UNIVERSITY OF TURKISH AERNAUTICAL ASSOCIATION

INSTITUTE OF SCIENCE AND TECHNOLOGY

An Evaluation of Absorption Refrigeration System Using Solar Energy as a Heat Source in Iraq Environment

MASTER THESIS

ALI ISMAEL JALIL

ID: 1403730028

Institute of Science and Technology

Mechanical and Aeronautical Engineering Department

Master Thesis Program

April 2017

UNIVERSITY OF TURKISH AERNAUTICAL ASSOCIATION INSTITUTE OF SINENCE AND TECHNLOGY

An Evaluation of Absorption Refrigeration System Using Solar Energy as a Heat Source in Iraq Environment

MASTER THESIS

ALI ISMAEL JALIL

Ref.NO: 10147551

IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE

DEGREE OF MASTER OF SCIENCE IN MECHANICAL AND AERONAUTICAL ENGINEERING

Supervisor: Assist. Prof. Dr. Sudantha Balage

CO. Supervisor: Dr. Ahmed Shihab Al-Samari

ABSTRACT

An Evaluation of Absorption Refrigeration System Using Solar Energy as a Heat Source in Iraq Environment

ALI ISMAEL JALIL

Master, Department of Mechanical engineering

Thesis Supervisor: Dr. Sudantha Balage

April 2017, 89 pages

The absorption refrigeration system has a significant advantage could be used with the renewable energy resources, because of it uses heat as an important energy source to complete its refrigeration cycle. This advantage is extremely useful when using solar energy as a renewable energy resource. The main goals of this research are first, design and build a parabolic trough solar collector (PTSC) using materials that available locally in Iraq. The reason to choose building the PTSC in this way is to give an encouragement for whom interesting in this topic, to reproduce these results with minimum costs. However, the using of cheap reflecting materials may lead to low efficiency, but we have to keep in mind that the solar energy is 100% free and unused lands are very wide in Iraq also. Second goal is to simulate an absorption refrigeration system using programming package, such as Matlab to calculate the heat needs per T.R.

The research outcome shows that the PTSC efficiency was 13.75 %, which is low in comparison to the ultimate PTSC but keep in mind that the cost is about 15% of the best one. Moreover, the area of PTSC per ton of refrigeration (T.R) needs was 75 m^2/T.R. Finally, these results are showing promising alternative refrigeration systems as renewable energy resources applications.

Keywords: Renewable energy, Solar Energy and absorption Refrigeration system**.**

Irak Ülke Koşullarında Isınma Kaynağı Olarak Güneş Enerjisini Kullanan Soğurma Soğutma Sistemi Üzerine Bir Değerlendirme

YÜKSEK LİSANS TEZİ

ALI ISMAEL JALIL

KİMLİK NO: 1403730028

Danışman Dr. Sudantha Balage Co-advisor
Yardımcı Danışman Dr. Ahmed Shehab Al_samarai

2016 / 2017

ÖZET

Irak Ülke Koşullarında Isınma Kaynağı Olarak Güneş Enerjisini Kullanan Soğurma Soğutma Sistemi Üzerine Bir Değerlendirme

ALI ISMAEL JALIL

Makine Mühendisliği Bölümü Yüksek Lisans Tezi

Tez Danışmanı: Dr. Ahmed Al-Samari

Nisan 2017, 89 sayfa

Soğurma soğutma sistemi, yenilenebilir enerji kaynakları ile birlikte kullanılabilen son derece önemli bir avantaj saplayan bir sistemdir çünkü kendi soğutma döngüsünü tamamlamak amacıyla ciddi bir enerji kaynağı olan ısıyı kullanır. Bu imkân, güneş enerjisini yenilenebilir bir enerji kaynağı olarak kullanırken inanılmaz derecede kullanışlıdır. Bu araştırmanın temel hedefleri öncelikle tasarım ve Irak'ta yerel anlamda mevcut ve bulunabilen maddeleri kullanan parabolik oluk tipi güneş kollektörü inşa etmektir. Bu şekilde parabolik oluk tipi güneş kollektörü inşa etmenin seçilmesindeki sebep bu konu ile yakından ilgilenen kişileri en düşük maliyetle bu sonuçlara ulaşmaları için cesaretlendirmektir. Ancak ucuz yansıtıcı maddelerin kullanımı düşük verimliliğe sebebiyet verebilir ama yine de güneş enerjisinin %100 seviyesinde bedava ve Irak'ta kullanılmayan alanların da son derece geniş olduğu unutulmamalıdır. İkinci amaç ise T.R. başına ısıyı hesaplamak için Matlab gibi yazılım paketlerini kullanarak soğurma soğutma sistemini taklit edebilmektir.

Bu araştırmanın çıktıları göstermektedir ki; parabolik oluk tipi güneş kollektörü verimliliği %13,75 olurken ki bu seviyenin parabolik oluk tipi güneş kollektörünün en iyi seviyesine göre düşük olduğu karşılaştırmalarla ortaya konulsa bile en iyisinin dahi maliyetinin yaklaşık %15 seviyesinde olduğu unutulmamalıdır. Dahası, ton başına soğutma(T.R.) ihtiyaçlarının parabolik oluk tipi güneş kollektörü alanına göre ihtiyaç durumu ise 75 m2/T.R. olarak gerçekleşmiştir. Sonuç olarak, bu sonuçlar göstermektedir ki; alternatif soğutma sistemleri yenilenebilir enerji kaynakları uygulamaları açısından umut verici bir görüntü sergilemektedir.

Anahtar Kelimeler: Yenilenebilir enerji, Güneş enerjisi ve soğurma soğutma sistemi

بسم الله الرحمن الرحيم

اللَّهُ نُورُ السَّمَاوَاتِ وَالْأَرْضِ مَثَلُ نُورِهِ كَمِشْكَاةٍ فِيهَا مِصْبَاحٌ الْمِصْبَاحُ فِي زُجَاجَةٍ الزُّجَاجَةُ كَأَنَّهَا كَوْكَبٌ دُرِّيٌّ يُوقَدُ مِن شَجَرَةٍ مُّبَارَكَةٍ زَيْتُونِةٍ لَّا شَرْقِيَّةٍ وَلَا غَرْبِيَّةٍ يَكَادُ زَيْتُهَا يُضِيءُ وَلَوْ لَمْ تَمْسَسْهُ نَارٌ نُّورٌ عَلَى نُورٍ يَهْدِي اللَّهُ لِنُورِهِ مَن يَشَاءُ وَيَضْرِبُ اللَّهُ الْأَمْثَالَ لِلنَّاسِ وَاللَّهُ بِكُلِّ شَيْءٍ عَلِيمٌ

صَدَقَ اللّٰهُ العَظِيم

سورة النور–الآية ٣٦

Türk Hava Kurumu Üniversitesi Fen Bilimleri Enstitüsü'nün 1403730028 numaralı Yüksek Lisans öğrencisi "Hussein AL-Gburi" ilgili yönetmeliklerin belirlediği gerekli tüm şartları yerine getirdikten sonra hazırladığı "An Evaluation of Absorption Refrigeration System Using Solar Energy as a Heat Source in Iraq Environment" başlıklı tezini, aşağıda imzaları bulunan jüri önünde başarı ile sunmuştur.

Tez Danışmanı **:Yrd. Doç. Dr. Sudantha Balage**
Türk Hava Kurumu Üniversitesi ....

Jüri Üyeleri

: Yrd. Doç. Dr. Mohamed S. Elmnefi ....
Türk Hava Kurumu Üniversitesi

: Yrd. Doç. Dr. Munir Elfarra ....
Ankara Yıldırım Beyazit Üniversitesi

:Yrd. Doç. Dr. Sudantha Balage
Türk Hava Kurumu Üniversitesi ....

Tez Savunma Tarihi: 21.04.2017

STATEMENT OF NON-PLAGIARISM PAGE

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

Alı Ismael Jalıl

21.04.2017

Acknowledgment

First praise be to my Allah Who has inspired me to pass all the difficulties that I have got in my work.

I am so grateful to my supervisor Dr. Sudantha Balage, and my co-adviser Dr. Ahmed Shihab Al-Samari for his advice, wisdom and his guidance for improving my work.

When I think to thank the people who were so close by their support, the first two who have helped me to pass all my difficulties easily, are my dear wife (Noora Sh. Ahmed), and my dearest son (Nooraldeen).

Last, but not least my thanks go to all people who have supported this research directly or indirectly.

ALI ISMAEL JALIL

Table of Contents

CHAPTER THREE
METHODOLOGY

CHAPTER FOUR
EXPERIMENTAL WORK

CHAPTER FIVE
RESULTS AND DISCUSSION

CHAPTER SIX
CONCLUSIONS AND RECOMMENDATIONS

Terminology

Symbol	Dimensions and physical quantities	Units
A	Area	m^2
m	Mass of the water	Kg
T	Temperature	C°
t	Time	s
h	Enthalpy	Kj/kg
Q	Amount of heat	Kj
q	Amount of heat generated	Kw
q_h	Amount of heat gain from sun per one meter	w/ m^2
Ib	Intensity radiation	w/ m^2
ŋ	Collector efficiency	
P	Pressure	Kp_a
x	Solution concentration	%
q_e	Amount of heat for the ton evaporator	Ton
q_c	Amount of heat rejected on condenser	Ton
q_a	Amount of heat rejected to absorbent	Ton
q_{gen}	Amount of heat added to generator	Ton

q_{add}	Total Amount of heat added to the cycle	Ton
q_{rej}	Total Amount of heat rejected from the cycle	Ton
PTSC	Parabolic Trough Solar Collector	Ton

List of Figures

List of Tables

CHAPTER ONE
INTRODUCTION

1. Introduction

There are a lot of countries with a significant need of food preservation, and climate conditioning. It would be valuable to have simple solar refrigerators working independent of an electrical energy source. Although the solar powered refrigeration using cells store is a valuable, they are at a higher- cost level. The solar radiation distribution has a significant value in Iraq climate, and the statistics give us the estimation about beam solar radiation distributions contained as (5-6), (6-6.5), (6.5-7) Kw.hr/m2 from north, middle to south respectively [1]. Above facts give us the possibility of converting this energy for several advantages for examples: heating, cooling, food preservation, furnaces, drying, desalination….. atc.

There are two types of refrigeration systems. The first one combines the solar energy and the refrigeration system. The second separates the solar power unit from the refrigeration unit. This thesis focuses on the second type. The first part is the solar power unit the basic concepts of the solar power unit are either focusing collectors or flat plate collectors [2]. Second part, which uses energy obtained by the first part, is the absorption refrigeration system.

There are two ways to operate an absorption refrigeration system; intermittent or continuous operation. The intermittent absorption has two main operations, one of them is regeneration and the other is refrigeration. To complete the refrigeration absorption cycle is to motivate the refrigerant and absorbent mixing concentration to separate, then the refrigerant become liquid from vapor phase in the condenser rejecting the latent heat. All this process is called regeneration operation. In continuous operation method, the refrigerant is vaporized directly without the use of a regenerated cycle [2].

The Absorption cooling systems are methods based on the amount of heat that is absorbed from the collector enabling the regenerator fluid (Li-Br-water) to separate the strong concentration flow into two streams. The first is a superheated vapor and go through the condenser to reject the amount of heat that the refrigeration fluid gained from the generator, then the above heat is rejected to the ambient.

The liquid moves from the high pressure components (generator, condenser) through the expansion valve to low pressure components, and enters the third parts of refrigeration cycle that is called an evaporator.

The evaporator is the part in the refrigeration cycle that has the ability to remove an amount of heat from the place where we want to condition it's environment, after that, the refrigeration fluid leaves the evaporator as a saturated vapor to enter the forth part of the cycle – absorbent. There the refrigeration fluid reacts with the weak concentration (Li-Br) to become the strong concentration to complete the cycle while motivating the refrigeration cycle. [3,4]

One of the first studies that used Li-Br-water in absorption refrigeration system was conducted in Spain by Rosiek and al. in 2012. [4], their project reported estimation of power, maximum COP, solar absorption cooling plant to be 70KW, 0.6, 35KW respectively .After that Ali and al. improved the performance until they achieved the efficiency approximately value 49.3 and COP until 0.81. [4]. Work by Hammad and Zurigat.[5] reported estimation of 1.5 ton solar cooling unit with the collector area of 14 m2. It employed five shells and tubes heat exchanger and the experiment was performed in April and May in Jordan, the maximum cop reported was 0.85.

G.Cascales and al. in 2016. [6] studied Li-Br-water closed system that was successfully applied to a classroom condition in Murcia, Puerto Lambrera. At Hong Kong University Yeung et.al.[7] designed and built a solar collector and found the values of collector power, absorption cooling power, and efficiency to be 1.33T.R, 37.55%respectively for a collection area of 38.3 m2. Their annual system had the efficiency value of 7.8%.

The evaluation of that project was made by Rosiek, and it was made with MATLAB code to simulation between the solar energy and the absorption refrigeration system. Simulation by using MATLAB code to calculate the total heat needed per each ton of refrigeration (kW heat/ T.R). This study includes an estimation of cooling load for small building and then calculate the area of the parabolic solar collector that needed to guarantee the amount of heat energy. The main idea is using solar energy in evaluations of absorption refrigeration systems benefits.

The major topic of this thesis is heat transfer (solar energy) and compression refrigeration systems (absorption refrigeration systems). The experiment was performed in Dyala Iraq. A MATLAB code is developed to facilitate the coputations.

1.2 The Main Goal of the Thesis

This study reports on an estimation of a cooling load for a small building and then proceeds to calculate the area of the parabolic solar collector that is needed to guarantee the amount of required energy for air conditioning. The parabolic solar collector was designed and built to perform the experiments to obtain the solar heating. A computer program, written in MATLAB, was developed to simulate the absorption refrigeration cycle.

The main idea is to demonstrate the viability of the absorption refrigeration in inexpensive, simple solar refrigeration systems that is accessible to rural or remote populations.

CHAPTER TWO

LITERATURE SURVEY

2.1 Introduction

The contemporary trends of energy focus on permanent renewable and alternative unlimited sources which have less pollution and avoids probable drain of other resources. The solar energy source has taken up priority over the other sources because of its ability to achieve human needs of energy. Moreover, it has a large active contribution in future since it has no pollution harm or damage and it is able to transform into other kinds of energy such as: electrical, mechanical and thermal energy. Recently, the researches of energy of solar system has witnessed rapid expansion in applications in most world regions including the Arab countries, concentrating on ways which leads to increased and developed its efficiency of solar system applications. There is no doubt that the regions that has suitable moderate climate with high temperature, have got many opportunities and ability to obtain the maximum energy of solar system with high efficiency. Iraq is one of that countries, it can receive solar rays with approximate time of 4000 hours per year in convenient places of solar energy. Two basic choices exist when we deal with solar energy, the first is photovoltaic, and it is a direct solar energy conversion to electrical energy. The solar thermal systems are second choice, in which heat to a thermodynamic system is provided by the solar radiation, which results in the generating a mechanical energy can be transformed into electricity. The efficiencies of photovoltaic systems are in the region of 10 to 20 percent, while it is possible to achieve efficiencies as high as 30 percent in solar thermal system [8]. In the current study we have worked on thermal utilization of solar intensity. One or more solar collectors form an atypical thermal solar utilization system; they are connected to a storage and distribution system. A solar collector is device that utilizes the solar radiation to heat a fluid, which can then be used for suitable applications [9]. Basically, thermal solar collectors have three types: concentrators, flat-plate and evacuated tube.

2.2 Solar Thermal Collection - Concentrating Solar Power

The solar thermal collection main principle is as the solar radiation is falling on surfaces (such as of black – bodies) part of this falling radiation is absorbed increasing the surface temperature. As the temperature of the body increases, the energy from the surface will be transformed to the surroundings at an increasing rate. The steady – state is achieved when the rate of solar heat gained is balanced to the rate of heat loss to the surroundings [10]. The amount of incident energy intensity on the absorbing surface is increased by the solar concentrators as compared to amount of incident energy on the concentrator aperture. Achieving this increase is done by the use of any optical means such as reflecting surfaces which focuses the incident radiation onto a proper receiver / absorber. Therefore, generally, the main solar concentrator components are (i) a focusing device, (ii) a receiver / absorber provided without or with a transparent cover, and (iii) a tracking device to continuously follow the sun. Concentrating Solar Power (CSP) technologies are usually categorized in three different concepts, as shown in figure (2-1). They work as follows [11]:

- Towers: a big number of heliostats are used to make the sun's rays focus on to a center receiver placed atop a tower.
- Troughs: parabolic trough – shaped mirror reflectors concentrates sunlight onto receiver tubes linearly. The transfer fluid within the receiver tube absorbs the heat and transport it to the required system.
-
- Dishes: parabolic dish – shaped reflectors focuses sunlight to a point where it will be converted into another form of energy or transported as thermal energy to another system via a transfer fluid.
- Typically, the obtained ratio of solar flux concentration is at the level of 30 – 100 for trough, 100 – 1000 for tower, and 1000 – 10000 for dish system [12].

