inclined plane by Equation (17). The sky model will be used to evaluate the diffuse irradiance which will be summed with bream irradiance and ground reflected irradiance to get the total irradiance.

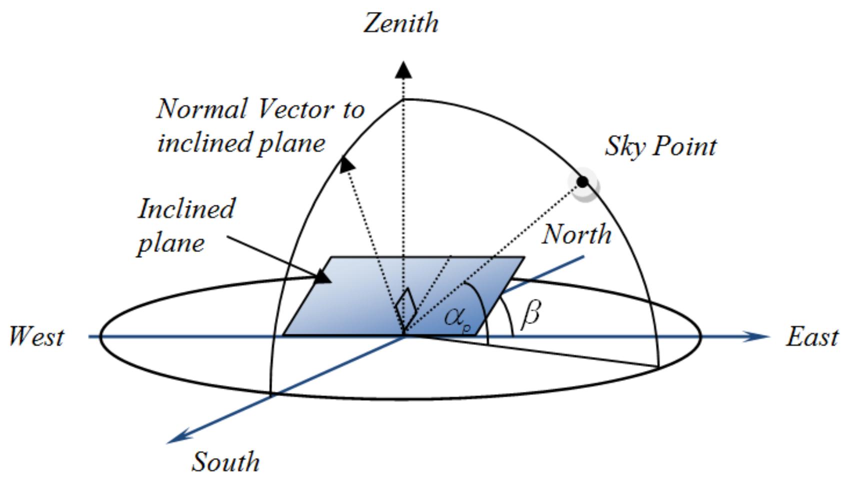


Figure 3-1: Sky point contribution on inclined plane

$$\cos(a_{pn}) = \frac{V_{pn}.V_{plane-n}}{\|V_{pn}\|\|V_{plane-n}\|} \tag{16}$$

$$V_p \, normal \ to \ plane = V_p.\cos(a_{pn}) \tag{17}$$

3.2. Description of the System

3.2.1. Vapor Absorption System

3.2.1.1. System Configuration

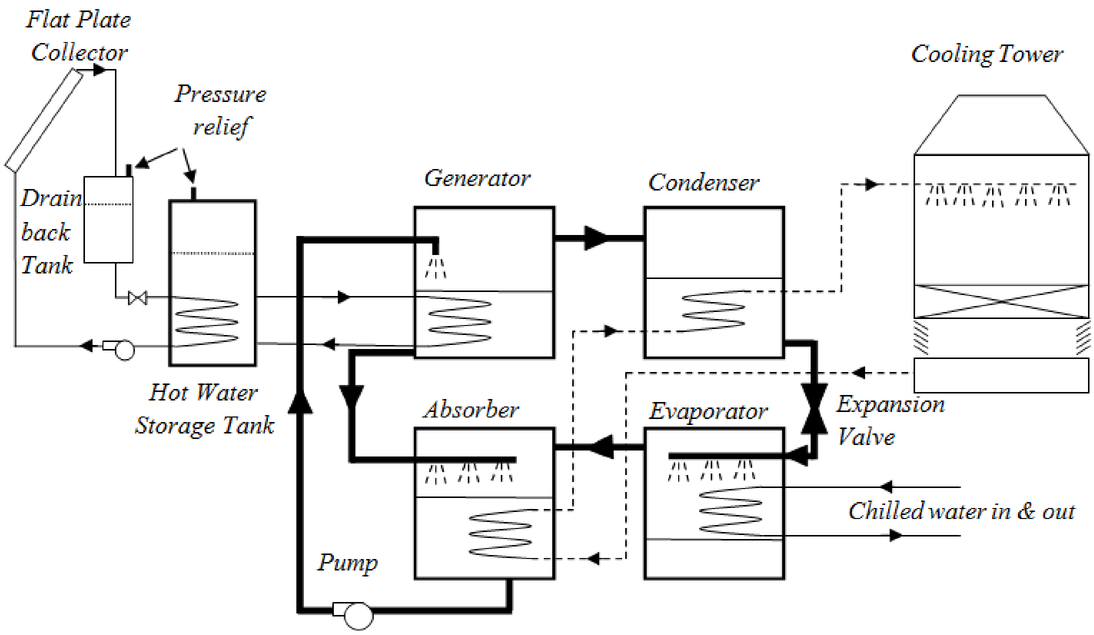


Figure 3-2: Absorption system configuration

The absorption system was configured as shown in Figure **3-2**. For the simulation, 2 scenarios are considered: one was an open loop system as shown in Figure **3-2**, and a direct loop system, where there is no drain back tank, so that the hot fluid remains in the collector when not in operation.

3.2.1.2. Sizing the System

a. Absorption Chiller

A single effect Li-Br chiller system with 4.7kW rated power was used in this study. The specifications of the system were obtained from a model developed by Li and Sumathy (2001) for a chiller manufactured by Yazaki (Chiller model: WFC-400S)[35]. The chiller had a generator inlet temperature range of 75-100°C and cooling water inlet temperature range of 24-31°C. The system was said to achieve its rated power under the following conditions:

- Generator inlet temperature=88°C
- Cooling water inlet temperature=29.5°C
- Chilled water outlet temperature=9°C.

b. Solar hot water collector

A collector area of around $38m^2$ ($8.0m^2/kW_{th}$) was used by García Cascales *et al.* (2011) in Spain [36] and another system of similar size was used by Best and Ortega (1999) from a study by Yueng *et al.* (1992) in Hong Kong [37]. The collector area reported by by Eicker & Pietruschka (2009) was 0.2 to $5m^2/kW_{th}$ with $2.5m^2/kW_{th}$ being the average [38]. Figure **3-3** shows the average Solar Insolation figures obtained for the above referred 2 locations and Bangkok. They were obtained from irradiance calculator in Solar Electricity Handbook which gives the average insolation from data collected over a 22 year period [39]. The figure shows even though Spain has a higher peak of average insolation, they need a larger collector area to overcome the times with lower insoaltion, while Bangkok, the insolation remains at a more steady value when compared to Spain. Germany follows a similar trend to Spain. As for Hong Kong, the insolation is much lower than Bangkok.

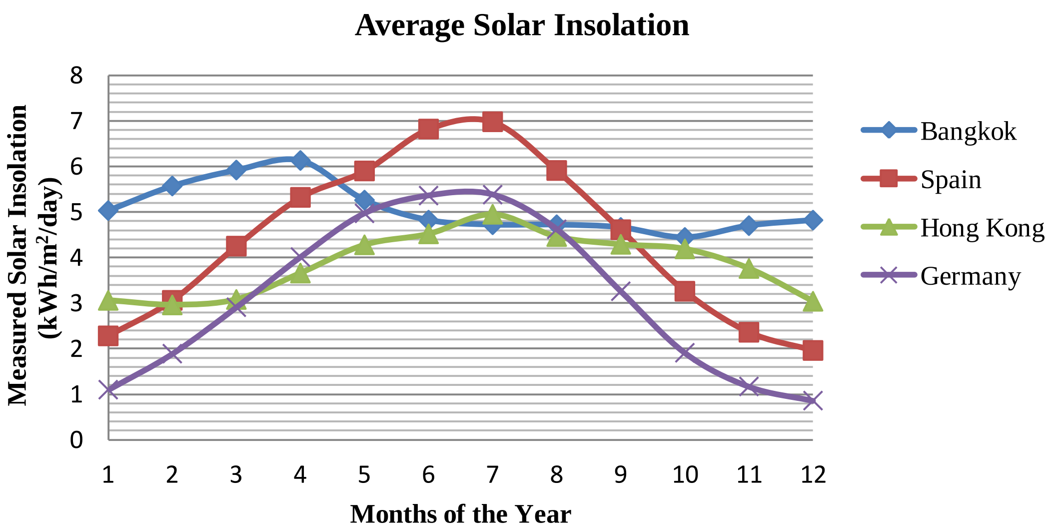


Figure 3-3: Average insolation for Bangkok, Beirut and Hong Kong

A proposed system for a 15,000 seating capacity stadium in Bangkok uses $3.57m^2/kW_{th}$; however, this system is capable of working as single effect or double effect, depending on the thermal energy, and uses compound parabolic concentrator medium thermal evacuated tube collectors [40]. In this study, $38m^2$ was used as a base system, and simulation was also done for $20m^2$ ($4.5m^2/kW_{th}$), $23m^2$ ($4.9m^2/kW_{th}$) and $26m^2$ ($5.5m^2/kW_{th}$).

c. Hot Water Storage Tank

In Beirut, Ghaddar *et al.* (1997) showed that 1000 to 1500 liters tank area is required to run a solar absorption system for 7 hours a day[41]. Atmaca and Yigit (2003) also agree on the size [42]. For this study, simulation was carried out using a 1500 liters tank and a 300 liters tank since 200 to 300 liter tanks are common for domestic purpose.

3.2.1.3. Modeling the System

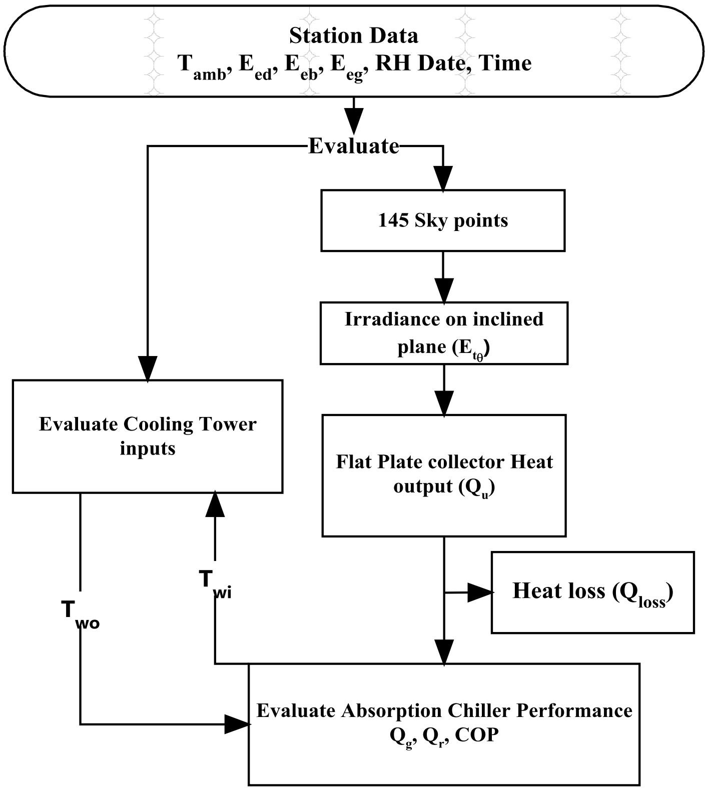


Figure 3-4: Overall flow diagram of absorption system model

The overall flow diagram of the absorption system model is shown in Figure **3-4**. First the thermal energy will be evaluated with the Sky Model and Flat Plate collector model using station data. Then the absorption system performance will be evaluated using the thermal energy input, and the cooling tower output which is also evaluated using the station data. Description of the models will be discussed individually later.

a. Solar Water Collector Model

Since a single effect absorption system was used for the simulation, a flat plate solar collector was used. Equation (18) shows the efficiency curve of the flat plate collector used (TitanPower-ALDH29 Flat Plate Solar Collector) [43].

$$\in = 0.732 - 3.1239\frac{(T_{Collector-out} - T_{amb})}{E} - 0.0143\frac{(T_{Collector-out} - T_{amb})^2}{E} \quad (18)$$

The useful heat gained from the collector is given by Equation (19), which could be written as Equation (20).

$$Q_u = mc_p(T_o - T_i) = AE \in \quad (19)$$

$$m_h c_w(T_{Collector-out} - T_{Collector-in}) = AE[\eta - a_1\frac{(T_{Collector-out} - T_{amb})}{E} - a_2\frac{(T_{Collector-out} - T_{amb}}{E} \quad (20)$$

Let $x = T_{Collector-out} - T_{amb}$, and Equation (20) can be reduced to a quadratic equation in the form of Equation (21). Equation (21) can now be solved as expressed in Equation (22) where the positive solution is the only real solution.

$$Aa_2x^2 + (Aa_1 + m_h c_{pw})x - [AE\eta + m_h c_{pw}(T_{Collector-in} - T_{amb})] = 0 \quad (21)$$

$$x = -\frac{b}{2a} \pm \frac{\sqrt{b^2 - 4ac}}{2a}$$, where

(22)

$a = Aa_2$, $b = Aa_1 + m_h c_{pw}$, and $c = -[AE\eta + m_h c_{pw}(T_{Collector-in} - T_{amb})]$

During night time, to calculate the standing loss if water left in the collector, the collector was modeled as copper tubes loosing heat to the surroundings. Tube length and surface area was determined by the volume of water the collector hold given in the manufacture data.

$$Q_{coll-loss} = U_{copper}A_{pipe}(T_{hot} - T_{amb}) \quad (23)$$

b. Absorption Chiller Model

A flow diagram of the absorption chiller model is shown in Figure **3-5**. As mentioned earlier, chiller system was modeled by using the performance curves of a chiller used in a study of Li and Sumathy (2001) [35]. Figure **3-9** and Figure **3-10** shows the performance curves used. Outputs of the chiller model were Refrigeration heat (Q_r) and the Coefficient of performance (COP). To perform the simulation, the chilled water output temperature will be set at a certain value, so once the hot water output from the solar collector is determined, and the cooling water input from the cooling tower, Q_r and COP is read from the graph data which will be fed to the program used. Assuming gravitational flow energy input from the generator (Q_g) will be given by Equation (24). Finally, cooling tower input (Q_{CL}) by Equation (25).

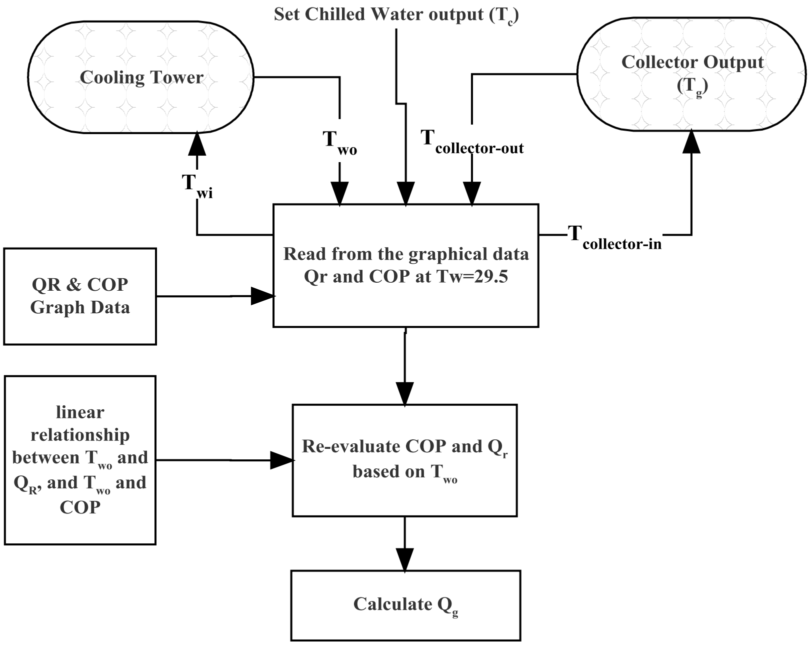


Figure 3-5: Absorption Chiller Model Flow Chart

$$Q_g = \frac{Q_r}{COP} \tag{24}$$

$$Q_{CL} = Q_r + Q_g \tag{25}$$

For simplification, a linear relationship was assumed for the effect of varying the cooling water inlet on COP and Q_r. The linear relationship was derived based on the model outputs of [35] and the performance curves given in [36] which also uses a similar size chiller system in their study. Figure **3-6** shows the curves given in Ref. [36], and Figure **3-7** shows the curves of COP and Q_r redrawn against cooling water inlet (T_{wo}). Note that Figure **3-6** is graph as it is from Ref. [36], and in Ref. [36], T_{wi} is water cooling water inlet to chiller, while in this study T_{wo} is the cooling water outlet to chiller from the cooling tower.

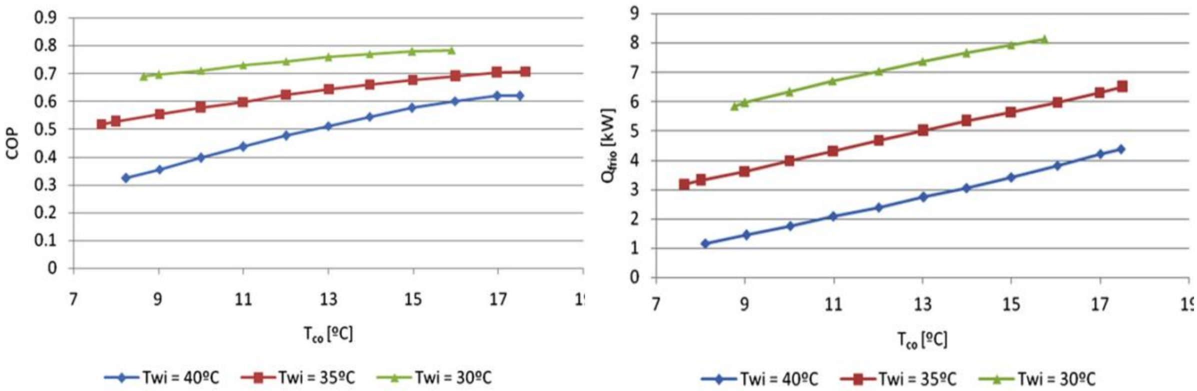


Figure 3-6: COP and Qr curves given in Ref. [36]

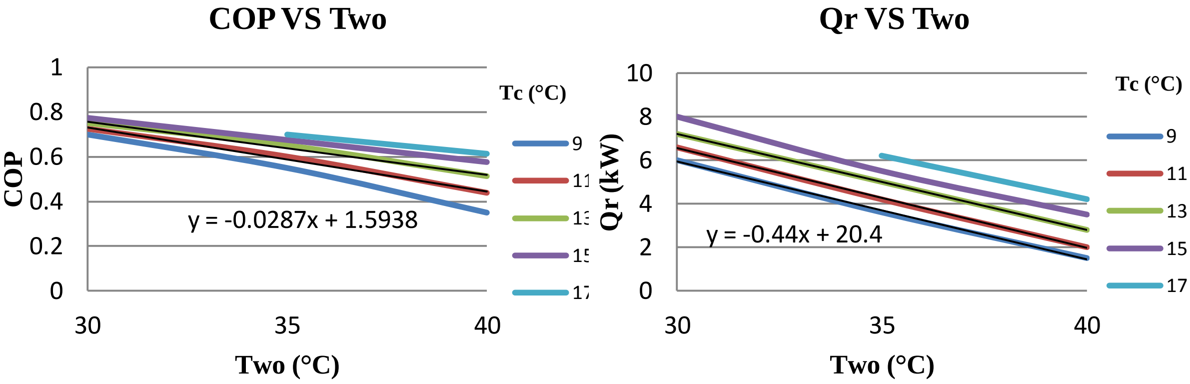


Figure 3-7: COP and Qr curves re-plotted against cooling water inlet temperature

Figure **3-7** shows that a linear relationship could be estimated for the variation of COP and Q_r with condenser water inlet. For example, referring to Figure **3-8**, based on the linear relationship, it could be assumed that each slope represents a particular T_g and T_c. From the information available in Figure **3-9**, the Q_r can be found for a given T_g and T_c at T_{wo} 29.5°C. Using this Q_r and Equation (26), first the $y_{intercept}$ of the Equation (26) will be evaluated to find which line represents the particular T_g and T_c on Figure **3-8**. Once the $y_{intercept}$ known Q_r can be re-evaluated for the T_{wo} calculated from the cooling tower model. The same procedure will be followed to re-evaluate COP at a given T_{wo}, once it's found at 29.5°C.

