GAZIANTEP UNIVERSITY

GRADUATE SCHOOL OF

NATURAL & APPLIED SCIENCES

EXPERIMENTAL INVESTIGATION ON PERFORMANCE OF A HEATING AND COOLING SYSTEM WITH GROUND COUPLED HEAT PUMP

M. Sc. THESIS

IN

MECHANICAL ENGINEERING

By

AHMED SAADALLAH SALMAN

DECEMBER, 2012

Experimental investigation on performance of a heating and cooling system with ground coupled heat pump

M. Sc. Thesis

in

Mechanical Engineering

University of Gaziantep

Supervisor

Assoc. Prof. Dr. Recep YUMRUTAŞ

By

Ahmed Saadallah SALMAN

December, 2012

T.C.

UNIVERSITY OF GAZIANTEP

GRADUATE SCHOOL OF

NATURAL & APPLIED SCIENCES

DEPARTMENT OF MECHANICAL ENGINEERING

Name of the thesis: Experimental investigation on performance of a heating and cooling system with ground coupled heat pump.

Name of the student: Ahmed Saadallah SALMAN

Exam date: 31/12/2012

Approval of the Graduate School of Natural and Applied Sciences

Assoc. Prof. Dr. Metin BEDİR
Director

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science.

Prof. Dr. L. Canan DULGER
Head of Department

This is to certify that we have read this thesis and that in our opinion it is fully adequate, in scope and quality, as a thesis for the degree of Master of Science.

Assoc. Prof. Dr. Recep YUMRUTAŞ
Supervisor

Examining Committee Members	Signature
Prof.Dr.M. Sait SÖYLEMEZ	....
Prof. Dr. Hüsamettin BULUT	....
Assoc.Prof.Dr.Recep YUMRUTAŞ	....

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.

Ahmed Saadallah SALMAN

Signature

PREFACE

The need to conserve energy has become an everyday feature of our lives, at home, in offices and in factories. Energy saving is an area of activity which draws people together, as is shown by the activities of the International Energy Agency (IEA) and the European Economic Community (EEC) in funding joint energy projects.

A device which can make a significant contribution to energy conservation is the heat pump. By raising low grade (or low temperature) heat to a more useful temperature, "new" source of heat pump became available, for example ambient air, ground, soils, pond water or groundwater, solar, .etc. These new renewable and sustainable sources can couple the heat pump in order to amendment and reinforcement the efficiency and performance of the running system, In addition, their friendly direct impact on the environment. Hence, these commingled techniques can be one of the effectual key technologies involved in achieving the ultimate in the rational use of energy in the future.

The author hopes that this work can be access to identify the facts of matters and the real challenges, as well as the development and creation occasion workarounds and practical applications to get to climax of performance. And costs commensurate with newness decade. Furthermore, negative side effects on the environment.

Ahmed Saadallah SALMAN

ABSTRACT

EXPERIMENTAL INVESTIGATION ON PERFORMANCE OF A HEATING AND COOLING SYSTEM WITH GROUND COUPLED HEAT PUMP

Ahmed Saadallah SALMAN
M.Sc. in Mechanical Engineer
Supervisor: Assoc. Prof. Dr. Recep YUMRUTAŞ

December 2012, 122 pages

In this thesis, an experimental investigation on thermal performance of a space heating and cooling system with a Ground Coupled Heat Pump (GCHP) is performed. Therefore, a testing system under investigation was constructed in Laboratories site of the Mechanical Engineering Department, University of Gaziantep (37.1^{o}N), Turkey. The system was mainly consisted of a test room to be heated in winter and cooled in summer season, vertical slinky pipe buried in the ground, heat pump with variable capacity, circulation pump, heat exchanger, temperature and pressure measuring elements, water flow meter, data logger and a personnel computer (PC).

Thermal performance of the heating and cooling system was obtained by using measurement parameters for the room, slinky pipe, compressor, circulating pump and fan motors, condenser and evaporator. The parameters are temperatures and pressures of the refrigerant (R410A) circulated in the heat pump cycle, current and voltage for the each motor, temperature and mass flow rate of water-antifreeze circulated through the slinky pipe and heat exchanger, and outside and inside air temperatures. They were measured simultaneously during 24 hours in one minute interval, then collected data were transferred and saved to the PC. The measurement data for heating season from February 20 to April 1, 2012 and for cooling season from June 3 to September 1, 2012 were performed and analyzed. Average values of Coefficient of Performance for the GCHP (COP_{HP}) and the system (COP_{S}) were obtained as 3.8, 3.3 and 3.2, 3.0 for both seasons, respectively. The results show that this type of system can be used safely, reliably and efficiently for Gaziantep climatic conditions.

Keywords: Heat pump; Coefficient of Performance (COP); heat exchanger; heating; cooling.

ÖZET

TOPRAK KAYNAKLI ISI POMPALI ISITMA VE SOĞUTMA SİSTEMİ VERİMLERİNİN DENEYSEL İNCELENMESİ

Ahmed Saadallah SALMAN
Yüksek Lisans Tezi, Makine Mühendisliği Bölümü
Tez Yöneticisi: Doç. Dr. Recep YUMRUTAŞ

Aralık 2012, 122 sayfa

Bu çalışmada, toprak kaynaklı ısı pompalı bir ısıtma ve soğutma sisteminin ısıl verimleri deneysel olarak incelenmiştir. Bu yüzden, bu çalışmayı gerçekleştirmek için Gaziantep Üniversitesi Makine Mühendisliği Bölümü laboratuarları yanına bir test sistemi kurulmuştur. Test sistemi temel olarak kışın ısıtılacak yazın soğutulacak bir oda, spiral şeklinde toprak altı ısı değiştirgeci, değişken kapasiteli bir ısı pompası, devir daim pompası, ısı değiştirici ile debi, sıcaklık ve basınç ölçüm elemanları, veri toplayıcı ve bilgisayardan oluşmaktadır.

Isıtma ve soğutma sisteminin ısıl verimleri; oda, toprak altı ısı değiştirgeci, kompresör, pompa ve fan motorları, buharlaştırıcı ve yoğuşturucu ile ilgili parametrelerin ölçülmesi ile elde edilmiştir. Bu parametreler ise ısı pompası devresinde dolaşan soğutucu akışkanın (R410A) sıcaklık ve basınçları, her motor için akım ve gerilim, toprak altı ısı değiştirgecinde dolaşan antifirizli suyun debi ve sıcaklığı ile oda ve dış hava sıcaklıklarıdır. Bu parametreler 24 saat boyunca bir dakika aralıklarla ölçülmüş ve bilgisayar tarafından kaydedilmiştir. Ölçümler ısıtma mevsiminde 20 Şubat-1 Nisan 2012, soğutma mevsiminde ise 3 Haziran-1 Eylül 2012 tarihleri arasında gerçekleştirilmiş ve elde edilen verilerin analizleri yapılmıştır. Bu tarihlerde yapılan ısıtma ve soğutma testlerinde, ısı pompası (COP_{HP}) ve sistemin (COP_S) etkinlik katsayıları ortalama olarak sırasıyla 3.8, 3.3 and 3.2, 3.0 olarak elde edilmiştir. Bu sonuçlardan Gaziantep'in iklim şartlarında böyle bir sistemin emniyet ve güvenle, verimli bir şekilde kullanılabileceği anlaşılmıştır.

Anahtar Kelimeler: Isı pompası; etkinlik katsayısı (COP); ısı değiştirgeci: ısıtma; soğutma.

ACKNOWLEDGEMENTS

The presented dissertation would not have been possible without the guidance and the help of several individuals who in one way or another contributed and extended their valuable assistance in the preparation and completion of this study.

First and foremost, my sincere gratitude to my supervisor, Assoc. Prof. Dr. Recep YUMRUTAŞ whose sincerity and encouragement I will never forget. He has supported me throughout my thesis with his patience and knowledge. I attribute the level of my Master degree to his encouragement and effort and without him this thesis, too, would not have been completed or written.

I would like to thank Assist. Prof. Dr. Önder KAŞKA for his thoughts and suggestions during the progress of this thesis.

I would like to show my gratitude to the examining committee members for spending their valuable time for attending my M. Sc. Thesis.

I cannot find the right words to express to appreciation to my parents and family. They have been behind me and supportive throughout all of my studies. I would like to give my utmost gratitude to my parents for their valuable sacrifices in every stage of my life.

Lastly, I offer my regards and blessings to all of those who supported me (Emrah KARAYAZI, Serhat YILMAZ, Hanafi AŞGIN, Abdullah KAYMAKÇI, Feti YÜCEL) and the laboratory staff in any respect during the construction and completion of this project.

Table of Contents

LIST OF FIGURES

LIST OF TABLES

Page

NOMENCLATURE

Cp	Constant pressure specific heat of (Kj kg^{-1} $^{o}C^{-1}$)
k	Thermal conductivity of ground. (W/m $^{o}C^{-1}$)
To	Outside air temperature (^{o}C)
Ti	Inside room temperature (^{o}C)
COP	Coefficient of performance
h	Specific enthalpy of refrigerant (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{Q}$	Heat transfer rate (kW)
P	Pressure (bar)
s	Specific entropy (kJ/kg $^{o}C^{-1}$)
$\dot{W}$	Power (kW)
Tc	Condensation Temperature (^{o}C)
Te	Evaporation Temperature (^{o}C)
Pc	Condensation pressure (bar)
Pe	Evaporation pressure (bar)
Ic	Compressor power (kW)
Iv	Compressor voltage (V)
Ip	Circulating pump power (kW)
If	Fan power (kW)

GREEKS

α	Thermal diffusivity of the ground (m^2/s)
ρ	Density of the ground (kg/m3)
η	Isentropic efficiency of the compressor

Subscripts

o	Outside
i	Inside
e	Evaporator
c	Condenser
w	Water

Abbreviations

COP_{HP}	Coefficient of performance of the heat pump alone
COPs	Coefficient of performance of the whole system
HP	Heat pump
GCHP	Ground coupled heat pump
GCHPS	Ground coupled heat pump system
GWHPS	Ground water heat pump system
ASHPS	Air source heat pump system
AAC	Autoclaved Aerated Concrete
SEER	Seasonal energy efficiency ratio
EER	Energy efficiency Ratio

CHAPTER 1

1.1. INTRODUCTION

The world has witnessed a vast amount of technological developments and population growth since the industrial revolution; there has been a corresponding increase in the use of resources. Hence, society is becoming increasingly aware of the effects of pollution, toxic waste, global warming, deforestation, resources and ozone depletion. The global energy crisis has led to the development of a number of new low energy systems for building heating and cooling. These systems provide viable alternatives to conventional energy systems, and have the capability to significantly reduce electrical energy usage. Ground Source Heat Pump Systems (GSHPS) have received considerable attention in the recent decades as an alternative energy source for residential and commercial space heating and cooling applications. GSHPS have been the subject of many previous investigations, and have also found practical applications in the past. A Ground Source Heat Pump (GSHP) heating system uses a heat exchanger buried in the ground. The GSHP system has many advantages over the air source heat pump systems which are well known and commonly used in many applications. The advantages are mentioned in the following paragraphs.

Currently, GSHPS are perhaps one of the most widely used renewable energy resources. GSHPS use the earth's relatively constant temperature as a heat sink for cooling and a heat source for heating. From a thermodynamic perspective, using the ground as a heat source or sink makes more sense than the ambient air because the temperature is usually much closer to room conditions. Seasonal variation of temperature in the earth is small relative to the variation in air temperature. For that reason, the earth is normally at a more favorable source temperature than the outside air. Also, the use of liquid instead of air as the source/sink fluid for the heat pump cycle promotes higher efficiency, which can be attributed to the decrease in difference between the source/sink temperature and the refrigerant temperatures. In addition, the specific heat of water is more than four times greater than that of air.

Instead of producing heat through the combustion of fossil fuels, GSHPS function by concentrating naturally existing heat by collecting the Earth's natural heat using loops installed below the surface of the earth or submersed in a lake. The fluid circulating through the loop system is used to transfer the heat to the building. The distribution system of the building is then used to distribute the conditioned air to the various rooms. Furthermore, the heat deposited or rejected (depending on the season) from the home is collected by the means of the circulating fluid in the loops and return to the Earth. GSHPS applications are one of three categories of geothermal energy resources as defined by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) [1]. These categories are: (i) high temperature (>150^{o}C) electric power production, (ii) intermediate and low temperature (<150^{o}C) direct use applications, and (iii) GSHPS applications generally (< 32^{o}C). GSHPS applications are distinguished from the others by the fact that they operate at relatively low temperatures. The term GSHP has become an all-inclusive term to describe a heat pump system that uses the earth, ground water, or surface water as a heat source and/or sink. GSHPS in general consist of three loops or cycles as shown in figure 1.1 and 1.2. The first loop is on the load side and is either an air/water loop or a water/water loop, depending on the application. The second loop is the refrigerant loop inside a water source heat pump. Thermodynamically, there is no difference between the well-known vapor compression refrigeration cycle and the heat pump cycle; both systems absorb heat at a low temperature level and reject it to a higher temperature level. The difference between the two systems is that a refrigeration application is only concerned with the low temperature effect produced at the evaporator, while a heat pump may be concerned with both the cooling effect produced at the evaporator as well as the heating effect produced at the condenser. The third loop in the system is the ground loop in which water or an antifreeze solution exchanges heat with the refrigerant and the earth. GSHPS rely on the fact that, under normal geothermal gradients of about 30^{o}C/km [2]. The earth temperature is roughly constant in a zone extending from about 6.1 to 45.7 m deep [3]. This constant temperature interval within the earth is the result of a complex interaction of heat fluxes between the sun, atmosphere, and the earth. As a result, the temperature of this interval within the earth is approximately equal to the average annual air temperature [3].

The other important substances for the heat pump systems are heat transfer fluids. One of them is refrigerant which flows through the heat pump cycle. Generally, R134A, R410A and etc., which are discussed in Chapter 3 in detail, are used in the heat pump cycle [56]. But, type of heat transfer fluid in cycles between evaporator/condenser and ground, and also between condenser/evaporator and space sides are important. Water is the most commonly used as heat transfer fluid in heat pump applications because it possesses numerous desirable qualities, such as low cost, readily availability and high specific heat. The use of the water instead of air as the source/sink for the heat pump also promotes higher efficiency, which can be attributed to the decrease in difference between the source/sink temperature and the refrigerant temperatures. Moreover, the specific heat of water (4.18 kJ/kg ^{o}C) is more than four times greater than that of air [4].

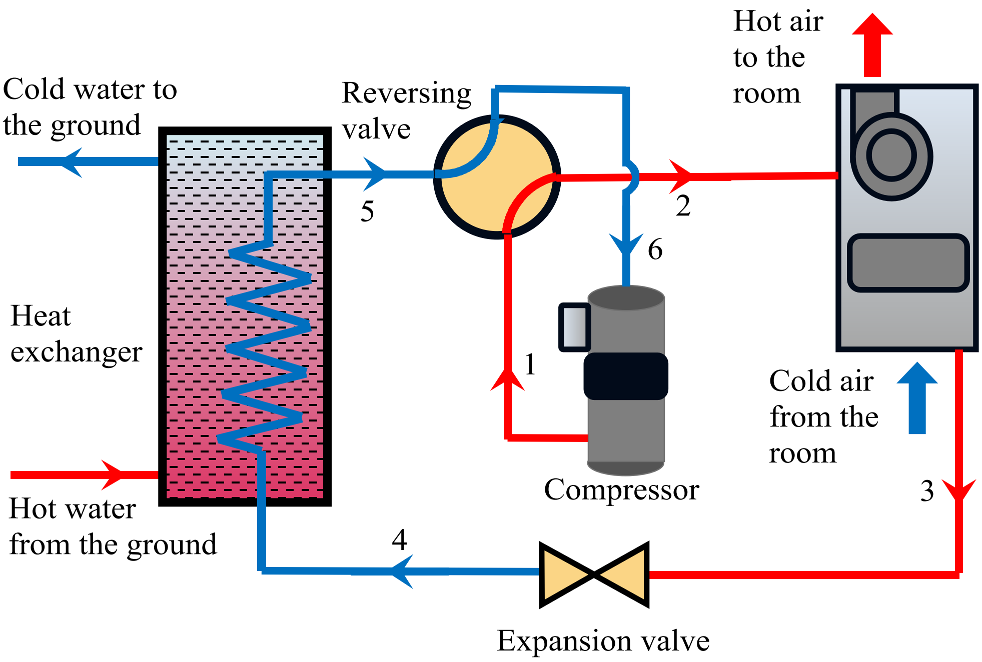

Figure 1.1. Schematic of GSHPS cycle in heating mode with fluid flow direction.

Another important advantage of the GSHPS is efficiency. Efficiency of a GSHPS is known as Coefficient of Performance (COP). COP's of the GSHPS are much greater than conventional air source heat pump systems [5]. Water source heat pumps tend to have a longer service life, as they are not subjected to refrigerant pressures as high or low as those of conventional air source heat pumps. A higher COP can be achieved by GSHPS because the source/sink earth temperature is relatively constant compared to air temperatures. The COP of vapor compression type heat pumps can be more than 3.0, when wasteful or renewable energy is used as the heat source. While in case heat source is supplied to heat pumps via water instead of air, the COP reaches even 4.0 to 5.0 [6]. The main reason of this phenomenon is the water medium has relatively high heat capacity than the air medium.

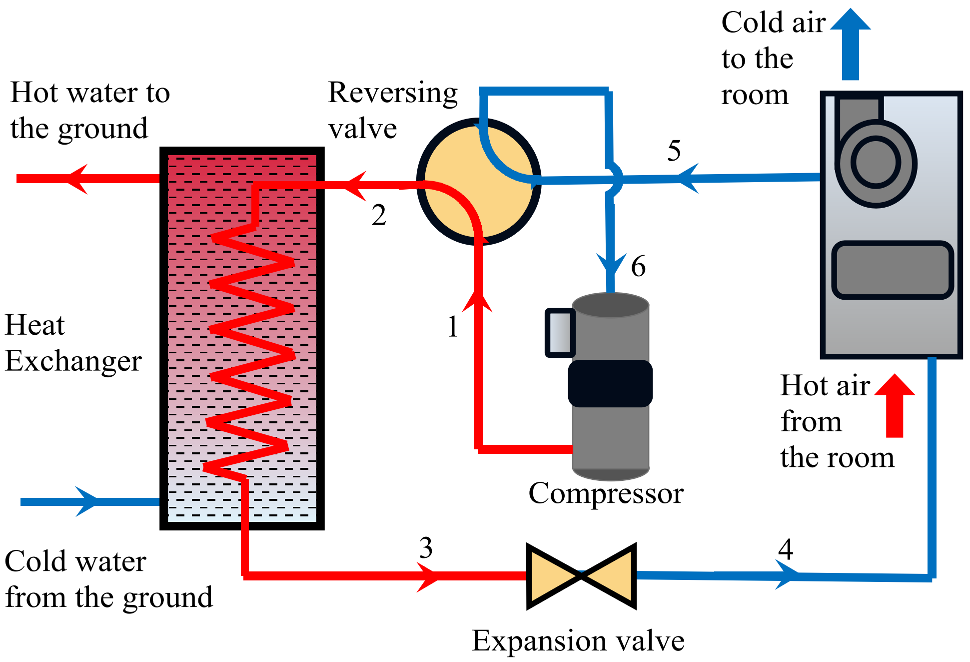

Figure 1.2. Schematic of GSHPS cycles in cooling mode with fluid flow direction.

GSHPS can play a more significant role in assisting the government reduces the CO_2 emissions associated with space and water heating, especially domestically, and with the right support encouraging recent growth trends should continue. Considering this, it is clear that the technology should be utilized alongside other low carbon heating methods such as combined heat and power and biomass, aided by greater utilization of energy efficiency measures, to reduce the carbon footprint of space heating/cooling, and water heating.

In addition of reducing purchased energy consumption and low CO_2 emissions, GSHPS have a number of environmental and operational advantages which are:

- High reliability (few moving parts, no exposure to weather).
- High security (no visible external components to be damaged or vandalized).
- Long life expectancy (20-25 years and up to 50 years for the ground coil).
- Low noise.
- Low maintenance costs (no regular servicing requirements).
- No boiler or fuel tank.
- No combustion or explosive gases within the building.
- No flue or ventilation requirements.
- No local pollution.

Nowadays, GSHPS are one of the fastest growing applications of renewable energy in the world, most of this growth occurs in USA and Europe, and also in other countries like Japan and Turkey. At 2010, the worldwide installed capacity was estimated at almost 14 GWth with an annual energy use of 24 TWh. There is around two million GSHPS units have been installed worldwide; with annual increases of 30% have occurred in about 40 countries over the past 10 years [7].

1.2. THESIS OBJECTIVE AND SCOPE

In the present study, experimental investigation on performances of the GCHPS for space heating and cooling is conducted. For this purpose, slinky type pipe in which water-antifreeze solution was flowed to extract or reject heat from or to the ground, was buried in vertical position, and a testing room was built. The testing room was constructed in the University of Gaziantep Campus (37^oN), Department of Mechanical Engineering, with 16 m^2 floor area and 2.7 m height (4 m width* 4 m length*2.7 m high) were constructed. The testing room is equipped with condenser or evaporator and measurement elements. The measurement elements are used to obtain nodal temperatures, pressures, voltage and currents. Temperatures and pressures of the refrigerant flowed through the heat pump cycle, voltages and ampere for the motors of pump, compressor and fans, ambient and inside air temperature with mass flow rate of the water-antifreeze solution were measured instantaneously for the heating and cooling operations. The performance parameters of the GCHP and GCHPS were calculated using the experimentally measured data. In addition, performance parameters of the heat pump were compared with those of similar heat pump systems with the same operating conditions. Heat extracted from the ground and heat rejected to the testing room with the power consumed by the compressor, circulating pump, and condenser or evaporator fan were calculated. Finally, the results were discussed and given graphically.

This thesis consists of six chapters which are organized in such a way that; in the second chapter, comprehensive literature survey of research on the subject is summarized. Historical background on the concept of using the ground as a heat source for a heat pump, types of GSHPS, basic ground coil design configurations, theoretical and experimental studies on the different coil or pipe buried in earth, and results from these studies, GSHP heating systems, energy extraction or rejection rates from or to underground heat exchangers are reviewed.

In the third chapter, there are explanations about thermodynamically operation of a heat pump, its basic components and basic mentality of heat sources at low temperature, the term Coefficient of Performance “COP”. Some issues such as how a unit is operated and integrated into a building heating system will be discussed alongside other related factors such as electrical requirements. Explanations about how a heat pump can reduce CO_2 emissions, how a heat pump technology operates and running costs are also outlined. Also, the myriad of different ground loop configurations available are given. Moreover, the importance of accurate design, sizing and considerations which need to be taken into account in estimating a borehole length to meet building demand with various methods are mentioned to produce a borehole length, coil diameter, etc. Estimations about technical developments in the field with various commercially available sizing programs are discussed.

Experimental setup for the GSHPS and system characteristics are presented in Chapter 4. In addition, calibrations of measuring sensors and instruments of the experimental setup are explained, and tables about the calibrations are given at the appendix of the thesis. Expressions for calculating performance of the GCHPS are also derived in this chapter. Heat transfer and energy requirement for components of the GSHPS such as condenser, evaporator, compressor, fan and circulating pump are expressed, and then COP_{PH} of the heat pump unit and COP_{SYS} of the GSHPS are expressed by using the heat transfer and energy consumption of the compressor, fan and pump.

Chapter 5 presents for the experimentally measured data that have been recorded during the heating and cooling seasons. The gathered data are analyzed and calculated. The results calculated are turned into graphics and tables in order to see and understand thermal behavior of the experimental GCHPS during heating and cooling seasons. In addition, some figures are depicted for comparing thermal performance of ASHPS and GWHPS. By these figures, their advantages and disadvantages of these systems under near climatic conditions are discussed clearly.

Conclusions of the experimental investigation and recommendations for further experimental study about the GSHPS for heating and cooling applications are presented in Chapter 6.

CHAPTER 2

BACKGROUND AND LITERATURE SURVEY

2.1. Introduction

Ground source heat pump systems (GSHPS) are efficient and alternative to conventional heating and cooling of any spaces or buildings because they utilize the ground as an energy source or sink instead of using the ambient air. The GSHPS is recognized to be prominent heating, cooling and water heating systems. They offer significant savings of electrical energy use and reductions in demand, provide high levels of comfort, need very low levels of maintenance and are environment friendly [8,9]. Ground behaves as a low grade heat storage medium which can employ thermal energy to heat pumps to upgrade the energy level and improve ground coupling energy exchange. The heat pumps known as GSHP's which extract or reject heat from or to the ground. These types of heat pumps have been commonly used since twenty years. But studies related with the GSHP and their systems have been continued since last fifty years. This chapter deals with literature review of developments in use of the ground energy by heat pumps and their systems, focusing on problems related to the designing, installing and performance of GSHP heating and cooling systems.

2.2. History of ground coupling heat pumps and development stages

Heat pump applications that use ground as a heat source have been introduced from a long time ago. The basic principle of the heat pump derives from the work of Carnot cycle in the early 19^{th} Century. Lord Kelvin first proposed a practical heat pump system or "heat multiplier" as it was then known, indicating that a refrigerating machine could also be used effectively for heating. In his system, he used air as the working fluid. The ambient air was drawn into a cylinder where it was expanded, thus reducing both its temperature and pressure. The air was then passed through an air-to-air type heat exchanger located outside where the cooled air was able to pick up heat from ambient air. Prior to being expelled into the building to be heated, the air was compressed back to atmospheric pressure, resulting in an increase in temperature above that the ambient level. Lord Kelvin claimed that his heat pump was able to produce heat using only 3 per cent of the energy which would be needed for direct heating [10]. Schematic representation of Lord Kelvin heat pump was indicated in the Figure 2.1.

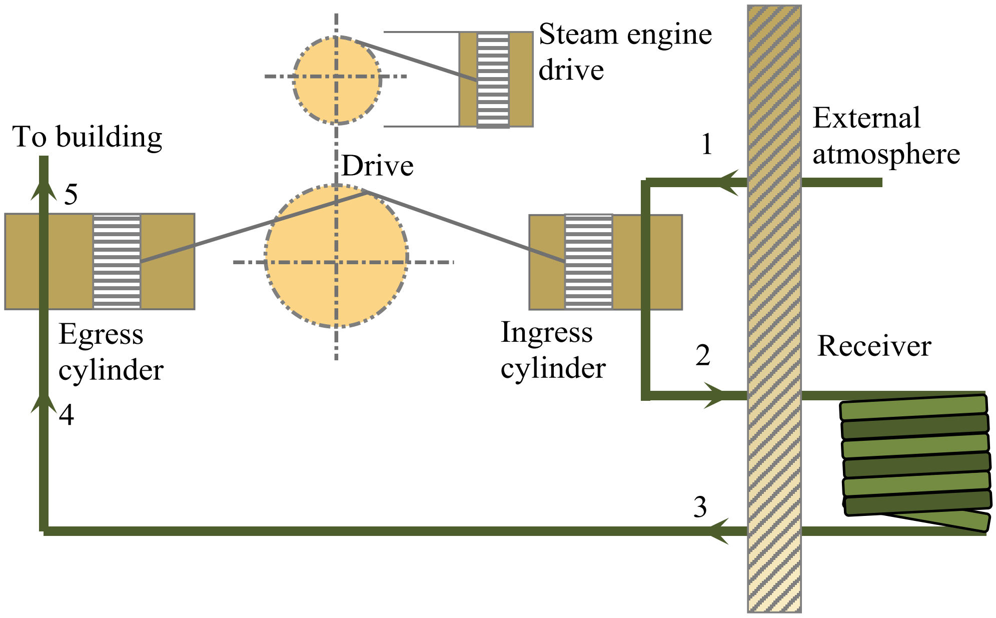

Figure 2.1. Layout of Lord Kelvin heat Multiplier [10].

Later, some works on domestic heat pumps were performed. One of the earlier domestic heat pumps operated successfully in the UK was installed in Sumner's [11] own house. Figure 2.2 shows schematic representation of Sumner's heat pump system. He used the same system used in Norwich electricity department for space heating [12]. The heat source was a river and the heat sink was circulating hot water and delivered at a temperature of 49 ^{o}C. In his heat pump system, Sulphur dioxide (SO_2) was used as the refrigerant, and COP of the order 3 was achieved. For Sumner heat pump, the heat was extracted from the outside air during the first few years of operating, but later a ground coil extracting heat from below ground surface at a depth of approximately 1m was used. Heat dissipation into the house was via copper pipes located in the concrete floor. Coefficient of Performance (COP) of his heat pump was achieved as 2.8.

Table 1.1. Heat pump development stages [13]

Location & Date	Heat source	Heat sink	Power in, kW	Power out, kW	Costs £/kWh		Mean COP
					Capital	Running	
Norwich office 1945	water	water	40-80	120-240	-	10.2	3
Royal Hall 1949	water	water	522	2700	103,20 1953	4200	1.5
Norwich house 1950	concrete	water	1.3	3.74	-	-	2.8
BEAIRA 1951	water	water	3	7-15	-	-	2.2-5
ERA 1952	soil	water	7.5	25	2,252 1952	89 1955	3
Fridge heater 1954	air	water	0.4	0.7-1.3	141 1954	-	3
Solar house 1956	air	air	-	6-12	325 1956	29 1956	-
Brantford house 1957	air	water	9	-	-	40	-
Nuffield EAO 1961	sewage	water	31	150	9,310 1962	896 1962	3.98

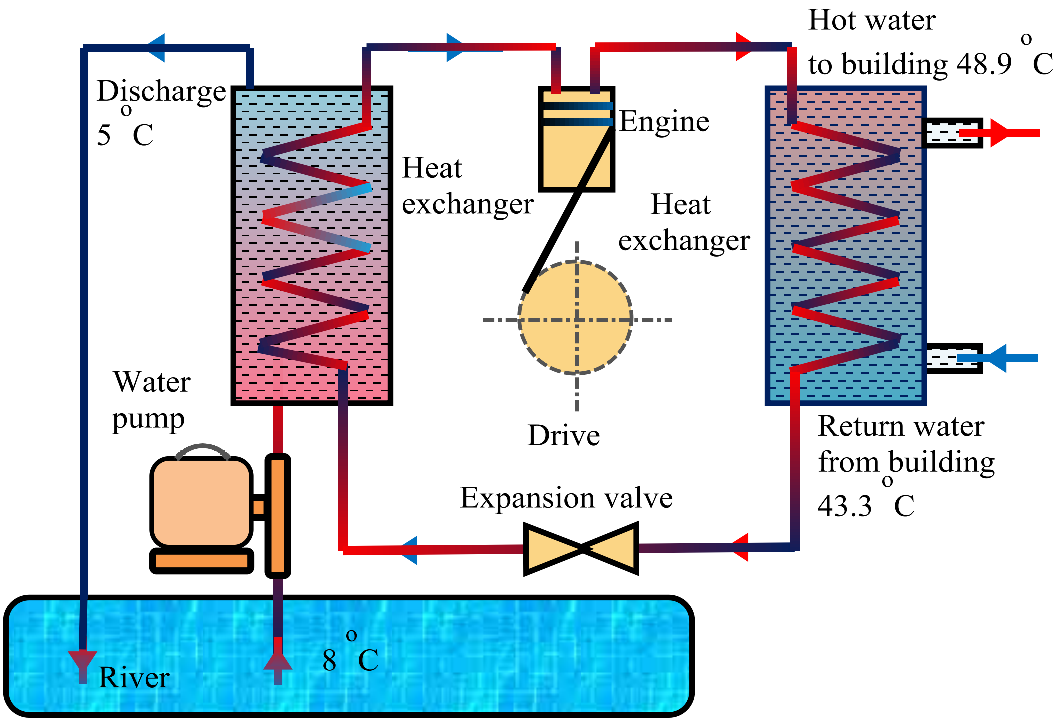

Figure 2.2. Schematic representation of Sumner's heat pump [12]

Some references state that first known heat pump using the ground as a heat source is in a Swiss patent issued in 1912 to Zoelly. Despite Zoelly's 1912 patent, it was not until the Second World War that serious ground coil development efforts began in both North America and Europe. This surge in interest lasted until the early 1950s when gas and oil became widely used as heating fuels. In the U.S.A. the first heat pump installed at Nuffield College, Oxford was conceived in 1950 [13]. Development stages of the heat pumps were given in his study and presented in Table 2.1. The system used low grade heat available in sewage as the heat source. The temperature of the sewage was between 16 and 24^{o}C, and its compressor was driven by a 31 kW diesel engine. In result of the experimental study, overall heat pump COP was obtained as about 4. The waste heat utilized in the heat pump system was extracted from exhaust heat of the engine to water circulated through a cycle between heat pump evaporator and the engine cooling jacket. The heat was distributed throughout the College via a hot water circuit at 49^{o}C, and provided 150 kW of the total design heating load of 450 kW.

After the Second World War, particularly following the energy crisis in the early 1970s was this concept revived for commercial use. Crandall [14] introduced the Zoelly's plan. Ingersoll and Plass [15] developed the classic theory for ground pipe heat conduction. They adapted Kelvin‘s heat source theory for flow of heat from soil to buried pipes. Hooper [16] reported the first research installation during this period in Canada. At the end of 1950s it was marked by installation of a large number of experimental ground source heat pumps. Results of these experiments published by Kidder and Neher [17] are examples of research efforts in the United States at this period. Besides encompassing practical demonstrations, the researches provided information on the heat extraction rates in practical applications.

2.3. Ground source heat pump system (GSHPS) applications in the world.

GSHPS applications are one of the fasting growing applications of renewable energy in the world [18], most of this growth happening in USA and Europe. Other countries such as Japan and Turkey are also taking off. By the end of 2004, the worldwide installed capacity was estimated at almost 1.7 million ground source heat pump systems and annual increases of 20% have occurred in about 30 countries over the past 10 years. In USA alone, over 100,000 ground source heat pump units are sold each year, with a majority of these for residential applications as installed units, 85% are closed loop systems (46% vertical, 38% horizontal) and 15% open loop systems.

Sanner et al. [19] reviewed the current status of GSHP and underground thermal energy storage in Europe. He gave some recent data for the number of installed heat pump units in some European countries. It is observed from figure 2.3 that the extremely high number for Sweden in 2002 is the result of a large number of exhaust-air and other air-to-air heat pumps; however, Sweden also has the highest number of GSHPS among the European countries.