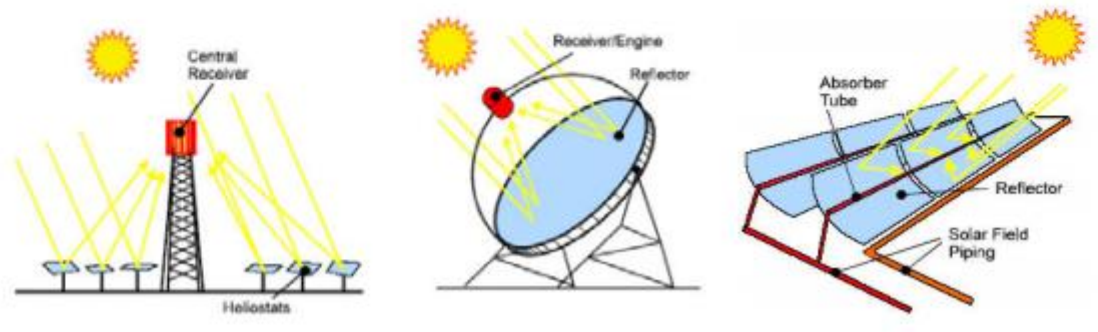


Fig (2-1) Schematic diagrams of the three CSP systems (Tower, Dish, and Trough) [12]

2.3 Solar Potential in Iraq

The available solar activities for investment in Iraq depend on many factors such as:

2.3.1 The Intensity of Solar Radiation

In term of solar radiation exposure Iraq ranks second behind Saudi Arabia in the world. The solar radiation exposure of Iraq is as shown in figure (2-2). The daily averaged solar insolation contour map of Iraq as shown in figure (2-3) have established that almost all of Iraq has the potential areas for establishing large – scale solar utilities. In Iraq, the annual average of beam solar energy is between 4.5 – 5.4 kWh/m^2. Figure (2-3) shows the daily average for solar exposure in Iraq which is high degree thus; when it comes to solar applications, Iraq is among the most suitable countries [13].

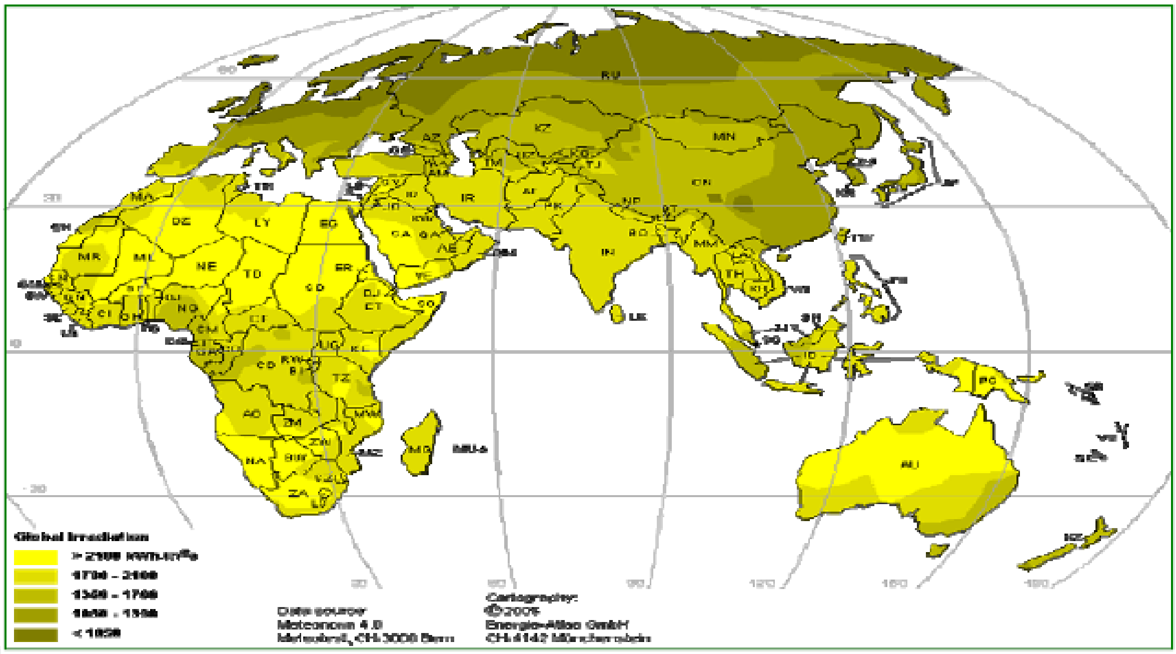


Fig (2-2) Classification of solar radiation exposure [14]

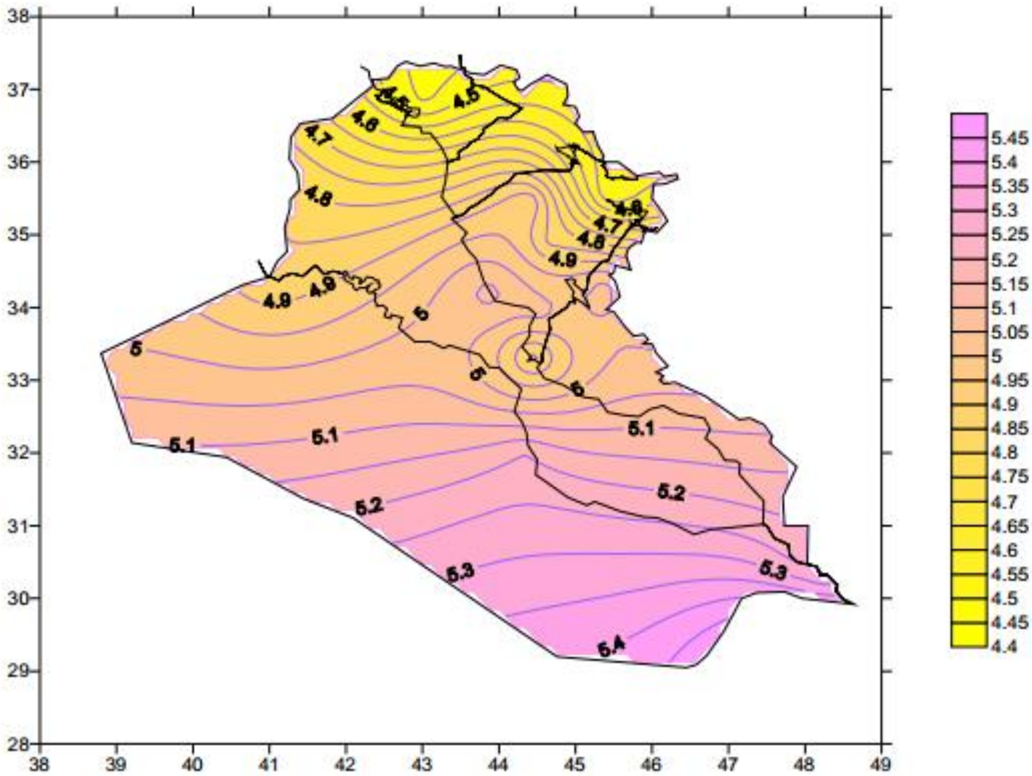


Fig (2-3) Daily-averaged solar insolation for different locations in Iraq [13].

2-3-2 Solar Energy Exposure

Solar energy exposure period in Iraq is available for a long period as shown below:-

Table (2-1) solar exposure period in Iraq [13]

Quantity.	*Solar Exposure period*
Sunny hours	4100 hours
Sunny days	333.6 days
cloudy days	31.4 days

2-3-3 Solar Energy Applications

Technically, Iraq's weather is convenient for all solar applications but from the economic aspect there are some factors that must be taken in to consideration while choosing the application. First, the Iraqi environment has particular weather conditions such as dust spread for a long period each year. The thermal solar energy utilization is less affected in accordance to this type of environment than the utilization of solar energy in direct generation of electricity by using photovoltaic (PV) technology. Because the PV absorb the visible part and little from NIR part of solar spectrum (range 400-1100 nm), the visible part intensity is influenced obviously by dust atmosphere while the thermal system works mainly on long wave-length infrared of spectrum which is less affected by dust. Second, the Iraqi family's needs of energy is distributed between (65-70%) for heating and cooling equipment per a year and about 30% for lighting and other electrical needs, so there is a need for thermal system that has efficiency of conversion the solar energy to thermal of about (60-70%). Third, the cost of electrical production of PV system is about 20 Cent/kWh while it costs 10 Cent/kWh with thermal systems. Fourth, the storage of energy of thermal system is better in quantity, cost and age than PV system.

2.4 Fundamentals of Solar Radiation -Solar Geometry.

2.4.1 Sun Earth Geometrical Relationship

The earth orbits around the sun every 365.25 days on an elliptical path that falls on the equatorial plane, and it completes a full rotation around its axis every 24 hours. The earth – sun distance is the smallest on December 21 (perihelion, 1.47 x 10^{11} m) and the largest on June 21 (aphelion, 1.52 x 10^{11} m) [15]. The tilt of the rotation axis of the earth is at an angle of 23.45° to its orbital plane, as it can be observed below. (2.4). [16].

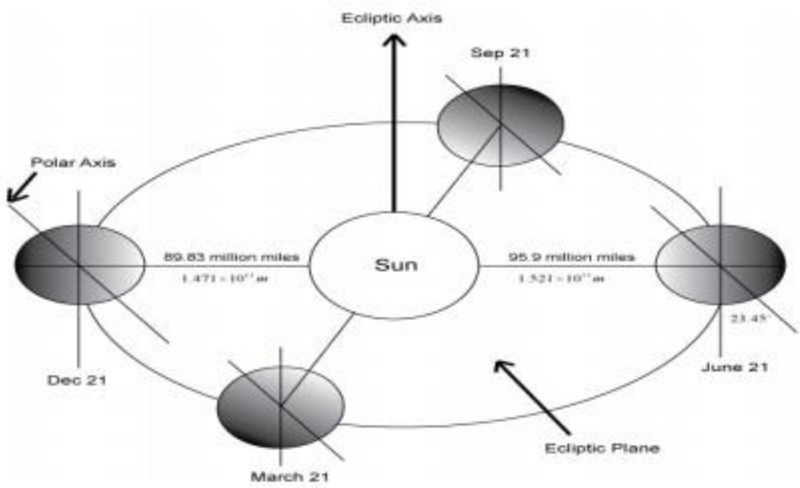


Fig (2-4) Motion of the earth about the sun [17]

2.4.2 Basic Earth – Sun Angles

A general point p on the surface of the earth can be coordinatized with two angles as shown in the figure (2-5), the angles are defined relative to the suns rays.

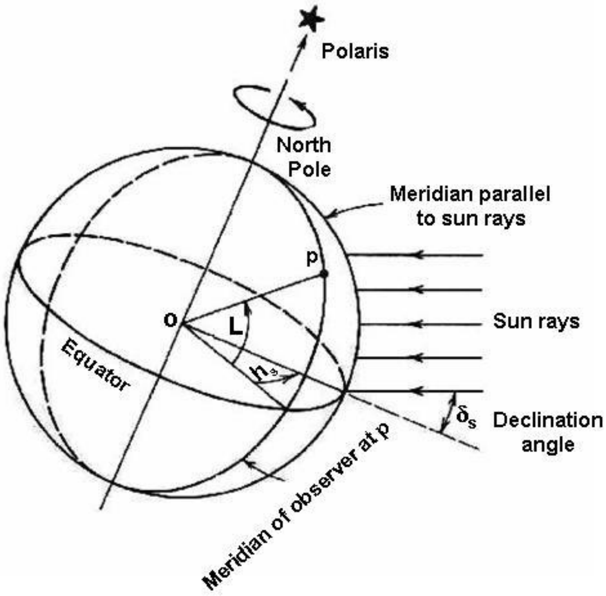


Fig (2-5) Basic earth – sun angles [18].

The **Latitude angle** (L): angular distance of the point p measured from the equator, the latitude angle L is the angle between the radius vector op and the projection of it on the equatorial plane.

The **Hour angle** (hs): is the angle between the projection of the line that joins the sentence of the earth and the sun on to the equatorial plane and the projection of op on to the equatorial plane.. The hour angle is:

$h_s = 15(ST - 12)$……………… (2-1)

The rotation of the earth $h_{s,}$ where ST represents the solar time in hours, differs at the rate of 15 per hour, and $h_s = 0$ at solar noon [19]. The conversion between the local time and the solar time necessitates obtaining the local standard meridian, the day of the year, and the location (longitude) as in the following equation [19]:

$ST = LST + 4(L_{st} - L_{loc}) + ET$……………… (2-2)

Where LST is the local standard time, L_{st} is the standard meridian for local time zone (45^o for Baghdad), L_{loc} is the longitude for the position (44^o for Baghdad), and ET is the equivalence for the time in minutes and equal to:

ET = 9. 87 Sin (2B)– 7.35 Cos (B) – 1.5 Sin (B)…………….. (2.3)

Where B, in degree is defined as:

$B = 360(d_n - 81)/364$…………………….. (2.4)

Where d_n represents the number of the day throughout the year (1≤ d_n≤ 365). The values of the angle representing the hour are negative if east due south (morning), and positive if west due south(afternoon). [19].

The **Sun's declination angle** (δ_s) is the angular distance of a sun's ray north (or south) of the equator. It illustrates the angle between a line that spreads from the center of the earth to the center of the sun and that line's projection on the equatorial plane of the earth. The angle of declination (δ_s) is calculated with the help of the following equation: [20].

$\delta_s = 23.45\, Sin\left[360\,\frac{284+d_n}{365}\right]\ degree$……………. (2.5)

2.4.3 Derived Sun – Earth Angles

In addition to the three basic angles latitude, hour and sun's declination angles, some other angles are used to define the sun's position in relation to the surface as the below figure illustrates [21]:

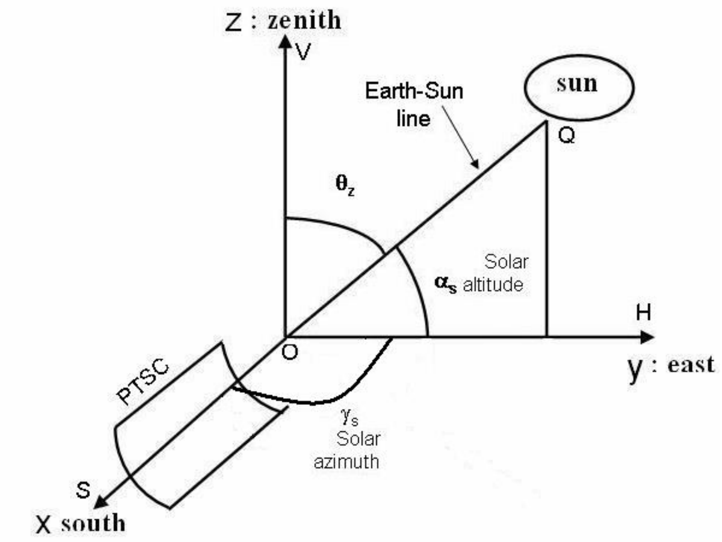


Figure (2-6) Derived sun – earth angles [21]

The zenith angle (θ_z): is the angle QOV which is situated between a vertical line on the horizontal surface at O and the sun rays.

The solar altitude angle (α_s): is the angle QOH on a vertical plane between the sun rays and its projection on the horizontal plane, i.e. the complement of the zenith angle. It follows that:

$\alpha_s + \theta_z$= 90……………….. (2.6)

The same can be attained using the following [22].

$Sin(\alpha_s) = Sin(L)Sin(\delta_s) + Cos(L)Cos(\delta_s)Cos(h_s)$……….. (2.7)

The solar azimuth angle ($\boldsymbol{\gamma_z}$): the angle HOS is the angular displacement that begins from the south to the horizontal projection of the rays of sun. To find the azimuth angle, the following equation is used [23].

$Cos(\boldsymbol{\gamma_z}) = \frac{\boldsymbol{Sin(\alpha_s)Sin(L)-Sin(\delta_s}}{\boldsymbol{Cos(\alpha_s)Cos(L)}}$…………………….. (2.8)

2.5 The Solar Radiation

The sun is a spherical source with a diameter of about 1.39 million Km. Due to its immense, but finite size, it has an angular diameter of 0.53° (32') as shown in figure (2-7). The solar radiation is the energy flowing noticeable from the sun jointly and constantly in all directions. The solar radiation can be divided into two types, extraterrestrial and terrestrial [24].

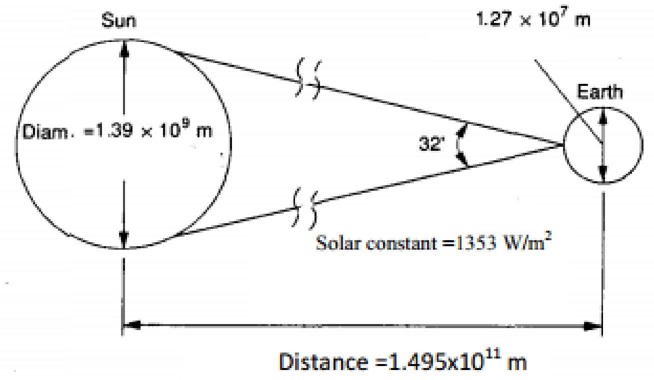


Figure (2-7) Sun-earth relationship [24].

2.5.1 Extraterrestrial Solar Radiation

Extraterrestrial solar radiation represents the solar radiation outside the earth's atmosphere (ETR) (Io). Many factors determine the ETR on top of the earth's such as the orientation and distance. ETR at the mean sun earth distance, Dm is called the solar constant, the French scientist Pouillet was the first one who introduced Isc in 1837 [24] and current accepted value from NASA is said to be 1353 W/m2, and the mean sun earth distance is 1.496×1011m. There is a seasonal solar due to elliptical orbit of the earth about the sun, and because its variation, the earth sun distance varies with 1.7 percent. Io varies by the inverse square law, as shown in the following equation [22].

$$I_0 = I_{sc} \left[\frac{D_m}{D_{e-s}}\right]^2 \text{……………..} (2.9)$$

Where D_{e-s} is the distance between the sun and the earth, the value of I_0 is the d Extraterrestrial solar radiation corresponding to the De-s. An empirical formula for the above equation can be given as [24]:

$I_0 = I_{sc}\left[1 + 0.034\ Cos\left[\frac{360 d_n}{365.25}\right]\right]$………….. (2.10)

2.5.2 Terrestrial Solar Radiation

There are a lot of parameters decreases the a mount of radiation that reaches the earth's surface such as absorption, reflection, and light scattering.