$$Qr = -0.44 \times T_{wo} + y_{\text{intecept}} \tag{26}$$

$$COP = -0.0288 \times T_{wo} + y_{\text{intecept}} \tag{27}$$

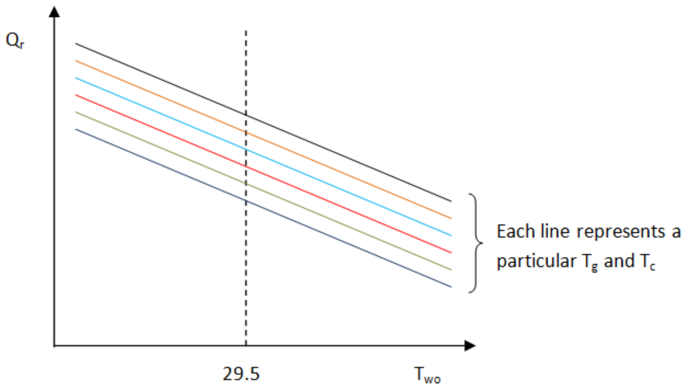


Figure 3-8: Re-evaluating Q_r for new T_{wo}

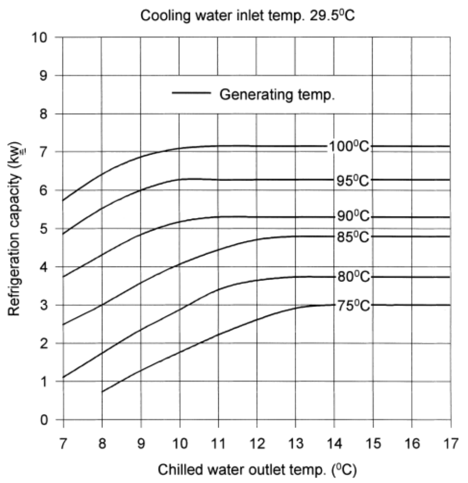


Figure 3-9: Refrigeration heat of chiller

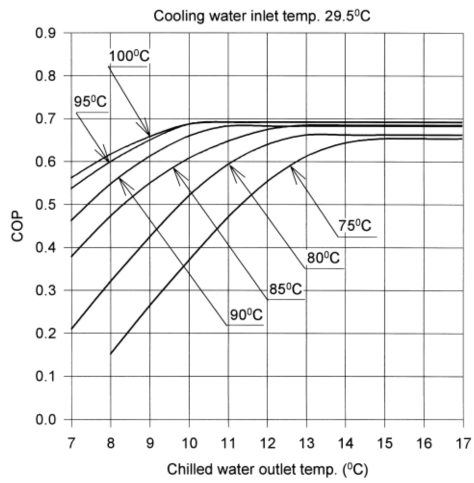


Figure 3-10: Refrigeration COP of the chiller

c. *Cooling Tower Model*

Cooling tower models were very complicated until Braun *et al.* (1989) found a correlation between Number Transfer Units NTU and the ratio of mass flow rate of cooling water by the mass flow rate of air given by Equation (28) [44]. In this equation, the coefficient *a* has a value between 0.5 to 5, and *n* has a value between -0.1 to 0.65. The model used in this study is illustrated in Chirarattananon (2005) [45] which uses correlation of Braun *et al*, and the values of a and n used in this study are 0.9325 and 0.4887 respectively. Inputs for the model are ambient temperature (T_{amb}), relative humidity (RH), water inlet temeperature (T_{wi}), air flow rate and mass flow rate. Figure **3-11** shows the flow diagram of the model.

$$NTU = a\left(\frac{m_c}{m_a}\right)^n \tag{28}$$

For a given T_a and RH, the enthalpy of air is evaluated by using the set of equations below [46], where RH=100% for saturated enthalpies.

$$p_{vs}(Ta) = \exp[53.5224 - \frac{6834.27}{Ta} - 5.17\ln(Ta)] \tag{29}$$

$$p_v = p_{vs} \times RH \tag{30}$$

$$w = \frac{0.622 p_v}{101.3 - p_v} \tag{31}$$

$$h = 1.006T_a + (25501 + 1.805T_a)w \tag{32}$$

The ideal case of heat transfer at a cooling tower is modeled with the enthalpy of air leaving as the enthalpy of saturated air at the temperature of inlet water, as given in Equation (33). Effectiveness of heat transfer is given as ε.

$$Q_{CL} = \varepsilon m_a (h_{sat,wi} - h_{ai}) \tag{33}$$

Additional relations used in the cooling tower model include the Number of Transfer Units (NTU) which is given by Equation (34).

$$NTU = \frac{\ln[(1-\varepsilon)/(1-\varepsilon R)]}{1-R} \tag{34}$$

$$R = \frac{m_a C_s}{m_c C_{pw}}, \text{ where } C_s = \frac{h_{sat,wi} - h_{sat,wo}}{T_{wi} - T_{wo}} \tag{35}$$

Additional relations used in the model are given below.

$$h_{ao} = h_{ai} + \varepsilon m_a (h_{sat,wi} - h_{ai}) \tag{36}$$

$$h_{sat,e} = h_{ai} + \frac{h_{ao} - h_{ai}}{1 - \exp(-NTU)} \tag{37}$$

$$W_{ao} = W_{sat,e} + (W_{ai} - W_{sat,e})\exp(-NTU) \tag{38}$$

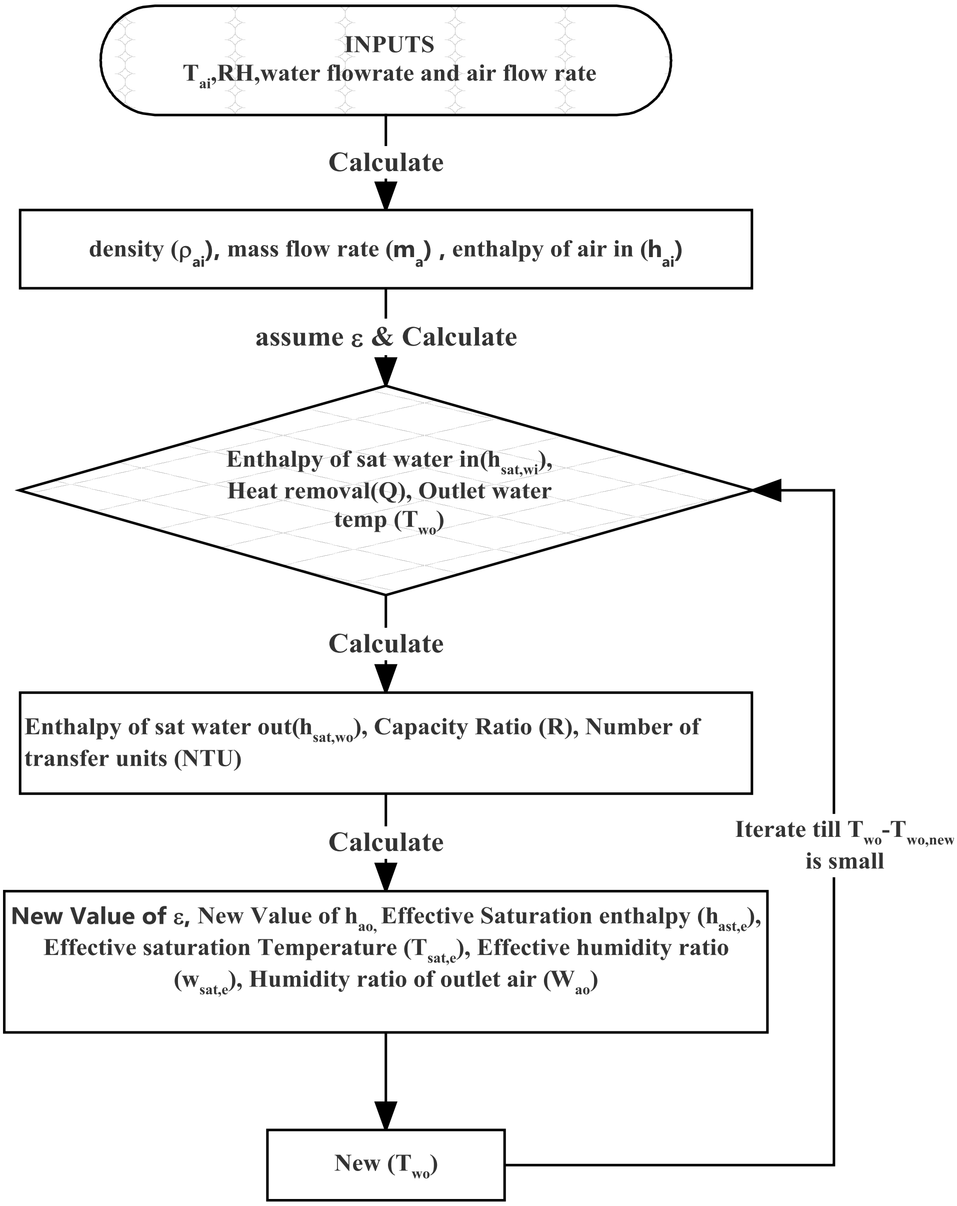


Figure 3-11: Cooling tower model

d. Storage Tank and Connection Pipes

For simplification, the storage tank was assumed to be cylindrical. Heat exchange between the tank and the outside is given by Equation (40), where A is the surface area of the tank or pipe, and R is the insulation in h.ft^2.°F/Btu. Final temperature of the tank after standing heat loss at night was evaluated using the Equation (41). For the simulation, insulation value used for the storage tanks is R-15, and the pipe is R-3. Length of the connection pipes used is 25m for collector to tank and 22 m from tank to absorption system generator. At any given time, Heat at the tank is given as Equation (39), where Q_{lost} is heat lost from the tank and the pipes. Maximum temperature of the tank was set to 100°C based on the chiller specification and range specified by Wang and Zheng for a single effect LiBr absorption system [47].

$$Q_{tank} = Q_u - Q_g - Q_{lost} \tag{39}$$

$$Q_{loss} = \frac{A}{R \times 0.176110}(T_{hot} - T_{amb}) \tag{40}$$

$$Q_{loss} = mc(T_1 - T_2) \tag{41}$$

3.2.2. Electric Chiller

3.2.2.1. Sizing the System

The base system used in this study is a 4.7kW$_{th}$ absorption chiller system. Looking at the performance curves, even though the nominal capacity of absorption chiller is 4.7kW$_{th}$ the system is able to operate at 7kW$_{th}$. Hence for comparison; an electric chiller rated at 7.1kW$_{th}$ was used.

3.2.2.2. Modeling the System

Attempts were made to model an actual vapor compression system. However, in the new systems available now, the manufacturers are not willing to share neither the system performance curves nor the technical details of the sub-systems, such as the compressor. Without these information a simplified model such as the model described by Cabello *et al* (2005) also cannot be used without carrying experiment to find out necessary parameters [48].

For all the above reasons, a 7.1kW system with a COP of 2.29 was used. This is based on the electrical chiller of LAUDA (Model no: UC 0060ST). For simplification it was assumed to be working at constant COP. Its rated electrical power is 3.1kW

3.2.3. Photovoltaic System

3.2.3.1. Sizing the System

In this section, estimated sizes of the systems were determined by referring to [49]–[51]. Aspects considered in sizing the battery bank are listed below in the table.

a. Battery Bank Sizing

Table 3-1: Battery bank sizing

Description	***Value***		***Unit***	***Remarks***
Chiller Power Consumption		3.1	kW	
Run Time/Day		11.0	Hrs	
Total kWh/Day	*a*	34.1	kWh	
Charger Efficiency	*b*	0.9		
Desired days of autonomy	*c*	0.5	Days	
Temperature Compensation by Battery Manufacturer	*d*	0.9	%	estimating the adjusted capacity at 15.6 degree Celsius
Allowable Depth of Discharge	*e*	0.75		% between 50 to 80
Desired Battery nominal voltage	*f*	48.0	V	
Battery Bank Size		584.7	Ah	*((((a/b)xc)/d)/e)/f*

Efficiency of the inverter: there is a loss associated in turning DC into AC. Since for most battery-based systems efficiency is closer to 90%, this was used in determining the battery bank size [50].

Number of days/hours the battery has to last: also known as the days of autonomy indicates the time the battery bank is able to supply power with one charged cycle.

Batteries operating temperature: operating temperature of the battery. A temperature derate factor of 90% is used, which corresponds to battery temperature of 60^{o} F, and indicates that at this temperature, the battery will be able to deliver 90% of its rated value [50].

Depth of Discharge: the amount of energy drawn from the battery bank. Since the number of the charge cycles and depth of discharge are related and give determines the life of the battery, 50 to 80% depth of discharge gives good battery life. Hence, 75% was used in this study to determine the battery bank size.

Nominal Voltage: nominal voltage of the battery bank set to 48 V.

b. PV Array Sizing

Factors considered in sizing the PV array are as follows.

PV and Battery Efficiency: losses involved in the battery include internal losses, as well as the ability for the charge controller to charge the battery. As for the PV array, loses include, temperature of the array, how dirty the modules are, voltage losses in the wiring etc. So, an estimated value of 85% for the battery efficiency and 75% for the PV array was used in determining the array size.

Total available solar resource: a factor involved with the losses due to the shading effects, array tilt and azimuth. Factor of 95% is used in the calculation, since the PV array was set to collect the maximum solar resource available.

Peak Sun Hours: time frame during which maximum solar power is available. 5 hours used for calculation purposes.

Table 3-2: PV array sizing

Description		***Value***	***Unit***	***Remarks***
Chiller Power Consumption		3.1	kW	
Run Time/Day		11.0	Hrs	
Total kWh/Day	*a*	34.1	kWh	
PV & Battery Efficiency	*b*	0.64	%	Assuming 85% for battery and 75% for PV
TSRF	*c*	0.95	%	Assuming 10% loss due to shading, array tilt and orientation
Peak Sun Hours	*d*	5	Hrs	
Array Size		11.2	kW	*(a/(bxc))/d*

c. Inverter Sizing

The chiller system in this study was operated at 220 V, therefore this was the inverter output voltage. Rated power of the chiller system to be used is 3.1 kW. PV equipment suppliers such as LEONICS recommend for a standalone system the inverter should 20-30% higher than the rated power. Specially if the appliance type is motor or compressor, it should be minimum 2 to 3 times higher than the rated value [52]. Hence, an optimal size of inverter for this system can be 6 kW.

3.2.3.2. Modeling the System

An hourly energy flow model, illustrated by Khatib and Elmenreich (2014), was used to model the PV system in this study [53]. The power output P_{PV} of the PV system at a given time is by Equation (42), and the variables in these equations are given below. Table **3-3** shows the data from the PV panel manufacturer catalogue data which were used in the simulation.

$$P_{PV}(t) = \left[P_{peak}\left(\frac{E(t)}{E_{st}}\right) - \alpha_T [T_c(t) - T_{st}] \right] \times \eta_{inverter} \times \eta_{wire} \tag{42}$$

$$T_c(t) = T_{amb}(t) + \frac{NOCT - 20}{800} \times E(t) \tag{43}$$

Where:

P_{peak}	:	Peak power of the PV system
E_{st} & T_{st}	:	Standard test conditions
E	:	Radiation at a given time
T_c	:	Collector temperature which is determined by Equation (43)
α_T	:	Temperature coefficient of PV module Power
$\eta_{inverter}$ & η_{wire}	:	Inverter and wire efficiencies
$NOCT$	:	Nominal operation cell temperature measured under 800W/m^2 solar radiation and 20°C of ambient temperature

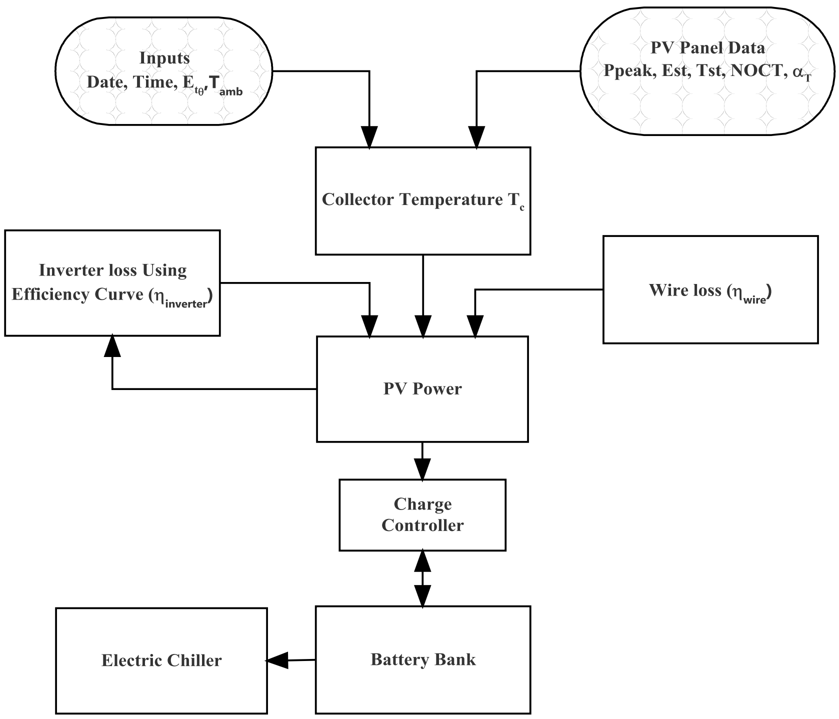


Figure 3-12: Flow diagram of electric chiller operated by PV system

Inverter efficiency curve was modeled on the relation given by Demoulias (2010) [54] for a 5kW inverter manufactured by SMA Solar Technology.

$$\eta_{inverter}(P_{PV}) = 97.644 - 1.995 \times P_{PV} - \frac{0.445}{P_{PV}} \tag{44}$$

Table 3-3: PV panel manufacturer catalogue data

Description	**Value/Unit**
P_{peak}	11.2kW
I_{st} & T_{st}	100 W/m2 & 25 ^{o}C
α_T	-0.44 %/^{o}C
η_{wire}	0.95
NOCT	47.5 ^{o}C

3.3. Computer Simulation

3.3.1. Absorption System

To carry out the computer simulation, meteorological data taken in Bangkok in the year 2000 was used. During the simulation, collector areas used are 20, 23, 26 and 38m^2, and storage tank volumes used are 300, 900 and 1500 liter. For all the above options, open loop and direct loop was also considered.

3.3.2. Electrical Chiller

As mentioned earlier in Chapter 3.2.2.2, a 7.1kW$_{th}$ chiller working at a constant COP of 2.29 was used in this study. The chiller was assumed to run at rated capacity if the PV power was available.

During the life cycle cost estimation, for further comparison, another scenario was also considered where the electric chiller was operated using the utility power.

CHAPTER 4

RESULTS AND DISCUSSION

4.1. Solar Irradiance on the Inclined Surface

Figure **4-1** shows the Total irradiance on a horizontal plane evaluated from the sky model compared to measured global irradiance on the horizontal plane. Figure shows the sky model values are in compliance with the measured station data. An average deviance of 0.29% was found between the two.

Figure **4-2** shows a comparison between the evaluated total irradiance on an inclined plane and the irradiance on an inclined plane calculated using Equation (2). The figure shows that the results from Equation (2) which assumes uniform sky also gives a good approximation of the irradiance on the inclined plane. An average deviance of 1.56% was found between the sky model results and that from the Equation.

