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Energy Consumption Evaluation of Air Cooled Chiller With Cold Storage System Powered by Photovoltaic (PV) Modules

By

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Thesis submitted in partial fulfillment of requirements of the degree

Master of Science in Renewable Energy & Sustainability

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The undersigned hereby certify that they have read, examined and recommended to the Deanship of Graduate Studies and Scientific Research at Palestine Polytechnic University and the Faculty of Science at Al-Qdus University the approval of a thesis entitled:

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BY: Zaid Jammal Alnather

ABSTRACT

Renewable energy becomes an appealing technology that used in many applications in our life. Environmentally it reduces the CO_2 emissions and enhance the systems sustainability. This research study the beneficial of using PV-system with thermal storage tank (TST) to power an air cooled chiller, associated with three different scenarios.

The simulation methodology is adopted in this research to study the various scenarios of the combination of the utility, PV-system, thermal storage tank and air cooled chiller. The scenarios are based on the annual simulation building library of the TRNSYS simulation software. The three scenarios investigated in this study include supplying an air cooled chiller using PV-system with the grid, PV-systems with grid and TST and finally fully supplying the system by PV-system and TST .

The first scenario gives a reduction in energy consumption from the grid by 81%, and the CO_2 emissions by 72%, in addition the payback period equal to 9 years with 4,350$ total profit along the project life cycle. The second scenario saves 75.6% of the utility energy consumption and decrease the CO_2 emissions by 68%, moreover the payback period becomes 12.4 years with 3,202$ total profit. The final scenario, chiller is 100% supplied from the extended PV-system size and TST volume, which leads to the best reduction in the amount of CO_2 emissions by 89.5%, furthermore the payback period equal to 12.5 years with 4,206$ total profit.

تقييم استهلاك الطاقة للمبرد الذي يتم تبريده باستخدام الهواء مع نظام لتخزين الطاقة الحرارية على شكل برودة والذي يتم تشغيله باستخدام الواح الطاقة الشمسية

اعداد: زيد جمال الناظر

الملخص:

اصبحت الطاقة المتجدد تكنولوجيا جذابة يمكن استخدامها في العديد من التطبيقات في حياتنا العملية. من ناحية بيئية تساهم الطاقة المتجددة في تقليل انبعاث غاز ثاني اكسيد الكربون بالإضافة الى تحسين استدامة الانظمة المستخدمة معها . هذا البحث يدرس الفائدة من استخدام نظام الالواح الكهروضوئية مضافا له خزان للطاقة الحرارية بهدف تزويد المبرد الذي يتم تبريده عن طريق الهواء بالطاقة الكهربائية وذلك من خلال ثلاثة سيناريوهات مختلفة.

استراتيجية المحاكاة تم اعتمادها في هذا البحث بهدف دراسة السيناريوهات الثلاثة من خلال الربط بين شبكة الكهرباء الرئيسية ، نظام الالواح الكهروضوئية ، خزان الطاقة الحرارية بالإضافة الى المبرد المراد تغطية الحمل الكهربائي له. هذه الاستراتيجية تم تنفيذها باستخدام برنامج المحاكاة TRNSYS. السيناريوهات الثلاثة التي تم استخدامها في هذه الدراسة تتمثل في تغذية المبرد باستخدام نظام الالواح الكهروضوئية والشبكة الرئيسية ، الالواح الكهروضوئية والشبكة الرئيسية بالإضافة الى استخدام خزان للطاقة الحرارية و اخيرا تزويد المبرد بالطاقة الكهربائية بشكل كامل باستخدام الالواح الكهروضوئية وخزان الطاقة الحرارية فقط.

السيناريو الاول يعطي تقليل في استهلاك الطاقة الكهربائية من خلال الشبكة الرئيسية بنسبة 81% و ايضا تقليل في انبعاث غاز ثاني اكسيد الكربون بنسبة 72% ، بالإضافة الى ان فترة الاسترداد لهذا السيناريو تساوي 9 سنوات مع 4,350 دولار كفائدة لاستخدام هذا السيناريو خلال فترة المشروع. السيناريو الثاني يخفض 75.6% من الطاقة المستهلكة من الشبكة بالإضافة الى تقيل انبعاث غاز ثاني اكسيد الكربون بنسبة 68% وفي هذا السيناريو فترة الاسترداد تساوي 12.4 سنة مع 3,202 دولار قيمة الفائدة من استخدام النظام. السيناريو الاخير يعمل على تغطية الطاقة الكهربائية للمبرد بشكل كامل من خلال زيادة عدد الخلايا الكهروضوئية وحجم الخزان الحراري ، وهذا من شانه ان يقودنا الى افضل تقليل في انتاج غاز ثاني اكسيد الكربون بنسبة 89.5% ، بالإضافة الى فترة استرداد تساوي 12.5 سنة مع 4,206 دولار كفائدة لاستخدام هذا السيناريو .

DECLARATION

I declare that the Master Thesis entitled" **Energy Consumption Evaluation of Air Cooled Chiller With Cold Storage System Powered by Photovoltaic (PV) Modules** " is my own original work, and herby certify that unless stated, all work contained within this thesis is my own independent research and has not been submitted for the award of any other degree at any institution, except where due acknowledgement is made in the text.

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DEDICATION

To my Family ………….……. For their support

To my Teachers ……........... For help me until the end

To my friends ………………. Who give me Positive sentiment

To oppressed people throughout the world and their struggle for social justice and egalitarianism

To our great Palestine

To my supervisor Dr Ishaq Sider

To all who made this work is possible

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Finally, my ultimate thanks go to the great edifice of science (Palestine Polytechnic University).

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LIST OF ABBREVIATIONS

AC	Alternating Current
C.T	Cooling Time
COP	Coefficient of performance
CRF	The Constant Rate Factor
CTES	Cooling thermal energy storage
DC	Direct Current
DSC	Differential Scanning Calorimeter
EGWS	Ethylene Glycol Based Water Solutions Tank
EPS	Environmental Process Systems Limited
E_{Annual}	Total electrical energy obtained using each scenario.
h	hour
i	Inside
ITS	Ice thermal storage
MPPT	Maximum Power Point Tracking
NIS	New Israeli shekel
No	Number
NOCT	Normal Operating Conditions Test
o	outside
O&M	Operation & Maintenance Cost
PBP	Payback Period
PCM	Phase change material
PV	Photovoltaic
RM	Malaysian Ringgit
SPV	Solar photovoltaic
STC	Standard Test Condition
TES	Thermal energy storage
TFM	Transfer Function Method
TR	Refrigeration Ton
TRANSYS	Transient System Simulation program
TRNBuild	TRANSYS Building input data visual interface
TST	Thermal Storage Tank
WHO	World Health Organization
Wp	Watt Peak

LIST OF SYMBOLS

Variables	Units	Description
A	m^2	Surface area
a	1/h	Times of air change
Cp	kJ/kgK	Specific heat
E_{annual}	Wh	Annual electrical energy
h	$W/m^{2o}C$	Conviction heat transfer coefficient
h_{fg}	kJ/kg	Air enthalpy
i	----	Loan intrest
k	W/m^oC	Thermal conductivity
m	kg	Mass
m^*	kg/sec	Mass flow rate
n	Years	Project life cycle
η_{exe}	----	Exergetic efficiency
P_{elec}	W	Electrical Power
P_{peak}	W	Peak Electrical Power
Q	W	Thermal cooling load
q_c	kJ/h	Conviction heat flux
Q_{inf}	kW	Infiltration heat gain
q_{it}	kJ/h	Infiltration heat flux
q_{prod}	kJ	Product cooling load
Q_{prod}	kW	Product gain
q_s	kJ/h	Conduction heat flux
Q_{total}	W	Thermal cooling load for chamber
T	^oC	Temperature
T_r	$	Tariff price
U	$W/m^{2o}C$	Overall heat transfer coefficient
V	m^3	Volume
V_f^*	m^3/sec	Volumetric flow rate
ω	Kg/kg dry air	Air humidity ratio
ρ	kg/m^3	Air density
ΔT	^oC	Temperature difference
Δx	m	Layers thickness

CHAPTER 1
INTRODUCTION

1.1. Background

As the innovation develops and the cost of fossil fuel assets develops quickly, the expanding concentrate on renewable energy assets is watched [1]. Due to the growing concern and awareness of environmental issues among the scientific community, so increasing energy demand with associated increase in CO_2 emissions is expected in the near future and hence it is necessary to reduce the energy demand and CO_2 emission by improving energy efficiency and utilizing energy waste. Using power generation from renewable energy sources, particularly solar energy, become significantly important for the last few decades.

One of the principal worries of usage of refrigeration system , which have wildly been utilized in residential and industrial sectors, is electrical energy utilization for gas compression which not only is cost effective but also leads to more greenhouse gases, absence of fossil fuels in the near feature, day by day ascend in fuel costs and ecological difficulties, which are the main motivators to find ways for using energy more effectively especially in residential sectors[2].

Solar photovoltaic power for refrigerators has great potential for lower running costs, greater reliability and a longer working life than kerosene refrigerators or diesel generators, which have been generally used in remote areas. Over the past five years, at least 3000 photovoltaic medical refrigerators have been installed [3].

Thermal energy storage (TES) systems could play a remarkable role in energy saving via shifting it from on-peak load (day-time) to off-peak load (night-time) for cooling by the TES system, which is one of best methods for power management and economic advantages. Dincer [4] introduced various information and useful examples for cooling thermal energy storage (CTES) and analyzed them from energy and exergy aspects and mentioned their environmental and economic advantages.

1.2. Statement of The Problem

Nowadays, the rising level of CO_2 , global warming , ozone layer depletion and the cost of fossil fuel, appeals the attention to renewable energy resources, including solar energy, wind energy and thermal storage system. In this study, a blended system was build which comprises the usage of on-grid PV system and thermal cold storage tank by using (EGWS). Thus, during the on peak radiation hours the compressor will be fed via the PV system, and using (EGWS) tank as cooling thermal energy storage system the surplus energy will be used during the off peak radiation hours.

1.3. Study Objectives

a. Energy consumption evaluation for the used three scenarios in this study, which are on-grid PV system, PV powered air cooled chiller with (TST), and full load coverage using PV system with TST.

b. Economical and Ecological study for the air cooled chiller based on vapor compression refrigeration system for each three scenarios .

1.4. Methodology and Thesis Scope

1- Obtaining the thermal cooling load for one cubic meter refrigeration chamber using annual building simulation in TRNSYS software .

2- Developing a simulation model using TRANSYS software for an air cooled chiller in order to compensate the needed thermal cooling load for the refrigeration chamber.

3- Developing a COP file for the selected chiller in order to achieve the needed electrical power that used to run the chiller .

4- Calculating the annual electrical power that needed to run the chiller.

5- Designing the PV system using annual electrical power method, selecting the stabile PV module in order to cover this power and simulate this system using TRNSYS software.

6- Calculating the extracted thermal power from the PV system in the first scenario and determine the cooling load compensation percentage for the refrigeration chamber.

7- Designing an (EGWS) storage tank in order to shift the thermal power (Cooling Load) from on-peak hours to off-peak hours and simulate this system using TRANSYS software.

8- Calculating the extracted thermal power from the PV system with storage tank in the second scenario and determine the cooling load compensation percentage for the refrigeration chamber.
9- Designing the new PV system and storage tank for the third scenario and simulate this system using TRNSYS software.
10- Calculating the extracted thermal power from the PV system with storage tank in the third scenario and determine the cooling load compensation percentage for the refrigeration chamber.
11- Calculating the payback period and total profit for the study scenarios.
12- Calculating the amount of CO_2 emissions that saved using the three scenarios.

1.5. Thesis Structure

This research divided in to eight chapters. The previous study that deals with refrigeration system driven by solar energy with thermal storage tank are given in chapter 2. The description of the refrigeration chamber, its walls construction, load sources, and the thermal cooling load are given in Chapter 3. Chapter 4 includes a general description of TRNSYS software simulation environment and chamber cooling load simulation using Type56 and TRNBuild library. In Chapter 5, chiller working principle and simulation using TRNSYS have been discussed. Chapter 6 includes three coverage scenarios for the air cooled chiller load which are on-grid PV-system, PV-system with thermal storage tank and fully coverage load using PV-system and thermal storage tank, this chapter discusses the molding and results for the three scenarios. The economical and environmental assessments are given in chapter 7. Chapter 8 includes the conclusions and future work for this study.

CHAPTER 2
LITERATURE REVIEW

2.1. Introduction

The sun is the source of nearly all our energy (with the exception of radioactive sources and the tides) and will continue to be the most important nuclear fusion and fission reactors to be used [5]. This chapter divided in two sections. The first section deals with studies on refrigeration system driven by PV-system and the second one discusses studies on refrigeration system with thermal storage tank.

2.2. Refrigeration System Driven By PV-system

Many researchers developed studies and experiments on vapor compression refrigeration system driven by photovoltaic cells, **Deshmukh1 et al.**[6]**,** developed a Performance Evaluation of Photovoltaic System Designed for DC Refrigerator. Performance of photovoltaic system at no load and full load condition were carried out to assess its technical viability.

This study indicated the necessity and usefulness of energetic and exergetic techniques to evaluate the performance of the solar photovoltaic (SPV) refrigerator. The average photovoltaic conversion efficiency and exergy efficiency found nearer to 8.5% and 11% respectively in both no load and full load condition for May month. This indicates that the full load condition does not affect the PV system as shown in figure 2.1 . It was observed that the PV module temperature had a great effect on the exergy efficiency, could be improved by maintaining module temperature close to ambient and that could be achieved by removing the heat from PV module surface .It was concluded that the exergy losses increased with increasing module temperature [6].

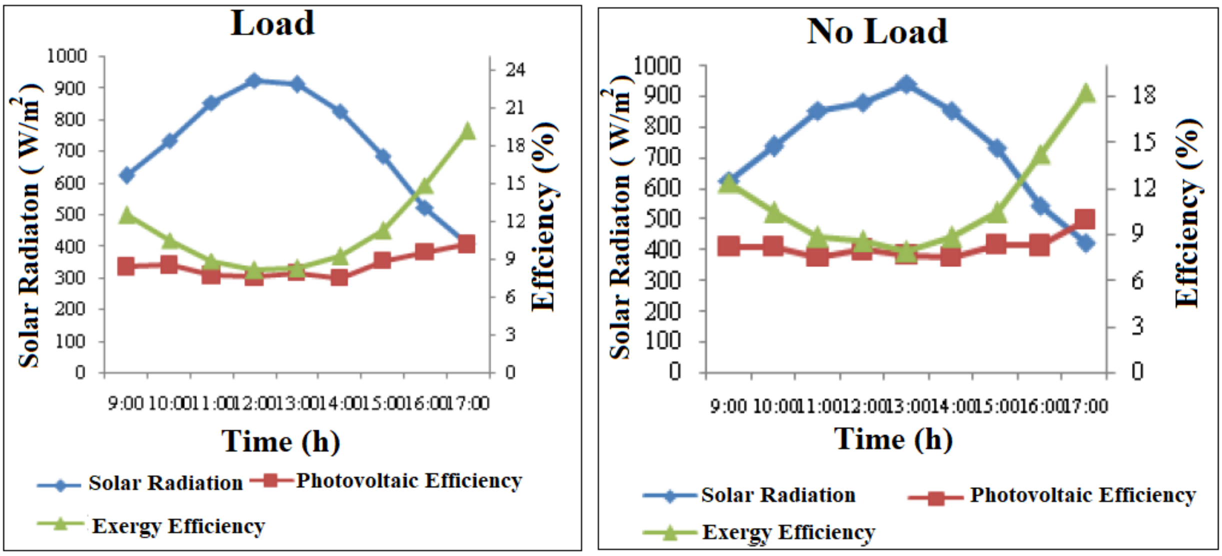

Fig.2.1. Variation of photovoltaic efficiency, exergy efficiency and solar radiation with time [6].

Kalbande et al.[7]**,** a photovoltaic system for DC refrigerator was designed and developed in meeting the needs of most rural areas which have no access to national grid or with unstable and erratic supplies of electricity. The solar photovoltaic operated DC vapor compression refrigeration system under test was able to maintain the temperature as specified by the World Health Organization (WHO) for the vaccine preservation (2-8°C). The average photovoltaic conversion efficiency and exergy efficiency of refrigerator found nearer to about 12.05% and 14.20% on full load condition in November 2015.

Solar radiation was less, the power output was also low, it was recorded maximum 60.57 W at 12:30 and it varied from 5.71 – 60.57 W and recorded minimum 5.71 W at 17:00, that shown in figure 2. 2 [7] .

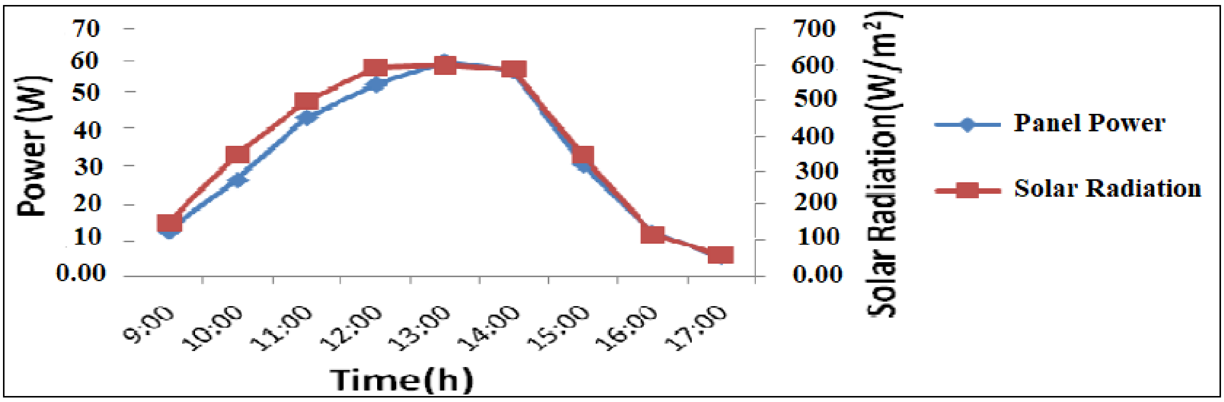

Fig.2.2. Variations in Power Output and Solar Radiation with Time [7].

Fatehmulla et al. [8], conducted that the performance of the refrigeration system with PV module is significantly good. Cost comparison between the PV (Photovoltaic energy or solar energy) and Conventional energy (electrical energy) demonstrates the economic effectiveness of the energy efficient low power PV refrigeration system which green, clean and safe, in view of the calculations and the initial cost of our PV system including the initial electrical installation cost to run the low power refrigeration system, a plot showing the comparison of the energy cost incurred with PV and conventional systems and years of operation has been drawn in figure 2.3.

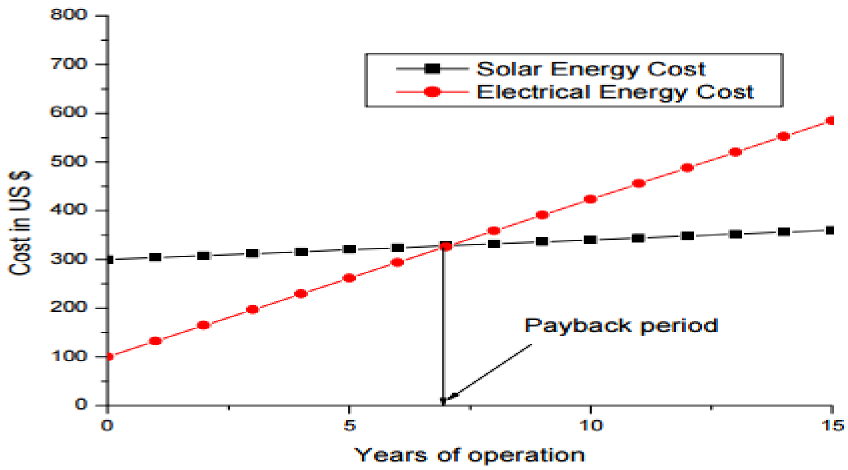

Fig.2.3. Cost comparison between the PV and Conventional energy in operating a low power refrigeration system [8].

2.3. Refrigeration System With Thermal Storage Tank

Thermal storage tank is an appealing technology that used to transfer the thermal power in the periods that we don't need this power to another periods that we need it**, Rismanchi et al.** [9], indicates the economical analysis of the cost benefits is carried out for a system including chiller and storage system.; hence the study was conducted for a range of 100–2000 tons of refrigeration (TR) (352–7034 kW) for two storage strategy of full storage and load leveling storage strategy as shown in figure 2.4 .

The installation costs are mainly dominated by the total system capacity. Generally, in the Malaysian market, if the total cooling capacities are 1000 TR (3514 kW) and above, a rule of thumb of $370/kW (RM 3500/TR) can be adopted, which includes the supply and installation of the chillers, pumps, cooling towers, piping and valve fittings, electrical and control system cost and chemical treatment. For system capacities of less than 1000 TR (3514 kW) the installation costs vary from $470 to $670/kW. The same rule of thumb may also be applied for the installation cost of the ice thermal storage (ITS) systems. If the total cooling capacities are 1000 TR (3514 kW) and above, the installation cost of $700/kW (RM6000/TR) can be adopted, which includes the supply and installation of chillers, pumps, cooling towers, ice tanks, piping and valve fittings, electrical and control system cost, glycol and chemical treatment. As for cooling capacities less than 1000 TR (3514 kW), the price varies from $700 to $1100/kW [9].

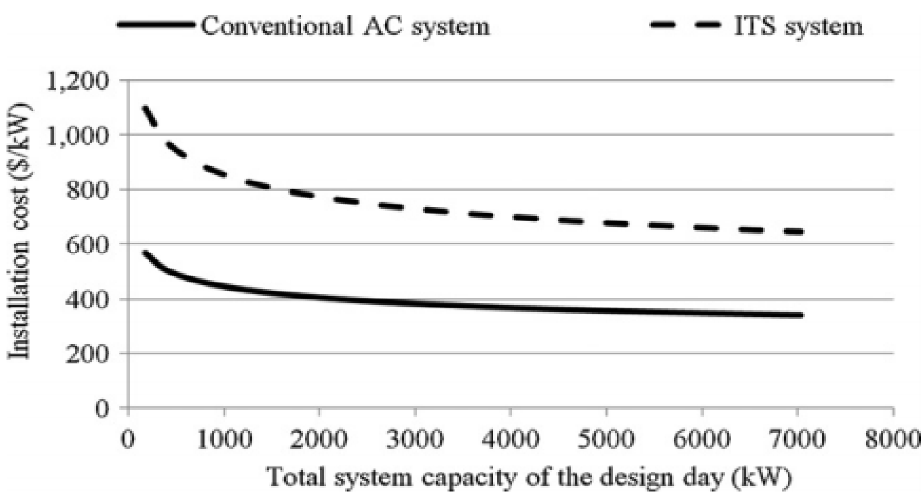

Fig.2.4. Installation cost trend for conventional AC system and ITS system [9].

