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THERMODYNAMIC ANALYSIS OF AIR, WATER AND GROUND SOURCE HEAT PUMPS FOR DIFFERENT REFRIGERANTS

M.Sc. Thesis by

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M.Sc. THESIS EXAMINATION RESULT FORM

We have read the thesis entitled **"THERMODYNAMIC ANALYSIS OF AIR, WATER AND GROUND SOURCE HEAT PUMPS FOR DIFFERENT REFRIGERANTS"** completed by **OZAN DEMİR** under the supervision of **ASSIST. PROF. DR. KEMAL BİLEN** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

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THERMODYNAMIC ANALYSIS OF AIR, WATER AND GROUND SOURCE HEAT PUMPS FOR DIFFERENT REFRIGERANTS

ABSTRACT

In this theoretical study, energy and exergy analysis of the air, water and ground source heat pumps used for residential heating in Gölbaşı district of Ankara were carried out. The lowest air and soil temperatures seen in the last ten years in the Gölbaşı district were provided from the Turkish State Meteorological Service and the lowest water temperature (the lowest temperature seen in Mogan Lake) seen in the last ten years was provided from General Directorate of State Hydraulic Works. In the heat pump cycle, R134a, R407a, R410a, R600a and R1234yf refrigerants were chosen as the working fluid. In the study, the coefficient of performance (COP) and exergy efficiency (η_{II}) of each heat pump were calculated and compared at Gölbaşı conditions by selecting the same ambient temperature for the residence to be heated for each heat pump. According to this, it was concluded that the ground source heat pump has a better COP value than the air and water source heat pumps and air source heat pump has a better η_{II} value than the water and ground source heat pumps for all refrigerants used as working fluid in this study at Gölbaşı conditions. In addition, it was observed that R600a refrigerant has a better performance than the other refrigerants for all type heat pumps.

Keywords: Heat pumps, HP, air, water, ground, R1234yf, COP, exergy efficiency.

HAVA, SU VE TOPRAK KAYNAKLI ISI POMPALARININ FARKLI AKIŞKANLAR İÇİN TERMODİNAMİK ANALİZİ

ÖZ

Bu teorik çalışmada; Ankara ilinin Gölbaşı ilçesinde konut ısıtması için kullanılacak hava, su ve toprak kaynaklı ısı pompalarının enerji ve ekserji analizi yapılmıştır. Gölbaşı ilçesinde son on yıl içerisinde görülmüş olan en düşük hava ve toprak sıcaklığı Meteoroloji Genel Müdürlüğünden, son on yıl içerisinde görülmüş olan en düşük su sıcaklığı (Mogan Gölünde görülmüş olan en düşük sıcaklık) ise Devlet Su İşleri Genel Müdürlüğünden temin edilmiştir. Isı pompası çevriminde iş akışkanı olarak R134a, R407a, R410a, R600a ve R1234yf soğutucu akışkanları seçilmiştir. Çalışmada; ısıtılacak ortam sıcaklığı, her bir ısı pompası için aynı seçilerek Gölbaşı şartlarında her bir ısı pompasının ısıtma tesir katsayısı (ITK) ve ekserji verimi (η_{II}) hesaplanmış ve sonuçlar karşılaştırılmıştır. Buna göre; bu çalışmada iş akışkanı olarak kullanılan tüm soğutucu akışkanlar için Gölbaşı şartlarında, toprak kaynaklı ısı pompasının, hava ve su kaynaklı ısı pompalarına göre daha iyi bir ITK değerine sahip olduğu görülmüştür. Diğer taraftan hava kaynaklı ısı pompasının su ve toprak kaynaklı ısı pompalarına göre daha iyi bir η_{II} değerine sahip olduğu sonucuna ulaşılmıştır. Ayrıca; tüm ısı pompası türleri için, R600a soğutucu akışkanının bu çalışmada kullanılan diğer soğutucu akışkanlara göre daha iyi bir performans sağladığı görülmüştür.

Anahtar kelimeler: Isı pompası, IP, hava, su, toprak, R1234yf, ITK, ekserji verimi.