The total radiation that is incident on the surface of earth consist of two components: The first, beam radiation, Ib, is solar radiation falling on the surface that has passed through the atmosphere without being scattered in an appreciable way [24]. The second is diffuse radiation, Id, that reaches the surface after being significantly scattered by the atmosphere [24]. Terrestrial solar radiation or global (total) radiation refer to the sum of the beam and diffuse radiation [25]. Hottel proposed [26] simple clear sky model to calculate the beam solar radiation at normal incidenc. Shah provides the following equation for Ib [27]:

$I_b = I_0\left[a_0 + a_1 e^{-(KAM)}\right]$…………………… (2.11)

Parameters a_0, a_1 and k are empirical constants, and given by Duffie and Beckman [24].

$a_0 = 0.94[0.4237 - 0.00821(6 - AL)^2]$…………………. (2.12)

$a_1 = 0.98[0.5055 - 0.00595(6.5 - AL)^2]$………………... (2.13)

$K = 1.02[\,0.2711 - 0.01858(2.5 - AL)^2]$………………... (2.14)

Where AM is the air mass (which equal to 1/cos θ_z or 1/sinα_s) and AL, is the altitude's location above mean sea level (km).

For tilted surface the beam radiation received is related to incident angle, θi given by [24]:

$I_{bt} = I_b Cos\theta_i$……………. (2.15)

By the use of the equation (2.16) the diffused solar irradiance can be determined on a horizontal surface [19]:

$$I_d = I_0 Cos\theta_z\left[0.2710 - 0.2939\left(a_0 + a_1 e^{-(KAM)}\right)\right] \ldots\ldots (2.16)$$

The diffused radiation has no effect on the design of the concentrating collector calculation, while the fraction of direct radiation is particularly important to the performance of focusing or concentrator collector.

2.5.3 Spectral distribution of Solar Radiation

When the solar radiation reach earth surface, two phenomena cause the wavelength of the spectral circulation of the solar radiation to go beyond the range of 0.2 - 50μm of the atmosphere. This range is reduced to 0.3 – 3μm. The primary phenomenon is the scattering of the radiation while it passes through the atmosphere, which is instigated by the interaction between the radiation and the water (vapor and droplets), dust and also, air molecules, as illustrated in figure (2-8). The degree of scattering depends on the number of particles and their sizes relative to the wavelength of the solar radiation. The scattering occurs in accordance with the theory of Rayleigh (i.e., Rayleigh scattering depends on wavelength to the negative fourth powerλ^{-4}). The Rayleigh distribution

becomes significant at shorter wavelengths less than 0.6 µm [28].

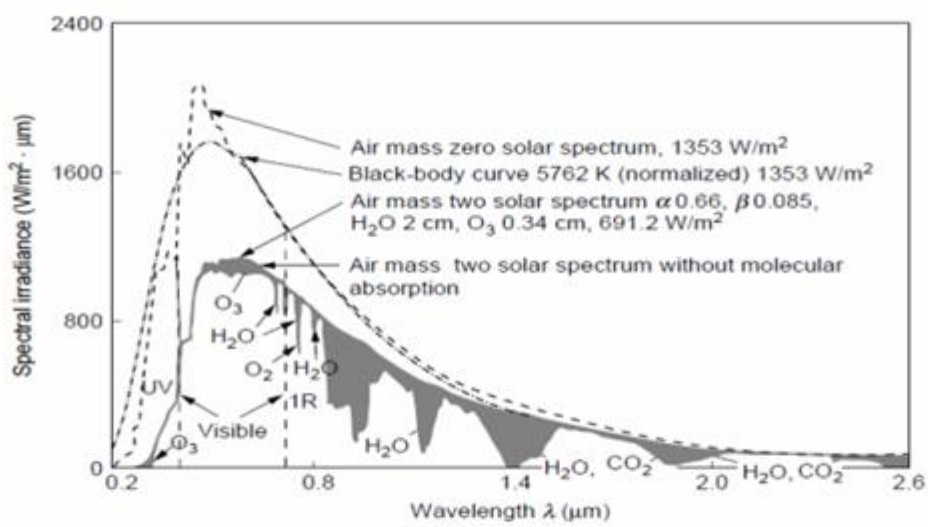


Figure (2-8) The spectral distribution of the solar radiation [28].

Carbon dioxide and water vapor absorb in the infrared region of the solar radiation and Ozone absorbs in the ultraviolet. Water vapor absorbs strongly in bands centered 1.0, 1.4 and 1.8 μm. beyond 2.5μm.

the spectral circulation of the solar radiation as a function of wavelength is shown below on table (2-2)

. It can be noticed from this table that the solar radiation contains about 47% of the visible range and this ratio is decreased by the dust effect especially for the Iraq environment.

Table (2-2) the spectral distribution of solar radiation [21].

Range	Wavelength (nm)	Percentage%	Solar radiation W/m^2
UV	0-380	7	95
Visible	380-780	47.29	640
IR	780-3000	45.71	618

2.6 Geometry of Parabolic Trough Solar Collector

Parabolic Trough Solar Collector (PTSC) which can be also called cylindrical parabolic collector employs linear imaging concentration. The receiver tubes are present in these collectors, which are situated lengthwise the focal line of the parabola. As well, a cylinder-shaped concentrator of parabolic shapes that are cross-sectional is present. A section of a PTSC is displayed in figure (2-9).

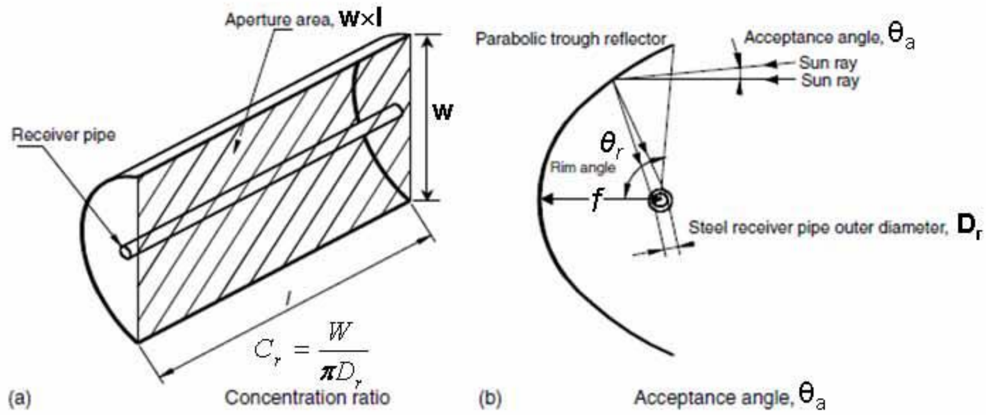


Fig (2-9) The cross-sectional view of PTSC [29].

Basically, it consists of (i) a parabolic reflector of about 1-6 m aperture width, (ii) an absorber (receiver) tube made of steel or copper with diameter 1.5-5 cm and coated with selective coating, and (iii) a concentric tubular glass cover surrounding the receiver with a gap of about 1-2 cm which is evacuated [29].

The cylindrical parabolic reflector focuses all the incident sunlight onto a metallic tubular or flat receiver placed along its length in the central plane. The heat transfer fluid is permitted to flow throughout the receiver. The parabolic reflector is defined by its aperture diameter (W), rim angle (θ_r), and receiver's shape and size.

At a random placement, the radius of the parabola is demarcated by r, and is designated the "mirror radius". At its external edge, the maximum mirror radius is able to occur, and is therefore called the "rim radius" or the parabolic radius. The rim angle, θ_r, resembles the beam radiation reflected from the external rim of the concentrator. The focal length, f, is associated with the rim angle, and aperture width, W, as [30]:

$W = dftan\left[\frac{\theta_r}{2}\right]$………………… (2.17)

The mirror radius found at the incident point of the beam radiation can define the dimension of a reflected solar image at the focal point. A simple calculation for the image width $W_{im}$was made by Jeter [31].

$W_{im} = r\theta_s$…………………….. (2.18)

Where θ_s stands for the width of the angle of the incident beam radiation of 0.53^0 (≈0.00925 rad), acceptance half – angle θ_a of 0.267^0, the reflected beam trail length being equivalent to the parabolic radius, r. Accordingly, for obtaining an almost normal incidence, taking place more commonly in the summer months, the calculation (2-18) can be redrafted as:

$W_{im} = 0.00925r$……………………. (2.19)

The geometric concentration ratio is given as [22]:

$C = \frac{Effective\ aperature\ arae}{Receiver\ tube\ area} = \frac{W - D_{r,o}L}{\pi D_{r,o}L}$…………… (2.20)

The concentration ratio (C) is related to θ_r can also be defined as [29]:

$C = \frac{Sin\theta_r}{\pi Sin\theta_a}$……………………. (2.21)

For intercepting the entire solar image, the size of the receiver can be figured. The diameter D_r of a cylinder-shaped receiver is illustrated below [24]:

$D_r = 2rSin\theta_a = \frac{WSin0.267}{Sin\theta_r}$……………….. (2.22)

For a flat receiver in the focal plane of the parabola the width W_f is determined below [24]:

$W_f = \frac{2rSin\theta_a}{Cos(\theta_r + 0.267)} = \frac{WSin0.267}{Sin\theta_r Cos(\theta_r + 0.267)}$……………………… (2.23)

2.7 The optical Performance for PTSC

The optical analysis of solar collectors with a parabolic reflector must take into consideration various effects, such as the optical properties of materials, the relative size of the receiver and concentrator and the type of tracking and corresponding losses.

2.7.1 Incidence Angle Modifier

Other losses from the collector, added to the losses caused by the angle of incidence, can be correlated to the incidence angle. The tracking of errors in the concentrating collector, the effects of the errors and the errors in the dislocation of the receiver from the focus all direct to deviated or enlarged images and effect ~~on~~ the intercepting factor. For counting these inaccuracies, the incidence angle modifier K (θ_i) is used, it is set as an empirical fit to experimental data for a particular collector type. The incidence angle modifier for the LS-3 collector is [31,32].

$K(\theta_i) = Cos\theta_i + 0\ 000884(\theta_i) - 0\ 00005369(\theta_i)^2$................. (2.24)

Where θ_i, the incidence angle, is provided in degrees.

2.7.2 Optical Efficiency of the PTSC

The optical proficiency μ_0 represents the part of the solar radiation occurring on the aperture of the collector, then assimilated at the exterior of the receiver tube [33].

$\mu_0 = \frac{S}{I_b}$................................ (2.26)

By using all of the changes that the solar radiation based on them? So, the values of the amount of radiation(S), or the absorbed radiation present onto the receiver can be determined by a simple calculation [34,35]:

$S = I_b\left(\rho_a \tau_g \alpha_r \gamma\right) K(\theta) X_{END}$.................... (2.27)

The optical efficiency of the PTSC embodies numerous significant concentrators' optical characteristics, as well as the mirror surface reflectance, ρ_a, the receiver (glass) transmittance, τ_g, the receiver surface absorption, α_r, and the intercept factor γ, which is a representation of the section of the reflected radiation that intercepts the receiver.

2.8 The Thermal Performance and Losses of PTSC

A working fluid is utilized to remove the energy from the receiver, inside a thermal conversion system. The thermal effectuation of PTSC depends on its thermal effectiveness which is termed as the ratio of the valuable energy distributed to the energy occurring at the concentrator aperture. The thermal losses for a PTSC are due to the convection and the radiation from the receiver tube to the ambient [29,34].

2.8.1 Overall Heat Loss Coefficient (U_L)

The general loss coefficient (U_L) merges the thermal losses into one coefficient. Assuming that the area amongst the cover (glass) and the receiver (absorber) tube is a stationary amount of air, $U_L$is one of the crucial parameters and can be determined from the below equation [36].

$$U = \left[\frac{A_r}{A_g\left(h_{c,g-amb}+h_{r,g-amb}\right)} + \frac{1}{h_{r,r-g}}\right]^{-1} \text{………………. (2.28)}$$

Where: A_r area of the receiver (absorber) tube and A_g area of glass cover. $h_{c,g-amb}$= the convection heat transfer coefficient in the middle of the glass and the ambient air which depends on the wind [34,36].

$$h_{c,g-amb} = h_w = \frac{N_{Ua}K_a}{D_g} \text{……………………… (2.29)}$$

Nusselt number (N_{Ua}) of the air can be defined by two equations [34]:

$N_{Ua} = 0.4 * 0.54 * {Re_{ea}}^{0.53}$ for $0.1 < Re_{ea} < 1000$……….. (2.30)

$N_{Ua} = 0.0.3 * \ {Re_{ea}}^{0.6}$ for $1000 < Re_{ea} < 50000$……….... (2.31)

Reynolds number (Re_{ea}) of the air is calculated by the following equation [37]:

$$Re_{ea} = \frac{\rho_a \vartheta_a D_g}{\mu_a} \text{…………………… (2.32)}$$

Where ρ_a , ϑ_a and μ_a are the density, the velocity and the viscosity of air, respectively. D_g is the cover (glass) diameter.

$h_{r,g-amb}$= the radiation heat transfer coefficient between the glass and the ambient.

$$h_{r,g-amb} = \varepsilon_g \delta\left(T_g + T_{amb}\right)\left({T_g}^2 + {T_{amb}}^2\right) \text{……………….. (2.33)}$$

$h_{r,r-g=}$ the radiation heat transfer coefficient between the receiver tube and the glass tube.

$$h_{r,r-g} = \frac{\delta\left(T_r+T_g\right)\left({T_r}^2+{T_g}^2\right)}{\frac{1}{\varepsilon_r}+\frac{A_r}{A_g}\left(\frac{1}{\varepsilon_g}-1\right)} \text{…………….. (2.34)}$$

Where T_r stands for the temperature of the receiver, T_g represents the temperature of the glass,ε_r is the emittance of the receiver, ε_g represents the emittance of the glass and δ stands for the Stefan Boltzman constant which is: 5.67×10^{-8} W/m^2k^4.

2.8.2 Heat Transfer to Fluid

The heat conveyance starting from the receiver tube to the fluid (HTF) must be categorized by turbulent or laminar flow conditions consequently, the evaluates the Reynolds number, Ref of the fluid [38].

$Re_f = \frac{4m^{\cdot}}{\pi D_{r,i}\mu_f}$………………… (2.35)

The Nusselt number of the fluid Nu_f for the laminar flow is given by equation (1-33) and for the turbulent flow by equation (1-34).

If $Re_f < 2200$

$Nu_f = 3.7$…………………. (2.36)

If $Re_f > 2200$

$Nu_f = \frac{(f_f/8)Re_f Pr_f}{1.07+12.7\left(\sqrt{f_f/8}\right)[Pr_f{}^{2/3}-1]}$………………….. (2.37)

The friction factor, f_f , for the smooth pipes is determined by:

$f_f = \left(0.79ln\left(Re_f - 1.64\right)\right)^{-2}$

The heat transfer coefficient, h_f, to the fluid is furtherly assessed [32]:

$h_f = \frac{Nu_f k_f}{D_{r,i}}$………………….. (2.38)

Where. $m^{\cdot}$, μ_f , k_f, and Pr_f signify the mass flow rate, the viscosity, the thermal conductivity and the Prandtl number of the fluid, correspondingly. $D_{r,i}$, is the receiver's inner diameter.

2.8.3 Overall Heat Transfer Coefficient and Factors

The overall heat transfer coefficient (U_0) is described as the coefficient for the heat transference from the environs to the fluid, according to the outer diameter of the receiver tube$D_{r,0}$, this is found by the following equation [36,39]:

$$U_0 = \left[\frac{1}{U_l} + \frac{D_{r,0}}{h_f D_{r,i}} + \frac{D_{r,0} ln\left(\frac{D_{r,0}}{D_{r,i}}\right)}{2K}\right]^{-1} \text{……………………... (2.39)}$$

Where: K is the thermal conductivity of the receiver tube material.

Defining a collector efficiency factor (F^{-}) in the most appropriate way would be as: the ratio of actual valuable energy collected to the useful energy collected, with the condition that the total absorber surface is at the mean fluid temperature.

$$F^{-} = \frac{U_0}{U_L} \text{……………………… (2.40)}$$

Now eq. 2-39 is reworked as it follows: [39]:

$$F^{-} = \frac{1/U_L}{\frac{1}{U_L} + \frac{D_{r,0}}{h_f D_{r,i}} + \frac{D_{r,o} \ln(D_{r,0}/D_{r,i})}{2k}} \text{.................................. (2.41)}$$

The **heat removal factor** or the correction factor, FR, having value ranges between 0<F_R <1, could have an interpretation as the ratio of the actual useful energy collected to that which would be collected if the entire absorbed surface is at the temperature of the fluid entering the collector. When seen as a heat exchanger, FR means a measure of the effectiveness of the receiver, which is actually the effectiveness with which the absorber radiation energy can be relocated to the working fluid. Its value is controlled through the working fluid flow rate and its properties along with the thermal characteristics of the receiver material [40].

$$F_r = \frac{m_f \cdot C_p}{A_r U_L}\left[1 - exp\left(-\frac{A_r U_L F^{-}}{m_f \cdot C_p}\right)\right] \text{………………… (2.42)}$$

Where: C_p is the precise heat of the fluid.