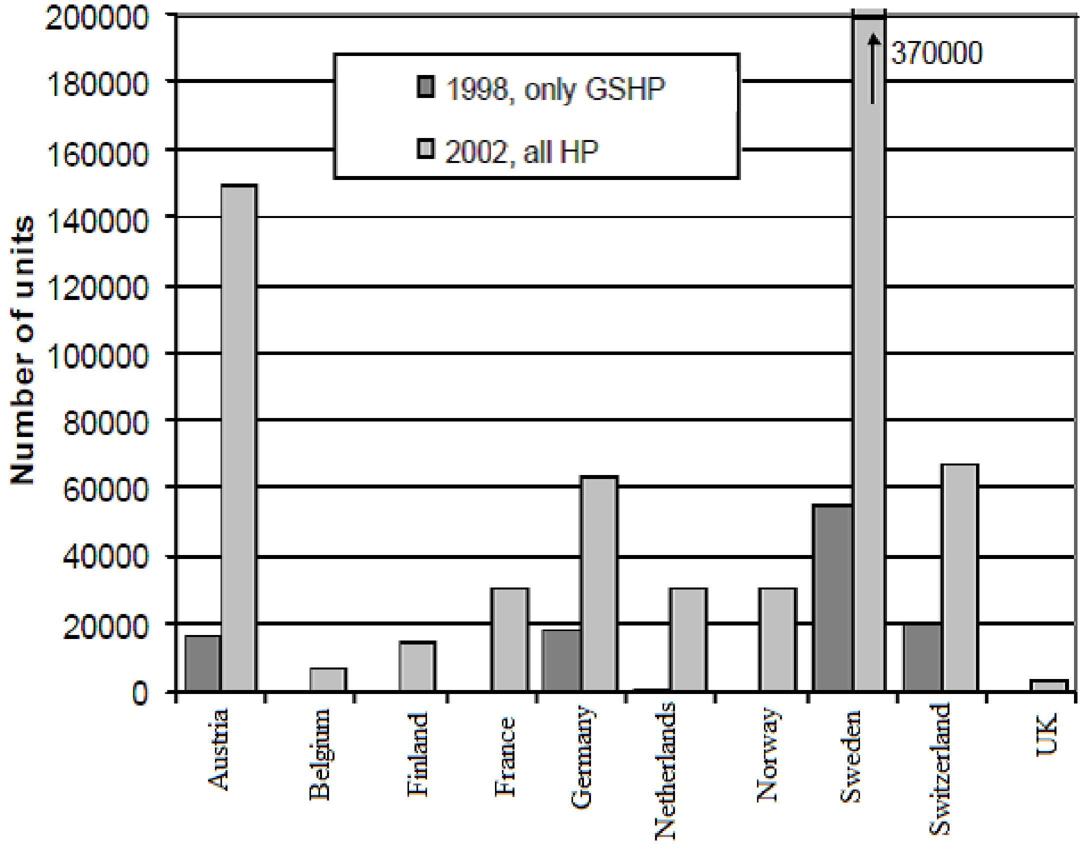

Figure 2.3. Number of installed heat pump units in Europe countries [19]

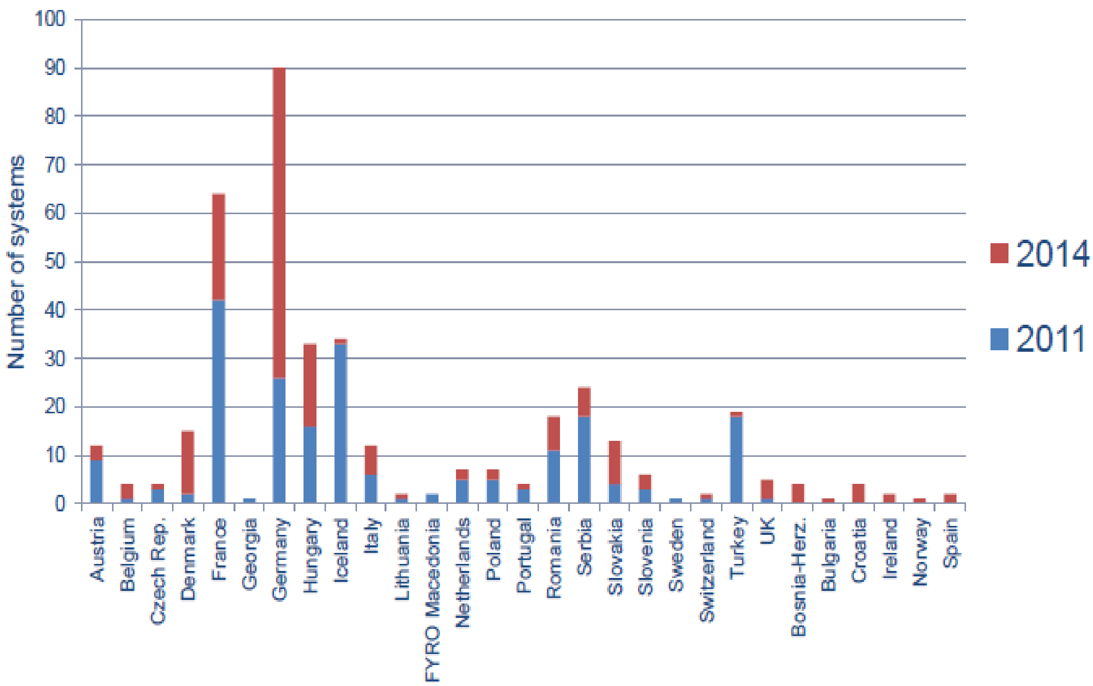

Figure 2.4. Number of installed heat pump units in European countries [20]

While figure 2.3 indicates the number of installed heat pump in European in 2001, it was clearly that Sweden has the largest share. Figure 2.4 clarify the new map for heat pump systems installation in Europe at 2011 and 2014. Germany, France, and Turkey countries are having growth significantly. In the meanwhile Figure 2.5 indicates the ongoing rising remarkable at 2011 for the installed capacity for heat pump systems in Europe. Again Turkey one of the advanced countries in this matter regards with a capacity about 700 MWth.

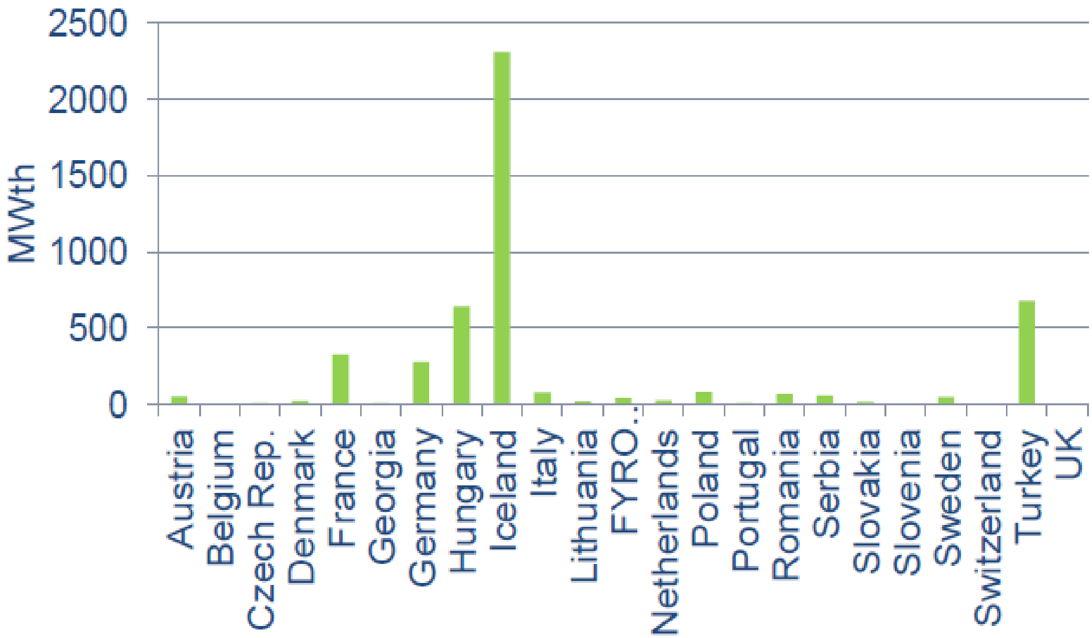

Figure 2.5. Number of installed capacity heat pump units in European countries [20]

2.4. Ground source heat pump (GSHP) systems

GSHPS include several different variations, which can be distinguished as having three main types of earth connection systems that have been adopted [1,21]. All of which reject heat and/or extract heat from ground as follow:

1. Surface Water Heat Pump (SWHP) systems. These use surface water bodies such as lakes, ponds and streams as a heat source and heat sink.
2. Ground Water Heat Pump (GWHP) systems. These use underground water as a heat source and heat sink, such as aquifer. Generally these kinds of systems divided into two systems which are:-

a. Standing column well (SCW) systems.

b. Open loop ground water (OLGW) systems.

3. Ground Coupled Heat Pump (GCHP) systems. These use the ground as a heat source and heat sink, with vertical, horizontal, or slinky ground heat exchangers.

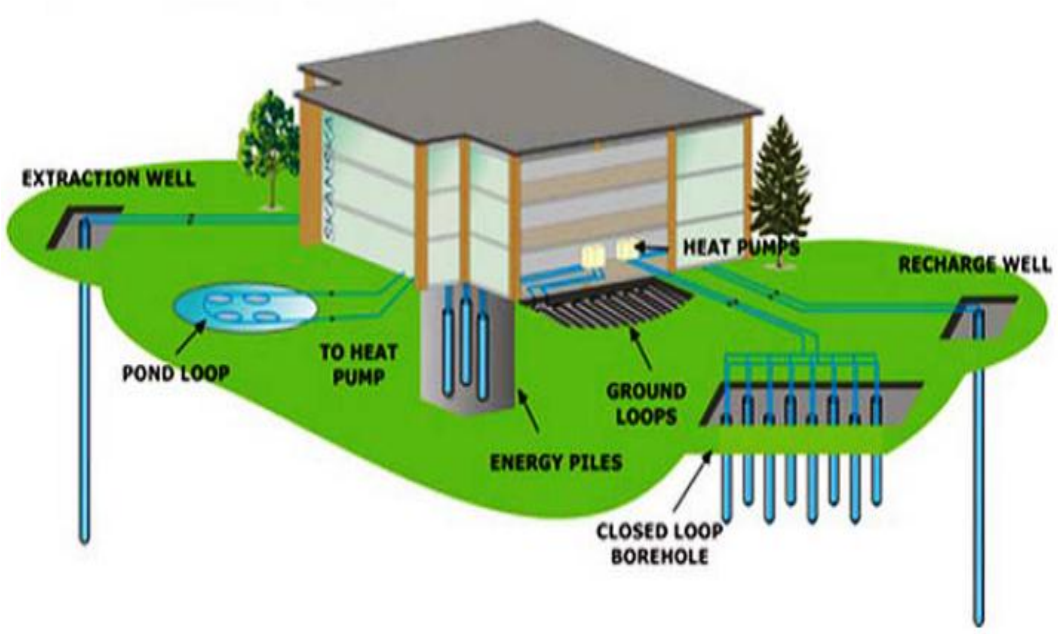

Figure 2.6. Schematic of GSHPS types [21]

2.4.1. Surface water heat pump (SWHP) systems

The SWHPs are a viable and relatively low cost option of GSHPS. A series of coiled pipes are submerged below the surface of a lake or pond as the heat exchanger. SWHPs can also be either closed loop or open loop systems. Open loop systems cannot be used in colder climates, thus an antifreeze mixtures are required to prevent freezing of the fluid. The heat transfer fluid is pumped through the pipes in a closed loop avoiding adverse impacts on the aquatic ecosystem.

2.4.2. Ground water heat pump (GWHP) systems

The GWHPs are open loop systems where they use a constant supply of ground water as the heat transfer fluid. A GWHP earth connection consists of water wells were ground water from an aquifer is pumped directly from the well to the heat pump's water connection to the refrigerant heat exchangers or to an intermediate heat exchanger. The heat exchanger transfers the heat from the open groundwater loop to a closed building loop. After leaving the building, the water is pumped back into the same aquifer via a second well referred to as the injection well. However, this type of GSHPS is not used where the water table is especially deep since the power required for pumping renders the system prohibitively expensive.

2.4.3. Ground coupled heat pump (GCHP) systems

In the GCHPS, a series of buried pipes circulates a heat transfer fluid in a closed loop, where the fluid never leaves the system but rather travels back and forth in a cyclic loop between the earth connection and the heat pump. This circulating fluid can be either water or an antifreeze solution if freezing temperatures are expected. The ground heat exchanger can make use of a series of deep vertical holes (boreholes) or a horizontal pipes arrangement. Horizontal collectors are generally more appropriate for small installations, where the collector pipe is buried in a trench at a depth between 0.25 to 1.8 m. A spiral coil can reduce the surface area required for horizontal earth connections and is sometimes referred to as a slinky coil installation. Vertical collectors are applied where land area is limited and require less pumping and pumping energy. The collector pipe is buried in a trench at depth between 15 to 150 m [22]. Knowing that working principle and the key portions are similar with presence pros and cons are different for each type. Below some of the experimental and theoretical work execute describes each type of GSHPS. Types of the GSHPS according to heat exchanger type buried underground are known as horizontal, vertical, and the slinky types which are explained shortly in the following subsections.

2.4.3.1. Vertical ground coupled heat pump systems

In the GCHPS with vertical borehole, ground heat exchanger configurations typically consist of one to tens of boreholes each containing a U-shaped pipe through which the heat transfer fluid is circulated through the U-shaped pipe. If land area is at a premium or the soil is too shallow for trenching a vertical borehole may be required for the ground loop in order to maximize heat gain for the space available. Fitting a vertical collector will be more expensive due to high drilling costs, but thermal performance of the GSHPS with U-shaped pipe will be higher than that of the others. The deeper the loop gives the more stable ground temperatures, and higher the system efficiency [23].

Hepbaşli [24] worked on an experimental study of a closed loop vertical ground source heat pump system. He used classical bin method for calculating the average heat gain or loss per hour, using R-22 in the heat pump cycle as working fluid. He obtained that overall COP of the system was 0.85 and the COP of the heat pump was record as 1.09. The substandard outcomes were extremely low when compared to other heat pumps operating under conditions at or near design values. The primary reasons for this situation were over sizing some system parts. Because some parts of the heat pump system was established with big sizes. The big size decreased the thermal performance of the system.

Ozyurt and Ekinci [25] made an experimental study on vertical GSHPS performance evaluation for space heating. The experimental apparatus mainly consisted of a ground heat exchanger, buried on the depth of 53 m, a liquid-to-liquid vapor compression heat pump. Experimental results were obtained during January to May within the heating season of 2007. The experimentally outcome results were used to compute the system performance. The COP_{HP} and COP_{SYS} were found to be in the range of 2.43 to 3.55 and 2.07 to 3.04, respectively. Their study indicates that these systems proposed could be used for residential heating in the province of Erzurum which is one of the coldest climate regions of Turkey.

Ozgener and Hepbaşli [26] conducted a study covers two various GSHPS namely a solar assisted vertical ground source heat pump and horizontal ground source heat pump. The study deals with the energetic and exergetic modeling of ground source heat pump systems for the system analysis and performance estimation. Tabulated results were given for both energetic and exergetic analysis of the systems. Average COPs estimate for the solar assisted vertical ground source heat pump were 3.64 for heat pump alone and 3.43 for whole system. While COPs of horizontal ground source heap pumps were 3.12 and 2.72, respectively. The exergy efficiency highest values for both whole systems on a product/fuel basis were in the range of 80.7% and 86.13%.

2.4.3.2. Horizontal ground coupled heat pump systems

The higher initial coast of installing of the GSHPS is one of the main causes for the horizontal ground coupled heat pump system to become increasingly important. It can be reduced installation cost compared to that of other GSHPS as no drilling is necessary and only a trench 1-2 m depth is required. Thermal characteristics of GCHPS with horizontal pipe are similar to those of vertical ones. The main difference is that horizontal ground loop heat exchangers are more affected by weather and air temperature fluctuations due to their proximity to the earth's surface. There have been many studies related with the GCHPS with horizontal pipe. Some of them and their results are given here.

Inallı and Esen [27] performed an experimental work to evaluate thermal performance of a horizontal GSHPS for space heating in Fırat University, Elazig, Turkey. The GSHPS was connected to a test room with 16.24 m^2 floor area. The heating and cooling loads of the test room were 2.5 and 3.1 kW at design conditions, respectively. The experimental results were obtained from November to April in heating season of 2002 - 2003. The average COP_{SYS} of the system for the horizontal ground heat exchanger in the different trenches, at 1 and 2 m depths, were obtained to be 2.66 and 2.81, respectively. It was evaluated that the COP_{SYS} values were low values for this type of heating system.

Babur [28] designed a single horizontal pipe coupled heat pump system operating with the Refrigerant R-12 on the ground to air basis and constructed using available equipment in the Mechanical Engineering Department, Middle East Technical University (METU). He performed 44 runs of experiments during the 1985–1986 heating season under varying climatic conditions to determine the COP and changes in soil temperatures. Approximately 10 m length of ground coil was installed at 1.5 m depth with spacing of 0.6 m. The COP_{SYS} value for heating was found to vary from 1.1 to 1.3.

Benlia and Durmuş [29] performed an experimentally study on a horizontal GSHPS with phase change material (PCM) with latent heat storage system were developed to use natural energy for thermal environment control of the greenhouse. COP_{HP} of the heat pump and the overall system performance COP_{SYS} have been determined based upon the measurements made on the field during the heating mode from 1 September 2005 till 30 April 2006 in Elazığ, Turkey, the average heating COP_{HP} of the GSHPS unit and the overall system COP_{SYS} were obtained to be in the range of 2.3 - 3.8 and 2 - 3.5, respectively.

2.4.3.3. Slinky ground coupled heat pump systems

It is also known as spiral loops, this is a different piping arrangement which can be used with either horizontal or vertical systems. Typically a slinky loop will require more pipes per W of heat (43 – 87 m, i.e. 12-25 W/m) extraction but conversely less trench space by 20 % [30], thus saving on installation costs due to the fact ground excavation is more expensive than the pipe material, which will be used frequently at the expense of other species.

There are many studies related with slinky type pipe used for evaporator or condenser of GSHPS. A first recorded slinky heat exchanger coupled GSHPS was developed by Bose and Smith [31] at Oklahoma State University. They indicated that the slinky heat exchanger enhanced the heat transfer area within a limited space when compared to a horizontal buried heat exchanger. Subsequently improving rise the system performance.

Yupeng Wu et al. [32] examine an experimental measurement of slinky horizontal ground coupled heat exchangers system. Results from the experimental measurements of GSHPS at Talbot Cottage at Drayton, Oxford shire, UK climate were presented. Thermal performance for different coil diameters and slinky interval distances was investigated. Results from a two month period of monitoring the performance of the GSHPS clarify that the COP decreased with the running time. The average COP of the system was 2.5. Also, there was no significant difference in the specific heat extraction rate of the slinky heat exchanger at different coil diameters. However, the larger the diameter of coil, the higher the heat extraction rate per meter length of soil.

Doherty et al. [33] installed and undertook experimental measurements of slinky horizontal ground heat pump system GCHPS installed at the University of Nottingham, UK. The experimentally measurements for three months on the cooling seasons were record and COP_{HP} and COP_{SYS} were found to be approximately 2.7 and 2.3 respectively.

2.5. GSHPS and ground heat exchanger (GHE) heat flux

When circulating fluid on both heat pump cycle normally (Refrigerant used) and the ground loop cycle normally (water or water antifreeze used) there will be a heat extraction or gain from the refrigerant or the water antifreeze depending on the weather condition. This energy should be known as it is vital for correct sizing of ground heat exchanger and heat pump itself to achieve an accurate prediction of their long term performance over their useful working life. There are several investigations on the GSHP and GHE in order to determine heat extraction or gain from or to the ground. In this subsection, some of them are summarized.

Penrod [34] obtained a heat flux of up to 50 W/m for 25 mm diameter copper pipes buried horizontally at a depth of 1.5 m.

Vestal and Fluker [35] achieved similar heat extraction rate for copper pipes 13 and 25 mm in diameter buried at a depth of 1.6 m in various soils in Texas, they extracted up to 100 W/m with larger 50 mm diameter copper pipes during 7 days.

Griffith [36] determined rates of heat extraction using horizontal copper pipes buried in clay at depths of 1.25, 1.8, and 2.5 m in London, the extraction rates was 30-60 W/m for steady state conditions and 2 hour transient heat flows respectively.

Jalaluddin et al. [37] worked on experimental study for several types of ground heat exchanger systems GHES installed in a steel pile foundation; including double tube, U-tube, and multi tube GHES. The test was carried out at Saga University. The performance of the installed GHES was investigated under actual operation in the cooling season with water flow rates of 2, 4, and 8 liters/min. He found that the double tube had the highest rate of heat exchange among the multi tube and U-tube GHES. The average heat exchange rate of GHES over 24 hours continuous operation with a flow rate of 4 l/min was 49.6 W/m for the double tube while it was 34.8 W/m for the multi tube, and 30.4 W/m for the U tube. Increasing flow rate increased the heat exchange rate of the GHES.

2.6. Types of refrigerant used in the GSHP cycles

To ensure better system performance on of the major items that should be accurately chosen is the type of refrigerant used in the heat pump cycle. The refrigerant must meet high temperatures without unduly high pressure. The following factors should be considered when selecting the refrigerant to use as the working fluid [38]:

- Evaporator/condenser, and compressor pressures; the lower compressor pressure ratio the higher the efficiency. The pressure in the evaporator must be above atmospheric pressure.
- An ideal refrigerant will have a high enthalpy of transformation (ratio of latent heat to specific volume) while vapor will have a low specific volume at the evaporator and thereby decrease compressor work and size.
- Ratio of latent to sensible heat ideally the maximum amount of liquid refrigerant should be converted to vapour at the temperature in the evaporator coil.

There have been several studies about selection of better refrigerants for each heat pump cycles used for heating and cooling applications. Summary of results for the some of these studies are presented in the subsection.

P.C. Zhao [39] performed an experimental study to overcome the biggest barrier of findings a proper working fluids, which work in the condition of 80-100 ºC as condensation temperature and 25-35 ºC as evaporation temperature and at the same time keep the highest pressure in the system under 2500 kPa. Experimental results indicated that the new working fluids R123/R290 (50/50 by mass %) and R123/R290/R600a (40/50/10 by mass %) will meet the requirement of the new working condition. The COP was above 3.0. This result is quite good for condensation temperature above 80 ºC and a condensation pressures below 2500 kPa.

Man-Hoe [40] presents results of performance tests for R-22 and four alternative fluids (R134a, R 32/134a (30/70 %), R-407C, and R-410A) at operating conditions typical for a residential air conditioner. The study was performed in experimental breadboard water to water heat pump in which a water/ethylene glycol mixture was used as the heat transfer fluid. The zeotropic mixtures, R07C and R32/R134a (30/70) have the closest performance characteristics to R-22, with R32/R134a having a slightly better COP. The important operating parameters (evaporator, condenser pressures and compressor discharge temperature) did not deviate significantly from the R-22 values. The binary near-azeotrope R-410a, displayed a 44 % higher capacity than R-22 when tested at the same compressor rpm.

Yılmaz [41] presented a comparison between the pure refrigerants R-12, R-22, R-114 and mixtures of R-12/R-22 (mass fractions of R-12 are 37.5 %, 50 %, 62.5 %, 75 %) and R-12/R-114 (mass fractions of R-12 are 25 %, 40 %, 50 %, 75 %), on the performance analysis of an air to water vapor compression heat pump system using pure refrigerants and zeotropic refrigerant mixtures. He noticed that COP of the system is generally increasing as the evaporator source inlet temperatures increases. He remarked that COP depends significantly on the mixture ratio, as an example, the maximum COP occurs at a mixture ratio of around 75–25 % R12/R22. It is important to evaluate refrigerant mixtures which do not deplete ozone and have no greenhouse effects. Moreover, zeotropic refrigerant mixtures would have smaller ozone depletion potential, minimum greenhouse effect, and higher COP.

2.7. Comparison of performances for GSHPS

Esen et al. [42] carried out a study on a performance comparison between a ground coupled heat pump system GCHPS and an air source heat pump system ASHPS. The two systems were connected to a laboratory test room in Firat University, Elazig, Turkey. The testing rooms were designed and constructed for space cooling. The performances of the GCHPS and the ASHPS were experimentally determined. The experimental results were computed from June to September of 2004. The average cooling COP of the GCHPS for the horizontal ground heat exchanger system HGHES in the different trenches at 1 and 2 m depths were found to be 3.85 and 4.26 respectively, and that for the ASHPS was determined to be 3.17.

Lam and Wilco [43] examine two case studies were conducted to assess the energy performance of air to water and water to water heat pumps in two city hotels in Hong Kong. The former provided heating for an outdoor swimming pool during the heating season while the latter was used to complement an existing boiler system for hot water supply. The COP of the air to water heat pump varied from 1.5 to 2.4 during the 6-12 month heating season. The water to water heat pumps operated with a mean COP of 3.0. It was obtained that the COP for the water to water type heat pump was higher than that of the air to water type. The payback period was just about two years. Although the case studies were conducted in subtropical Hong Kong only, it was believed that the seasonal and yearly performance found, and the COP with payback periods estimated from this study can give a good indication of the likely energy performance and cost implication of heat pumps in locations with similar development and prevailing climates.

Yonghui Guo et al.[44] made an experimentally study on a comparison between a direct expansion ground source heat pump system (DX-GSHPS) and a secondary loop ground coupled heat pump system (SL–GCHPS). The DX-GSHPS and SL-GCHPS were designed and installed in parallel for the same space cooling load in a demonstrating building in Hunan province, China. The performances of the two systems were experimentally determined from June to September in cooling season of 2009. The average cooling performance coefficients COP_{SYS} of the DX-GSHPS was obtained to be 6.03, while the COP_{SYS} for the SL-GCHPS was determined to be 5.63.

Kara [45] investigated both theoretically and experimentally the utilization of low temperature resources for space heating of a health resort center in Erzurum, Turkey. By using a GSHPS coupled to geothermal wells, the temperature of the disposed water from the baths is around 30 to 35 ^{o}C. Considering these temperature limits, Kara designed a water to water GSHPS running with R-22 space heating and developed a computer simulation for the system, water produced at 45^{o}C temperature for a floor heating system by using the geothermal resource at 35^{o}C. He obtained that COP for heating was 2.8, and results from computer simulations were in good agreement with experimental results.

Swardt and Meyer [46] compared performance of a reversible ground coupled heat pump system (GCHPS) to a municipality water reticulation system with simulations to a conventional air source heat pump system (ASHPS) for space cooling and heating. The experimental set up was operated during five years in Pretoria, he depicted variation of monthly ambient air and ground properties, and annual cooling COP for air and ground system for different depth, as indicted in figures 2.7 it is obvious from the figure for a depth between 0.13 m and 1 m the COP for the cooling operation changes from 2.4 and 2.7, and COP for the heating operation changes from 3.4 and 3.7. Difference between heating and cooling system COP is approximately equal to 1; that is reasonable value. When the earth depth increases, COP for the both system increase. Since, heat loss increases when the underground pipe approaches to the ground surface. This figure indicates that depth of the pipe should be higher than 1 m for a reasonable COP for both systems.

Bose [47] performed a performance comparison between GSHPS and ASHPS at low ambient air temperatures. He obtained that the GSHPS had significant capacity and efficiency improvements over ASHPS. The comparison revealed that the deeper the municipality ground source water the greater the improvement in capacities and COPs. GSHPS will have a payback period of less than 2 years. It was concluded that the utilization of municipality water reticulation systems as a heat source or sink was a viable method of optimizing energy usage in the air conditioning industry. Figure 2.8 shows that COP of the ground system varies between the values of 3.4 and 4.3 as dependence of ground depth, also amplitude of the COP decreases with increasing the depth. Amplitude of COP for the ASHPS is higher than that of the GSHPS for all depths. Amplitude of COP for the GSHPS increases with depth of the pipe buried underground.

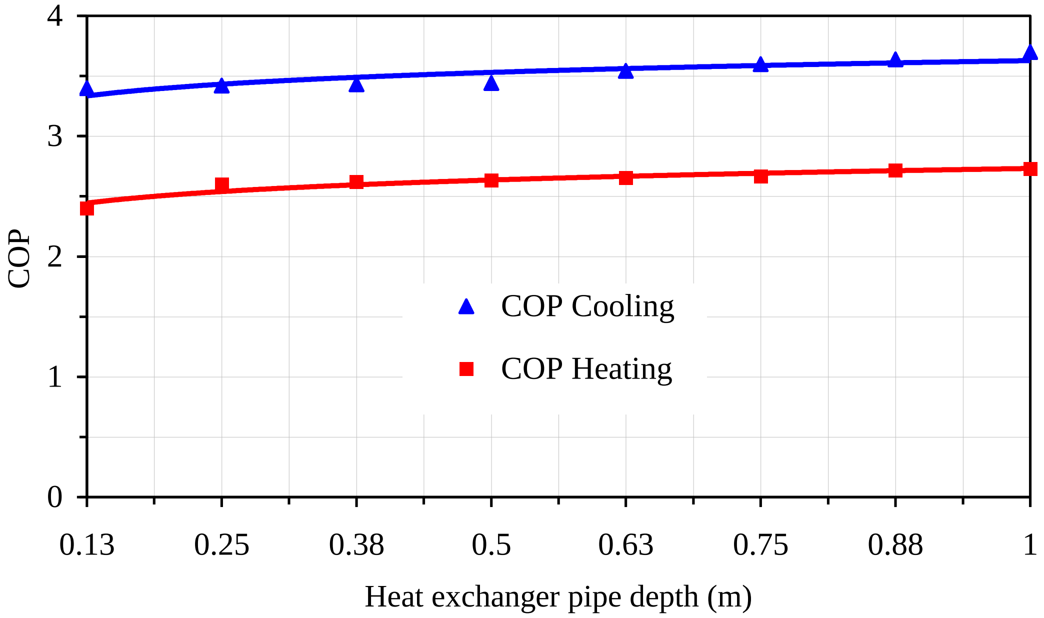

Figure 2.7. Effect of ground heat exchanger pipe depth on COP [46]

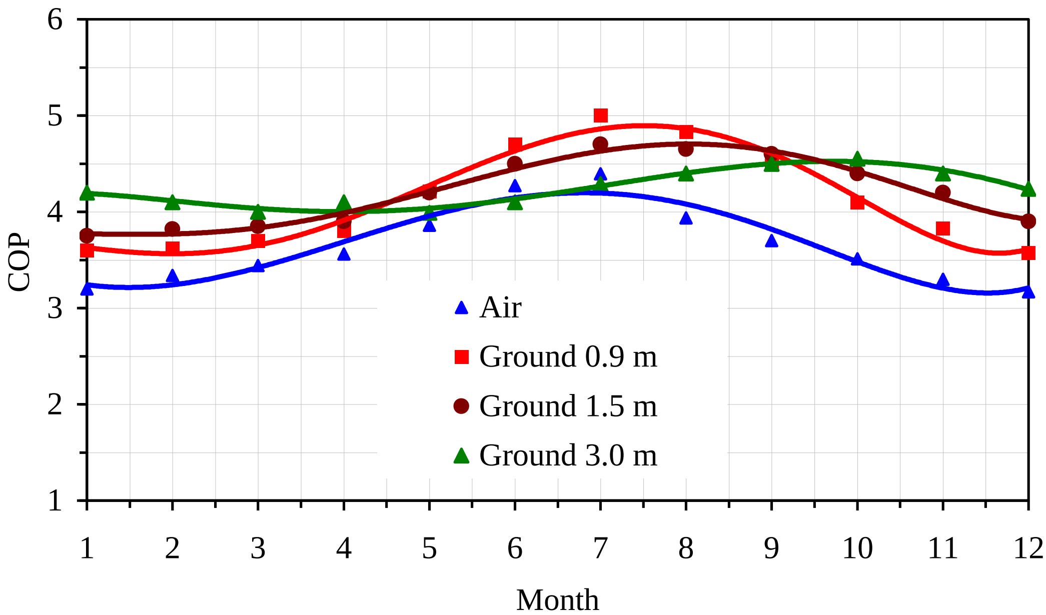

Figure 2.8 Annual cooling COP for air and ground systems [47]

Healy and Ugursal [48] investigated a typical GSHPS for space heating and cooling of house, which is similar to our experimental study given in this thesis. Figure 2.9 represents a schematic of their system. They stated that in cold climates such as in Canada, GSHPS represented a viable alternative to conventional space heating and cooling systems than ASHPS because of their higher operating efficiency, especially during the heating season. COP of the system for cooling operation was 3.6 due to the relatively constant ground and water temperatures with respect to ambient air temperature as shown in figures 2.10 and 2.11. It seen from the figure 2.10 that amplitude of the ground water temperature is lower than that of the others. Also mean temperature of the ground water is higher than that of the ground and air temperatures. Variations of COP and power consumed are seen in figure 2.11 when the power consumed increases, COP decreases with ground heat exchanger area.

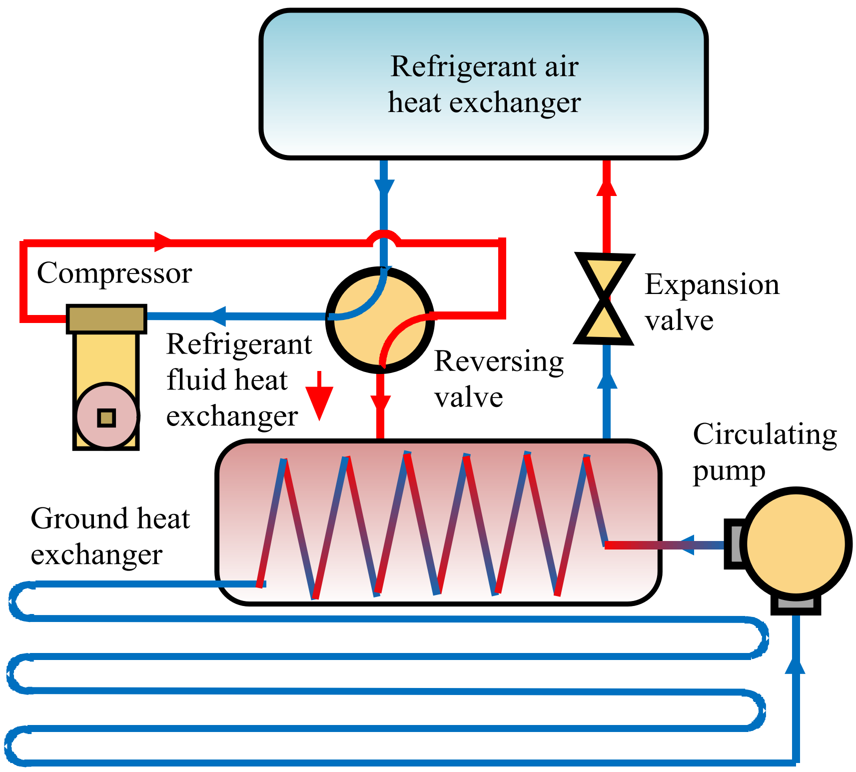

Figure 2.9. Schematic representation of GSHP [48]

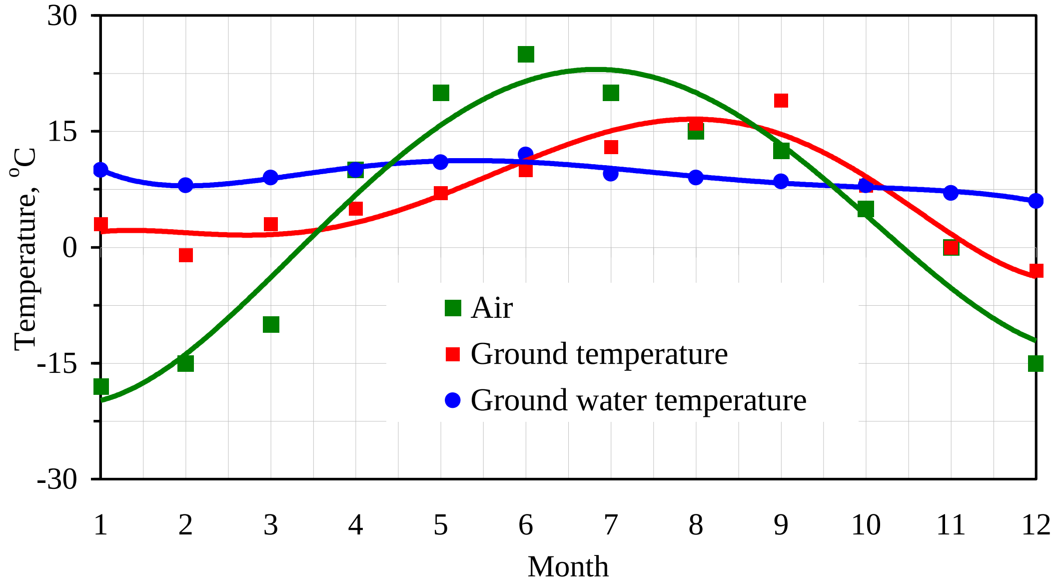

Figure 2.10. Annual temperature distributions of ground water, ground and ambient air for Nova Scotia, Canada [48]

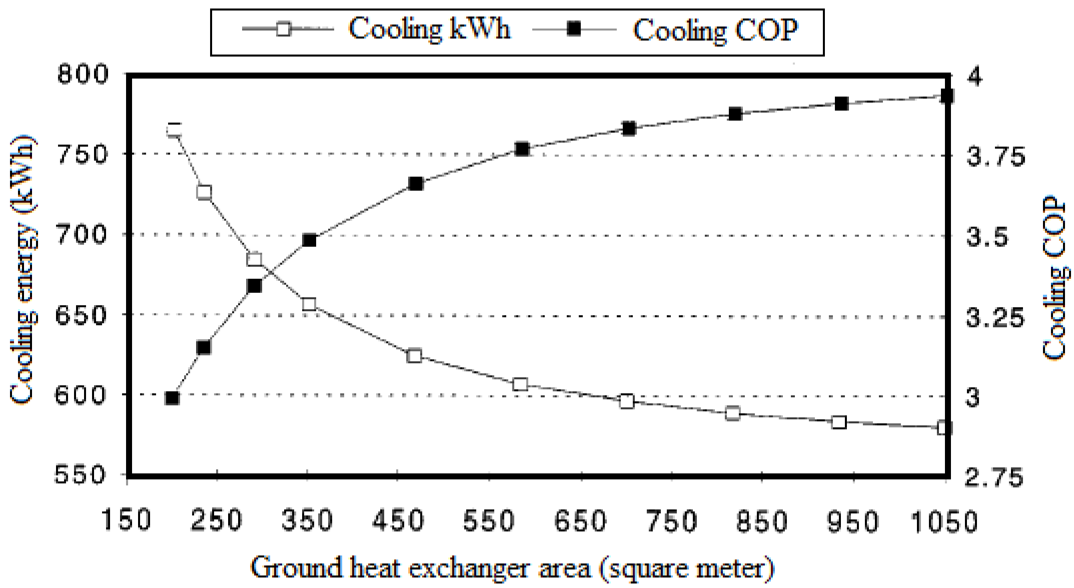

Figure 2.11. Effect of ground heat exchanger area on cooling performance [48]

2.8. Section conclusion

GSHPS utilizes the earth, ground water, or surface water as a heat source or sink for providing heating and cooling. The GSHPS is generally recognized to be one of the most outstanding technologies of heating and cooling in both residential and commercial buildings. Since, it provides high coefficient of performance “COP” up to 3-4 for an indirect heating system and 3.5-5 for a direct heating system. The main benefit of using the GSHPS is that the temperature of the subsurface is not subject to large variations experienced by air.