Also, The results indicate that considering the special off-peak tariff rate of $0.06/kWh for the total system capacities of 500 and 1500 TR (1758 kW and 5275 kW), the annual cost saving varies from $230,000 to $700,000 and from $65,000 to $190,000 for full storage and load leveling storage strategy, respectively , as shown in figure 2.5 , the overall results show that the full storage strategy can reduce the annual costs of the air conditioning system by up to 35% while this reduction is limited to around 8% for a load leveling strategy. [9] .

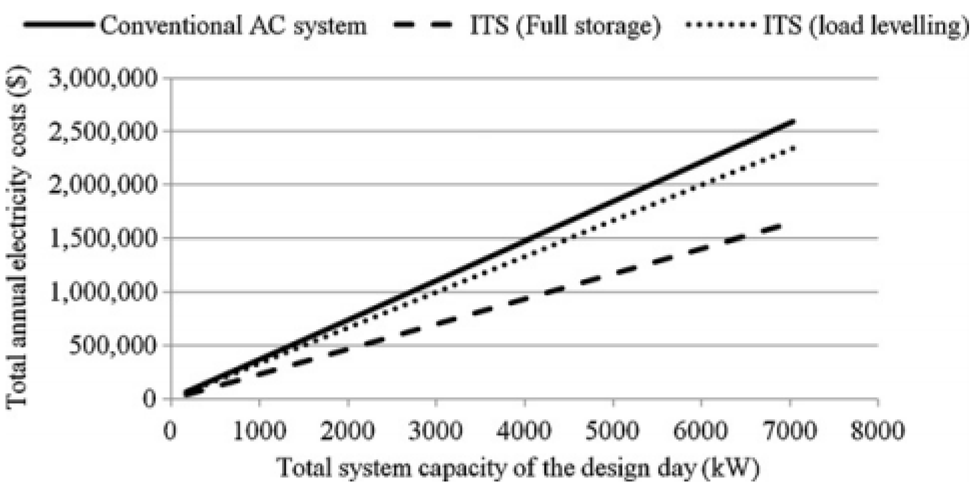

Fig.2.5. Total annual electricity costs for conventional, full load (ITS) and load leveling (ITS) systems [9] .

Figure 2.6 shows the comparison study in payback period, reveals that for the full storage strategy the payback period varies between 3 and 6 years while the payback period for the load leveling strategy varies between 1 and 3 years. It was concluded that the ITS system can play a vital role in consuming the natural resources in a more efficient [9].

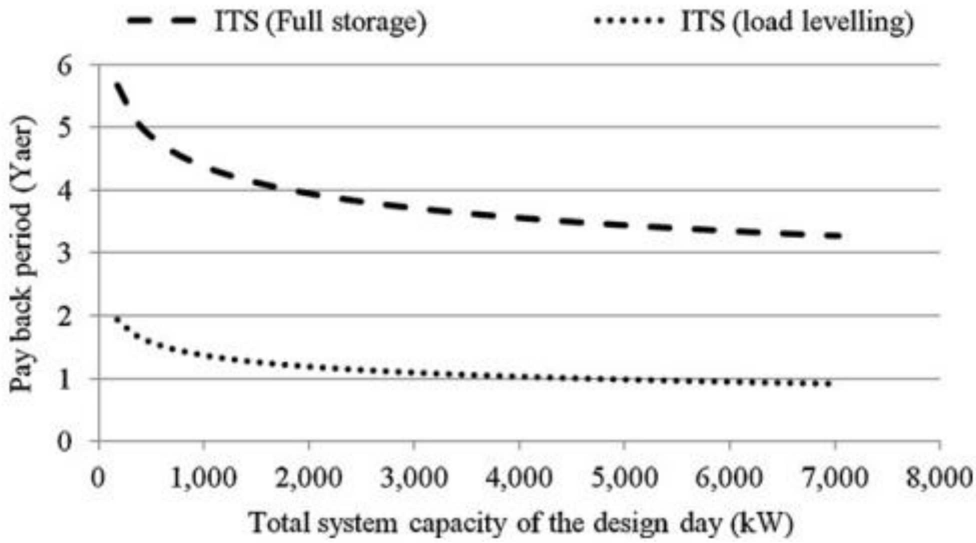

Fig.2.6. Payback period for full load and load leveling ITS systems[9].

On the other hand, by having the annual energy saving, the emission reduction potential of utilizing ITS system was estimated based on the potential of the natural gas to produce CO_2 emission. The results show that the annual CO_2 emission reduction for load leveling strategy varies from 3000 to 60,000 ton for the total system capacities of 352 and 7034 kW [9].

Hoseini Rahdar et al.[10], a vapor compression A/C system has been analyzed via two strategies of hybrid systems. First, an ice thermal energy storage (ITES) system is used in the a.m. hybrid system; and thereafter a phase change material (PCM) - PCM popireties shown in table 2.1 tank is used as a full storage system in order to shift the load from on-peak to off-peak mode. This A/C system is modeled and analyzed from exergetic, economic and environmental point of views for both cases.

Table2.1: Thermo-physical properties of used PCM (RT3HC) [11]

Melting point (°C)	3
Density (liquid) (kg m^{-3})	770
Density (solid) (kg m^{-3})	880
Specific heat (kJ kg^{-1} K^{-1})	2
Melting latent heat (kJ kg^{-1})	200
Toxic	No

The results have illustrated that the payback period of PCM system compared to the conventional system was estimated to be 5.56 years, while for the ITES system, it was 3.16 years, the power consumption of ITES and PCM systems are 4.59% and 7.58% lower than the conventional system respectively as shown in figure 2.7 . Moreover, CO_2 emission production for ITES and PCM systems are 17.8% and 27.2% lower than conventional system respectively [10] .

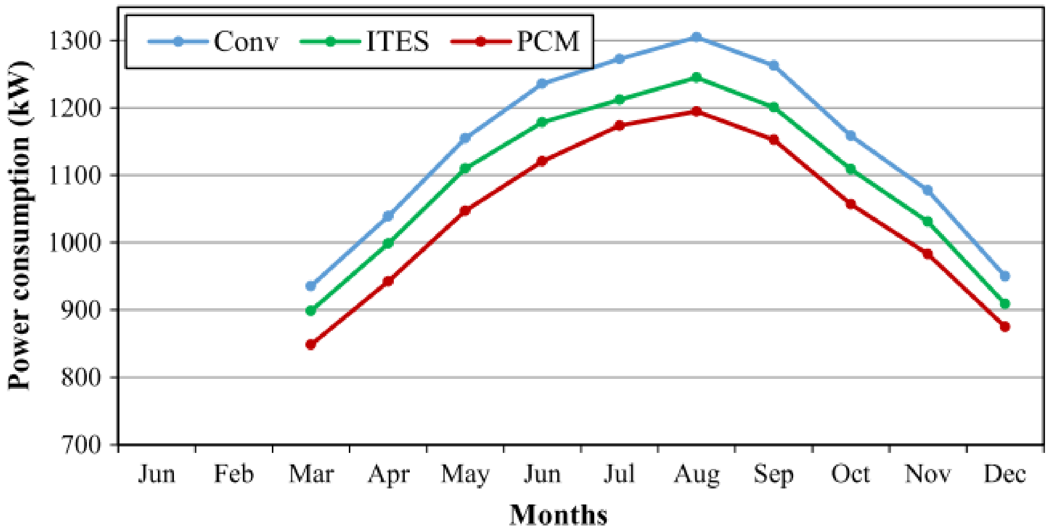

Fig.2.7. Power consumption of conventional, ITES and PCM systems [10].

Eduard Oró1 et al.[12]**,** This paper provides an overview of the existing Spanish and European potential energy savings and CO_2 mitigation by incorporating TES systems to cold storage and transportation systems. The total energy demand for cold applications in Spain and in Europe was calculated, and after that the energy reduction and therefore CO_2 emissions mitigation was determined assuming a full implementation of the phase change materials (PCM) TES systems. The industry sector shows the highest potential of all the sectors analysed. Related to economical savings, Spain could save between 2,309 and 11,674 GWh/year , and yearly CO_2 emissions may get to be cut down from 1195 to 5,902 [1000 tCO_2/year] as shown in figure 2.8 .

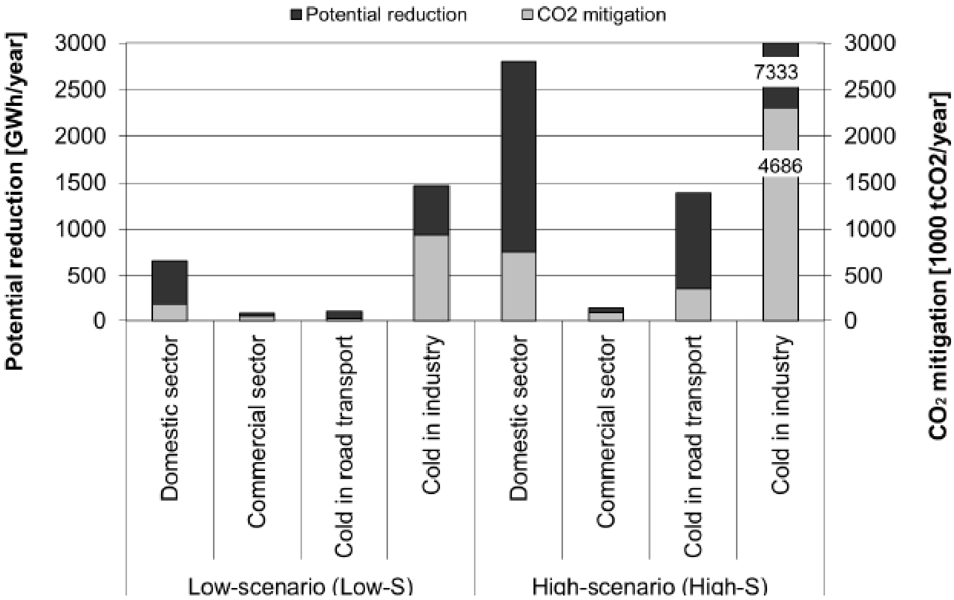

Fig.2.8. Potential reduction and CO_2 mitigation in different sectors by using TES systems in Spain [12]

Furthermore Europe save between 27,405 and 136,440 GWh/year , and acoodring to Europe CO_2 emissions mitigation, those values become 15,578 and 75,371 [1000 tCO_2/year] as shown in figure 2.9, depending on the scenario evaluated [12] .

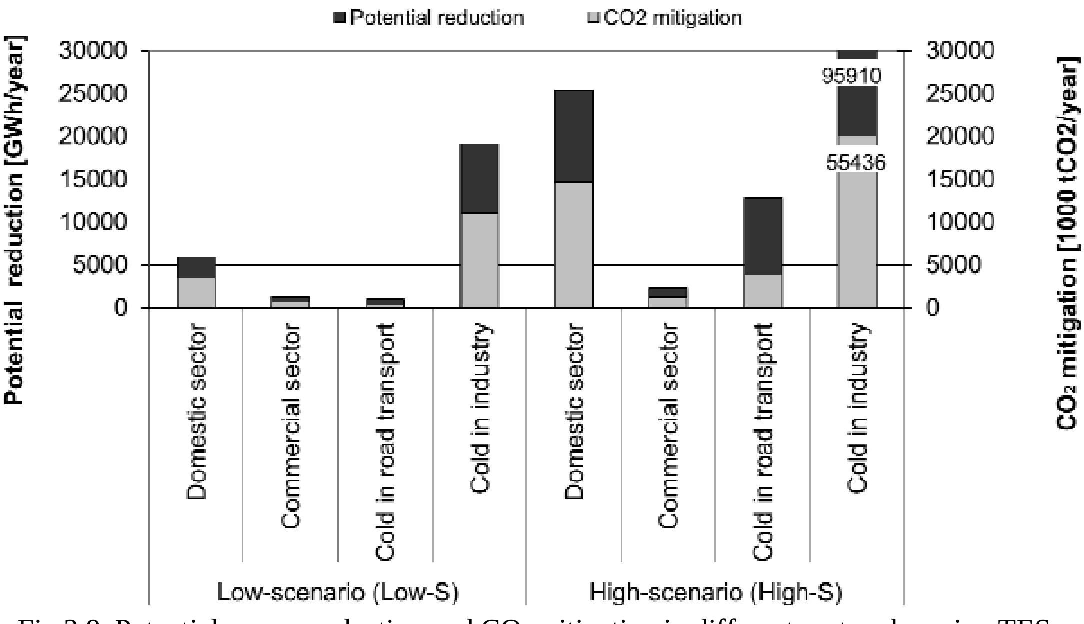

Fig.2.9. Potential energy reduction and CO_2 mitigation in different sectors by using TES systems in Europe [12] .

Liu et al. [13], An innovative refrigeration system incorporating phase change material (PCM) is proposed to maintain refrigerated trucks at the desired thermal conditions. In addition, the system consumes less energy and produces much lower local greenhouse gas (GHG) emissions. As the temperature of the refrigerated space needs to be maintained at -18 ^{o}C, the proposed PCM needs to have a melting temperature below -18 ^{o}C. To minimize the required heat transfer area when cooling the space, the melting point should be as low as possible. However, considering that the PCM needs to be charged, the melting point should not be too low; otherwise the refrigeration unit will be more expensive and operates at a lower efficiency.
There are several available commercial products, such as SN.29 and SN.33 supplied by Cristopia Energy [14] and E-29, E-32 and E-34 supplied by Environmental Process Systems Limited (EPS) [15] .
Given that there was no suitable PCM commercially available that was both reliable and low cost, so a new PCM was developed (inorganic salt-water solution). PCM samples were tested by a Differential Scanning Calorimeter (DSC) (PerkinElmer DSC 8000) and the DSC curve is present in Figure 2.10 . The measured melting point is -26.8 ^{o}C and the latent heat of fusion is 154.4 kJ kg^{-1}. The cost of this new PCM is less than one-fifth of that available from Cristopia and comparable to that manufactured by EPS [13].

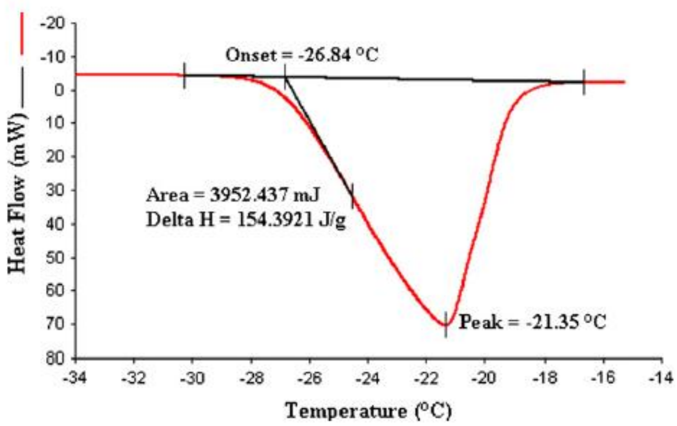

Fig.2.10. DSC curve for the developed PCM [13].

The objective of this research is to merge between the studies that consider supplying the air cooled chiller from the PV-system as mentioned in [6-8] and the results of researches that regard using the thermal storage tank with the air cooled chiller as listed in [9-13]. The combination between the two approach is evaluated for three different scenarios for an air cooled chiller, to study the energy consumption, economical feasibility and environmental aspects.

CHAPTER 3
COOLING LOAD DEMAND

3.1. Introduction

The total heat required to be removed from refrigerated space in order to bring it at the desired temperature and maintain it by the refrigeration equipment is known as cooling load. The purpose of a load estimation is to determine the size of the refrigeration equipment that is required to maintain inside design conditions during periods of maximum outside temperatures. The design load is based on inside and outside design conditions and its refrigeration equipment capacity to produce and satisfactory inside conditions.

3.2. Reference Location Climates

The chamber cooling load demands are influenced by the outdoor ambient air temperature . In this study, the cooling load calculations depends on the selected locations climate for Hebron city in Palestine.

Hebron city is located south of Palestine at latitude 31.31^{o} North and longitude 35.8° East. The climate of Palestine may be divided into two main categories which are west bank climate and Gaza climate. West bank climate divided into five zones, Hebron city considered as a part of zone four which is **Warm sub-humid summer, cold winter – Mediterranean climate**, this zone is about 1314.6 km^2 in area with population of 876971 persons, which represents approximately 47% of the West Bank population. Zone four has mean annual temperature of 16 ^{o}C, mean annual average of relative humidity of 60% and 715 mm of maximum mean annual rainfall [17] .

3.2.1. Meteorological Data for Reference Location

The meteorological data file of Hebron city received in Energy Plus Weather (EPW) format from Palestinian Meteorological Department contains measurement data in one hour intervals for the year 2015. It is includes, the horizontal solar radiation (beam, diffuse and global), the total tilted surface radiation (tilt angle 30^{o}), ambient air temperature, relative humidity, wind speed and etc … [18].

Figure 3.1 and figure 3.2 represent the total tilted surface radiation and ambient air temperature for each Hebron city.

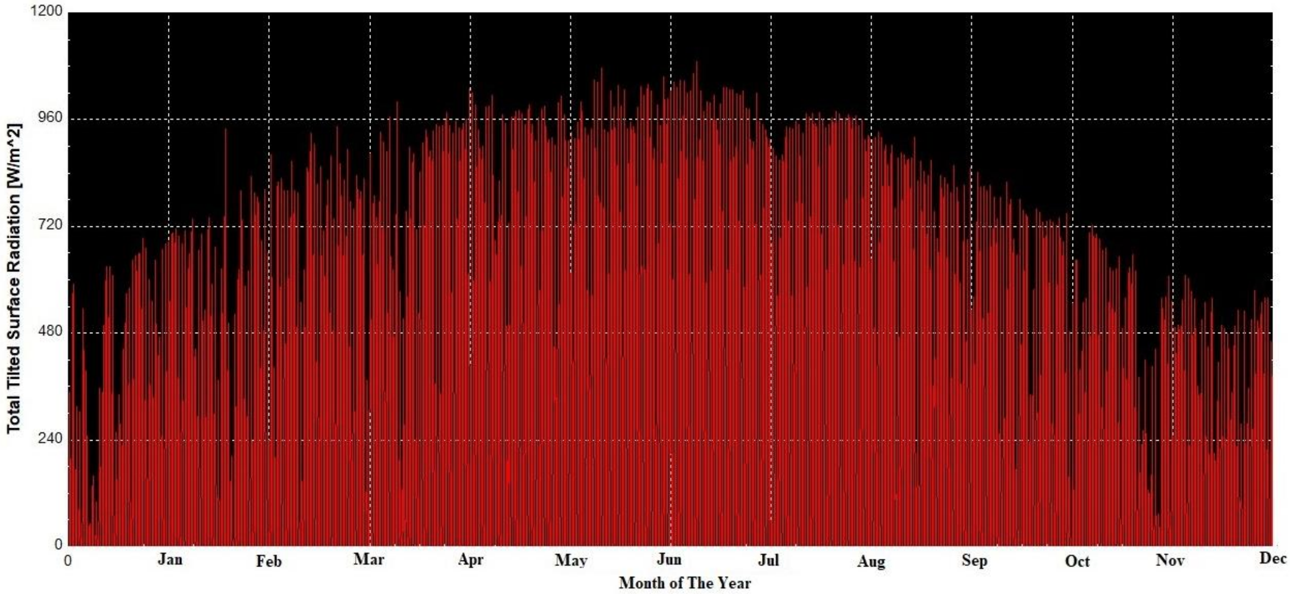

Fig.3.1. Annual distribuation of total tilted surface radiation for Hebron city [18].

Figure 3.1. illustrates, distribution of the total tilted solar radiation for Hebron city, the maximum radiation in summer season reaches near to 1090 W/m^2 and in winter 600 W/m^2.

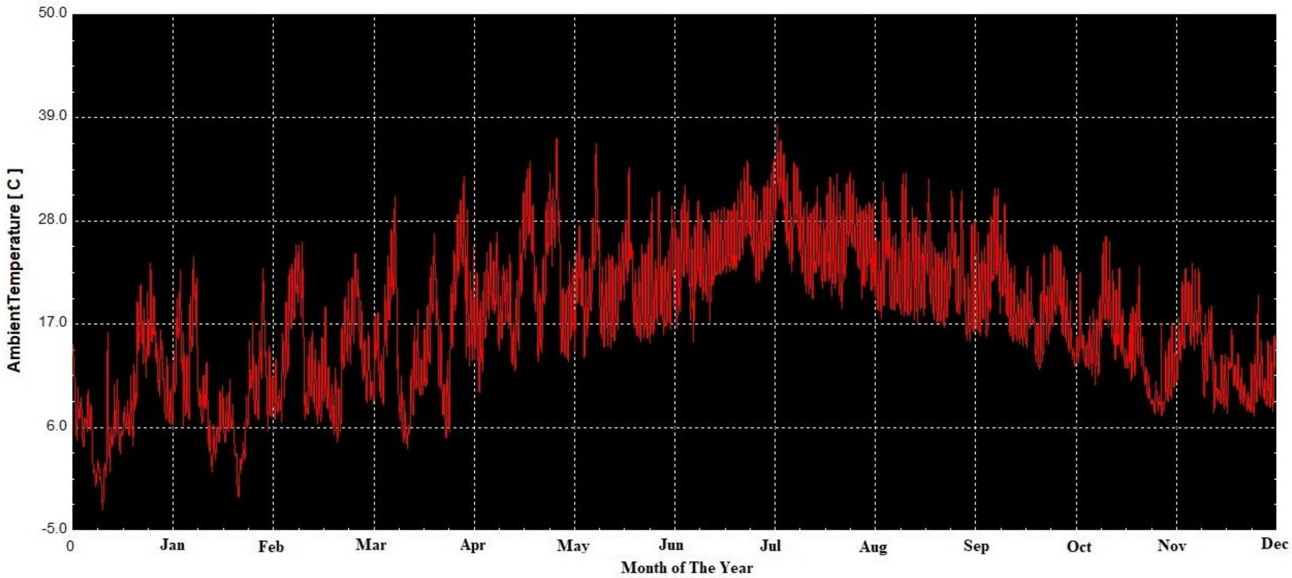

Fig.3.2. Annual distribution of ambient temperatures for Hebron city [18].

Figure 3.2 shows, the annual distribution of ambient temperature in Hebron city. in summer season the daily maximum peak temperature reaches to 39 ^{o}C, in winter it change between -3 ^{o}C to 23 ^{o}C, so the cooling demand during the summer season more than the demand during the winter .

3.3. Load sources

The cooling load seldom results from any one single source of heat. Rather , it is the summation of the heat which usually evolves from several different sources. some of the more common sources of heat that impose the load on refrigerating equipment are the wall heat gain, the product heat gain, infiltration heat gain, packing heat gain, defrosts heater heat gain and fan motor heat gain [16].