The collector flow factor F'' is then determined in the subsequent calculation:

$$F" = \frac{F_r}{F^-} = \frac{m_{f}{\cdot}C_p}{A_r U_L F^-}\left[1 - exp\left(-\frac{A_r U_L F^-}{m_{f}{\cdot}C_p}\right)\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2.43)}$$

2.8.4 Thermal Efficiency of a PTSC

The instant thermal efficacy μ_{th} of a solar concentrator could be found from an energy balance on the receiver. The valuable heat gain, Q_u, distributed by the receiver has the possibility to be written according to the optical and thermal losses, the optical losses being symbolized by the optical efficiency, μ_0 [22,35].

$$Q_u = \mu_0 I_b A_a - U_L(T_r - T_{amb})A_r \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2.44)}$$

Where A_a is the aperture area, as the receiver surface temperature can be challenging to find out, it is expedient to define the Qu in terms of the inlet fluid temperature by way of the heat removal factor F_R as [42]

$$Q_u = A_a F_R\left[S - \frac{U_L(T_{f,i} - T_{amb})}{C}\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2.45)}$$

The valuable heat associated with the flow rate can also be determined in accordance with the fluid difference temperature as below: [29]

$$Q_u = m{\cdot}C_p(T_{f,0} - T_{f,i}) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2.46)}$$

Where: $T_{f,i}$, $T_{f,0}$ and T_{amb} represent the inlet fluid, the exit fluid and the ambient temperatures, respectively.

The thermal efficiency of the solar thermal collector can likewise be defined plus simplified as the ratio of the beneficial heat Q_u, delivered per A_a, and the insolation, I_b, which is occurring on the aperture.

$$\mu_{th} = \frac{Q_u}{A_a I_b} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2.47)}$$

The thermal efficiency of the collector can now be reworked from eq. 2-42 and eq. 2-44 as follow [42,43]:

$$\mu_{th} = F_R \left[\mu_0 - \frac{U_L(T_{f,i} - T_{amb}}{I_b C}\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.48)$$

The thermal efficiency is determined by two kinds of quantities, which are the parameters that characterize the conditions during the operation and the concentrator design parameters. Where the solar flux, the inlet fluid temperature and the ambient temperature define the operating circumstances [29], the optical efficacy, the heat loss coefficient and the heat removal factor are the design dependent parameters. The exit fluid temperature, Tf,o the temperature rise, (Tf,o-Tf,i) and the efficacy can be found out by the use of the below given equation [29,43].

$$\mu_{th} = \frac{m \cdot C_p (T_{f,o} - T_{f,i})}{I_b A_a} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.49)$$

2.9 Heat Transfer Fluid for PTSC

With the support of the transfer fluids, the heat storage containers which are present in the solar heating and in the cooling systems are carried heat from the HCE. The water, the hydrocarbons, the glycol, the oil and the air are the most regularly utilized fluids. Upon the selection of a transfer fluid, there are significant criteria that should be taken into consideration: the viscidness, the coefficient of expansion, the boiling point, the freezing point and the thermal capacity. The next are some of the most ordinarily used heat transfer fluids and their characteristics.

- **Air:** it cannot boil or freeze, and it is non-corrosive. On the other hand, due to its extremely low heat capacity, it has the tendency to leak out of the dampers, the collectors and the ducts.
- **Water** is low-priced and non-toxic. Its characteristics are that it has a very low viscosity and a high particular heat, which are making it easy for it to be pumped. Inopportunely, water possesses a fairly high freezing point and a low boiling point. Moreover, it can be caustic if the PH (acidity / alkalinity level) is not kept at neutral levels.

- **The hydrocarbon** oils possess a lower particular heat than water and a high viscosity; but then again more energy is necessary to pump these oils. These oils have a low freezing point and are somewhat economical.
- The heat transfer fluid found in air conditioners, the heat pumps and the refrigerators are the most frequently used refrigerants/phase change fluids. A characteristic is that they mostly have a high heat capacity in addition to a low boiling point. The absorption of the heat takes place when the refrigerant boils (changes phase from liquid to gas) in the solar collector.

The key fluids used by refrigerator, air-conditioner and heat pump producers are the chlorofluorocarbon refrigerants (CFC), such as Freon, due to the fact that they are not flammable, low in toxicity, firm, not corrosive and they cannot freeze. [29,44].

2.10 Storage of Thermal Energy

Thermal energy may be deposited in the forms of: 1) sensible heat, 2) latent heat. The dissimilarity between these methods is that they differ in the quantity of heat that can be stored per unit weight or volume of the storage media and the operating temperatures [45].

2.10.1 Sensible Heat Storage

In the environment of a sensible heat storage, changing the temperature of the storage medium leads to the storage of the thermal energy. To rise the temperature of one unit of a material, it is required one degree of its specific heat (heat capacity), however, the temperature change, the amount of storage media and the heat capability of the media being employed will determine the amount of heat stored. Water and pebbles are the most universal materials for the low temperature energy storing and the hydrocarbon oil has been proposed for high temperature [28,46].

2.10.2 Latent Heat Storage

In the latent heat storage, the means of a reversible alteration of state (phase change) are utilized for storing the thermal energy in the media. The most commonly utilized are solid-liquid transformations. Both of the solid-gas and liquid-gas phase changes consist of the maximum energy of the promising latent heat storage methods. Molten salt is a commonly used media for the latent heat storage in the solar thermal system [47].

2.11 Introduction to absorption refrigeration cycle

Generally, the refrigeration cycle has two types, the first being the vapor compression refrigeration and the second the vapor absorption refrigeration cycle. The vapor compression cycle is the conventional one, of which consumption of electrical energy in which CFC's are utilized as refrigerants, is big. As a considerable amount of electrical energy is needed, the total of the fossil fuels that has to be burnt will be great, leading to more CO2 emissions. Secondly, the working pairs used are corrosive and toxic in their nature and also result in ozone layer depletion. With a rapid economic growth, more concern about energy utilization and environmental pollution will be raised by these factors, ~~and~~ therefore human beings have to face energy and environmental issues more seriously. In order to solve these problems, methods are developed to enhance the energy utilization efficiency, to utilize renewable energy resources and so on.

The absorption heat pumps or the absorption chillers both can be powered by low-grade thermal energy which makes them important energy saving devices, they could be derived using geothermal energy, solar energy and industrial waste heat resulting from the industrial processes, hence the devices will have a major part in reducing the environmental pollution and the emission of carbon dioxide and in the improvement of the energy utilization effectiveness. A lot of attention was paid to the absorption refrigeration technology all around the globe, since it is pleasant to the environment and could be advantageous because of the low-grade energy, when it comes to the energy set in the exhaust steam of low temperature and low pressure and being ignored [48].

In the present, it is clear to us that the vapor absorption cycle is superior to the conventional vapor compression cycle; now, the working pairs for the absorption refrigeration cycle must be explored. However, first of all, the succeeding section will explain clearly the dissimilarities b/w the vapor compression cycle and the vapor absorption cycle [48].

2.12 Differences between the absorption refrigeration cycle and the compression refrigeration cycle

- Between these cycles, a noteworthy difference is the manner in which the compression is reached. It is known that in the vapor compression cycle, the compression is accomplished by utilizing a compressor, of which consequence is a high electrical energy consumption. However, in the vapor absorption cycle, the compression is accomplished by utilizing a generator-absorber solution circuit, which is made of a heat exchanger, a generator, an absorber, and a pump.
- Corrosive and toxic refrigerants are used in the vapor compression cycle, while in the vapor absorption cycle the H2O and the ionic liquid based on the working pairs can be utilized, both of them being environmentally friendly.
- In the vapor absorption cycle, the work input is low, meaning that the pumping involves liquid, but in the vapor compression cycle, the work for achieving the compression is extraordinary.
- Despite the fact that there is not as much equipment in the vapor compression cycle as in the vapor absorption cycle, the latter is economically reasonable as it utilizes a low grade source of heat like solar energy, geothermal energy and other industrialized sources of heat.
- In order to execute the vapor compression work, a high pressure needs to be maintained, while the vapor compression cycle can work in low pressure using specific working pairs [48].

2.13 Low grade Energy

From sustainable resources like the solar energy or the geothermal energy, the energy that is derived, together with the energy passed over from the industries, are implanted in the exhaust steam of low temperature and pressure, which therefore cannot be used in order to achieve work at all. These energies are denoted as low grade heat energies. Its imperative role is to lessen the carbon dioxide emissions, the environmental contamination as well as enhancing the energy consumption effectiveness. For lowering the consumption of the electrical energy to a great degree, low grade thermal energy should be utilized [48].

2.14 Working pairs

The conformation of the refrigeration cycles is not the only factor that is able to control the performance of the cycle. Moreover, the thermodynamic characteristics of the working pairs habitually made of absorbent and refrigerant partake. A few frequently applied working pairs in the absorption cycles are aqueous solutions such as either lithium bromide water ammonia or water. Nevertheless, their most important drawbacks in the industrial applications are the high working pressure, the crystallization, the corrosion and the toxicity, which are all detrimental.

Therefore, the focus of the research in the past two decades was on seeking for further beneficial working pairs that have a good thermal stability, with minimum corrosion and without crystallization.

For the compliance with the above mentioned properties, the focus is on the ionic liquids(ILs**)** that have appealed owing to their exceptional properties, for instance good solubility, staying in the liquid state over a wide temperature range from room temperature to about 300°C, non-flammability, low melting points, negligible vapor pressure and thermal stability. The previously mentioned favorable properties of ILs have been seen as an impulse for carrying out the needed research on the absorption refrigeration cycle using IL- based working pairs in which IL is utilized as an absorbent and water is utilized as a refrigerant [48].

The following are the looked-for properties of the working fluids of the absorption cycles: - [48]

- The dissimilarity in the boiling point amongst the pure refrigerant and the absorber solution, at an identical pressure, ought to be as great as possible.
- Within the absorbent, the refrigerant should have a high concentration and additionally a high heat of evaporation so that the low circulation ratio could be preserved between the generator and the absorber per unit of chilling capacity.
- The viscosity, the thermal conductivity and the diffusion coefficient are all transport properties that should be preferable.

➢ It is of high importance that the refrigerant and the absorbent to be reasonably priced, ecologically friendly and noncorrosive.

2.15 The working principle of the absorption refrigeration cycle

The evaporator, the generator, and the condenser are the elementary constituents of the absorption cycles of the absorber. While the waste heat passed from the industries at the equivalent temperature supplies the generator, the evaporator is provided by the surroundings that need to be chilled. The augmented heat conveyed from the condenser and the absorber is used for condensing the refrigerant, which is water. A description for the operating system of the absorption cycle is in Figure 2.10, as follows [48].

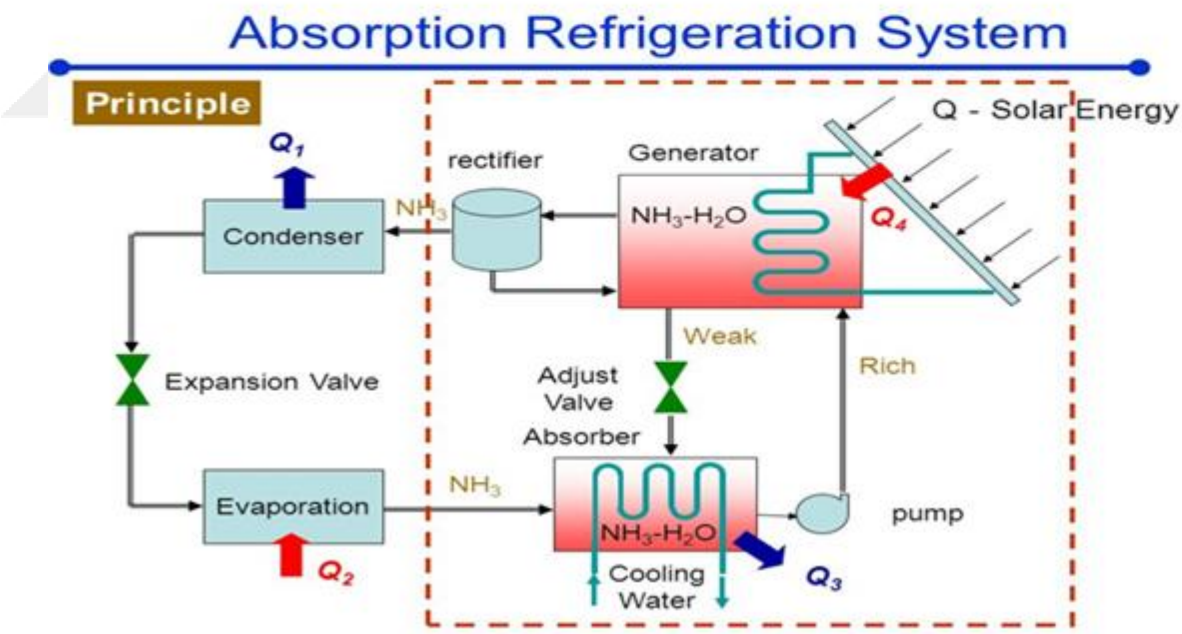


Figure.2.10 Vapor absorption refrigeration scheme

2.16 Literature Review

The following is a summary of the previous experimental and theoretical work related to the current study on the subject of the solar concentrator. Nowadays, the technology of the parabolic trough is the most extended solar system for producing electricity or for generating steam for the industrial practices.

In the records it is revealed that starting from 1774 there have been efforts to captivate the energy of the sun for the power production [49]. The French chemist Lavoisier designed a furnace which attained the significant temperature of 1750 C°. A 1.32m lens was used by the furnace, plus a secondary 0.2m lens to focus the sun's rays onto a test tube [50]

A large solar engine was built between 1907 and 1913 by an American engineer called F. Shuman and C.V. Boys. Its characteristics consist of it being over 50hp, with a 1200m2 array of parabolic troughs which has the purpose of pumping the irrigation water from the Nile River near Cairo, Egypt. It was closed during WWI [51, 52].

As the solar concentrator rose to fame yet again in the 1960's, a technique has been developed with the purpose of enhancing the scheme of the focusing solar collectors. It has been accomplished by means of a study that elaborated the energy balances meant for a parabolic-cylindrical reflector with tubular receivers of different diameters, and it was studied by G.Lof 1962 [53].

The result of this investigation shows that the increase of the size of the receiver leads to an increase of the thermal losses and of the intercept factor, explained as "the fraction of radiation specularly reflected from the reflector intercepted by the receiver"

Later, Lof 1963 [54] studied an analysis of the factors and the methods involved in design optimization, and a set of graphical relationships which may be used in designing focusing solar-collectors' system of maximum efficiency and minimum size depending on the width ratio "the fraction of width of receiver to the parabolic cylinder reflector aperture".

The result of the analysis states both the maximum energy delivery and the intercept factor where the width ratio is (0.02-0.025). The influence of the target geometry on the maximum concentration has been theoretically studied by Edward and Cherng 1976

[55]. The results show that the elliptic cylindrical target and the flat plate will achieve a maximum concentration as the optimum target for the same value aperture.

Parmpal and Cheema’s 1976 [56] study had as its purpose analyzing the performance of a cylindrical-parabola collector on the subject of the collected energy quantity, the optimization of its different parameters and concluded that it seems to be more logical to use the aperture of a cylindrical-parabola collector as a characteristics’ dimension than the use of the focal length.

There was a complete prediction for the performance of the collector. The term stagnation temperature refers to the maximum temperature accomplished by the absorber for the zero energy withdrawal.

Ramsey and Gupta 1977 [57] evaluated the performance of the PTSC by means of the utilization of three absorbers that differ from each other; additionally, a heat pipe with a selective solar absorber layer applied to its surface, a black dyed tube intended for operating near the ambient temperature and a heat pipe which had its surface covered in a nonselective black paint.

When the incoming solar energy to the collector is usual, with the condition that the heat losses don’t occur, the peak efficiency for the collector can be approximately 62%. There have been some losses that arose at very high temperatures, such as 300Oc, which had the outcome of reducing the peak efficacies to 30% and 50% respectively, for the black painted tubes, selectively coated.

Derrick 1979 [58], analyzed and compared between the compound of simple parabolic and parabolic solar collectors on account of their individual reflector arc-lengths and because they are capable of accepting non-direct radiation. The simple parabolic concentrator is advantageous for the concentration ratios bigger than 10, because of the fact that the compound parabolic reflector’s expense is even higher than 4.4 times as costly. However, the compound parabolic concentrator may be less affordable than the simple parabolic trough.

The study of Clark 1982 [59] was based on the main design features that influence the yield of PTSC. Features like the incident angle modifier, the mirror-receiver tube the intercept factor, the end loss, and the spectral directional reflectivity of the mirror system, the receiver tube misalignment, were considered for the analysis and the effect of tracking errors.

The geometrical effects on the PTSC performance were studied by eter 1983 [60], he concentrated on end-effect. It was concluded that when short troughs are considered, the significance of end-effects particularly increases and that eliminating this effect is important for obtaining the test results.

Luz International Ltd. is an American company founded in 1979, which conceived three PTSC generations, called LS-1, LS-2 and LS-3, all fit in the Solar Electric Generating System (SEGS) plants. On the same structure length, the LS-1 and LS-2 are manufactured. They are generations of collectors that consisted of similar assemblies, with the difference that the aperture width of the LS-2 collector was twice of LS-1 collector. Their structure's base is made on a rigid structural support tube, designated as the torque tube, a tube that wires the steel profiles to which the parabolic mirrors are connected. In the LS-3, a metal lattice framework substitutes the torque tube, besides the opening width is 14% wider than the LS-2 and the collector length is doubled [61].

A sample structure of PTSC was created by Homas 1994 [62], with the purpose of learning its deflection and its optical properties under various load circumstances. The test offers enough information about the way the wind load affects the optical performance of a PTSC, all in the absenteeism of the wind tunnel facilities.

The performance examination of PTSC with artificial oil and water designated as the working fluids was done by Odeh et al. 1998 [63]. For the efficacy of solar parabolic trough collectors, the preparations have been created according to the absorber's wall temperature, for calculating the performance of the system with any working fluid. The thermal losses from the trough collector have been described in terms of the absorber, the emissivity, the wind speed, the absorber wall temperature and the radiation level.