GSHPS running costs are lower than conventional heating and air conditioning systems. As a result, GSHPS have increasingly been used for building heating and cooling with annual rate of increase of 20% in recent years. With increasing worldwide awareness of the serious environmental problems due to fossil fuel consumption, efforts are being made to develop energy efficient and environmentally friendly systems by utilization of non-polluting renewable energy sources, such as solar energy, industrial waste heat or geothermal water. Heat pump applications using the renewable energy sources increase from day to day. GSHPS are suitable for heating and cooling of buildings, and so will play a significant role in reducing fossil fuel consumption and CO_2 emissions.

CHAPTER 3

OVERWIEV ON GROUND COUPLE HEAT PUMP SYSTEMS

3.1. Introduction

This thesis is mainly focusing on experimental investigation on thermal performance of Ground Coupled Heat Pump System (GCHPS) for heating and cooling of a space. It is necessary to give basic information about heat pump technology. Therefore, in this chapter, it will be given basic information about heat pump technology, and theory and components of the heat pump systems. Also, design factors for the GCHPS which a designer or installer must consider in selecting correct size of each component of the GCHPS are outlined alongside explanations. Finally, various methods available to improve design accuracy in respect to ground loop length are introduced.

3.2. Thermodynamic explanations for a heat pump system

In this section, basic thermodynamic explanations for a heat pump system such as Clausis statement, Carnot cycle with the vapour compression refrigeration cycle, vapor compression refrigeration cycle and heat sources of heat pump systems will be summarized. Processes and states for the heat pump cycle will be explained and discussed shortly.

3.2.1. Clausis statement

The Clausius statement of the second law of thermodynamics states that it is impossible to operate a cyclic device in which the only effect is the transfer of heat from a cooler body, Tc to a hotter body, Th. Operating principal of a heat pump is shown in figure 3.1. This still holds true but it has been found that the addition of an energy input can produce a net heat transfer from Tc-Th. The first mention of exploiting this by means of a heat multiplier is attributed to Lord Kelvin in 1852 as a part of his theory of the dissipation of energy. He outlined basic principal of a heat pump which transfers heat from the surroundings to a warm space in order to maintain that space at a higher temperature than its surroundings.

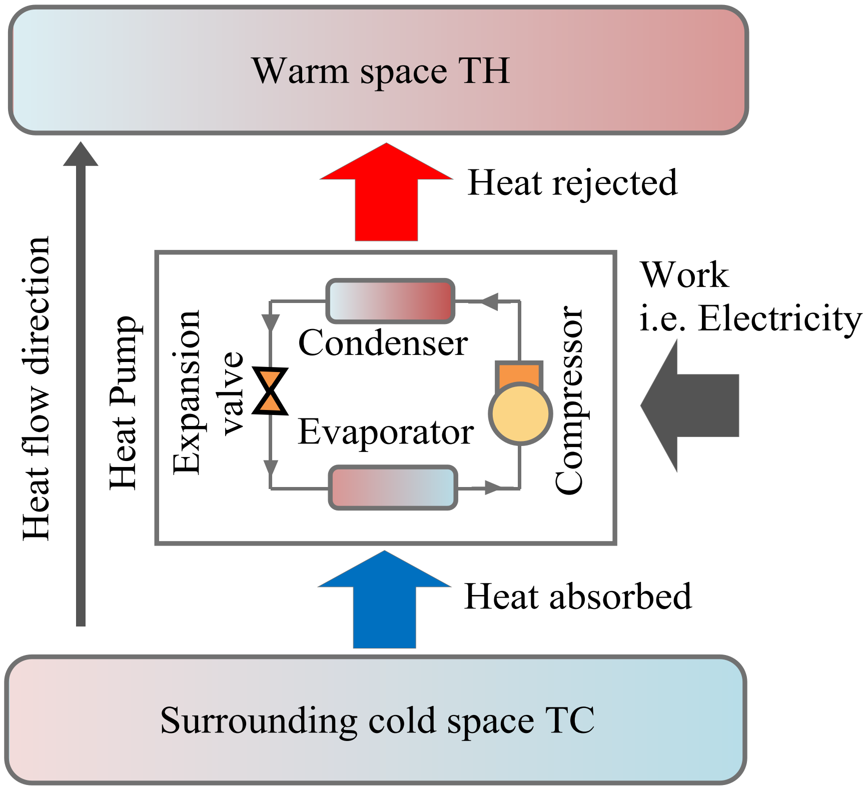

Figure 3.1. Basic premise of heat pump [53]

Although there are several different thermodynamic cycles which can feasibly perform heat pumping, the great majority of heat pump systems work on a vapour compression cycle [53], and this is considered as state of the art. Other methods include absorption, adsorption, vuilleumier, stirling cycles, single phase cycles and hybrid systems. Since heat transfer usually occurs from a hotter body to a cooler body until they reach equilibrium it can be seen that a heat pump is moving heat in a direction it would not normally travel. In this case, it is known that temperature is simply a term for heat energy. The heat energy or enthalpy of a body cannot be raised exempt for adding energy to it. A heat pump achieves this in the form of work for compression. This is in correlation with the first law of thermodynamics which states that it is always possible to convert any given quantity of mechanical energy into its equivalent heat energy.

3.2.2. The Carnot cycle

Figure 3.2, shows schematic representation of the reversed Carnot cycle (a) and its temperature entropy (T-s) diagram (b). Basic working principal of the reversed Carnot cycle can be explained by tracking the movement of a working fluid in figure 3.2. The working fluid used in the cycle is called as refrigerant which goes form 1-2 and 3-4 as isentropically, i.e. constant entropy processes. A device between processes 1 and 2 called as compressor, and the other one between processes 3 and 4 is called as expander. Processes 2-3 and 4-1 take place at constant temperature, which are known as isothermal processes. The process 4-1 where heat is extracted from a heat source such as air, river, lake, ground etc. The device used between the processes 4 and 1 is known as evaporator in which the refrigerant evaporates by taking energy from the energy source. Condenser is used through the process 2-3 where heat is rejected from it to heat sink. A space, house or building can be used as a heat sink for heating operation. In cooling operation or reversed cycle, the space, house or building are known as heat source. Furthermore air, river, lake, ground etc. are known as heat sink. All stages in these processes in the Carnot cycle are assumed to be reversible. In reality, it is not possible to achieve this ideal reversible cycle. Firstly, difficulties are presented by trying to compress a two phase liquid and therefore evaporation will continue to the saturated vapor line. Secondly during the compression stage heat will be lost due to frictional losses. Moreover the work output from the expander is relatively small compared to the work input during compression, Therefore it is usually replaced by a simple throttle valve [54].

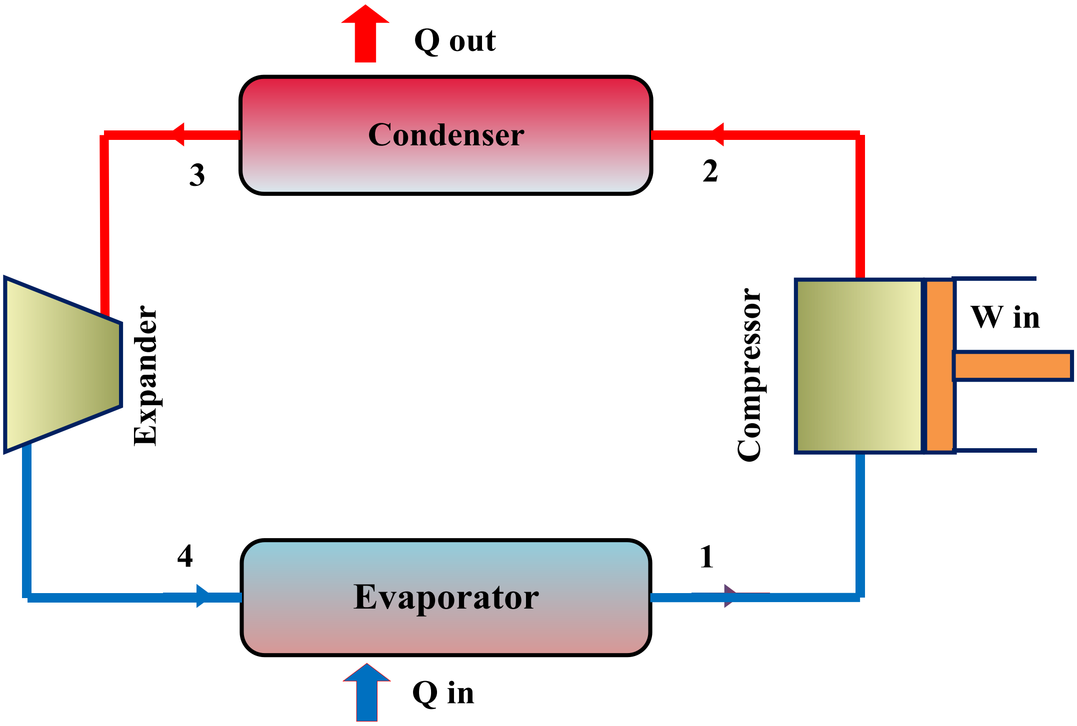

Figure 3.2a

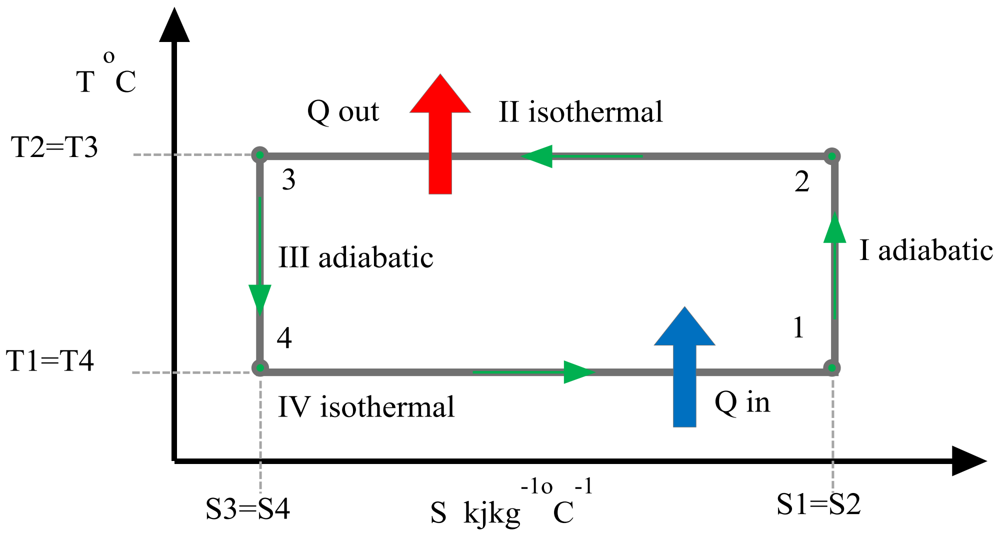

Figure 3.2. Schematic of Carnot cycle (a) and its temperature-entropy (T-s) diagram (b) [54].

3.2.3. The vapor compression refrigeration cycle

There are many difficulties associated with the reversed Carnot cycle can be eliminated by vaporizing the refrigerant completely before it is compressed. A throttling or expansion device is replacing the expander or turbine. The cycle that results is called ideal vapor compression refrigeration cycle. Pressure and enthalpy (P-h) diagram for an ideal vapor compression cycle is indicated in figure 3.3. There are two constant pressure processes and one constant enthalpy process. The vapor-compression refrigeration cycle is the most widely used cycle for refrigerators, air-conditioning systems, and heat pumps. In actual, it is not possible to operate the ideal vapor compression cycle ideally. A cycle is necessary to use in actual, which is called as actual vapor-compression refrigeration cycle. The actual vapor-compression refrigeration cycle differs from the ideal one in several ways, owing mostly to the irreversibilities that occur in various components. Two common sources of irreversibilities are fluid friction that causes pressure drops and heat transfer to or from the surroundings [56]. Schematic of an actual vapor compression heat pump cycle is shown in figure 3.3.

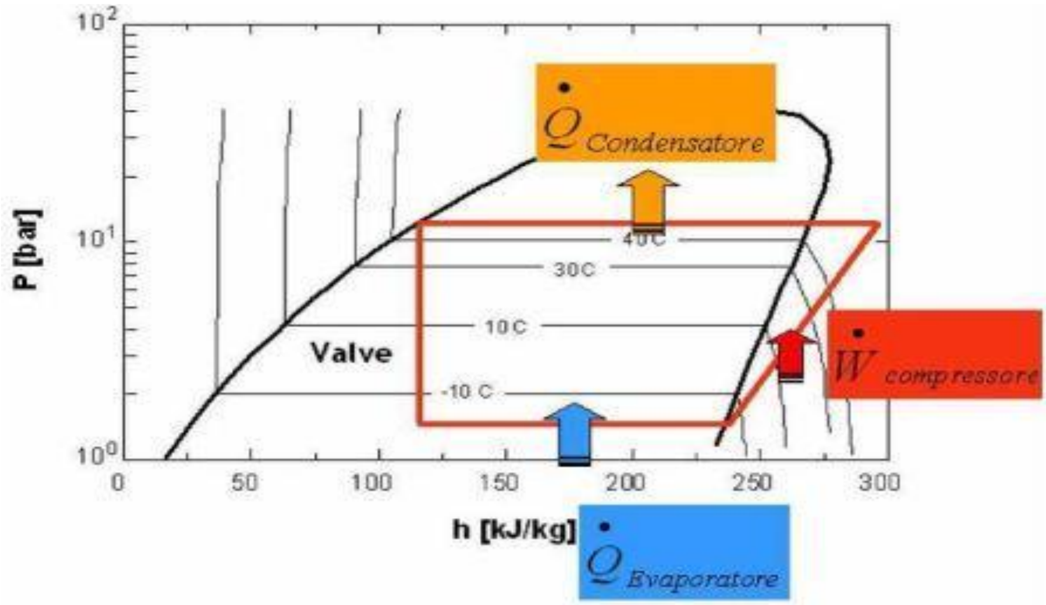

Figure 3.3. Vapour compression cycle on a pressure and enthalpy (P-h) diagram [53]

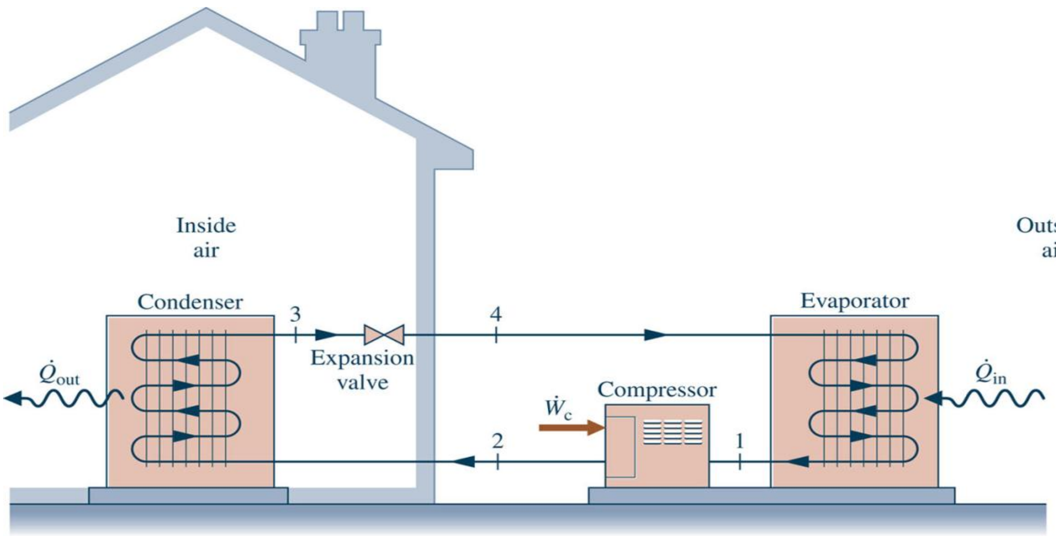

Figure 3.4. Schematic of an actual vapor compression heat pump cycle [54]

3.2.4. Heat sources of the heat pump systems

The technical and economic performance of heat pump is closely related to characteristics of the heat source [53]. An ideal heat source should be:

- High and stable temperature during the heating season.
- Abundant.
- Not corrosive or polluted.
- Have favourable thermophysical properties.
- Require low investment to exploit.
- High specific heat per unit volume “small temperature drop during-extraction”.

There is a variety of different sources from which a heat pump can draw heat during the evaporation process in the outer coil. The option selected will depend on local circumstances, the location of the building and its heat demand. The most popular heat sources are air, water (i.e. river, lake, pond, and ground) or the ground soil and rock. A lot of heat pump systems have been designed to increase performance of the systems using different the most popular heat sources such as exhaust air, sea water, waste water and effluent etc.

Source temperature is one of the most important characteristic of the source. Since, the level of temperature of the heat source will affect performance of the heat pumps. For example; for an air source heat pump system, heat is extracted from the ambient air, environment or surroundings for a heating purpose. When the ambient air temperature is higher, and then performance of the heat pump will be higher for heating operation. Air also has a lower thermal mass than the water or ground which hampers heat transfer. In many cases, air source heat pumps will have lower COPs than those using water or the ground as a heat source, due to more variable temperature ranges shown in figure 3.5 and table 3.1. The heat pumps using the ground as a heat source operate in less operation time at the optimal design point, lower capacities at low temperature and additional energy requirements to defrost the evaporator coil.

It is seen from the figure 3.5 that amplitude of the ambient air temperature is higher than the ground temperature. When the depth of the ground increases, and then amplitude of its temperature decreases, and it stays constant at a depth of 75 m. The air temperature is less than the ground temperature during winter or heating season, and is higher than it during the summer or cooling season. The stability of the source temperature increases performance of the heat pump systems as general. Also, amplitude or temperature ranges of the heat sources are presented in table. It is seen from the table that the highest temperature amplitude is for the ambient air.

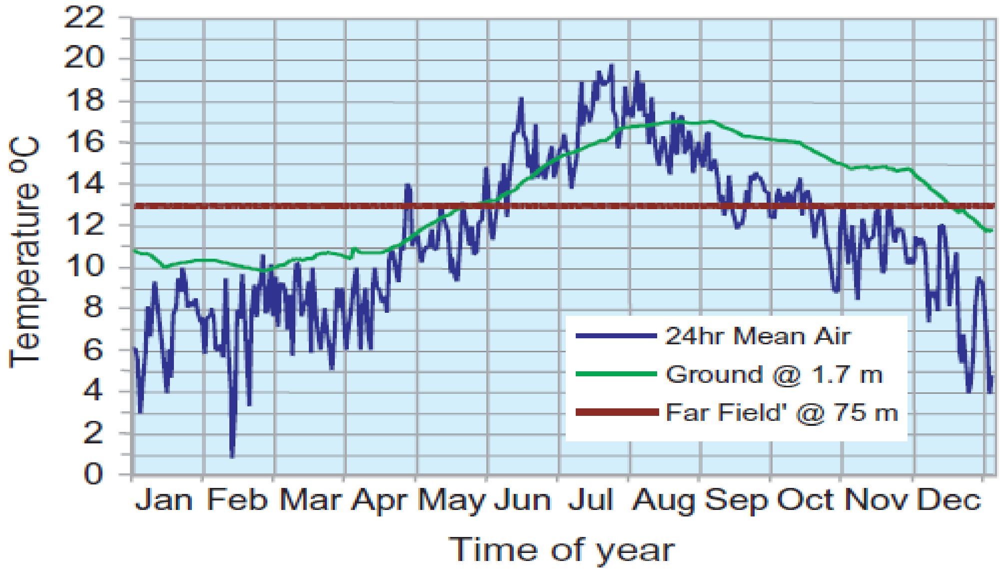

Figure 3.5. Annual variation ground and ambient air temperature [54]

Table 3.1. commonly used heat sources [53]

Sources	Temperature ranges, °C
Ambient air	-10-15
Exhaust air	15-25
Ground water	4-10
Lake water	0-10
River water	0-10
Sea water	3-8
Rock	0-5
Ground	0-10
Waste water and effluent	>10

3.3. Heat pump components

The main components of a heat pump system are the compressor, two heat exchangers referred to as evaporator and condenser, the expansion valve, reversing valve and accumulator. The components are connected to form a closed circuit. Working fluid that is known as Refrigerant circulates through these components. It is necessary to explain these components in order to understand working principals. For that reason, description and operating conditions will be explained in this section.

3.3.1. The compressor

The compressors are one of the most important parts or heart of any heat pump cycle [55]. The compressor compresses the refrigerant, which flows to the condenser, where it gets cooled. It then moves to the expansion valve, and the evaporator and it is finally sucked by the compressor again. For the proper functioning of the heat pump cycle, the refrigerant must be compressed to the pressure corresponding to the saturation temperature higher than the temperature of the naturally available air or water. It is the crucial function that is performed by the compressor. Compression of the refrigerant to the suitable pressure ensures its proper condensation and circulation throughout the cycle. There are five types of compressor used in the heat pump cycle, which are:

a) Reciprocating type compressors,

b) Scroll type compressors,

c) Rotary type compressors,

d) Helical or screw type compressors,

e) Centrifugal compressors.

Improvements in compressor technology have resulted in improved heat pump performance. Efficiencies for different types of compressor are shown in figure 3.6. Modern hermetically sealed scroll compressors have brought improved system performance. The compressor work principle builds upon using an explanation of how a scroll compressor operates. A more important parameter effecting performance of any compressor is compression ratio. The compression ratio can be defined as the ratio of absolute discharge pressure to absolute suction pressure. This ratio will vary for cooling and heating applications due to differing temperatures and suction pressures, which will affect the system performance.

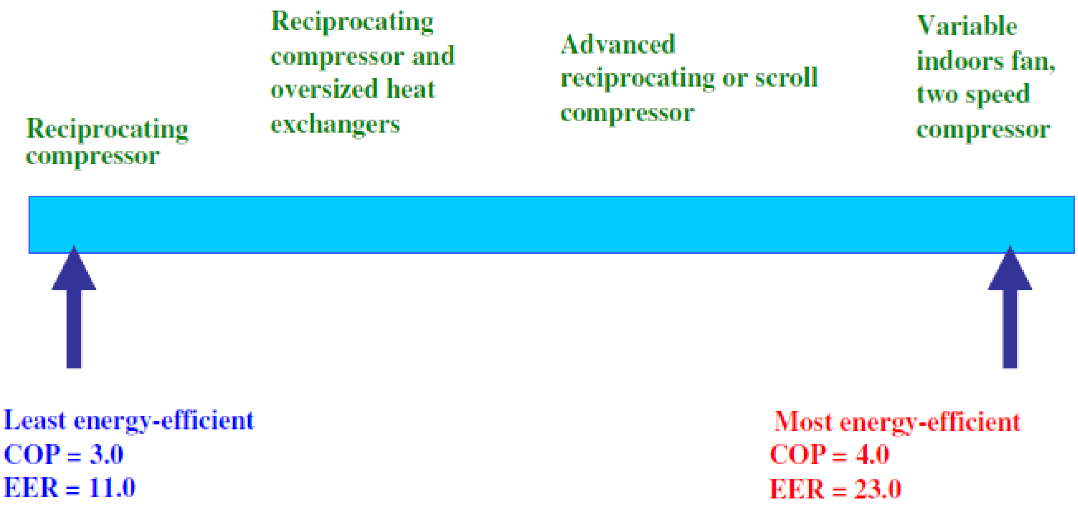

Figure 3.6. Efficiencies for different types of compressor [54].

3.3.2. Condenser

Condenser used in a heat pump cycle is a type of heat exchanger in which energy is rejected from the refrigerant flowing through the condenser to the coolant, which can be air or water. After passing through the condenser the refrigerant gets condensed but still remains at high pressure. It comes out in a partially liquid and gaseous state and then enters the throttling or expansion valve. There are three types of condensers: air cooled, water cooled and evaporative as described below:

i) Air cooled condensers: Which are used in small units like household refrigerators, water coolers, deep freezers, small packaged air-conditioners etc. These are used in fields where the cooling load is small and the total quantity of circulating refrigerant in the cycle is small. They are also called coil condensers as they are usually made of aluminum or copper coil. These types overrun a somewhat larger space than the other types. These are assorted in two kinds in which heat transfer take place forced convection and natural convection. For natural convection type, air flows over the surface in natural way whereby on the condenser coil temperature. In forced air type, a fan run via a motor blowing air through the surface coil.

ii) Water cooled condensers: Which are used for large refrigerating fields, large packaged air-conditioners, central air-conditioning stations, etc. They are used in fields where cooling loads are overly high and a multitude of refrigerant flows via the condenser. Water cooled condensers divided into three categories: Which are double pipe type or tube-in-tube, shell and coil type, and shell and tube type. In all of them the refrigerant flows via one side of the piping however the water flows via the other piping, in order to cool and condense the refrigerant.

iii) Evaporative condensers: Which are ordinarily used in the ice stations. They are a combination of air cooled condensers and water cooled. In these systems the hot refrigerant flows via the coils while water is sprayed along the coils. Altogether, the fan directed air from the lower side of the condenser and discharges it from its upper side. The spray water comes in touch with the condenser coil obtain evaporated in the air and it sucks the heat from the condenser in order to cool and condense the refrigerant. These systems have the benefits of both air cooled and water cooled condensers, thus it needs less space. Yet, safeguard the evaporative condenser clean and free of range is so difficult and needs a lot of maintenance.

3.3.3. Evaporator

Evaporator shown in figure in 3.7 is a type of heat exchanger that is used in a heat pump cycle. The process of heat removal from the substance to be cooled or refrigerated is done in the evaporator. The liquid refrigerant is vaporized inside the evaporator (coil or shell) in order to remove heat from a fluid such as air, water etc. Evaporators are manufactured in different shapes, types and designs to suit a diverse nature of cooling requirements. Furthermore, we have a variety of types of evaporators, such as prime surface types, finned tube or extended surface type, shell and tube liquid chillers, etc.

3.3.4. Expansion valve

An expansion valve is a piece of equipment that reduces pressure in a system. The most common form of expansion valves is a thermal expansion valve which is used in heating, ventilation, and air conditioning systems. The two main types of air conditioner expansion valves are thermostatic expansion valves and capillary tubes. A thermostatic expansion valve controls the flow of refrigerant and can function at various temperatures or pressures. The capillary tube is responsible for transmitting the pressure levels in the system to the sensing tube attached to the thermostatic expansion valve and basically acts as a throttle.

Figure 3.7. Plate type evaporator [55].

3.3.5. Reversing valve

A reversing valve is an important element within a heat pump system. Schematic representation of the reversing valve is shown in figure 3.8. It changes the direction of the refrigerant flow within the heat pump system. It is located along the tubes or pipes that connect the major operating components of the heat pump, and helps direct the flow of refrigerant through these pipes. This valve allows the heat pump to act as both a cooling system and a heating system. For heating the inner coil acts as a condenser and the outer coil an evaporator, in cooling mode however the refrigerant flow is reversed and the roles of the inner and outer coils switch. The heat pump reversing valves eliminate the need for separate heating and cooling systems, which typically results in lower costs, less maintenance, and improved efficiency.

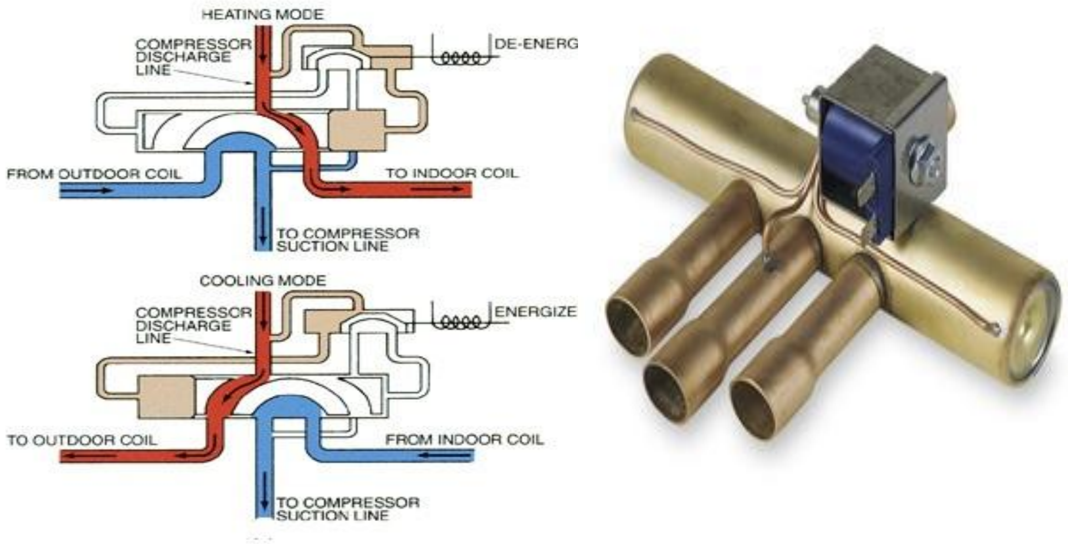

Figure 3.8. Schematic of reversing valve in cooling and heating mood [54]

3.3.6. Accumulator

It plays a key role in maximizing the performance of the heat pump. Compressors are designed to compress vapour and not liquids excessive liquid returned to the compressor could cause the possible dissolution of the compressor lubrication oil. The accumulator acts to trap the cool low pressure refrigerant and allow liquid to evaporate prior to entering the compressor [54]. The accumulator is usually located after the reversing valve and before the compressor. The accumulator can also be coupled to a heat exchanger to increase efficiency. An accumulator-heat exchanger has three key functions [55]:

- Add sub cooling to high pressure liquid on its way to the evaporating coil.
- Provide a positive separation of low pressure liquid and vapour from the evaporating coil so only dry, vapour reaches the compressor suction.
- Assures positive oil return to the compressor at all times during operation.

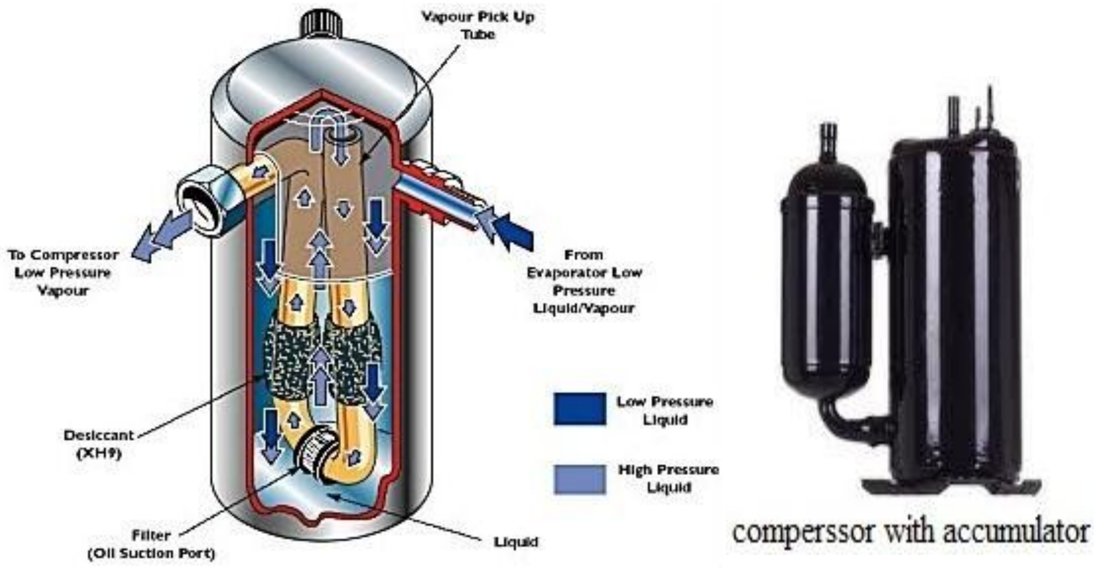

Figure 3.9. Schematic diagrams of accumulator located on the compressor [55]

3.4. Miscellaneous elements of the heat pumps

There are several miscellaneous elements of the heat pumps, which are connecting pipes, control unit, refrigerants, dryer etc. These parts are necessary to be able to operate any heat pump unit. For example; if there is no refrigerant in the heat pump cycle then, it will not be possible to heat or cool the domain by the heat pump. Furthermore, some of these components will shortly describe in this section.

3.4.1. Connection Pipes

Usually made from aluminum or copper, their aim functions are accountable of circulating the refrigerant throughout the system. The length, diameter, thickness, properties and nature of the metal, welding type, and the fixing and supporting method should be chosen and done in the correct way to insure better operation condition and to avoid unnecessary problems of leakage and heat transfer losses due to friction and bad pipes insulation. And taking into account the expansion and contraction with compression ratio which will affect the performance of the system.

3.4.2. Control unit

The control circuit aims to operate the heat distribution system at the lowest temperature which meets require comfort conditions [56]. This will maximize efficiency, and ensures better stable operation condition. The control unit is linked and synchronized it function with the protection system device. Furthermore avoiding any defect extraordinary occurs on the system to safeguard the running system devices. Modern heat pump units have a similar standard of controllability i.e. weather compensated control that ensures the heat pump never works harder than necessary through utilizing a sensor for gauging the outside air temperature, a selection of heating curves, timers and specific operating and fault messages.

3.4.3. Refrigerants

Ideal refrigerant has favorable thermodynamic properties, is noncorrosive to mechanical components, and is safe (including nontoxic, nonflammable, and environmentally benign). They are more suitable for a particular application by choice of operating pressure.

CFC "Chlorofluorocarbons" were developed in the early 1930's. Their chemical stability and low toxicity meant they were suitable for residential use and furthermore they were relatively inexpensive. The most common types of these refrigerants are [56]:

- CFC-12 – low/medium temperatures (max 80°C)
- CFC-114 – high temperatures (max 120°C)
- R-500 – Medium temperatures (max 80°C)
- R-502 – Low-Medium temperatures (max 55°C)

In 1974 however links were drawn between CFC's and ozone depletion [56] and it has since been proven that the chlorine content of CFC's and their chemical stability means these substances have a high global warming and ozone depletion potential (GWP and ODP, respectively). Since chlorine has a long life it is transported by winds into the stratosphere and there are no natural processes to remove it. For this reason the CFC's such as R-11/12/13, R-113/14/15 and R-500/502 are now prohibited as refrigerants.

After CFC's were prohibited HCFC's (Hydrochlorofluorocarbons) were introduced. These also contain chlorine but have a far lower ODP and GWP, typically 2-3% and 12% of CFC-12, respectively, this is due to lower atmospheric stability and the fact they are not so readily transported into the upper atmosphere [55]. These (R-22, R-401/402/203) are considered transitional refrigerants however, and can only be utilized in retrofit systems. Under the 1987 Montreal Protocol and 1995 Vienna Convention all CFC's and HCFC's are to be phased out by 2020 [56].

Long term options may be HFC's (Hydrofluorocarbons) which are chlorine free. Examples of these are R-134a, R-152a, R-32, R410a, and R-507. Since these don't contribute to ozone depletion they are seen as a crucial alternative to R-12, R-22 and R-502.

Table 3.2. Potential HFC's and their Properties [56]

HFC Refrigerant	Properties
HFC-134a	Similar thermo physical properties and achievable COP to CFC-12
HFC-152a	Component in blends. Flammable however so only used in small systems.
HFC-32	GWP close to zero. Can replace R-502 and HCFC-22.
HFC-125/143a	Similar properties to R-502 and HCFC-22. Three times GWP of HFC-134a.

3.5. Ground couple heat pump systems (GCHPS)

There is potential for GCHPS to be used for space heating and cooling. The GCHPS are used in domestic, industrial and commercial applications, which are summarized in this section.