3.3.1. Chamber Overview

- Storage temperature is 5 ℃ .
- Surrounding temperature is varied according to whether file for Hebron city.
- Mass of the product (Water bottle) $= 100\ kg$
- Volume of the product (Water bottle) = 100 Liter
- Water bottle dimensions (D = 10 cm , h = 36 cm)
- Chamber dimensions (1 x 1 x 1) meter
- Chamber volume = 1 m^3 = 1000 Liter .

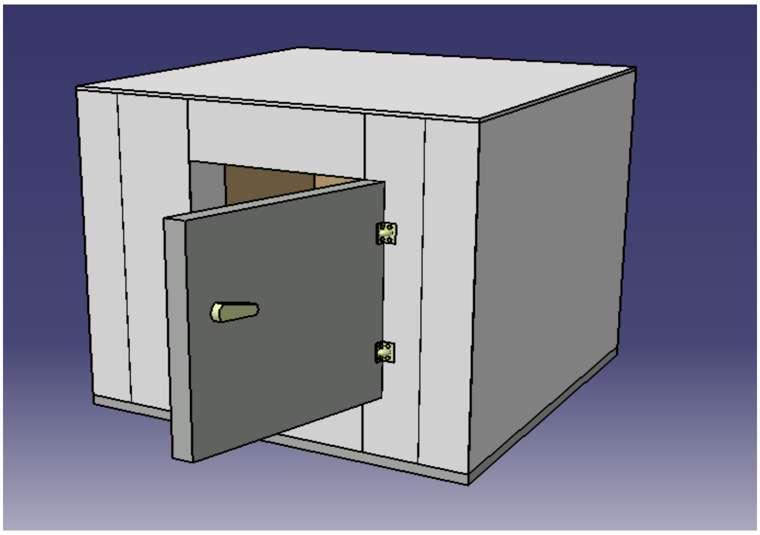

Fig.3.3. Chamber design.

In this study many heat sources forms the cooling load demand of the system which are wall heat gain, product heat gain, infiltration heat gain and etc… , the product heat gain represents the largest value of the thermal cooling load .

3.3.2. The Wall Heat Gain

The wall heat gain load , sometimes called the leakage load, is a measure of the heat flow rate by conduction through the walls of the refrigerated space from the outside to the inside. Since there is no perfect insulation, there is always a certain amount of heat passing from the outside to the inside whenever the inside temperature is below than the outside. The wall gain load is common to all refrigeration application and is ordinary a considerable part of the total cooling load [16].

$$Q_{wall} = U * A * \Delta T \qquad \text{.......3.1}$$

Where :-

A : Outside Surface Area of The Wall [m^2].

U : the overall heat transfer coefficient [$W/m^2\ °C$].

ΔT: the temperature differences across the walls [°C].

Overall heat transfer coefficient is computed by the following :

$$U = \frac{1}{\frac{1}{h_i}+\frac{\Delta x1}{k_1}+\frac{\Delta x2}{k_2}+\cdots+\frac{1}{h_o}} \qquad \text{.......3.2}$$

Where :

U : the overall heat transfer coefficient [$W/m^2\ °C$].

Δx : the thickness of the layer of the wall [m] .

k : the thermal conductivity of the material [$W/m\ °C$] .

h_i: the convection heat transfer coefficient of inside air [$W/m^2\ °C$].

h_i: Forced convection inside the chamber by using fan (20 – 50), taken 25[$W/m^2\ °C$] [19].

h_o : the convection heat transfer coefficient of outside air [$W/m^2\ °C$].

h_o: Free convection outside the chamber (5- 20) , taken 5 [$W/m^2\ °C$] [19].

All walls are constructed of a three layers as shown in Figure 3.4.

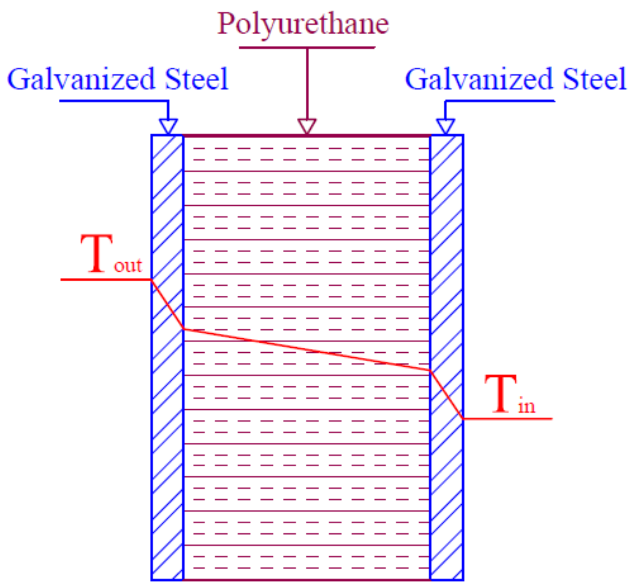

Figure 3.4. Chamber wall layers.

The various material of the chamber envelope layers, the thicknesses, density and specific heat listed in Table 3.1 [20].

Table 3.1: Construction components of the chamber layers

Layer	Thickness [cm]	Thermal Conductivity [$W/_{m.°C}$]	Specific Heat [kJ/kgK]	Density [kg/m^3]
Galvanized steel	0.1	50	0.510	8050
Polyurethane	5	0.03	1	40
Galvanized steel	0.1	50	0.510	8050

The overall heat transfer coefficient (U) for the chamber walls calculated by equation 3.2, equals to 0.554 [$W/_{m^2\,°C}$].

3.3.3. The Product Heat Gain

The heat emitted from the product to be stored is very important in case of cold storages. The product cooling load depends upon the mass of the product, specific heat of the products, entering product temperature, final product temperature desired and the cooling time. This heat gain can be calculated by the following equation [21]:

$$q_{Prod.} = m^* . c_p . \Delta T \quad \text{.......} 3.3$$

Where :

$q_{Prod.}$: Cooling product load in [kJ].

m : Mass of the product in [kg]. (100 kg of water used in this study)

C_p : Specific heat of the product in [kJ / kg . ℃]. (Cp for water equal to 4.18 kJ/kgK)

ΔT : Temperature deference for the product [^{o}C].

The total product cooling load can be calculated from equation 3.4.

$Q_p = (q_{Prod}) / $ C.T ……………………………………………………………………3.4

Where :

C.T : Cooling time [Seconds].

The water use in this study taken from a real case in central supermarket, in this case the consumption of water every six hours equal to 100 kg of water, so that each 100 kg water stay four hours in the refrigeration chamber. According to this case the cooling time equal to four hours (14400 seconds) .

3.3.4. Infiltration Heat Gain

In the practical operation of refrigerated facility, doors must be opened at many times in order to move the product in and out of the chamber. The infiltration load is one of the major loads in the refrigerator. The infiltration air is the air that enters a refrigerated space through cracks and opening of doors . This is caused by temperature difference between the inside and outside air [19].

Q_{inf} = m . C_p . (T_o - T_i) ………………………………………………………………..…3.5

$Q_{inf} = \rho * V_f^* * C_p * (T_o - T_i)$……………………………………..…………………3.6

Where :

ρ : Air density [1.25 kg / m^3] [20].

C_p : Specific heat of the air [1000 J / kg . ℃] [20].

V_f^* : Volumetric flow rate of infiltrated air [m^3/s].

T_o : Outside temperature [℃].

T_i : Inside temperature [℃].

V_f^* = number of air change * volume of room ……………………………………………3.7

Number of air change = 0.6 [times / h] [19]

Volume of room = 1 x 1 x 1 = 1 m^3

$V_f^* =$ 0.294 * 0.6 = 0.1764 m^3/ hr

Other cooling load sources like fan motor, door lamp, and etc… considered as a very small loads, so it covered by using factor of safety 15% added to the total cooling load. Whatever the total annual cooling loads discussed in this chapter modeled and simulate using TRNSYS simulation software as described in the next chapter.

CHAPTER 4
CHAMBER ANNUAL THERMAL COOLING LOAD SIMULATION

4.1. Introduction

The main objective for this chapter is to calculate the annual thermal cooling loads for the refrigeration chamber. That is in order to investigate the main objective of this study, which is the coverage of the annual thermal cooling load for the chamber using PV modules and thermal storage tank. This chapter has two sections. The first section discusses and describes the annual thermal cooling loads simulation by using TRANSYS Software. The second section includes the thermal cooling load simulation results and analysis of the results.

4.2. TRNSYS Software Simulation Environments

TRANSYS is a transient system simulation program. The software includes a large library of built-in components, often validated by experimental data [22-24] . Components (or types) are individual engineering systems such as a boiler, thermal storage tank, PV panels, a pipe and etc… that are defined by a discrete set of inputs, outputs, parameters, and the mathematical functions which govern their operation. It is a complete and extensible simulation environment for the transient simulation of systems.

The program allows the users to create and design complex energy engineering systems by adding and dropping components from the software library to a simulation map and connects this components inputs and outputs together [24]. This makes TRANSYS a very capable tool to simulate the cooling load demand for refrigeration chamber. TRNSYS consists of suitable programs. In this study, only two of these programs have been deployed which are TRNSYS simulation studio and Multi-zone building (TRNBuild) [25].

4.3. Description of Type 56 Components

TYPE 56 (Multi-zone building model) in TRNSYS is selected to simulate the heat conduction and convection through surfaces of the chamber envelop. In order to use this type in two separate processing program. The first process, TRNBuild program reads in and processes a file containing the chamber description and generate two files described in section 4.3.2. The second process occurred in the TRNStudio program, the two generated files will be used by the Type 56 component during a TRNSYS simulation.

4.3.1. Type 56 Mathematical Description

The TRNSYS mathematical model calculations are affected by the outdoor climatic conditions, the indoor design conditions and the chamber envelop structure. The heat balance method is used by TRNSYS as a base for all calculations. For conductive heat gain at the surface on each wall, TRNSYS use Transfer Function Method (TFM) as a simplification of the arduous heat balance method [26,27]:

$$\dot{q}_{s,i} = \sum_{k=0}^{nb_i} b_s^k T_{s,o}^k - \sum_{k=0}^{n_{cs}} C_s^k T_{s,i}^k - \sum_{k=1}^{nd_s} d_s^k q_{s,i}^k \quad \text{......4.1}$$

$$\dot{q}_{s,o} = \sum_{k=0}^{na_s} a_s^k T_{s,o}^k - \sum_{k=0}^{n_{bs}} b_s^k T_{s,i}^k - \sum_{k=1}^{nd_s} d_s^k q_{s,o}^k \quad \text{......4.2}$$

Where :

q_s : Conduction heat flux throw the wall [kJ/h]

a_s, b_s, c_s, and d_s : z-transforms of the surface temperature and heat flux determined by the z-transfer function routines of literature[28].

k : refers to the term in the time series, and it specified by the user within the TRNBUILD description.

The Heat gain by convection is calculated the following equation [26]:

$$q_c = h_{convc,s,o}(T_{as} - T_{s,o}) \quad \text{......4.3}$$

Where :

q_c : conviction heat flux [kJ/h]

h: conviction heat transfer coefficient [W/m^{2o}C]

T: surface temperature [^{o}C]

The Latent heat gain by the ventilation or infiltration is calculated by using [19,28]:

$$q_{it} = V\, a\, \rho\, (\omega_o - \omega w) h_{fg} \quad \text{......4.4}$$

Where :

qit: infiltration heat flux [kJ/h]

V: volume [m^3]

a: Times of air change [1/h]

ω: air humidity ratio [kg/kg dry air]

h_{fg}: air enthalpy [kJ/kg]

Other data for the mathematical model used in this type (type 56) can be obtained from TRNSYS software manual [27].

4.3.2. Chamber Load Modeling With Type56 and TRNBuild in TRNSYS

TRNBuild is used to enter the chamber wall layers and infiltration data that discussed in chapter three section (3.3), this input data used to create the chamber description file (Chamber.bui). This file includes all the information required to simulate the chamber where (Chamber.bui) file used to generate three new files, the (Chamber.bld) -the file containing the Geometric information about the chamber-, (Chamber.trn) -the file containing the wall transfer function coefficients-, and information file (Chamber.inf) files which are used by type 56 during the simulation process in TRNStudio program.

The chamber wall's layers definition using TRNBuild program, as shown in the figure 4.1. the simulation program requires a sample of data from the user such as layer thermal conductivity, specific heat, and density, then this layers can be used in the construction of chamber walls.

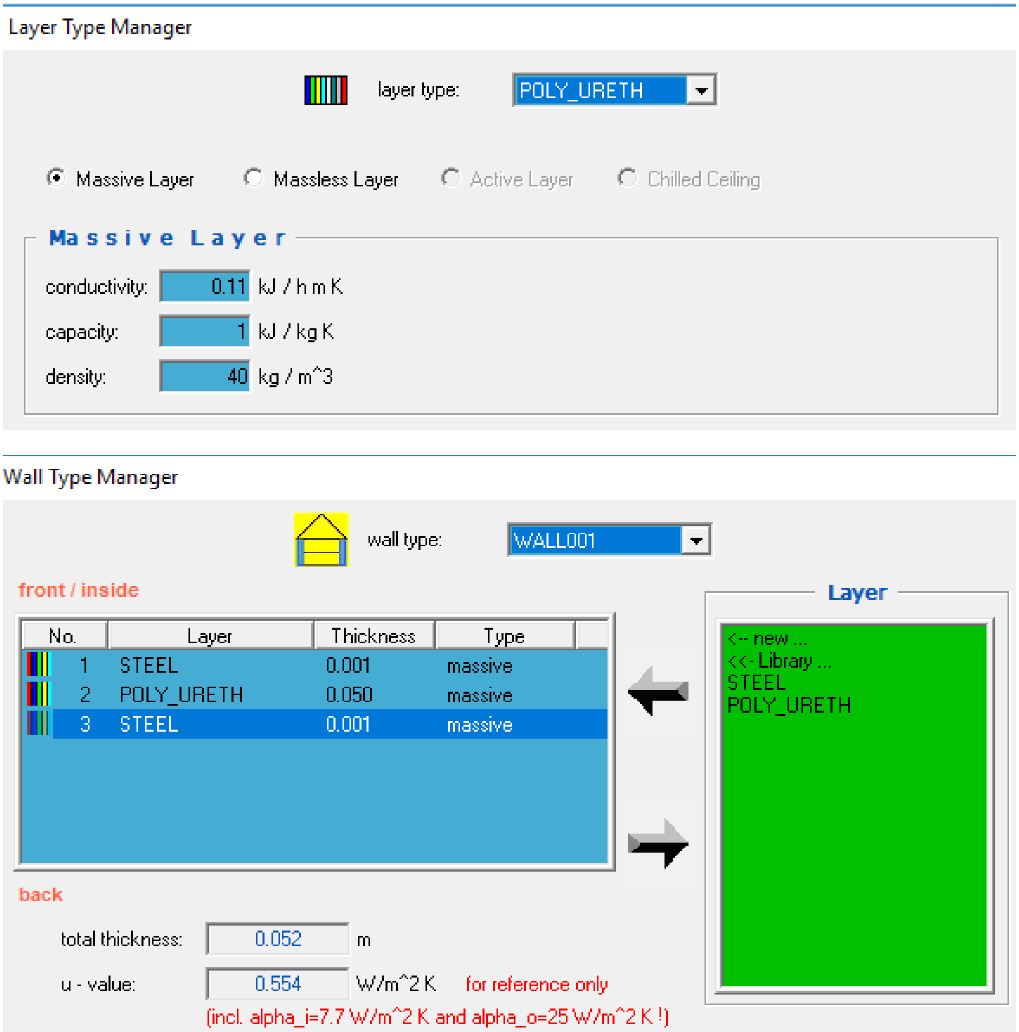

Fig.4.1: Wall layers definition using TRNBuild.

Figure 4.2 shows that, TRNBuild allows the users to specify all the chamber structure in details that is needed to simulate the thermal cooling loads of the refrigeration chamber such as geometry data, wall construction data, infiltration data, cooling data and etc. Furthermore, it needs schedule information which define the internal heat gain from the cooling equipments such as fan motor, electrical heaters and etc… .

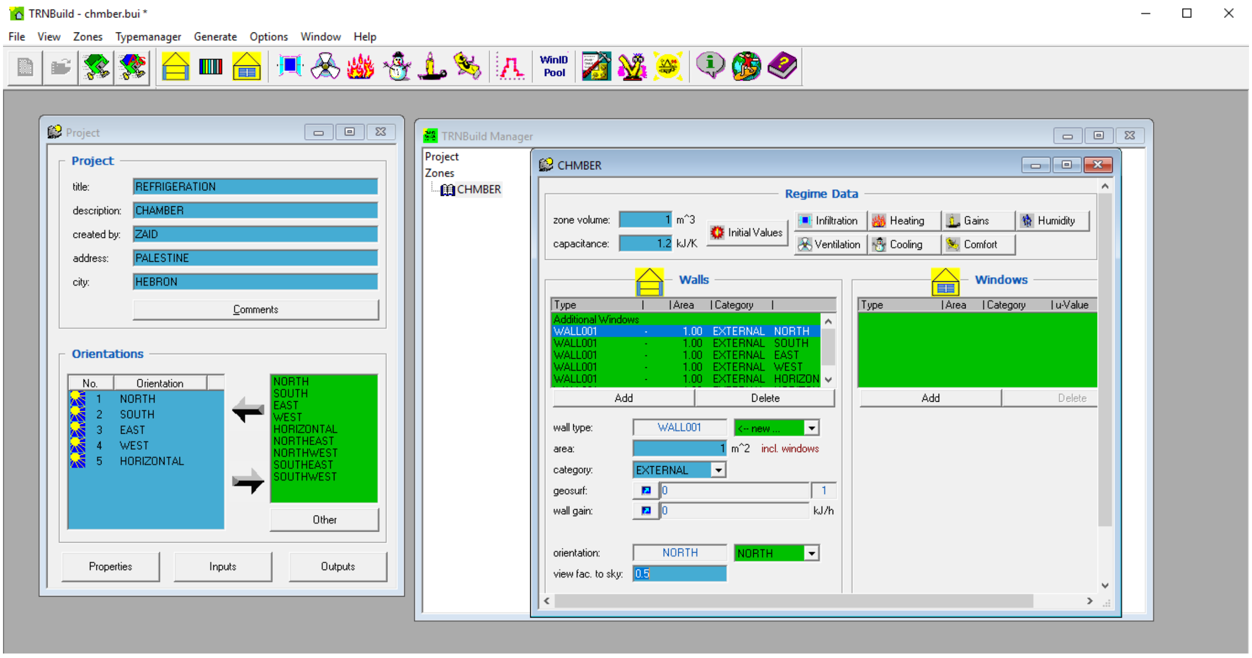

Fig.4.2. Chamber Load Modeling using TRNBuild Program.

4.4. Thermal Cooling Load Simulation Results

The TRNBuild program used to model the chamber load data from walls conduction, conviction, infiltration and other small gains, that discussed in chapter three with respect to different ambient temperature along the year. On the other hand TRNStudio program used to model the product load inside the chamber by using equation 3.3 and 3.4 that discussed in the previous chapter. The total annual load for the refrigeration chamber is the summation between the walls load from TRNBuild program using type56 and the product load using equation type in TRNStudio program. Figure 4.3. shows the connection between walls load and product load in TRNStudio program.

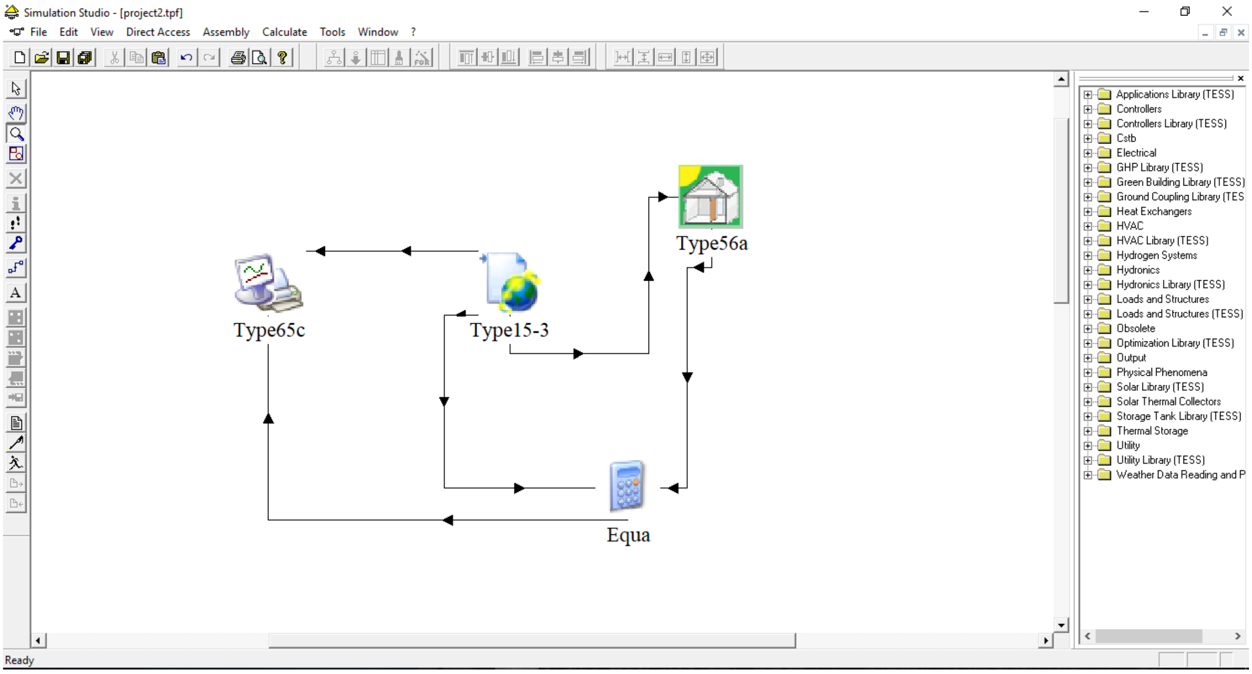

Fig.4.3. Chamber load model (Type 56) connections with product load in TRNStudio.

Figure 4.4. Illustrates the walls load obtained from TRNSbuild program, the product load obtained from TRNStudio program and the total annual cooling load for the refrigeration chamber, Type65 in TRNStudio program using to show the output data for the chamber load along the year, for more details about equation type or type65 used in TRNStudio program, see the TRNSYS16 manual [27].