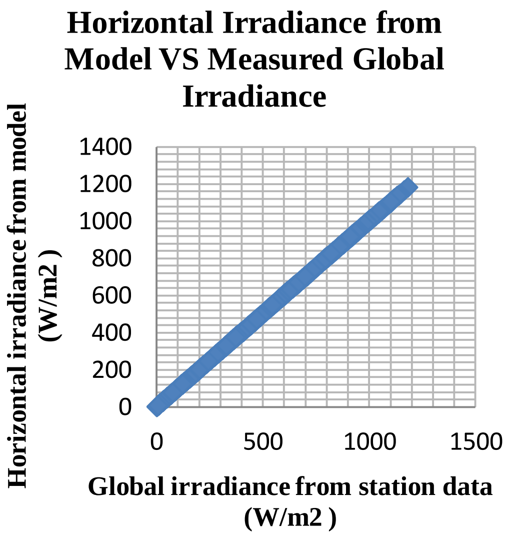


Figure 4-1: Measured horizontal irradiance against irradiance evaluated from sky model

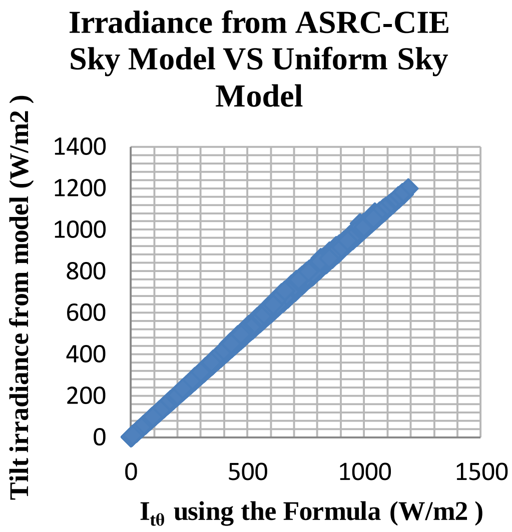


Figure 4-2: Irradiance on inclined plane from ASRC-CIE sky model against uniform sky model

Figure **4-3** shows the irradiance calculated from the ASRC-CIE sky model on the inclined plane, and Figure **4-5** shows the ambient temperature averaged per hour per month for the whole year from 7:00 AM to 6:00 PM. The average global irradiance on the horizontal plane for the reference time was found to be 427.63 W/m^2, and for the inclined plane, 472.8 W/m^2. On the graphs, below the x-axis shows the average values per hour for the whole year. It should be noted that despite the average values, during the whole year irradiance values varied from 0 to 1197.41 W/m^2 in the reference time. As for the ambient temperature, it varied from 16.85°C to 40.66°C.

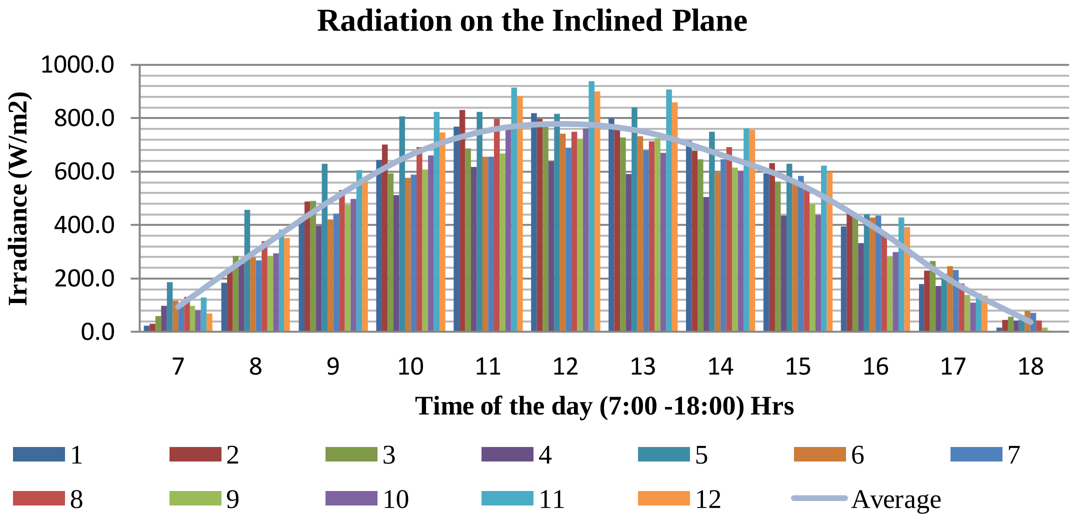


Figure 4-3: Irradiance on the inclined

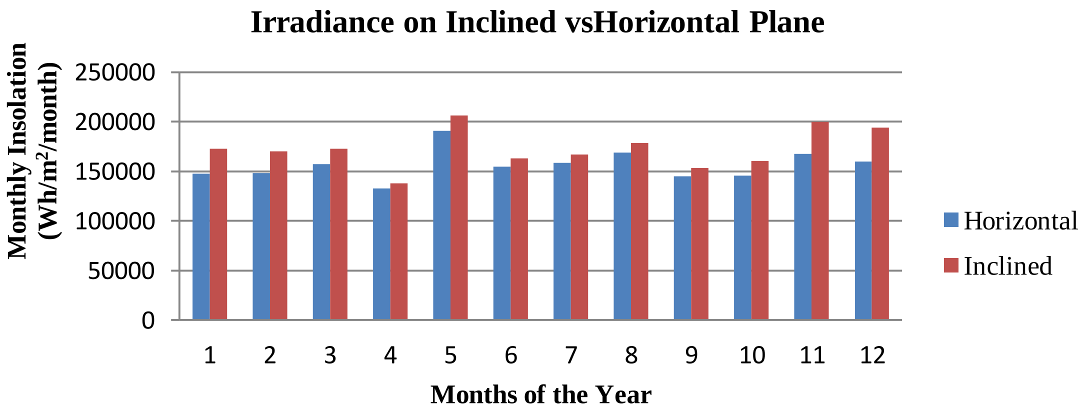


Figure 4-4: Irradiance on the inclined plane versus irradiance on the horizontal plane

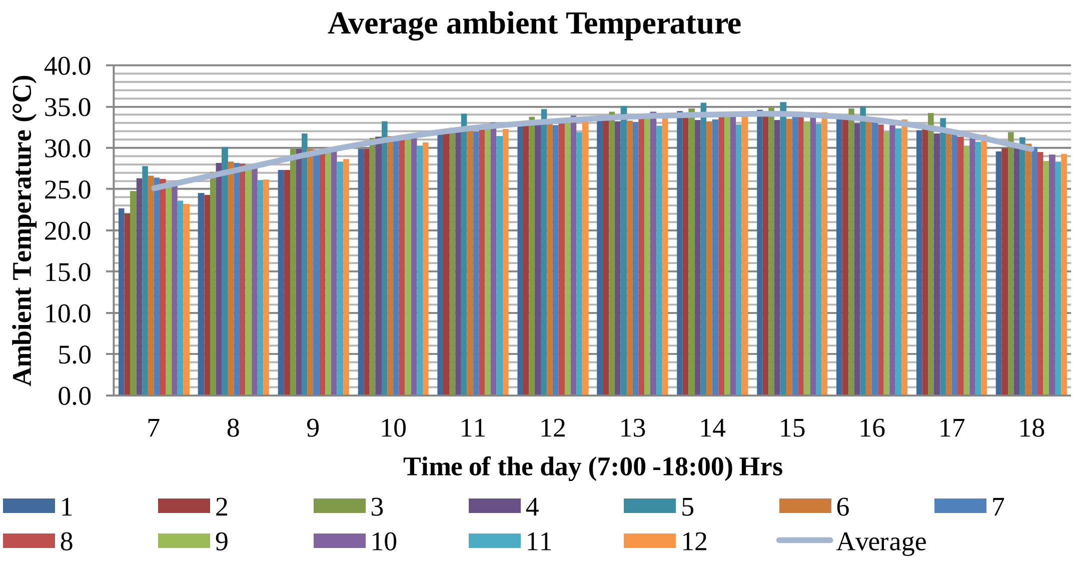


Figure 4-5: Measured hourly average ambient temperature

Figure **4-4** shows a comparison of the monthly solar insolation on the inclined plane and the horizontal plane. The figure shows different amounts of gain in insolation on the inclined compared to horizontal plane for different months. This is due to the different path of the sun taken over the year. Yearly insolation on the horizontal plane was found to be 1.88MWh/m^2/year. Calculated yearly solar insolation for the inclined plane was found to be 198kWh/m^2/year higher than that on the horizontal plane giving a total insolation of 2.08 MWh/m^2/year. Variation of the irradiance on the inclined plane compare to horizontal plane along the year can be explained as follows.

The axis of the Earth's rotation is tilted 23.4 degrees. This tilt results in different paths of the sun along the North-South plane at different times of the year, as shown in Figure **4-6** . Southern sun is more perpendicular to the inclined than the horizontal plane, while northern sun is more perpendicular to the horizontal plane than the inclined plane. Hence, there is a variation in the additional amount of irradiance found on the inclined plane compared to horizontal plane at different times of the year. In addition to this, ground reflectance will also contribute to the inclined plane, while it will be blocked for the horizontal plane.

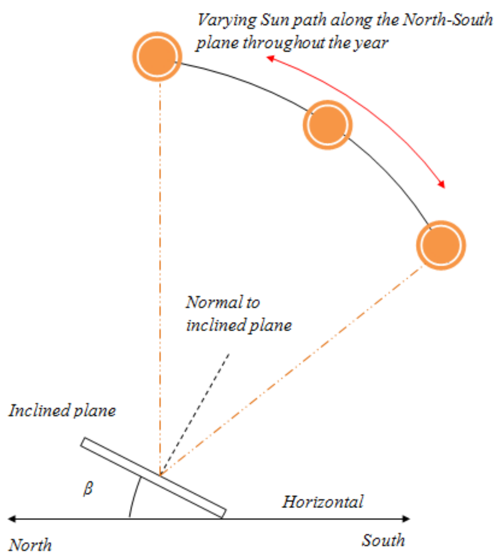


Figure 4-6 : Sun path along the year with respect to the inclined plane along the North-South plane

4.2. Absorption System

4.2.1. Hourly simulations

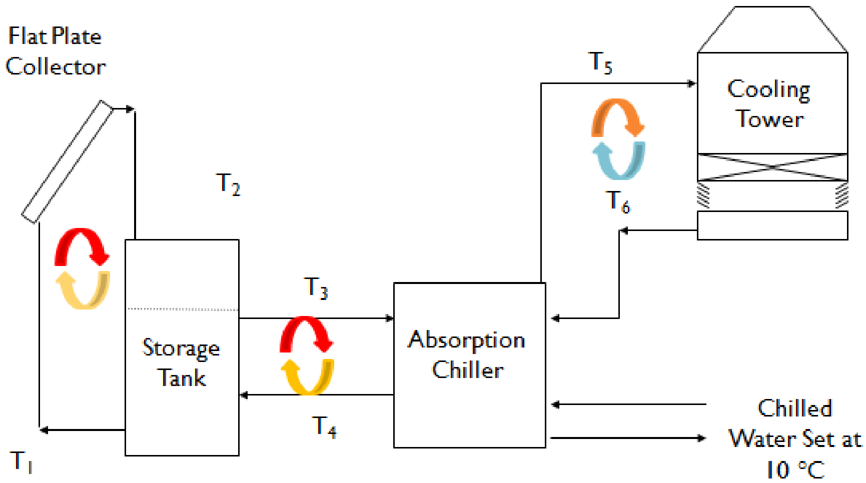


Figure 4-7 : Temperatures of the components in hourly simulation

Figure **4-7** and Table **4-1** show the inlet and outlet temperatures of the flat plate collector, the absorption chiller, and the cooling tower for 4th January 2000, with a collector area of 38m^2 and a hot water storage tank of 1500 liters. The selected time frame is 8:00 to 15:00 hrs.

Table 4-1 : Temperature of the components in hourly simulation

Hour	$E_{t\theta}$	T_{amb}	T_1	T_2	T_3	T_4	T_5	T_6
08:00	351.5	27.18	71.01	72.33	71.01	71.01	24.66	24.66
09:00	621.61	30.5	72.83	77.02	72.83	72.83	24.66	24.66
10:00	823.34	33.1	78.79	84.86	78.79	75.97	28.92	25.84
11:00	949.36	33.98	83.38	90.5	83.38	80.42	30.48	26.58
12:00	1020.58	35.41	89.28	96.83	89.28	85.95	31.94	27.18
13:00	731.61	36.06	95.26	99.48	95.26	91.49	33.32	27.75
14:00	747.83	36.93	95.8	100.21	95.8	92.01	33.87	28.09
15:00	605.74	36.93	96.59	99.48	96.59	92.74	34.27	28.26

Figure **4-8** shows the hourly performance of the absorption chiller for the same day and configuration. At 8:00 and 9:00 AM the system is not able to run due to low thermal storage and low thermal energy from collector. At 10:00AM the chiller is able to operate. The curves show that the irradiance has to be around 600 W/m^2 for this system to operate. However, it should be noted that the time it takes to bring the tank to a desired temperature depends on the size of the storage tank.

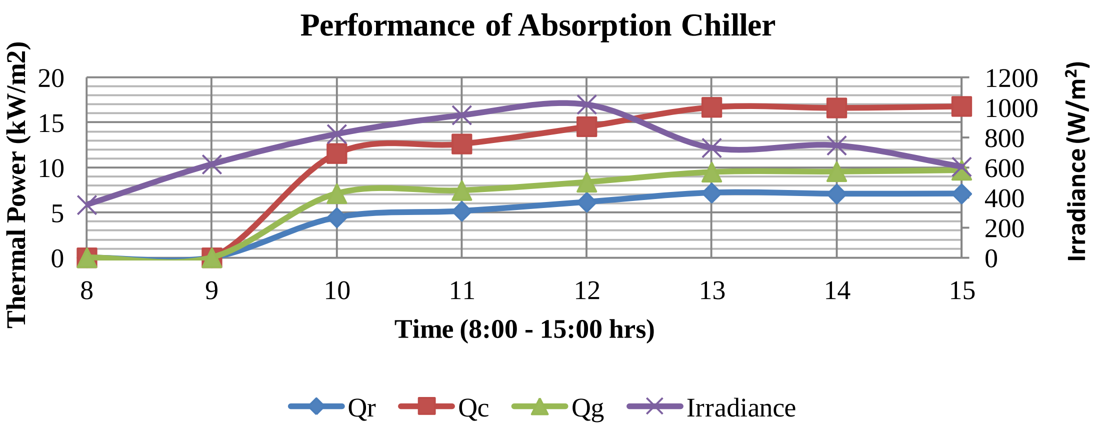


Figure 4-8: Hourly performance of absorption chiller

4.2.2. Simulation Results for the whole year

Figure **4-9** shows the Qr averaged at per hour per month, and below the x-axis is the Qr averaged per hour for the whole year. This figure is for a collector area of 38m^2 and storage tank of 1500 liters. As seen from the hourly simulation results in the previous section for 12th March 2000, for most of the months, the collector input is sufficient for operation at 9:00 AM, and before that the operation is using the thermal energy stored from previous day. Unlike the previous configuration, the results show that in average 1500 liter tank can operate the system for 12 hours.

Figure **4-10** shows the number of hours that the system is unable to operate either due to insufficient produced thermal energy or insufficient stored thermal energy. Open loop and direct loop systems are shown this figure, and for this configuration, the open loop allows additional 44 hours of operation and 444.5 kW cooling during the whole year. Further comparison on size of tanks, collector area, and open loop and closed loop will be done in the next section.

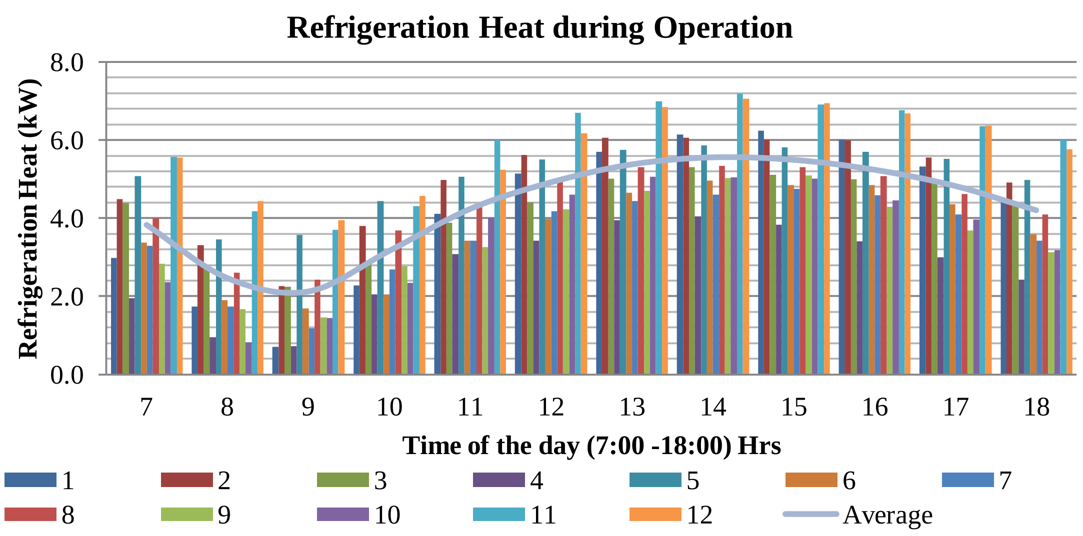


Figure 4-9: Refrigeration heat output per hour per month

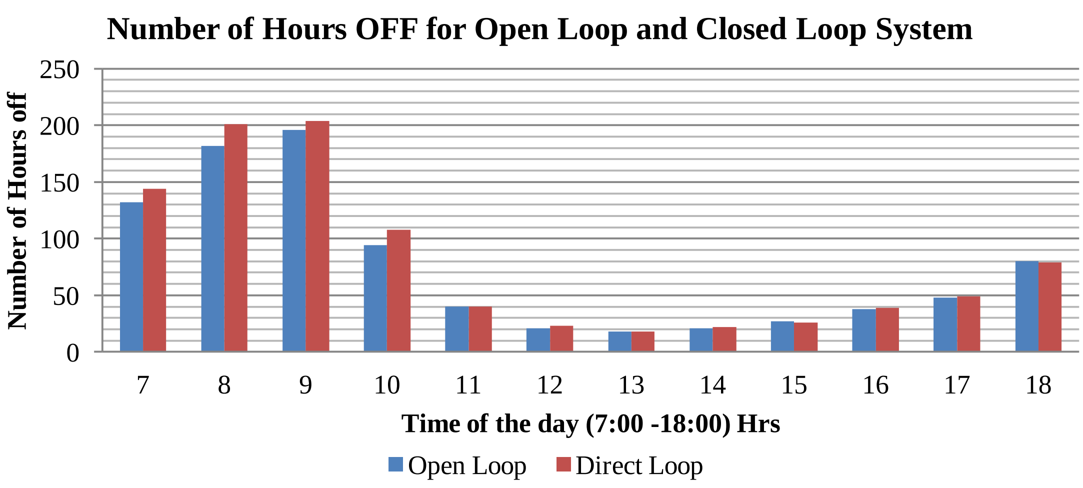


Figure 4-10: Number of OFF hours for open loop and direct loop system

4.2.2.1. Varying Collector Area

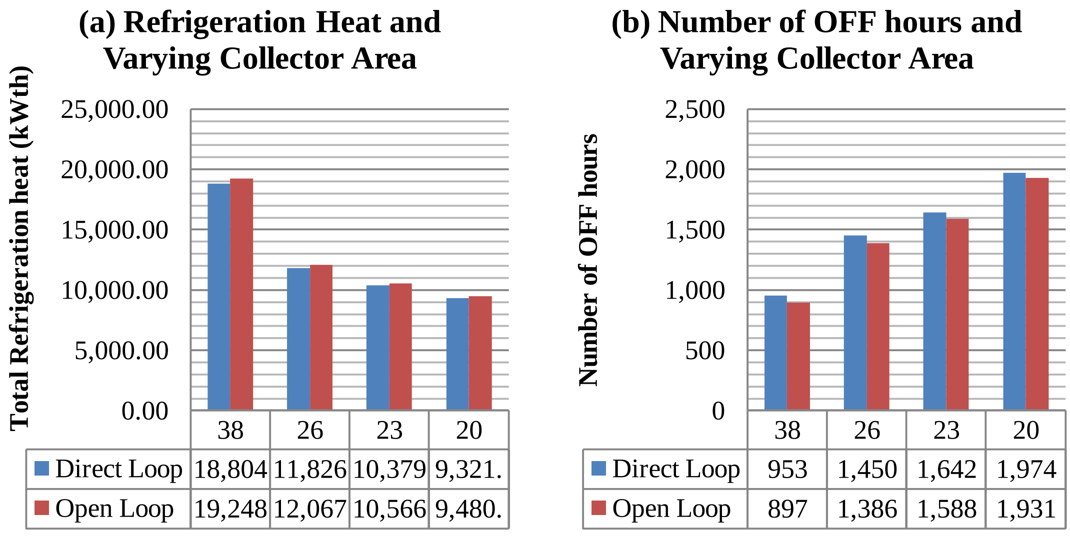


Figure 4-11: (a) Refrigeration heat and (b) Number of OFF hours for varying collector area

Figure **4-11** shows the total refrigeration heat produced in the year and the total number of hours the system is unable to operate for both direct loop system and open loop system. Looking at the charts you can see an increase in total refrigeration heat output and increase in operating hours when the collector area increases. Even though the daily insolation (**Figure 3-3**) in Bangkok shows a more steady insolation than do other countries, a collector of $8m^2/kW_{th}$ shows a significant improvement in performance over $5m^2/kW_{th}$.