The GCHPS are used in domestic applications which are:

- House or building heating.

- House or building cooling.
- Water heating.

The GCHPS are used in industry in a number of different applications [58]:

- Space heating.
- Heat/Cooling of process streams.
- Steam production and Water heating for washing/sanitation/cleaning.
- Drying, dehumidification, and evaporation.
- Distillation

The GCHPS are used for commercial applications. Sports centers are also a promising option for the technology especially if they have swimming pools which require a constant heating/dehumidification load or if cooling is required. Limitations on an extended use of GCHPS are [59]:

- Most commercial premises are leasehold and owners would have to be willing to allow drilling/disruption close to the building.
- Land in shopping and office areas is of high value and there is unlikely to be large areas available for collection systems.
- Buildings tend to be bigger and as such will require large collection systems.
- Payback is an investment norm and higher capital costs may rule out GCHPS.

Literature survey and description about GCHPS technology built on the main heat pump information is mentioned in chapter 2. This study is about the experimental study that is to find thermal performance of the GCHPS. The heat pump used in the GCHPS extracts energy from the ground. Therefore, it is required that thermal characteristic of the ground should be explained. Firstly, the varying of the earth thermal and physical properties, different methods of extracting the heat in terms of direct, indirect and open systems and the types of loop configuration are outlined. Finally potential applications for GCHPS are discussed in this section.

3.5.1. Material of the ground heat exchanger (GHE)

System performance is significantly affected by the material in which the Ground Heat Exchanger (GHE) is laid. Factors which will determine performance are [56]:

- Subsurface temperature and thickness, nature of superficial deposits i.e. soil.
- Rock properties, depth to groundwater, seasonal variations in ground water.

3.5.2. Ground temperature

The solar radiation received by the ground is absorbed at its surface. The absorbed heat is transferred to a finite depth of the ground. As a result of the heat transfer, ground temperature will increase. The heat transfer from its surface to finite depth depends on thermal and physical properties of the ground. There are several studies about this subject in literature. Heat transfer and temperature variation with respect to depth and seasons were investigated. Seasonal and mean variation of the ambient air, earth and groundwater temperature with respect to months and depth of the earth are given in Figure 3.10, respectively. It is seen from the first figure that the amplitude takes place for the ambient air, but the lowest seasonal amplitude of the temperature happens for the groundwater. It is understood from the figures that the lowest COP for a heat pump will be obtained for the ambient source heat pump system. But the highest COP will be obtained for groundwater and then earth. The second figure shows seasonal variations down to a depth of approximately 15 meter. More variations occur in the first two meters. Below this level the temperature is fairly constant and will roughly equate to the mean annual air temperature of the region, due to the fact that earth has a high thermal mass/inertia of soil and is able to store the heat absorbed. After heat is extracted it is regenerated by solar irradiation, precipitation and on a smaller scale thermal gradient in the ground [57].

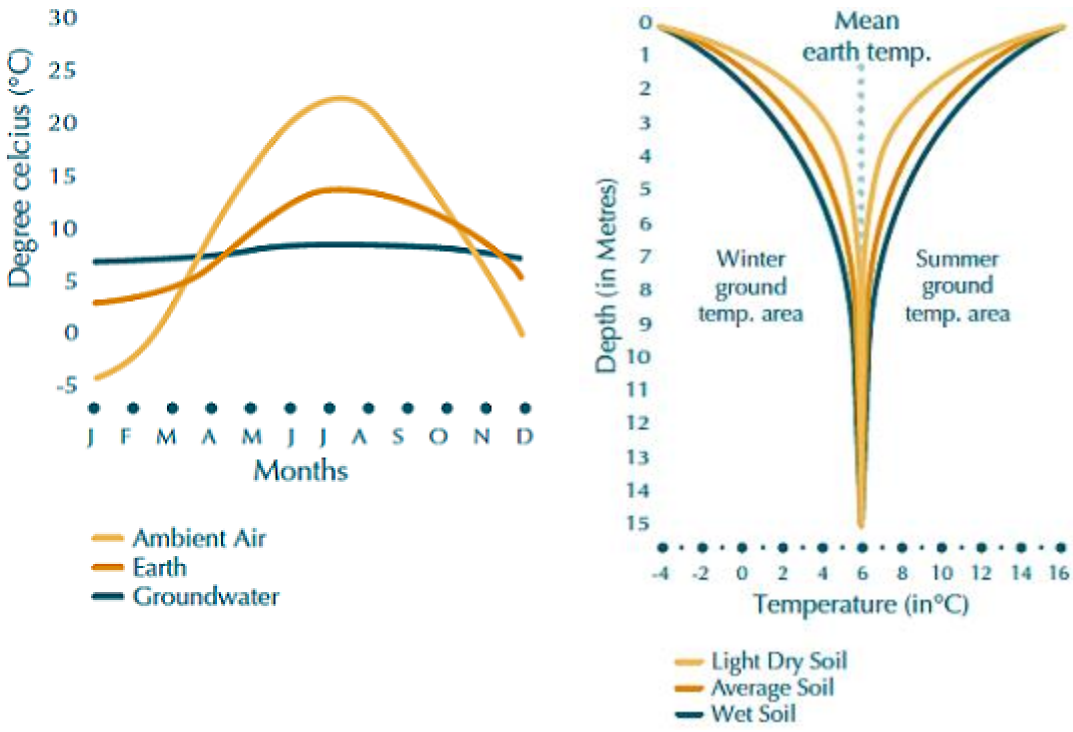

Figure 3.10. Variation of temperature for ambient air, earth and groundwater [57]

3.5.3. Thermal and physical properties

GCHPS capacity for heating and cooling will depend on, not just the size of the system but also the thermal and physical properties of the ground. For that reason, thermal and physical properties of the ground are very important for COPs of the GCHPS. Since temperature difference between the ground and the fluid in the ground heat exchanger drives the heat transfer. In order to design a GCHPS, the knowledge of ground thermal properties are important for correct functioning of the system. The thermal properties are known as thermal conductivity, pipe or borehole thermal resistance, undisturbed soil temperature, specific heat capacity, etc. These factors will also have a strong influence on the capital costs associated with installing the ground heat exchanger, which can account for 30-50% of total capital costs [56].

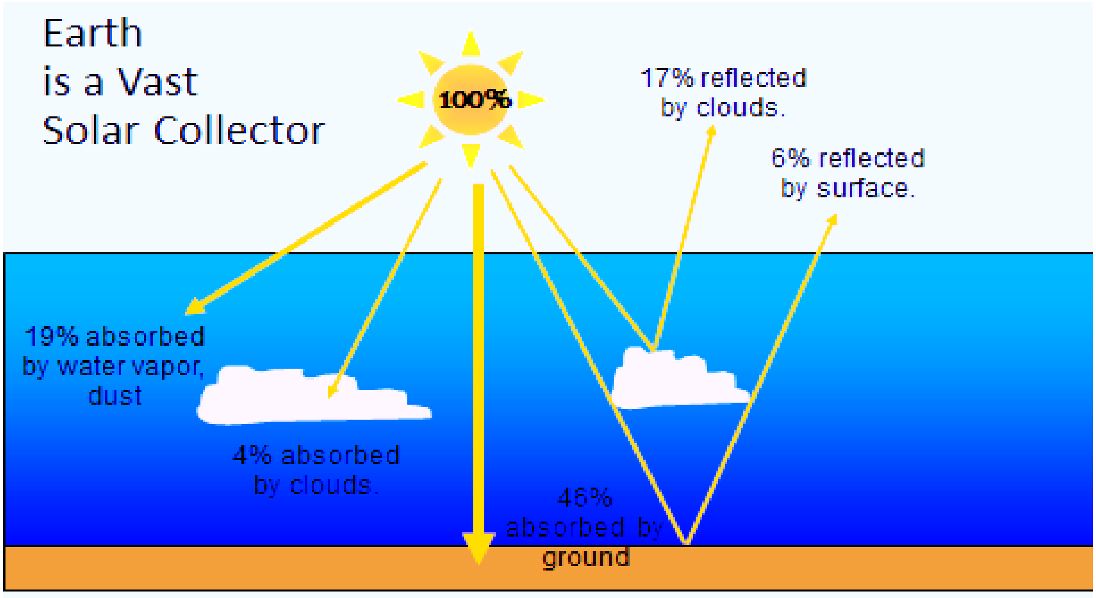

Figure 3.11. Solar distributions on the earth [57]

3.5.4. Thermal conductivity

It is a measure of the quantity of heat transmitted per unit area, per unit temperature gradient and in unit time, under steady state conditions. Multiplying this factor by the thermal gradient will give the ground heat flow. Considering thermal conductivity in rocks factors such as porosity, composition and the nature of any saturating liquids will determine its value. Generally, the larger the extent of porosity the lower the thermal conductivity. Thermal conductivity can vary for rocks most commonly found near the surface and even more great for the range of sediments found in area. Mainly, rocks have higher thermal conductivity values than soils.

Table 3.3. Thermal conductivity of typical rocks and sediments [55]

Material	Typical Thermal Conductivity, $Wm^{-1}K^{-1}$
Low porosity sedimentary rocks (<30%) i.e. shale, sandstone, siltstone	2.2-2.6
Quartz sandstone (5% & 30% porosity)	6.5, 2.25
Schist, Serpentine	2.9
Igneous plutonic rocks i.e. granite, gabbro	3.0
Quartzite	5.5
Sand (gravel), saturated sand	0.77, 2.5
Silt	1.67
Loam	0.91
Clay, saturated	1.11, 1.67
For Comparison:	Water = 0.6, Air = 0.0252

3.5.5. Thermal diffusivity

It is a measure of ground thermal conduction in relation to thermal capacity. This links thermal conductivity (k), specific heat (Cp) and density (ρ). Density multiplied by specific heat is termed volumetric heat capacity. As shown in the following formula $\alpha=k/(\rho Cp)$, m^2/s, a high thermal diffusivity value is desirable since this means the material will quickly adjust temperature to that of the surrounding environment since heat is conducted rapidly relative to thermal mass.

Table 3.4. Thermal diffusivity of typical rocks and sediments [55]

Material	Typical Thermal Diffusivity (m^2 day^{-1})
Basalt	0.059
Granite	0.086
Gneiss	0.106
Quartzite	0.255
Clay	0.082
Limestone	0.091
Sandstone	0.143

3.6. Types of ground source heat pump systems (GSHPS)

There are several types of ground source heat pump systems (GSHPS). These are direct and indirect systems classified as closed and open systems, horizontal and vertical closed loops cycles slinky or spiral loops cycle. Description of these types of systems will be summarized in this section.

3.6.1. Direct and indirect systems

It is important to clarify the difference between direct and indirect systems. Indirect system the circulating fluid, to which heat is transferred to or from the ground, will be a water or antifreeze mixture. For indirect systems the ground loop is made from plastic, usually high density polyethylene or polybutylene, which has a long life. Copper is utilized as the piping material for direct systems. A direct system is more viable the smaller the heat pump capacity. An alternative configuration is to circulate the refrigerant through the ground loop to pick up the low grade heat called a direct expansion DX system. This offers several advantages over an indirect system [56]:

- Increased thermal contact with the ground, hence raise efficiency.
- No requirement for a circulation pump.

- Elimination of a heat exchanger between ground coil liquid and refrigerant.
- Shorter ground coil required.

3.6.2. Open loop cycle systems

Open loop systems using wells and a pond are seen from figure 3.12. In open loop system, groundwater or lake water is used as a heat carrier and brought directly to the heat pump evaporator for the heating operation. It is consist primarily of extraction or re-injection wells and surface water systems. The water is drawn from a source i.e. the primary aquifer, over heat exchanger and then discharged or re-injected into a separate aquifer well. Since the heat exchanger exposed to the groundwater, it is subject to fouling, corrosion and blockage. Nevertheless, there are many advantages as drilling requirements are lower than closed loop system and performance can be improved since the groundwater will deliver heat at the ground temperature removing any losses through heat exchange to a circulating fluid. Open systems are subject to the highest pumping power requirements of GSHPS which can make costs excessive.

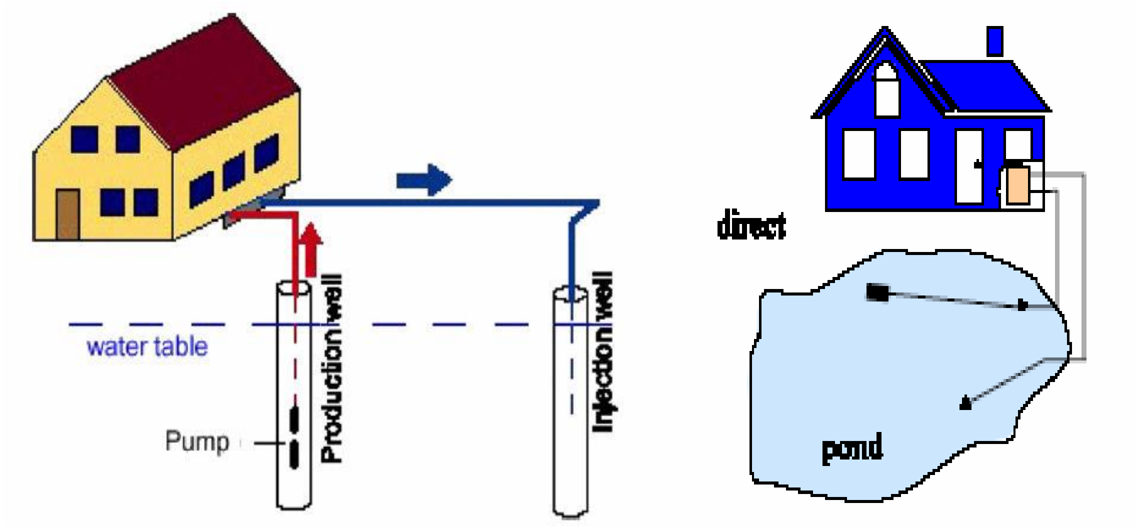

Figure 3.12. Open loop systems using wells and a pond [55]

3.6.3. Horizontal and vertical closed loops cycles

The choice of horizontal or vertical system depends on available land area, local ground conditions and excavation temperatures [56]. If land area is at a premium or the soil is too shallow for trenching, vertical borehole is required for the ground loop in order to maximize heat gain for the space available. Fitting a vertical collector will be more expensive due to high drilling costs, but it will increase thermal efficiency, require less pipe material and pumping energy. In a horizontal system, the pipes are buried beneath the ground at a depth of 1.2 and 2m. Also, all pipe runs at the same length; to ensure that the collector field has the same pressure drops. Approximately 35-60 m of length is required for each kW of heating i.e. a heat extraction of 15-30 W/m [55]. Horizontal systems can be laid in either series or parallel, these dense patterns offer maximum heat extraction for the space available. A distance of approximately 3m should be kept between pipe runs, to avoid thermal interference. Configurations of different types of GHE are given in figures 3.13-3.16. The horizontal series and parallel systems are shown in Figure 3.13, schematically. Figure 3.14 and 3.15 indicates horizontal trench collector systems and various configurations of horizontal ground heat exchangers, respectively. Figure 3.16 shows vertical borehole heat exchangers double U-pipe [55].

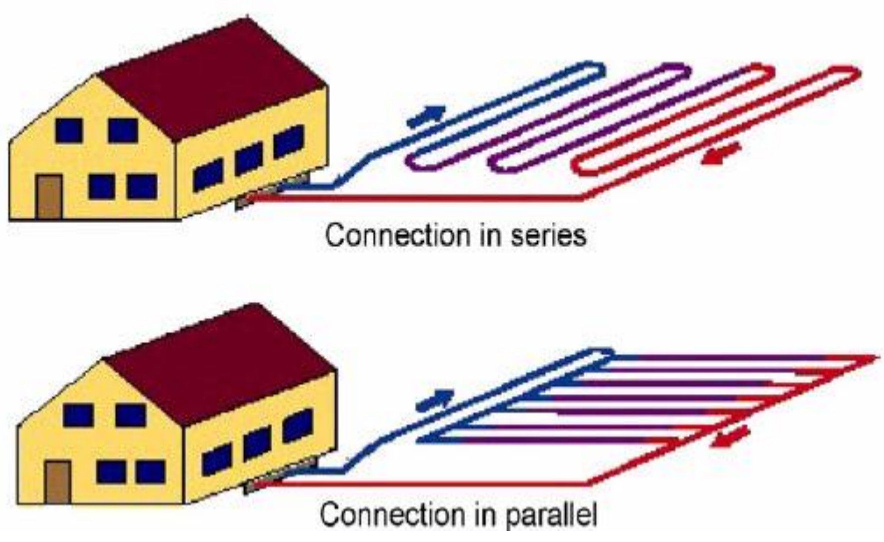

Figure 3.13. Horizontal series and parallel ground loop configurations [54]

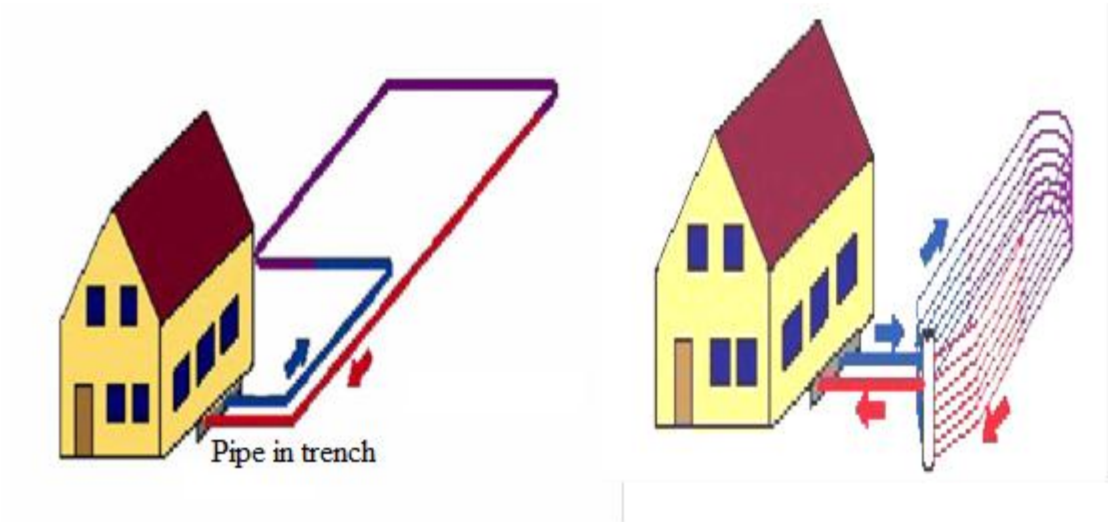

Figure 3.14. Horizontal trench collector systems [54].

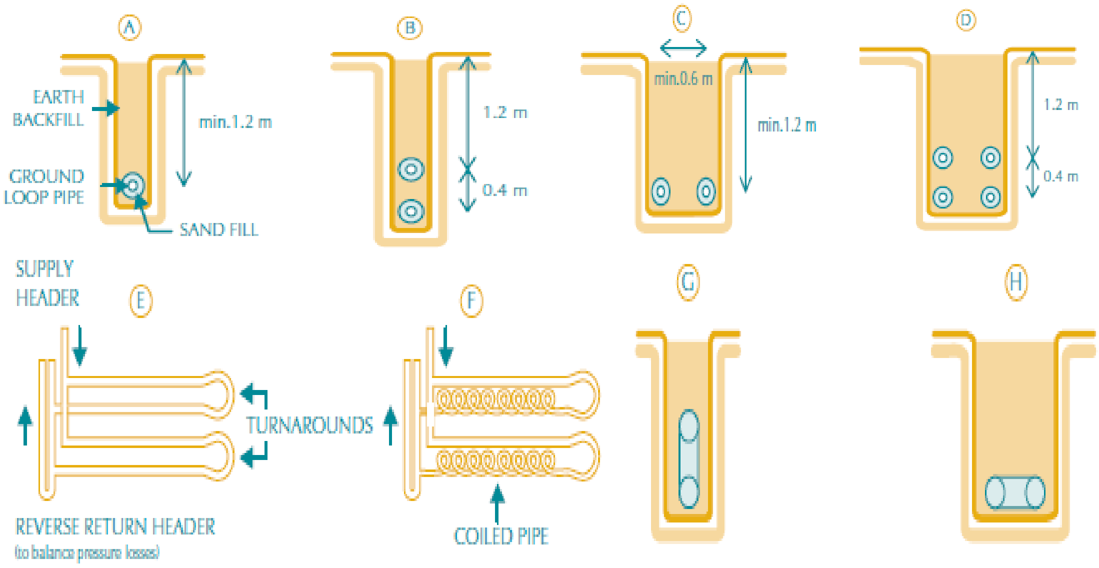

Figure 3.15. Various configurations of horizontal ground heat exchangers [54].

Where: A. Single pipe. B. Stacked two-pipe C. Parallel two-pipe. D. Stacked parallel four-pipe. E. Layout of parallel two-pipe showing turnarounds and header. F. Coiled pipe lay horizontally in a wide trench or vertically in a narrow trench. G. Coiled pipe lay vertically in a narrow trench. H. Coiled pipe lay horizontally in a wide trench.

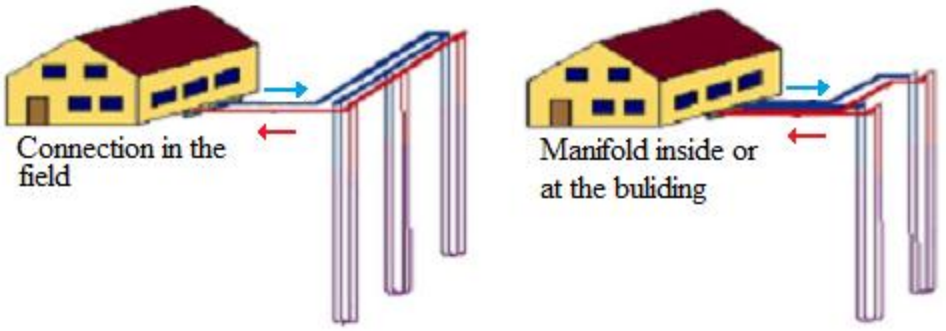

Figure 3.16. Vertical borehole heat exchangers double U-pipe [55].

3.6.4. Slinky or spiral loop cycle

A slinky closed loop field is a type of horizontal or vertical closed loop where the pipes overlay each other. A photo for a slinky horizontal and closed loop type is shown in figure 3.17. Slinky loop field is used if there is not adequate room for a true horizontal or vertical system, but it still allows for an easy installation rather than using straight pipe, slinky coils use overlapped loops of piping laid out horizontally or vertically along the bottom of a wide trench. Depending on soil, climate and the heat pump's run fraction, slinky coil trenches can be up to two thirds shorter than traditional loop trenches. Slinky coil ground loops are essentially a more economical and space efficient version. Moreover, it is best suited to areas where natural recharge to the ground is not essential [56].

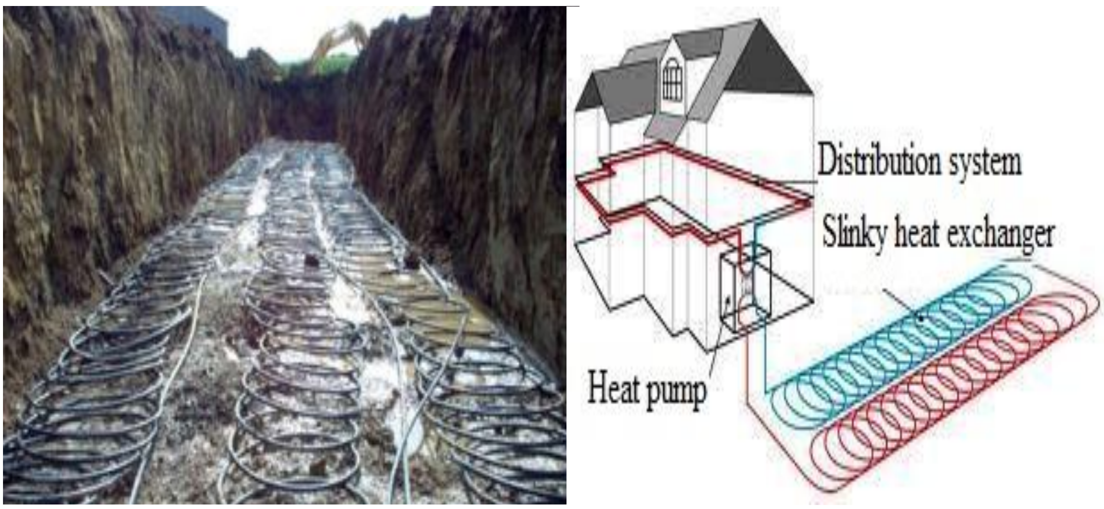

Figure 3.17. Slinky pipes in a trench [56]

3.7. Ground heat exchanger (GHE) design consideration

Here, it will firstly outline why correctly sizing the length of ground collector is fundamental in ensuring optimal ground source heat pump performance. The implications of over and under sizing will be covered. Many different factors which a designer or installer must consider in selecting the correct size are outlined alongside explanations of why they must be considered. Sizing the GHE is one of the most important tasks in the design of GCHPS [58], and is critical to achieve good performance. It is therefore essential that calculations are done accurately to ensure that it is not under or oversized. If the loop is undersized it will result in:

- Poor efficiency.
- Decreased comfort levels.
- Nuisance heat pump lockouts and safety control activation.

GHE designs need to balance long term issues such as heat buildup or depletion while also catering for short term peak loads (during which temperatures can increase between 5-10°C in 1-2 hours). With an undersized GHE there is a risk of it not being able to meet the building heat load. This will result in the utilization of auxiliary heating, i.e. direct electric systems, thus reducing system efficiency, running cost savings and increasing carbon dioxide emissions.

3.8. GCHPS design factors

In designing GCHPS, there are many factors which need to be taken into account in order to estimate the length and diameter of the ground loop required. Firstly, it is essential that the building loads and the building heat loss are known, it is related to energy consumption profile and the domestic hot water requirements if needed [59]. This should include the design loads i.e. peak demand, which for heating will be the coldest winter period. Heat gain should also be considered and the system balance point determined. GCHPS design should be based on loads for a whole year and not just peak heat/cooling demands. When using a borehole heat exchanger for a GCHPS, the length required satisfying building load is largely dependent on the thermal properties of the ground, soil or rock on site i.e. thermal conductivity, diffusivity, volumetric heat capacity etc. The moisture content of the ground will affect design considerations through altering thermal conductivity.

3.8.1. Climatic conditions

Significantly affect the performance of heat pump systems [55] through determining the temperature of heat source and the extent of thermal recovery after heat extraction have taken place. GCHPS underground loops extraction rates is dependent on the temperature difference between the circulating fluid and far field ground temperature. Thus, the following items should be specified:

- Pipe material, diameter, wall thickness.
- Loop configuration i.e. horizontal, slinky, vertical, or any combined types.
- Space between boreholes and ground loop pipes.
- If a borehole is used with a U-tube pipe, separation distance between them should be known.
- The thermal properties of the backfill i.e. grout. Backfill will affect heat transfer region.
- Heat transfer fluid (antifreeze or brine) utilized.

3.8.2. Operating characteristics

Operating characteristics of the heat pump itself need to be known. What are the minimum and maximum entering temperatures, what flow rate of refrigerant inside the unit is required; the latter will in turn affect pumping requirements and auxiliary energy consumption. Excessive pumping energy will limit expected running cost savings and lower the seasonal efficiency of the system. The COP_{HP} and COP_{SYS} are function of the antifreeze temperature from the ground coil (a factor of ground temperature, pump speed and the design of the coil) and the distribution temperature with the amount of energy that the ground heat exchanger delivers.

3.9. Design tools

Design tools will assist in the design and installation of more cost effective, reliable and efficient systems in the future [59]. The core of any sizing package is to model the heat transfer between the heat transfer fluid of brine or antifreeze, and surrounding ground (soil or rock .etc.) and hence to select a heat exchanger length that limits the water temperature exiting the loop. In order to perform these calculations quickly, design programs use various simplifying assumptions. There are a multitude of different software packages or programs available. These vary in terms of calculation method and assumptions utilized and the inputs required performing sizing.

An overview of different programs which are available to aid the user in sizing GCHPS will outline. Firstly, the general calculation methods behind programs will be introduced. Design software is a suitable compromise between rules of thumb and tables on one hand and time consuming numerical simulation on the other [60]. Early programs utilized for this task, which calculate the fluid temperatures for a single or multiple borehole system at an arbitrary time. The heat extraction rate is given by twelve steps each with arbitrary lengths. The process repeated cyclically after a period of time, normally one year, and the programs calculate the inlet and outlet temperatures when the mean fluid temperature is given. These programs require at least a base level of engineering knowledge to use. The methods used for designing of GHE are summarized in the following subsections.

3.9.1. Cylinder and line source method

In 1947, Carslaw and Jaeger developed an equation for heat transfer from a cylinder based in the earth. This approximates the time varying nature of heat addition and extraction from the ground to the heat exchanger during cooling and heating operation. This leads to a steady state solution and effective thermal resistance [60]. This method allows for calculation of thermal interaction between boreholes and estimation of long term heat buildup and depletion. An alternative approach is the one dimensional line source heat transfer equation. The borehole can be modeled in this one dimensional manner since the length of the pipe is far greater than its diameter. The amount of heat extracted or rejected by the heat exchanger is treated as a constant for each time step. This is possible since the temperature variation inside the borehole is usually slow and minor [58]. The line source analysis is conducted for a single pipe then the results are reproduced for a multi-borehole system shown in below figure 3.18.

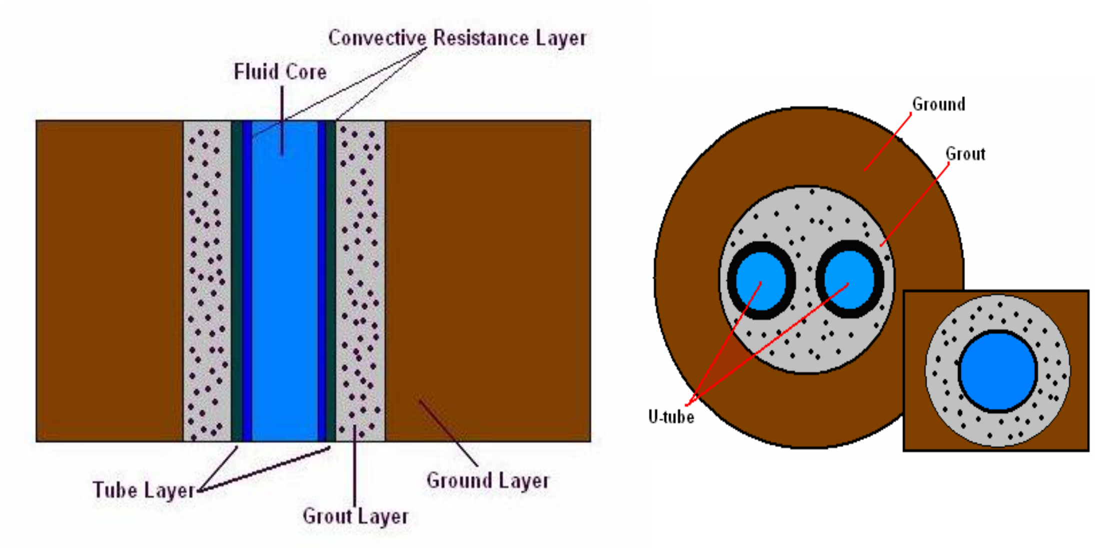

Figure 3.18. Heat transfer model implications and bird’s eye view [58].

3.9.2. G-Functions method

An algorithms have been derived for the estimation of required ground heat exchanger length to satisfy a given heat load from modeling exercises and parameter studies. It has produced an analytical solution to assess heat flow with different functions for particular borehole patterns and geometry. These are called G-functions and are dependent on the defined spacing of boreholes, depth of the borehole and space between the top of the heat exchanger and ground surface. Probably, the most computational effective method for performing ground response yearly simulations even at very short time steps of superposing in time the effects due to different heat loads by the way of pre-calculated temperature. A g-function represents the dimensionless temperature variation of a ground volume (with respect to the ground undisturbed and initial temperature and evaluated at the BHE periphery) to a stepwise, continuous and constant heat pulse lasting in time.

3.9.3. Eskilson method

Eskilson’s approach based on long time step response function was developed to determine the temperature distribution around a borehole loop model. As G-functions are non-dimensional temperature response factors, this allows the temperature fluctuation at the borehole wall to be calculated in response to changes in heat input over a specified time period. Eskilson’s research has been built on by various parties to develop shorter time step G-functions which can be used by sizing software packages to predict the nature of the borehole or ground relationship over shorter time periods. In multiple borehole G-functions are produced for fixed spacing. The thermal interaction between boreholes is stronger as the number of boreholes in the field is increased and as the time of operation increases [58].

3.9.4. Calculation method for ground loop heat exchanger (GLHE)

The G-functions utilized in GLHE are pre-computed using a finite difference model. The response to a peak pulse is estimated from the line source model with a simple analytical approximation [59]. GLHE has 307 pre computed borehole configurations, in which the depth of the borehole can be assessed. The user must provide:

- Monthly heat and cooling loads.
- Monthly peak loads (optional), with number of peak hours.
- Specifics of heat pump system (entering water temperature, inputs and outputs i.e. performance map).
- Thermal properties of the ground (thermal conductivity, volumetric heat capacity and undisturbed ground temperature).
- Configuration of heat exchanger, borehole diameter, U-tube diameter, material, ground thermal properties.
- Volumetric heat capacity, density and its flow rate of working fluid.

When the data has been entered, the sizing calculation can be performed for a time period of 25 years to give the minimal depth required to meet specified minimum and maximum temperatures of the heat transfer fluid entering the heat pump. The recommended borehole length with maximum and minimum average ground temperature. Maximum and minimum temperatures under peak conditions with a specified depth, heat rejection rate (W/m), work (kWh), maximum and minimum entering water temperatures.

3.9.5. GS2000 calculation method

Twelve different configurations can be modeled, seven horizontal and five vertical. The heat exchanger i.e. borehole diameter, distance between them (if more than one), and depth of the borehole under the ground. The pipe material, pressure rating, wall thickness, ground temperature properties and heat pump characteristics are also a requirement as an input for this method. To outline these following inputs:

- Maximum/Minimum entering fluid temperature and flow rate through loop.
- Heat pump heating capacity and respective coefficient of performance.
- Heat pump cooling capacity and respective energy efficiency ratio (EER).

Either a single year or multi-year analysis can be run. The results of the sizing simply give the recommend depth/length of the heat exchanger and accurate sizing heat pump systems.

3.9.6. Earth energy designer (EED) calculation method

Ground properties such as thermal conductivity, volumetric heat capacity, surface temperature and geothermal heat flux must be entered in this method. The pipe arrangements which can be considered are:

- Coaxial (one tube inside another).
- Single, double or triple U-pipe(s) per borehole. For these shank spacing is also required i.e. distance between the center of the up and down tubes.

The simulation period can be selected (up to 25 years) as well as the starting month. The output file from an EED simulation should include:

- Design data entered and required length of boreholes.
- Average monthly specific heat extraction rate (W/m)
- End of month mean fluid temperatures for years 1, 2, 5, 10 and 20.
- Minimum/maximum means fluid temperature with month of occurrence.

3.9.7. GCHP calc method

This program utilizes Kavanaugh's cylindrical source method; variation in load and the switching on/off of the heat pump is represented by four (hour, daily, monthly and annual) cyclic pulses of building heat. Heat losses and gains need to be entered as well as the equivalent full load heating and cooling hours. The program specifies the minimum allowable entering water temperature allowable to the heat pump for heating (maximum for cooling). The following key factors must be entered:

- Design inlet heat pump heat/cool temperatures and flow.
- Undisturbed ground temperature.
- Thermal conductivity, diffusivity of ground.
- Tube qualities i.e. diameter, flow regime (laminar, transient turbulent etc.).
- Number of boreholes, distance between them and arrangement.

The program output gives required bore lengths for minimal or a high rate of groundwater movement alongside a summary of the design data inputted.

3.9.8. Earth coupled analysis (ECA) method

The calculation methodology is utilized by ASHRAE in their design manual for closed loop GCHPS. The heat load is required along with temperature range where no heating or cooling will take place i.e. heat pump inactive. The programs output is a calculation of the required pipe loop length necessary for heating and cooling a building with the heat pump, soil, and weather conditions specified. Pressure drops are also reported. An option can be exercised to input costs for drilling, pumps, antifreeze etc. and therefore generate a complete bill of materials for the project.