The researchers Volker and Franz 2001 [64] processed an experimental and theoretical study of solar power plants in southern California working with a concentrator of PTSC (Solar Electrical Generating System) (SEGS) where these concentrators were competent in the production of the thermal energy crisis of the work of the electrical system, and considered the plants (SEGS) as some of the most profitable plants in terms of the economy where the price of kWh during peak hours is over 100 USA-cents.

A new design of PTSC called (Euro Trough) was developed by a group of researchers named Rafael et al. 2002 [65], from European countries, through the development of a generation of solar concentrators and reduced cost. The production of the two types of concentrates (ET 100 & ET 150), was developed and designed for utilizing it to produce the steam necessary for the applications of the solar thermal electric power generation. As the electric power generation system in California could complement the use of these models has been the rehabilitation of this plant is actually in the years (2000-2002) and use the same materials in the manufacturing of the models, but the difference between the lengths of PTSC in the area and the number of the absorber tubes, with a length of the form (ET150) (148.5m)area (817.5m2) both the model (ET 100) was long (99.5m) and the area (545m2).

Balbir and Fuaziah 2003 [66] studied the performance of a PTSC and the use of processed information meant for creating a replicated model with the utilization of the same meteorological information. The results' conclusion pointed out that there could be a probable symmetry accomplished amongst the growing thermal losses with the growing aperture area, and the growing optical losses with the declining aperture area.

Thomas and Michael 2005 [67] developed a PTSC comparable in terms of the dimensions to the smaller-scale marketable modules for use in a South African solar thermal research program. The collector length is 5m, the aperture width is 1.5m and the rim angle is 82o. Two receivers were fabricated for the comparative testing, including one enclosed in an evacuated glass cover. The peak efficiencies of 55.2% and 53.8% were obtained with the unshielded and the glass-shielded receivers respectively.

Umamaheswaran 2005 [68] offered particulars of the study of the assemblage, and also of the testing and of the examination of PTSC designed for a small scale home purpose water distillation application. The heating of the ground water is done by the solar radiation, as it moves along the solar collector within its absorber pipe, with the aim of producing the steam straightly into the absorber pipe.

Miguel, Javier 2006 [69] fabricated the concept of a PTSC called (SENER) and the aim of the research was to reduce the cost of the construction of the concentrators. Two sample modules of the SENER trough have been assembled and verified at the CIEMAT-PSA amenities. In October 2005, the first sample of the SENER parabolic collector was assembled and verified in CIEMAT-RSA amenities. The obtaining of the assembling experience was the aim, as well as the need to have a general impression of the operational performance while comparing it to other collectors. This initial sample consisted of a torque tube and cantilever arms fabricated with welded tube profiles. Each cantilever arm of the total amount of 28 were incorporated in the torque tube by the means of a manual jag. The following sample of the SENER parabolic collector was assembled in February 2006. In this secondary scheme, the ideal solution for the cantilever arms is contained within.

In the subsidiary prototype, the knowledge assimilated from the first prototype with regards to the mounting procedures was applied. The same tests were performed, more exactly, the optical and thermal tests, and they have shown that this model has more of an accurate structure which has resulted to a furtherly improved performance, and the addition of these stamped cantilever arms are able to decline the likely miscalculations in the situation of the mirror support points.

A new PTSC for hot water production was developed by Valan and Samuel 2006 [70]. The variation of the collector water outlet temperature, along with the water temperature of the storage container, have increased. The water temperature of the storage tank increased from 36 oC to 73 Oc.

The prediction of Kassem 2007 [71] was a natural convection heat transfer in an annular space between a glass envelope of a PTSC a circular receiver tube.

Solitem PTC-1800, the parabolic trough collectors that are solar thermal, was studied by Dirk et al. 2008 in order to supply heat for generating electricity, cooling and desalination. It was conducted from the results that the collector thermal testing has discovered substantial optical losses in addition to comparably low thermal losses. The applicability of the collector is high entirely for the medium temperature applications changing between 150° to 190°C.

CHAPTER THREE
METHODOLOGY

3.1 Introduction

A parabolic trough solar collector (PTSC) has been designed to collect solar energy (heat) and rise up the temperature of the inlet water.

The data have been gathered from the collector using more than one water inlet temperature and more than one water outlet temperature. The collector operated under the effect of the Iraq's climate. Moreover, thanks are given for engineers Murad, Mustafa and Maraw (mechanical dep., engineering college, Diyala university) who helped me building this device and collecting the data.

3.2 Extracted Data

The temperature has been the measure for the several time interval, from 0 to 35 min. using thermocouple type 2k (Chromel / Alumel), with the range of -50 to 1300 C^o and accuracy + - 2.2 C^o with the following description:-

- Five digits with LCD feature, it has overload show warning when input date be outside the boundary limitation.
- To measure the maximum and minimum temperature, and calculate the average between them if it necessary.
- It is an ability to take the temperature in C or F by just clicking the trigger to choice one between two temperatures reads.

Fig.3.1 Foxpic 6802II Digital Dual Two Channel 2K-Type Thermometer Thermocouple Temperature Meter Tester Sensor Probe

3.3 Theoretical Calculations

The collected data has been used to calculate the amount of heat collected through the concentrator using the following equation,

$Q = \dot{m}(h_1 - h_2)$……………….. (3.1)

Where:

Q = the collected heat kW.

$\dot{m}$ = the mass flow rate kg/sec.

h = enthalpy

The water flow rate measured by filling the tube in the PTSC with a specific amount of water (kg) and measure the time that needed to heat up. Then the mass flow rate was calculated from the following equation.

$mass\ flow\ rate = \frac{water\ mass\ (kg)}{total\ time\ (\mathrm{sec})}$ ……………………… (3.2)

3.4 Manufacture of trough solar collector

A polished aluminum sheet of a parabolic cylinder is used by the parabolic trough solar collector in order to concentrate and reflect sun radiations towards a receiver tube which is placed in the focal line on the absorber supporting. The arriving radiations are absorbed by the receiver and transformed into thermal energy; the solar radiation is collected and transported by a fluid which moved in horizontal axis toward the receiver tube. There are a lot of benefits of concentrated solar collection method to minimized the cost. So, that that can be used either for generating electricity, for thermal energy collection or them together, as long as it is a perfect side to exploit solar energy immediately.

There are two ways to manufacturing the reflected material. The first one is using rips put under the reflecting surface based on the frame; it looks like a house to hold the reflected. the second one is depending on elasticity and flexibility of the material.in our case we used the second way to made our sample with easy way by just curved the material like a parabolic and put the absorber pipe in the focal line then look at the pipe it was shiny, after that we measured the different parameters and gained amount of heat at last we assumed it is a parabolic. : [73]

- Low cost and simplicity. Really it is the easy way to get high performance parabolic trough solar collectors locally that does not necessary to use complicated equipment to achieve it. It is not specified by only reducing production cost, but the other thing is manufacturing methods of parabolic troughs, therefore, it made any fossil fuel cost more than the solar energy collecting substantially. It is possible that the social and economic significations of the method are huge. [73]
- Quality and better performance of the product. Though smaller parabolic troughs have the ability to be better than bigger ones because they have a lot of benefits, but there is an amount of realizing that as small as a parabolic trough is lower than bigger in performance is going to be. All that give us some information that the performance of a parabolic trough solar collector has fundamental characteristics; they are its optical efficiency and its concentration ratio. In this time, the values of the concentration ratio of a parabolic trough collector that has dimensions of width between (1m _ 2m) are limited to about

50 times under industrial manufacturing conditions and with big cost level. So, our design can achieve acceptable efficiency with acceptable heat gain .and put in our consideration the cost.

There are several usages for the solar collector characterizations here, and that will change by the located of the receiver tube. Depended on the low-temperature receiver give us imaginary of in this act that can be used for the water heater, the situation of the heater or heat exporter for an absorption cooler. [73]

Figure.3.2 The trough solar collector [73]

3.5 Support Stand of solar parabolic trough collector (SPTC)

The support stand was made of Mild Steel. It consists of ‘L’ and rectangle shaped cross sectioned bars welded together and two ball bearings fixed with the inner race with a rod. The outer race is rotary and mounted in the housing of absorber supporting plate.

Figure .3.3 Support stand

3.6 Supporting frame of solar parabolic tough collector (SPTC)

The frame modeled was of the form of 'L' cross section and made of Mild Steel. The Specifications of the support frame are as given in Table **1.**

Fig.3.4 Support frame

Table3.1 Specifications of Support

Dimension	Value
Length of the Support frame	**2000 mm**
Breadth of the Support frame	**1000 mm**
Thickness of the Support frame	**2 mm**

3.7 Absorber Supporting Plate

There are 2 supporting plates welded to the two opposite sides along the length of the support frame. It houses the absorber pipe at both ends while the lower end of the plate has two ball bearings mount which in turn connected to the supporting stand with one rotational degree of freedom.

Table3.2 Specifications of absorber

Dimension	Value
Length of the Plate	**400 mm**
Breadth of the Plate	**60 mm**
Thickness of the Plate	**3 mm**
Hole diameter for Absorber Pipe	**25 mm**

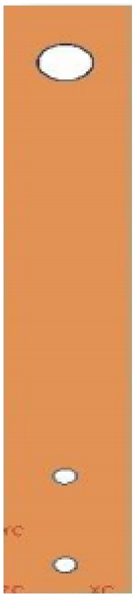


Fig.3.5 Absorber supporting plate

3.8 Absorber pipe

This part was designed by the designed way to matching the dimensions of the collector, another important thing on made it that we should put in our mentality mutual practice when we choice it like piping, working fluid velocity, fabrication, and heat loss. With this way the absorber is fabricated by the seamless chromium pipe, with the inner diameter of 23 mm, outer diameter 25 mm and 2100 mm in length, it is galvanized iron pipe painted with not shine black paint to absorb maximum heat as possible.[74]

Fig.3.6 The Absorber pipe

Table3.3 Specifications of Absorber Pipe

Dimension	Value
Inner diameter of the Pipe	23 mm
Outer diameter of the Pipe	25 mm
Length of the Pipe	2150 mm

The width of the absorber, 'D', is defined by:

$D = 2r_r sin0.267$…………………… (3.3)

Moreover, the collector has not the ability to move because it is fixed in its placed; therefore, there is a relationship between the altitude angle and the image of all the times. It may be store clearly about 10° per day.

Overall width of the absorber 'w' is 98 mm [(25x2) + 48].[74]

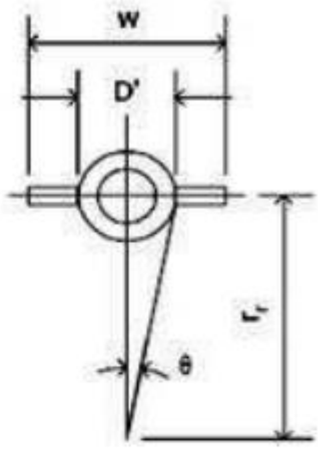


Fig.3.7 Cross section of the absorber pipe

$D\ develop = r_r.sin$…………… (3.4)

The relation of concentration ratio is suited to:

$C = \frac{[(a-b)\times L]}{[w\times L]}$………….... (3.5)

3.9 Parabolic trough

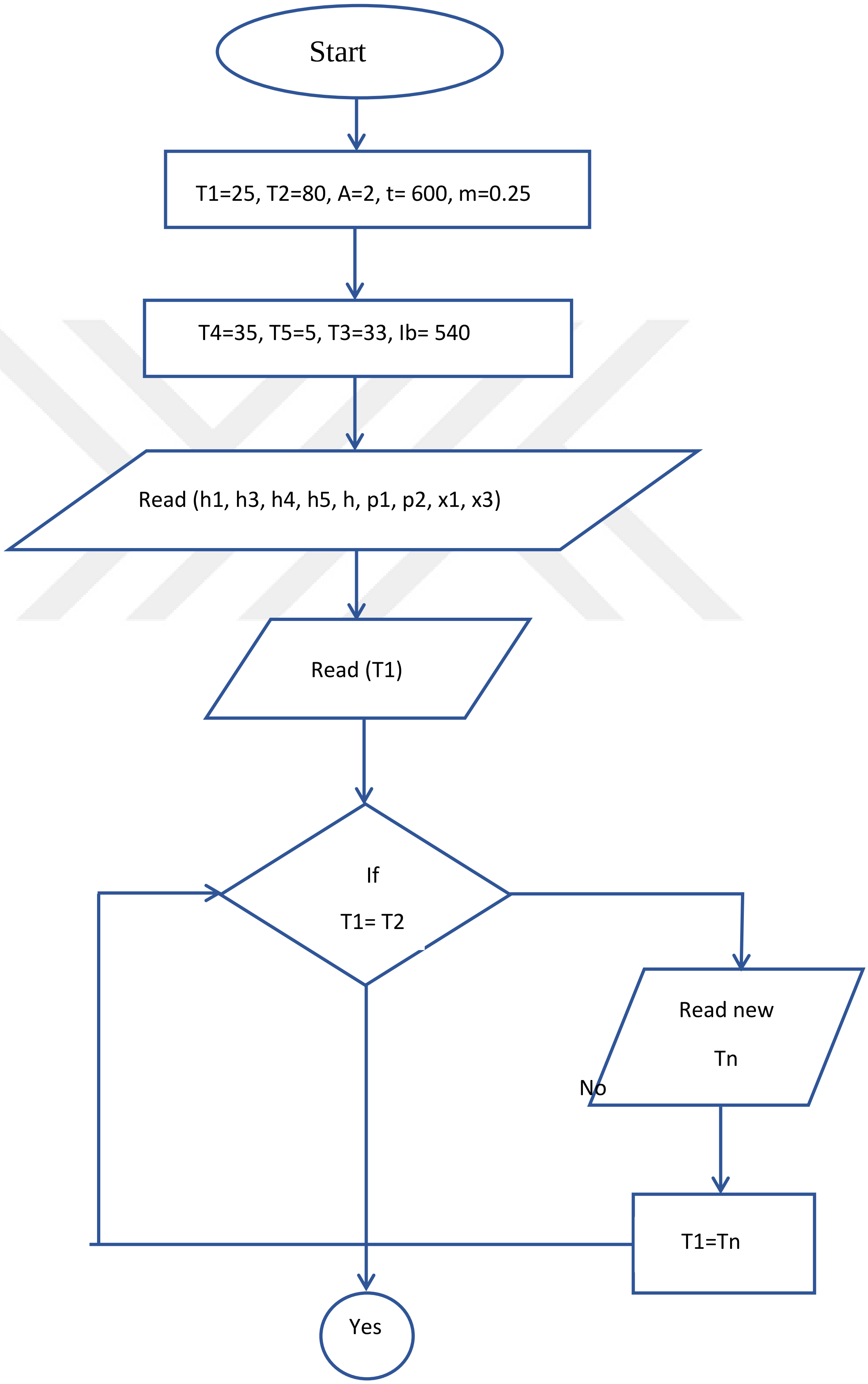
ALUMINIUM TROUGH: To obtain the desired dimensions one highly polished Aluminum sheets is used. The highly polished Aluminum sheets dimensions in table4.[74]

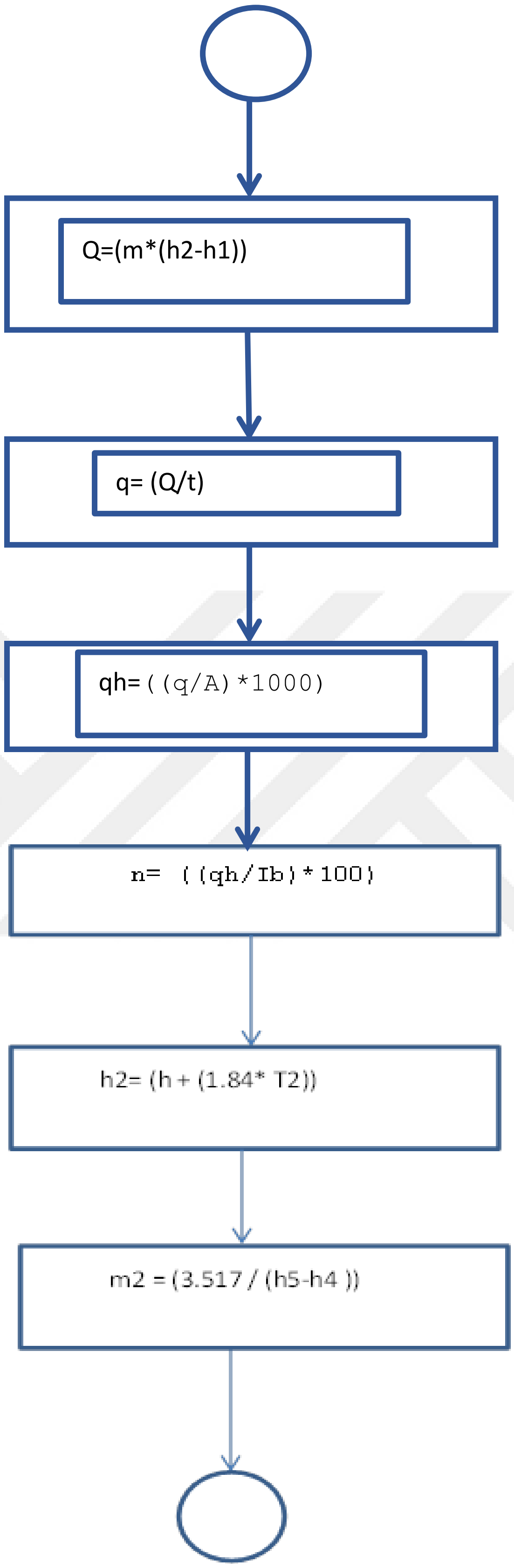
Table3.4 The highly polished Aluminum sheets