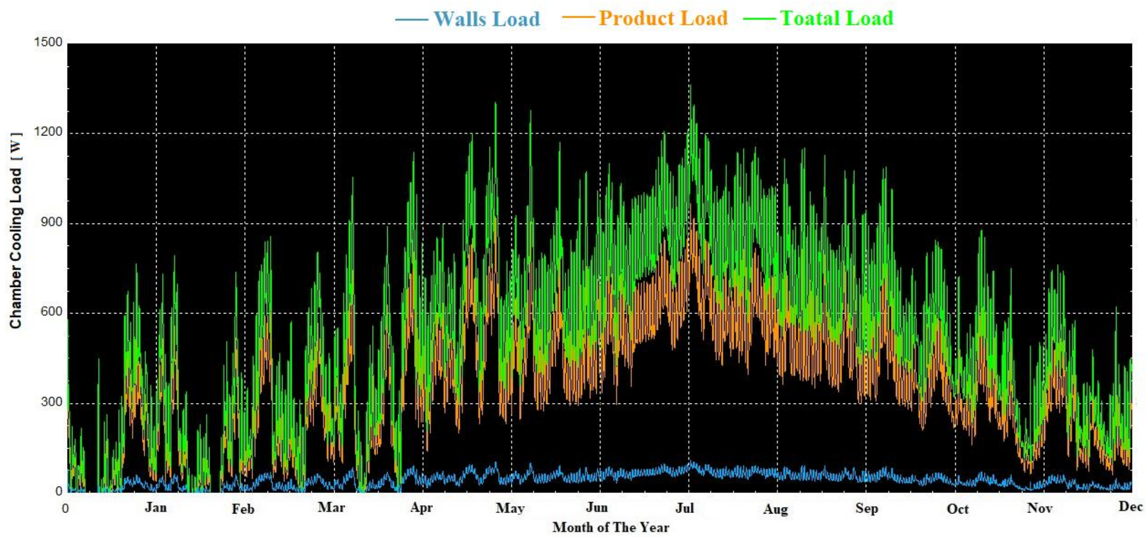

Fig.4.4. Walls product load and total annual cooling load for the chamber.

The maximum cooling load for the refrigeration chamber equal to 1,365 W occurs in July where the ambient temperature reach its maximum values, on the other hand sometimes (at the beginning of the year) the cooling load equals to zero watt which mean the ambient temperature equal or less the set point temperature for the refrigeration chamber as shown in figure 4.4.The total annual cooling load energy is 4,595.4 kWh/year.

CHAPTER 5
AIR COOLED CHILLER SIMULATION USING TRNSYS PROGRAM

5.1. Introduction

The first step in this chapter is selecting the suitable air cooled chiller that used to cover the chamber load demand. Air cooled chiller based on vapor compression refrigeration system this system is the dominant system today for cooling and refrigeration and is being used in almost all kind of applications. It is available for a wide range of sizes from 0.5 up to 10 TR [29].

5.2. Chiller Working Principle

Air cooled chillers absorb the heat from process water, then it transfer this heat into the air around the chiller unit. The cycle starts with the evaporator, which has a liquid refrigerant that enter a heat exchanger in order to cool the chilled water that used for cooling the refrigeration space. In this process, heat is absorbed from the chilled water circulating through the heat exchanger. Then the compressor compress the refrigerant vapor from the evaporator to the air cooled condenser, where the heat rejected to the surroundings, The high-pressure liquid out from the condenser moves through the expansion device and into the evaporator, in the process the refrigerant pressure is reduced along with the temperature. To complete the continuous cycle, the refrigerant flows back over the chilled water coils in the heat exchanger and absorbs more heat [16]. Figure 5.1. Shows the vapor compression refrigeration cycle that used for the operation of the air cooled chiller.

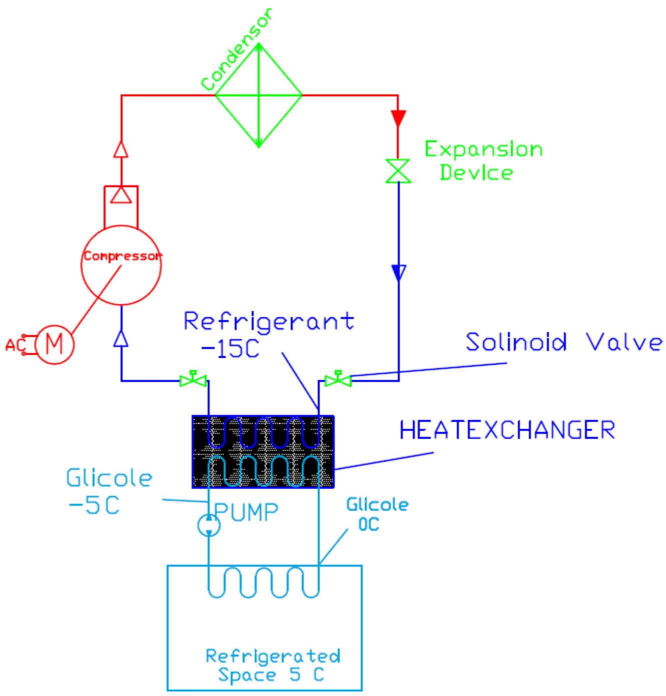

Fig.5.1. Air cooled chiller refrigeration cycle.

5.3. Chiller Selection

In this study, a half refrigeration ton (1.75 kW) air cooled chiller selected for covering the cooling load demand. This chiller produced by ChillX company with the model (CXF050DRS), the company data sheet is attached in appendix (A) [29].

The chiller used ethylene glycol water mixture in order to decrease the mixture freezing point under -10 ^{o}C, when the glycol percentage equal to 40% (by the volume) in the water mixture the freezing point temperature equal to -23.8 ^{o}C as shown in figure 5.2. and the specific heat for the mixture at this point equal to 3.3 kJ/kg K [30].

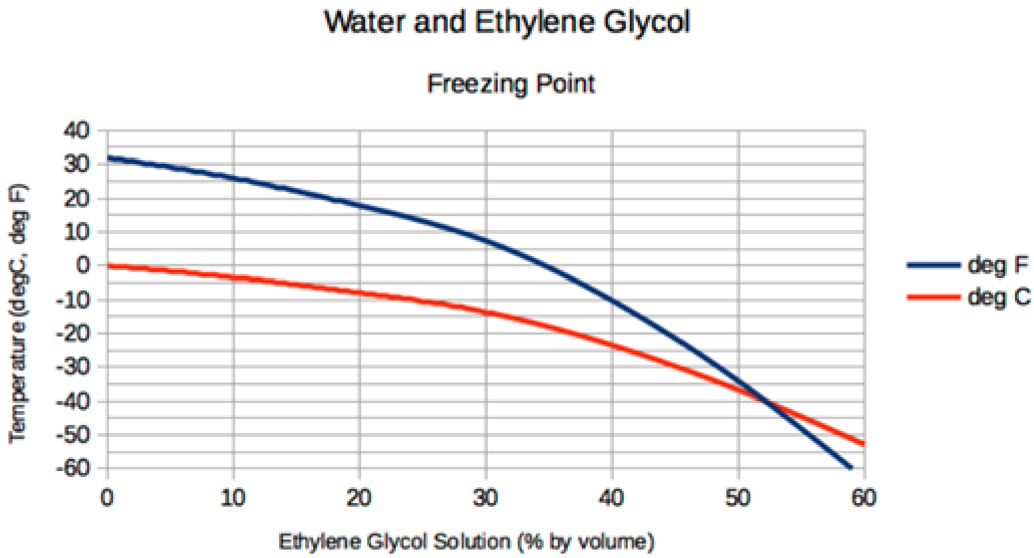

Fig.5.2. Water ethylene glycol freezing point [30].

5.4. Chiller Modeling Using TRNStudio Program

Type655 used to model a vapor compression air cooled chiller. This type require input parameters which are the inlet fluid (Water ethylene glycol) temperature, inlet fluid flow rate, set point temperature, ambient temperature and fluid specific heat in order to calculate the annual thermal cooling load demand.

Inlet fluid temperature taken zero ^{o}C, inlet fluid flow rate calculated using equation 5.1, set point temperature taken -5 ^{o}C, ambient temperature taken from whether data file and the specific heat of the fluid equal to 3.3 kJ/kgK. Figure 5.3 shows the chiller connections in TRANStudio program in order to calculate the total thermal cooling load.

$$m^* = \frac{Qtotal}{C_p \times \Delta T} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 5.1$$

Where :

$Qtotal$: Thermal Cooling load for the chamber [W].

m^* : Mass flow rate [kg/sec].

C_p : Specific heat of the water ethylene glycol in [kJ / kg .℃].

ΔT : Temperature deference for inlet and outlet fluid [°C].

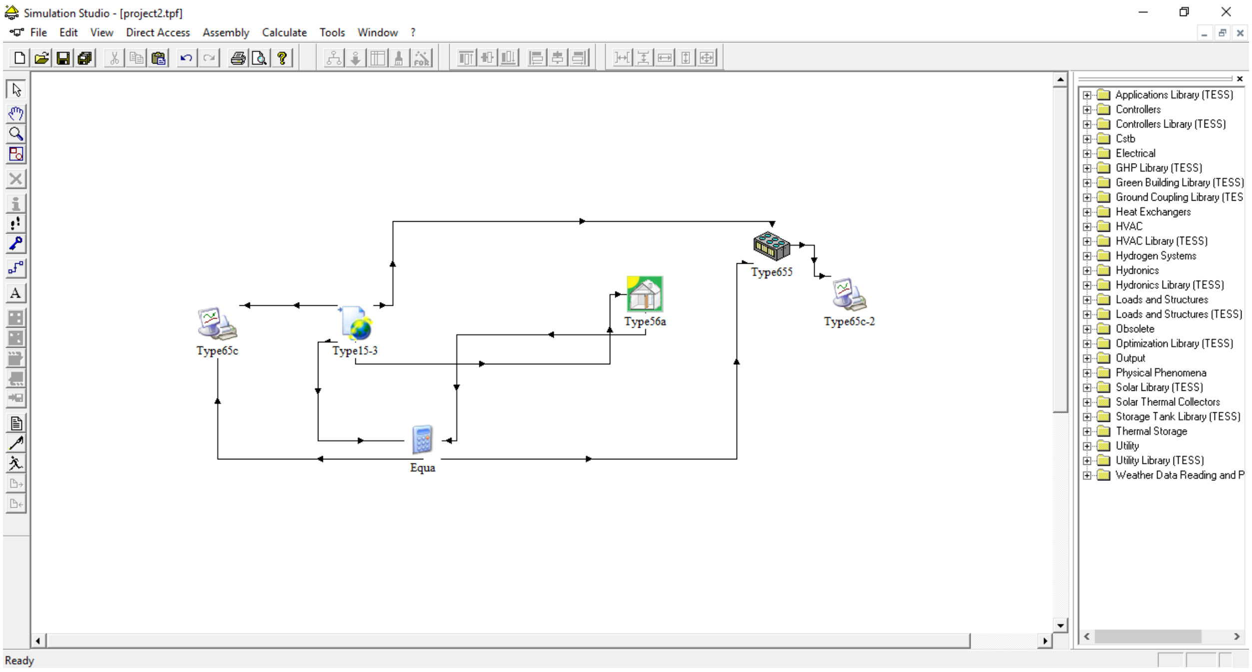

Fig.5.3. Chiller connection in TRANStudio Program .

In order to calculate the total electrical power that needed to cover the total thermal cooling load, type655 require a sample data for the COP of the used chiller. This data entered the simulation model (type655) using Text file. Type655 calculate the electrical power using equation 5.2 .

$$Pelec. = \frac{Qtotal}{COP} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 5.2$$

Where :

$Qtotal$: thermal Cooling load for the chamber [W].

$Pelec.$: Electrical Power [W].

COP : Coefficient of performance for the chiller.

The needed text file for the COP is developed using the COP values that illustrated in table 5.1, these values are varied according to the ambient temperature and the outlet water glycol mixture. The COP values are taken from the selected chiller data sheet attached in appendix (A) [29].

Table 5.1: The COP values for the selected chiller.

Glycol T	Ambient T	10 ºC		26.66 ºC		35 ºC		40.5 ºC	
		50 ºF		80 ºF		95 ºF		105 ºF	
		EER	COP	EER	COP	EER	COP	EER	COP
-9.44 ºC	15 ºF	5.1	1.49	4.3	1.26	3.6	1.06	3.1	0.91
-6.67 ºC	20 ºF	5.8	1.70	4.9	1.44	4	1.17	3.5	1.03
-1.11 ºC	30 ºF	7	2.05	5.9	1.72	4.9	1.44	4.3	1.26
4.44 ºC	40 ºF	7.7	2.26	6.8	1.99	5.7	1.67	5	1.47
10.00 ºC	50 ºF	---	---	7.6	2.23	6.3	1.85	5.5	1.61
15.55 ºC	60 ºF	---	---	8.4	2.46	7	2.05	6.1	1.79

In order to use the COP values shown in table 5.1 in TRANStudio program, this value must be converted to the require form in a text file that can be read by type 655 as shown in figure 5.4. As mentioned in the table 5.1. The air cooled chiller does not work when the ambient temperature less or equal the glycol temperature.

```
-9.44  -6.67  -1.11  4.44  10  15.55    ! Chilled Water Temperatures (C)
0    10    26.66    35     40.5         ! Ambient Air Dry-Bulb Temperatures (C)
 0                                              ! COP (All Power) Ratio at -9.44/0
 1.495                                          ! COP (All Power) Ratio at -9.44/10
 1.26                                           ! COP (All Power) Ratio at -9.44/26.66
 1.055                                          ! COP (All Power) Ratio at -9.44/35
 0.91                                           ! COP (All Power) Ratio at -9.44/40.5
 0                                              ! COP (All Power) Ratio at -6.67/0
 1.699                                          ! COP (All Power) Ratio at -6.67/10
 1.436                                          ! COP (All Power) Ratio at -6.67/26.66
 1.172                                          ! COP (All Power) Ratio at -6.67/35
 1.025                                          ! COP (All Power) Ratio at -6.67/40.5
 0                                              ! COP (All Power) Ratio at -1.11/0
2.05                                            ! COP (All Power) Ratio at -1.11/10
1.729                                           ! COP (All Power) Ratio at -1.11/26.66
 1.436                                          ! COP (All Power) Ratio at -1.11/35
 1.465                                          ! COP (All Power) Ratio at -1.11/40.5
 0                                              ! COP (All Power) Ratio at 4.44/0
 2.256                                          ! COP (All Power) Ratio at 4.44/10
 1.992                                          ! COP (All Power) Ratio at 4.44/26.66
 1.67                                           ! COP (All Power) Ratio at 4.44/35
 1.465                                          ! COP (All Power) Ratio at 4.44/40.5
 0                                              ! COP (All Power) Ratio at 10/0
 2.579                                          ! COP (All Power) Ratio at 10/10
 2.227                                          ! COP (All Power) Ratio at 10/26.66
 1.846                                          ! COP (All Power) Ratio at 10/35
 1.611                                          ! COP (All Power) Ratio at 10/40.5
 0                                              ! COP (All Power) Ratio at 15.55/0
 2.637                                          ! COP (All Power) Ratio at 15.55/10
 2.461                                          ! COP (All Power) Ratio at 15.55/26.66
 2.051                                          ! COP (All Power) Ratio at 15.55/35
 1.787                                          ! COP (All Power) Ratio at 15.55/40.5
```

Fig.5.4. Text file for the chiller COP values.

Figure 5.5 illustrates the total thermal power for the refrigeration chamber, the total electrical power that needed to cover the thermal power and the COP values.

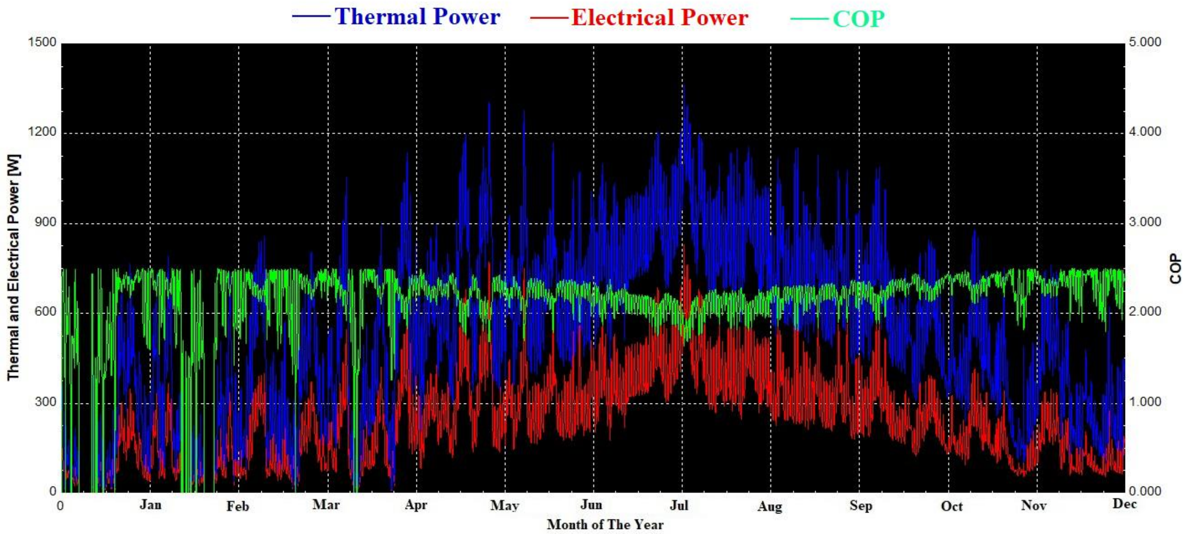

Fig.5.5. Chiller thermal power, electrical power and chiller COP values.

As shown in figure 5.5 the maximum electrical power value equal to 835 W in July where the thermal power reach its maximum value. The total annual electrical energy is calculated by integrate the electrical power profile (red profile) this value equal to 2107 kWh/year. In some areas the value of the COP equal zero which mean the ambient temperature equal or less the chamber temperature so the chiller does not work.

CHAPTER 6
SYSTEM LOAD COVERAGE SCENARIOS

6.1. Introduction

In order to operate the air cooled chiller that used to cover the total annual cooling load for the refrigeration chamber three scenarios used to achieve this goal. The three scenarios are on grid PV system, PV system with thermal storage tank and full load coverage using PV system and storage tank. The third scenario developed from the second one to compensate all thermal load during off-peak period.

6.2. First Scenario (On Grid PV System)

In this scenario only PV system used to operate the air cooled chiller during on peak periods, in this case the system contain PV modules, on grid inverter and air cooled chiller as shown in figure 6.1.

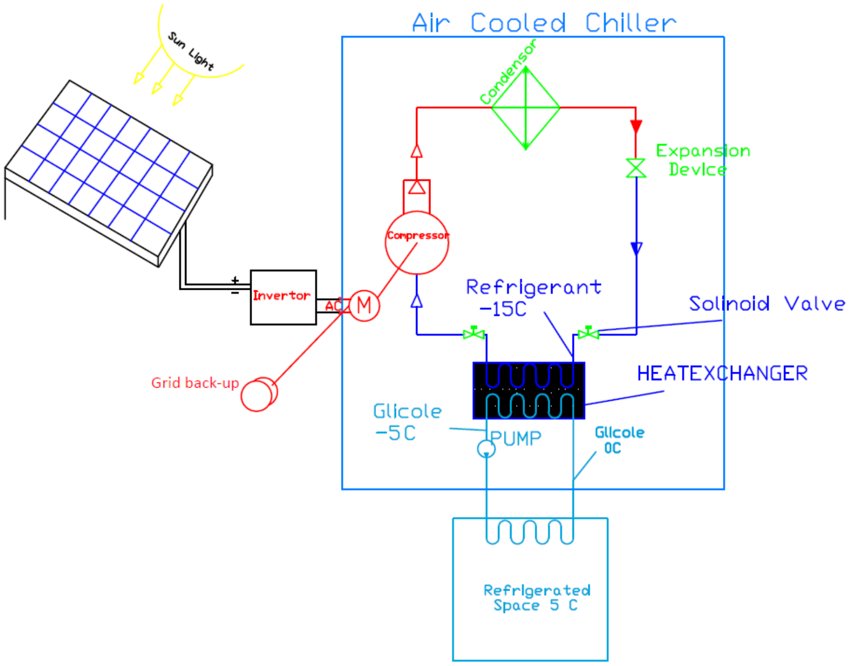

Fig.6.1. System components in the first scenario .

The PV module converts solar radiation into electric power as direct current (DC). The inverter converts DC into alternating current (AC) which is needed to drive the chiller compressor. The chiller converts the AC power to the needed thermal cooling power. The air cooled chiller is supplied as a back-up with an electric AC power from the grid, when there is not enough DC power from the PV-array, especially at night, evening and morning of the day when there is no enough solar radiation to drive the chiller.

6.2.1. PV Array Sizing and Design

Due to the shift in the peak power of the chiller and the PV-system with the seasonal variation, the total electrical energy method used to calculate the number of the PV modules for the system, instead of the chiller peak power (835 W). As mentioned in the previous chapter the total annual electrical energy that needed to cover the thermal power equal to 2107 kWh/year. According to the equation 6.1 the needed peak power equal to 1.27 kWp .

$$P_{peak} = \frac{Total\ annual\ electrical\ energy}{Total\ annual\ elctrical\ energy\ produced\ by\ 1\ kWpeak/year} = \frac{2107}{1650} = 1.27\ kWp \quad \text{……..…6.1.}$$

The total annual electrical energy produced by one kW peak power can be calculated by knowing the number of peak hours in Hebron city which are equal to 5.4 hours, the annul energy calculated using equation 6.2 which is equal to 1944 kWh/year, due to the power losses the value equal to 1650 kWh/year in Hebron city[18].

$$E_{annual} = 1\ kWp \times 5.4\ hours \times 30\ day \times 12\ month = 1944\ kWh/year \quad \text{…………….6.2.}$$

Different sizes of PV modules produce different amounts of power. The power produced depends on the size of the PV system and the radiation at the site. The QCELLS solar module (Q.PLUS L-G4.2 320-345) 325 (Wp), is produced by QCELLS solar company in Germany where the Module's efficiency is 16.3%.It was selected for all PV scenarios in this study. The data sheet of the PV module is attached in Appendix (B)[31].

The number of PV modules required is calculated by equation 6.3. The air cooled chiller should be powered by 4 PV-modules with 325 Wp.

$$No.of\ PV\ modules = \frac{Peak\ Power\ Needed}{Peak\ Power\ of\ PV\ Module} \quad \text{………………………………………….6.3.}$$

$$No.of\ PV\ modules = \frac{1.27}{0.325} = 3.9 \cong 4\ Modules\ .$$

The total PV array's area required is calculated by using equation 6.4 where the PV module area equal 1.994m^2 as shown in Appendix (B).

PV array's area $= Module\ Area \times No.of\ modules$ …………………………..……....6.4.

PV array's area $= 1.994 \times 4 = 7.976\ \text{m}^2$.

The four PV modules connected in series to obtain the needed power along the year.

6.2.2. Inverter Selection

On-grid inverter is used to convert (DC) current from PV-array to (AC) current in order to run the air cooled chiller. For the suggested PV system, the required inverter should supply 1.21 kW. The selected inverter EVERSOL TL, 2 kW with a peak efficiency of 97%. This inverter is produced by the ZEVERSOLAR Company, the data sheet is attached in Appendix (C) [32].