CONTENTS

NOMENCLATURE

A	Area, m^2
Bo	Boiling number
c_1, c_2, c_3, c_4, c_5	Constants
Co	Convection number
COP	Coefficient of performance
c_p	Constant pressure specific heat, kJ/(kg·K)
d_i	Inside diameter of tube, m
E	Total energy, kJ
F_{fl}	Constant of fluid dependent
Fr	Froude number
G	Mass flux, kg/(m^2·s)
g	Gravitational acceleration, m/s^2
$\bar{h}$	Average heat transfer coefficient, W/(m^2·K)
h	Specific enthalpy, kJ/kg
k	Thermal conductivity, W/(m·K)
ke	Specific kinetic energy, kJ/kg
KE	Total kinetic energy, kJ
m	Mass, kg
$\dot{m}$	Mass flow rate, kg/s
P	Pressure, kPa
pe	Specific potential energy, kJ/kg
PE	Total potential energy, kJ
Pr	Prandtl number
q''	Heat flux, W/m^2
Q	Total heat transfer, kJ
$\dot{Q}$	Heat transfer rate, kW
$\dot{Q}_H$	Heat transfer with high temperature body, kJ
$\dot{Q}_L$	Heat transfer with low temperature body, kJ
$\dot{Q}_{CV}$	Net heat exchange per unit time in control volume, kW
Re	Reynolds number
s	Specific entropy, kJ/(kg·K)

$\dot{S}_{gen}$	Entropy generation, kJ/K
t	Time, s
T	Temperature, °C or K
T_c	Condensation temperature, °C or K
T_e	Evaporation temperature, °C or K
T_H	Temperature of high temperature body, K
T_L	Temperature of low temperature body, K
T_o	Surroundings temperature, °C or K
U	Total internal energy, kJ
v	Velocity, m/s
$\dot{W}_c$	Energy consumption in compressor, kW
$\dot{W}_{CV}$	Net work exchange per unit time in control volume, kW
x	Quality
X_{dest}	Total exergy destruction, kJ
$\dot{X}_{dest}$	Rate of total exergy destruction, kW
z	Elevation, m

Greek Letter Symbols

Δ	Difference
η_{II}	Second law efficiency
μ	Dynamic viscosity, (N·s)/m^2
v	Specific volume of fluid, m^3/kg
ρ	Density, kg/m^3
ρ_l	Liquid density, kg/m^3
ρ_v	Vapor density, kg/m^3
φ	Specific closed system exergy, kJ/kg

Subscripts

act	Actual
comp	Compressor
CV	Control volume
gen	Generation
H	High temperature

HP	Heat pump
i	Inlet condition
in	Input
L	Low temperature
l	Liquid
max	Maximum
min	Minimum
out	Outlet, output
r	Refrigerant
rev	Reversible
s	Source
v	Vapor
1, 2, 3, 4	Important points of the heat pump cycle

Superscripts

˙ (overdot)	Quantity per unit time

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CHAPTER 1

INTRODUCTION

Energy is needed in many areas in the universe. Energy is required in many places such as heating and cooling of buildings, machinery used in factories, electrical devices and lighting systems. Energy need, increase over time as a natural result of the increase in the world population. Nowadays, in some areas, fossil fuels such as oil, natural gas and coal are generally used as energy sources. It is known that fossil fuel use is both more costly and causes environmental pollution. Also, because fossil fuels are a nonrenewable energy source; petrol, natural gas and coal reserves are estimated to be exhausting shortly time. For this reason, the use of new and renewable energy sources is an inevitable situation to meet energy needs the worldwide. Therefore, there are many researches in the area of alternative energy sources. In general these researches, it concentrates on energy types such as solar energy, wind energy, geothermal energy, wave energy and biomass energy. In order to benefit from these alternative energy sources, respectively; solar panels, wind turbines, regional geothermal heating systems, wave generators and biogas generators are being developed. In addition to these energy conversion systems, heat pumps, it attracts attention with its features such as low energy consumption, high performance coefficient and environmental friendliness.

Heat pumps are machines that can be used for heating and/or cooling purposes by making use of energy sources such as air, water and ground. Ground and water source heat pump, in winter it carries the heat stored under the earth to the building and in summer it carries the heat inside the building to the underground. In summary, underground, it acts as a heat source in winter and a heat well in summer. The same is true for the air. In other words, the atmosphere acts as a heat source in winter and a heat well in summer. In order for the heat pump system to operate stably, the temperature of this source or well should not much change over time. In addition, these resources should be abundant and clean.