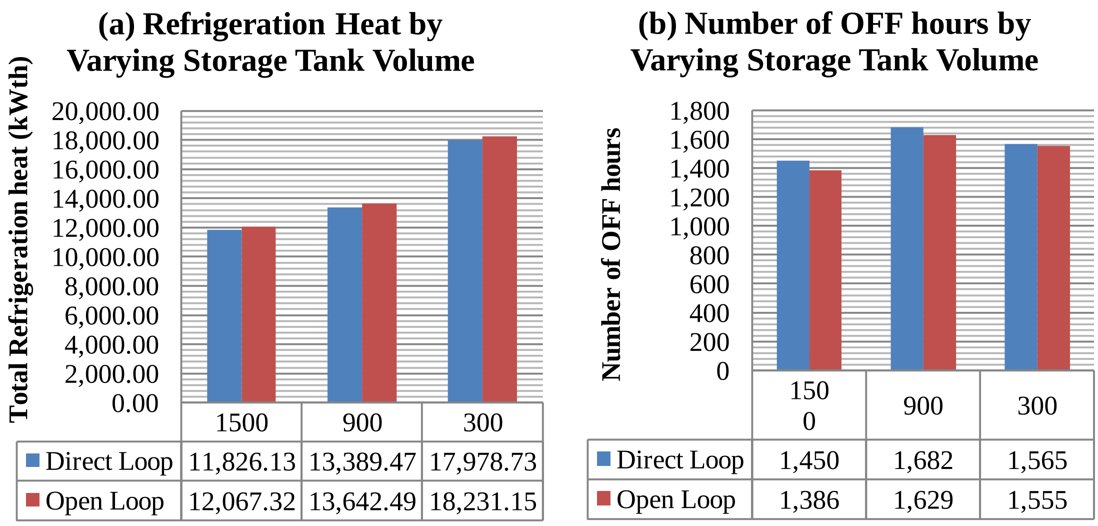


Figure 4-12: (a) Refrigeration heat output and (b) Number of OFF hours for varying storage tank size

4.2.2.2. Varying Storage Tank Size

Figure **4-12** shows the effect on refrigeration heat output and the number of operating hours when the storage tank size was varied. Figure **4-12** (a) shows that the total refrigeration heat output increases when the storage tank size decreases. This is because keeping collector area constant, when the storage tank size is reduced, less time is required to bring the tank to the desired temperature when compared to a larger tank. This reduction in charging the storage tank size allows the tank to operate at higher cooling capacity longer compared to a larger tank. Hence, the total refrigeration heat is higher for the smaller tank.

As for the number of operating hours, even though the total refrigeration heat output is higher for a smaller tank, Figure **4-12** (b) shows that the number of operating hours is the highest for 1500 liter tank, while the number of operating hours for 300 liter tank is more than that of 900 liter tank.

The above mentioned results for the varying storage tank size can be further explained by Figure **4-13** and Figure **4-14**. As mentioned earlier, it can be seen that the 300 liter tank operates at higher cooling capacity for longer time than 900 liter tank. Figure **4-13** shows 300 liter tank is unable to operate at 8:00AM due to insufficient thermal storage backup and thermal energy production, but it can operate at 7:00AM from the backup. As for 900 liter tank, Figure **4-14** shows refrigeration heat output is less from 7:00AM to 9:00AM with sometimes no output. This and the longer time to charge the tank explains the less number of operating hours for 900 liter tank when compared to 300 liters.

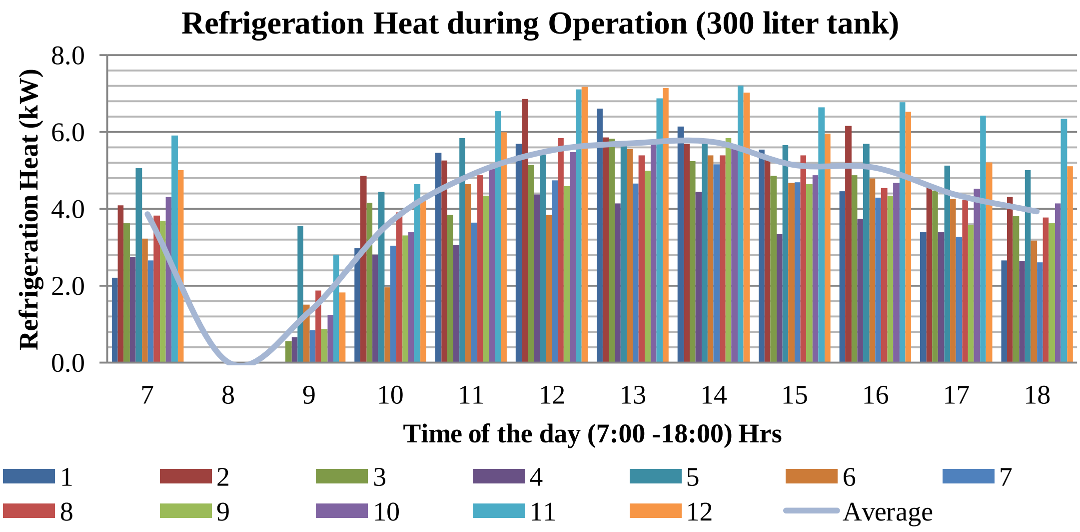


Figure 4-13: Refrigeration heat output for 300 liter tank

4.2.2.3. Open loop VS Direct loop for varying collector area and varying tank sizes

In this section, the open loop system is compared to the direct loop system for all the configurations used in this study. Figure **4-15** and Figure **4-16** show the additional total refrigeration heat and additional operating hours available from an open loop system when compared to direct loop system. When collector size was varied, additional refrigeration heat increases with the increase of collector area and number of operating increases for 26m^2 but tends to decrease for 38m^2. Similarly, when tank size was varied, additional operating hours increased with the increase in tank volume. As for the additional refrigeration heat, it tended to decrease from 900 to 1500 liters tank, while increasing from 300 to 900 liters tank.

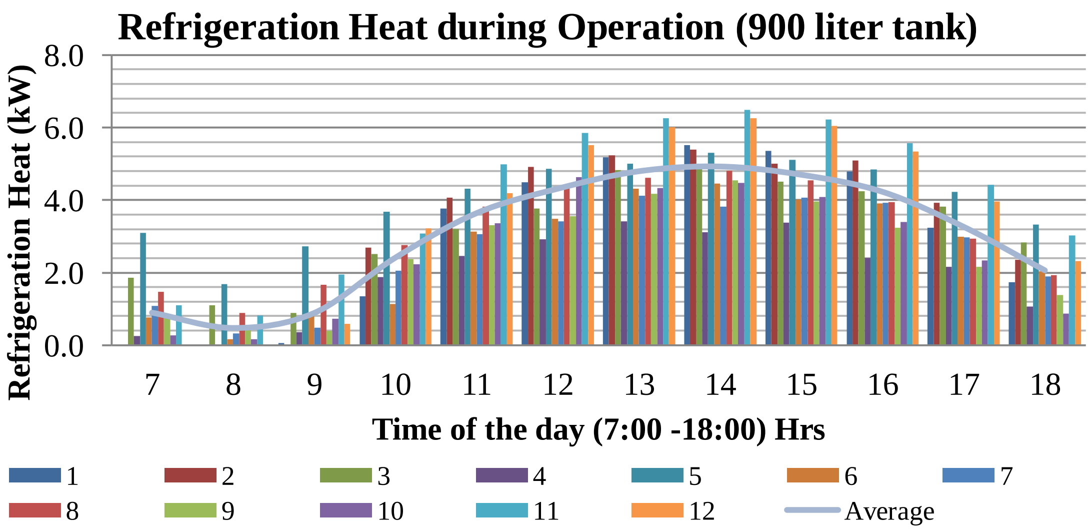


Figure 4-14: Refrigeration heat output for 900 liter tank

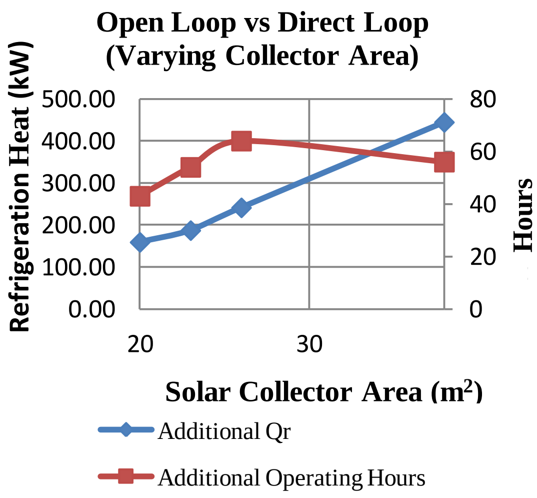


Figure 4-15: Open loop vs direct loop (varying collector area)

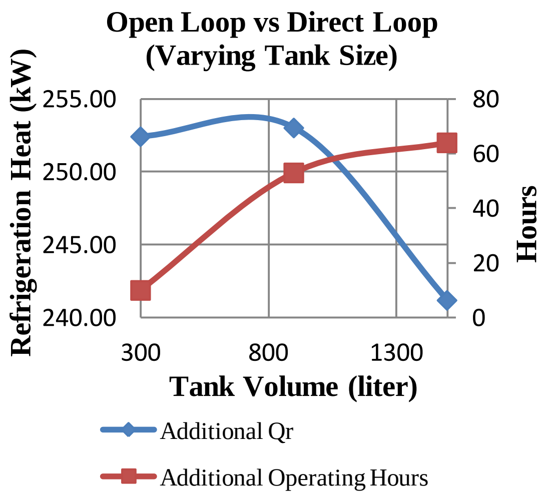


Figure 4-16: Open loop vs direct loop (varying tank size)

The open loop system allows the water to be stored in an insulated tank at night time to be used the next day. So the effects on the performance of an open loop system, when compared closed loop system, becomes more visible depending on the collector area and storage tank. When the storage tank volume is much larger than the volume of water in the collector, the effect of improvement in performance is less compared to smaller storage tank volume and same amount water in the collector. In the configurations considered for simulation, the maximum refrigeration heat gain is between 400 and 500 kW and the maximum gain of operating hours is between 60 and 70 hours in the whole year. So for the simulated configurations, a significant improvement in performance is not seen for an open loop system.

4.3. Electric Chiller Simulation Results

4.3.1. Photovoltaic Power Produced over the year

Figure **4-17** shows the energy produced by the PV panel, and Figure 4-18 shows the charging potential of the battery bank used in the PV electric system. Figure 4-18(a) shows the daily average charge cycle of the battery, and Figure 4-18(b) shows charge on the battery for the whole year. Simulation results shows that the battery bank capacity with a half day backup (Table **3-1**) is not capable to operate the chiller continuously from 7:00AM to 6:00PM throughout the year, even though the system is operation most of the time. Figure **4-19** shows specifically the times at which the system was not able to operate.

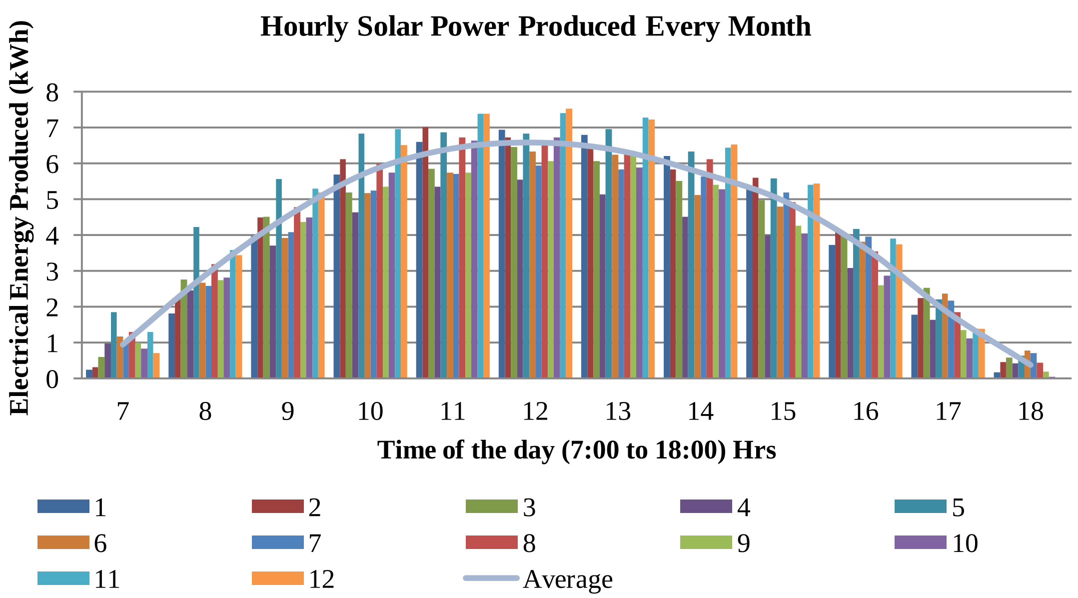


Figure 4-17: Hourly PV electric power produced by the panel

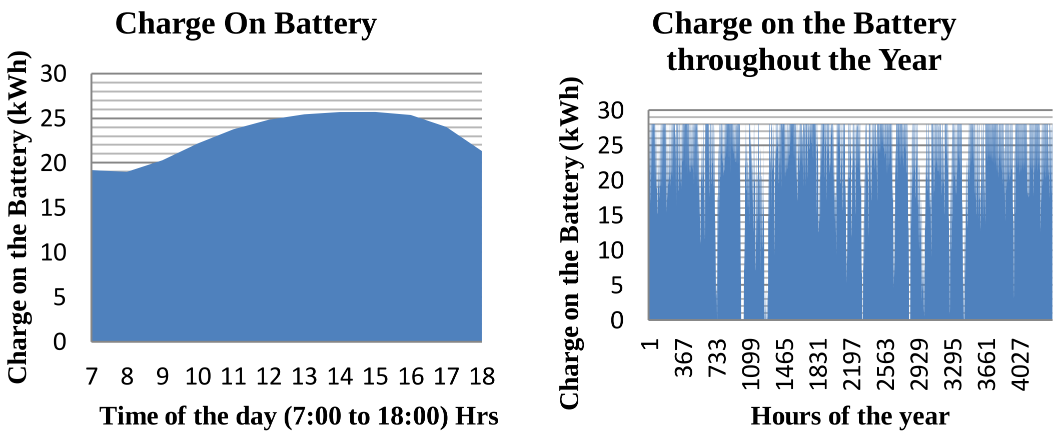


Figure 4-18: (a) Hourly average energy stored in the battery per day; (b) Charge on the battery throughout the year

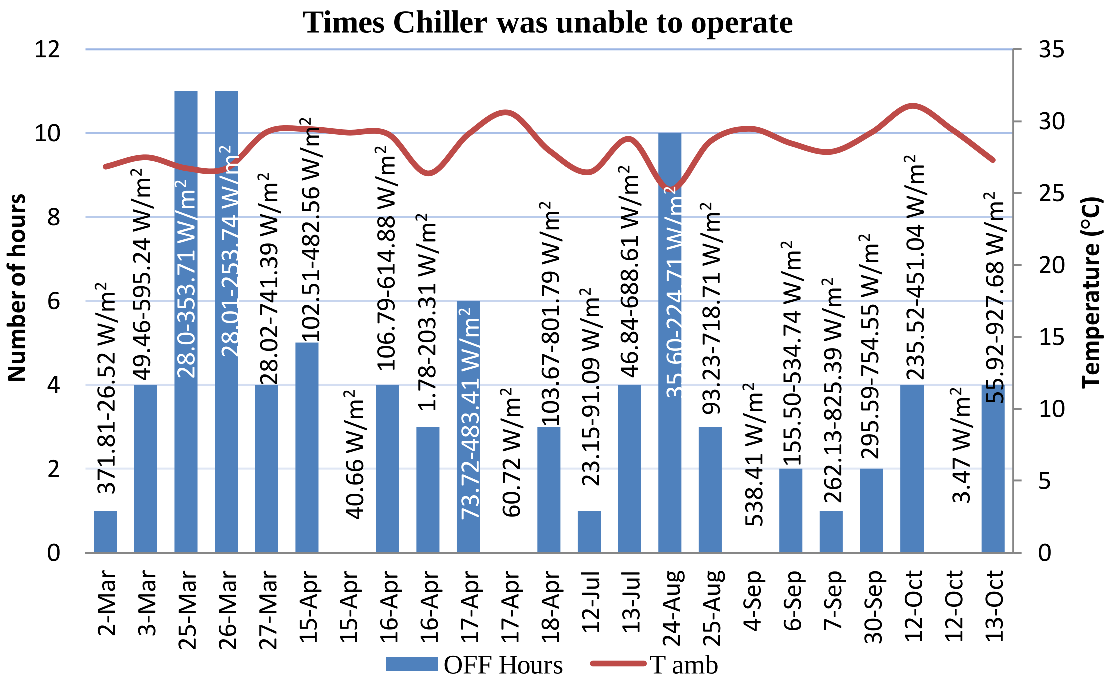


Figure 4-19: Hours electric chiller was unable to operate

4.4. Life Cycle Costing

4.4.1. Absorption System Details and its Life Cycle Costing

Table 4-2: Absorption chiller details used for life cycle costing

Description	Cost	Lifetime (Years)	Reference
Absorption Chiller (4.7kW$_{th}$)	$23,500.00	30	[55], [40]
0.5 kW Pump			
Flat Plate Collector (26m2)	$5,965.30	25	[56], [43]
Cooling Tower (17kW)	$1,185.00	15	[57], [58]
CT Fan Motor (0.12kW)			
CT Pump (0.54kW)			
Storage Tank (200 Gallon)	$1,290.00	10	[59], [60], [40]
Storage Tank (500 Gallon)	$1,500.00		
Total	***$32,018.30***		

Table **4-2** shows the costs and the life spans of the sub-systems used in the lifecycle costing of the solar absorption system. For the lifecycle costing, collector area of 26m^2 and storage tank of 1500 liter was used. Even though 1500 liter tanks are available in the market, since the price of exact volume couldn't be found, the cost of the storage tank was estimated to be $1,368.00 based on a 200 and 500 gallon tank.