6.2.3. Back-up System

When there is no enough electric power coming from the PV-array, to cover the cooling power demands, the back-up system is designed to deliver the remaining needed AC power. The back-up system is assumed to have a direct connection to the electricity grid and to the air cooled chiller. This connection is considered and designed for all PV scenarios.

6.2.4. First Scenario Simulation In TRNSYS

Type 194b (PV-inverter) is used to model the PV array and the inverter for all scenarios in this study, the model is based on the calculation method presented by DeSoto et al [33]. Type 194b need many input parameters from whether data file such as total radiation on tilt angle, beam radiation, sky diffuse radiation, ground reflected diffuse radiation, slope of surface, wind speed and ambient temperature. On the other hand the model need the electrical characteristics from the PV module data sheet[31], such as short circuit current and open circuit voltage at Standard Test Condition (STC), module voltage and current at Maximum Power Point (MPPT), temperature coefficient at short circuit current and open circuit voltage, number of cells in series, number of module in series, number of module in parallel, Normal Operating Conditions Test (NOCT) and the module area. Also the inverter parameters efficiency and power are used in type 194b. Figure 6.2 shows the connections of first scenario using type 194b in TRNStudio program.

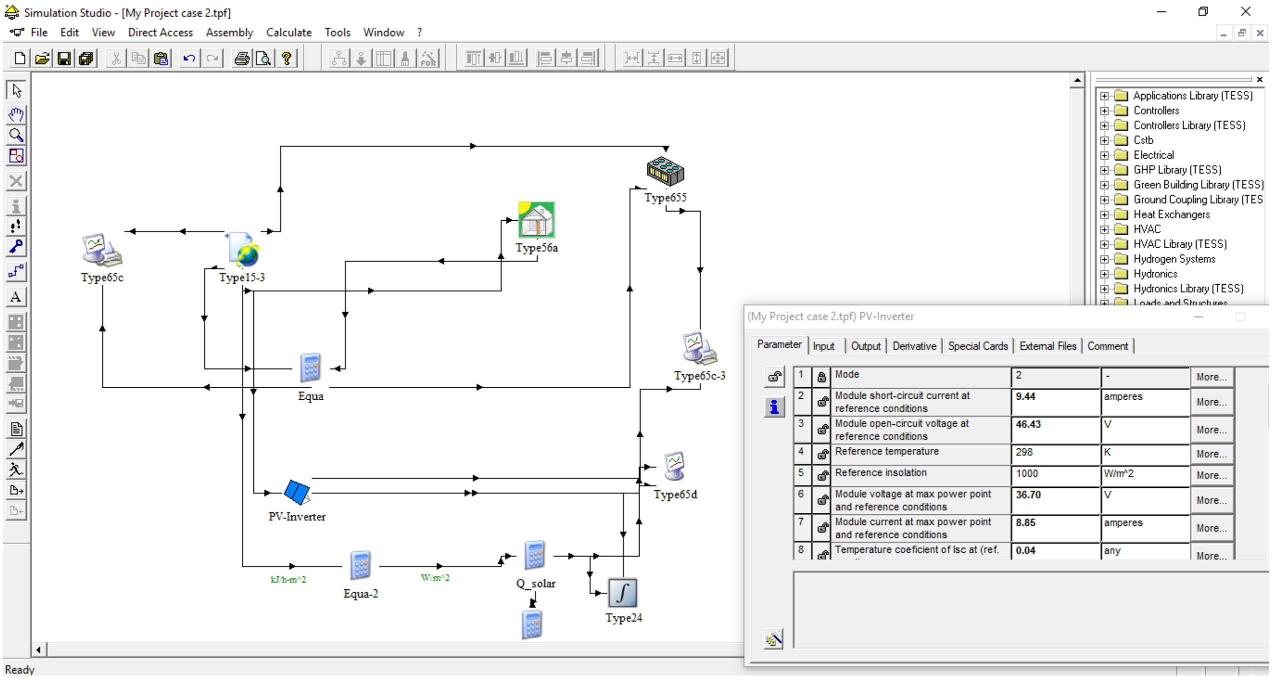

Fig.6.2. First scenario modeling using type 194b in TRNStudio program.

By running the simulation program for the first scenario illustrated in the figure 6.2., the output electrical power obtained from the PV array along the year shown in figure 6.3.

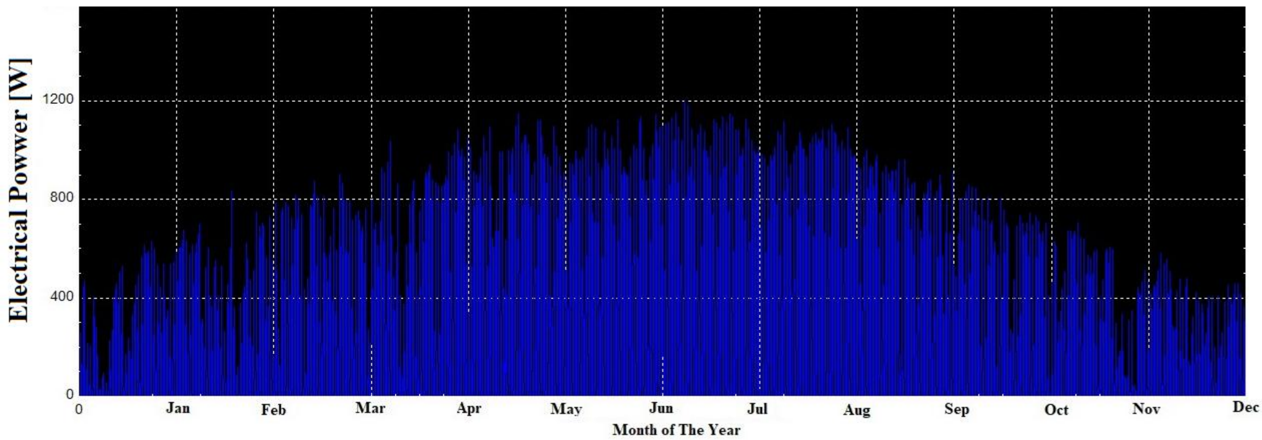

Fig.6.3. Annual electrical power obtained from the PV array.

As shown in figure 6.3 the maximum power produced by the PV array equal to 1,210 W in Jun, this power values moved to excel sheet file in order to compare between the chiller power and the power produced by the array. Figure 6.4 to Figure 6.7 illustrate the weekly distribution of electrical powers of the system in winter, spring, summer and in autumn for the chiller and the PV array.

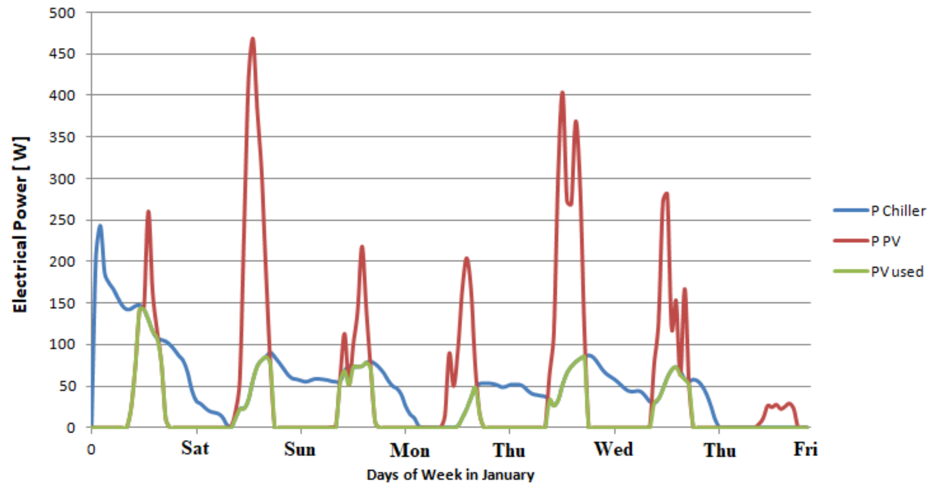

Fig.6.4. First scenario electrical powers in winter week.

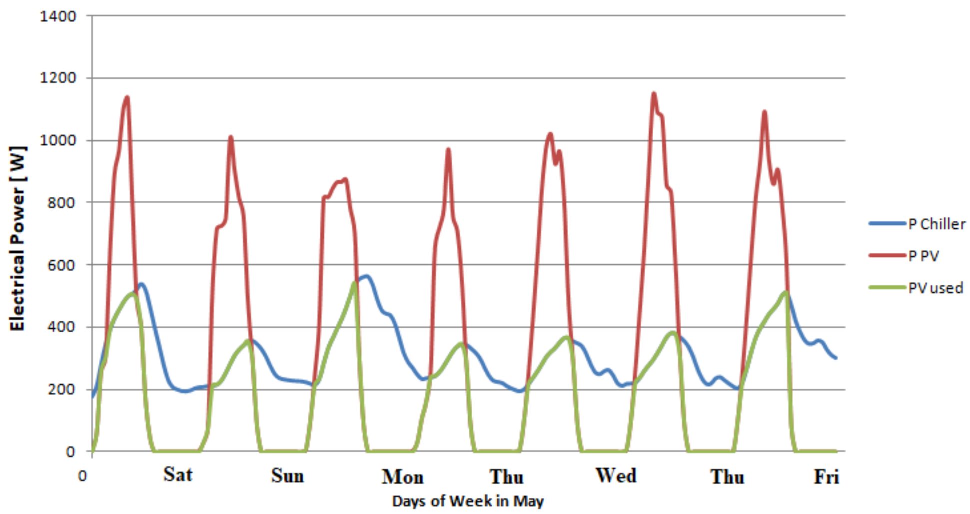

Fig.6.5. First scenario electrical powers in spring week.

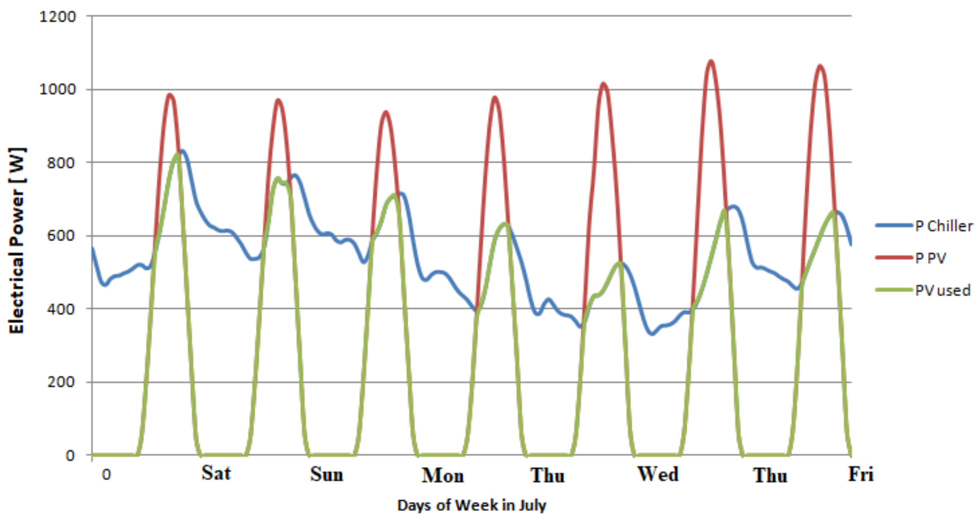

Fig.6.6. First scenario electrical powers in summer week.

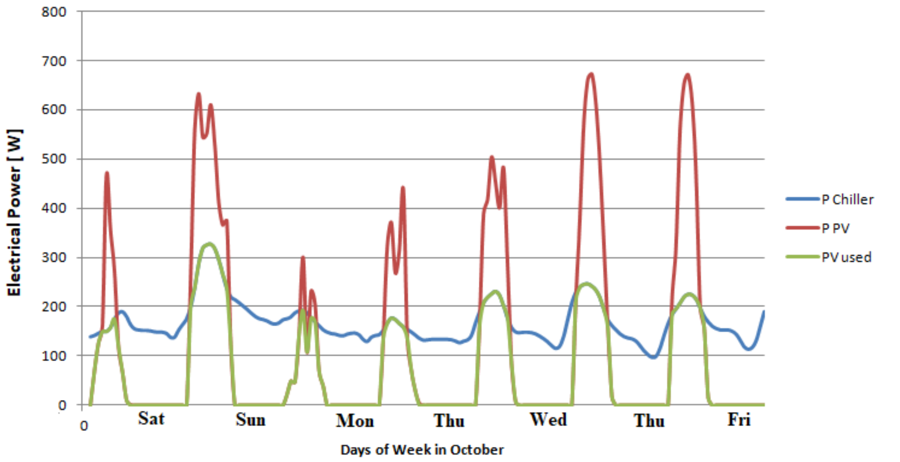

Fig.6.7. First scenario electrical powers in autumn week.

As shown in figures 6.4 to 6.7, the red profile represents the power produced by the PV array, the blue profile represents the chiller power and the green profile represents the used power obtained from PV array. In winter the produced power by the PV between 0 to 470 W, also the chiller power is low between 0 to 250 W, the used power from PV array equal 35% of the total needed power for the chiller. In spring the PV array reach its peak power which around 2,000 W so the array cover 50% of the chiller power. In summer the needed power for the chiller reach its maximum values so the PV array cover 43% of the needed power. Finally in autumn the array cover 41% of the chiller power.

6.2.5. First Scenario Results

The power that direct used from PV array in order to run the chiller in the on-peak hours is represented in figure 6.8 in the blue profile, this profile integrated using excel file to calculate the total annual power that direct used from the PV array, which equals to 950.5 kWh/year, this value represents 45% of the total annual power (2107 kWh/year) that needed to run the chiller along the year as illustrated in figure 6.8.

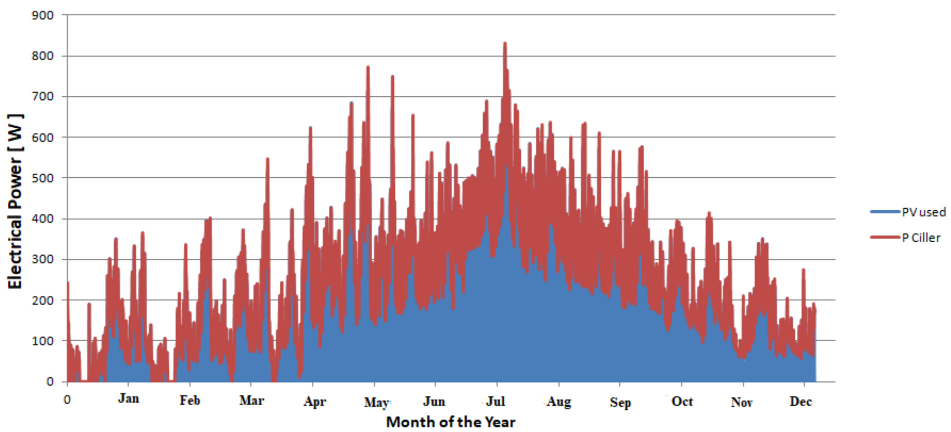

Fig.6.8. Total annual power for the chiller and total annual power direct used from PV.

In this scenario the excess power from PV array is supplied to the grid, getting an equal value for it during night hours - in case of electricity consumption for the chiller is greater than the PV system production in one month - based on Power Authority laws in Palestine, this value equals to 752 kWh/year which represents 36% of the total annual power needed. Based on the first scenario result's, 81% of the total annual electrical power needed to run the chiller was covered as shown in figure 6.9.

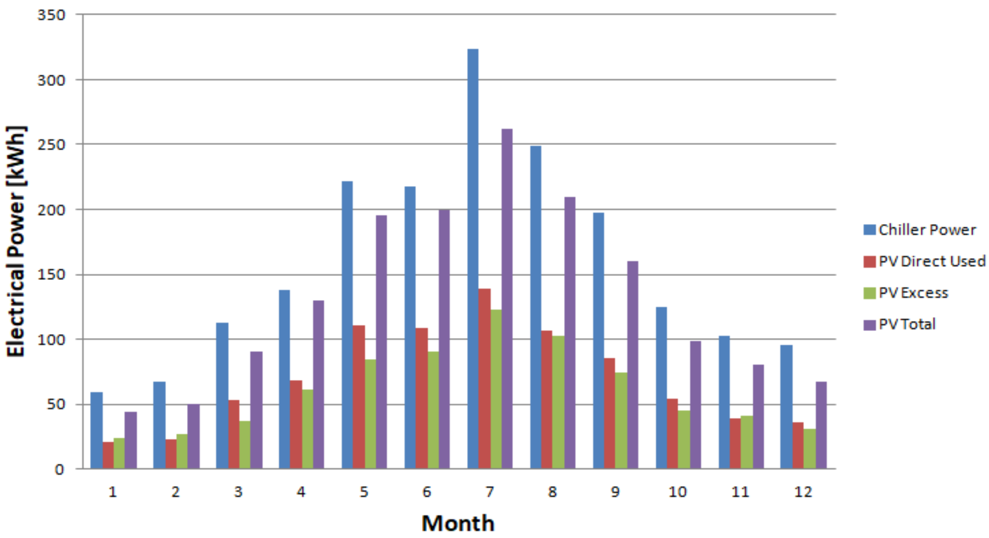

Fig.6.9. Monthly distribution for the electrical power consumption by the chiller and production by the PV system.

Figure 6.9. depicts the electrical power consumed by the chiller and the power produced by the PV-system, the blue profile represents the total electrical power needed to run the chiller, the purple profile represents the total electrical power produced by the PV-system, the red profile represents the electrical power direct used from the PV array at on-peak periods and the green profile represent electrical power that transferred to the grid and used at the off-peak periods. On the other hand, 19% of the annual electrical power must be covered by running the chiller using utility electrical power.

6.3. Second Scenario (On Grid PV System With Thermal Storage Tank)

In this scenario PV system used in order to run the air cooled chiller during on peak periods, also thermal storage tank used in order to shift the excess power from on-peak periods to off-peak periods, this case contains the same components of the system in the previous scenario in addition to the thermal storage tank as shown in figure 6.10.

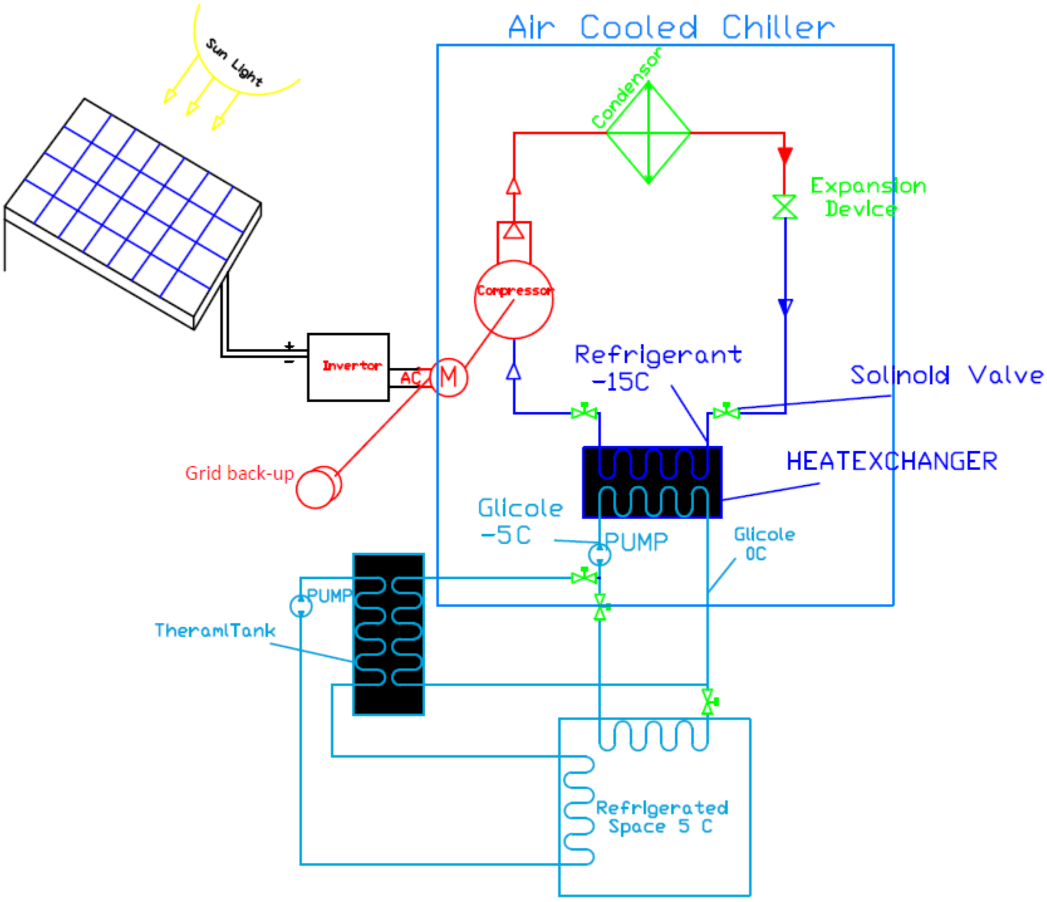

Fig.6.10. System components in the second scenario.

6.3.1. Second Scenario Components

The same components that used in the first scenario such as four PV modules (QCELLS), Inverter (EVERSOL TL), back-up system and the air cooled chiller are used in this scenario. The addition in this case is the thermal storage tank that used to store the excess cold power during the PV on-peak periods and using this power during the off-peak periods (night hours).

6.3.2. Thermal Storage Tank Design

In this case the excess power produced by the PV array at the on-peak period also used to run the chiller and the excess amount of the thermal power produced by the chiller in this periods stores in thermal storage tank. That's to provide cold energy for a few hours in the afternoon when solar radiation already decreases but internal cooling load demand is still high. Thermal load is used to compare between the chiller load and the load produced by the PV array and storage tank in this case, this because the storage tank deals only with thermal power.

The working fluid used in storage system is the same fluid that used for the chiller which is glycol water mixture (40% glycol) with 3.3 kJ/kgK specific heat and 1110 kg/m^3 density [30]. The volume of the thermal storage tank is designed according to the maximum excess of the thermal power produced by the PV during one day, this value equal 8000 Wh/day in Jun. By using equation 6.5 the tank volume equal 1.5 m^3 (1500 L).

$$V = \frac{m}{\rho} \quad \text{......} 6.5.$$

Where : V : Volume of the tank [m^3].
m: Fluid mass [kg].
ρ : Fluid density [kg/m^3].

The fluid mass calculated using maximum excess the thermal power by using equation 6.6.

$$m = \frac{Q_{excess}}{Cp \times \Delta T} \quad \text{......} 6.6.$$

Where : m : Fluid mass [kg].
Cp: Fluid specific heat[kJ/kgK].
ΔT : Fluid Temprature difrance [^{o}C].

$$m = \frac{Q_{excess}}{Cp \times \Delta T} = \frac{8\,[\text{kW}] \times 3600\left[\frac{\text{hr}}{\text{sec}}\right]}{3.3\left[\frac{\text{kJ}}{\text{kgK}}\right] \times 5\,[\text{C}]} = 1745.45 \text{ kg}.$$

$$V = \frac{m}{\rho} = \frac{1745.45}{1110} = 1.5 \text{ m}^3$$

The selected tank is (HF1500) produced by Reflex company in Germany . The tank data sheet attached in appendix (D)[34]. The heat loss of this tank equal 5.1 kWh/24h (212 Wh) at worst case.