The use of heat pumps is becoming widespread in the world and in our country. The geographical conditions of the environment where the heat pump is used are very important for the efficiency of the heat pump. For example, the efficiency of the heat pump used for heating is lower in cold climates than in hot climates. This efficiency also varies according to the source where the heat pump is used.

Air is among the heat sources that can be used, is the most preferred heat source due to its easy availability. However, this source is not preferred much due to it has low efficiency in very cold weather. Water can be used as a heat source in environments where it can be obtained and water has some advantages over air. In areas where the weather conditions are bad and water sources are not available, ground can be preferred as the heat source. Ground source heat pumps are the subject of very important projects nowadays and many ground source heat pump projects are being carried out in Europe [1].

In the same climate conditions, ground source heat pumps have higher efficiency values than air source heat pumps [2]. The reason for this is that the ground temperature changes less with time than the air temperature.

In winter months, the fact that the air temperature is very variable and generally low, decreases the efficiency of air source heat pumps and thus increases the energy consumption. In contrast, throughout the year, the ground temperature at certain depths of the underground and the water temperature at certain depths of the underground do not vary much and have higher values than the air temperature. Also, the absence of water everywhere makes ground source heat pumps more preferred than water source heat pumps.

The purpose of this thesis is compare air, water and ground source heat pumps by performing energy analysis and exergy analysis under the same conditions. R134a, R600a, R410a, R407a and R1234yf refrigerants are used in heat pump systems. At the same time a comparison was made between these refrigerants. Thus in this study, it is aimed to show that the selection of heat source according to geographical conditions is important for the efficiency of the heat pump, to determine the suitable heat pump source for Ankara province and to see the usability of R1234yf refrigerant in heat pumps.

1.1 Heat Pumps

Heat pump is a machine that carries heat energy from a lower temperature environment to another environment with a higher temperature and is powered by electricity. As is known, energy can be neither created nor destroyed, it just changes form or moves from one place to another. The heat pump takes its name from the ability to pump or transport heat energy from one place to another place.

Heat pump first appeared as a concept in 1824. Carnot realized that the heat taken from the environment could be transferred to another environment by reversing the steam power cycle. However, it was first proposed by Lord Kelvin in 1852 as an idea. Lord Kelvin used air as a working fluid and built a heat pump. In this machine, the ambient air is expanded by pulling it into the cylinder, thereby reducing the pressure and temperature of the air. This low temperature and pressure air was passed through an air-to-air heat exchanger placed outside and heat was drawn from the ambient air. This heated air was raised above atmospheric pressure before it was introduced into the building. This machine designed in Switzerland has been shown to be a successful application [1].

The first heat pump application in Europe was designed in 1927 to heat a house in Scotland. In this application, air was used as the heat source. Heat pumps started to be used in America after Europe, spread rapidly until 1960. Heat pumps were started to be used in South America in 1963. However, they have some problems when used in cold regions. This has caused the reliability of the heat pump systems to decrease. In 1973, there was a great improvement in the heat pump industry and interest in the heat pump increased again. In developed countries such as America, Canada and Germany, heat pumps were widely used in both domestic and industrial areas. Heat pumps started to be used in our country in 1990, cannot be said to be very common, but heat pump applications have increased gradually in recent years [1].

When we compare heat pumps with traditional heating systems, heat pumps are more advantageous than traditional heating systems with its features such that heat pumps can be used both for heating purposes in winter and for cooling purposes in summer, making maintenance costs lower and having less environmental pollution. However, the initial investment costs of heat pumps are very high and the decrease in the efficiency of the heat pump in regions with cold weather conditions is the deficiency of heat pumps.

Heat pumps are used for both heating and cooling purposes. The working principle of the heat pump used for heating can be summarized as follows: First, thanks to the heat source, the refrigerant in the evaporator heats up and it begins to evaporate. The evaporated refrigerant is drawn and compressed by the compressor. As a result of compression, the pressure and temperature of the refrigerant is increases. Refrigerant with high temperature and pressure is sent to the condenser. Air or water used in the area to be heated is heated by the condenser. Thus, the heat energy needed by the area to be heated is met. Then refrigerant enters the expansion valve. In the constant enthalpy of the refrigerant passing through the expansion valve, the pressure and temperature are reduced and sent to the evaporator. Here, the refrigerant draws heat from the source and evaporates, then enters the compressor again. This working principle is same for air, water and ground source heat pumps. The schematic picture of the heat pump cycle is shown in Figure 1.1.