Table **4-3** shows the life cycle costs calculated for the absorption system. A discount rate of 7% was used for the calculation. Annualized cost of the whole system was calculated to be $3,494.75 with a present value of the system of $40,066.03. Annual cost of absorption chiller weighs more than 60% total annual costs.

Table 4-3: Life cycle costing for absorption chiller

Description	SPWF	CRF	Annuity	Annual Energy cost	Annual Cost	Present Value
Absorption Chiller	12.41	0.08	$1,893.78	$392.40	$2,223.38	$27,587.54
Solar Collector	11.65	0.09	$511.89	$0.00	$511.89	$5,965.30
Cooling Tower	9.11	0.11	$130.11	$434.81	$564.91	$5,145.19
Storage tank	7.02	0.14	$194.77	$0.00	$194.77	$1,368.00
				Total(USD)	***$3,494.75***	***$40,066.03***

4.4.2. Electric Chiller Details and its Life Cycle Costing

Table **4-4** shows the costs and life spans of the sub-systems of PV electric system used for the life cycle costing. Table **4-5** shows the life cycle costs calculated for the PV electric chiller system. The system has an annualized cost of $3,123.89 and a net present value of $25,434.00.

For further comparison of life cycle costs, life cycle costing was done for the electrical chiller if it was to be operated on grid power at rated conditions. The electricity tariff was taken to be THB 4.5 ($0.15)/kWh. Table **4-6** shows the annualized cost and present value for the grid operated chiller.

Table 4-4: PV electric chiller details used for life cycle costing

Description	Cost	Lifetime (Years)	Reference
Electrical Chiller (7.1 kW_{th})	$1,900.00	15	[40],[61],[62]
Inverter (6kW)	$8,239.00	20	[63][64]
Charge Controller	$1,480.00	15	[65],[66]
Photovoltaic Panel (12 kW_p)	$16,800.00	25	[67],[68]
Battery Bank (18 kWh)	$4,995.00	5	[69][70],[71]
Total(USD)	***$25,434.00***		

Table 4-5: Life cycle costing for PV electric chiller

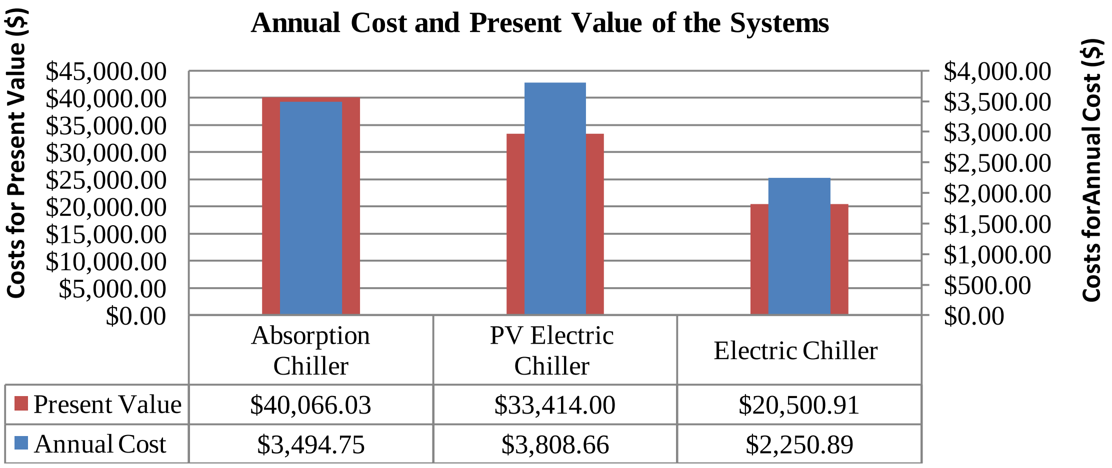
Description	SPWF	CRF	Annuity	Annual Cost	Present value
Electric Chiller	9.11	0.11	$208.61	$208.61	$1,900.00
Inverter	10.59	0.09	$777.70	$777.70	$8,239.00
Charge Controller	9.11	0.11	$162.50	$162.50	$1,480.00
PV Panel	11.65	0.09	$1,441.62	$1,441.62	$16,800.00
Battery Bank	4.1	0.24	$1,218.23	$1,218.23	$4,995.00
			Total(USD)	***$3,808.66***	***$33,414.00***

Figure **4-20** shows the net present value and annual costs of the systems for comparison. The chart shows that net present value and annualized cost of the absorption chiller is larger than the PV driven electric chiller and grid operated chiller, and the net present value and annualized cost of PV-driven electric chiller is larger than that of grid operated chiller.

Therefore, for small cooling capacity, grid operated chiller is economically more feasible compared to PV-driven electric chiller and solar powered absorption chiller.

Table 4-6: Life cycle costing for electric chiller operated on grid power

Power	**SPWF**	**CRF**	**Annuity**	**Annual Energy cost**	**Annual Cost**	**Present Value**
At Rated Power	9.11	0.11	$208.61	$2,042.28	***$2,250.89***	***$20,500.91***



	Absorption Chiller	PV Electric Chiller	Electric Chiller
Present Value	$40,066.03	$33,414.00	$20,500.91
Annual Cost	$3,494.75	$3,808.66	$2,250.89

Figure 4-20: Comparison of annual cost and present value for the systems

Additional analysis was done for the small absorption system to see how much the chiller cost should decrease to be comparable to grid power-operated chiller, assuming the prices of all other sub-systems have landed or remained the same. Table **4-7** shows the how much the initial cost of the absorption chiller should decrease if it were to be compared to a 7.1kW$_{th}$ system, and Table **4-8** shows how much the initial cost of absorption chiller should decrease if it were to be compared to a 4.7kW$_{th}$ system [61]. The system is compared with its same rated size system and a 7.1kWth system because even though absorption chiller is rated at 4.7kW$_{th}$, with higher generator temperature the system is able to operate at 7kW$_{th}$ (Refer to section 3.2.1.3 & 3.2.2.1).

The analysis shows that the small absorption chiller with its current cost ($5000/kW$_{th}$) has to decrease to a rate of about $1,713/kW$_{th}$ if compared to 7.1kW$_{th}$ system and $1,428/kW$_{th}$ if compared to a 4.7kW$_{th}$ system. However, it should be noted that the 4.7kW$_{th}$ system used here is an industrial chiller, so there are much cheaper electrical chillers available in the market for the absorption chiller to be economically competing with. Hence, the reduction in cost will be higher than that mentioned earlier. Such reductions could be possible because current prices in the market shows that for larger capacity absorption chillers (351.7kW$_{th}$ to 1,230.9 kW$_{th}$), the cost of absorption chillers are around \$145-200 /kW$_{th}$ [72]. Life cycle costing of one such system will be presented in the next section.

Table 4-7: New life cycle costing for absorption chiller comparing annual cost to grid operated electric chiller (7.1kW$_{th}$)

Description	**Initial Cost**	**Annuity**	**Annual Energy cost**	**Annual Cost**	**Present Value**
Absorption Chiller	$8,053.00	$648.96	$329.40	$978.36	$12,140.54
Solar Collector	$5,965.30	$511.89	$0.00	$511.89	$5,965.30
Cooling Tower	$1,185.00	$130.11	$434.81	$564.91	$5,145.19
Storage Tank	$1,368.00	$194.77	$0.00	$194.77	$1,368.00
			Total (USD)	***$2,249.93***	***$24,619.03***

Table 4-8: New life cycle costing for absorption chiller comparing annual cost to grid operated electric chiller (4.7kW$_{th}$)

Description	**Initial Cost**	**Annuity**	**Annual Energy cost**	**Annual Cost**	**Present Value**
Absorption Chiller	$6,716.00	$541.22	$329.40	$870.62	$10,803.54
Solar Collector	$5,965.30	$511.89	$0.00	$511.89	$5,965.30
Cooling Tower	$1,185.00	$130.11	$434.81	$564.91	$5,145.19
Storage Tank	$1,368.00	$194.77	$0.00	$194.77	$1,368.00
			Total (USD)	***$2,142.19***	***$23,282.03***

4.5. Life cycle costing and simulation of a larger system

4.5.1. Model Variables and Simulation Results

For further analysis, this section will present a simulation of a large absorption system and its lifecycle cost compared to an electrical chiller. The absorption system used in this study is based on the system proposed in the paper of Bukoski *et al* (2014) for a stadium of 15,000 seating capacity in Thailand [40]. The system proposed was a 560kW$_{th}$ absorption chiller capable of working as a single effect and double effect system depending on the thermal energy available. Compound parabolic concentrator medium thermal evacuated tube solar collectors were used with a 6000 liter pressurized tank. However, for this simulation, single effect absorption chiller, with flat plate collector will be used. Constants for NTU correlation derived by Braun et al (1989) [44] for cooling tower derived based on the catalogue data of Liang Chi Industry (Thailand) co.ltd [73]. Equation (45) shows the NTU correlation derived for the simulation.

$$NTU = 1.7785\left(\frac{m_c}{m_a}\right)^{0.2859} \tag{45}$$

The same absorption chiller model was used under the assumption that single effect LiBr absorption chillers follow the same performance curves. Following this, Equation (46) can be used to evaluate for Qr of 560kW$_{th}$ system.

$$Qr_{560} = \frac{Qr_{4.7}}{4.7} \times 560 \tag{46}$$

Figure **4-21** shows the refrigeration heat output from the simulation. The chart shows that 6000 liter tank with a flat plate collector is not sufficient to operate the system for 12 hours. However, it should be noted that in the proposed system by Bukoski *et al.* (2014), proposed collectors will be able to generate water at higher temperatures, and the pressurized storage tank will be able to store water at higher thermal energies [40]. In addition to this, chilled water storage tanks could also be used as backups.

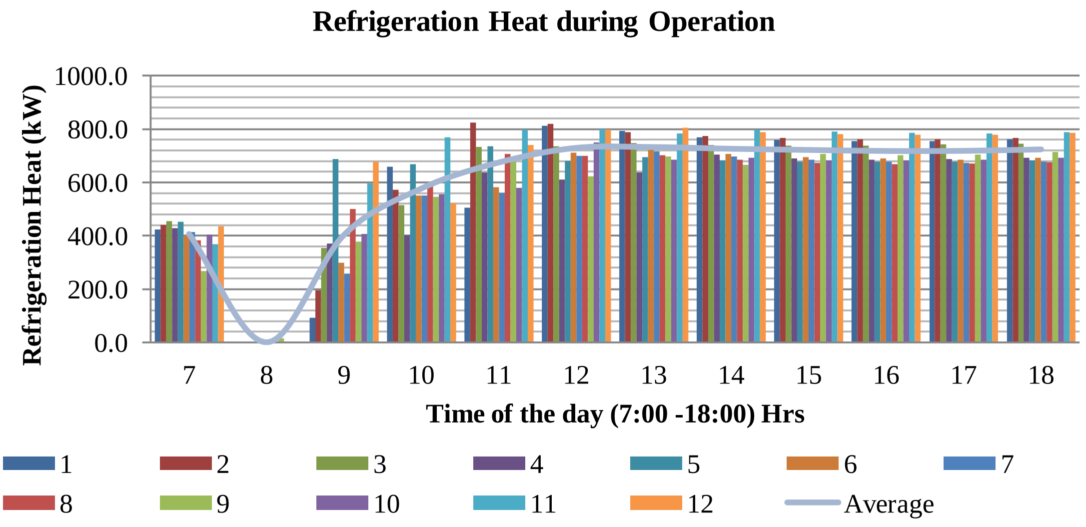


Figure 4-21: Refrigeration heat output for 560kW$_{th}$ system

4.5.2. Life Cycle Costing and Comparison

Table **4-9** shows the costs and lifespan used for the sub-systems in the absorption system in order to do the life cycle costing. For a 6000 liter tank, cost was estimated to be around $9,950.00 based on the costs of 350 and 10,000 gallon tank. Table **4-10** shows the life cycle costing done for absorption system.

The 560kW$_{th}$ absorption system was compared with a 531kW$_{th}$ electrical chiller operating on grid power. The chiller is from a Chinese company called Guangdong Head-Power Air Conditioning Co.,Ltd [74]. In addition to that, life cycle costing was done for the same electric chiller driven by PV. Same steps in section 3.2.2.1 were followed in the sizing the PV system and the same life span for the sub-systems as in Table **4-4** were used.

Figure **4-22** shows a comparison of the annual costs and present values for large cooling systems. Analysis shows that 560kW$_{th}$ absorption chiller gives a low net present value and annual cost compared to PV-driven electric chiller and grid operated electric chiller. Net present and annual cost of PV-driven electric chiller is higher than that of grid operated electric chiller

Table **4-11** shows the life cycle costing for the PV-driven system, and Table **4-12** shows the life cycle costing for the chiller if it were to be operated on grid power.

Table 4-9: 560kW$_{th}$ Absorption chiller details used for life cycle costing

Description	Cost	Lifetime	Reference
Absorption Chiller (560kW$_{th}$)	$120,000.00	30	[40]
Flat Plate Collector (2,000m^2)	$458,869.00	25	[56], [43]
Hot Water Tank (350 gallon) Hot Water Tank (10,000 gallon)	$7,000.00 $10,000.00	10	[75], [40]
Cooling Tower (1,899 kW$_{th}$)	$50,940.00	15	[76], [58]
Hot Water Pump (11kW)	$1,265.44	2	[40],[77]

Table 4-10: Details of the 560kW$_{th}$ absorption system

Description	SPWF	CRF	Annuity	Annual Cost	Present value
Absorption Chiller	12.41	0.08	$9,670.37	$9,670.37	$120,000.00
Flat Plate Collector	11.65	0.09	$39,375.79	$39,375.79	$458,869.00
Cooling Tower	9.11	0.11	$5,592.94	$5,592.94	$50,940.00
Storage tank	7.02	0.14	$1,416.66	$1,416.66	$9,950.00
Water Pump	1.81	0.55	$699.90	$7,946.70	$14,367.79
			Total ($)	***$64,002.46***	***$654,126.79***

Table 4-11: Life cycle costing for 531kWth PV driven electric chiller

Description	Initial cost	Ref	Annuity	Annual Cost	Present value
Electric Chiller (531kW$_{th}$/155kW)	$28,320.00	[74]	$3,109.38	$3,109.38	$28,320.00
Inverter (190kW)	$190,000.00	[78]	$17,934.66	$17,934.66	$190,000.00
Charge Controller	$3,163.00	[79]	$347.28	$347.28	$3,163.00
PV Panel (612kW)	$856,800.00	[68]	$73,522.45	$73,522.45	$856,800.00
Battery Bank (306 kWh)	$212,787.00	[71]	$51,896.77	$51,896.77	$212,787.00
			Total ($)	***$146,810.50***	***$1,291,070.00***

Table 4-12: Life cycle costing of 531kW$_{th}$ electrical chiller operated on grid power

Description	**Life**	**Initial cost**	**Electrical Power**	**SPWF**	**CRF**	**Annual Cost**	**Present value**
Electric Chiller	15	$28,320.00	155	9.11	0.11	***$105,223.40***	***$958,365.50***

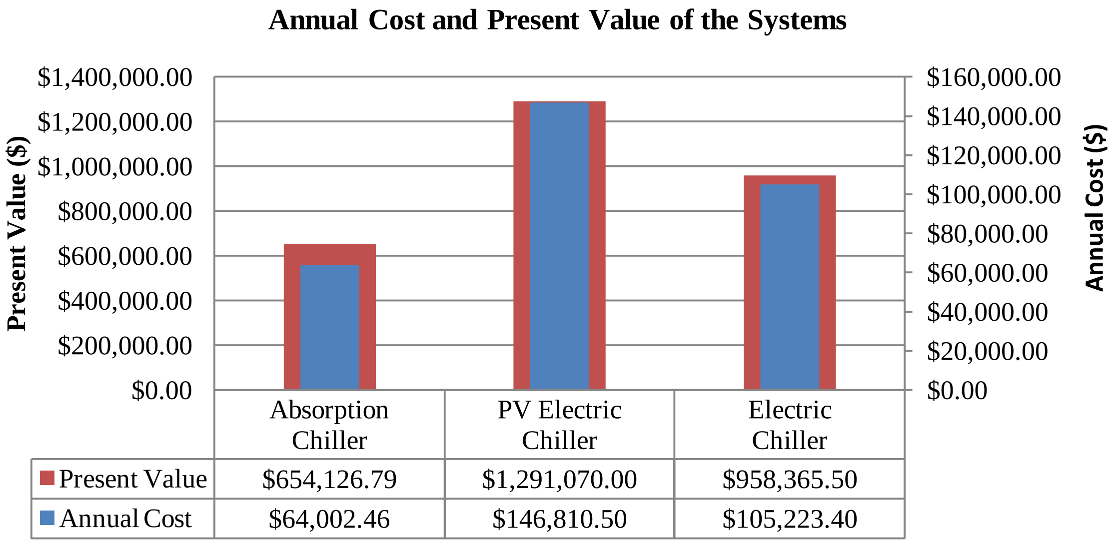


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Figure 4-22: Comparison of annual costs and present values for the large systems

4.6. Further comparison between absorption chillers and electric vapor compression chillers

- The COPs of absorptions chillers are lower than those of electric chillers. Single effect absorption chillers are reported to have COP of 0.6 to 0.8, while double effect absorption chillers have a COP of 1.0 and triple effect systems with a COP of 1.4 to 1.6. As for electric chillers, COP can range between 1.0 to 6.0[72].
- Absorption chillers are larger in terms of floor area and height that are equivalent electric chillers. Therefore, larger floor area and room height is required to equip an absorption chiller compared to an electric chiller.
- Due to its large size, absorption chillers are usually shipped in sections requiring field welding for assembly. Absorbent charging is also done on the field for absorption systems. On the other hand, for electric chillers, they are shipped fully assembled with the refrigerant already charged.

- Electric chillers produce loud noise and strong vibrations while absorption chillers are quiet and almost vibration-free.
- Due to possible crystallization of LiBr (absorbent) in absorption chillers when it cools down, the condenser water temperature must be kept within a certain range depending on LiBr/water concentration, while there is no such limitation for electrical chiller
- Absorption chillers require hot water storage tanks or chilled water storage tanks for backup. As seen from the simulation, sometimes the backup is not enough, and increasing the backup capacity could be costly and space-consuming. So emergency power source may be required for cooling and to ensure LiBr doesn’t crystallize within the chiller due to low ambient temperature.
- Heat is rejected at the condenser for LiBr absorption chillers larger than equivalent electric chiller; therefore, larger cooling towers and larger condenser water pumps with high water flow rates are needed for absorption chiller.
- LiBr absorption chillers use natural refrigerants, such as water, unlike electric chillers, which require CFC or HCFC refrigerants with high global warming potential.

CHAPTER 5
CONCLUSIONS

The study presented simulation results and life cycle costs of a solar powered absorption chiller, a PV-driven electric chiller, and a grid-operated electric chiller. Due to the huge variants in investment costs for absorption chillers, a small system (4.7kWth) and a large system (560kWth) were considered in the analysis. From the simulation and the calculations, this study arrives at the following conclusions.