3.10. Section conclusion

This introductory overview should have explained the generic theory behind heat pumps, and how they operate and the main components which make up these systems have been covered as have how they can be utilized, i.e. for heating or cooling, and in terms of operational modes with the benefits of utilizing a heat pump system have been outlined. A description on how the earth can be used as a heat source and the key geological and hydrological factors to consider when planning GCHPS. From this point, the different methods of extracting heat have been described. From direct and indirect systems, open heat pumps using ground water as the heat carrier and the different types of ground loop collectors presented. Also, it's highlighted that there is a plethora of different systems available to assist the installer or designer in selecting a suitably sized ground heat exchanger for a GCHPS. In order to do this, the software must be able to assess the heat transfer relationship between the working fluid and surrounding ground the packages highlighted vary in terms of calculation method. Finally, a brief overview of potential uses for GCHPS technology has been given.

CHAPTER 4

SYSTEM EXPERIMENTAL SETUP AND THEROTICAL ANALYSIS

4.1. Introduction

The objective of this chapter is to clarify GCHPS components experimental set up performed for heating and cooling operations and computing the performances of system. GCHPS was installed at the south side of laboratories of mechanical engineering department in University of Gaziantep and composed of a test room which is consisted of mainly four different walls and a roof, a vertical slinky ground heat pump system, measurement elements, data logger, and personal computer. Each components of system with instrument and sensors used will be outline. Moreover, calibration procedure will set forth.

The second part of this chapter is giving the methodologies for calculating and a performance analysis of GCHPS. The methodology consists of calculation of heat transfer at condenser, evaporator, heat loss or gain in liquid pipeline from closed loop ground cycle to condenser or evaporator unit, the temperature of the aqueous mixture at different depth under the ground, the pressure and mass flow rate for both of the aqueous mixture in the ground cycle and the refrigerant in the heat pump cycle, and energy consumption via circulating pump, fan, and compressor. Furthermore, estimation of COP_{HP} alone and COP_S were presented.

4.2. Experimental system and setup for the GCHPS

The experimental system is about installation of the Ground Couple Heat Pump System (GCHPS). There are several components of the GCHPS, which are a test room, a vertical slinky type Ground Heat Exchanger (GHE), measurement elements, thermocouples or sensors, data logger, and personal computer. Each components of the GCHPS with instrument and sensors used will be explained and calibration procedure for some of the measurement elements.

4.2.1. Testing rooms

A testing room having dimensions of 4 m x 4 m x 2.7 m was constructed. The room has four different walls and a flat roof, which are commonly used in Turkey [61]. Figure 4.1 represents a photograph of the testing room. The test room has four different walls and a flat roof. Schematic representations of the walls are shown in Figure 4.2. The walls, which are commonly used in Gaziantep and Turkey, are brick, blokbims, autoclaved aerated concrete (AAC), and XPS walls. All walls have no insulation materials except the XPS wall. The roof of the test room is consisted of concrete, plaster, insulation material and plaster. The plaster is only applied on inside surface of the roof, which is seen from the figure 4.2

Figure 4.1. Photograph of the testing room.

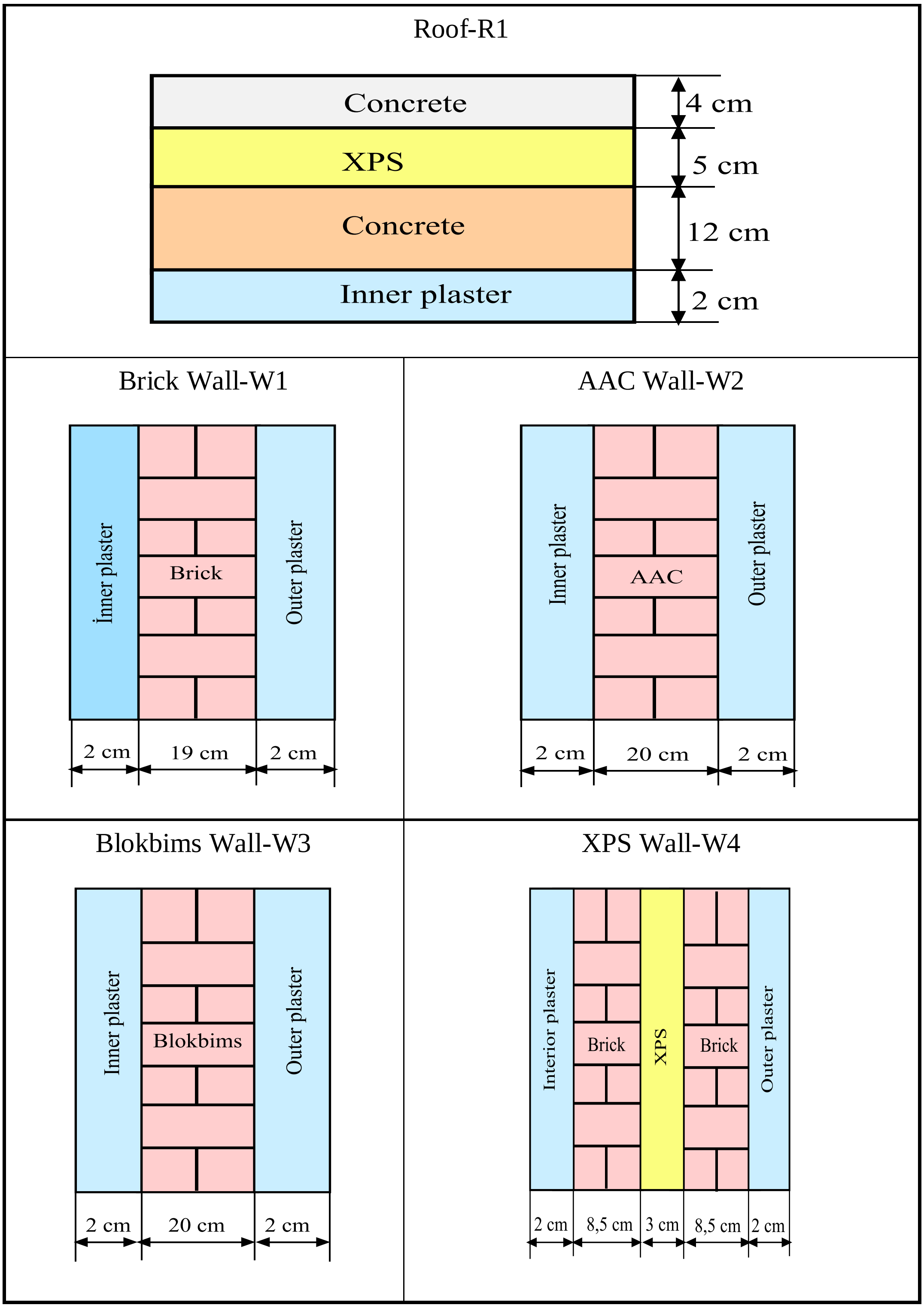

Figure 4.2. Schematic representation of the walls and roof with dimensions [61]

4.2.2. The ground source heat pump (GSHP)

The Ground Source Heat Pump (GSHP) is the second main important part of the Ground Couple Heat Pump System (GCHPS). Schematic representation of the GCHP is indicated in figure 4.3. The heat pump consists essentially of compressor, condenser, evaporator, reversing and expansion valves, dryer and flash tank. Indoor heat exchanger is used as a condenser for heating, and used as an evaporator for cooling operation for the test room. Heat exchange between the room air and refrigerant in the indoor unit is performed by using a fan operating with variable speed. The fan, which operates with variable speed depending on heat requirement of the room, locates on the indoor heat exchanger. Outdoor heat exchanger is used as an evaporator during heating season or winter season and as condenser during summer or cooling season. The outdoor heat exchanger is a plate type heat exchanger which transfers heat from the ground to refrigerant or from the refrigerant to the ground. In order to perform heat exchange between the ground and refrigerant, water is circulated through slinky pipe buried vertically in the ground. The ground properties for drilled pit were within the slandered range. The ground type was clay with some rocks. The moisture content was subjected to the climate condition. The inside design air temperatures which were taken as 20°C in heating season and 25°C in cooling season. The heat pump has a heating capacity of 4 kW and cooling capacity of 3.5 kW. The room cooling load was computed as 3.2 kW [62].

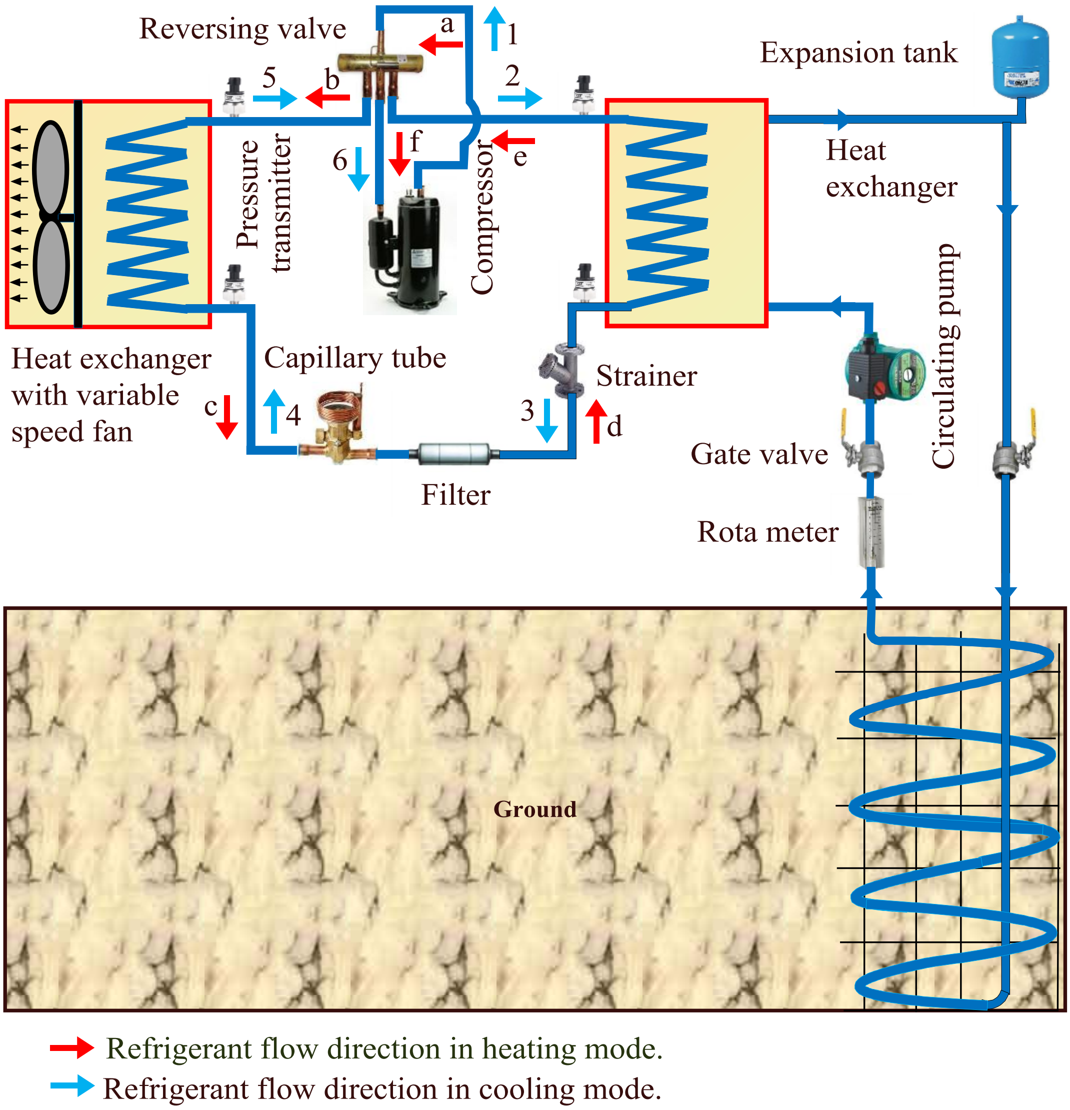

Figure 4.3. Schematic diagrams of the GCHPS.

4.2.3. Slinky type ground heat exchanger (GHE)

Pictures of the slinky type Ground Heat Exchanger (GHE) are shown in figure 4.4 and 4.5 before and after burying, respectively. It is used as a heat exchanger under the ground. Depth of the GHE is about 3.0 meter from the ground surface. It is spirally shaped with a diameter of 3.5 meter, and connected with steel bars to be able to locate with equal depth. Pipe is made from polyethylene, which doesn't require extra fittings to change direction. It has 1.905 cm internal diameter and 50 meters length. It is buried under the ground, and connected to outdoor heat exchanger or plate type heat exchanger. It used to extract heat from the ground during heating season, and to reject heat to the ground during the cooling season.

Circulation pump used in the GHE cycle is a type of wet-rotor circulation pump produced by Grundfos. An air purger exists on the pipeline to purge air in circulated liquid to ambient. Flash tank was connected to pipe line to compensate pressure changes due to cooling and warming of liquid and leakage as well.

Figure 4.4. Slinky type ground heat exchanger before burying

4.2.4. Measurements of performance parameters

Measurements should be performed to obtain thermal performance of each part of the GCHPS, and GCHP and system. In order to measure all nodal temperatures and pressures, mass flow rate of the aqueous mixture, and power consumed by circulating pump, fan and compressor during the experiment, the testing system was installed. The testing system with the measurement elements was indicated in figure 4.7. It was composed essentially of power and pressure transducers, thermocouples, flow meters, data logger and personnel computers PC.

Power consumptions of electrically driven compressor, fans, and circulating pump of the GCHPS were measured via voltage and current transducers. Thus, energy consumption of each device was calculated by using the measured values of the currents and voltages. Compressor used in the heat pump cycle is DC brushless rotary compressors. Discharge thermistor also exists to prevent compressor over loading.

Inlet and exit pressures of refrigerant flowing through the GCHP components which are compressor, condenser, expansion device and evaporator were measured via pressure transducer. The pressure transmitters used in the test system have range of 0-30 bar gauge and 4-20 mA output.

All nodal temperatures on the GCHPS cycle, ambient and inlet room air temperatures were measured using K-type thermocouples, and recorded for each minute on computer disc. Inlet and outlet states of the refrigerant can be determined if two independent thermodynamic properties are known. Two independent measurable variables at the inlet and outlet states are pressure and temperature. Temperatures and pressures of the refrigerant at inlet and outlet states of the indoor heat exchanger were measured in the same manner just expressed. All the thermocouples were connected to IOTEH 3005 labeled data logger with expansion module that was connected to the PC to collect the data.

The mass flow rate of the aqueous (or antifreeze-water) mixture was measured by a Rotameter. The inlet and outlet temperatures of the aqueous mixture were measured by K-type thermocouples immersed into the suction and discharge pipes. Operation pressure through the pipeline was kept as 1 bar gauge throughout the whole experiment period.

On the control side, the heat pumps have their own control unit. Electronic board of heat pumps sense indoor temperature, indoor coil temperature, ambient air temperature and control heat transfer by regulating compressor speed and linear expansion valve opening.

After connecting the measuring equipment on the GCHPS, they were calibrated. Before starting the data transfer, measuring system was checked for each of the nodal points. The measurement was not started until the system run faultless.

Measurements of the performance parameters which are pressures, flow rate, temperatures, currents and voltages will be explained in the following subsections.

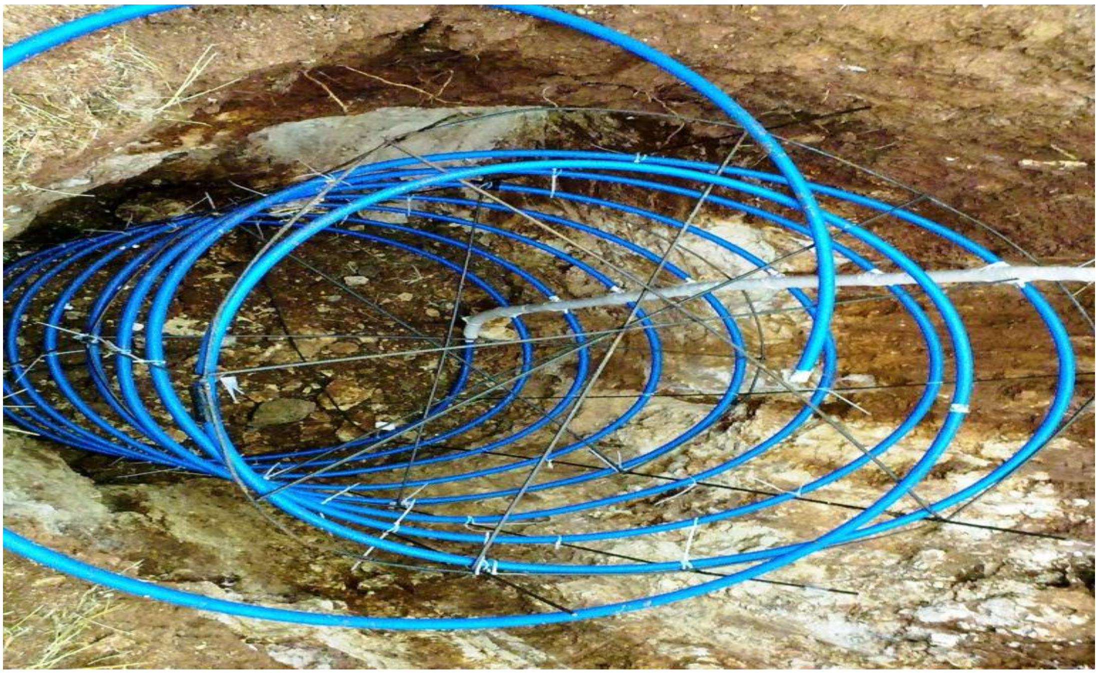

Figure 4.5. Slinky type ground heat exchanger with steel frame located vertically in the ground.

4.2.5. Pressure measurements

Pressure measurement in the GCHPS at condenser and evaporator inlets and outlets were measured by Carel SPKT0031C0 coded piezo-resistive pressure transmitter having 0-30 pressure measurement range and 4-20 mA output. Schematic representation of the pressure probe developed by CAREL for the application in the refrigeration and air conditioning sectors is shown in figure 4.6. Technical specifications of the transmitter are given as:

- Power supply : 8-28 V DC
- Output : 4-20 mA
- Operating conditions : -40 to 135 °C
- Linearity: ± 0.5% Fs typical ± 1%FS max.
- Total precision : ± 1% FS typical, ± 2% FS max (0-to 50 ºC),± 4%FS max (-20 to 50ºC).

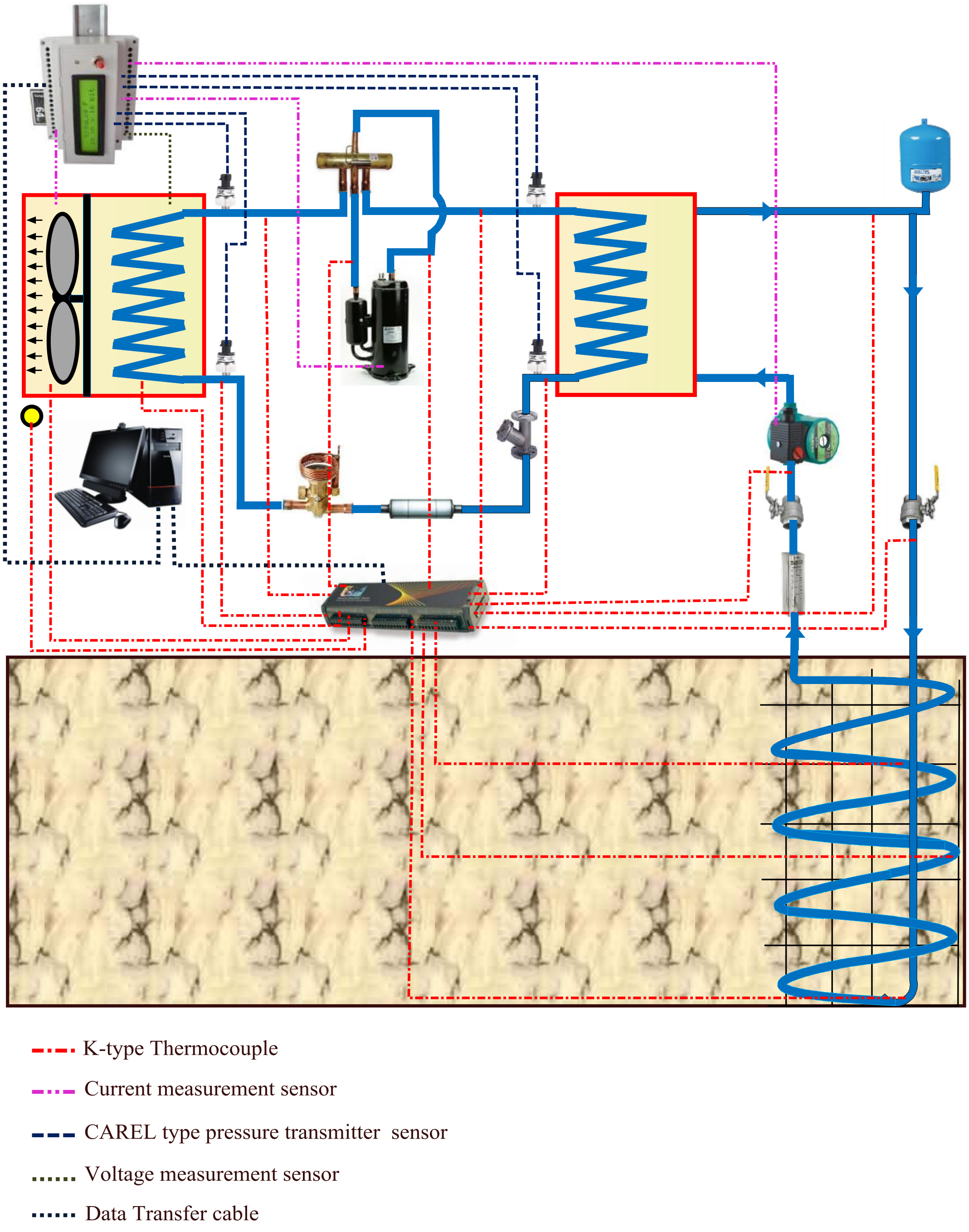

Figure 4.6. Schematics of the measuring elements on the GCHPS

4.2.6. Flow rate measurement

Flow rate is one of the required parameter to be known for analysis of energy systems. Mass flow rate of the GCHPS unit line of aqueous mixture was measured by dwyer series of visi-float flow meters with direct reading scales for air or water, (figure 4.7). Operation temperature of the line didn't exceed 48°C. Specifications of the used flow meter are given below:

- Service : Compatible gases and liquids
- Body : Acrylic plastic
- Float : Stainless steel
- Pressure and Temperature limits : 6.9 bar, 48oC
- Accuracy : 2% of full scale
- Mounting orientation : Vertically

Followings cautions given by manufacturer were taken into consideration during location of the flow meter. It is good practice to approach the flow meter inlet with as few as elbows, restrictions and size changes as possible. In order to avoid turbulence which can occur as result of drastic size changes, inlet piping should be as close as to the flow meter connection size. Discharge piping should be at least as large as the flow meter connection. For pressure fed flow meters on water back pressure will not affect the calibration of the instrument since flowing medium is incompressible**.**

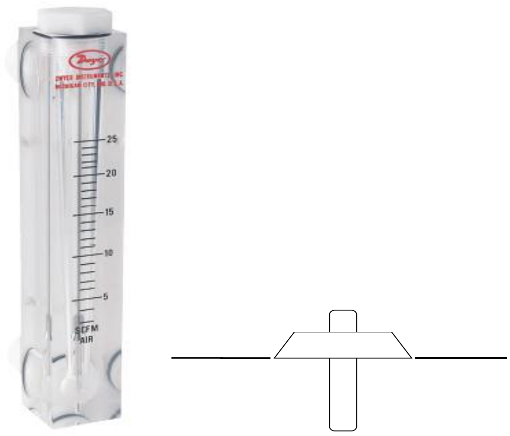

Figure 4.7. Rotameter and its visible float position during reading.

4.2.7. Instrumentation calibration

Temperature, pressure, mass flow rate, electrical current and voltage were measured during the experimental study, which can be put in order as fallows temperature measurement: K-type of thermocouples is used for measuring temperature. Pressure measurement: 0-30 Bar ranged piezoresistive pressure transducers having output of 4-20 mA were used. Flow measurement: A Rota meter was used for measuring flow rate of aqueous mixture. Electrical current measurement: 0-5 ampere ranged current transducer having 0-20 mA output. Voltage measurement: 0-230 ranged voltage transducer having 0-20 mA output.

It is known that calibration is very important for correct measurements of the performance parameters. They are temperature, pressure and mass flow rate. For that reason, thermocouple, pressure and flow meter calibrations will be explained in the following subsections.

4.2.8. Thermocouple calibration

Water bath, heater with thermostat, and thermometer were used to calibrate the thermocouples. Water was heated until the set temperature of the thermostat via a heater. A mixer was used to blend the water to obtain uniform temperature. Mercury in glass thermometer having sensitivity of 0.1^oC and range of $0\text{-}100^oC$ was used in calibration. For calibration purposes, the temperature range used was $15\text{-}75^oC$ with temperature increments of 10^oC. Each of thermocouples connected to IOTECH 3005 personal daq immersed in to the water in the bath. IOTECH data logger allows the user to select thermocouple types such as T and K type, and to apply an internal linear correction to each channel mv readings. Firstly, K-type thermocouple readings were checked for each channel simultaneously. After water temperature becomes stable, temperature is noted, and then allowed the data logger to receive data for 30 seconds with one second increment. The procedure was repeated for three different set temperature values. Data for 30 temperatures were collected for each set; data were averaged and compared to readings of mercury in glass thermometer. The average temperature readings by data logger show that for each channel about $\pm0.50^oC$ deviation from the readings of mercury in glass thermometer. Appendix A.

4.2.9. Pressure measurement calibration

Carel SPKT0031C0 pressure transducers were used to measure pressure at inlet and outlet of the condenser and evaporator of the heat pump cycle. Transmitter’s range is expressed in relative bars. Gigalog data logger was allowed to make scan for 20 second for each test measurement. With the aid of a calibrated voltmeter to take the direct readings, when the system is on within the same time interval. Average of each scan with fabric calibration results was done for these pressure ranges. 0-10 vdc output of the transmitter was changing almost linearly with change voltmeter readings. The error rate was about 2% of the full display. Appendix B.

4.2.10. Flow meter calibration

The flow meter was calibrated in-situ using a stopwatch, bucket and precision weight scale. First, the empty bucket was placed on the weight scale and zeroed. An outlet valve to the piping system then was adjusted until the desired flow rate was reached. The stopwatch and water was started simultaneously and the bucket was filled to a predetermined point. Once this point reached, the stopwatch and water flow to bucket was stopped and then bucket was weighed and value recorded. The recorded for each one was then used to calculate the actual flow rate given by equation.

$$\dot{m} = \frac{m}{t} \tag{4.1}$$

Where m: mass of water or aqueous mixture filled into bucket, t: time passed to fill m, kg of aqueous mixture to bucket, and: calculated mass flow rate of the aqueous mixture. 2% of the full indicator is the error percentage of flow meter.

4.3. Thermal analysis of the GCHPS

In this section, thermal analysis about the Ground Coupled Heat Pump System (GCHPS) is presented. The GCHPS consists of two cycles which are heat pump (HP) cycle and closed loop water in the ground cycle for (GSHPS). Refrigerant and water are circulated through the HP with closed loop ground water cycle. Main components of the HP cycle are evaporator, compressor, condenser, expansion device. GSHPS cycle is composed of a circulation pump, plate type heat exchanger and polyethylene pipe. Properties of these components will be given below; expressions related with the components of the GCHPS will be presented in the following sections.

Table 4.1. Main specification and characteristics of the GCHPS components.

Main circuit	Element	Technical specification
GCHPS circuit	Heat ground heat exchanger with total length 50 m [54].	Slinky ground coupled heat pump lead vertically in a Pit (1.5 width * 3.00 depth) m, plastic pipe material Polyethylene (PE) blue color, TS 418-2, EN 12201-2 ,ISO 9001:2008 nominal pipe diameter 0.01905 m.
	water-antifreeze mixture circulation pump	Type of wet-rotor circulation pump produced by Grundfos, Liquid temperature range: -25 - 110 °C, Maximum operating pressure: 10 bar, with 3H speed, maximum power input: 140 W.
Refrigerant circuit	Compressor	Type: DC brushless rotary compressors; model: KNB073FDVHC; 65 – 194 V d. c. inverter 75 – 345 Hz. volumetric flow rate: 7.6 m3/h; speed: 2900 r/min; the rated power of electric motor driving: 2 HP (1.4 kW); watt range (2310 – 700), refrigerant: R-410A; single phase; capacity: 4 – 3.5 kW (at evaporating/ condensing temperature of 0/54.5 _°C); connection: suction line: 5/8 in, discharge line: 3/8 in.
	Heat exchanger (HE)	Manufacturer: Tunc _ Technique; type: TTE 3; capacity: 8.2 kW (with R 410A), material: copper and inner cooling aluminum
	Capillary tube	Copper capillary tube, 2 m long with inside diameter: 0.0015 m
	Expansion tank	Manufacturer: Zimmet, type: 520/L, capacity: 15 l; max pressure: 3 bar
condenser fan circuit or air circuit	Air conditioning indoor with variable speed air fan	Manufacturer: Mitsubishi Thailand, model: MUZ-GA35VA, input power 0.25 Amp when compressor is off.

4.3.1. The GCHPS cycle for heating operation

Basic principle of a GCHPS is heat removing from low temperature heat sources that can be underground soil, scree, etc. and heat rejection to the high temperature heat sink that air in required medium for the heating season. Although the heat rejection is major aim in heating season, temperature of the heat source play excellent role in performance of a heat pump system. Simply the higher temperature of heat source means higher COP of the system. In cooling application, heat source and heat sink are interchanged and major aim becomes heat removing from a medium. But heat rejection temperature is one of the main factor acting on the COP of the system. Lower the temperature of heat rejection the higher the COP of the system. The temperature of the ground source for closed loop type doesn't vary with seasons as rapidly as air source temperature. So, ground soil is a preferable as a heat source and heat sink compared to air. Since the higher temperature of the ground cycle give the higher COP of the GCHPS. Heat rejection to the ground sink from refrigerant and heat removing from ground by refrigerant can be made using heat exchanger. To perform heat exchange process, water ground source must be transported by irrigation. Calculation procedure for finding heat transfer will be presented during the transportation. Design of any heat pump unit completely is depend on the heat rejection rate from the condenser if the heat pump is used for heating purposes. Heat transfer occurs at the evaporator unit if the purpose is cooling. Heat transfer from the ground soil is computed using simple energy equation.

$$\dot{Q}_{ev} = \dot{m}_b C_{pb} \left[T_{s,b} - T_{r,b} \right] \qquad (4.2)$$

From this equation, we can elicitation:-

Qev : Rejected heat from aqueous mixture to refrigerant 410A.

Ts,b : Supplied aqueous mixture to evaporator

Tr,b : Return aqueous mixture from evaporator

Cpb : Specific heat of aqueous mixture in the pipe line.

Ethylene-glycol basis antifreeze mixture has to be added to the closed cycle using the ground source because of preventing water to be frost. Thus, evaporator side heat exchanger cannot be damaged. Equation to calculate specific heat of pure ethylene glycol is given in Touloukian [63]. The equation presented below gives absolute deviation of 1 % for pure ethylene glycol.

$$C_p = \left(\sum_{n=0}^{3} a_n \left(T + T_{ref} \right)^n \right) 4.184 \qquad (4.3)$$

From this equation, we can elicitation:-

C_p : Specific heat of the ethylene glycol

T : Temperature (°C)

T_{ref} : Reference temperature

The constants of a_n; $a_0 = 0.016884$, $a_1 = 0.00335083$, $a_2 = -0.000007224$, $a_3 = 0.000000007618$

The value of 4.184 is the conversion coefficient used to convert unit from calorie/gram°C to kJ/kg. K. Specific heat capacity of aqueous mixtures is calculated by following equation:-

$$C_p = (C_{p1}N_1 + C_{p2}N_2)(1 + A_1N_1N_2) + A_2(A_3 - T)N_1N_2 + A_4(A_3 - T) \tag{4.4}$$

From this equation, we can elicitation:-

N denotes mass fractions

$A_1 = 0.223375729$

$A_2 = -0.005033512$

$A_3 = 122.398371$

$A_4 = -0.00119436$

Equation (4.4) is valid in temperature range of - 35°C to125 °C. Here, brine, mixture of ethylene glycol and water are used as heat transfer fluid. Subscript b denotes brine, and e is used for evaporator. Temperature of the brine at inlet and outlet of the evaporator and mass flow rate of the aqueous mixture are measured so that heat rejection to the aqueous mixture can easily be calculated. Heat transfer rate rejected from aqueous mixture has to be equal to heat removed from R-410A. Thus, Heat rejected to refrigerant can also be expressed by product of mass flow rate of refrigerant and enthalpies at inlet and outlet states, which is given as:

$$\dot{Q}_{ev} = \dot{m}_r(h_1 - h_4) \tag{4.5}$$

In this equation, unknown parameters are enthalpies of the refrigerant. Enthalpies of states can be obtained from thermodynamics table or EES (Engineering Equation Solver) by entering measured temperature and pressure of refrigerant at inlet and exit of the evaporator. After determining mass flow rate of refrigerant, since inlet and outlet pressure and temperature of the compressor are measured. Energy loaded to refrigerant by compressor can also be determined by:

$$\dot{Q}_{comp} = \dot{m}_r \left(h_2 - h_1\right) \tag{4.6}$$

This is added energy to refrigerant by compressor but this does not mean compressor consume that much of energy. Energy consumed by compressor both heating season and cooling season can be calculated by electrical power relation.

$$\dot{W}_{comp} = \frac{I_c V_c \cos\varphi}{1000} \tag{4.7}$$

Where, I_c is the electrical current, V_c is the voltage of the compressor, and $\cos\varphi$ is the power factor of the compressors. The compressor efficiency can be defined using input energy by the electrical energy measured and actual energy, which can be expressed as:

$$\eta_{comp} = \frac{\dot{Q}_{comp}}{\dot{W}_{comp}} \tag{4.8}$$

Heat rejected to indoor air from refrigerant is

$$\dot{Q}_{cond} = \dot{m}_r \left(h_3 - h_2\right) \tag{4.9}$$

Enthalpies refrigerant at the inlet and the outlet of the condenser are obtained from thermodynamic property tables or EES entering measured pressure and temperature values. Coefficient of performance of the heat pump unit alone is:-

$$COP_{HP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \tag{4.10}$$

Coefficient of performance “COP” of the GCHPS is equal to heat rejected by the condenser over total energy consumed by the compressor, circulation pump and fan. It is given by the expression:

$$COPs = \frac{\dot{Q}_{cond}}{\dot{W}_{comp} + \dot{W}_{pump} + \dot{W}_{fan}} \tag{4.11}$$

Where, electrical powers consumed by the pump and the fan for both heating and cooling season are:-

$$\dot{W}_{pump} = \frac{I_p V_p \cos\varphi}{1000} \tag{4.12}$$

$$\dot{W}_{fan} = \frac{I_f V_f \cos\varphi}{1000} \tag{4.13}$$

Where, I_p, V_p and I_f, V_f are electrical currents and voltages of pump and fan respectively.