6.3.3. Second Scenario Simulation

The excess electrical power produced by the PV array in the first scenario used to run the chiller in TRANStudio, then chiller produced thermal power that stores in thermal storage tank as shown in figure 6.11. The storage tank supply this power to the refrigeration chamber during the night hours.

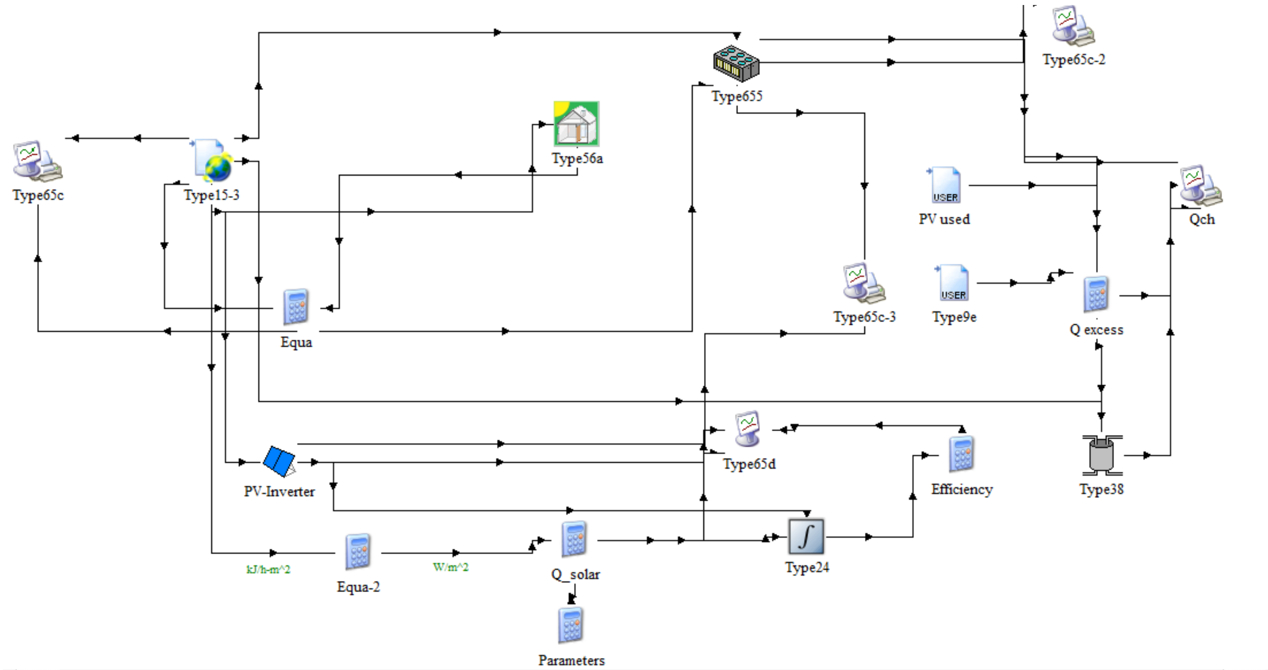

Fig.6.11. Second scenario simulation using TRANStudio program.

The output thermal load from the thermal tank depending on its heat losses which equal 212 Wh, the output thermal load calculated using excel sheet file. Figure 6.12 shows the thermal tank input and output powers.

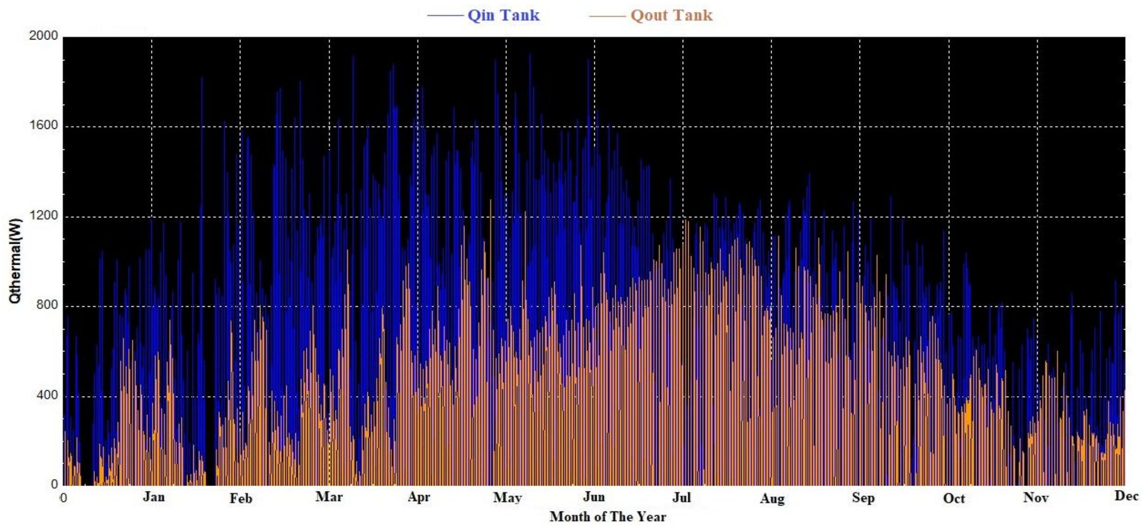

Fig.6.12. Thermal tank input and output power.

Figure 6.13 to Figure 6.16 illustrate the weekly distribution in winter, spring, summer and in autumn for the total thermal power produced by the PV array, thermal power direct used from the PV array and thermal power used from tank in order to cover the chiller thermal load.

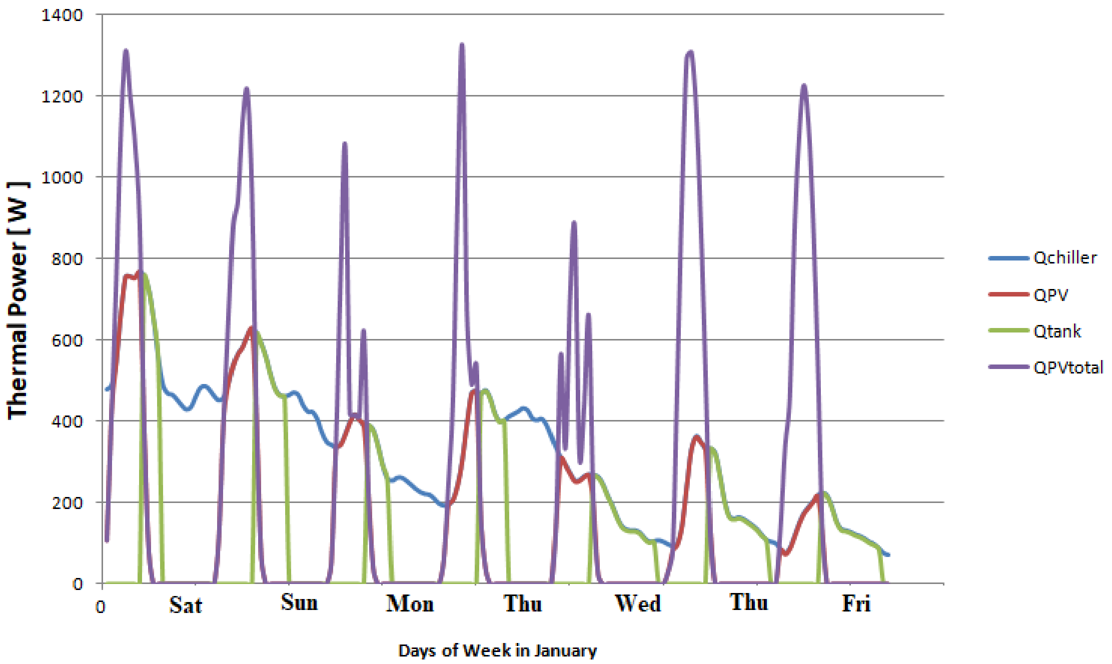

Fig.6.13. Second scenario thermal powers in winter week.

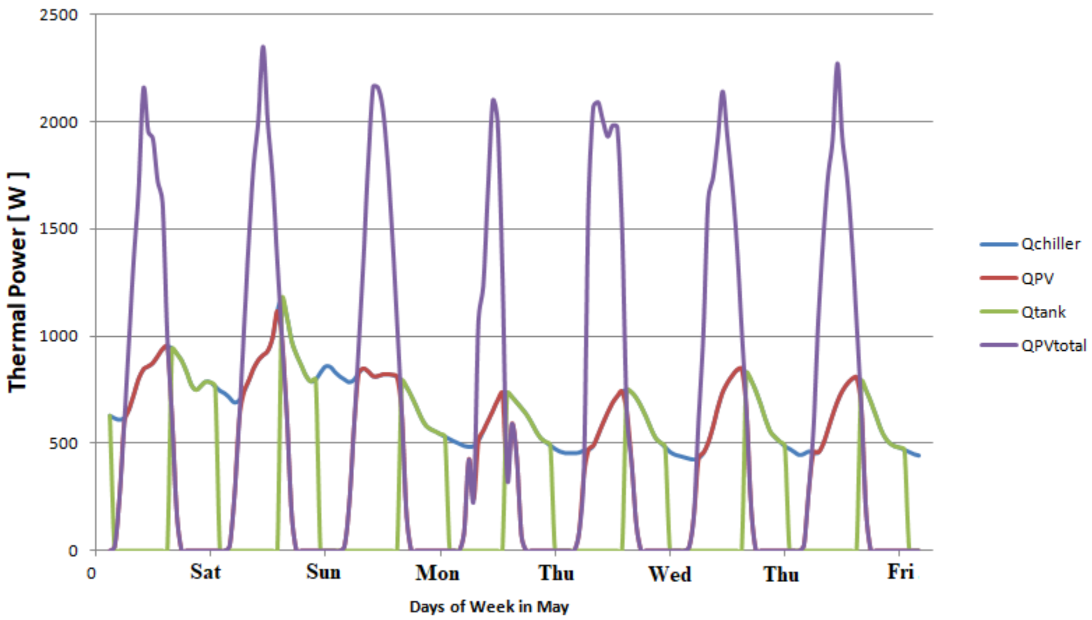

Fig.6.14. Second scenario thermal powers in spring week.

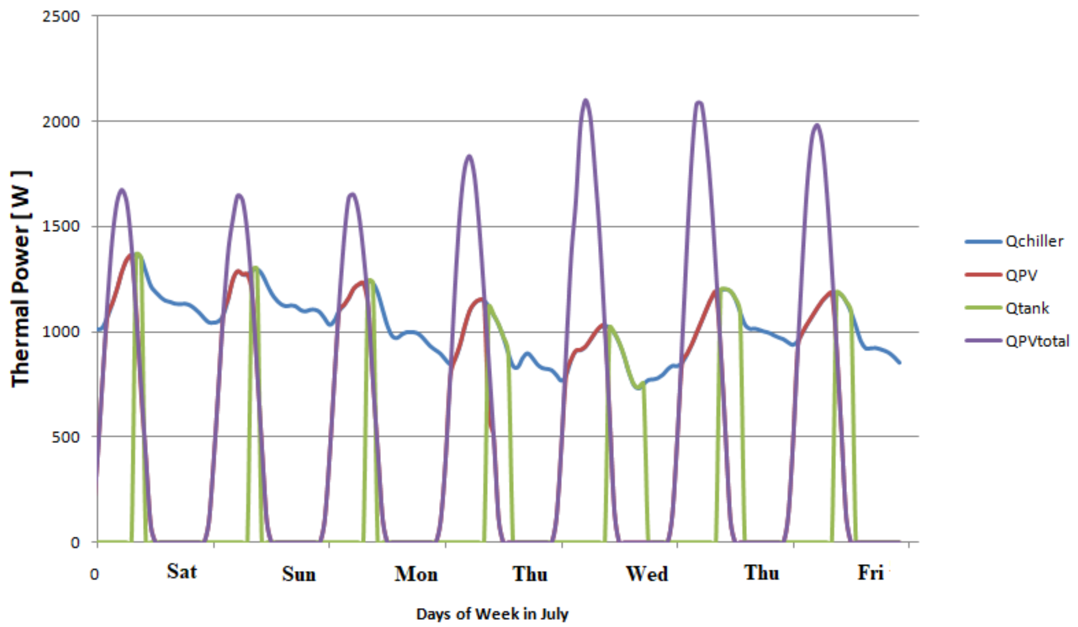

Fig.6.15. Second scenario thermal powers in summer week.

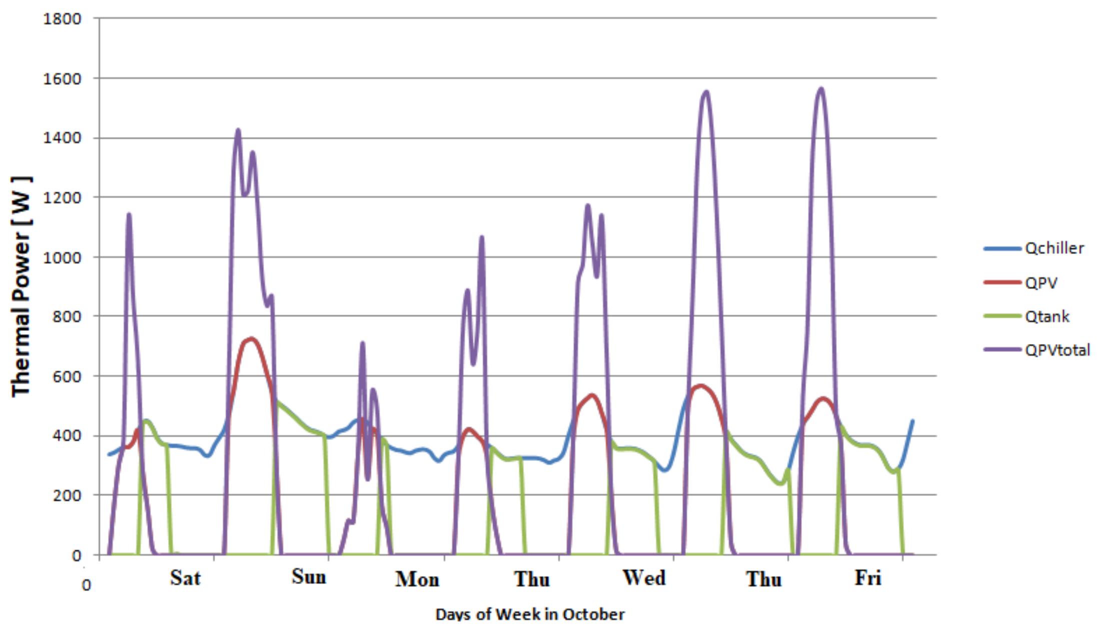

Fig.6.16. Second scenario thermal powers in autumn week.

As shown in figures 6.13 to 6.16, the purple profile represents the total thermal power produced by the PV, the red profile represents the thermal power direct used from the PV array at on-peak periods, the green profile represents thermal power obtained from storage tank at off-peak periods and the blue profile represents the chiller thermal power. In winter, 70% of the chiller load was covered, 38% direct from PV array and 32% from storage tank. In spring, 89% of the chiller load was covered, 49% direct from PV array and 40% from storage tank. In summer 59% of the chiller load was covered, 42% direct from PV array and 17% from storage tank. Finally In autumn 74% of the chiller load was covered, 40% direct from PV array and 34% from storage tank.

6.3.4. Second Scenario Results

In this scenario the excess power from PV array is moved to the thermal storage tank in order to use it at night hours, as shown in figure 6.17 the total needed load expressed using blue profile, and the total thermal power that covered using PV and storage tank expressed in red profile. The red profile integrated using excel file to calculate the total annual thermal power that direct used from the PV array and storage tank, which equal to 3,474 kWh/year, this value represent 75.6% (45% PV and 30.6% Storage Tank) of the total annual thermal power (4,595.4 kWh/year) that needed to run the chiller along the year as illustrated in figure 6.17. The thermal losses in storage tank equal to 310.6 kWh/year.

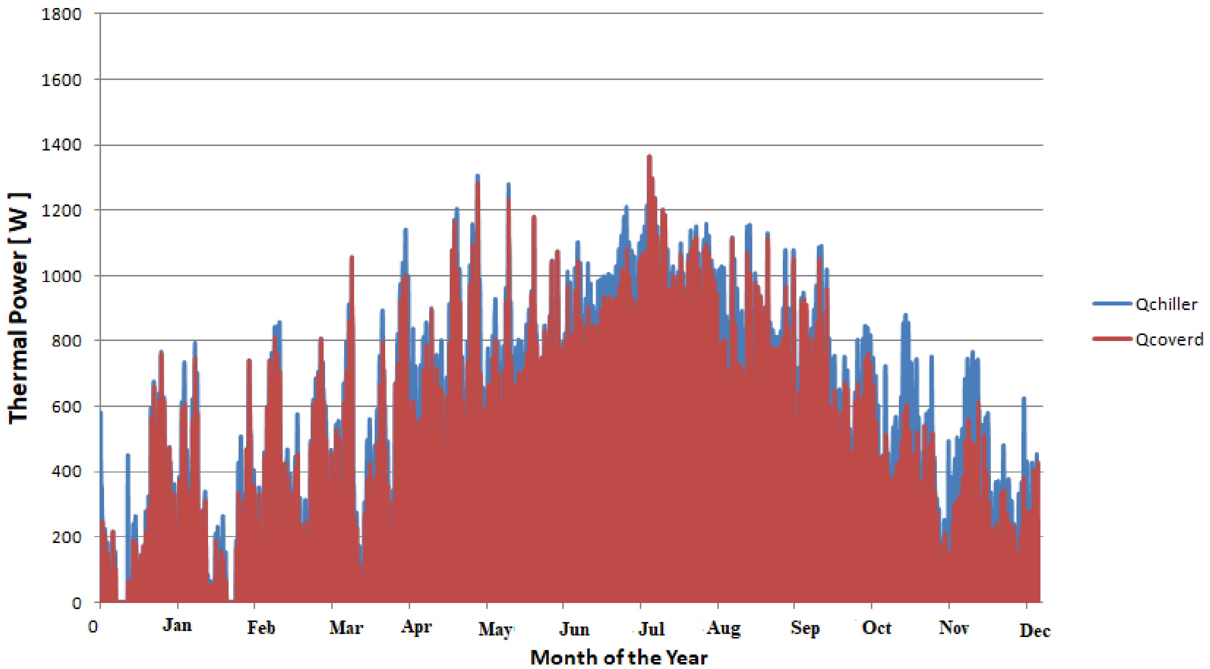

Fig.6.17. Total annual thermal power for the chiller and total annual thermal power covered by the PV and storage tank.

According to the second scenario results', the value of the electrical power that reduced in this scenario equal to 1593 kWh/year, this power obtained by using PV array and storage tank which represent 75.6% of the total electrical power needed to run the chiller (2107 kWh/year).

6.4. Third Scenario (Full Load Coverage Using PV System And Storage Tank)

This scenario uses the same components that used in the second scenario as shown in figure 6.10 and use the same process according to the load coverage operation. The deference between the two scenarios are the number of PVs and the storage tank volume, this values designed in order to cover the total annual load for the chiller.

6.4.1. PV Array Sizing and Design

In this case, the total energy was not covered using PV array in first scenario (1157 kWh/year) added to the total annual energy that needed to run the chiller (2107 kWh/year) in order to calculate the needed peak power in this scenario which equal 1.86 kWp according to the equation 6.1. The same module QCELLS solar module 325 (Wp) used in this scenario and the number of PV modules required is 6 PV modules calculated by equation 6.3. Due to the needed peak power (1.86 kW) the same inverter is used to convert (DC) current from PV-array to (AC) current in order to run the air cooled chiller.

6.4.2. Thermal Storage Tank Design

This case need larger storage tank in order to cover the total annual load for the chiller. The working fluid used in the storage system is the same fluid that used in the second scenario. The maximum excess of the thermal power produced by the PV during one day in this scenario is equal 10200 Wh/day in jun. By using equation 6.5 the tank volume equal 2 m^3 (2000 L).

The selected storage tank is (HF 2000) from the same data sheet attached in appendix (D)[34]. The heat loss of this tank equal 5.9 kWh/24h (245 Wh) at worst case.

6.4.3. Third Scenario Simulation

This case use the same simulation method that used in the second scenario as shown in figure 6.10, but this model use 6 PV modules instead of 4 PV modules. Figure 6.18 shows the extracted power from the PV array.

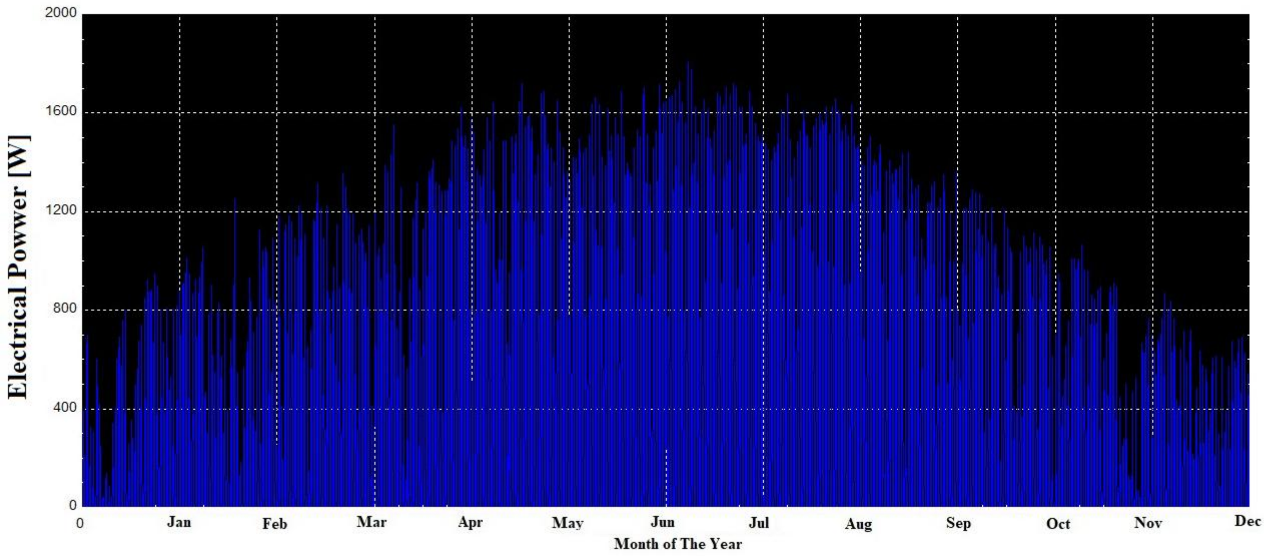

Fig.6.18. Annual electrical power obtained from the PV array.