- ACRC-CIE sky model gives a good approximation of diffuse irradiance, and the results are in close proximity with the uniform sky model on an inclined plane. For further verification, irradiance should be measured on an inclined plane, and then compared to the results of the sky model.
- Inclining the collector or PV array at the latitude of the location increased the yield by 10% (1.88MW/m^2/year to 2.08 MW/m^2/year).
- For the flat plate collector, 8m^2/kW_{th} showed a significant improvement in performance over the 5m^2/kW_{th}, so an optimum collector area should be evaluated based on cost and performance output.
- With varying tank sizes, the 300 liters tank produced higher total refrigeration heat output, as small size allowed it to gain the temperature faster than a larger tank could. As for the operational hours, 1500 liter tank gives more operational hours because it has a higher capacity to store energy compared to the smaller tanks.
- There was no significant improvement in the performance of open loop over direct loop, as noticed in the configurations used for the simulation. The effect depends on ratio of the collector area: storage tank volume. Larger the ratio, more visible the effect is.
- In the life cycle costing for small capacity systems, grid-operated electric chillers are the best option in terms of economic feasibility. Then comes the PV-driven electric chiller followed by absorption chiller which gives the highest lifecycle cost and net present value. However, for large capacity systems, absorption chiller gives the best life cycle cost followed by PV-driven electric chiller. For this case, grid operated electric chiller had the highest life cycle cost, thus making the absorption chiller the best option economically.

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APPENDIXES

Appendix A: MATLAB Script files

A.1 ASRC-CIE Sky Model

Main Program

```
% *************ASRC-CIESkyGen***************
% This script reads station data on solar radiation and daylight to generate
% radiance of 145 sky zones using ASRC-CIE sky luminance model. The
% predecessor is skylumcal.m
pskyiillum=zeros(148,5); %initializes output array.
ppskyillum=[];
jldatedtpnt=[];
output1=[];
fid=fopen('Stationdata15mins.txt','rt'); % open test datat file
ncount=1;
while ncount<=2
    linestring=fgetl(fid); % Remove title and heading
    ncount=ncount+1;
end
linestring=fgetl(fid); % Read location data
location(1:3)=strread(linestring);% Read standard longitude local longitude
% and local latitude
linestring=fgetl(fid); % Remove time heading
linestring=fgetl(fid); % Read Julian date and number of data pointst
linedata=strread(linestring);
dtpoints=linedata(1);   % number of data points of the day
numofdays=linedata(2);
% Initialize time array.
hrmisec=zeros(3);
% Read the next section
linestring=fgetl(fid); % Remove data heading
for count9=1:numofdays
    for incount=1:dtpoints % To read station data of the given set ( of
        % the given day)
        linestring=fgetl(fid); % Read one line of time, solar radiation and
        % daylight data
        linedata=strread(linestring);
        jldt=linedata(15);
        jldatedtpnt=[jldt dtpoints];
        % The following segment computes the equation of time and declination angle
        % for the given day
        beqt=pi*(jldatedtpnt(1)-81)/182;
        eqnt=(9.87*sin(2*beqt)-7.53*cos(beqt)-1.5*sin(beqt))/60;%Equation of time
        declin=23.45*pi*sin(2*pi*(jldatedtpnt(1)+284)/365)/180;
        %Read next section
        hrminsec=linedata(1:3); % time array
        skyrad=linedata(5);  % sky radiation data
        sunrad=linedata(6);  % sun radiation
        skydl=linedata(8);   % sky daylight
        sundl=linedata(9);   % sun daylight
        if skyrad>0
            [solalt,solazm]=fnsolarposition(location,jldatedtpnt,hrminsec,eqnt,declin);
```

```
            pclrindx=((sunrad+skyrad)/skyrad+1.041*(pi/2-solalt)^3)/(1+1.041*(pi/2-
solalt)^3);
            pbrgindx=skyrad/1367/sin(solalt);
            calhskyillm=0; % Set&reset calculated horizontal sky illum
            [pskyillum,calhskyillm]=lumgen145(solalt,solazm,pclrindx,pbrgindx);
            zlumfac=(skyrad)/calhskyillm;
            pskyillum(1:145,4:5)=zlumfac*pskyillum(1:145,4:5);% Structure for each time is:
            % line 1:145 [skyzoneno solalt solazm skylum]
            pskyillum(146,1:3)=hrminsec(1:3);% line 146 [hr min sec cumulative-horizontal
            pskyillum(146,4)=calhskyillm*zlumfac;    % -illuminance-up-to-that-point]
            pskyillum(147,1)=skydl;pskyillum(147,2)=sundl;% line 147 [skydl sundl pclrindx
            pskyillum(147,3)=pclrindx;pskyillum(147,4)=pbrgindx; % pbrgindx]
            pskyillum(148,1:4)=[skyrad sunrad solalt solazm];  %line 148 [skyrad sunrad
            % solalt aolazm]
        else
            pskyillum=zeros(148,5);
            pskyillum(146,1:3)=hrminsec(1:3);
        end
        ppskyillum=[ppskyillum;pskyillum];
    end
end
output1=[output1;ppskyillum];
[arraysize]=size(output1);
rows=arraysize(1,1)
row2=148*dtpoints*numofdays
fclose(fid);
fid2=fopen('skylumcalOP.txt','w');
for iindx=1:rows
    fprintf(fid2,'%9.2f %9.4f %9.4f %9.4f
%9.4f\n',output1(iindx,1),output1(iindx,2),output1(iindx,3),output1(iindx,4),output1(iindx,
5));
end
fclose('all')
```

Function 1: Calculates the Sky Irradiance

```
function pskylum=asrcciesky(solalt,solazm,pclrindx,pbrgindx,palt,pazm)
% Calculation of sky radiance
angdist=acos(sin(solalt)*cos(pi/2-palt)+cos(solalt)*sin(pi/2-palt)*cos(abs(solazm-pazm)));
if pclrindx<1.4
    plumoc=fnocsky(palt); %luminance of overcast sky
    plumin=fninsky(palt,solalt,angdist); % luminance of intermediate sky
    pclrterm=(pclrindx-1)/0.4;pbrgterm=(pbrgindx-0.15)/0.6;
    pclrbrgterm=max([pclrterm pbrgterm]);
    if  pclrbrgterm <= 1
        bin=pclrbrgterm;
    else
        bin=1
    end
    bov=1-bin;
    pskylum=bin*plumin+bov*plumoc;
elseif (pclrindx>1.4)&(pclrindx<3)
    plumin=fninsky(palt,solalt,angdist); % luminance of intermediate sky
    plumtclr=fntclrsky(palt,solalt,angdist);
    btc=(pclrindx-1.4)/1.6;
```

```
    bin=1-btc;
    pskylum=btc*plumtclr+bin*plumin;
elseif pclrindx>=3
    plumtclr=fntclrsky(palt,solalt,angdist);
    plumclr=fnclrsky(palt,solalt,angdist);
    pclterm=(pclrindx-3)/3;
    if pclterm<1
        bcl=pclterm;
    else
        bcl=1;
    end
    btc=1-bcl;
    pskylum=bcl*plumclr+btc*plumtclr;
end
```

Function 2: Function for Clear Sky

```
function [plumclr]=fnclrsky(palt,solalt,angdist)
% Function for clear sky
ncltermi=1-exp(-0.32/sin(palt));
ncltermi2=0.91+10*exp(-3*angdist)+0.45*(cos(angdist))^2;
dclterm=0.27385*(0.91+10*exp(-3*(pi/2-solalt))+0.45*(sin(solalt))^2);
plumclr=ncltermi*ncltermi2/dclterm;
```

Function 3: Function for Intermediate Sky

```
function [plumin]=fninsky(palt,solalt,angdist)
% Function for intermediate sky
ninterm1=0.4399*(1.35*(sin(3.59*palt-0.009)+2.31)*sin(2.6*solalt+0.316)+palt+4.799);
ninterm2=exp(-0.563*((palt+1.059)*(solalt-0.008)+0.812)*angdist);
dinterm1=0.9981*(sin(2.6*solalt+0.316)+2.792);
dinterm2=exp(-1.4806*(solalt+0.30077)*(pi/2-solalt));
plumin=ninterm1*ninterm2/dinterm1/dinterm2;
```

Function 4: Function for Overcast Sky

```
function [plumoc]=fnocsky(palt)
%Function for overcast sky
plumoc=(1+2*cos(pi/2-palt))/3;
```

Function 5: Function for Turbid Clear Sky

```
function [plumtclr]=fntclrsky(palt,solalt,angdist)
%Function for turbid clear sky
nctterm1=1-exp(-0.32/sin(palt));
nctterm2=0.856+16*exp(-3*angdist)+0.3*(cos(angdist))^2;
dctterm=0.27385*(0.856+16*exp(-3*(pi/2-solalt))+0.3*(sin(solalt))^2);
plumtclr=nctterm1*nctterm2/dctterm;
```

Function 6: Function to Calculate Sun Position

```
function [salt sazm]=fnsolarposition(loc,jdpnt,time,eqtime,soldec)
% This function calculates for each given time, solar altitude and solar
% azimuth
% loc is a row vector of standard longitude, local longitude and latitude
% jdpnt contains a row vector of Julian date and number of data
% time ia a row vector of gicen hour, minute and second
jd=jdpnt(1);
thour=time(1)+time(2)/60+time(3)/3600;
soltime=thour-(loc(1)-loc(2))/15+eqtime;
solhrangle=pi*((soltime-12)/12);
salt=asin(sin(pi*loc(3)/180)*sin(soldec)+cos(pi*loc(3)/180)*cos(soldec)*cos(solhrangle));
if loc(3)>soldec
    sazm=asin(cos(soldec)*sin(solhrangle)/cos(salt));
else
    sazm=-pi-asin(cos(soldec)*sin(solhrangle)/cos(salt));
end
```

Function 7: Function to Calculate Intensity of radiance of 145 Sky Points

```
function [pskyillum,calhskyillm]=lumgen145(solalt,solazm,pclrindx,pbrgindx)
% This function calculates intensity from radiance of 145 points in the sky
% using ASRC-CIE model.  The locations of the points are identical to the
% CIE standard locations of sky zones and the locations of EKO sky scanner.
calhskyillm=0;% Initialize calhskyillm
pskylum(147,4)=0;
for ip=1:60
    if ip<31
        palt=pi/30;  % Altitude of first 30 pts is 6 degrees
        pazm=(ip-1)*pi/15;
    else
        palt=pi/10; % Next 30 points 18 deg
        pazm=(ip-31)*pi/15;
    end
    dpalt=pi/15; % Increment of alt is 12 degrees
    dpazm=pi/15;   % Increment of azm is 12 deg
    plum=asrcciesky(solalt,solazm,pclrindx,pbrgindx,palt,pazm);
    pskyillum(ip,1)=ip;
    pskyillum(ip,2)=palt;
    pskyillum(ip,3)=pazm;
    dparea=dpalt*dpazm*cos(palt);
    pskyillum(ip,4)=plum*dparea; % This is actually the normal illuminance
    pskyillum(ip,5)=plum;
    calhskyillm=calhskyillm+pskyillum(ip,4)*sin(palt); % cumulate horizontal illum
end
for ip=61:108
    if ip<85
        palt=pi/6;  % Altitude of first 24 pts is 30 degrees
        pazm=(ip-61)*pi/12;
    else
        palt=7*pi/30; % Next 24 points 42 deg
        pazm=(ip-85)*pi/12;
    end
    dpalt=pi/15;
```

```
    dpazm=pi/12;
    plum=asrcciesky(solalt,solazm,pclrindx,pbrgindx,palt,pazm);
    pskyillum(ip,1)=ip;
    pskyillum(ip,2)=palt;
    pskyillum(ip,3)=pazm;
    dparea=dpalt*dpazm*cos(palt);
    pskyillum(ip,4)=plum*dparea; % This is actually the intensity
    pskyillum(ip,5)=plum;
    calhskyillm=calhskyillm+pskyillum(ip,4)*sin(palt); % cumulate horizontal illum
end
for ip=109:126
    palt=3*pi/10; % Altitude of these 18 pts is 54 degrees
    pazm=(ip-109)*pi/9;
    dpalt=pi/15;
    dpazm=pi/9;
    plum=asrcciesky(solalt,solazm,pclrindx,pbrgindx,palt,pazm);
    pskyillum(ip,1)=ip;
    pskyillum(ip,2)=palt;
    pskyillum(ip,3)=pazm;
    dparea=dpalt*dpazm*cos(palt);
    pskyillum(ip,4)=plum*dparea; % This is actually the illuminance
    pskyillum(ip,5)=plum;
    calhskyillm=calhskyillm+pskyillum(ip,4)*sin(palt); % cumulate horizontal illum
end
for ip=127:138
    palt=11*pi/30; % Altitude of these 18 pts is 66 degrees
    pazm=(ip-127)*pi/6;
    dpalt=pi/15;
    dpazm=pi/6;
    plum=asrcciesky(solalt,solazm,pclrindx,pbrgindx,palt,pazm);
    pskyillum(ip,1)=ip;
    pskyillum(ip,2)=palt;
    pskyillum(ip,3)=pazm;
    dparea=dpalt*dpazm*cos(palt);
    pskyillum(ip,4)=plum*dparea; % This is actually the intensity
    pskyillum(ip,5)=plum;
    calhskyillm=calhskyillm+pskyillum(ip,4)*sin(palt); % cumulate horizontal illum
end
for ip=139:144
    palt=13*pi/30; % Altitude of these 18 pts is 78 degrees
    pazm=(ip-139)*pi/3;
    dpalt=pi/15;
    dpazm=pi/3;
    plum=asrcciesky(solalt,solazm,pclrindx,pbrgindx,palt,pazm);
    pskyillum(ip,1)=ip;
    pskyillum(ip,2)=palt;
    pskyillum(ip,3)=pazm;
    dparea=dpalt*dpazm*cos(palt);
    pskyillum(ip,4)=plum*dparea; % This is actually the intensity
    pskyillum(ip,5)=plum;
    calhskyillm=calhskyillm+pskyillum(ip,4)*sin(palt); % cumulate horizontal illum
end
palt=pi/2;  % Zenith point
pazm=0;
dpalt=pi/30;
dpazm=2*pi;
```

```
plum=asrcciesky(solalt,solazm,pclrindx,pbrgindx,palt,pazm);
pskyillum(145,1)=145;
pskyillum(145,2)=palt;
pskyillum(145,3)=pazm;
dparea=dpalt*dpazm*cos(palt);
pskyillum(145,4)=plum*dparea; % This is actually the incremental illuminance
pskyillum(145,5)=plum;
calhskyillm=calhskyillm+pskyillum(145,4)*sin(palt); % cumulate horizontal illum
```

A.2 Calculating Irradiance on inclined plane and computing absorption chiller performance

Main Program

```
%Reads the output file skylumcalOP and stationdata15mins to get the
%radiation on an inclined plane. Then using the tiltradition, find the
%performance of an absorption system nomillay rated at 4.7kW.
clear
clc
```

Setting the Variables

```
beta=(13.6*pi)/180;     %inclination angle of the plane
plazm=(1*pi)/180;       %azimuth of the inclined plane
ro=0.1;
cw100 = 4.205;          %kJ/kg/C (specific heat of hot water at 100C)
pskyrad=zeros;
pskyrad1=[];
pskyrad2=[];
pskyrad3=[];
tempdRHdate=[];
systemoutput=[];
CoolingToweroutput=[];
STradiation=[];
Iglobv=[];
ccalhskyillmv=[];
testvector=[];
diffuseskymodel=[];
STdiffusev=[];
Testvector=[];
Two=30;                 %Setting initial value to do the simulation
Tcollectorin=75;
mc=0.46;                %Setting initial value to do the simulation
CollArea=38;            %Collector Area m2
tankV=1500;             %(Litres)Volume of tank
tankh=2;                %height of the tank (m)
tankr=((tankV/1000)/((22/7)*tankh))^0.5            %radius of the tank (m)
Qtank=(tankV*cw100*Tcollectorin)*(2.778/10000);    %Set Initial energy stored in tank to
begin simulation
Qg=1;
R=15;                   %Isulation Grading
```

Reading stationdata15mins file to get the number of data points

```
fid1=fopen('Stationdata15mins.txt','rt'); % open test datat file
ncount=1;
while ncount<=5
linestring=fgetl(fid1);     %Jumping to the no. of Data points
ncount=ncount+1;
end
linedata=strread(linestring);
datapnt=linedata(1);        %Julian date and no. of Data points
numofdays=linedata(2);
```

```
ncount2=1;
linestring=fgetl(fid1);     %Remove Heading
for ncount2=1:(datapnt*numofdays)
    linestring=fgetl(fid1);
    linedata=strread(linestring);
    Iglob=linedata(4);      %Global radiation measured
    Tamb=linedata(10);      %Ambient temperature
    RH=linedata(11);        %Relative Humidity
    year=linedata(12);      %Year
    month=linedata(13);     %Month
    day=linedata(14);       %Date
    tempRHdate1=[Tamb RH year month day];
    tempdRHdate=[tempdRHdate; tempRHdate1];
    Iglobv=[Iglobv;Iglob];
end
fclose(fid1);
```

Reading skylumcalOP data file to get the sky model values on a horizontal

```
%plane
Data=zeros;
fid2=fopen('skylumcalOP.txt','r');
formatspec='%f %f %f %f %f';
sizeA=[5 Inf];
skylumcalOP=fscanf(fid2,formatspec,sizeA);
fclose(fid2);
skylumcalOP=skylumcalOP';
sizeskylumcalOP=size(skylumcalOP)
```