4.3.2. The GCHPS cycle for cooling operation

The same analysis is followed during cooling mode. It is known that heat rejected from heat pump condenser to the ground side, during cooling season is higher than heat removed from ground cycle during heating season. Thus, mass flow rate of liquid circulated through ground-condenser loop during cooling season has to be higher than that of circulated through ground-evaporator loop during heating season for same temperature difference. Besides, it is not necessary to use aqueous mixture in the cooling mode, since ground closed loop-side liquid getting warmer because of heat rejected from condenser. Temperature of the water at inlet and outlet of the condenser and mass flow rate of the water are measured so that heat rejection to the water can easily be calculated.

$$\dot{Q}_{cond} = \dot{m}_w C_w \left[T_{r,w} - T_{s,w}\right] \tag{4.14}$$

Heat transfer rate removed by water has to be equal heat rejected by refrigerant of R410A. Thus, Heat rejected from the refrigerant can also be expressed by product of mass flow rate of refrigerant and enthalpy difference between inlet and outlet states.

$$\dot{Q}_{cond} = \dot{m}_r \left(h_3 - h_2 \right) \tag{4.15}$$

Here, unknown parameters are enthalpies and mass flow rate of the refrigerant. Enthalpies of the states can be obtained from thermodynamics table or EES (Engineering Equation Solver) by entering measured temperature and pressure of refrigerant at inlet and exit of the condenser. So that, mass flow rate of the refrigerant is deduced from above equation. After determining mass flow rate of refrigerant, since inlet and outlet pressure and temperature of the compressor are measured. Energy loaded to refrigerant by compressor can also be determined by:-

$$\dot{Q}_{comp} = \dot{m}_r \left(h_2 - h_1 \right) \tag{4.16}$$

Heat removed from indoor air by refrigerant is:-

$$\dot{Q}_{ev} = \dot{m}_r \left(h_1 - h_4 \right) \tag{4.17}$$

Enthalpies refrigerant at the inlet and the outlet of the evaporator are obtained from thermodynamic property tables or EES by entering measured pressure and temperature values. Coefficient of performance of the heat pump unit alone in the cooling mode is:-

$$COP_{HP} = \frac{\dot{Q}_{ev}}{\dot{W}_{comp}} \tag{4.18}$$

Coefficient performance of the whole GCHPS during cooling season is

$$COPs = \frac{\dot{Q}_{ev}}{\dot{W}_{comp} + \dot{W}_{pump} + \dot{W}_{fan}} \tag{4.19}$$

4.4. Section conclusion

The experimental setup for the GCHPS was outlined, and the system characteristics were explained as well as measuring elements or sensors and instrument calibration. Moreover, the methodology for calculating performance parameters of the GCHPS was given. Heat transfer at condenser, evaporator, heat loss or gain in liquid pipeline from ground heat exchanger (GHE) to condenser or evaporator unit, and energy consumption of pump, fan and compressor were calculated by using the given expressions in Section 4.3. And also, COP_{PH} of the heat pump units alone, COP_S, and heat loss and gain of the room were computed from the experimental measured data.

CHAPTER 5

RESULTS AND DISCUSSIONS

5.1. Introduction

In this section, variation of major performance parameters related with the experimental system which is called as "Ground Coupled Heat Pump System GCHPS" are presented in graphical and table forms for heating and cooling operations. The performance parameters are computed by using measurement values from the experimental GCHPS. The measurement values are the ambient and inside room temperature, temperature of antifreeze water mixture in the ground loop cycle at different ground depth, pressure and temperature of refrigerant R-410A used in the heat pump cycle at each inlet and exit states of the heat pump components, currents and voltages during operating of the circulating pump, fan, and compressor, and the variation of antifreeze-water flow rate.

The performance parameters for both of the heating and cooling systems are heat gain or rejection to the testing room, heat gain or rejection from or to the ground, power consumed by the circulation pump, condenser or evaporator fan and compressor, performance of the evaporator or condenser, coefficient of performance for the heat pump (COP_{HP}) and for the whole system (COP_S). All of these parameters are computed by using the measurement values. The results obtained from computation are given as figures and tables, and they are discussed in this chapter. The results for the heating and cooling operations are given in two different main sections of this chapter. In addition, comparisons of some performance parameters for the Ground Heat Pump System (GHPS) with the same parameters for the other heat pump systems are presented in this chapter.

5.2. Experimental results for the GCHPS for heating operation

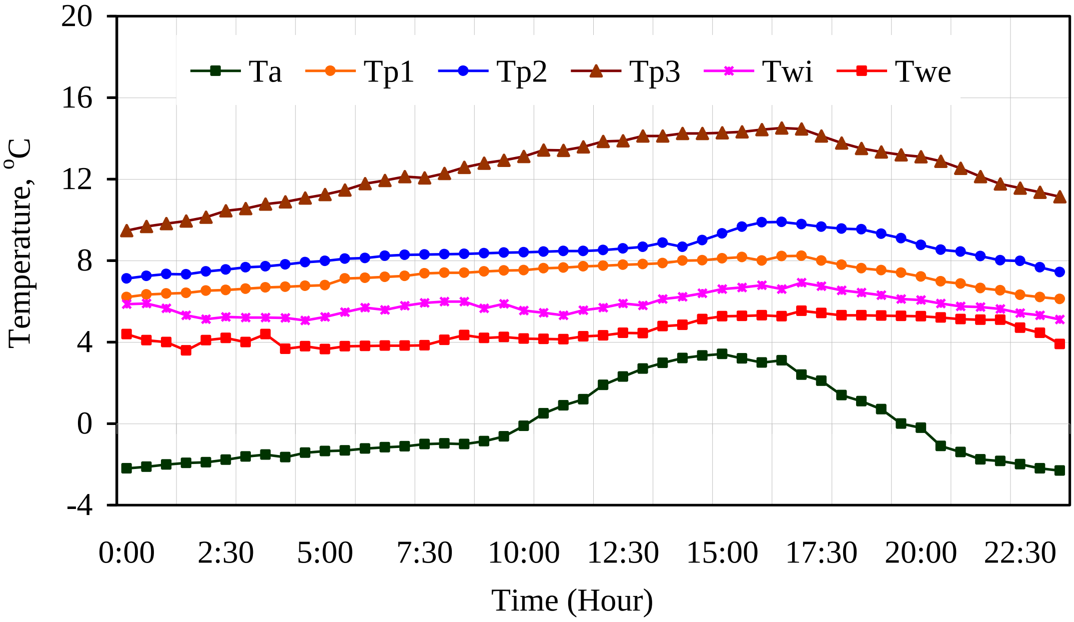

Figure 5.1. Hourly variation of temperatures for ambient air, water, and ground. (February 22, 2012).

Figure 5.1 illustrates variation of hourly inlet "Twi" and exit "Twe" temperature for the aqueous mixture that flows through the evaporator, and ambient air and ground temperatures at a different depth. It is seen from the figure that ambient air temperature changes between about -3 ^{o}C and 3 ^{o}C. That is, mean ambient air temperature is about 0 ^{o}C. Temperature variations in the ground at a different depth are shown in the figure 5.1. "Tp1", "Tp2", and "Tp3" are pipe surface temperatures at depth of 0.9 m, 1.5 m and 3 m, respectively. While the maximum ambient air temperature "Ta" didn't exceed 3.7 ^{o}C, these temperatures were measured as 8.8 ^{o}C, 11^{o}C and 14.5 ^{o}C respectively. As it is examined, temperature increases with respect to depth. Source temperature has important effect on COP_{HP} and COP_{S} for the GCHPS. Since, it is understood that the main effective parameter on the GCHPS performance is the variation of ground (source) temperature.

Ground or antifreeze water "aqueous" mixture temperature at different depth is very important. Since the temperature which gives an idea for heat extraction or rejection and COP of the heat pump or GCHPS. For that reason, figure 5.2 is depicted for the relationship between the aqueous mixture temperature at 3 m depth and the system performance COP with total electric consumption “Ws”. It is seen from the figure that COP_{HP} and COP_S increase gradually with the temperature of the aqueous mixture circulating in the ground loop cycle. At the same time, total power consumption increases but its slope is low. This indicates that heat gain from the ground is relatively higher than the power consumption. When the aqueous temperatures are about 11°C to 12°C, the COP values reaches maximum value of 3.8 and 3.2 for the COP_{HP} and COP_S, respectively. The total power consumed by the compressor, fan, and circulating pump is about 1.36 kW. The ground temperature rising in such away is accompanying with the solar radiation that provide energy to the ground. It can be attributed to the effect of increasing solar radiation near the ground surface. As a result, the higher source temperature gives the higher heat pump and system performances and vice versa.

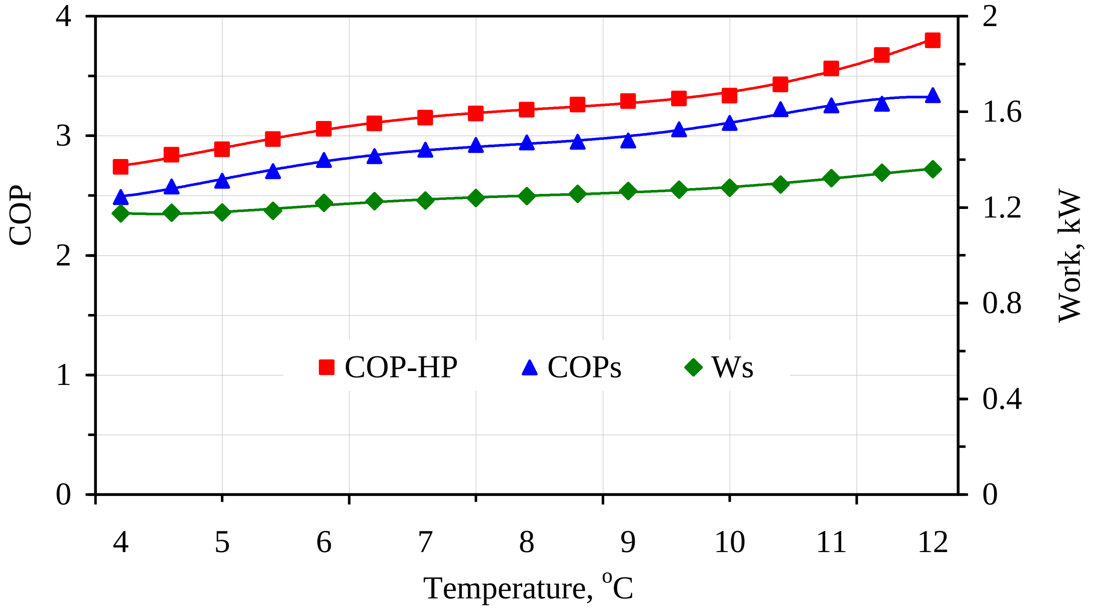

Figure 5.2. Variations of COP_{HP} and COP_S and total system work with aqueous mixture temperature. (February 22, 2012)

Figure 5.3 explains hourly variation of works consumed by circulation pump, fan and compressor. The evaporator energy extracted from the ground by GHE, and condenser energy supplied to the testing room to keep the inside room temperature as 20 °C during March 22, 2012. The heat rejected from the condenser to the room decreased with the time up to 17:00 pm. While the heat extracted by evaporator from the ground increase and reach about 0.625 kW at the same time. It was obtained that the heat extraction from the ground per unit length of GHE was 12.5 W/m. For the same period, heat rejected from the condenser to the room “Qcon” was higher and reach its maximum value 3.325 kW at the early daytime and after the sunset. The work consumed by the heat pump compressor “Wc” was 1.10 kW (increasing and decreasing with room load variation) at almost the same time. The condenser fan “Wf” and circulating pump “Wp” were consuming nearly the same amount 0.080 kW. However, compressor tends to more power consumption to meet higher heating requirements. In fact, while evaporator temperature is kept constant, compressor consume more electricity to reach higher condensation temperature. Generally, increase in compressor power consumption overcomes the useful gain at the evaporator, and so the COP decreases. The main reason for that the evaporator energy and the rate of heat extraction were so low, is the low temperature difference of aqueous between the outlet and inlet of the ground heat exchanger and oversizing some parts of the system.

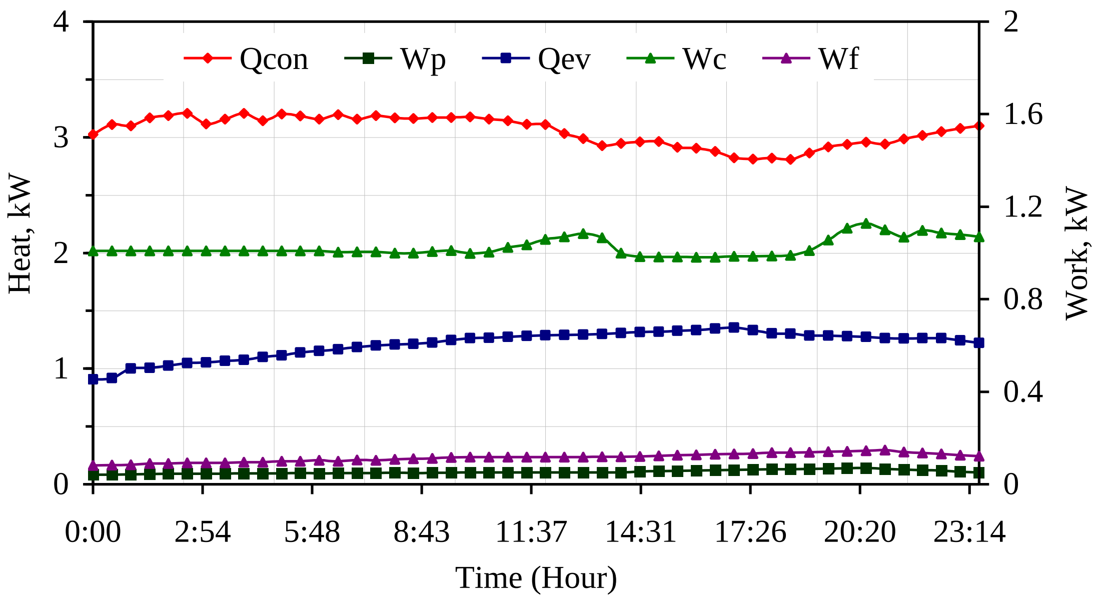

Figure 5.3. Hourly variation of heat extracted and rejected with work consumed by compressor, pump, and fan. (March 22, 2012).

Hourly variations of measured temperatures of aqueous and refrigerant with calculated COP for the heat pump and the GCHPS are indicated in figure 5.4. There are augmentation for the COP of the heat pump and system nearly in parallel reaching the maximum value between 15:45 and 16:23 pm. The COP_{HP} and COP_S increase up to value of 3.75 and 3.3, respectively. This increase was associated with the rise of aqueous mixture temperature entering the evaporator "Twi" reaching almost 11.5°C, the condenser refrigerant outlet temperature "Trco" decreases from its average value of 45°C at the peak heat requirement to reach about 41°C. The ambient air temperature reaches about 3.5°C at the same time. The inlet aqueous solution temperature of the evaporator, i.e. the outlet aqueous solution temperature of the GHE, is lower than the natural temperature of the ground, due to the heat extraction from the ground to the circulating aqueous solution. This is the most representative parameters of the ground coupling effectiveness and heat pump. In other word, the actual performance of the equipment is a function of the aqueous solution temperature produced by the GHE. Condensation temperature increases after 15:00 pm, while evaporation temperature decreases. This period shows that heating requirement of the room increases, and compressor starts to consume much power. This leads to COP of the system after this point slightly decreases.

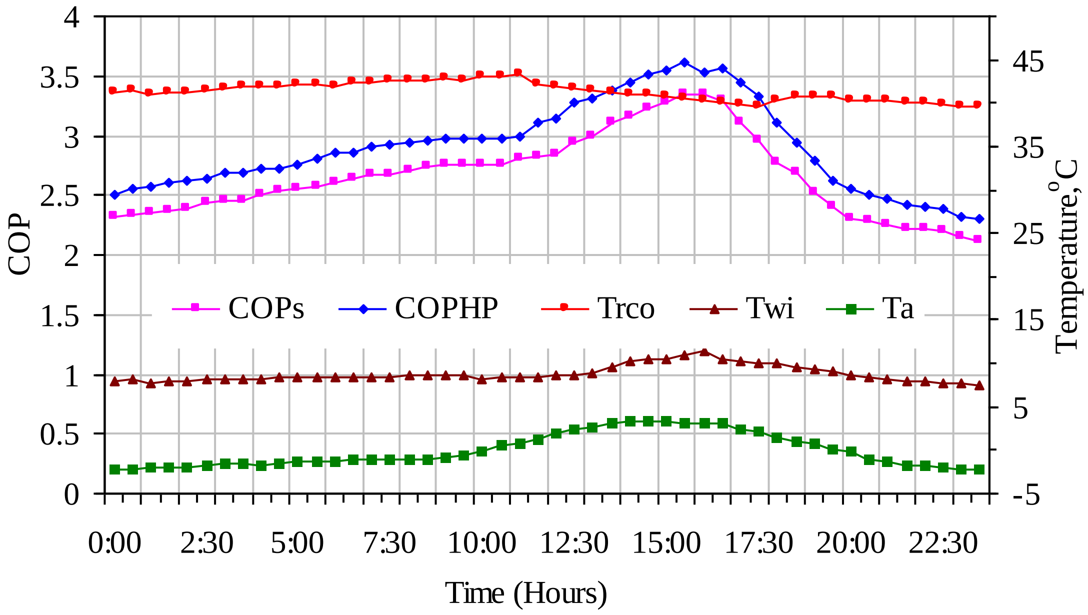

Figure 5.4. Hourly variations of COP and aqueous, ambient, and R-410A temperatures. (March 21, 2012).

Figure 5.5 represents the hourly variations in condensation temperatures of the ASHP, GWHP, and GCHP. Sharply decrease in temperature shows defrost period of the ASHP. Defrost is the one of the main problem of the ASHP during heating season. Freezing occurs at evaporator coils. Existing ice on the coils decreases heat transfer from refrigerant to air and block by air flow. Heat pumps have defrosted cycle to melt ice on the evaporator coil. The defrost cycle means that heat pump refrigerant cycle is reversed and outdoor heat exchanger become condenser. As it is clear from the figure that neither GWHP nor GSHP units doesn't require defrost since temperature of aqueous mixture supplied to evaporator is higher enough to prevent evaporator coil from frozen. Thus, net energy savings and more stable working conditions would be obtained. The outcome is longer system life. The experimental work shows the benefits of using GWHPS and GSHPS instead of the ASHPS, which are the most commonly used in all over the world.

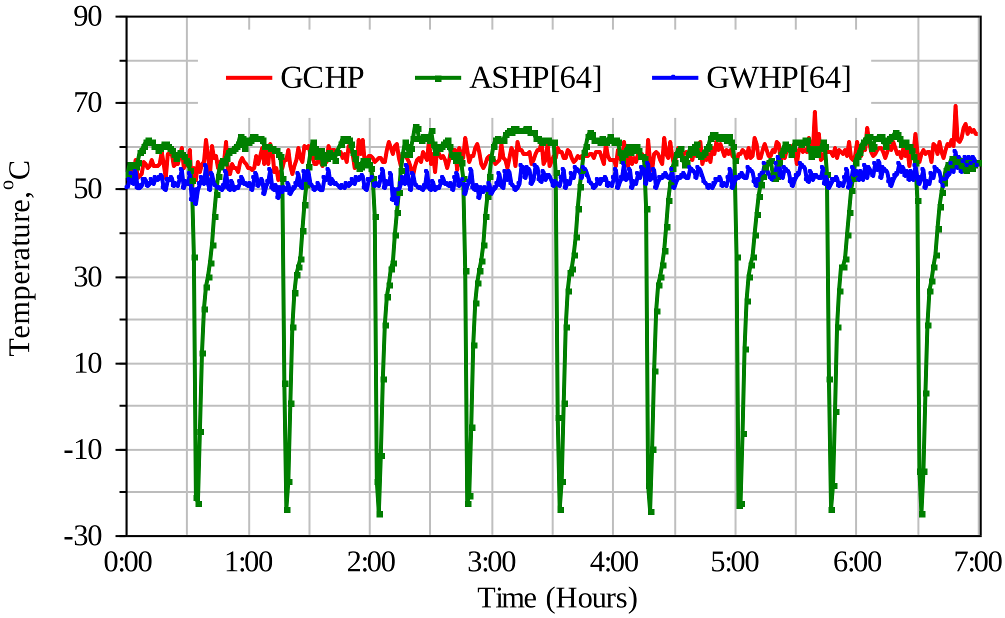

Figure 5.5. Variation of condensing temperatures for GWHP [64], ASHP [64], and GCHP. (March 23, 2012).

Figure 5.6 indicates comparison of COP for three types of heat pump systems which are the GWHP, ASHP and GCHP under the same or near climatic conditions. In the meantime, the COP_{HP} and COP_S for the three systems are computed. COP values for the three systems increases gradually with time. COPs of the all heat pumps are higher than those of the heat pump systems due to extra power consumed by the fan and circulating pump. The COP for the ASHPS is the lowest among the others. The main reason is that evaporation temperature decreases below $0\ ^{o}C$ while air temperature nearly constant, and then water vapor in the air freezes on evaporator coil. The frozen water reduces the air flow through evaporator since it fills spaces between fins, and reduce heat transfer rate from air to refrigerant. As a result refrigerant becomes colder and thus, the heat pump returns defrost cycle to melt the ice on the evaporator coils, and this situation will defiantly reduce the COP of the system. The reason of higher COP for the GWHPS and GCHPS is source from higher evaporation temperature caused by higher source temperature of aqueous mixture than air. Discontinuity in COP lines shows defrost cycle period for ASHPS. The performance of ASHPS is shown higher just after defrost period. However, heat pump unit cannot reach its steady operation, and cannot supply enough heat rejection from condenser to the testing room.

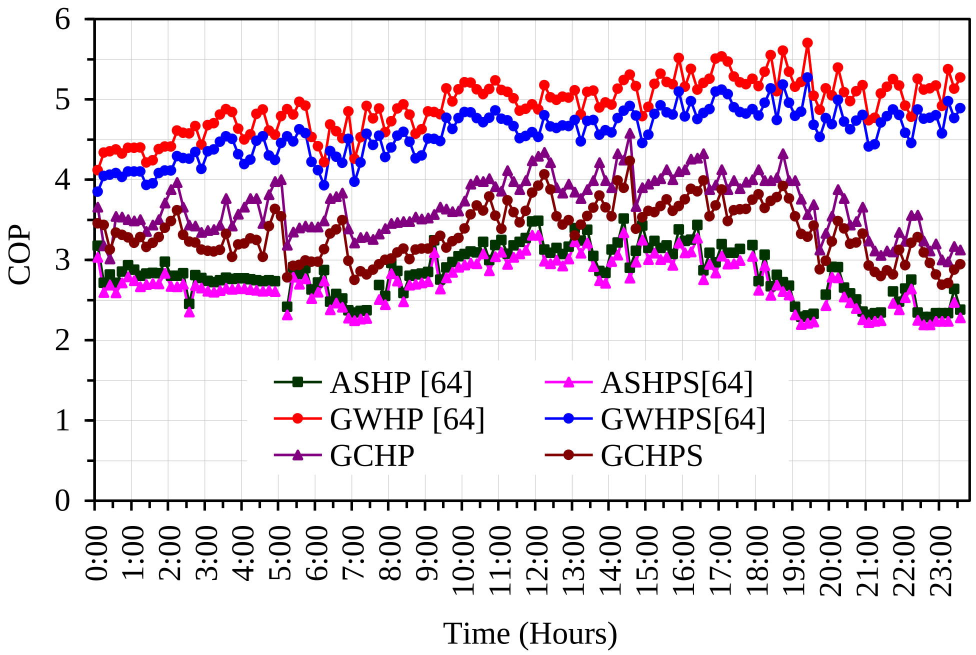

Figure 5.6. Comparisons of COP_{HP} and COP_{S} for ASHP [64], GWHP [64], and GCHP (March 27, 2012)

Figure 5.7 supports the results obtained from figure 5.6. In addition, it gives the operation conditions for each type of the compared systems. For the ASHP the ambient air is given, when the air temperature increase the system work more steady and there is no need for the defrost cycle. But when the air temperatures decrease the system performance decrease, due to the increase in power consumption by the system to meet the room load requirements. For the GWHP the aqueous mixture at the evaporator exit is given. The source heat shows that this type of systems will have the higher COP among the others. For the GCHP the pipe surface temperature of the aqueous mixture “Tp3 “at 3.0 m depth is given. The relative stability of the ground provides the circulated aqueous mixture at evaporator exit with the heat needed to be exchange with R-410A at the GHE. The COP is higher than the ASHP and lower than the GWHP. The main reason for this is the source heat type.

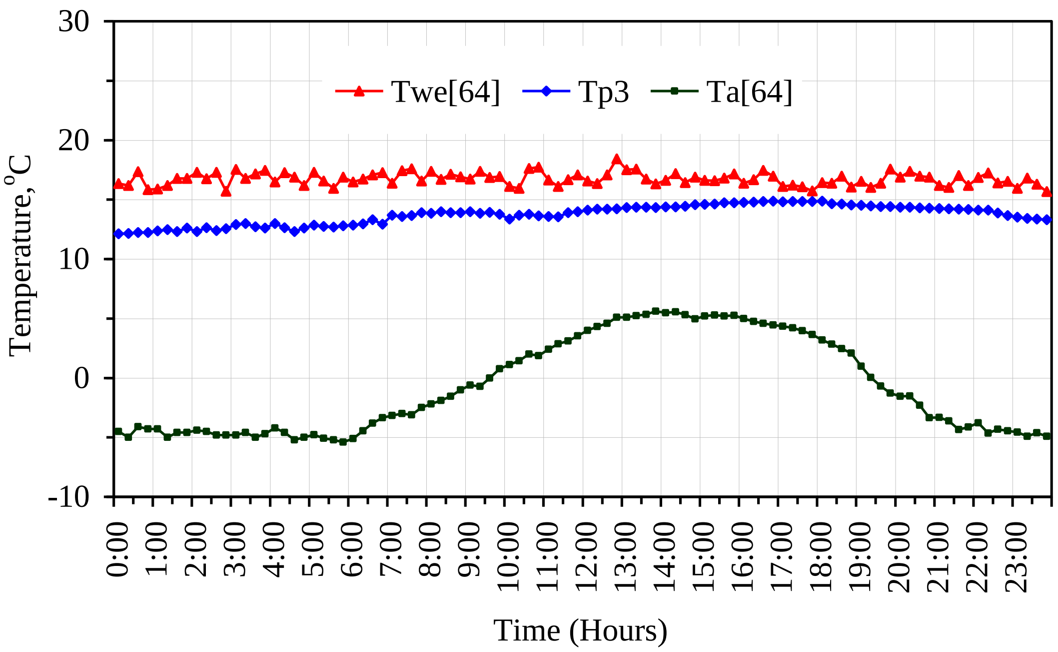

Figure 5.7. Hourly variation of aqueous mixture and ambient air. (March 27, 2012).

Figure 5.8 indicates hourly variation of COP_{HP} and COP_{S} for GCHPS. COP for both system and heat pump are lower during night period of the day. The COP increases in the midday period of time between 14:00 and 16:00 pm hours since ambient air temperature, “Ta” is higher during this time interval. In addition, this augmentation is coincided with the temperature increase of aqueous mixture through the evaporator and condenser reaching evaporation temperature, “Twe” about 2.5°C and condensation temperature, “Tce” of about 46.4°C. While the system performance decreases in the trends during night hours due to the surrounding parameters diminishing and the absence of the solar energy. It is understood from the figure that the higher source or ground temperature gives the higher COP of the heat pump, which results energy savings.

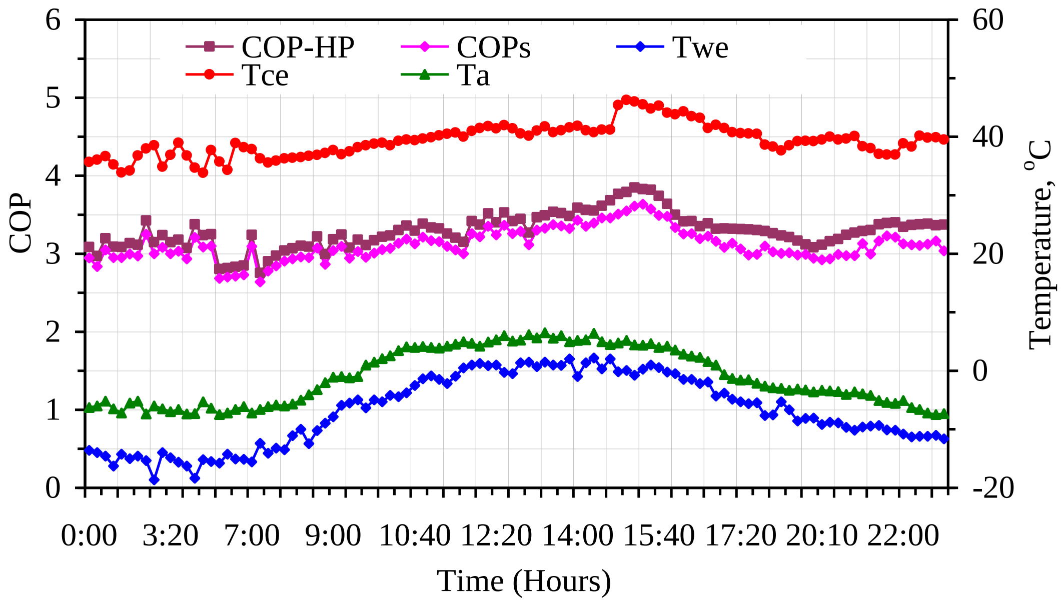

Figure 5.8. Hourly variation of evaporation, ambient air, condensation temperatures, and COP_{HP} and COP_S for the GCHPS (March 27, 2012).

Daily variations of COP for the heat pump and whole system and temperatures for the ground "pipe surface temperature" Tp3 at 3 m depth, aqueous mixture entering and leaving the evaporator heat exchanger are shown in figure 5.9 for the whole test period of heating season. It is understood from the figure that temperatures increase with days, and then the COPs increase as depending on the temperatures. The ground temperature, Tp3 reaches to 16.3°C at the end of the test period. When the heat requirement of the room decreases due to end of the heating season, less power for compressor, pump and fan is need. That condition reveals increment in heat pump and system COP. The data calculated using measurement parameters shows that the COP_{HP} and COP_S reach about 4.3 and 3.4, respectively mostly at the end of March while the aqueous mixture maximum entering temperature to the evaporator, Twi was about 8.8°C, and the maximum aqueous mixture exit temperature from the evaporator, Twe is about 5.6°C.

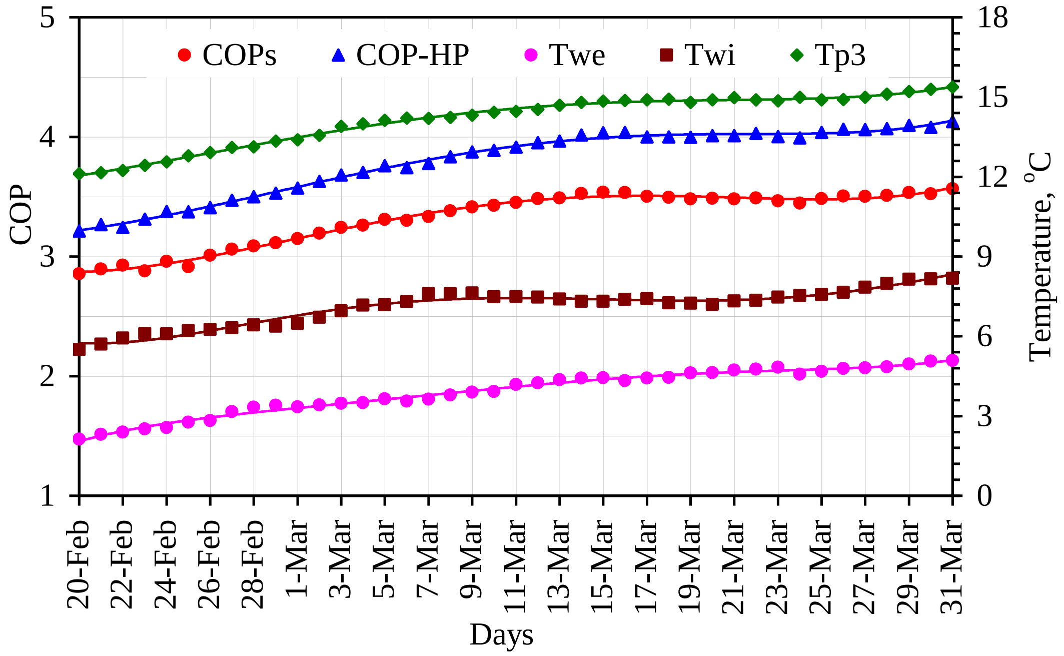

Figure 5.9. COP and temperature variations throughout the 40 test days.

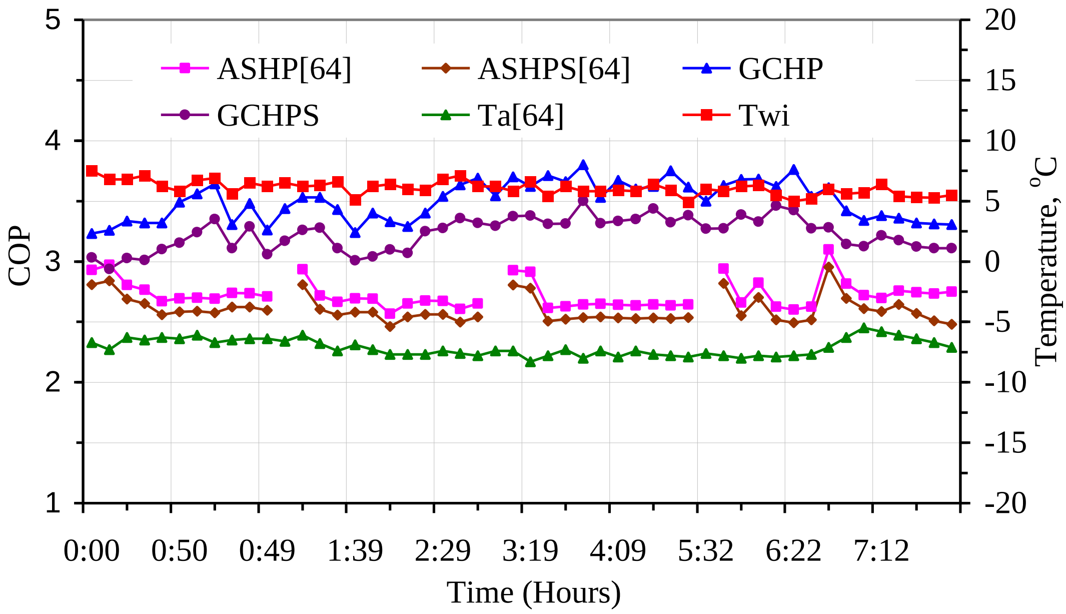

Figure 5.10. Comparison of COP of GCHP and ASHP [64] units.

During March 23, 2012, figure 5.10 is depicted for comparison of the COP for GCHP and ASHP units. As it is seen from the figure, the COP_{HP} and COP_{S} for GCHP are higher than those of the ASHP. The main reason for that is higher evaporation temperature caused by higher source temperature of aqueous mixture than ambient air. Discontinuity in COP lines shows defrost cycle period for ASHP. COP of ASHP is shown higher just after defrost period, however heat pump unit cannot reach its steady operation, and cannot supply enough heat rejection from condenser to the room. Whereas, the other unit operates in a stable manner and the rates heat absorption and extraction are visualizations with the equipped load.

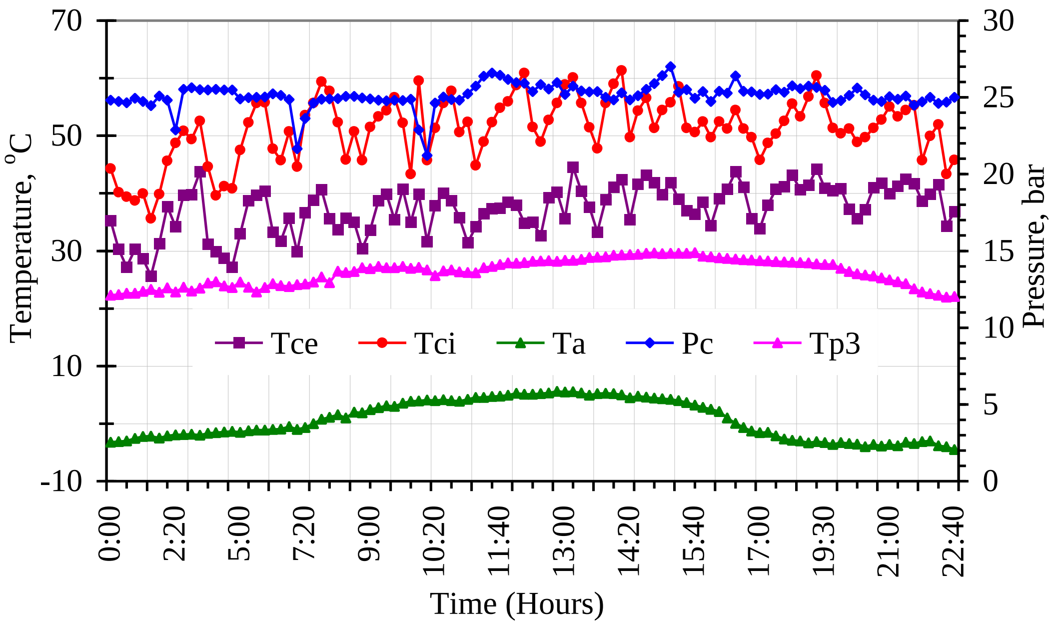

Figure 5.11. Hourly variation of ambient air, ground and refrigerant temperatures (March 22, 2012)

Figure 5.11 illustrates hourly variation of ambient air “Ta”, ground “pipe surface temperature” Tp3 temperatures, and refrigerant temperatures at the inlet “Tci”, and exit “Tce” state of the condenser. It is clearly that ambient air temperature is below zero degrees at night but it is above the zero degree at the daytime. Refrigerant temperatures at condenser inlet are higher than the temperatures at the condenser outlet due to the nature of the heat exchange among the refrigerant and the room heat requirement. The condenser pressure Pc vary around 21-25 bar. The condenser pressure has a direct correlation with the temperature behavior, since the rate of increase occurs on both sides in a proportionate way. The effect of the ground depth and the ambient air are mentioned as well as.