Dimension	Value
Length	2000 mm
Breadth	1000 mm
Thickness	1 mm
Length of the reflector	2000 mm

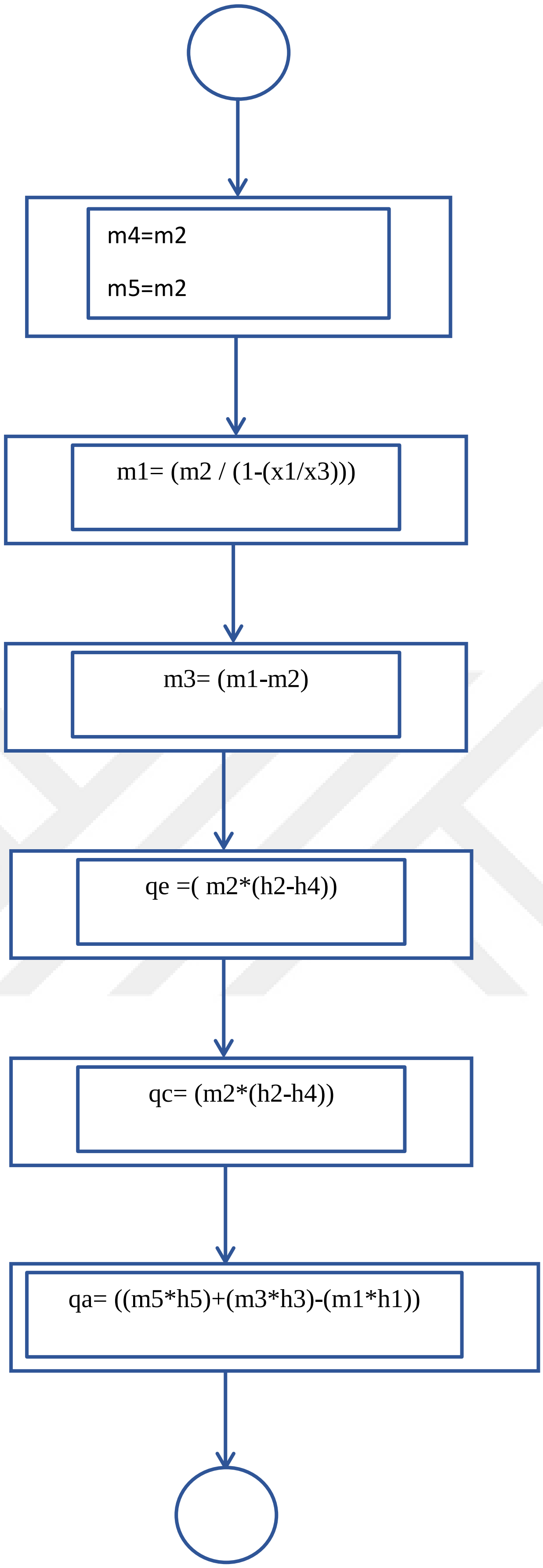
Fig.3.8 Aluminum trough

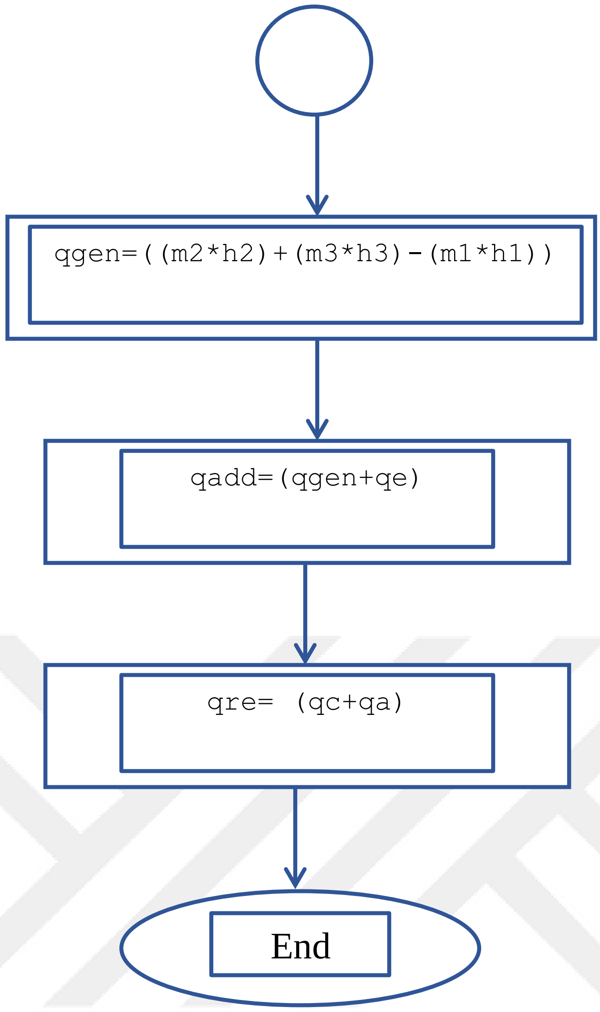
The calculation procedure can be clear vision by the following this flowchart diagram





Q=(m*(h2-h1))
q= (Q/t)
qh=((q/A)*1000)
n= ((qh/Ib)*100)
h2= (h+(1.84* T2))
m2 = (3.517 / (h5-h4))


m4=m2
m5=m2
m1= (m2 / (1-(x1/x3)))
m3= (m1-m2)
qe =(m2*(h2-h4))
qc= (m2*(h2-h4))
qa= ((m5*h5)+(m3*h3)-(m1*h1))


qgen=((m2*h2)+(m3*h3)-(m1*h1))
qadd=(qgen+qe)
qre= (qc+qa)
End

CHAPTER FOUR

THE EXPERIMENTAL WORK

Introduction

In this chapter, we will explain the effects of several parameters on the performance of the PTSC are analyzed. The results were obtained from outdoor experimental tests through the selected when the sky is clear in months of October, November, and December without draw-off HTF. The performance of the PTSC was evaluated using a metallic receiver.

4.1 Solar Radiation Intensity Calculations

The characteristics and the nature of solar radiation incident upon the earth surface are considered as the essential requirements for designing solar energy systems. The location of the PTSC in DIYALA is the latitude of 33.77° north, longitude of 45.14° east, and a standard time meridian of 45°. All results in this section are stored for just one day, October 10th (dn is a number of day, equal 283), in the evening , as we know the hour angle is at 0 degree. The declination angle was found to be -7.724°,

The sun located (position) in any period of the day by two angles, they are the solar altitude angle, α_s and the solar azimuth angle, γ_s. The solar altitude angle is 49.89°. However, they need to be related to fundamental angular quantities, as the sunrise and sunset hour angles, latitude and declination angle. The hour angles for DIYALA on October 10th were $\pm 84.89^{\circ}$; negative for morning and positive for the afternoon and the time from solar noon is calculated to be 5 hours 39 minutes and 33 seconds. However, due to the irregularity of the earth's motion about the sun, a correction factor is 13.8 minutes which is given by the equation of time (refer Eq. 1-3). Applying the equation of time correction factor, the sunrise and sunset local standard times are 6:21 AM and 17:39 PM, respectively, resulting in a day length of 11 hours and 19 minutes.[74]

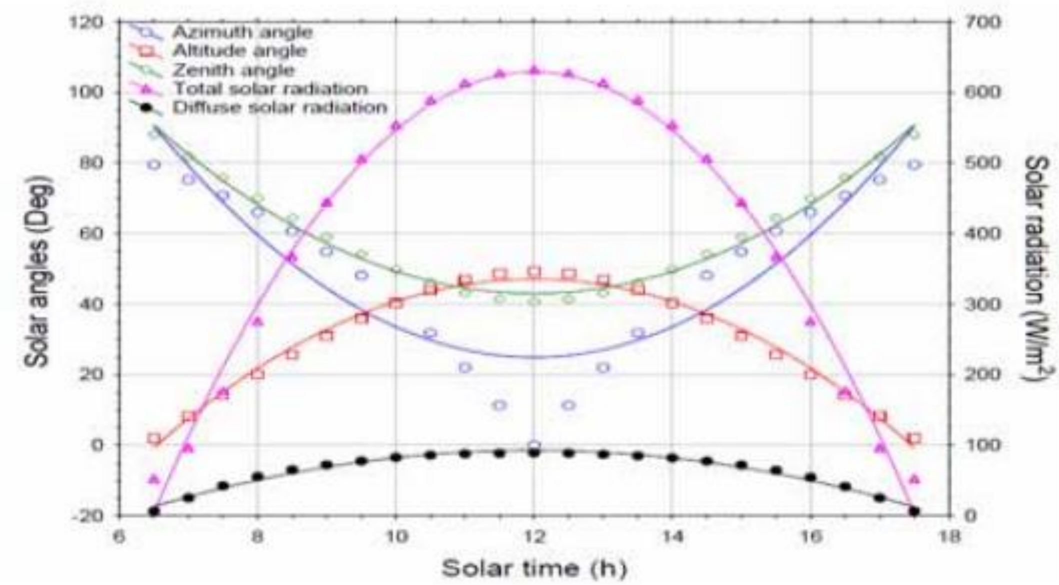


Figure (4-1) Variation of solar angles and solar radiation versus solar time.[74]

In the evening, the beam solar radiation dropping on the PTSC was measured to be estimable 633 W/m^2. So, the radiation that reflected our collector, approximately, 540 W/m^2 of solar radiation.[74]

4.2 Measuring the PTSC Heat Gain

After we created a solar collector in a simple manner and with the local materials, as shown in chapter two, we took the readings which listed in the (table 4.1) below for the month of February at 11:00 am by placing(0.25 kg) of water by using the thermocouple device, which was described in Chapter two also.

Table 4.1: Pressure and temperature change inside the pipe of PTSC versus

Time (min)	Pressure(bar)	Temperature(C°)
0	0	25
5	0.1	68
10	1.7	110
15	2.5	121
20	3.1	126
25	3.5	131
30	3.7	135
35	4	136

4.3 Sample of Calculations

4.3.1 Heat Gain from PTSC

To calculate the heat gain from PTSC we should find the temperatures, area, time, enthalpies and the mass of the water that putted in this collector inside and outside the collector.

When we store the different values in the table (4.1), so we can find the amount of heat that gain from PTSC and its efficiency by

$T1= 25\ ^oC$ degree initial temperature of inlet water in the absorber pipe

$T2= 80\ ^oC$ degree initial temperature of outlet water that outlet from the absorber pipe

$m= 0.25$ kg the amount of mass inside the pipe

$A=2\ m^2$ the area of the reflected material

$t= 600s$ the time that needed to rising the outlet temperature that needed to heat the fluid in the generator in the refrigeration system.

Now we can find the values of inlet enthalpy h_1

h1=104.81 kJ/kg from saturated water in T=25 C°, we find enthalpy

h2=461.41 kJ/kg from saturated water in T=80 C°, we find outlet enthalpy

To find the amount of heat that gain from the sun

$Q = m*(h_2-h_1) = 89.15$ kJ

$q=Q/t = 89.15/600 = 0.1486$ kJ/sec amount of heat generated per one second

To convert the amount of heat that gain from the sun from kw to w/m^2

$qh =[(q/A)*1000] = 74.2917$ w/m^2

From the latest studies that found the intensity radiation = 540 w/m^2

So the efficiency of the PTSC

Ib= 540 w/m^2 intensity of radiation.

$\eta = [(qh/Ib)*100] = 13.75\%$ collector efficiency.

4.3.2 Heat needed for absorption refrigeration system

The condenser and the evaporator have just the water and the water vapor, so the pressure is perspective to the saturated pressure to the saturated vapor water from the steam table. So:-

$P_1=P_5=0.8725$ kpa

$P_2 = 5.6291 = P_3 = P_4$

Finding the concentration in the solution (x1) in the point one is by dropping T1 and P1 on the chart (Appendix) and find the concentration of the solution (x3) by dropping T3 and P3 on the chart (Appendix), so we can find the values of

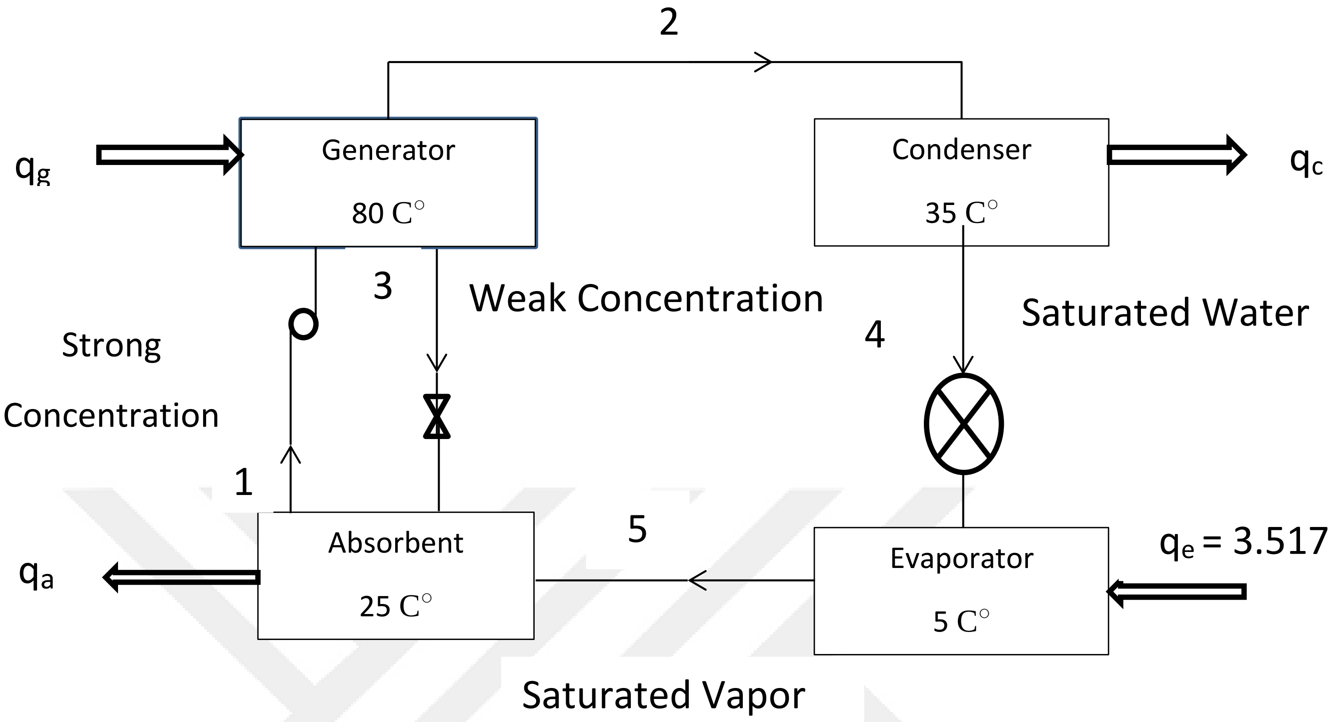


Figure (4.2) Schematic of Refrigeration System

x_1 = 0.535

x_3=0.6

To calculate the enthalpies in point one and point two from the (figure 4.2), so:

h_1= 75 kj/kg

h_3= 215 kj/kg

To find the enthalpies in point 4 and 5 for saturated water and saturated water vapor from the steam table, so:

h_4=164.64 kJ/kg

h_5=2510.1 kJ/kg

The enthalpy of the superheated vapor in the point (h_2) with simple equation:

h_2=h + (1.84*T2)

When h= 2510.1 kJ/kg taken from the table of saturated vapor so,

h_2=2648 kJ/kg

To find the mass of the refrigeration fluid which moved in the cycle is fixed, so we can find it by dividing the refrigeration required has the unit one kilo watt to the value of the refrigeration cycle and we can obtain it by:

$m_2=m_4=m_5= 3.517/(h_5-h_4)$

$= 0.0015$ kg/s

Then we can find the amount of mass proportional to the Bro- lithium- water solution which is distributed between the generator and the absorber submit to the mass equilibrium by solving two equations, the first one is mass equilibrium between generator and absorbent and the second equation is the total mass equilibrium of the cycle to gather.

$m1*x3 = m1*x1$ (1)

$m1= m2+m3$ (2)

m2= 0.0015 kg/s

m1= 0.0138 kg/s

m3= 0.0123 kg/s

Now we use the easy method to calculate the amount heat in all the parts for refrigeration cycle:-

(1) Evaporator for one ton amount of heat absorption

qe= 3.517 kw /T.R

(2) Condenser

$qc = m2*(h2 - h4)$

= 3.7241 kw /T.R

(3) Generator

qgen = ((m2*h2)+ (m3*h3) – (m1*h1))

= 5.5864 kw/T.R

(4) Absorbent

qa=((m5*h5)+(m3*h3)- (m1*h1)

= 5.3793 kw /T.R

Then we can find the total amount of heat added and rejected to/from the cycle

qadd = qgen + qe

= 9.1034 kw /T.R

qrej = qc + qa

= 9.1034 kw /T.R

So we get equilibrium of the cycle that (qrej=qadd)

4.3.3 The PTSC area needs per T.R

At last we can calculate the PTSC area that is needed for each T.R from absorption refrigeration system:-

qgen /qh

when

qgen = amount of heat that added in generator per T.R

qh = amount of heat that gain from the sun per m^2

Therefore,

qgen/ qh = (5.5864* 1000/ 74.2917) $(W/T.R)/(W/m^2)$

= 75 m^2/T.R (**which this is the main goal of our research)**

CHAPTER FIVE
RESULTS AND DISCUSSIONS

5.1 Results and Discussions

The voice that demands us to find a cheap, and available electrical power gives the researchers the motivation to develop absorption refrigerate system run by alternative power in the deserted places that has no access to the electric grid. This system will help us to dispense from fuel fossil, so we can reduce the environment contamination.