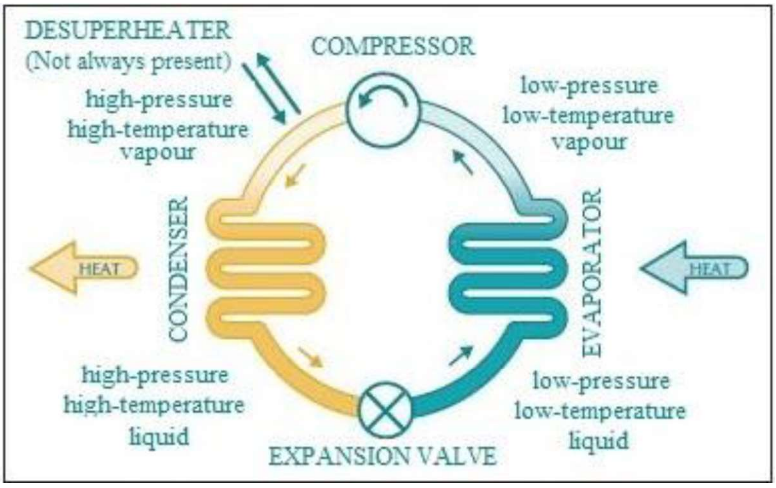

Figure 1.1 Schematic representation of the heat pump cycle [3]

1.2 The Ideal Vapor Compression Refrigeration Cycle

The vapor compression refrigeration cycle is the most used cycle in refrigeration machines, air conditioning systems and heat pumps. Schematically and *T-s* diagram of the ideal vapor compression refrigeration cycle are given in Figure 1.2.

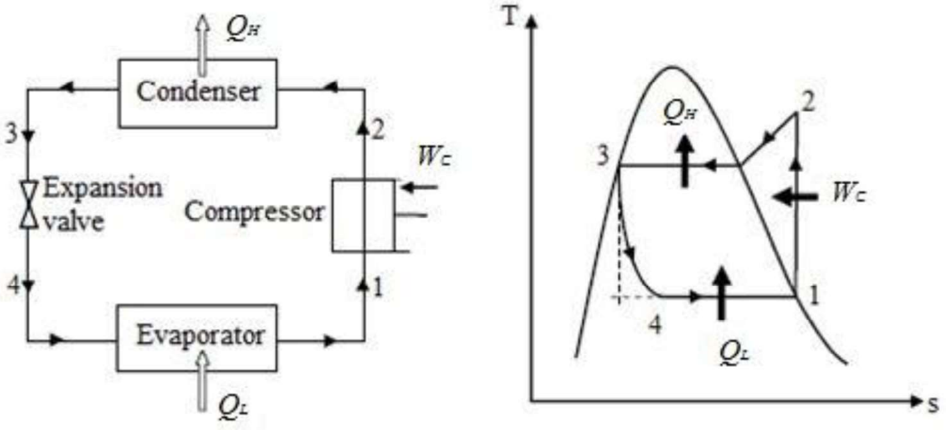

Figure 1.2 Schematic and *T-s* diagram for the ideal vapor-compression refrigeration cycle

This cycle consists of four operations: Cycle is completed by isentropic compression in a compressor (1–2), constant-pressure heat rejection in a condenser (2-3), throttling in an expansion device (3-4), constant-pressure heat absorption in an evaporator (4-1).

In an ideal vapor-compression refrigeration cycle, the refrigerant enters the compressor at state 1 as saturated vapor and is compressed isentropically to the condenser pressure. The refrigerant then enters the condenser as superheated vapor at state 2 and leaves as saturated liquid at state 3 as a result of heat rejection to the surroundings. The temperature of the refrigerant at this state is still above the temperature of the surroundings. The saturated liquid refrigerant at state 3 is throttled to the evaporator pressure by passing it through an expansion valve or capillary tube. The temperature of the refrigerant drops below the temperature of the refrigerated space during this process. The refrigerant enters the evaporator at state 4 as a low-quality saturated mixture and it completely evaporates by absorbing heat from the refrigerated space. The refrigerant leaves the evaporator as saturated vapor and reenters the compressor, completing the cycle [4].