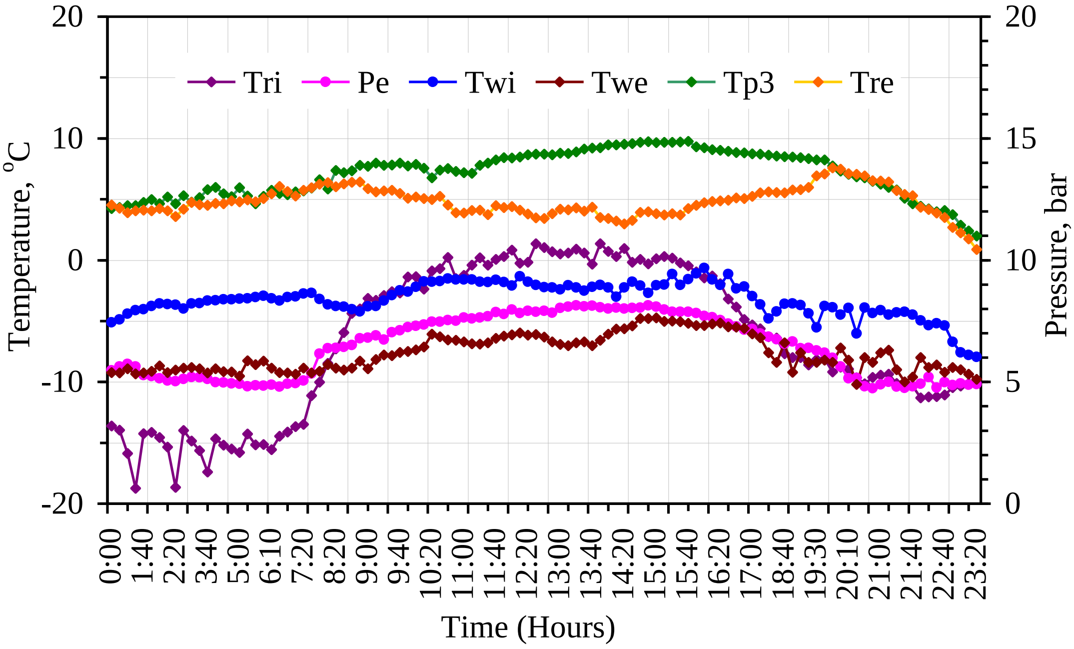

Figure 5.12. Hourly variations of evaporator pressure and temperature with ground and aqueous mixture temperatures (March 25, 2012).

Figure 5.12 shows hourly variation of evaporator pressure "Pe" and evaporator temperature for refrigerant at inlet and exit "Tri", "Tre" with ground "Tp3" and aqueous mixture temperature at evaporator inlet "Twi" and exit "Twe". It is seen from the figure that pressure of the refrigerant increases regularly with together temperature. This means that aqueous mixture supplying temperature increases, and heat transfer to the refrigerant in the evaporator will increase. As a result of the increase in temperature of the refrigerant evaporation, its pressure will increase. The increments in evaporation temperature and pressure have positive effects on COP of the heat pump and system. Therefore, at the midday time when the aqueous mixture temperature difference at the evaporator inlet and outlet increase, it will raise the evaporation pressure. In addition to that, the system performance will increase. "Pe" reaches the maximum pressure of 8.2 bar, and the minimum of 4.8 bar. It is understood from the figure that the higher source temperature gives the higher "Pe". Thus a higher COP of the heat pump is achieved, which results energy savings. Also, it is clear from the figure that refrigerant exit temperature from the evaporator, "Tri" increases with respect to the evaporator aqueous mixture inlet temperature "Twi". This temperature level is very good situation for heat pump evaporators. Since evaporation temperature takes relatively higher value, COP of heat pump and whole system will have higher value.

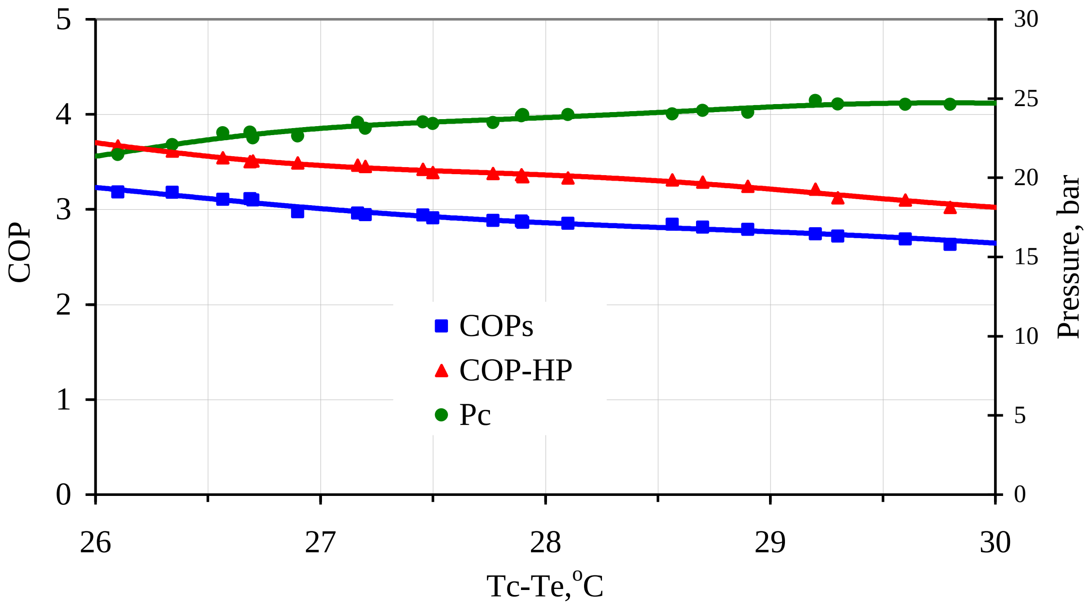

Figure 5.13. COP variations with difference between condensation and evaporation temperature and the condensation pressure. (March 26, 2012).

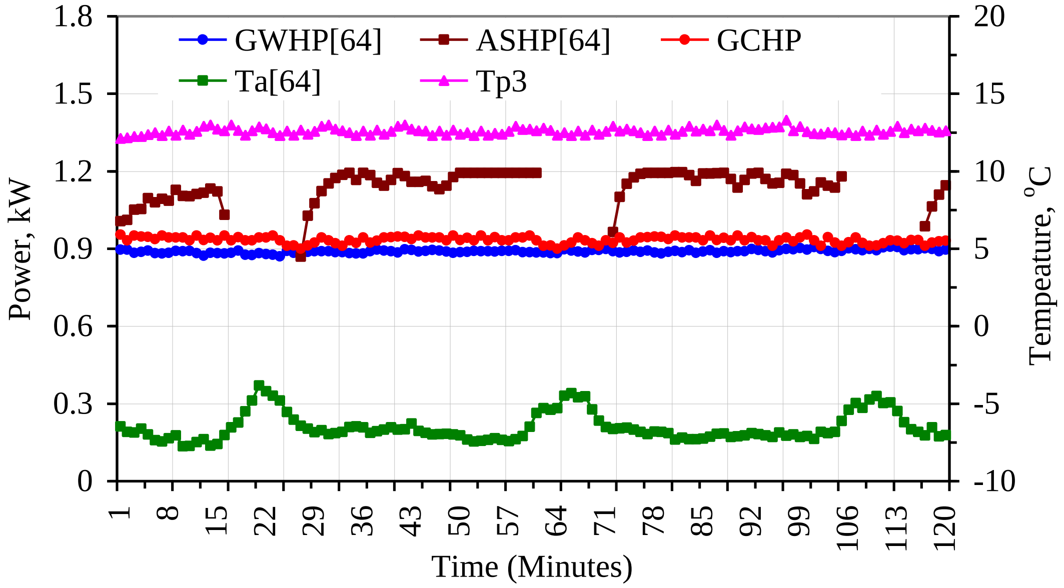

Figure 5.14. Compressor power consumption for GCHP, ASHP [64] and GWHP [64] (February 24, 2012).

Figure 5.13 indicates that the COP for both the heat pump (COP_{HP}) and the system (COP_S) decrease with the increment of the difference between the condensation “Tc” and evaporation “Te” temperatures. In addition, the COP_{HP} and COP_S decrease with increment of condensation pressure “Pc”. Since, the increment of the “Pc” causes in increment in “Tc”. When the “Pc” and “Tc” increase, electric power consumption by the system will increase to meet heat requirement of the room. Thus, the COP will decrease due to the increment in power consumption.

Compressors, which are heart of the heat pump systems, consume great amount of energy in heat pumps and their systems. The power consumed can be different for each type of the heat pump system. For that reason, a comparison for the power consumption of (ASHP), (GWHP) and present GCHP is held. From the point of views, figure 5.14 shows compressor power consumption for the GCHP, ASHP, and GWHP and ground, Tp3 during February 24, 2012 running period. Compressor of the ASHP unit consumes more power than the GWHP and GCHP units compressors, and it also consumes energy during defrost cycle. Powers for the GWHP and GCHP are closer to each other and they operate steadily. However, there is a problem for the ASHP compressor, which has sometimes on state and sometimes off states. Aqueous mixture and ground temperatures are nearly constant. The fluctuation in the compressor power for the ASHP is sourced from fluctuation in ambient air temperature. As it is seen from the figure, there is no fluctuation on ground and antifreeze-water temperatures for the GWHP and GCHP systems, but ambient air temperature has high daily changes. The source temperatures have great effects on the compressor powers and COP of the heat pumps and their systems.

As a result of the experimental study during heating season for the period of February 20 to April 01, 2012. Table 5.1 and 5.2 are given. Table 5.1 indicates climatic conditions taken from Turkish State Meteorological Service located in Gaziantep on the same heating season, and Table 5.2 illustrates averages of measured and calculated parameters for the same heating period. Names of the measured and calculated parameters are given in the first column. Average values of these measured and calculated parameters for the period of February 20 to March 1 and March 1 to April 1, 2012 are presented in second and third columns, respectively.

Table 5.1. Climatic conditions for Gaziantep on heating season for the period of February 20 to April 01, 2012.

Data	February	March
Average maximum outdoor temperature (°C)	6.5	12.2
Average minimum outdoor temperature (°C)	-1.0	3.6
Average outdoor temperature (°C)	3.8	8.4
Average sunshine duration (hours) [65]	2.8	5.1
Average relative humidity [65] (%)	71.3	64.7
Average wind velocity [65] (m/s)	2.3	2.8
Average solar radiation [65] (cal/cm^2 day)	244.31	312.45

Table 5.2 Averages of measured and calculated parameters for the period of February 20 to April 01, 2012.

Item	February	March
Measured parameters		
Condensation pressure (MPa)	2.62	2.82
Condensing temperature (°C)	44.5	47.3
Evaporating pressure (MPa)	0.759	0.832
Evaporating temperature (°C)	-2.0	1.35
Temperature of water-antifreeze solution at GHE inlet (°C)	5.99	8.90
Temperature of water-antifreeze solution at GHE outlet (°C)	4.28	7.27
Soil temperature at depth 0.9 m (°C)	9.09	10.33
Soil temperature at depth 1.5 m (°C)	11.49	12.65
Soil temperature at depth 3.00 m (°C)	14.31	15.93
Outdoor air temperature (°C)	2.8	8.53
Mass flow rate of water-antifreeze solution (l/s)	0.07	0.1
Mass flow rate of air (l/s)	0.31	0.35
Compressor electric current input (Amp)	4.46	4.84
Condenser fan electric current input (Amp)	0.451	0.663
water-antifreeze solution circulating pump electric current input (Amp)	0.462	0.632
Total Current of systems (Amp)	5.373	6.132
Voltage of the system (V)	220	220
Temperature of air at condenser fan inlet (°C)	21.4	21.63
Temperature of air at condenser fan outlet (°C)	28.7	30.7
Power factor	0.92	0.92
Calculated parameters		
Power input the compressor (Watt)	953.984	978.88

Power input to the condenser fan (Watt)	110.1864	134.32
Power input to the water-antifreeze solution circulating pump (Watt)	118.3888	128.06
Total power input to the systems (Watt)	1182.559	1241.2
Heat extraction rate (W/m)	9.119943	12.467
Frequency of the water-antifreeze solution circulating pump (Hz)	36.25	44.0
COP_{HP}	3.331536	3.8147
COP_S	3.054889	3.3259

5.3. Experimental results for the GCHPS for cooling operation

Maximum and minimum monthly ambient air temperatures variations for Gaziantep are shown in figure 5.15. It is clear that the maximum temperature occurs at July to reach about 45°C, and minimum temperature takes place about 12°C low for this month. In winter season, maximum ambient air temperature occurs at January, which is about 19 °C and also minimum temperature is about -17 °C. It is observed that temperature difference between minimum and maximum ambient air temperatures changes about 30-36 °C except December.

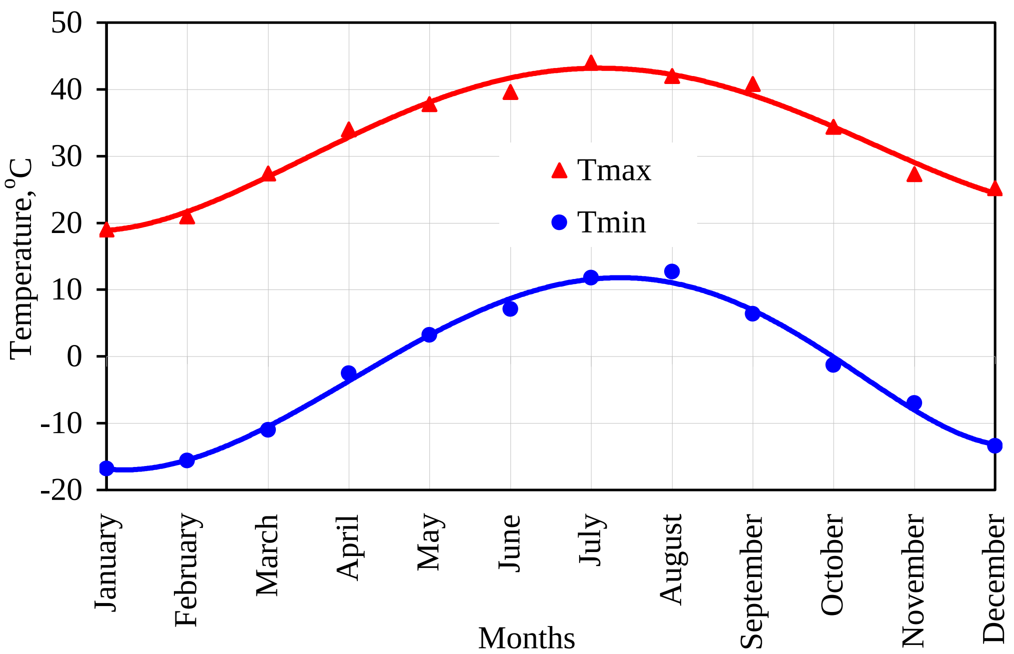

Figure 5.15. Monthly average temperature variations for Gaziantep, Turkey [65]

For a test date of July 14, 2012, figure 5.16 indicates the hourly temperature distribution for the ambient air "Ta", ground "pipe surface temperature" TP at different depths and the antifreeze-water mixture flows through the evaporator at inlet state, "Twi" and exit state, "Twe". As it is seen from the figure, hourly and mean ambient air temperatures are lower than ground temperature for all hours of the day. The air temperature can change sharply, but the ground temperature is more stable and its range of change is moderate. Ground temperatures decrease from the ground surface to the deeper. Ground temperature Tp1 at depth of 0.9m is greater than the temperature, "Tp2" at depth of 1.5m, which are greater than the temperature, "Tp3" at a depth of 3m. Average temperature differences are about 2-4 degrees. The aqueous mixture circulating through the evaporator (in-out) is almost in parallel with each other, in which temperature difference is about 2-3.5 degrees. When the system operates consistently, the outlet and inlet aqueous solution temperatures drastically rise. The daily average temperatures of the ground slowly rise, while the room air temperature decreases slowly at daytime. When the system is stopped at end of the day (i.e. when the compressor, circulating pump and evaporator fan are not on run), the outlet and inlet aqueous temperatures drastically drop while the room air temperature drastically increases.

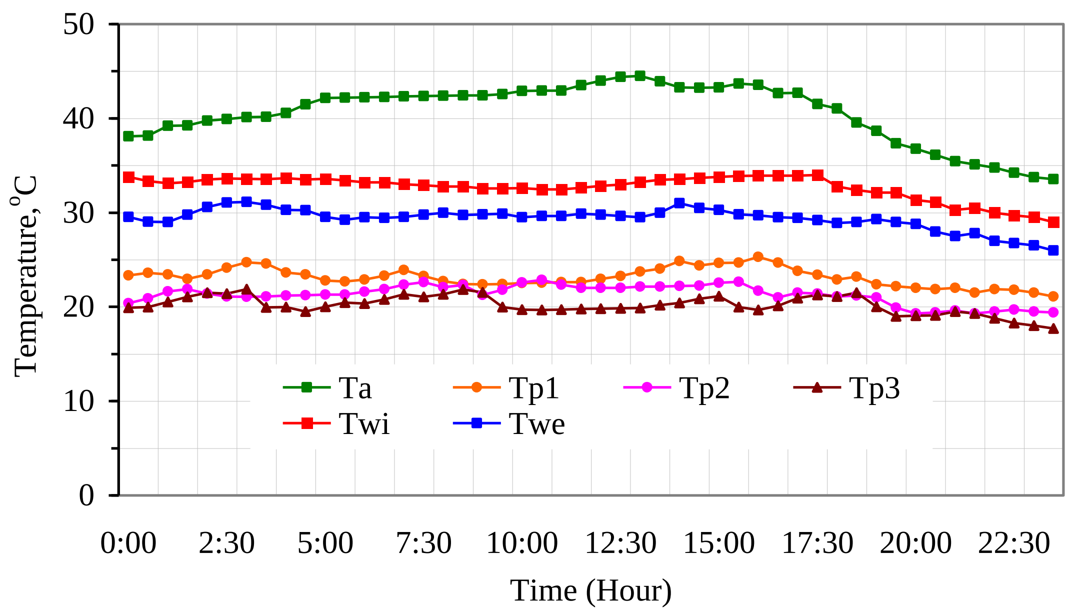

Figure 5.16. Hourly variations of temperature for ambient air, ground, and antifreeze-water mixture in GHE. (July 14, 2012).

Figure 5.17. shows variation of compressor power consumption, and ground, ambient air, antifreeze-water temperatures at evaporator inlet and exit, and difference between evaporation and condensation temperatures. Compressor power changes between 0.9 kW and 1 kW, which operates steadily. Reason of increase in power of the compressor is to meet higher cooling requirements. If there is constant cooling requirement, then constant power will be consumed. In another words, if ambient air temperature increases, then heat gain will increase depending on the degree of increment in the ambient air temperature. At the same time, compressor power will operate until the room inside air temperature is equal to the indoor set temperature, and power consumption of the compressor, fan and pump will increase. As a result, COP of the heat pump and system will decrease.

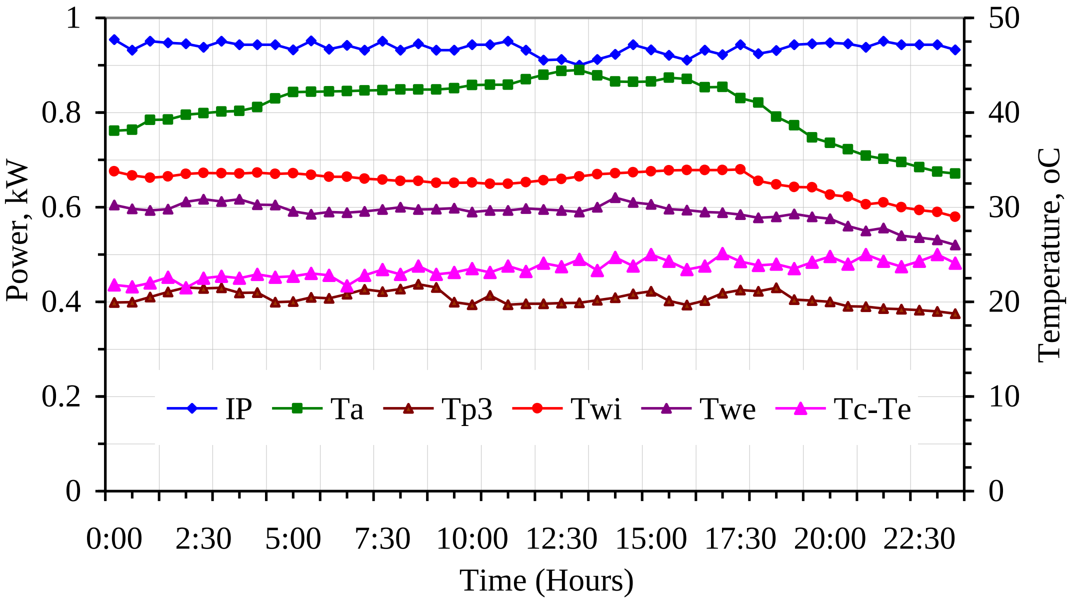

Figure 5.17. Hourly variations of compressor power and ground, ambient air, antifreeze-water, condensation, and evaporation temperatures.

Figure 5.18 represents hourly heat addition to the heat pump system and heat rejected from the system to the ground, and the power consumed by the system during a day of July 16, 2012. While the maximum heat rejection from the condenser to the ground, "Qev" is about 0.7 kW, maximum heat addition to the room, "Qcon" is about 3.5 kW. The power consumption via the compressor increases with the peak load, and reaches about 1.13 kW. While the power consumed by the circulating pump “Wp” is about 0.13 kW, and for the evaporator fan “Wf” is about 0.127 kW. At this time, minimum COP is achieved.

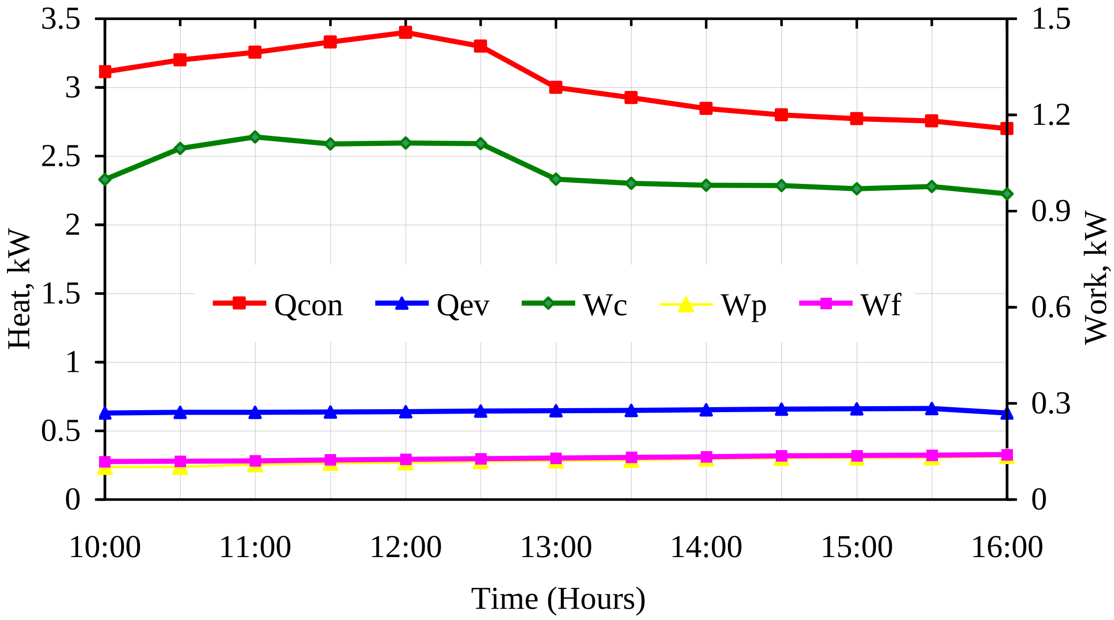

Figure 5.18. Hourly supplied and removed heat with system power consumption.

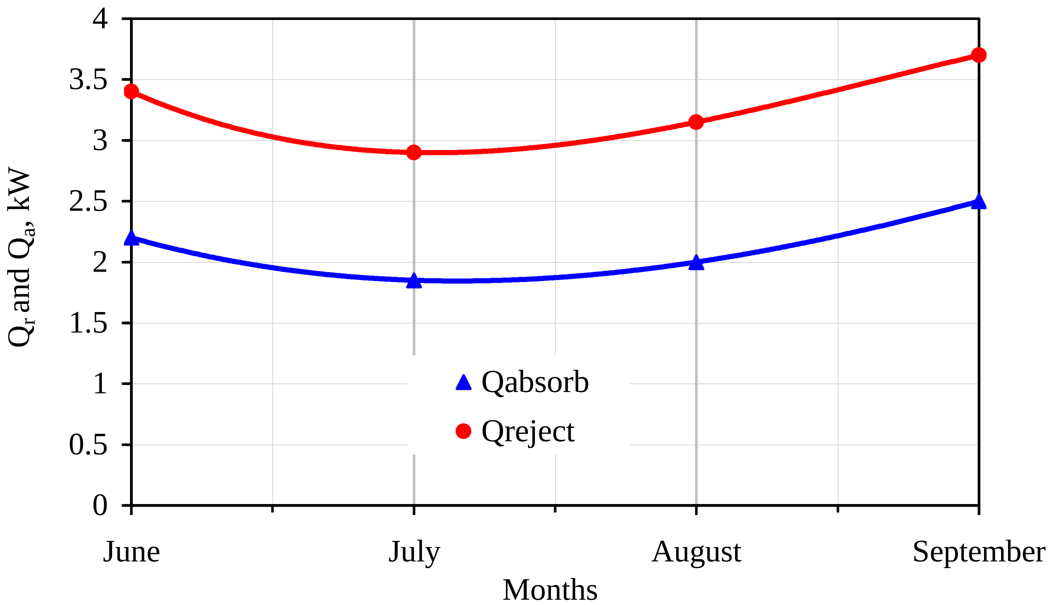

Figure 5.19. Monthly variation of heat absorption from the room and rejection to the ground.

Figure 5.19 represents the monthly variation of the heat absorption from the room "Qa" and heat rejection to the ground "Qr". It is obvious that the higher heat absorption from the room and heat rejection from the condenser to the ground occurs at June and the beginning of September 3.4, 2.2 and 3.8, 2.5 kW, respectively. The lower values of the heat absorption and rejection occur at July and August 3 kW and 1.9 kW, respectively.

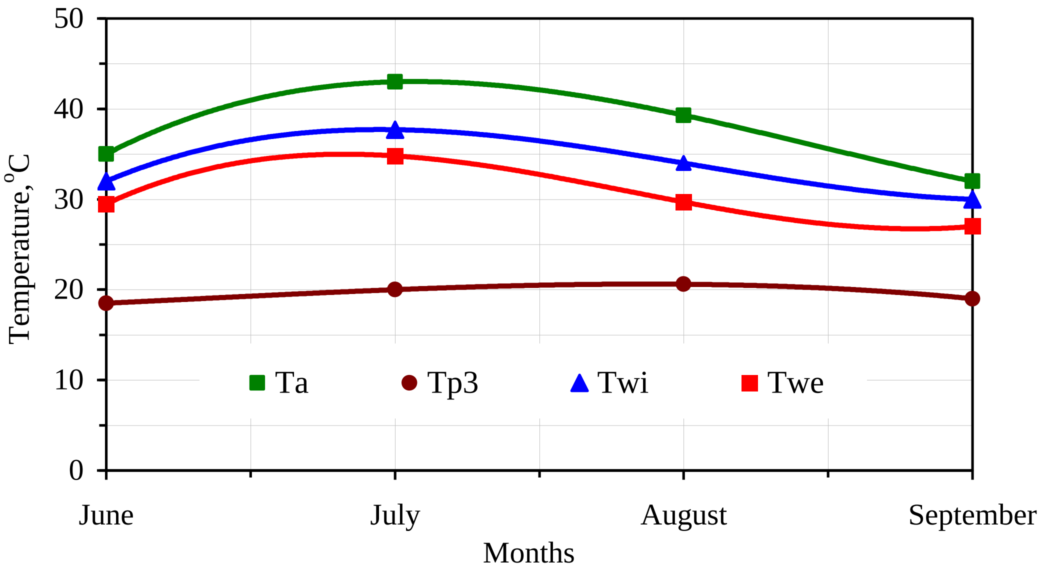

Figure 5.20. Monthly variation of average temperatures at the cooling season.

Figure 5.20 it can be noted that the monthly maximum value of evaporator aqueous solution temperature "Twi" over the four months period is nearly 35^{o}C, the average being about 32.67^{o}C while the minimum value of "Twe" is approximately equal to 29.4^{o}C. The mean values of outdoor air "Ta" and the ground "Tp3" temperatures are found as 43 and 21 ^{o}C, respectively. The difference between the average values of "Twi" and "Twe" is found to be approximately about 2.35^{o}C.

Figure 5.21 gives variation of hourly condensation temperature and pressure of the refrigerant in the ASHP and GCHP during the same time period (July 12, 2012). Tendency of the pressure "Pc" curves for the both heat pumps is in accordance with tendency of temperature "Tc" curves. The "Pc" and "Tc" of the ASHPS is bigger than those of the GCHPS. The data collected was taken during maximum cooling requirement period. Due to the higher temperatures and pressures on the ASHPS, this will lead to more energy consumption especially when the defrost cycle takes place, and ultimately decrease the COP. However, the condensation pressure and temperatures for the GCHPS is lower than those of the ASHP. As a result of the GCHP cycle occurs at lower pressure and temperature, the system need less compressor work. And also COP for the GCHP will be higher and the situation lead to energy saving.

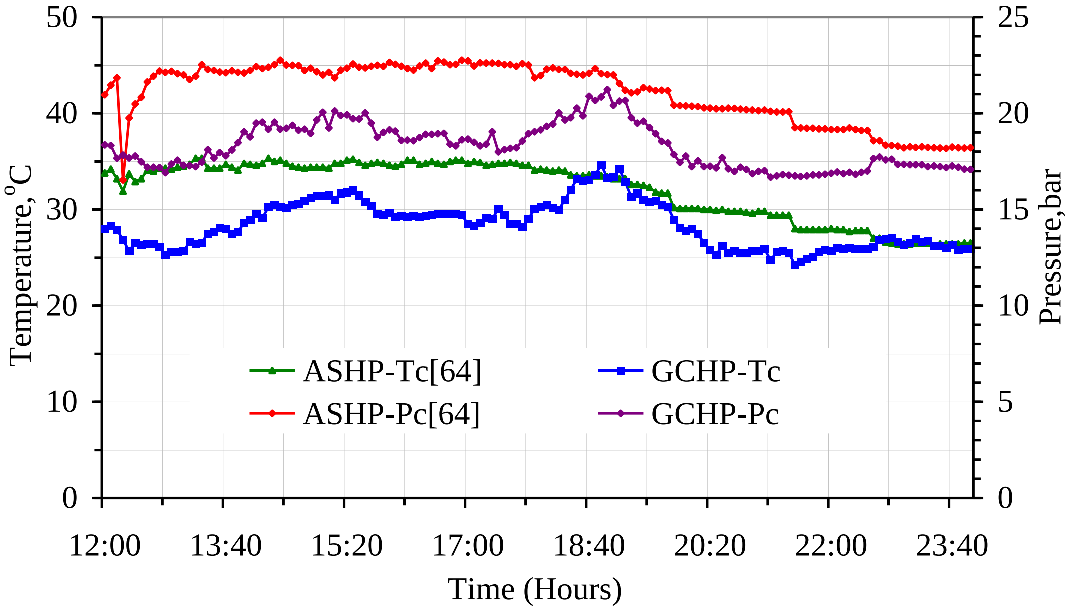

Figure 5.21. Hourly variations of condensation temperature and pressure for the ASHP [64] and GCHP during the same time period (July 12, 2012).

Figure 5.22 shows hourly variation of condensation pressure for both ASHP and GCHP, and temperatures of ground “Tp3”, ambient air “Ta”, and aqueous mixture “Twi” entering the evaporator during July 14, 2012. Condensation pressure tendency agrees to source temperatures tendency. Condensation pressure for the ASHP occurs at higher pressure during daylight hours because of higher ambient air temperatures. It is understood from the figure that cooling requirement of the rooms decrease during night hours. At the early morning hours, it decreases sharply in both condensation pressures, which shows that there is no cooling requirements for the testing room. Condensation pressure for the GCHP also well agrees to temperature tendency of water circulated through the ground loop heat exchanger. Increase in ground temperature leads to increase in condensation pressure. During the night hours, since cooling requirement is low, GCHP compressor consumes less power and thus, heat rejection from the refrigerant to aqueous mixture in the heat exchanger is relatively lower.

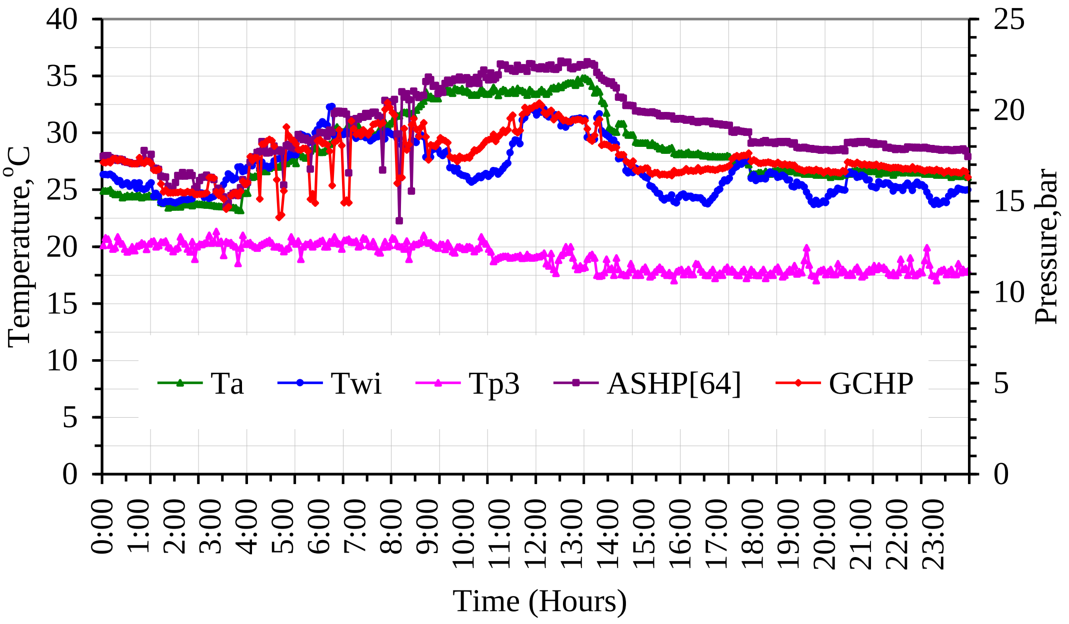

Figure 5.22. Hourly variation of condensation pressures, ground, ambient, and water mixture temperatures for the ASHPS [64] and GCHPS.

Figure 5.23 illustrates the hourly variation of condensation temperature and work consumption of the compressor for both GCHP and ASHP systems. For the ASHP cycle a reduction in condensation temperature after about 19:00 pm caused by reduction in ambient air temperature. Although, ambient air temperature decreases, compressor consumes nearly same power with previous hours. The main reasons of this, indoor cooling load requirement exist due to thermal inertia of the room. For the GCHP cycle the condensation temperature is higher when compressor work consumption is higher. Sharply decrease in condensation temperature occurs about the same time mentioned above will follows decrease in compressor work consumption. Compressor decreases work consumption when indoor air temperature closes the set temperature. Thus, if a comparison held between the two systems from the efficiency point of view and according to the data-inspired from the figure, GSHP has higher COP than air source heat pump during day light while ASHP higher COP during a part of night hours. In fact, ASHP's COP becomes higher when ambient temperature reduces below of ground supply water temperature.