The main results of the present work are:

1) Heat gain and efficiency of the PTSC :-

   Iraq is one of the countries that has a lot of area that is unused and available. Iraq's geothermal environment has a good quality of solar radiation with Ib intensity radiation of 540 w/m^2

   Figures 5.1 shows the experimental results of the PTSC. The time evolution of the gage pressure is plotted. Within a 35 minutes of exposure to the sun. PTSC outlet pressure increased to 4 bar gage. In the same period total of 0.25 kg of water pass through the PTSC and was converted to vapor. The pressure was measured at a closed container attach to the end of the pipe.

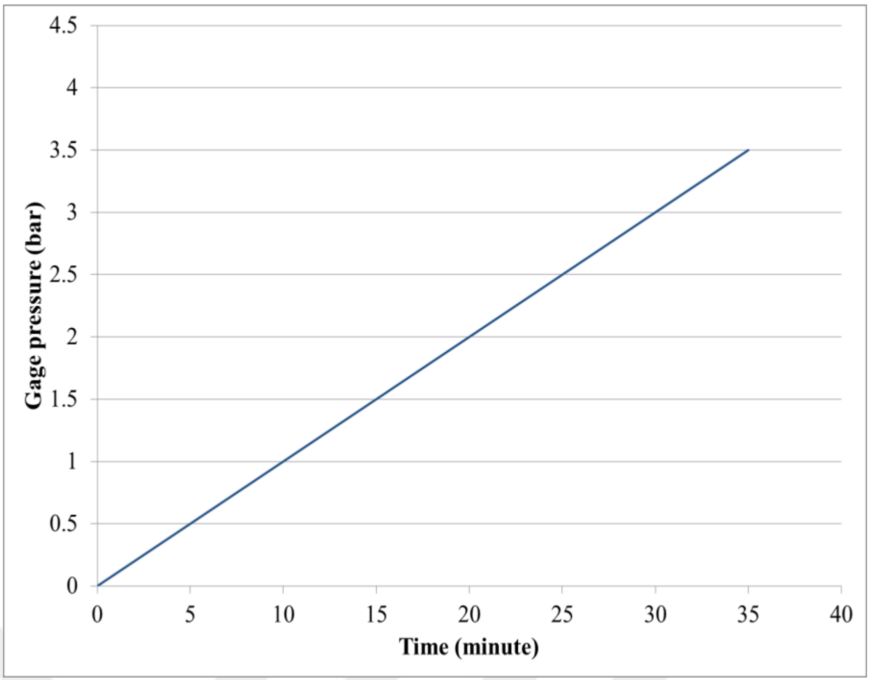


Figure (5.1) the outlet pressure increasing versus time for the water inside the pipe of the PTSC system

Figures 5.2 shows the temperature measurements of the PTSC. Within a 35 minutes of exposure to the sun, the PTSC outlet temperature increased to 136 C.

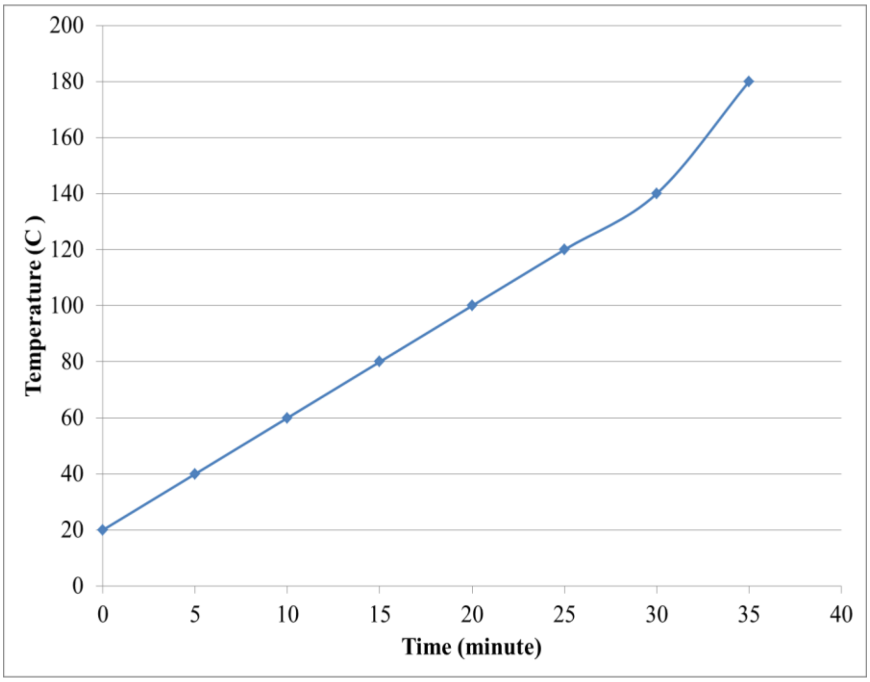


Figure (5.2) the outlet temperature increasing versus time for the water inside the pipe of the PTSC system

2) Absorption refrigeration system

The absorption refrigeration system requires a minimum inlet fluid temperature, at it's evaporator (see fig 4.2), of 80^o C. at this temperature the refrigerant has a sufficient energy to motivate itself through the cycle.

In order for the absorption refrigeration system to provide 1 T.R we need to compute the required added heat to the system. This heat to be obtained from the PTSC. The Matlab based simulation (see ch. 4) provides 9.1034 kw of heat added for 1 T.R. At steady state operation the heat rejected from the system is equal to the heat added.

$$q_{add} = q_{rej} = 9.1034 \ kw/T.R$$

The heat is added to the evaporator at Te = 5^o C. The temperature of the condenser is Tc = 35 C$^\circ$

2) The PTSC area needed to operate the system:

The view factor, coefficient of reflection, absorptivity and the temperature of the tube are the key factors that determines the efficiency of the PTSC system. The state of the art PTSC systems have efficiencies between 50- 54.6 % making them about 75% more efficient than our system. However, such system cost about 7 times more than our system.

Figure (5.3) shows cost and efficiency comparison between our system and the state of the art system. The cost efficiency is computed as follows:

$$\text{our } (\eta) = \frac{\text{altimate } \eta - \text{our}\eta}{\text{ultimate } \eta} *100$$

$$= \frac{55-13.75}{55} * 100$$

$$= 75\%$$

$$\text{Our cost} = \frac{\text{ultimate cost} - \text{our cost}}{\text{ultimate cost}} *100$$

$$= \frac{350\$ - 50\%}{350\$} *100$$

$$= 85\%$$

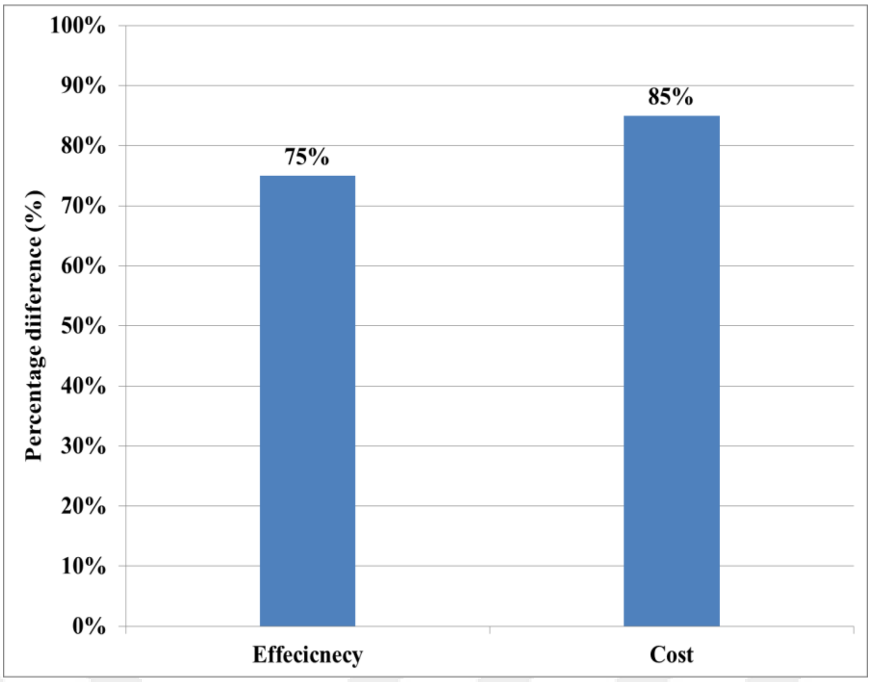


Figure (5.3) cost and efficiency comparison of our system with the state of the art system.

Alternatively, thermal and cost efficiencies could be computed via the following simple formulae:

our (η) = the optimal (η) / our (η)

= 25% from the optimal efficiency .

Our cost = optimal cost/ our cost

= 15% from the optimal cost

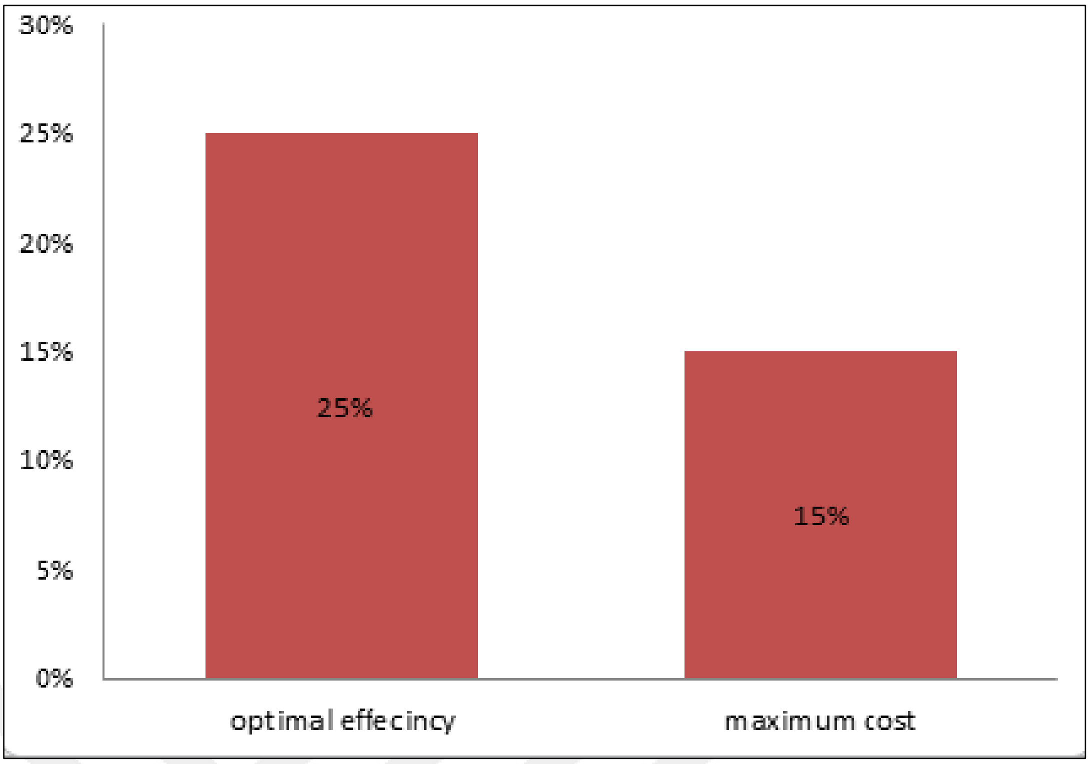


Figure (5.4) Cost and efficiency ratio to optimal cost and efficiency.

CHAPTER SIX
CONCLUSIONS AND RECOMMENDATIONS

Conclusions

Present goal was to come up with a functional design for a very simple and inexpensive refrigeration system that could potentially benefit the maximum number of people. Locally available and cheap material were exclusively used in this project. The design enabled easy assembly without special training and tools.

The research outcome shows that the PTSC efficiency was 13.75 %, which is low in comparison to the ultimate PTSC. However, the goal of this work is not so much an optimization of the efficiency as a single parameter, but to provide a highly accessible (in terms of coast and simplicity of manufacture) system with an acceptable efficiency. The area of PTSC per ton of refrigeration needed was 75 m^2/T.R. This area is a small fraction of total roof area of an average house in Iraq. Finally, these results show that the combination of PTSC and the absorption refrigeration system as a promising alternative system using renewable energy.

Recommendations

Using these results as a baseline, further improvements on the efficiency of the system is possible. The cost and simplicity should be used as constraints in future optimization to maintain the accessibility of this technology to the widest population.

REFERENCES

[1] Abdul Hai, S and Abdul Jabbar, K, 1984, "Iraq's National Paper,Solar in Iraq", Symposium using solar energy in Heating and cooling, Organization of Arab Petroleum Exporting and Solar Energy Research Center in the Scientific Research Council, Baghdad, Iraq.

[2] Sorensen B, 2004, “Renewable Energy" ,third edition by Elsevier Science.

[3] Sury R. K., Al Madani K., Ayyash S., 1984, “Choice of thermal energy system for solar absorption cooling, Solar Energy”, Vol. 32, N°2, pp. 181-187.

[4] Darkwa J., Fraser S., Cow D.H.C., 2012,” Theoretical and practical analysis of an integrated solar hot water-powered absorption cooling system, Energy”, vol.39, Issue1, pp. 395-402.

[5] Florides G. A., Kalogiro S.A., Tassou S.AWrobel., L.C., 2002, “Modeling and simulation of an absorption solar cooling system for Cyprus, Solar Energy”, Vol. 72, N°1, pp. 43-51.

[6] Gomri R., 2013, ” Simulation study of the performance of solar/natural gas absorption cooling chillers”, Energy Conversion and Management, Vol.65, pp-675-681.

[7] Yeung M.R., Yueu P.K., Dunn A., Cornish L.S., 1992, “Performance of a solar powered air conditioning system in Hong Kong, Solar energy”, Vol.48, N°5, pp. 309-319.

[8]obviously Rapp, Donald, “Solar Energy”, Prentice-Hall, Englewood Cliffs, NJ, 1981.

[9] Gavin D. J. Harper, "Solar Energy Projects", Copyright 2007 by the McGraw-Hill Companies, Inc. All right reserved. Manufactured in the USA.

[10] Duffie John A., and Beckman W. A., "Solar Engineering of Thermal Processes", 2nd edition. Wiley, New York, 1991.

[11] Pablo F. Ruiz, "European Research on Concentrated Solar Thermal Energy", © European Communities, 2004

[12] Philibert C., "The Present and Future Use of Solar Thermal Energy as a Primary Source of Energy", Copyright 2005 by InterAcademy Council All rights Reserved.

[13] NASA, "Surface Meteorology and Solar Energy – Available Table" Atmospheric science Data Center, 2008.

[14] Mukund R. Patel, "Wind and Solar Power Systems", Copyright 1999 by CRC press LLC, USA.

[15] Goswami D., Kreith F. and Kreider J., "Principles of Solar Engineering", 2nd edition. Taylor and Francis, Philadelphia, PA, 2000.

[16] Zekai Sen, "Solar Energy Fundamentals and Modeling Techniques" © Springer – Verlag London Limited 2008.

[17] Sol Wieder, "An Introduction to Solar Energy for Scientists and Engineers", Copyright © 1982, by John Wiely & Sons, Inc.

[18] Gavin D. Harber, "Solar Energy Projects for Evil Genius" Copyright © 2007 by The McGraw-Hill Companies.

[19] Geyer M., Stine B. William "Power from the Sun", Copyright ,2001 by J. T. Lyle Center.

[20] Braun J. E. and Mitchell J. C."Solar Geometry for Fixed and Tracking Surfaces" Solar Energy, 31 (1983) No. 5, 439-444.

[21] G. Barakos "Design, Simulation and Performance of Reflecting Parabolic Collector" Socrates Programme a European Summer School on System Engineering 18 July 2006.

[22] Omer O. Badran "Design and Implementation of Electro– Mechanical Sun Tracking Parabolic Trough", Desalination 220(2008) 669-676.

[23] Tomas Markvart, "Solar Electricity", Second edition, University of Southamton, UK, Madrid, Jon Wiley and sons Ltd. ISBN, 2000.

[24] Iqbal, M., "An Introduction to Solar Radiation", Academic, Toronto (1983).

[25] Donatelli M., Garlini L., and Bellochi G., "A software Component for Estimating Solar Radiation", Environmental Modeling and Software 21(2006) 411-416.

[26] Hottel HC. "A Simple Model for Estimating the Transmittance of Direct Solar Radiation through Clear Atmosphere". Solar Energy; 18 (1976) 129–134.

[27] Shah A., S. C. Kaushik, S.N. Garg, "Computation of beam Solar Radtiation at Normal Incidence Using Artificial Neural Network", Renewable Energy 31(2006) 1483-1491.

[28] John R. Howell, R. Bannerot, G. Vliet, "Solar – Thermal Energy Systems", Copyright by McGraw-Hill, Inc. 1982.

[29] Garg. H. P. and Prakash J., "Solar Energy Fundamentals and Applications", Copyright 2008, Tata McGraw-Hill Publishing Company Limited, Tenth reprint.

[30] Riveros H. G. and Oliva A. I., "Graphical Analysis of Sun Concentrating Collectors", Solar Energy, 36 (1986) No.4, 313-322.

[31] Jeter, Sheldon M. "Optical and Thermal Effects in Linear Concentrating Collectors", Diss. George Institute of Technology 1979.

[32] Dudley, V., Kolb, G.J., Mahoney, A.R., Mancini, T.R., Matthews, C.W., Sloan, M., and Kearney, D. Test Results: SEGS LS-2 “Solar Collector. Sandia National Laboratories”, SAND94-1884. December 1994.

[33] Lippke, Frank. “Simulation of the Part Load Behavior of a SEGS Plant.”, Prepared for Sandia National Laboratories, Albuquerque, NM, SAND95-1293. June 1995.