1.3 Classification of Heat Pumps According to the Heat Sources

As known, heat pumps are machines that transfer heat from a low-temperature medium to a high-temperature medium. Heat pumps are classified as air, water and ground source heat pump according to heat sources.

1.3.1 Air source heat pumps

Air is the most widely used source as it is an easy and abundant heat source. In addition, the initial investment cost of the air source heat pump is lower than the water and ground source heat pump. For this reason, air source heat pump is both the first applied in the history and the most researched type of heat pump. Air source heat pumps have advantages as well as disadvantages. These are very variable air temperature and icing problem on the evaporator. The installation of the air source heat pump system is shown in Figure 1.3.

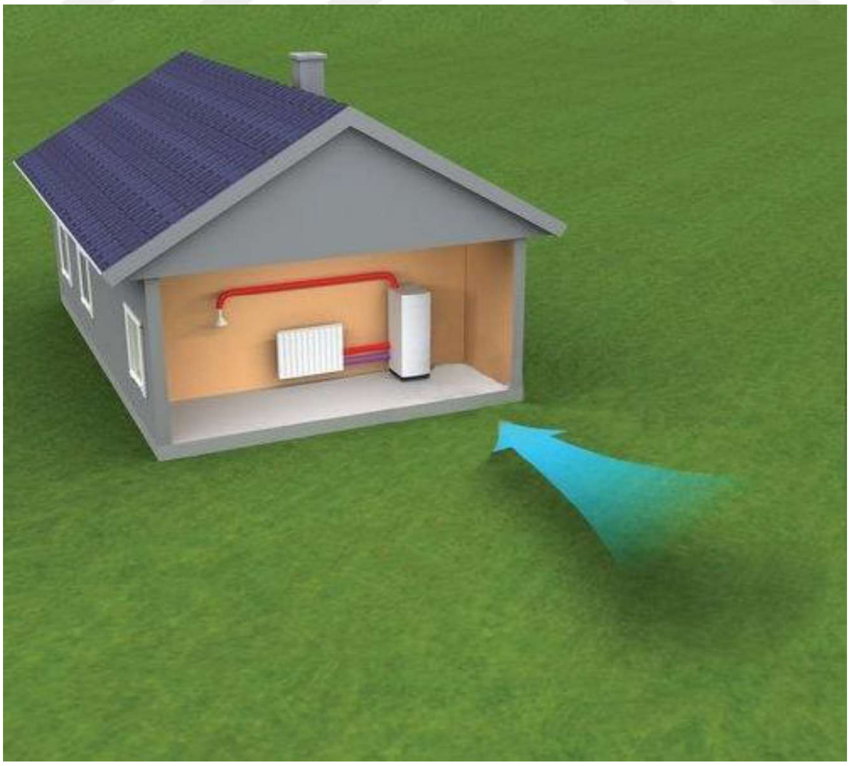

Figure 1.3 Air source heat pumps [5]

The working principle of the air source heat pump system is described in Section 1.1. The only difference is that thanks to the air heat source, the refrigerant in the evaporator heats up and it begins to evaporate. This system can take heat from the air even if the air temperature is too low. But in continental climates, especially in winter, the weather does not have stable temperature. The temperature of the air can change frequently even during the day. This circumstance is an undesirable feature in the heat pump system. Because, the heat pump has low performance and heat capacity in this situation. In such a case, the unmet heat requirement can be met by another heat source. Furthermore, the fact that the air temperature is very variable makes project design and equipment selection difficult. Considering all this, both energy consumption and cost increases in air source heat pumps.