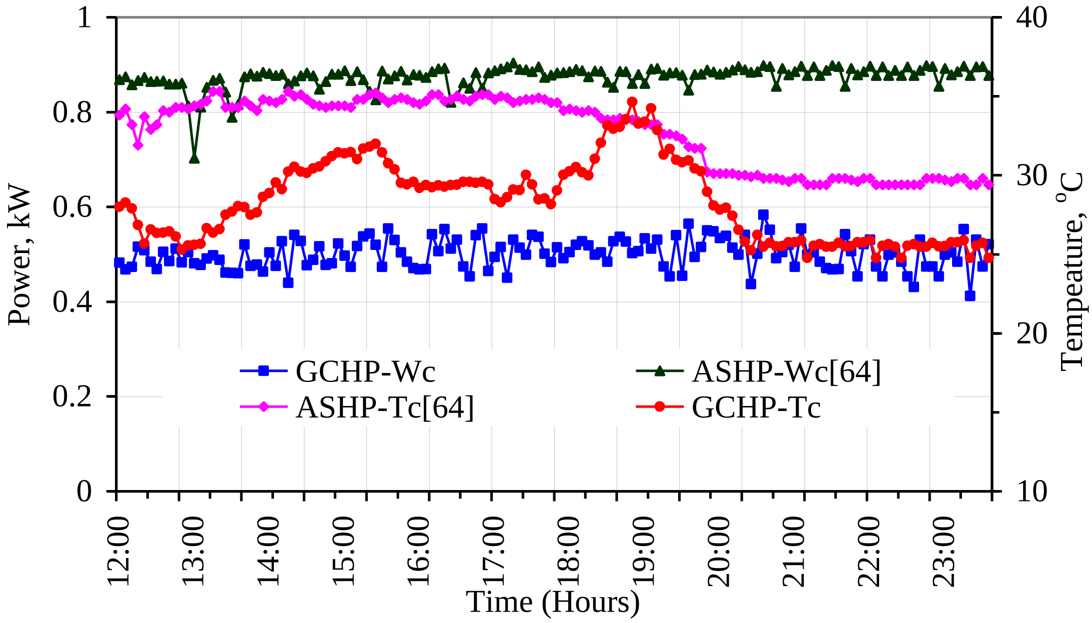

Figure 5.23. Hourly variation of condensation temperature and compressor work for ASHP [64] and GCHP units (July 15, 2012).

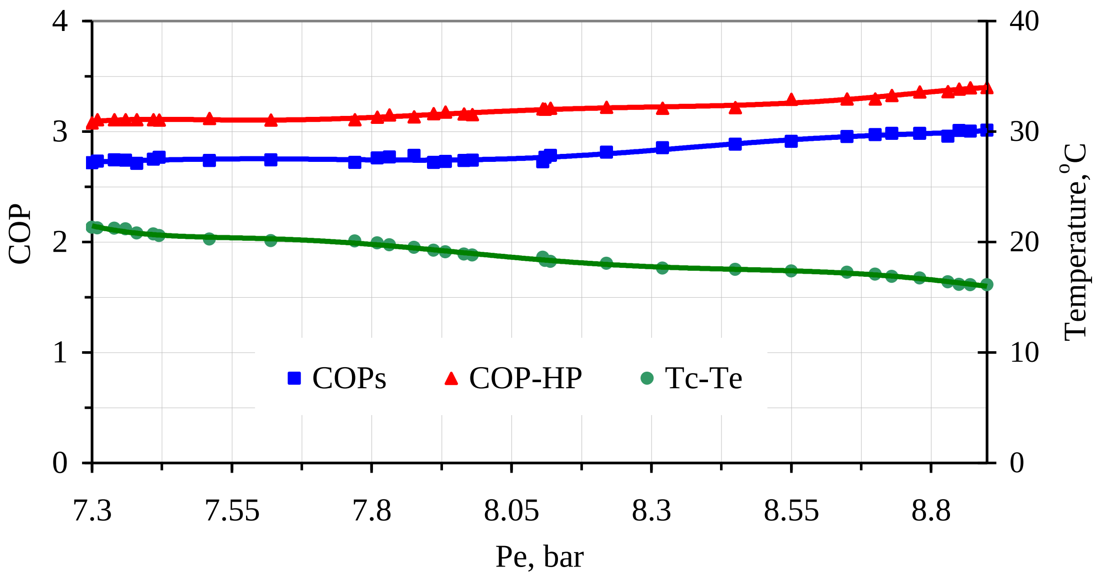

Figure 5.24. COP variations with temperature difference between condenser and evaporator with evaporator pressure.(July 22, 2012).

Figure 5.24 shows the COP for both the heat pump and the whole system with respect to the evaporation pressure "Pe". Moreover, the COP_{HP} and COP_{S} increase with the reduction in temperature difference between the condensation "Tc" and evaporation "Te" temperatures. The system performance increases with the evaporation pressure. As a result, the temperature difference "Tc-Te" decreases with the increments of the "Pe" and COP of the heat pump and system.

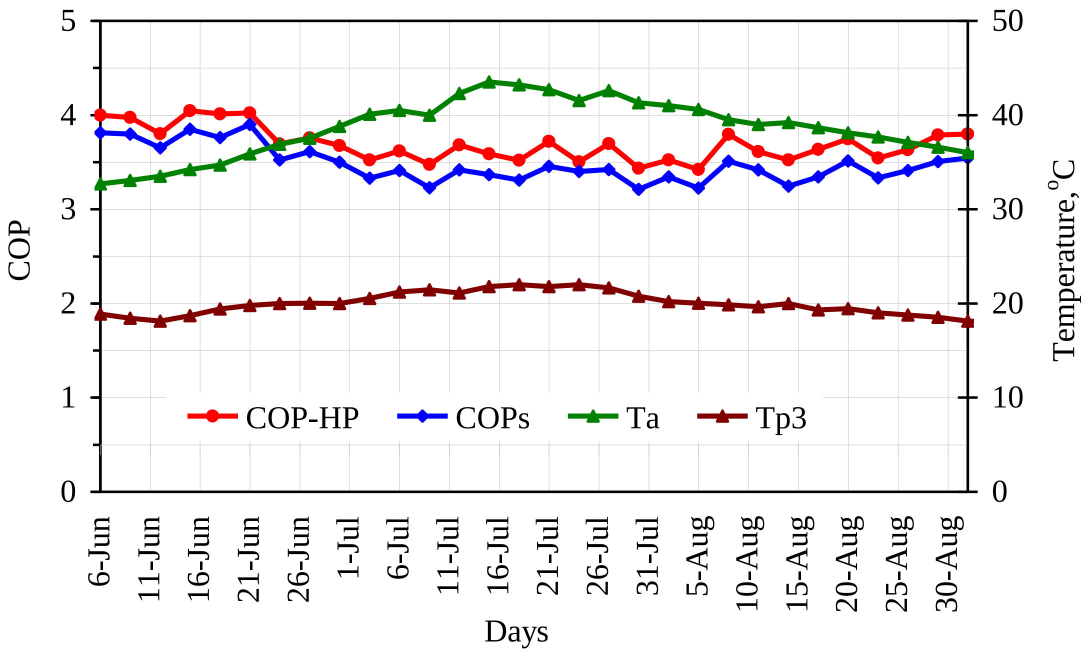

Figure 5.25. Monthly variations COPs for the heat pump and system, ground and ambient air temperatures.

Figure 5.25 shows the monthly variation of COP_{HP} and COP_{S} values ranging from 3 to 3.3. The lowest COP values are obtained in July and August while the highest COP values are found at June and September. The ambient air temperature "Ta" also reaches the highest value at the hottest months about 43°C, and falls progressively on the coldest moths about 33°C, which are seen from the figure, clearly. The temperature "Tp3" is the pipe surface temperature at depth of 3.0 m, and reaches its maximum value about 21.5 °C, at July and August.

Figure 5.26 shows variation of electrical work consumption via the compressor of GCHP and corresponding COP of the heat pump cycle. While the work consumption by the compressor is diminishing, this process will be associated with dwindling the condensation temperature. As a result, the system performance will be enhanced, and the COP of the heat pump will be higher. Compressor work consumption decreases when indoor air temperature closes to the set temperature. COPs of the system and heat pump are lower when the higher work consumption is required.

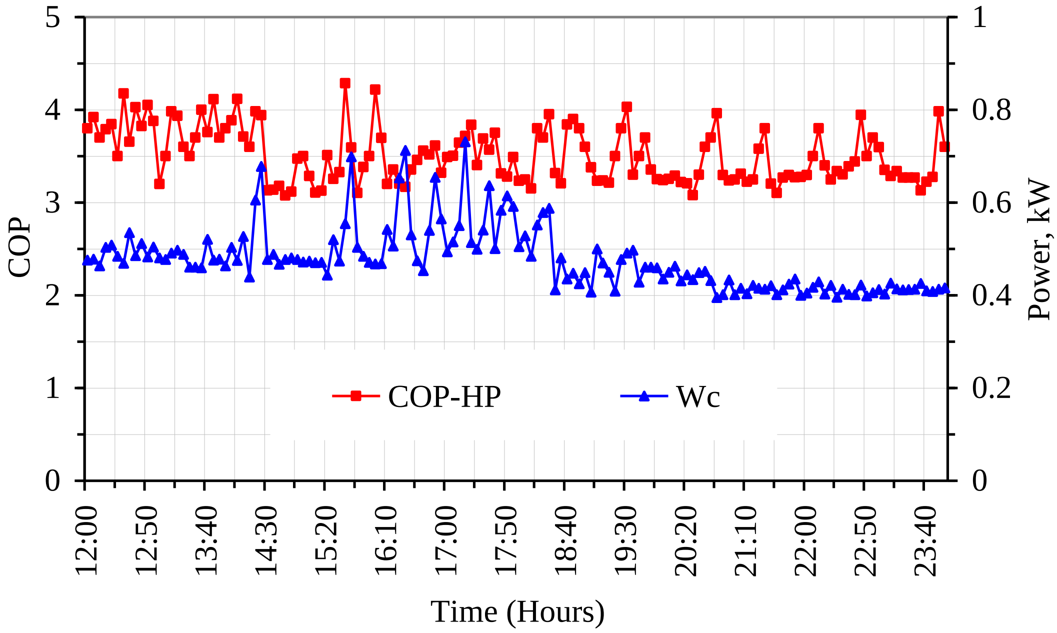

Figure 5.26. Hourly variation of COP_{HP} and compressor work for the GCHP.

Figure 5.27 indicates monthly variation of COP and Energy Efficiency Ratio (EER) for the GCHP and GCHPS. COPs values decrease until July, and then it increases. Monthly variation of COP_{HP} and EER are similar to those of heat supplied and removed. The monthly mean value of COP_S is obtained as 3.0. The highest COP for the system is found to be 3.3 in June and September while the lowest COP for the system is found as 3.0 in July. EER values are ranging from 8 to 10. By comparison, based on the values of the US Department of Energy (USDOE), commercially available systems have COP_S values of 3.52-4.92 [66]. So, the performance of the heat pump system is low due to a poor design of the system when compared to the values given above. The COPS increases with increasing the buried depth of GHE. Thus, the much more proper GHE design gives higher enhancement rates for COP_S.

The cooling equipment systems used in residential and small commercial buildings often express cooling system efficiency in terms of the energy efficiency Ratio (EER) and/or seasonal energy efficiency ratio (SEER) [67]. These coefficients are defined by the cooling effect in Btu (not in tons) divided by the power use in watts (not in kW) for the peak day (EER), or the seasonal average day (SEER) can be derived from the following equation.

Equation: $EER = 3.412\ COP_{HP}$ (5.1)

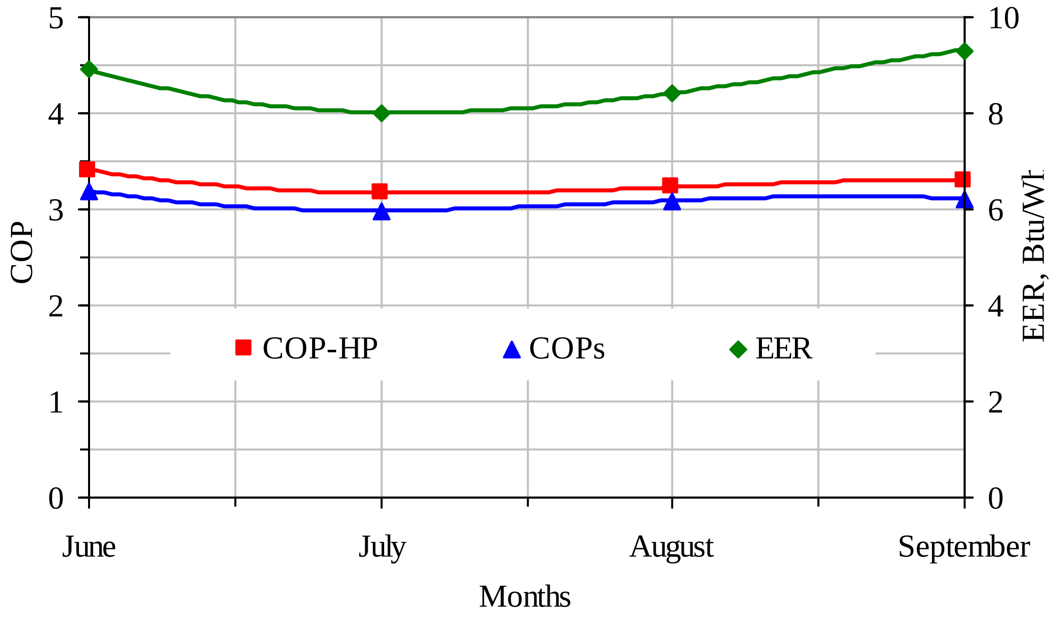

Figure 5.27. Monthly variation of the COP_S, COP_{HP} and EER.

The experimental study was performed during cooling season for the period of June 3 to September 01, 2012. During this study, measurements were taken under the climatic conditions which are given in Table 5.3 and 5.4. Table 5.3 indicates climatic conditions taken from Turkish State Meteorological Service located in Gaziantep on the same cooling season, and Table 5.4 illustrates averages of measured and calculated parameters for the same cooling period. Names of the measured and calculated parameters are given in the first column. Average values of these measured and calculated parameters for the period of June 3 to September 01, 2012, are presented in second and third columns, respectively.

Table 5.3. Climatic conditions for Gaziantep on cooling season

Item	June	July	August
Average max. outdoor temperature, °C	24.1	27.9	27.5
Average min. outdoor temperature, °C	31.4	36	35.5
Average outdoor temperature, °C	17.1	21.1	21.0
Average sunshine duration (hours) [65]	10.3	10.5	10.1
Average relative humidity [65] %	2.2	0.7	0.5
Average wind velocity [65] m/s	6.7	2.7	2.7
Average solar radiation [65] cal/cm^2 day	336.2	397.3	387.1

Table 5.4. Averages of measured and calculated parameters for the cooling period.

Item	June	July	August
Measured parameters			
Condensation pressure (MPa)	2.66	2.769	2.73
Condensing temperature (°C)	42.5	45.2	44.6
Evaporating pressure (MPa)	0.77	0.815	0.781
Evaporating temperature (°C)	-5.4	-6.2	-6.0
Temperature of water-antifreeze solution at GHE inlet (°C)	30.99	33.84	32.85
Temperature of water-antifreeze solution at GHE outlet (°C)	32.28	35.2	34.18
Soil temperature at depth 0.9 m (°C)	23.09	25.18	24.56
Soil temperature at depth 1.5 m (°C)	22.39	25.22	24.96
Soil temperature at depth 3.00 m (°C)	20.31	21.15	20.8
Outdoor air temperature (°C)	24.1	27.9	27.5
Mass flow rate of air (l/s)	0.32	0.33	0.325
Compressor electric current input (Amp)	4.26	4.55	4.52
Condenser fan electric current input (Amp)	0.431	0.55	0.521
water-antifreeze solution circulating pump electric current input (Amp)	0.422	0.561	0.525
Total Current of systems (Amp)	5.583	5.98	5.766
Voltage of the system (V)	220	220	220

Temperature of air at condenser fan inlet (°C)	21.5	21.6	21.2
Temperature of air at condenser fan outlet (°C)	28.7	29	28.2
Power factor	0.92	0.92	0.92
Calculated parameters			
Power input the compressor (Watt)	850.984	930.44	914.848
Power input to the condenser fan (Watt)	90.1864	110.02	105.4504
Power input to the water-antifreeze solution circulating pump (Watt)	93.3888	115.2104	109.26
Total power input to the systems (Watt)	1259.5592	1321.6704	1326.5584
Heat extraction rate (W/m)	12.5	12	11.5
Frequency of the water-antifreeze solution circulating pump (Hz)	34.25	36.20	35.90
COP_{HP}	4.015	3.219	3.1803
COP_S	3.848	3.032	3.0015

5.4. Section conclusion

This section focuses on the experimentally measured data for the GCHPS that were recorded during the cooling and heating season's operations. The experimentally measured parameters for both heat pump cycle "R410A" and the GHE cycle "water-antifreeze". The gathered data were collected via the measuring sensors located on each separate device of the GCHPS. Data were transferred and saved to PC. In addition to that, data were analyzed and calculated then the results were depicted into figures and tables in order to get the full image on the nature of the work of the system. Moreover, a comparison with ASHP and GWHP systems under the same or near to the climate condition were done on some figures to illustrate the difference between the systems and to understand the advantages and disadvantages of these systems, and to clarify the match of these systems with the sustainability change of the surrounding items.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

The ground “mother earth” acts as a very large store of heat energy. It can be used as a heat source in winter or a heat sink in summer. The ground can be used to moderate the temperature in buildings standing on it. A Ground Coupled Heat Pump (GCHP) can be used to extract heat energy from the ground in winter season to transfer the heat into buildings. At the same time, it can be used to provide a very efficient mechanism for heat rejection from buildings down into the ground in summer season. A GCHP provides a clean way to heat buildings, free of all carbon emissions on site. It can make use of solar energy stored in the ground to provide one of the most energy-efficient ways of heating or cooling buildings. Since, most heat pumps unit having reverse cycle can be used for both heating and cooling of the buildings. The GCHP systems are environmental friendly, economic, space saving, safely, reliably, less maintenance, long cycle life, and effective devices.

In order to examine advantages of the heat pump system experimentally, In this thesis, an experimental system to determine performance of the GCHPS was installed in scope of the thesis investigation on performances of the GCHPS used slinky type pipe as ground heat exchanger (GHE) for space heating and cooling is performed. All nodal temperatures, pressures, mass flow rates voltages and currents were automatically measured and collected by a personnel computer during heating season (February 20-April 1, 2012) and cooling season (June 3-September 1, 2012). Performance parameters for the GCHPS were computed using the measurement values. Results obtained from the experimental study were depicted as figures and discussed in Chapter 5. For measurement days in heating and cooling season, the most important results and some recommendations are presented in this chapter.

1. During heating and cooling days, average values of COP_{HP} and COP_S for the heat pump and system are obtained as 3.8, 3.3 and 3.2, 3.0 for the both seasons, respectively.

2. Mean values of ground at 3.0 m depth and ambient air temperature for both heating and cooling season were experimentally obtained as nearly 15, 5.6 and 21.5, 43 ^{o}C, respectively. Also, inlet and outlet aqueous solution temperatures for both seasons for evaporator were calculated as nearly 8.90, 7.3 and 33.84 35.2 ^{o}C, respectively.

3. While mean evaporation pressures and temperatures for the heating and cooling seasons were calculated as 7.59 bar, -2.0 ^{o}C and 8.15 bar, -6.2 ^{o}C, condensation pressures and temperatures were obtained as 28.2 bar, 47.3 ^{o}C and 26.6 bar, 42.5 ^{o}C for both seasons, respectively.

4. Both, average cooling and heating COP of GSHPS is higher than ASHP.

5. For the heating mode, when the inside temperature of the testing room reset to a new ambit between 21-24 ^{o}C. The circulating pump and compressor continue running within the range. When in the inside room temperature override 24^{o}C, just the indoor unit fan still in work, as a result the inside room temperature progressively decline. Until it reaches under 21^{o}C, the circulating pump and the compressor restart up again. When the system in run, the inlet and outlet water-antifreeze solation temperature significantly drop. Moreover, water-antifreeze solation temperatures boost when the circulating pump and the compressor are not in action.

6. Since ground temperature is stable and usually higher during heating season and lower during cooling season than ambient air temperature, ground is better heat source or heat sink for heat pumps.

7. In the GCHPS, except for the buried heat exchanger, the entire system can be located indoors. This reduces the wear and tear of the system experiences, which gives long life.

8. The main advantage of the GCHPS over the other conventional systems is that the heat extract or reject is maintain with in the circulating fluid through the ground cycle, not release it to the ambient and pollutants the environment.

9. Experimental results show that ground temperature near around the pipe (GHE) increases in cooling season and decrease in heating season as a result of heat transfer with intermediate fluid. Increase in temperature during cooling season is not desired condition since increase in source temperature causes lower heat pump or system COP. This situation is also valid for heating season. Since, the lower source temperature gives the lower heat pump or system COP.

10. Mass flow rate of the water through the GHE has effect on evaporation temperature. Especially, if parallel flow heat exchangers are used, temperature difference of water at inlet and outlet of the evaporator should be kept relatively lower. This means higher mass flow rate of the water. But, since pumping power has one of the important effects on the system COP, optimum mass flow rate of the water has to be predetermined, or a circulating pump with variable speed should be used.

11. Freezing of water in evaporator during heating season is one of the parameters to be avoided. Freezing of water can be prevented by adding antifreeze solution to in it. But, amount of antifreeze solution is important by heat extraction and system efficiency point of view. Thus, optimum amount of antifreeze has to be used.

12. Heating or cooling load of buildings is not constant through heating and cooling operation period. For this reason, heat pump units have to be varying capacity units. This is important by energy consumption point of view, since compressors consume higher energy at starts. Power consumption of the compressor has to be automatically controlled depend on the load so that energy saving is obtained whether COP of the system remain constant.

13. The parts of the GCHPS should be checked in terms of energy efficiency. Thus, it is necessary to conduct a pre-design analysis to determine optimal

system parameters that will ensure minimum energy consumption and favourable costs.

14. For a clearer perception of the system performance, in terms of calculating heat pump performance we do not just consider COP but also Seasonal Performance Factor (SPF). This will take into account energy for circulation, variable loads and source temperatures over time and it is a useful method of comparing the performance of a heat pump with more conventional heating systems.

15. Soil properties are important factors should be taken into consideration in the design and construction of a GHE for a heat pump application to ensure long GHE life and reduce the installation costs and maintenance and thus, ensure long life and longer life of the system. The properties should be tested by excavation of the earth at different depth before construction of the GCHPS.

16. Economic analysis has to be made before deciding type of heat pump by considering all environmental effect. GCHP units have higher installation cost but lower operation cost because of their higher COP. Payback period should be estimated before selecting unit.

17. The financial importance of good design should be emphasized in terms of keeping capital costs associated with the heat pump and borehole to a minimum.

REFERENCES

[1] ASHRAE, 1995. *ASHRAE Handbook*, *HVAC Applications*. American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc. Atlanta, GA.

[2] Grant, M.A., I.G. Donaldson, and P.F. Bixley, 1982. *Geothermal Reservoir Engineering*. Academic Press.

[3] Hart, D.P. and R. Couvillion, 1986. *Earth-Coupled Heat Transfer*. National Water Well Association, Dublin, OH.

[4] Hellstrom, G. 1991. Ground Heat Storage. *Thermal Analyses of Duct Storage Systems.* Lund, Sweden: University of Lund, Department of Mathematical Physics.

[5] Geo-Heat Centre. Bulletin: *Geothermal (Ground Source) Heat Pumps*: A World Overview. Oregon Institute of Technology 2004.

[6] ASHRAE, 1997. *ASHRAE Handbook*, *Fundamentals*. American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., Atlanta, GA.

[7] IEA Heat Pump Centre. About Heat Pumps. IEA Heat Pump Centre Website 2009 [cited 2010 May 24]; Available from: URL: www.heatpumpcentre.org.

[8] Kavanaugh S.P., Rafferty K. (1997). Ground-source heat pumps: design of geothermal systems for commercial and institutional buildings. *ASHRAE Transactions*, 167.

[9] Department of Trade and Industry. The Energy Challenge: Energy Review. Norwich: The Stationary Office; 2006.

[10] Langley BC. Heat Pump Technology. Virginia: Reston Publishing Company, 1983.

[11] Sumner, J.A. A summary of heat pump development and use in Great Britain. *J Inst. of fuel,* pp 318-321, Jan 1953.

[12] Von Cube, H.L. and Steimle, F. Warmepumpen. Grundlagen and Praxis. VDI- Verlag GmbH, Dusseldorf, 1978.

[13] Kell, J.R. and Martin, P.L. the Nuffied College heat pump *J.Inst.Heating & Ventil. Engnrs.* , pp 333-356, Jan. 1963.

[14] Crandall, A. C., (1946). House heating with earth heat pump, *Electrical* World, **126**, 94–95.

[15] Ingersoll and Plass, (1948). Theory of Ground Pipe Heat Source for the Heat Pump. *Heating, Piping, and Air Conditioning*, **20**, 119–122.

[16] Hooper, F. C. (1952). An Experimental Residential Heat Pump. *Canadian Journal of Technology*, **30**, 180–197.

[17] Kidder A. H. and Neher J. H. (1952). Residential Heat Pump Experiments in Philadelphia. Earth as a Heat Source. *AIEE Transactions, Part II, Application and Industry,* **71**, 343–350.

[18] Heat Pump Association. Heat Pump Applications. Heat Pump Association Website 2010 [cited 2011 May 24]; Available from: URL: http://www.heatpumps.org.uk/HeatPumpApplications.htm.

[19] Sanner B., Karytsas, C., Mendrinos, D. and Rybach, L.(2003), "Current status of ground source heat pumps and underground thermal energy storage in Europe", *Geothermics*, Vol. **32**, pp.579–588.

[20] Christian Boissavy, AFPG. Geothermal District Heating Market in Europe. Market Report @ GeoPower Europe 2011. Available from: URL: http://www.Geothermal.org.uk/GeothermalApplications.htm.

[21] ASHRAE. 1997. (Kavanaugh, S.P. and K. Rafferty, Authors) Ground Source Heat Pumps-Design of Geothermal Systems for Commercial and Institutional Building.

[22] Austin, W.A. III, C. Yavuzturk, and J. D. Spitler. 2000. Development of an In-Situ System for Measuring Ground Thermal Properties. Submitted for publication to ASHRAE Transactions.

[23] Chiasson, A. (1999). Advances in Modeling of Ground-Source Heat Pump Systems. *Master's thesis*. *Oklahoma State University. Stillwater, Oklahoma.*

[24] A. Hepbaşlı, Ozay Akdemir, Ebru Hancıoglu (2003). Experimental study of a closed loop vertical ground source heat pump system. *Energy Conversion and Management,* **44**, 527–548.

[25] Omer Ozyurt, Dundar Arif Ekinci. (2011). Experimental study of vertical ground-source heat pump performance evaluation for cold climate in Turkey. *Applied Energy*, 1257–1265.

[26] Ozgener O. and Hepbaşlı A (2007). A review on the energy and exergy analysis of solar assisted heat pump systems. *Renewable and Sustainable Energy Reviews* **11**, 482-496.

[27] Mustafa Inallı, Hikmet Esen (2004). Experimental thermal performance evaluation of a horizontal ground-source heat pump system. *Applied Thermal Engineering,* **24**, 2219–2232.

[28] Babur N. (1986). Design and construction of an earth source heat pump. *MSc thesis in Mechanical Engineering, Middle East Technical University, Ankara, Turkey, 119.*

[29] Huseyin Benli, Aydın Durmus (2009). Evaluation of ground-source heat pump combined latent heat storage system performance in greenhouse heating. *Energy and Buildings,* **41**, 220–228.

[30] P.D. Metz, Ground Coupled Heat Pump System Experimental Results. Brookhaven National Laboratory, Upton, 1983, Report No. BNL-33540.

[31] J.E. Bose, M.D. Smith, Performance of new ground heat exchanger configurations for heat pump, *Solar Engineering 1* (1992).

[32] Yupeng Wu, Guohui Gan , Anne Verhoef , Pier Luigi Vidale , Raquel Garcia Gonzalez (2010). Experimental measurement and numerical simulation of horizontal-coupled slinky ground source heat exchangers. *Applied Thermal Engineering,* **30**, 2574–2583.

[33] P.S. Doherty, S. Al-Huthaili, S.B. Riffat, N. Abodahab,(2004). Ground source heat pump description and preliminary results of the Eco House system, *Applied Thermal Engineering,* **24**, 2627–2641.

[34] Penrod (1954). Sizing Earth Heat Pumps. *Refrigerating Engineering,* **62**, 57-61.

[35] Vestal, D. M. and Fluker B. J. (1956). Earth as heat source and sink for heat pumps. *Heating, Piping and Air Conditioning,* **28**, 117–123.

[36] Griffith M. V. (1952). Heat transfer from soil to buried pipes, Tech. Report Y/Tn18. British Electrical and Allied Industries Research Association.

[37] Jalaluddin , Akio Miyara, Koutaro Tsubaki , Shuntaro Inoue , Kentaro Yoshida.(2011). Experimental study of several types of ground heat exchanger using a steel pile foundation. *Renewable Energy,* **36**, 764–771.

[38] Crane Co., (1957). Flow of Fluids: Technical Paper 410. 4th Printing, Chicago.

[39] P.C.Zhao, L.Zhao, G.L.Ding, C.L.Zhang, (2002). Experimental research on geothermal heat pump system with non-azeotropic working fluids. *Applied Thermal Engineering,* **22**, 1749–1761.

[40] Man-Hoe Kim (2005). Performance evaluation of R-22 alternative mixtures in a breadboard heat pump with pure cross-flow condenser and counter-flow evaporator. *Energy* **27**, 167–181.

[41] Mehmet Yılmaz (2003). Performance analysis of a vapor compression heat pump using zeotropic refrigerant mixtures, *Energy Conservation and Management*, **44**, 267–282.

[42] Hikmet Esena, Mustafa Inalli, Mehmet Esen. (2007). A techno-economic comparison of ground-coupled and air-coupled heat pump system for space cooling. *Building and Environment*, **42**, 1955–1965.

[43] C.Lam and Wilco Earth Heat Pump Research, *Part II, Eng. Exp*. Station Bull., University of Kentucky.5 (18).

[44] Yonghui Guo, Guoqiang Zhang, Jin Zhou , Jiasheng Wu , Wei Shen. (2012). A techno-economic comparison of a direct expansion ground-source and a secondary loop ground-coupled heat pump system for cooling in a residential building. *Applied Thermal Engineering*, **35**, 29–39.

[45] Kara Y.A., Yuksel B. (2001). Evaluation of low temperature geothermal energy through the use of heat pump. *Energy Conservation and Management*, **42**, 773–81.

[46] Swardt C. A., Meyer J.P. (2001). A performance comparison between an air-source and a ground-source reversible heat pump. *International Journal of Energy Research*, **25**, 899–910.

[47] Bose J.E., Parker J.D., McQuiston F.C. (1985). Design/data manual for closed-loop ground-coupled heat pump systems. Atlanta: ASHRAE.

[48] Healy P.F, Ugursal (1997). VI. Performance and economic feasibility of ground-source heat pumps in cold climate. *International Journal of Energy Research*, **21**, 857–70.

[49] Fleming W.S. (1998). Ground-source heat pump design and operation-experience within an Asian country. *ASHRAE Transactions*, **104**, 771–4.

[50] Kavanaugh S.P. (1992). Using existing standards to compare energy consumption of ground-source heat pumps with conventional equipment. *ASHRAE Transactions*, **98**, 599–606.

[51] TS, Turkish Standard prEN 255-3. Heat Pumps: Heat Pump Units with Electrically Driven Compressors for Heating only or for Heating and for Cooling- Part 3: Water/water and Brine/water-Air/water Heat Pump Units, Testing and Requirements for Marking. 1996. p. 9.

[52] C. Aprea, A. Greco, (2003). Performance evaluation of R22 and R407C in a vapour compression plant with reciprocating compressor. *Applied Thermal Engineering*, **23**, 215–227.

[53] J.R. Bose (1983). Design Methods for Ground-Source Heat Pumps. *ASHRAE Transactions, Part 2B,* 416–440.

[54] British Geological Survey. Ground Source Heat Pumps: Development of Geo-Reports for Potential Site Characterisation. British Geological Survey 2004; CR/05/217N; Issue 1.2.

[55] BLAST 1986. BLAST (Building Loads and System Thermodynamics). Urbana-Champaign: University of Illinois, BLAST Support Office.

[56] Viessmann UK. Technical Series: Heat Pumps. Viessmann 2009.

[57] Energy Efficiency Best Practice in Housing. Domestic Ground Source Heat Pumps: Design and Installation of Closed Loop Systems. *Energy Saving Trust.*2004; CE82.

[58] Shonder JA, Hughes PJ. Increasing Confidence in Geothermal Heat Pump Design Methods. In: Stiles L, editors. *2nd Stockton Geothermal Conference;* 1998 March 16-17; New Jersey; 1998. p. 41–57

[59] Zogou A, Stamatelos A. Effect of Climatic Conditions on the Design Optimization of Heat Pump Systems for Space Heating and Cooling. *Energy Congress.* 1996; **29 (7)**: 609–622.

[60] Rafferty K. Bulletin: Design Issues in the Commercial Application of GSHP Systems in the U.S. Oregon Institute of Technology 2010.

[61] Yumrutaş R., Önder K., and Yıldırım E. (2007). Estimation of Total equivalent temperature difference values for multilayer walls and flat roofs by using periodic solution. *Building and Environment*, **42**, 1878–1885.

[62] Önder KAŞKA, Recep YUMRUTAŞ, (2010). Kuyu suyu Kaynakli isi pompasi sistemi performansinin deneysel incelenmesi. *VIII. ulusal Temiz Enerji Sempozyumu , UTES'*. 1-5 Aralık, Bursa.

[63] Touloukian, Yeram S. (1970). Thermophysical properties of matter. New york. IFI/Plenum.

[64] Theoretical and Experimental Investigation of Total Equivalent Temperature Difference (TETD) Values for Building Walls and Flat Roofs Commonly Used in Turkey. *PhD Thesis, University of Gaziantep, Turkey 2008.*

[65] Gaziantep State Meteorological Station, Records for weather data Gaziantep, Turkey, 2011.

[66] U.S.D.O.E, United States Department of Energy, Ground-source Heat Pumps Applied to Federal Facilities, 2nd ed. Federal Technology Alerts. *The United States Department of Energy,* 2001.

[67] Yunus A. ÇENGEL (1998). Heat Transfer. New York: McGraw Hill.

APPENDIX A

Table A: Thermocouple readings and corresponding calibration equations of personaldaq 3005 data logger used in room 1

	Temperature (^oC)		
Channel	**22.6**	**45.9**	**64.2**
	Readings		
CH0	22.1	45.4	63.8
CH1	22.4	45.7	64.5
CH2	22.9	46.0	64.4
CH3	22.4	45.2	63.6
CH4	22.6	46.1	64.6
CH5	22.9	46.0	64.4
CH6	22.5	45.7	64.1
CH7	22.9	45.7	64.1
CH8	22.0	45.6	64.0
CH9	21.8	45.2	63.7
CH10	21.7	46.2	64.6
CH11	22.4	46.1	64.5
CH12	22.6	46.2	64.6
CH13	21.9	45.3	63.7
CH14	23.0	46.5	65.0
CH15	22.1	45.5	63.9
CH16	22.2	45.7	64.1
CH17	22.6	46.5	65.0
CH18	22.0	45.7	64.2
CH19	22.3	45.7	64.4
CH20	22.1	45.1	63.5
CH21	22.3	45.9	64.3
CH22	22.2	45.3	63.2
CH23	22.0	45.4	63.5

APPENDIX B

Table B1: Pressure transmitter readings and corresponding calibration equations for Gigalog dataloggers

	Gauge Pressure (bar)			
Transmitter	**0**	**1**	**5**	**10**
	Readings (mA)			
1	4.112	4.566	6.653	9.357
2	4.031	4.577	6.210	9.433
3	4.077	4.567	6.160	9.654
4	4.109	4.554	6.479	9.562

Table B2: Measurement point of pressure transmitter.

Pressure Transmitter	Measurement point
1	GCHP condenser inlet (Heating)
2	GCHP condenser outlet (Heating)
3	GCHP evaporator inlet (Heating)
4	GCHP evaporator outlet (Heating)