[34] Jacobson, E., Ketjoy, N., Nathakaranakule, S. and Rakwichian, W. "Solar parabolic trough simulation and application for a hybrid power plant in Thailand", 32 (2006) No. 2, 187-199.

[35] Garcia O. and Velazquez N., "Numerical Simulation of Parabolic Trough Solar Collector: Improvement using Counter Flow Concentric Circular Heat Exchangers", Journal of Heat and Mass Transfer 52 (2009) 597–609.

[36] Li M., L. L. Wang "Investigation of Evacuated Tube Heated by Solar Trough concentrating system", Energy Conversion and Management 47 (2006) 3591–3601.

[37] Egbo G., Sintali I. S., and Dandakouta H., "Analysis of Rim Angle Effect On the Geometric Dimensions of Solar Parabolic-Trough Collector", Int. Jor. P. App. Scs., 2 (2008) No.3, 11-20.

[38] Thepa S, Kirtikara K, Hirunlabh J and Khedari J "Improving Indoor Conditions of a Thai-Style Mushroom House by Means of an Evaporative Cooler and Continuous", Renewable Energy 17, (1999) 359-3693.

[39] A. Kahrobaian and H. Malekmohammadi, "Exergy Optimization Applied to Linear Parabolic Solar Collectors", Journal of Faculty of Engineering, 42 (2008) No. 1, 131-144.

[40] Norton B., D. E. Prapas, P. C. Eames, S. D. Propert, "Measured Performances of Curved Inverted-Vee, Absorbed Compound Parabolic Concentrating Solar – Energy Collectors", Solar Energy, 43 (1989) No. 5, 267-279.

[41] Soteris Kalogirou "Parabolic Trough Collector System for Low Temperature Steam Generation: Design and Performance Characteristics", Applied Energy, 55 (1996) No.1, 1- 19.

[42] Soteris Kalogirou " Use of parabolic trough solar energy collectors for sea-water desalination", Applied Energy 60 (1998) 65-88.

[43] Mazloumi M., Naghashzadegan M. and Javaherdeh K. "Simulation of Solar Lithium Bromide–Water Absorption Cooling System with Parabolic Trough Collector", Energy Conversion and Management 49 (2008) 2820–2832.

[44] Kalogirou, S.A.,"Solar Thermal Collectors and Applications". Progress in Energy and Combustion Science No. 30 (2004) 231-295. [45] Francis de Winter,"Solar Collector,Energy Storage, and Materials", MIT Press and AShRAE 1985.

[46] Lane, George A., "Solar Heat Storage: Latent Heat Material,"Volume 1. CRC Press, Boca Raton, FL, 1983.

[47] Mary Jane Hale, "Survey of Thermal Storage for Parabolic Trough Power Plants", NREL/SR-550-27925, Period of Performance: September 13, 1999.June 12, 2000.

[48] Senapati, R. K. (2015). "Design and Analysis of Absorption Refrigeration System Using H2o+[EMIM][TFA]", (Doctoral dissertation).

[49] Meinel AB, Meinel MP. Applied solar energy: an introduction. Reading, MA: Addison-Wesley; 1976.

[50]54]Soteris A. Kalogirou, "Solar thermal collectors and applications", Progress in Energy and Combustion Science 30 (2004) 231–295.

[51] Kryza, Frank, The Power of Light. McGraw-Hill, New York, 2003.

[52] Kreider JF, Kreith F., “Solar heating and cooling.”, New York: McGraw-Hill; 1977.

[53] Lof.G.O.G., "Energy Balance of Parabolic Cylindrical Solar Concetrator", J. of Engineering of Power (1962), 24-32.

[54] Lof.G.O.G., "Optimization of Focusing Solar Collector Design of Engineering for Power”, (1963), 221-228.

[55] Edward L. and Cherng J., "On Heat Exchanger used with Solar Concentrators", Jour.of solar energy, 18 (1976) 157-158.

[56] Parmpal.S. and Cheema L.S., "Performance and Optimization of Cylindrical-Parabola Collectror” of Solar Energy”, 18 (1976) 135-141.

[57] Ramsey J.W. and Gupta B.P., "Experimental Evaluation of a Cylindrical Parabolic Solar Collector” of heat transfer of ASME, 99 (1977) 163-168.

[58] Derrick G. A.,"A comparison of compound parabolic and simple parabolic concentrating solar collectors", of solar energy, 22 (1979).

[59] Clark JA, "An Analysis of the Technical and Economic Performance of a Parabolic Trough Concentrator for Solar industrial Process Heat Application", J. Heat Mass Transfer 25 (1982) 1427-1438.

[60] Jeter Sh, "Geometrical Effects on the Performance of Trough Collectors", Solar Energy,30 (1983) No. 2, 109-113.

[61] Pharabod F. and Philibert C., "Luz Solar Power Plants. Success in California and worldwide prospects.”, Tech. Rep. DLR for IEA-SSPS. SolarPACES; 1991.

[62] Thomas A., " Simple Structure for Parabolic Trough Concentrator", Energy Convers. Mgmt. 35 (1994), No.7, 569-573.

[63] Odeh SD, Morrison GL Behnia M., "Modeling of Parabolic Trough Direct Steam Generation Solar Collector", Solar Energy 62 (1998) 395-406.

[64] Volker Q. and Franz T., "Solar Thermal Power Plants for Hydrogen Production", Germany, (2001),

[65] Rafael Osuna, Antonio Esteban, (Spain), Michael Geyer and Paul Nava, (Solar International Germany)," EUROTROUGH – Parabolic Trough Collector Developed for Cost Efficient Solar Power Generation", (2002).

[66] Fauziah Sulaiman and Balbir Singh "Designing A Solar Thermal Cylindrical Parabolic Trough Concentrator by Simulation", Electrical and Electronics Engineering, Universiti Teknologi ETRONAS, Malaysia, (2003).

[67] Thomas Harms and Michael Brooks "Design, Construction and Testing of a Parabolic Trough Solar Collector for A developingCountry Application", Department of Mechanical Engineering, University of Stellenbosch, South Africa, (2005).

[68] Umamaheswaran M. "Solar Based Distillation System for Domestic Application", Ninth International Water Technology Conference, IWTC9 2005, Sharm El-Sheikh, Egypt.

[69] Miguel Domingo and Javier León "SENER Parabolic Trough Collector Design and Testing", SENER Ingenieríay Sistemas S.A.,ESPANA, (2006), Inc

[70] Valan A. and Samuel T., "Performance characteristics of the Solar Parabolic Trough Collector with Hot Water Generation System", Thermal Science, 10 (2006), No. 2, 167-174.

[71] Kassem T, "Numerical Study of the Natural Convection Process in the Parabolic Cylinderical Solar Collector", Desalination, 209 (2007); 144- 150.

[72] Dirk K., Yuvaraj P., Klaus H. and Mark S., "Parabolic Trough Collector Testing in the Frame of the REACt Project", Desalination 220 (2008) 612–618.

[73] Xiao, G. (2007). “Manual making of a parabolic solar collector”, Université de Nice.

[74] Pradeep Kumar, K. V., Srinath, T., & Venkatesh Reddy, A., 2013, ”Design Fabrication and experimental testing of solar parabolic trough collectors with automated tracking mechanism”, International Journal Of Research In Aeronautical And Mechanical Engineering, vol,1 Issue.4.

Appendix A

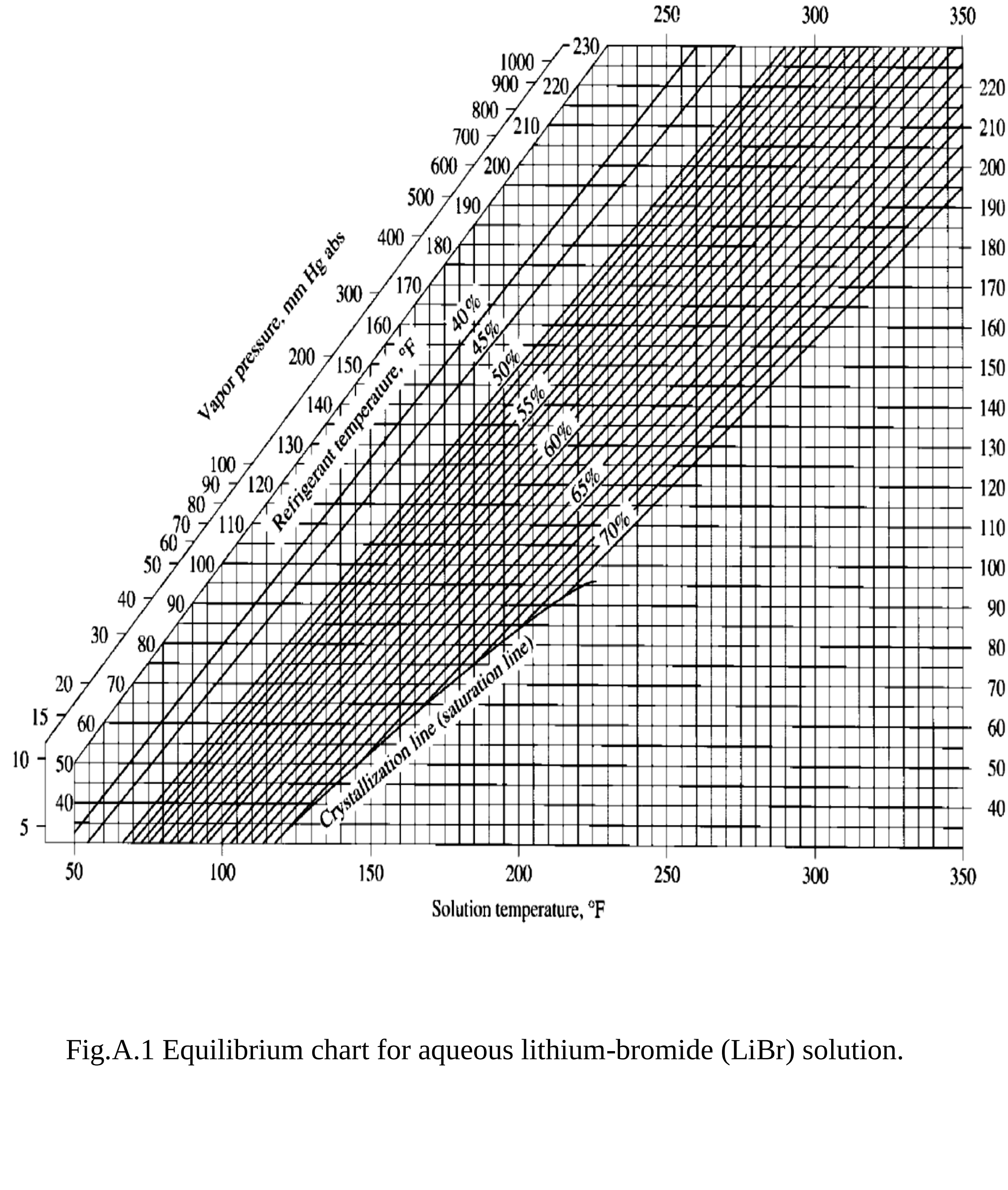


Fig.A.1 Equilibrium chart for aqueous lithium-bromide (LiBr) solution.

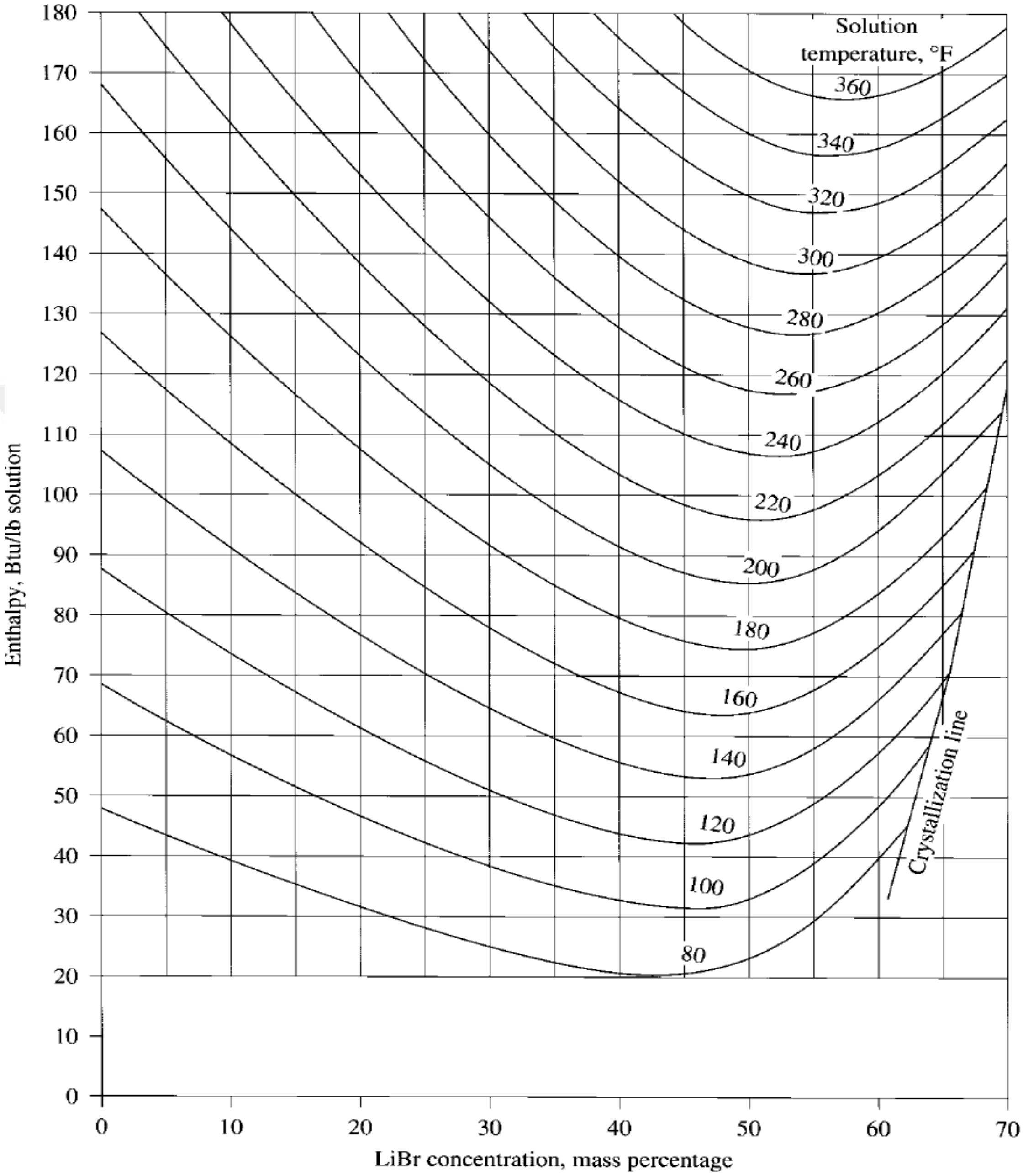


Fig.A.2 Enthalpy-concentration diagram for aqueous LiBr solution.

Appendix B

The code in Matlab

```
% calculate the solar absorber of parabolic trough solar collector

T1= 25            % in c degree initial temperature of inlet water
T2= 80            % in c degree initial temperature of outlet  water
m= 0.25           % in kg the mass inter the parabolic
A=2               %in m^2
t= 600            % the time in second
h1=104.81         % from saturated water in T=25 C,  we find enthalpy in  kj/kg
h2=461.41;        % from saturated water in T=80 C,  we find enthalpy in  kj/kg
Q=m*(h2-h1)       % in kj
q=Q/t             %in kj/s amount of heat generated per one second that mean kw
qh=((q/A)*1000)        % in w/m^2
Ib= 540                % w/m^2 intensity radiation.
n= ((qh/Ib)*100)       % n collector efficiency.
```

```
% calculating the absorption refrigeration cycle with those properities

T4=35 ;            %T in C degree in the  condenser.
T5=5 ;             %T in C degree in the evapourater.
T2=80 ;            %T in C degree in the generator.
T3=33;             %T in C degree in the absorbent.
P1=0.8725 ;        %P1 in kpa taken from table 2-1(water and satu. water ) at T5
P2= 5.6291 ;       %P2 in kpa taken from table 2-1(water and satu. water ) at T4
x1=0.535;          %x1 taken from chart 13.3 by droping T1 and P1.
x3=0.6;            %x3 taken from chart 13.3 by droping T3 and P3.
h1=75;             %h in kj/kg taken from chart 13.4 by droping x1 and T1.
h3=215;            %h in kj/kg taken from chart 13.4 by droping x3 and T3.
h4=164.64;         %h in kj/kg taken from table 2-1(water and satu. water).
h5=2510.1 ;        %h in kj/kg taken from table 2-1(water and satu. water).
h=2501;            %h in kj/kg taken from table 2-1(water and satu. water)at T0=0 C.
h2= h+ (1.84*T2 )
m2=(3.517/(h5-h4))       %  in kg/s
m4=m2;                   %  in kg/s
m5=m2;                   %  in kg/s
m1=m2/(1-(0.535/0.6))    %  in kg/s
m3=m1-m2                 %  in kg/s

qe=3.517                          %  in kw/T.R all calculates for one ton q evapourator
qc= m2*(h2-h4)                    %in kw/T.R amount of heat rejected on condeser
qa= (m5*h5)+(m3*h3)-(m1*h1)       % in kw/T.R amount of heat rejected on absopent
qgen= (m2*h2)+(m3*h3)-(m1*h1)     % in kw/T.R amount of heat add on generator
qadd= qgen+qe                     % in kw/T.R total amount of heat added to the cycle
qre= qc+qa                        % in kw/T.R total amount of heat rajected frpm the cycle
```