In winter, when the heat need is the most, the air temperature decreases much and this situation reduces the cycle efficiency. For example, large amounts of water vapor in the air condense on the evaporator and freeze at low air temperatures. This ice layer must be discarded through the evaporator. For this problem, defrost operation is performed. Because frost deposit and ice formed causes heat transfer area to decrease. If the ice is allowed to accumulate on the evaporator surfaces for a long time, heat transfer is prevented. This situation causes coefficient of performance and heat capacity of the heat pump to decrease. Also, the expansion of ice during the conversion of water to ice can damage the evaporator. On the other hand, research shows that ice accumulation (up to 9.8-14.6 kg/m^2) plays a role in increasing heat transfer. However, to prevent further ice accumulation, ice must be periodically thawed from the evaporator surfaces [6]. Another problem in air source heat pumps is the low heat transfer in the evaporator. Therefore, expanded surfaces (fins) and fans are used to increase heat transfer [6].

1.3.2 Water source heat pumps

Most water bodies can be used as an energy source for water source heat pumps. In terms of geographical conditions, water source heat pump is generally preferred in places close to large water bodies such as sea, lake, river and stream. It is an important advantage for the water source heat pump that the underground water too little changes at a depth of 10 meters or more throughout the year. But the change of the temperature of these water bodies according to the seasons may cause problems. This change is more stable than the change air. That is, it shows less variability over the time.

In this system, drilling and maintenance costs are very high and it is required to obtain the necessary permissions from government agencies to use underground water. When installing the system, either the evaporator should be buried in the water or a pipe system should be installed in the water to draw the heat from the water. If the system was set up in the form of the evaporator buried in water, some problems may arise. For example, the aquatic creatures can stick to the evaporator. The installation of the water source heat pump system is shown in Figure 1.4.

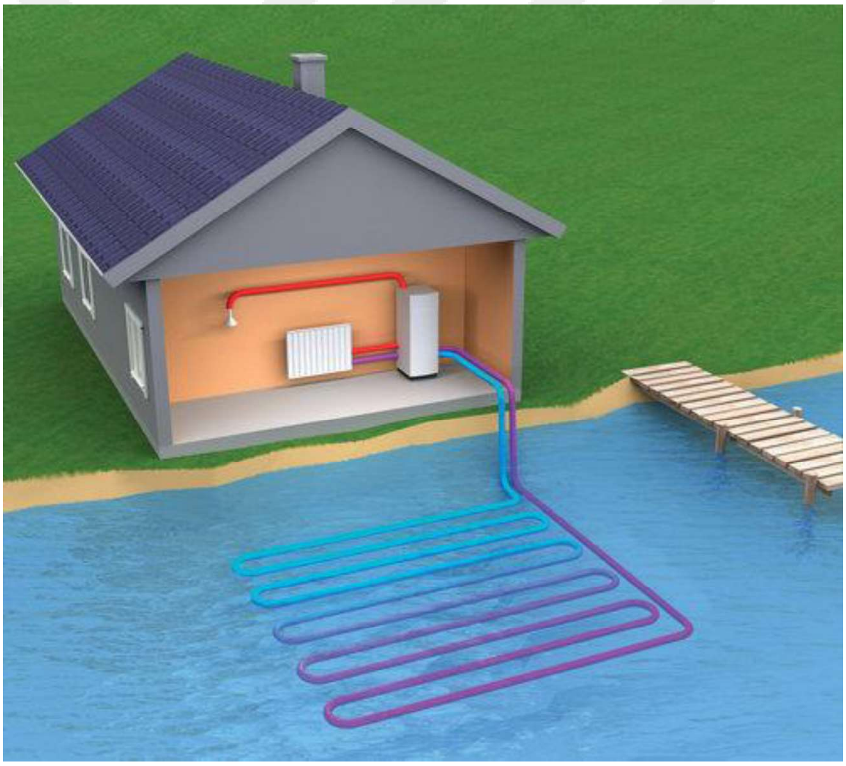

Figure 1.4 Water source heat pumps [5]

The water source will use as the heat source should be examined before the system is installed. Because minerals contained in some water sources can cause corrosion. All outdoor coils should be made of copper to prevent corrosion.

Water sources are generally in the temperature range of 4-12 °C. Also, water is a good source of heat for heat pumps through its high heat capacity and heat transfer. At the same time, its temperature never drops below the freezing point [6].