Assigning and Evaluation

```
Data=zeros;
%fid2=fopen('skylumcalOP.txt','rt');
caltiltrad=0;
ncount3=1;  %Counter for reading the 145 sky points
ncount4=1;  %Counter for reading the no.of data points
ncount5=1;  %Counter seperate each days data points
ncount6=1;  %Counter for the for loop to seperate days
ncount8=1;
row=0;      %counter for reading temerature
check=0;

for ncount6=1:numofdays
    for ncount4=1:datapnt
        row=row+1;
        caltiltrad=0;
        for ncount3=1:145
            pnum=skylumcalOP(1,1);                %point number
            palt=skylumcalOP(1,2);                %point altitude
            pazm=skylumcalOP(1,3);                %point azimuth
            phrad=skylumcalOP(1,4);               %point radiance contributing to
            %the horizontal plane
            costheta=vsvndotproduct(palt,pazm,beta,plazm); %cosineangle between the tilted
            %plane normal and point
```

```
            ptiltrad=phrad*costheta;         %normal radiance on the tilted plane
            caltiltrad=ptiltrad+caltiltrad;  %summing up the diffuse radiance values
            pskyrad=[pnum palt pazm phrad ptiltrad];
            pskyrad1=[pskyrad1;pskyrad];
            skylumcalOP(1,:)=[];
        end
        %Reading the Next Part of the data file
        %Time and cumulated horizontal sky radiance
        hrminsec(1:3)=skylumcalOP(1,1:3);   %linedata(1:3);
        calhskyillm=skylumcalOP(1,4);       %linedata(4);
        skylumcalOP(1,:)=[];
        %Sky luminance, sun luminance, clearness index and brightness index
        skydl=skylumcalOP(1,1);
        sundl=skylumcalOP(1,2);
        pclindx=skylumcalOP(1,3);
        pbrgindx=skylumcalOP(1,4);
        skylumcalOP(1,:)=[];
        %Sky radiance, beam radiance, solar altitude and solar azimuth
        skyrad=skylumcalOP(1,1);%linedata(1);
        sunrad=skylumcalOP(1,2);%linedata(2);
        solalt=skylumcalOP(1,3);%linedata(3);
        solazm=skylumcalOP(1,4);%linedata(4);
        testvector=[testvector; hrminsec skyrad sunrad solalt solazm];
        skylumcalOP(1,:)=[];
        %Ouput Array
        pskyrad1=[pskyrad1;hrminsec calhskyillm caltiltrad;skydl sundl pclindx pbrgindx
0;skyrad sunrad solalt solazm 0];
        %Adding the Beem radiance to the cumulated diffuse radiance calculated
        Ib=sunrad*sin(solalt);
        ccalhskyillm=(calhskyillm+Ib);% Beam radiance + Diffuse radiance from sky model
        ccalhskyillmv=[ccalhskyillmv;ccalhskyillm];
        %Beam Radiannce + Sky model diffuse radiance on the tilted plane
        numangle=((pi/2)-beta-solalt);
        angle1=((pi/2)-solalt);
        angle2=(solalt+beta-90);
        if (beta+solalt)>90
            denangle=angle2;
        else
            denangle=angle1;
        end
        Rb=cos(numangle)/cos(denangle); %Rb factor
        ccaltiltrad=caltiltrad+(Ib*Rb)+ro*((1-cos(beta))/2)*Iglobv(row,1);%summing beam and
diffuse
        Ittheta=(((1+cos(beta))/2)*skyrad)+(Ib*Rb)+ro*((1-cos(beta))/2)*Iglobv(row,1);
        pskyrad2=[hrminsec Ittheta ccaltiltrad];
        pskyrad3=[pskyrad3; pskyrad2];
        %Using the data to find the performance of absorption system

[Tg,Two,Tc,Qr,COP,Qg,Tcollectorout,Twi,ncount5,Qc,Twetb,Qtank,Tcollectorin,Ttankout,Ttankin
,Qcollector,Qtanki]=fnmodel(CollArea,tempdRHdate(row,1),tempdRHdate(row,2),ccaltiltrad,Two,
ncount5,mc,Qtank,cw100,tankV,Qg,tankr,tankh,R);
        systemoutput1=[Tcollectorout Tg Tcollectorin Twetb Two Twi Tc mc Qr Qg Qc COP];
        Testvector=[Testvector;Tcollectorin Tcollectorout Ttankout Ttankin Qtanki
Qcollector Qg];
        systemoutput=[systemoutput;systemoutput1];
        CoolingToweroutput1=[Two Twi];
```

```
        CoolingToweroutput=[CoolingToweroutput;CoolingToweroutput1];
    end
    check=check+1;
    Two=25;
    % Storage tank Standing Loss
    for ncount8=1:12 % Standing time without operation is 12 hours
        Qtank=fnstoragetank(Qtank,tempdRHdate(row,1),tankV,tankr,tankh,R);
        %if open loop fncollectorloss is exist, else 0
%         Qtank=fncollectorloss(Qtank,tempdRHdate(row,1),tankV,cw100,CollArea);
        pipelength=22;  %25 to the pipes to tank 22 from the tank
        Qtank=fnpipeloss(Qtank,tempdRHdate(row,1),tankV,cw100,pipelength);
    end
    ncount5=1;
end
skyradtempRHdate=[pskyrad3 ccalhskyillmv Iglobv tempdRHdate];
systemoutput;
CoolingToweroutput;
%check;
%Testvector;
% STradiation;
```

Generating Outputfile

```
%2 output files generted. One file give the radiation data, and the second
%file give the absorption system performance data.
fid3=fopen('tiltradtempRHdate.txt','w');
fprintf(fid3,'No.of data points/day  No.of days\n');
fprintf(fid3,'%9.0f     %9.0f\n',datapnt(1),numofdays(1));
fprintf(fid3,'hr min sec Ittheta Cum.Tilt.Rad Cum.Horz.Rad Iglob Tamb RH year month
day\n');
for iindx=1:(datapnt*numofdays)
     fprintf(fid3,' %9.0f %9.0f %9.0f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.0f %9.0f
%9.0f\n',skyradtempRHdate(iindx,1),skyradtempRHdate(iindx,2),skyradtempRHdate(iindx,3),skyr
adtempRHdate(iindx,4),skyradtempRHdate(iindx,5),skyradtempRHdate(iindx,6),skyradtempRHdate(
iindx,7),skyradtempRHdate(iindx,8),skyradtempRHdate(iindx,9),skyradtempRHdate(iindx,10),sky
radtempRHdate(iindx,11),skyradtempRHdate(iindx,12));
end
fclose(fid3);
%Second output file for the Absorption system performance
fid4=fopen('Systemperfmance.txt','w');
fprintf(fid4,'Tga Tg Tcolli Twetb Two Twi Tc mc Qr Qg Qc COP\n');
for td=1:(datapnt*numofdays)
    fprintf(fid4,' %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f
%9.2f\n',systemoutput(td,1),systemoutput(td,2),systemoutput(td,3),systemoutput(td,4),system
output(td,5),systemoutput(td,6),systemoutput(td,7),systemoutput(td,8),systemoutput(td,9),sy
stemoutput(td,10),systemoutput(td,11),systemoutput(td,12));
end
fclose(fid4);
fid5=fopen('hotwatergeneration.txt','w')
fprintf(fid5,'Tcollectorin Tcollectorout Ttankout Ttankin Qtank Qcollector Qg\n');
for ht=1:(datapnt*numofdays)
    fprintf(fid5,'%9.2f %9.2f %9.2f %9.2f %9.2f %9.2f
%9.2f\n',Testvector(ht,1),Testvector(ht,2),Testvector(ht,3),Testvector(ht,4),Testvector(ht,
5),Testvector(ht,6),Testvector(ht,7));
end
```

```
%Closing the opened files
fclose('all');
```

Function 1: Function for Absorption Chiller Model

```
function
[Tg,Two,Tc,Qr,COP,Qg,Tcollectorout,Twi,counter,Qc,Twetb,Qtank,Tcollectorin,Ttankout,Ttankin
,Qcollector,Qtanki] =
fnmodel(A,Ta,RH,I,Two,counter,mc,Qtank,cw100,tankV,Qg,tankr,tankh,Rtank)
%Absorption System Model from the Graph
```

Defining Variables Collector Variables

```
cw30 = 4.18;     %kJ/kg/C (specific heat of cooling water at 30C)
mh = 0.6;        %kg/s (mass flow rate of hot water)
va=0.7;          %m3/s (volume flowrate of air, 25m3/min)
CT = 3;          %CT+ (Correction factor for Condenser Temperature)
data=[];
Tg=[];           %Range 75-100 degrees C
Test2=[];
Tc=10;           %Range 7-17 degrees C
qrgraph=[];
copgraph=[];
Tmax=100;
Qtankmax=(tankV*cw100*Tmax)*1000*(2.778/10000000);
```

Reading the Graph Data files

```
%Qr Performance Curves
qrgraph=[];
fid1=fopen('qrgraph.txt','rt');
ncount=1;
for ncount=1:12
    linestring=fgetl(fid1);
    linedata=strread(linestring);
    qrgraph=[qrgraph;linedata];
end
fclose(fid1);
%COP Performance Curves
fid2=fopen('copgraph.txt','rt');
ncount=1;
for ncount=1:12
    linestring=fgetl(fid2);
    linedata=strread(linestring);
    copgraph=[copgraph;linedata];
end
fclose(fid2);
```

Calculating Tg, Tw, Qr, COP and Qg

```
    %Calculating Tw using Empiral formula
    Twetb=wetbulbtemp(Ta,RH);
```

```
    %Calculating Tg
    %When there is no radiation
    if I==0
        Tcollectorout=Ta;
        Tg=Ta;
        Qr=0;
        COP=0;
        Qg=0;
        Ttankin=Ta;
    else
    %When radiation is available calculating the Tg using the
    %Characteristic equation of solar collector
    Qtanki=Qtank;
    if Qtank<Qtankmax
        Ttankout=(Qtank/(2.778/10000000))/(1000*tankV*cw100); %input to absorption system
        Tcollectorin=Ttankout;
        Tcollectorout=solx(I,Ta,Tcollectorin,mh,cw100,A);
        Qcollector=mh*cw100*(Tcollectorout-Tcollectorin); %Heat into Tank
        if Qcollector<0
            Qcollector=0;
        else
            Qcollector=Qcollector;
        end
    else
        Ttankout=Tmax;%(Qtankmax/(2.778/10000000))/(1000*tankV*cw100); %input to absorption
system
        Tcollectorin=Ttankout;
        Tcollectorout=Tmax;%solx(I,Ta,Tcollectorin,mh,cw100,A);
        Qcollector=Qg;%mh*cw100*(Tcollectorout-Tcollectorin); %Heat into Tank
    end
    Tg=Ttankout;
        %Calculating Qr COP and Qgen
        %Absorption System working range is 75>Tg<100 for 8>Tc<17
        %and 80>Tg<100 for 7>Tc<17
        if Tc>=7 && Tc<8
            if Tg>=77
                %Calculating Qr and COP using the system performance curves
                Qr=refheat(qrgraph,Tg,Tc,Two);
                COP=performance(copgraph,Tg,Tc,Two);
                Qg=Qr/COP;
                Ttankin=Tg-(Qg/mh/cw100);    %Setting new Ti (Return to Tank)
            else
                Qr=0;
                COP=0;
                Qg=0;
                Ttankin=Tg;
            end
        elseif Tc>=8 && Tc<=17
            if Tg>=75
                %Calculating Qr and COP using the system performance curves
                Qr=refheat(qrgraph,Tg,Tc,Two);
                COP=performance(copgraph,Tg,Tc,Two);
                Qg=Qr/COP;
                Ttankin=Tg-(Qg/mh/cw100);    %Setting new Ti
            else
                Qr=0;
```

```
                COP=0;
                Qg=0;
                Ttankin=Tg;
            end
        end
    end
    Qc=Qg+Qr;
    if Qc~=0
        %Output of Cooling water with heat gain from Chiller System for the
        %next reading
        Twi=(Qc/(mc*cw30))+Two;
        %Output of Cooling tower to be the input cooling water for the next
        %reading
        [Two]=fncoolingtower(cw30,va,Ta,RH,Twi,mc);
    else
        Twi=Two;
        Two=Twi;
    end
    %Energy Balance at the tank
    Qgenerator=Qg;  %Heat out of Tank
    if Qtank<Qtankmax
        Qtank=Qtank-Qgenerator+Qcollector;  % Energy stored in tank
        Qtank=fnstoragetank(Qtank,Ta,tankV,tankr,tankh,Rtank);
        pipelength=22+25;
        Qtank=fnpipeloss(Qtank,Ta,tankV,cw100,pipelength);
    else
        Qtank=Qtankmax;
    end
%Counter to set the collector water temperature input = to previous days
%collector temperature output
counter=counter+1;
end
```

Function 2: Function to Evaluate Flat Plate Collector Output

```
function [Tg] = solx(I,Ta,Ti,mh,cw100,A)
%This function calcultes the Output Temperature of Solar Collector using
%the collector characteristic equation.
%%Defining Variables
n=0.732;
a1=3.1239; % design point of the system
a2=0.0143;
```

Solution using the Quadratic equation

```
%%Calculation
a=A*a2;
b=(A*a1)+(mh*cw100*1000);
c=-1*((A*I*n)+(mh*cw100*1000*(Ti-Ta)));
%Since the positive solution is the only real outcome only that is
%considered
solx1=(-b/2/a)+(sqrt((b^2)-(4*a*c)))/(2*a);
% solx2=(-b/2/a)-(sqrt((b^2)-(4*a*c)))/(2*a);
```

```
Tg=solx1+Ta;
end
```

Function 3: Function to do dotproduct of two vectors

```
function[costheta]=vsvndotproduct(palt,pazm,beta,plazm)
% This Function Calculates the normal vectore of the tilted Plane
% and Vector of the corresponding point. Then it performs dot product of
% the vectors to find angle between the two vectors and returns the
% costheta by checking if the angle is in the visible range of the plane.

% Calculate Vs and Vn Vector
Vs=[sin(palt);(cos(palt).*sin(pazm));(-cos(palt).*cos(pazm))];
Vn=[cos(beta);(-sin(beta).*sin(plazm));(-sin(beta).*cos(plazm))];
VsdotVn=dot(Vs,Vn);
Vsmod=(((Vs(1,1)^2)+(Vs(2,1)^2)+(Vs(3,1)^2))^(1/2));
Vnmod=(((Vn(1,1)^2)+(Vn(2,1)^2)+(Vn(3,1)^2))^(1/2));
costheta1=(VsdotVn/(Vsmod*Vnmod));

%Check if point is in visible range to the plane to contribute
if costheta1>0
    costheta=costheta1;
else
    costheta=0;
end
```

Function 4: Function to do interpolation of Qr and COP from the graph

```
function [y] = interpolate(x,x1,x2,y1,y2)
% This function calculates the value of y using interpolation.
% for the point (y,x), on the line of know coordinates (y1,x1) and (y2,x2)
 y=((y2-y1)*((x-x1)/(x2-x1)))+y1;
end
```

Function 5: Function Evaluate COP based on new Tw

```
function [COP] = coptwevaluation(COP,Tw )
%This function evaluates COP based on Tw
%Defining the Variables
COPold=COP;
gradient=-0.0288;
yintercept=0;
COPnew=0;
Twref=29.5;
%Evaluating the intercept
yintercept=COPold-(gradient*Twref);
%Re-evaluating COP based on the yintercept
COPnew=gradient*Tw+yintercept;
%Assigning the new COP
COP=COPnew;
end
```

Function 6: Function Evaluate Qr based on new Tw

```
function [Qr] = Qrtwevaluation(Qr,Tw )
%This function evaluates COP based on Tw
%Defining the Variables
Qrold=Qr;
gradient=-0.44;
yintercept=0;
Qrnew=0;
Twref=29.5;
%Evaluating the intercept
yintercept=Qrold-(gradient*Twref);
%Re-evaluating COP based on the yintercept
Qrnew=gradient*Tw+yintercept;
%Assigning the new COP
Qr=Qrnew;
end
```

Function 7: Function to Evaluate Performance from the graph data

```
function COP=performance(copgraph,Tg,Tc,Tw)
%This function Calculates the COP of the absorption system using the
%COP performance curve of the system at first at constant Tw (29.5), and
%then re-evaluates the value based on the Tw value.
```

Define the Variables

```
COP=0;
Tg1=0;
Tg2=0;
Tc1=0;
Tc2=0;
A=0;
B=0;
% copgraph=[];
```

Check the Tg & Tc value at constant Tw=29.5

```
%Find the line segment where the desired point lies to do the interpolation
i=2;
j=2;
while i<7
    if Tg==copgraph(1,i)
        Tg1=copgraph(1,i);
        break
    elseif Tg>copgraph(1,i) && Tg<copgraph(1,(i+1))
        Tg1=copgraph(1,i);
        Tg2=copgraph(1,(i+1));
        break
    end
    i=i+1;
end
 while j<12
```

```
        if Tc==copgraph(j,1)
            Tc1=copgraph(j,1);
            break
        elseif Tc>copgraph(j,1) && Tc<copgraph((j+1),1)
            Tc1=copgraph(j,1);
            Tc2=copgraph((j+1),1);
            break
        end
        j=j+1;
    end

copseg=[Tg1 Tg2;Tc1 Tc2];
```

Interpolation of the variables, on the respective line segment

```
A=0; B=0; C=0; D=0; E=0;
%Interpolate on Tg1 line
if Tc2>0
    COP1=interpolate(Tc,Tc1,Tc2,(copgraph(j,i)),(copgraph((j+1),i)));
    A=A+1;
    if Tg2>0
        COP2=interpolate(Tc,Tc1,Tc2,(copgraph(j,(i+1))),(copgraph((j+1),(i+1))));
        COP=interpolate(Tg,Tg1,Tg2,COP1,COP2);
        B=B+1;
    else
        COP=COP1;
        C=C+1;
    end
else
    if Tg2>0
    COP1=interpolate(Tg,Tg1,Tg2,(copgraph(j,i)),(copgraph(j,(i+1))));
    COP=COP1;
    D=D+1;
    else
    COP=copgraph(j,i);
    E=E+1;
    end
end

Checkji=[i j];
Checkvec=[A B C D E];

%%COP Re-evaluation based on Tw value
COP1=COP;
COP=coptwevaluation(COP,Tw);
end
```

Function 8: Function to Evaluate Qr from the graph data

```
function Qr=refheat(qrgraph,Tg,Tc,Tw)
%This function Calculates the Qr of the absorption system using the
%Qr performance curve of the system at first at constant Tw (29.5), and
%then re-evaluates the value based on the Tw value.
```

Define the Variables

```
Qr=0;
Tg1=0;
Tg2=0;
Tc1=0;
Tc2=0;
```

Finding the corresponding line segment to do the interpolation

```
i=2;
j=2;
while i<7
    if Tg==qrgraph(1,i)
        Tg1=qrgraph(1,i);
        break
    elseif Tg>qrgraph(1,i) && Tg<qrgraph(1,(i+1))
        Tg1=qrgraph(1,i);
        Tg2=qrgraph(1,(i+1));
        break
    end
    i=i+1;
end
 while j<12
     if Tc==qrgraph(j,1)
         Tc1=qrgraph(j,1);
         break
     elseif Tc>qrgraph(j,1) && Tc<qrgraph((j+1),1)
         Tc1=qrgraph(j,1);
         Tc2=qrgraph((j+1),1);
         break
     end
     j=j+1;
 end

Qrseg=[Tg1 Tg2;Tc1 Tc2];
```

Interpolation of the variables, on the corresponding line segment

```
A=0; B=0; C=0; D=0; E=0;
%Interpolate on Tg1 line
if Tc2>0
    Qrg1=interpolate(Tc,Tc1,Tc2,(qrgraph(j,i)),(qrgraph((j+1),i)));
    A=A+1;
    if Tg2>0
        Qrg2=interpolate(Tc,Tc1,Tc2,(qrgraph(j,(i+1))),(qrgraph((j+1),(i+1))));
        Qr=interpolate(Tg,Tg1,Tg2,Qrg1,Qrg2);
        B=B+1;
    else
        Qr=Qrg1;
        C=C+1;
    end
else
    if Tg2>0
```

```
    Qrg1=interpolate(Tg,Tg1,Tg2,(qrgraph(j,i)),(qrgraph(j,(i+1))));
    Qr=Qrg1;
    D=D+1;
    else
    Qr=qrgraph(j,i);
    E=E+1;
    end
end

Checkji=[i j];
Checkvec=[A B C D E];
```

Qr re-evaluation based on Tw Value

```
Qr1=Qr;
Qr=Qrtwevaluation(Qr,Tw);
end
```

Function 9: Performance graph data

0	75	80	85	90	95	100
7	0	0.222222222	0.3778	0.4667	0.5333	0.5667
8	0.155555556	0.322222222	0.4778	0.55	0.6	0.6222
9	0.266666667	0.422222222	0.56	0.611	0.65	0.6611
10	0.372222222	0.522222222	0.61667	0.6555	0.6889	0.6889
11	0.472222222	0.583333333	0.6444	0.69375	0.6889	0.6889
12	0.555555556	0.638888889	0.6778	0.69375	0.6889	0.6889
13	0.611111111	0.661111111	0.69375	0.69375	0.6889	0.6889
14	0.644444444	0.661111111	0.69375	0.69375	0.6889	0.6889
15	0.652777778	0.661111111	0.69375	0.69375	0.6889	0.6889
16	0.652777778	0.661111111	0.69375	0.69375	0.6889	0.6889
17	0.652777778	0.661111111	0.69375	0.69375	0.6889	0.6889