Figure 6.19. shows the input and output thermal power that obtained from the thermal storage tank in third scenario.

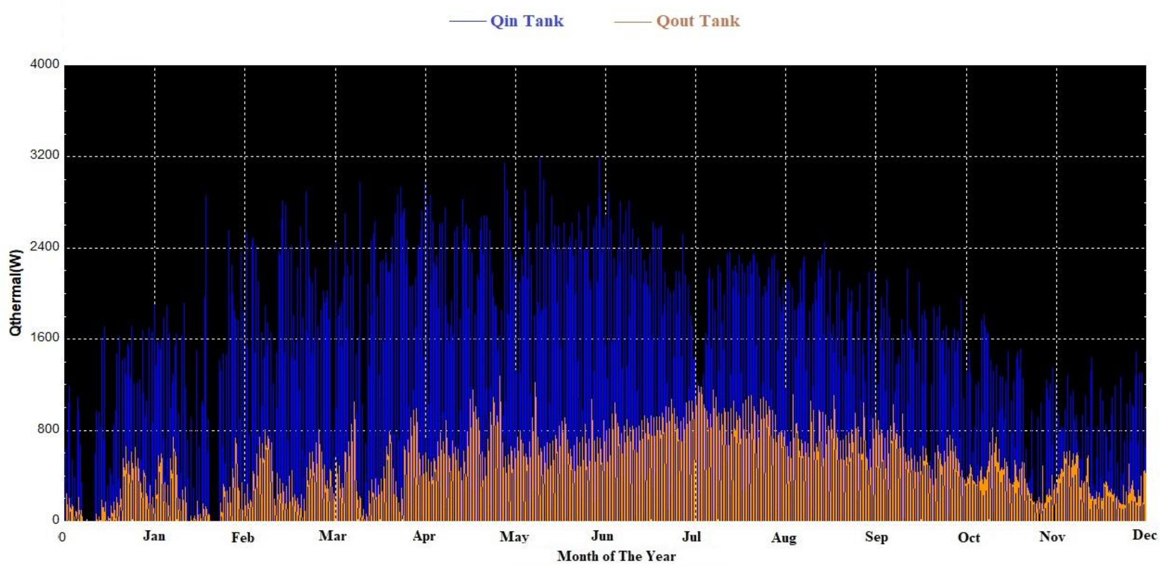

Fig.6.19. Thermal tank input and output power.

As shown in figure 6.19, the difference between the input thermal power and output power larger than the difference in the second scenario, this due to the larger amount of the thermal losses according the thermal storage tank.

Figure 6.20 to Figure 6.23 illustrate the weekly distribution in winter, spring, summer and in autumn for the total thermal power produced by the PV array, thermal power direct used from the PV array and thermal power used from tank in order to cover the chiller thermal load.

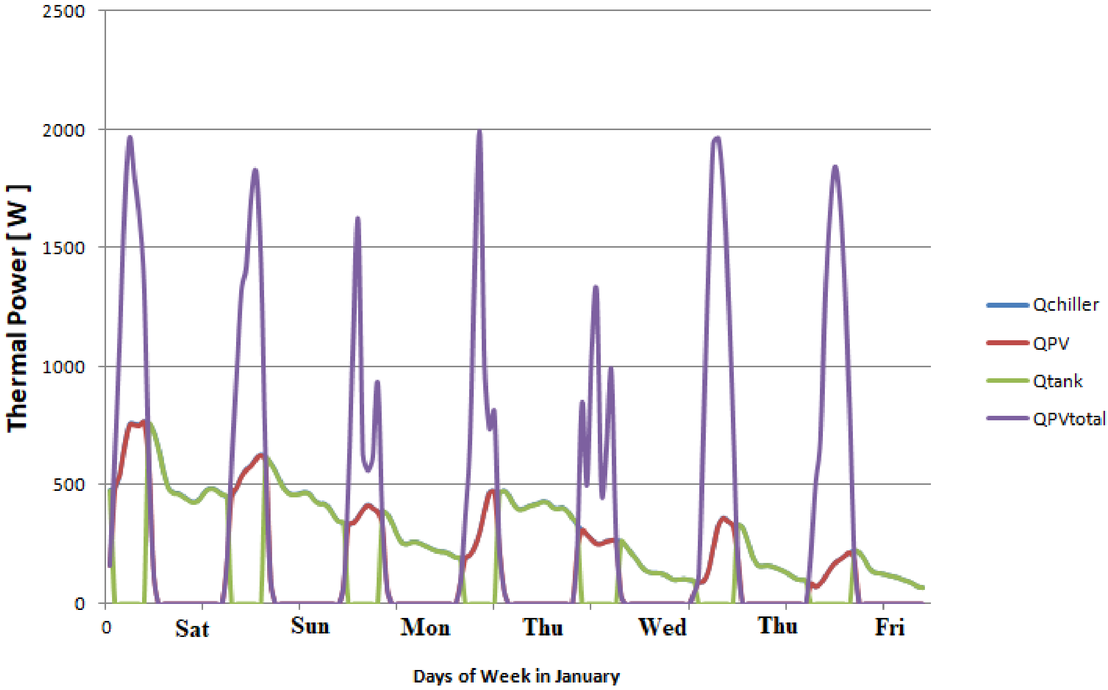

Fig.6.20. Third scenario thermal powers in winter week.

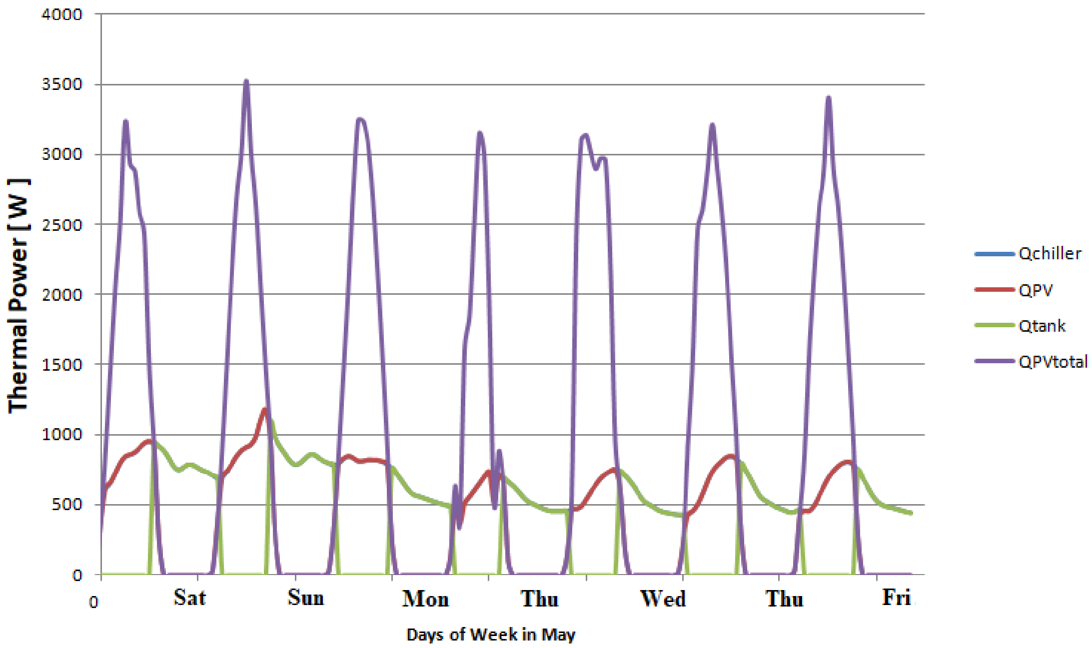

Fig.6.21. Third scenario thermal powers in spring week.

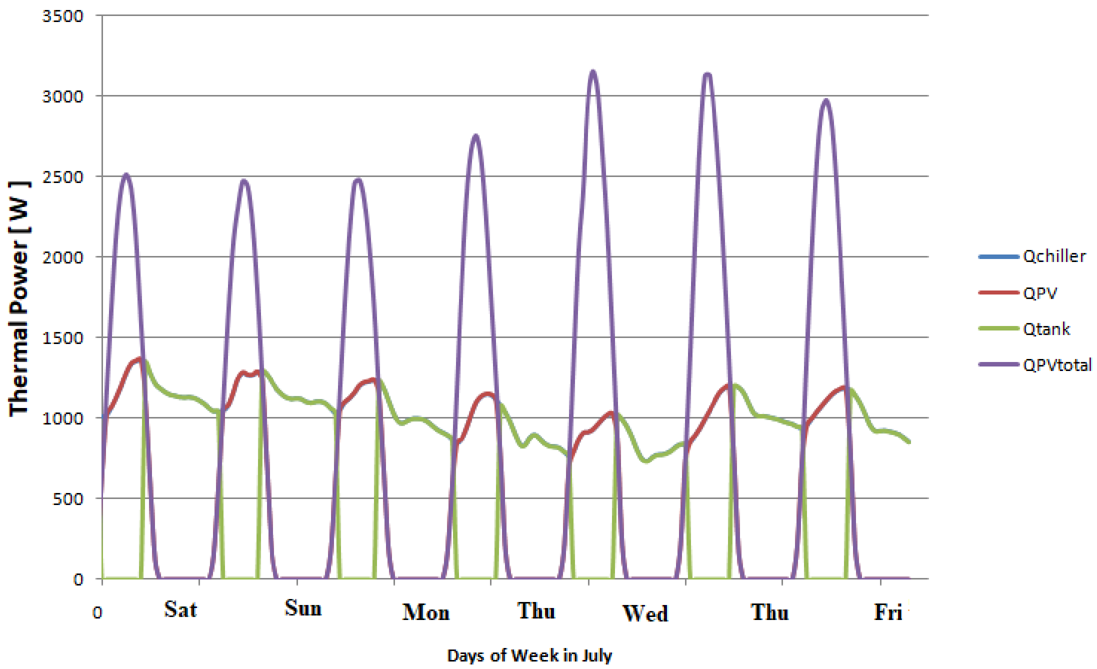

Fig.6.22. Third scenario thermal powers in summer week.

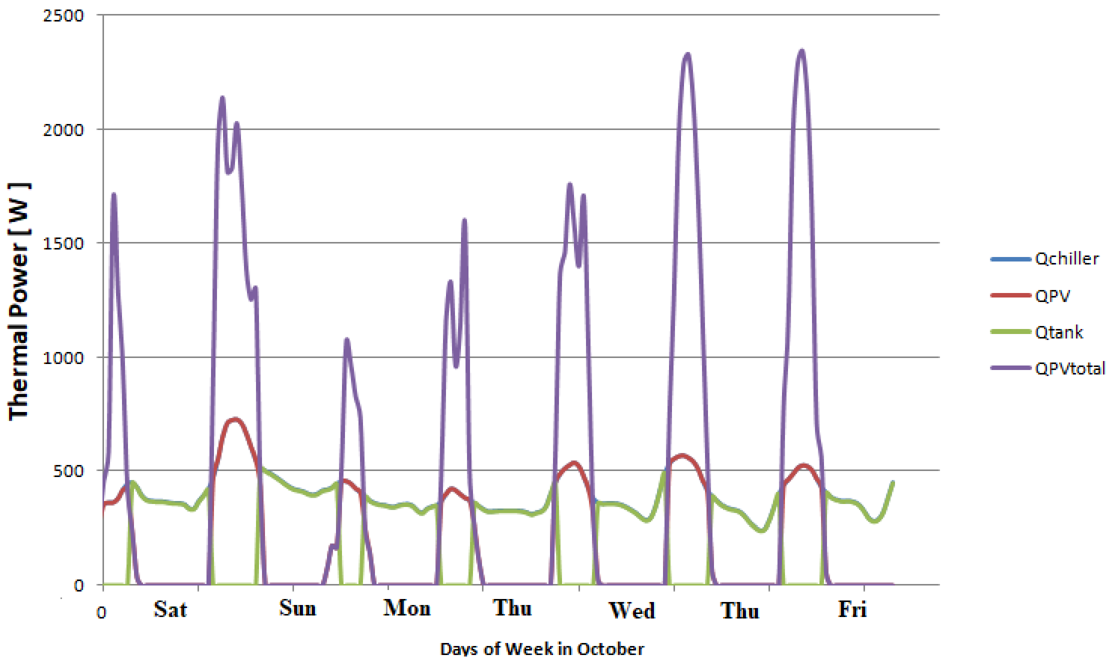

Fig.6.23. Third scenario thermal powers in autumn week.

As shown in figures 6.20 to 6.23, the purple profile represents the total thermal power produced by the PV, the red profile represents the thermal power direct used from the PV array at on-peak periods, the green profile represents thermal power obtained from storage tank at off-peak periods and the blue profile represents the chiller thermal power. The chiller profile doesn't appear in the figures this because the chiller load is fully covered using PV array and thermal storage system.

The chiller load is fully covered in all seasons. In winter 41% direct from PVs and 59% from storage tank, in spring 52% direct from PVs and 48% from storage tank, in summer 46% direct from PVs and 54% from storage tank, in autumn 43% from PVs and 57% from storage tank.

6.4.4. Third Scenario Results

This scenario used larger number of PV modules and larger storage tank volume in order to cover the total needed load for the chiller, as shown in figure 6.24, the total needed load expressed using blue profile but it doesn't appear in the figure due to the fully coverage using PV array and storage system, and the total thermal power that covered using PV and storage tank expressed in red profile. The red profile integrated using excel file to calculate the total annual thermal power that direct used from the PV array and storage tank, which equal to 4595.4 kWh/year, this value represent 100% (47% PV and 53% Storage Tank) of the total annual thermal power (4595.4 kWh/year) that needed to run the chiller along the year as illustrate in figure 6.24. The thermal losses in storage tank equal to 1060 kWh/year.

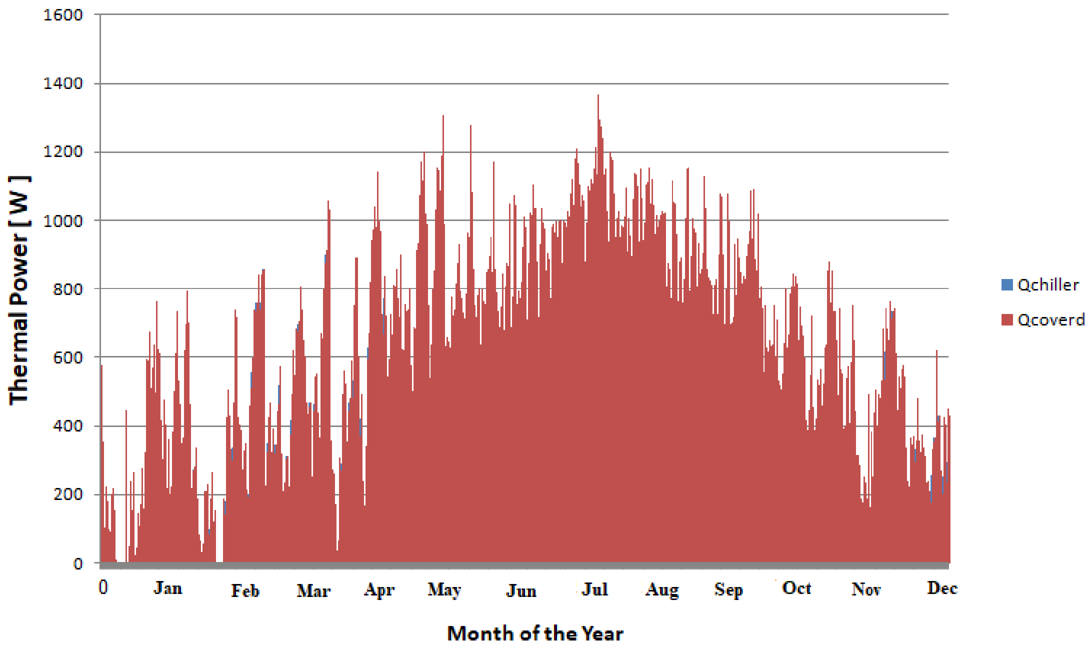

Fig.6.24. Total annual thermal power for the chiller and total annual thermal power covered by the PV and storage tank.

According to the third scenario results', the value of the electrical power that reduced in this scenario equal to 2107 kWh/year, this power obtained by using PV array and storage tank which represent 100% of the total electrical power needed to run the chiller (2107 kWh/year).

CHAPTER 7
ECONOMICAL AND ENVIRONMENTAL EVALUATION

7.1. Introduction

The economic feasibility of investments in PV systems with thermal storage tank conducted in this study for the different three scenarios. The selected indicators for this kind of assessment is the payback period (PBP) and total profit. On the other hand, the environmental advantage in comparison to traditional sources of energy is evaluated through the reduction of carbon dioxide emissions. The project life period is taken as 25 years according to the PV module performance guarantee.

7.2. Economical Evaluation

Payback period describes how long it takes the project to recover its initial cost and the total profit is the amount of dollars that decreased after the payback period, it can be calculated by knowing the annual revenue in the years after the payback period. This values can be estimated by calculate the total capital cost and total annual cost for all scenarios using the following equations.

$$\mathrm{CRF(i\%, n\ year)} = \frac{\mathrm{i(i+1)^n}}{\mathrm{(i+1)^n - 1}} \quad \text{7.1}$$

$$\mathrm{Anuul\ cost} = \mathrm{Initial\ Cost} \times \mathrm{CRF} \quad \text{7.2}$$

$$\mathrm{Total\ Annual\ Cost} = \mathrm{Anuul\ cost} + \mathrm{Operating\ Cost} + \mathrm{Maintenance\ Cost} \quad \text{7.3}$$

$$\mathrm{Total\ Capital\ Cost} = \frac{\mathrm{Total\ Annual\ Cost}}{\mathrm{CRF}} \quad \text{7.4}$$

$$\mathrm{Total\ Annual\ Saving} = \mathrm{E_{Annual}} \times \mathrm{T_r} \quad \text{7.5}$$

$$\mathrm{Payback\ Time} = \frac{\mathrm{Total\ Capital\ Cost}}{\mathrm{Anuual\ Saving}} \quad \text{7.6}$$

$$\mathrm{Total\ Profit} = \mathrm{Annual\ Revenue} \times (\mathrm{n} - \mathrm{Payback\ Time}) \quad \text{7.7}$$

Where :

CRF : The Constant Rate Factor is the default quantity CRF = 0.071 CRF/year .

n : Project life Period (25 years) .

i: Loan interest taken (5%), in this study it represents the annual depreciation of the system during the life of the project.

T_r: Tariff price in Hebron City (0.58 NIS [37]) .

E_{Annual} : Total electrical energy obtained using each scenario.

E_{Annual} equal 1,702.5 kWh/year for the first scenario (81% of the chiller electrical energy), 1,593 kWh/year for the second scenario (75.6% of the chiller electrical energy) and 2,107 kWh/year for the third scenario which represent 100% of the total energy needed to run the chiller.

7.2.1. Initial Cost

This cost contain the total cost that needed to operate the system at the beginning of the project life period such as PV modules cost (205 $/module [35]), inverter cost (752 $ [36]), storage tank cost (700 $ for HF1500, 900 $ for HF2000 [34]) and system installation cost. Installation cost found range from 0.064 to 0.1 $/Wp [38] (in this study taken 0.1), so it is equal 130 $ for the first and second scenario (4 modules x 325 Wp) and 195 $ for the third one (6 modules x 325 Wp). Table 7.1 shows the needed initial cost for all scenarios.

Table 7.1: Scenarios Initial Costs

Scenario Initial Cost	First Scenario	Second Scenario	Third Scenario
Modules Cost ($)	820	820	1230
Inverter Cost ($)	752	752	752
Storage Tank Cost ($)	0	700	900
Installation Cost ($)	130	130	195
Total initial Cost ($)	1702	2402	3077

7.2.2. Total Annual Cost

This cost contains the operation and maintenance cost which represent the annual money that needed to operate and maintain the suggested system (including PVs, Inverter and Storage Tank), according to the operating cost 1.5% [39] added to the annual cost that calculated using equation 7.2, and 0.04 $/Wp for system maintenance (52 $ for the first and second scenario and 78 $ for the third scenario). Then the total Annual cost calculated using equation 7.3.

7.2.3. Economic Assessment for The Three Scenarios

Using equation 7.1 to 7.7 in excel file sheet to calculate the needed payback period and total profit for each scenario as shown in table 7.2.

Table 7.2: Economic calculations for the study scenarios

	First Scenario	Second Scenario	Third Scenario
E annual (kWh/year)	1702.5	1593	2107
Initial Cos ($)	1702	2402	3077
CRF /year	0.071	0.071	0.071
Annual Cost ($)	120.8	170.5	218.5
Operating Cost ($)	1.8	2.6	3.3
Maintenance Cost ($)	52	52	78
Total Annual Cost ($)	175	225	300
Total Capital Cost ($)	2460	3170	4222
Total Annual Saving ($)	272	255	337
Payback Period /year	9	12.4	12.5
Total Profit ($)	4350	3202	4206

As illustrates in the table 7.2 the payback periods equal to (9 , 12,4 and 12.5 years) for the first, second and third scenario respectively. Furthermore the total profit values for the three scenarios equal to (4,350, 3202, and 4206 $) respectively. According to the economical assessment for the three scenarios the most economical scenario in this study is the first one, which is on-grid PV system without thermal storage tank.

7.3. Environmental Impact

Using renewable energy sources allows the reduction of environmental pollution. In order to evaluate the environmental advantages in this study, a comparison between CO_2 emissions released by the study scenarios and the ones released by using grid electricity produced by fossil fuels. The quantified emissions released by a PV system equal to 81 gCO_2/kWh, of which 93.7% are caused by PV panel manufacture [40]. On the other hand, fossil sources emissions were quantified in 771 gCO_2/kWh [41].

In this research, running the chiller using grid electricity only produced 1,624.5 kgCO_2/year. The first scenario used 1702.5 kWh/year from the PV-system and 404.5 kWh/year from grid electricity, so the CO_2 emissions equal to 450 kgCO_2/year. The second scenario used 1,593 kWh/year from PV system and 514 kWh/year from grid, so the CO_2 emissions equal to 525.3 kgCO_2/year. The third scenario used 2,107 kWh/year only from PV-system, so the CO_2 emissions equal to 170.7 kgCO_2/year.