Function 10: Refrigeration heat graph data

0	75	80	85	90	95	100
7	0	1.25	2.5	3.75	4.875	5.75
8	0.75	1.75	3	4.3125	5.5	6.375
9	1.25	2.375	3.5625	4.875	6	6.875
10	1.75	2.875	4.1875	5.1875	6.25	7.1875
11	2.1875	3.375	4.4375	5.3125	6.25	7.125
12	2.625	3.625	4.75	5.3125	6.25	7.125
13	2.9375	3.75	4.8125	5.3125	6.25	7.125
14	3	3.75	4.8125	5.3125	6.25	7.125
15	3	3.75	4.8125	5.3125	6.25	7.125
16	3	3.75	4.8125	5.3125	6.25	7.125
17	3	3.75	4.8125	5.3125	6.25	7.125

Function 11: Function for Cooling Tower Model

```
function [Two] = fncoolingtower(cw30,va,Ta,RH,Twi,mc)
% Cooling tower function to find the Outlet water temperature from cooling
% tower
```

Defining Variables

```
hai=[];
hsatTwi=[];
Pa=101325; %Pa
prow=1000; %density of water (kg/m3)
a=0.9325; %Parameters for calculating NTU
n=0.4887; %Parameters for calculating NTU
```

Calculation

```
Proa=(Pa/(287.058*(Ta+273.15)));
% Calculate mass flowrate of inlet air (kg/s)
ma=va*Proa;
% Calculate enthalpy of saturated water inlet
[hai,wai]=fnhsat(Ta,RH);
%Calculate of enthalpy of air leaving after saturation
[hsatTwi,wsatTwi]=fnhsat(Twi,100);
%Assume a value for effectiveness
eps=0.5;
Qcl=eps*ma*(hsatTwi-hai);
Two=Twi-(Qcl/(mc*cw30));
```

Iteration to get final Two

```
A=1;
newTwo=0;
while (abs(Two-newTwo))>0.01
    A=A+1;
    if newTwo~=0
        Two=newTwo;
    end
    %Calculate hsatwo effective specific heat of saturated air
    [hsatTwo,wsatTwo]=fnhsat(Two,100);
    %Effective heat of saturated air
    Cs=(hsatTwi-hsatTwo)/(Twi-Two);
    R=(ma*Cs)/(mc*cw30);
    %Finding number of transfer unit (NTU) using Braun et al. 1989
    NTU=a*((mc/ma)^n);
    %Calculating new value of effectiveness using regression model
    neweps=(1-exp(-NTU*(1-R)))/(1-(R*exp(-NTU*(1-R))));
    %Enthalpy of air outlet
    hao=hai+neweps*(hsatTwi-hai);
    %Effective saturation enthalpy
    hsate=hai+((hao-hai)/(1-exp(-NTU)));
    %Effective saturation Temperature
    [Tsate,wsate]=fniterateTsatenwsate(hsate);
    %calculating new value of wao
    wao=wsate+(wai-wsate)*exp(-NTU);
    %Calculating new Two
    newTwo=((ma*(hao-hai))-(mc*cw30*Twi))/((ma*cw30*(wao-wai))-(mc*cw30));
end
end
```

Function 12: Function to Evaluate Effective saturation Temperature and Humidity

```
function [Tsate,wsate] = fniterateTsatenwsate(hsat)
%This function finds Tsate and wsate from hsate by iteration

%Reference data for Simplify iteration
hsatdata=[20   57.051;21      60.49198;22    64.07701;23    67.81383;24    71.71061
25      75.77594;26    80.01888;27    84.44895;28    89.07622;29    93.9113;30     98.96536
31      104.2502;32    109.7784;33    115.5631];

%identifying point to start iteration
for i=1:14
    if hsat>=hsatdata(i,2)&& hsat<=hsatdata(14,2)
        if hsat<=hsatdata((i+1),2)
            beginiterate=hsatdata(i,1);
            break
        end
    elseif hsat<hsatdata(i,2)
        beginiterate=15;
    else
        beginiterate=33;
    end
end

%Defining Variable for iteration
hsat1=0;
Tsat1=beginiterate;
%Startin iteration
A=1;
while (abs(hsat1-hsat))>=0.001
    A=A+1;
    Tabs=(Tsat1+273.15);
    a=53.5224;
    b=6834.27/(Tabs);
    c=5.17*ln(Tabs);
    pv=exp(a-b-c); %saturation vapor pressure
    % At RH 100% saturation vapor pressure = partial pressure
    wsate=(0.622*pv)/(101.3-pv); % Humidity ratio
    hdryair=1.006*Tsat1; % enthalpy of dry air
    hwvapor=wsate*(2501+(1.805*Tsat1)); %enthalpy of water vapor
    hsat1=hdryair+hwvapor;  %enthalpy of saturated air
    Tsat1=Tsat1+0.0001;
    if hsat1>hsat
        break
    end
end
Tsate=Tsat1-0.0001;
end
```

Function 13: Function to Calculate Saturation Enthalpy

```
function [hsat,w] = fnhsat(T,RH)
%This function calculates the enthalpy of saturated air at given
%temperatures
Tamb=T;
```

```
Tabs=(Tamb+273.15);
a=53.5224;
b=6834.27/Tabs;
c=5.17*ln(Tabs);
pvs=exp(a-b-c); %saturation vapor pressure
pv=pvs*(RH/100); % partial vapor pressure
w=(0.622*pv)/(101.3-pv); % Humidity ratio
hdryair=1.006*Tamb; % enthalpy of dry air
hwvapor=w*(2501+(1.805*Tamb)); %enthalpy of water vapor
hsat=hdryair+hwvapor;  %enthalpy of saturated air
end
```

Function 14: Function to Evaluate Storage Tank Loss

```
function [Qtank] = fnstoragetank(Qtank,Ta,V,r,h,R)
% This function is the model for storage tank and calculates the
% temperature drop per hour.
```

Defining the Functions

```
%Tank Specifications (Cylindrical Tank)
cw100 = 4.205;          %specific heat of hot water
```

Calculations

```
U=1/(R*0.17611);            %Heat transfer coefficient of tank
A=(2*pi*r*h)+(2*pi*r*r);    %Surface Area of Tank
%Qtank is in kW, converting to kJs
QtankkW=Qtank;
Qtankkjs=(QtankkW/(2.778/10000000))/1000;
T1=Qtankkjs/(V*cw100);
Q=(U*A*(T1-Ta))*3.6;        %Heat Transfer (kJ/s)
Qtankkjs=Qtankkjs-Q;
QtankkW=(Qtankkjs*1000*(2.778/10000000));
Qtank=QtankkW;
end
```

Function 15: Function to Evaluate Pipe Loss

```
function [Qtank] = fnpipeloss(Qtank,Ta,tankV,cw100,L)
% Calculates the Energy loss in the water pipes
```

Defining Variables

```
cw30 = 4.18;    %kJ/kg/C (specific heat of cooling water at 30C)
r=0.022;       % pipe radius (m) (2inch)
Rpipe=3*0.17611;   % Thermal Resistanc  (m2K/W)
```

Calculations

```
Qtankmin=(tankV*cw30*Ta)*(2.778/10000); %Minimum Q in tank at Tam
QcurrenttankkW=Qtank;
QcurrenttankkJ=(QcurrenttankkW/(2.778/10000000))/1000;
Tpipe=QcurrenttankkJ/(tankV*cw100);
U=1/Rpipe;
pipesurfaceare=L*(22/7)*r*2;
Qpipelloss=(U*pipesurfaceare*(Tpipe-Ta))/1000; %kJ
QcurrenttankkJ=QcurrenttankkJ-Qpipelloss;
Qtank=(QcurrenttankkJ*1000*(2.778/10000000));
if Qtank<Qtankmin
    Qtank=Qtankmin;
end
end
```

Function 16: Function to Evaluate Collector Loss

```
function [Qtank] = collectorloss(Qtank,Ta,tankV,cw100,CollArea)
% Calculates the Energy loss if the water remains in the collector
```

Defining Variables

```
cw30 = 4.18;    %kJ/kg/C (specific heat of cooling water at 30C)
kal=385;    % Aluminum pipe conductivity(W/mk)
deltax=0.002;    %pipe thicknes (m)
r=0.0254;       % pipe radius (m) (2inch)
watervolume=(0.7924/1000); %per sq meter of coll area
```

Calculations

```
Qtankmin=(tankV*cw30*Ta)*(2.778/10000); %Minimum Q in tank at Tam
QcurrenttankkW=Qtank;
QcurrenttankkJ=(QcurrenttankkW/(2.778/10000000))/1000; %Convert to kJ
Tcoll=QcurrenttankkJ/(tankV*cw100); %Calcualte Temperature
CrossArea=(22/7)*r;
R=deltax/kal; %Thermal Resistanc  (m2K/W)
U=1/R; %Overal heat transfer coefficient
pipelength=(watervolume*CollArea)/CrossArea;
pipesurfaceare=2*pipelength*(22/7)*r;
Qcolloss=(U*pipesurfaceare*(Tcoll-Ta))/1000; %kJ
QcurrenttankkJ=QcurrenttankkJ-Qcolloss;
Qtank=(QcurrenttankkJ*1000*(2.778/10000000));
if Qtank<Qtankmin
    Qtank=Qtankmin
end
end
```

Function 17: Function to Calculate Natural Logarithm

```
function [x] = ln(y)
%returns the ln(x) of the value
```

```
x=(log(y)/(log(exp(1))));
end
```

Function 18: Function to Calculate Wet bulb Temperature

```
function [Tw] = wetbulbtemp(Ta,RH )
%Calculates Wetbulb Temperature from air Temperature (Ta) and Relative
%Humidity (RH) using empirical formula by Roland Stull, University of
%British Columbia, Vancouver, British Columbia, Canada (2011
%Empirical Formula
Tw=(Ta*atan(0.151977*((RH+8.31659)^(1/2))))+(atan(Ta+RH))-(atan(RH-
1.676331))+(0.00391838*((RH)^(3/2))*atan(0.023101*RH))-4.686035;
end
```

A.3 Life Cycle Cost Calculation of Absorption Chiller and New Absorption Chiller based on Electric Chiller cost

Main Program

```
% This Program Calculates the Life Cycle Cost of an Equipment
clear
clc
```

Defining Variables

```
i=0.07;             %Discount Rate, assuming 7%
ophrsperday=12;      %Operating hours per day
numofdays=366;      %No.of days per year to use for LCA
tarrif=0.15;         %Electricity Tarrif 3THB/kWh
% Absorption Chiller
numofequip=4;       %Num.of Equipments in the data file
% Setting Files IDs
fid1='chillertdetails.txt';
fid2='chillerlca.txt';
fid3='newchillerlca.txt';
% Other Variables
tempoutput=[];
electricchilleranncost=2142;
```

Reading the Data file

```
[Description life incost epower spower cpower]=textread(fid1,'%s %f %f %f %f
%f',numofequip);
for j=1:numofequip
    SPWF=(((1+i)^life(j,1))-1)/(i*((1+i)^life(j,1)));       %Series Present Worth Factor
    CRF=1/SPWF;                                             %Capital Return Factor
    Annuity=CRF*incost(j,1);                          %Annuity
    AnEnergycost=epower(j,1)*ophrsperday*numofdays*tarrif;  %Annual Energy Cost
    AnCost=Annuity+AnEnergycost;                          %Annual Cost
    Presentvalue=AnCost*SPWF;                              %Present Value
    tempoutput=[tempoutput;incost(j,1) epower(j,1) SPWF CRF Annuity AnEnergycost AnCost
Presentvalue];
end
% if numofequip>1
    Total=sum(tempoutput);
    NetPresentValue=Total(1,8);     %Net Present Value
    NetAnCost=Total(1,7);           %Net Annual Cost
```

Printing Original LCA Output file

```
%Compiling Output file
fields1=[];
Title=['Description' 'Life' 'Initial.cost' 'E.Power' 'SPWF' 'CRF' 'Annuity' 'An.Energycost'
'An.Cost' 'Presentvalue'];
for f=1:numofequip
fields=[Description(f,1) life(f,1) tempoutput(f,1) tempoutput(f,2) tempoutput(f,3)
tempoutput(f,4) tempoutput(f,5) tempoutput(f,6) tempoutput(f,7) tempoutput(f,8)];
```

```
fields1=[fields1;fields];
end
% Printing Output file
fid2=fopen(fid2,'w');
fprintf(fid2,'Description Life Initial.cost E.Power SPWF CRF Annuity An.Energycost An.Cost
Presentvalue\n');
formatspec='%s %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f\n';

for k=1:numofequip
    fprintf(fid2,'%s %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f
%9.2f\n',fields1{k,1},fields1{k,2},fields1{k,3},fields1{k,4},fields1{k,5},fields1{k,6},fiel
ds1{k,7},fields1{k,8},fields1{k,9},fields1{k,10});
end
fprintf(fid2,'Net.Present.Value=%9.2f\n',NetPresentValue);
fprintf(fid2,'Net.Annual.Cost=%9.2f\n',NetAnCost);
```

Comparing with Electric Chiller Annual Cost

```
A=0
tempoutput;
NetPresentValue
NetAnCost
while NetAnCost>electricchilleranncost
    A=A+1;
    tempoutput(1,1)=tempoutput(1,1)-1;
    NewAnnuity=tempoutput(1,4)*tempoutput(1,1);
    NewAnEnergycost=tempoutput(1,2)*ophrsperday*numofdays*tarrif;
    NewAnCost=NewAnnuity+NewAnEnergycost;
    NewPresentavalue=NewAnCost*tempoutput(1,3);
    tempoutput(1,1:8)=[tempoutput(1,1) tempoutput(1,2) tempoutput(1,3) tempoutput(1,4)
NewAnnuity NewAnEnergycost NewAnCost NewPresentavalue];
    %Summing Colums
    Total=sum(tempoutput);
    NetPresentValue=Total(1,8);     %Net Present Value
    NetAnCost=Total(1,7);           %Net Annual Cost
end
tempoutput;
```

Compiling Output file

```
fields2=[];
Title=['Description' 'Life' 'Initial.cost' 'E.Power' 'SPWF' 'CRF' 'Annuity' 'An.Energycost'
'An.Cost' 'Presentvalue'];
for f=1:numofequip
fields=[Description(f,1) life(f,1) tempoutput(f,1) tempoutput(f,2) tempoutput(f,3)
tempoutput(f,4) tempoutput(f,5) tempoutput(f,6) tempoutput(f,7) tempoutput(f,8)];
fields2=[fields2;fields];
end
```

Print File

```
fid3=fopen(fid3,'w');
fprintf(fid3,'Description Life Initial.cost E.Power SPWF CRF Annuity An.Energycost An.Cost
```

```
Presentvalue\n');
for k=1:numofequip
    fprintf(fid3,'%s %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f %9.2f
%9.2f\n',fields2{k,1},fields2{k,2},fields2{k,3},fields2{k,4},fields2{k,5},fields2{k,6},fiel
ds2{k,7},fields2{k,8},fields2{k,9},fields2{k,10});
end
fprintf(fid3,'Net.Present.Value=%9.2f\n',NetPresentValue);
fprintf(fid3,'Net.Annual.Cost=%9.2f\n',NetAnCost);

fclose('all')
```

A.4 Program for PV driven Electric Chiller

Main Program

```
% PV Electric Chiller
clear
clc
```

Defining the Variables

```
%PV variables
Ppeak=12000;        %Peak Power of PV panel (W)
Ist=1000;           %Irradiance at Standard Test Condition (W/m2)
alphaT=(-0.44);     %Power Temperature coefficient of PV module (%/degC)
Tst=25;             %Temperature at Standard Test Condition (degC)
nwire=0.95;         %Wire efficiency
NOCT=47.5;          %Nom.op.cell.temp under 800W/m2,20C Tamb,1m/s wind

% Battery Bank Size
volts=48;           %Voltage of PV system
amph=638;         %Amphour of Battery Bank
Bpv=6;              %Setting Initial Battery energy to start simulation
Bpvmax=volts*amph;  %Maximum energy stored in Battery Bank Wh

% Chiller Variables
Qrrated=7.1;             %Nominal colling power of chiller
Eprated=3.1;             %Nominal Electrical power of Chiller
COP=2.29;           %Nominal COP of Chiller

% Inverter Variables
coffa=97.644;           %inverter efficiency curver coefficients
coffb=(-1.995);
coffc=(-0.445);

% Other Variables
datamatrix=[];
power=[];
output=[];
ON=[];
reqdbatteryenergy=0;
```

Reading Data file

```
%file 1: Stationdata and tiltradiation
fid1=fopen('tiltradtempRHdate.txt','rt'); % open test datat file
ncount=1;
while ncount<=2
linestring=fgetl(fid1); % Jumping to the no. of Data points
ncount=ncount+1;
end
linedata=strread(linestring);
datapnt=linedata(1); % Julian date and no. of Data points
numofdays=linedata(2);
ncount2=1;
```

```
linestring=fgetl(fid1);     %Remove Heading
for ncount2=1:(datapnt*numofdays)
    linestring=fgetl(fid1);
    linedata=strread(linestring);
    hrminsec(1:3)=linedata(1:3);
    cumtiltrad=linedata(5);
    Tamb=linedata(8);
    yrmnthday(1:3)=linedata(10:12);
    datamatrix=[datamatrix;hrminsec cumtiltrad Tamb yrmnthday];
    %datamatrix=[hr min sec tiltrad Tamb yr mnth day]
end
fclose(fid1);
% file 2: Refrigeration heat
fid2=fopen('Systemperfmance.txt','rt');
linestring=fgetl(fid2);
for ncount3=1:(datapnt*numofdays)
    linestring=fgetl(fid2);
    linedata=strread(linestring);
    Qr=linedata(9);
    %for Same Qr as absorption system
    Epc=Qr/COP;
    Ep=Eprated;
    power=[power;Qr Ep Epc];
end
fclose(fid2);
datamatrix=[datamatrix power];
%datamatrix=[1hr 2min 3sec 4tiltrad 5Tamb 6yr 7mnth 8day 9Qr 10Ep 11Epc]

power;
```

Computations

```
for i=1:(datapnt*numofdays)
    Epv=datamatrix(1,10);
    Tc=datamatrix(i,5)+(((NOCT-20)/800)*datamatrix(i,4));
    Ppv=(Ppeak*(datamatrix(i,4)/Ist)-alphaT*(Tc-Tst))*(nwire/1000);
    ninv=coffa+(coffb*Ppv)+(coffc/Ppv);
    Ppv=Ppv*(ninv/100);
    % Operating the Electric chiller from Battery and Charging
    % Simultaneously
    if Bpv>Ep
        ONOFF=1;
    else
        ONOFF=0;
    end
    Bpv=Bpv+Ppv-Ep;
    if Bpv>(Bpvmax/1000)
        Bpv=(Bpvmax/1000);
    elseif Bpv<0
        Bpv=0;
    end

    ON=[ON;ONOFF];
    output=[output;datamatrix(i,6:8) datamatrix(i,1:3) Ppv datamatrix(i,10) Bpv
datamatrix(i,11)];
```

```
end

% Creating Output
output=[output ON];
summation=sum(output);
Onhours=summation(1,11)

% Printing Output
fid3=fopen('electricchillerperf.txt','w');
fprintf(fid3,'Year Month Day Hour Min Sec Ppv chillr.power Bpv Epc ONOFF\n');
for num=1:(datapnt*numofdays)
    fprintf(fid3,'%9.0f %9.0f %9.0f %9.0f %9.0f %9.0f %9.2f %9.2f %9.2f %9.2f
%9.2f\n',output(num,1),output(num,2),output(num,3),output(num,4),output(num,5),output(num,6
),output(num,7),output(num,8),output(num,9),output(num,10),output(num,11));
end
fclose(fid3);
```