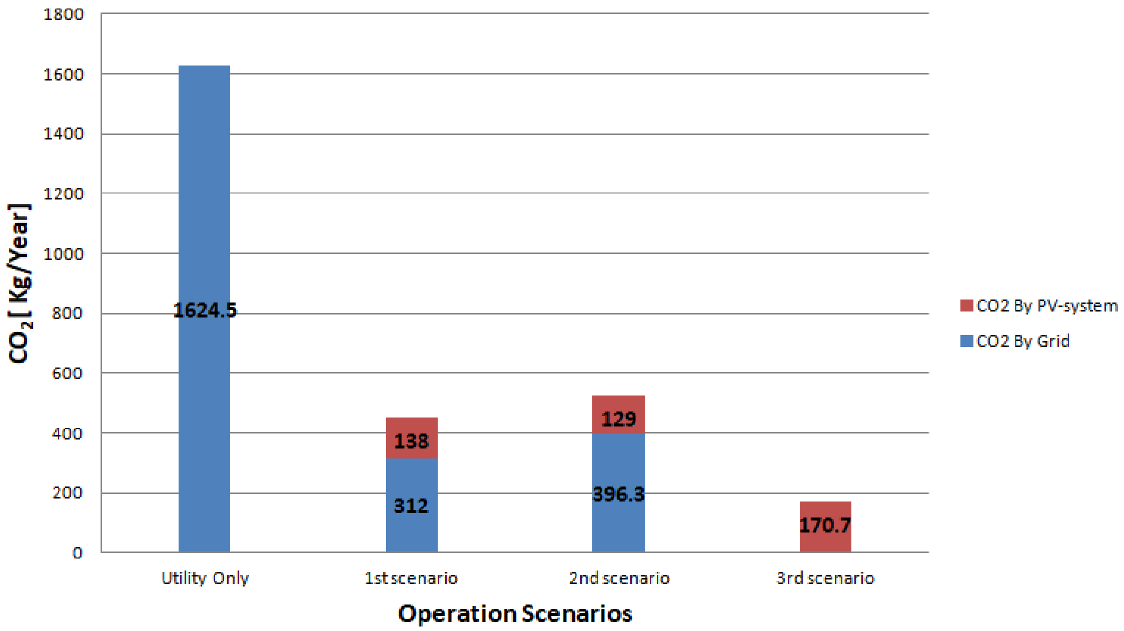

Fig.7.1. The amount of CO_2 emissions for the system scenarios.

Based on the CO_2 emissions for the system scenarios that shown in figure 7.1. The CO_2 emissions reduced by 72% for the first scenario compared with supplying the chiller from the grid only. In the same manner, the amount of CO_2 reduction for the second and third scenario equal to 68% and 89.5% respectively. According to these results the third scenario has the lowest impact on the environment.

CHAPTER 8
CONCLUSIONS AND FUTURE WORK

8.1. Conclusions

Investigating the using of PV-system with thermal storage tank to power an air cooled chiller shows a very effective results in reducing the electrical consumption from the utility, as well as reduction in CO_2 emissions and it enhance the payback period and the system profit.

Powering the chiller with the grid and PV-system saves 81% of the electrical power, compared with supplying it from the grid only. This result in a payback period of 9 years and 4,350$ profit from the system, in addition reducing in CO_2 emissions by 72%. By adopting thermal storage tank of 1.5 m^3 with the utility and PV-system the electrical consumption decreased by 75.6% and CO_2 emissions by 68%, also the payback period becomes 12.4 years, and the total profit equals to 3,202$. In order to fully supply the system from the PV-system and thermal storage tank without the need of the grid, the PV-system size is increased by adding another two PV modules and extended the volume of the storage tank to 2 m^3. This reduce the CO_2 emissions by 89.5% and the payback period becomes 12.5 years and 4,206$ total profit.

According to the economical and environmental study, the on-grid system is most economical scenario. On the other hand, the fully coverage scenario using PV-system and thermal storage tank has the lowest impact on the environment. The main findings of this work could be applied for larger systems and systems in remote areas, for reducing the electrical bills and the amount of CO_2 emissions. Furthermore, thermal storage units usage avoid the scheduled maintenance of the PV-system with electrical batteries.

8.2. Future Work

This study can be extended by immediate use of the thermal losses in storage tank for other cooling uses and calculate the economical and environmental benefits that extracted using this losses. Moreover, this research can be extended to analyze and compare between the PV-system with thermal storage tank scenario and the solar PV-system with electrical storage batteries scenario, in terms of energy reduction and economic assessments. Also, another study can be done by calculating the benefits of use the PV-system with thermal storage tank for other reference locations especially remote areas. Furthermore, analysis the suggested scenarios regarding the tilt angle of the PV-system modules since the most of the cooling load occurred during the summer season. Moreover, analyzing the system experimentally for the suggested scenarios in this work in order to validate the results.

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APPENDICES

Appendix A : Chiller Data Sheet

Appendix B : PV Module Data Sheet

Appendix C : Inverter Data Sheet

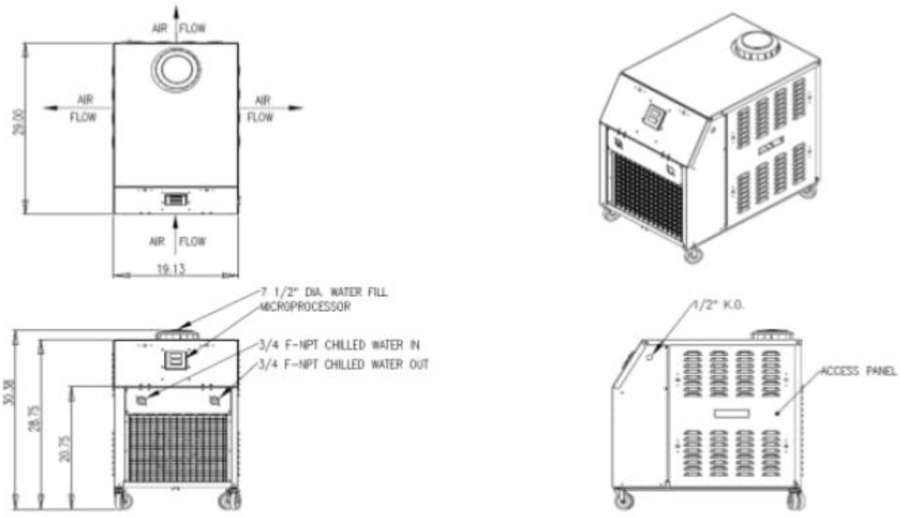
Appendix D : Thermal Storage Tank Data Sheet

Appendix (A): Chiller Data Sheet [29]

CHILLX WATER CHILLERS

MODEL
CXF050DRS

DIMENSIONAL DRAWINGS



LENGTH	WIDTH	HEIGHT	FLUID CONNECTIONS	TANK CAPACITY	WEIGHT
29"	19.25"	30.5"	3/4 F-NPT	2.5 GAL	125 LBS

PRODUCT PERFORMANCE INFORMATION

REFRIGERANT	COMPRESSOR OPER. TEMP. SUCTION RANGE	LFT	80°F AMB		90°F AMB		95°F AMB		100°F AMB		105°F AMB	
		°F	BTUH	EER	BTUH	EER	BTUH	EER	BTUH	EER	BTUH	EER
R134A	5°F to 50°F	15	2761	4.3	2495	3.8	2371	3.6	2240	3.4	2108	3.1
		20	3275	4.9	2967	4.3	2808	4.0	2655	3.8	2513	3.5
		30	4402	5.9	3990	5.2	3795	4.9	3614	4.6	3418	4.3
		40	5650	6.8	5163	6.0	4922	5.7	4681	5.3	4441	5.0
		50	6995	7.6	6419	6.7	6098	6.3	5812	5.9	5558	5.5
		60	8499	8.4	7839	7.4	7501	7.0	7173	6.5	6850	6.1

PO Box 66032 Austin, TX 7876 | 800.886.1353

ChillXChillers.com

CHILLX WATER CHILLERS

MODEL

CXF050DRS1

STANDARD FEATURES

FEATURE	BENEFIT
Listings	ETL 3046941, UL STD 1995, CAN/CSA STD, C22.2 No. 236
Microprocessor Control	Stand alone, simple, accurate control with over 10 years of field testing
Stainless Steel Brazed Plate Evaporator	Corrosion resistant evaporator wrapped in closed cell insulation
Hermetic Compressor	Industry leader, long lasting compressor with motor contactor
Reservoir Tank	Insulated polyethylene storage tank for increased process volume
Stainless Steel Pump	Fused, non-ferrous pump for process circulation
Copper Tube/Aluminum Fin Condenser	Rust resistant and high CFM
Painted Galvanized Steel Metal Cabinet	Sturdy, corrosion resistance, professional appearance
Polyethylene Storage Tank	Closed cell insulation for increase temperature control with reliable, corrosion resistant
Insulated Process Fluid Lines	Maximum thermal efficiency
Factory Tested	Ensures the unit meets the customer's performance designed criteria and arrives in working order
Full Refrigerant Charge	Hassle free and more efficient installation
NEMA 3R Rated Electrical	Indoor/outdoor use

Customization available, contact factory for information: 800.886.1353

AVAILABLE MODEL NUMBER VOLTAGE REFERENCE

S1 (115/1/60)

PRODUCT INFORMATION

MODELS AVAILABLE	COMPRESSOR			FAN MOTOR		CHILLER PUMP		RLA EA	LRA EA	MCA	MOP
R134A	MODEL	QTY	HP	QTY	FLA EA	HP	FLA				
CXF050DRS1	T6215Z	1	1/2	1	0.81	1/12	1.75	11	45	20	25

AVAILABLE MODEL NUMBER VOLTAGE REFERENCE

S1 (115/1/60)

PRODUCT INFORMATION

MODELS AVAILABLE	COMPRESSOR			FAN MOTOR		CHILLER PUMP		RLA EA	LRA EA	MCA	MOP
R134A	MODEL	QTY	HP	QTY	FLA EA	HP	FLA				
CXF050DRS1	T6215Z	1	1/2	1	0.81	1/12	1.75	11	45	20	25

Appendix B : PV Module Data Sheet [31]

Q.PLUS L-G4.2 320-345

Q.ANTUM SOLAR MODULE

The Q.ANTUM solar module Q.PLUS L-G4.2 with power classes up to 345 Wp is the strongest module of its type on the market globally. Powered by 72 Q CELLS solar cells Q.PLUS L-G4.2 was specially designed for large solar power plants to reduce BOS costs. But there is even more to our polycrystalline modules. Only Q CELLS offers German engineering quality with our unique triple Yield Security.

LOW ELECTRICITY GENERATION COSTS
Higher yield per surface area and lower BOS costs thanks to higher power classes and an efficiency rate of up to 17.6 %.

INNOVATIVE ALL-WEATHER TECHNOLOGY
Optimal yields, whatever the weather with excellent low-light and temperature behavior.

ENDURING HIGH PERFORMANCE
Long-term yield security with Anti-PID Technology[1], Hot-Spot-Protect and Traceable Quality Tra.Q™.

LIGHT-WEIGHT QUALITY FRAME
High-tech aluminum alloy frame, certified for high snow (5400 Pa) and wind loads (2400 Pa).

A RELIABLE INVESTMENT
Inclusive 12-year product warranty and 25-year linear performance guarantee[2].

EUPD RESEARCH TOP BRAND PV MODULES EUROPE 2015

Q CELLS YIELD SECURITY
ANTI PID TECHNOLOGY (APT)
HOT-SPOT PROTECT (HSP)
TRACEABLE QUALITY (TRA.Q™)

Photon
Q CELLS Best polycrystalline solar module 2013
Q.PRO-G2 235

THE IDEAL SOLUTION FOR:

Ground-mounted solar power plants

[1] APT test conditions: Cells at -1000V against grounded, with conductive metal foil covered module surface, 25°C, 168h

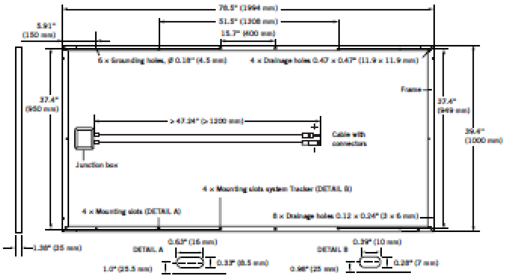
[2] See data sheet on rear for further information.

Engineered in **Germany**

Q CELLS

MECHANICAL SPECIFICATION

Format	78.5 in × 39.4 in × 1.38 in (including frame) (1994 mm × 1000 mm × 35 mm)
Weight	52.9 lb (24 kg)
Front Cover	0.13 in (3.2 mm) thermally pre-stressed glass with anti-reflection technology
Back Cover	Composite film
Frame	Anodised aluminum
Cell	6 × 12 Q.ANTUM solar cells
Junction box	3.35-4.13 in × 2.36-3.15 in × 0.59-0.67 in (85-105 mm × 60-80 mm × 15-17 mm), Protection class ≥ IP67, with bypass diodes
Cable	4 mm² Solar cable; (+) ≥ 47.24 in (1200 mm), (-) ≥ 47.24 in (1200 mm)
Connector	Amphenol H4, IP68

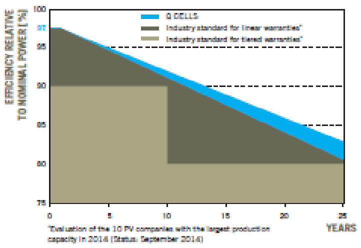


ELECTRICAL CHARACTERISTICS

POWER CLASS				320	325	330	335	340	345
MINIMUM PERFORMANCE AT STANDARD TEST CONDITIONS, STC[1] (POWER TOLERANCE +5W / -0W)									
Minimum	Power at MPP[2]	P_{MPP}	[W]	320	325	330	335	340	345
	Short Circuit Current*	I_{SC}	[A]	9.39	9.44	9.49	9.54	9.59	9.64
	Open Circuit Voltage*	V_{OC}	[V]	46.17	46.43	46.68	46.94	47.20	47.46
	Current at MPP*	I_{MPP}	[A]	8.79	8.85	8.91	8.97	9.03	9.09
	Voltage at MPP*	V_{MPP}	[V]	36.39	36.70	37.02	37.33	37.63	37.93
	Efficiency[2]	η	[%]	≥16.0	≥16.3	≥16.5	≥16.8	≥17.1	≥17.3
MINIMUM PERFORMANCE AT NORMAL OPERATING CONDITIONS, NOC[3]									
Minimum	Power at MPP[2]	P_{MPP}	[W]	237.2	241.0	244.7	248.4	252.1	255.8
	Short Circuit Current*	I_{SC}	[A]	7.57	7.61	7.65	7.69	7.73	7.77
	Open Circuit Voltage*	V_{OC}	[V]	43.08	43.32	43.56	43.81	44.05	44.29
	Current at MPP*	I_{MPP}	[A]	6.89	6.94	6.99	7.04	7.09	7.14
	Voltage at MPP*	V_{MPP}	[V]	34.44	34.72	35.01	35.29	35.56	35.83

[1] 1000 W/m², 25°C, spectrum AM 1.5G [2] Measurement tolerances STC ±3%; NOC ±5% [3] 800 W/m², NOCT, spectrum AM 1.5G * typical values, actual values may differ

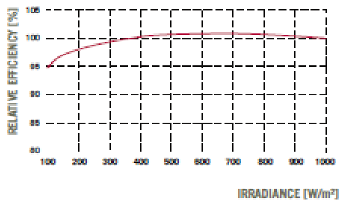
Q CELLS PERFORMANCE WARRANTY



At least 97 % of nominal power during first year. Thereafter max. 0.6 % degradation per year.
At least 92 % of nominal power after 10 years.
At least 83 % of nominal power after 25 years.

All data within measurement tolerances. Full warranties in accordance with the warranty terms of the Q CELLS sales organisation of your respective country.

PERFORMANCE AT LOW IRRADIANCE



The typical change in module efficiency at an irradiance of 200 W/m² in relation to 1000 W/m² (both at 25 °C and AM 1.5 G spectrum) is -1.5 % (relative).

TEMPERATURE COEFFICIENTS

Temperature Coefficient of I_{SC}	α	[%/K]	+0.04	Temperature Coefficient of V_{OC}	β	[%/K]	−0.29
Temperature Coefficient of P_{MPP}	γ	[%/K]	−0.40	Normal Operating Cell Temperature	NOCT	[°F]	113 ± 5.4 (45 ± 3 °C)

PROPERTIES FOR SYSTEM DESIGN

Maximum System Voltage V_{SYS}	[V]	1500 (IEC) / 1500 (UL)	Safety Class	II
Maximum Series Fuse Rating	[A DC]	15	Fire Rating	C / Type 1
Max Load (UL)[2]	[lbs/ft²]	75 (3600 Pa)	Permitted module temperature on continuous duty	-40 °F up to +185 °F (-40 °C up to +85 °C)
Load Rating (UL)[2]	[lbs/ft²]	33 (1600 Pa)	[2] see installation manual	

QUALIFICATIONS AND CERTIFICATES

IEC 61215 (Ed. 2); IEC 61730 (Ed. 1), Application class A
This data sheet complies with DIN EN 50380.

C US Certified UL 1703 (254141)

PACKAGING INFORMATION

Number of Modules per Pallet	29
Number of Pallets per 40' Container	22
Pallet Dimensions (L × W × H)	81.3 x 45.3 x 46.9 in (2065 x 1150 x 1190 mm)
Pallet Weight	1671 lbs (758 kg)

NOTE: Installation instructions must be followed. See the installation and operating manual or contact our technical service department for further information on approved installation and use of this product.

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Appendix C : Inverter Data Sheet [32]

Single-Phase String Inverters 1 kW to 3 kW

> Residential, Solar Inverters

Eversol TL Series

TL1000/1500/2000/3000

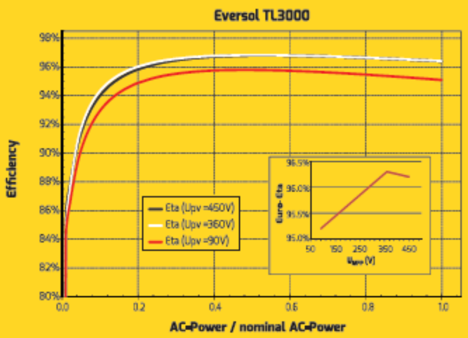
Introduction

We believe that the world would be a better place if everybody had easy access to the cleanest energy from the roof of their homes. By creating simple, easy to use, affordable and reliable inverters we are revolutionizing access to solar power and bringing energy to everybody. Ideal for residential applications, our Eversol TL single phase inverter.

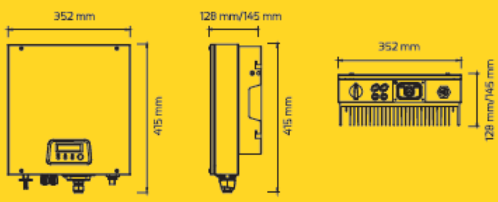
Features

- Efficiency 97%
- Maximum Powerpoint Tracking
- IP65 protection class
- RS485 communications
- Online web monitoring via ZeverCom & ZeverCom WiFi
- Grid management functions via ZeverCom & ZeverCom WiFi
- Easy handling for installation and maintenance

Conversion efficiency



Dimensions



JIANGSU ZEVERSOLAR NEW ENERGY CO., LTD.
No.198 Xiangyang Road, Suzhou 215011, China • t +86 512 6937 0998 • f +86 512 6937 3159
Sales: info@zeversolar.net

For contact information in your region please visit www.zeversolar.com

zeversolar

Single-Phase String Inverters 1 kW to 3 kW

Technical data	Eversol TL1000	Eversol TL1500	Eversol TL2000	Eversol TL3000
Input (DC)				
DC convertible power (@cosφ=1)	1150W	1750W	2250W	3350W
Max. input voltage	500V			
MPP voltage range / rated input voltage	90–450V / 360V			
Min. start voltage	80V			
Min. feed-in power	10W			
Max. input current per MPPT	12A	12A	12A	18A
Number of MPPTs	1	1	1	1
Number of independent MPP inputs	1	1	1	2
Output (AC)				
Rated active power	1000W	1500W	2000W	3000W
Max. apparent AC power	1100VA	1650VA	2140VA	3190VA
Nominal AC voltage / range	220,230,240V / 180–280V			
AC power frequency / range	50 / +-5Hz	50,60 / +-5Hz		
Rated power frequency / rated grid voltage	50Hz / 230V	50Hz / 230V		
Max. output current	5.5A	9A	11A	16A
Power factor (@rated power)	1	1	1	1
Adjustable displacement power factor	-	0.95 inductive ... 0.95 capacitive		
Feed-in phases / connection phases	1 / 1	1 / 1		
Harmonic distortion (THD) at rated output	< 3%	< 2%		
Efficiency				
Max. efficiency / European weighted efficiency	95.7% / 95%	97% / 96.5%		
MPPT efficiency	99.50%	99.50%		
Protective devices				
DC isolator	○			
PV iso / Grid monitoring	● / ●			
DC reverse polarity protection / AC short-circuit current capability	● / ●			
GFCI function	●			
Protection class (according to IEC 62103) / overvoltage category (according to IEC 60664-1)	I / II (DC). III (AC)			
General data				
Interfaces: RS485 / RS485[1] & Ethernet & WIFI & a.RJ45[2](DRED)	● / -			
Display	16 x 2 characters			
Dimensions (W x H x D)	352 x 415 x 128mm			352 x 415 x 145mm
Weight	11.5kg			14kg
Cooling concept	convection			
Noise emission (typical)	< 20 dB(A)@1m			
Installation	indoor & outdoor			
Mounting information	wall mounting bracket			
DC connection technology	SUNCLIX			
AC connection technology	plug-in			
Operating temperature range	-25°C...+60°C / -13°F...+140°F			
Relative humidity (non-condensing)	0% ... 100%			
Max. operating altitude	2000m			
Degree of protection (according to IEC 60529)	IP65			
Climatic category (according to IEC 60721-3-4)	4K4H			
Topology	transformerless			
Self-consumption (night)	< 1W			
Standby power	6W			

● standard ○ optional - not available

1) For connection to approved smart meters in zero export installations

2) Analog RJ45 interface to DRED in Australia & New Zealand

As of August, 2016 / Technical data is subject to revisions.

Appendix D : Thermal Storage Tank Data Sheet [34]

Storatherm Heat

Buffer tanks for heating & cooling*

Buffer tank for heating & cooling

H 200-5000
(Without flange and without coil)

H 300-5000/R
(With flange and without coil)

H 300-5000/1
(Without flange and with single coil)

H 500-2000/2
(Without flange and with double coil)

Buffer storage tank with jacket, without heating coil or inspection flange

Formerly, PHH

Storage Tank Type	Article No White	Article No Silver	Material Group	Ø D (mm)	Height H (mm)	Couplings 9x	Inclination Height mm	Weight kg	Heat Loss Kwh/24h	Outer Jacket Fire Classification
HF 200	8500000	8502000	63	660	1500	Rp 1 ½	1525	51.0	2.2	B2
HF 300	8500010	8502010	63	777	1320	Rp 1 ½	1355	59.0	2.8	B2
HF 500	8500020	8502020	63	777	1950	Rp 1 ½	1974	72.0	3.4	B2
HF 800	8500030	8502030	63	970	1825	Rp 1 ½	1870	124.0	4.0	B2
HF 1000	8500040	8502040	63	970	2115	Rp 1 ½	2153	139.0	4.4	B2
HF 1500	8500050	8502050	63	1180	2120	Rp 1 ½	2178	186.0	5.1	B2
HF 2000	8500060	8502060	63	1380	2122	Rp 1 ½	2200	266.0	5.9	B2