1.3.3 Ground source heat pumps

Ground is another easily accessible heat source. The sun is our biggest source of energy. Many energy sources are obtained from solar energy. Most of the solar energy is stored inside the ground. Thus, the temperature in the depths of the ground can be high even at very low air temperatures. In addition, the temperature in the depths of the ground does not very much. This is important for the performance of the heat pump. The source temperature of the ground source heat pump is more stable than the source temperature of the air and water source heat pump. Therefore, it does not require additional heat source even at low air temperatures. This feature makes the ground source heat pump have an advantage over others. The most important part in the ground source heat pump system is the heat exchangers placed in the ground. Ground density, thermal conductivity, moisture content and burial depth are important for the selection of heat exchanger to be buried in the ground. These features must be examined before the system is installed. Also, after the heat pump starts working, the temperature of the soil, the amount of moisture, etc. features may change over time. These cause difficulties in the project design phase. So, project phase is very important for the installation of the heat pump. If the pipes of the heat exchanger buried in the ground spread to large areas in order to draw heat from the ground, these effects will be reduced.

Heat exchangers to be buried in the ground can be placed horizontally or vertically. So there are two types of ground source heat pumps, horizontal type ground source heat pump and vertical type ground source heat pump.

In the horizontal type ground source heat pump system, heat exchanger pipes are buried horizontally in the ground. Horizontal type ground source heat pump draws the heat most efficiently when the ground depth is between 1 m and 2 m. In addition increasing the ground depth increases the cost as it makes digging difficult. For this reason, the cost should be taken into consideration when deciding on the depth. The installation of the the horizontal type ground source heat pump system is shown in Figure 1.5.

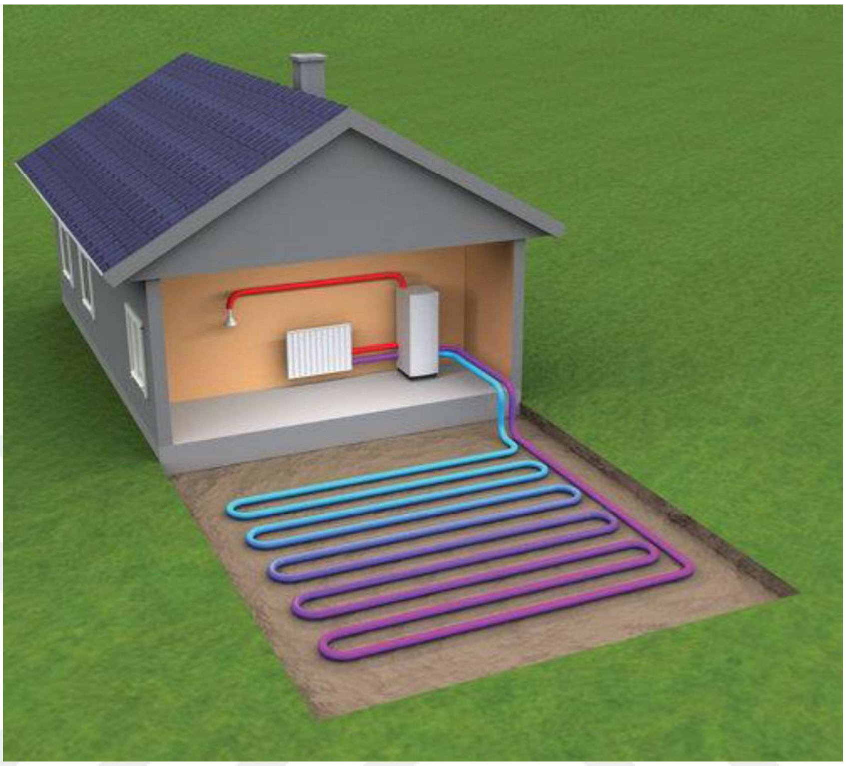

Figure 1.5 Horizontal type ground source heat pumps [5]

In the horizontal type ground source heat pump system, it is very important to compress the soil poured on the pipes after the heat exchanger pipes are placed. This operation makes the heat transfer between the ground and heat exchanger pipes more efficient. Moreover, growing grass and plants on the area where the horizontal type ground source heat pump system is placed does not harm the system.

In vertical type ground source heat pump system, heat exchanger pipes are placed vertically in the ground with the help of drilling machines. The ground depth for the vertical type ground source heat pump is between 30 m and 150 m [7]. In order to make the heat transfer between the ground and heat exchanger pipes more efficient, a filling material is filled around the pipes. The installation of the the vertical type ground source heat pump system is shown in Figure 1.6.

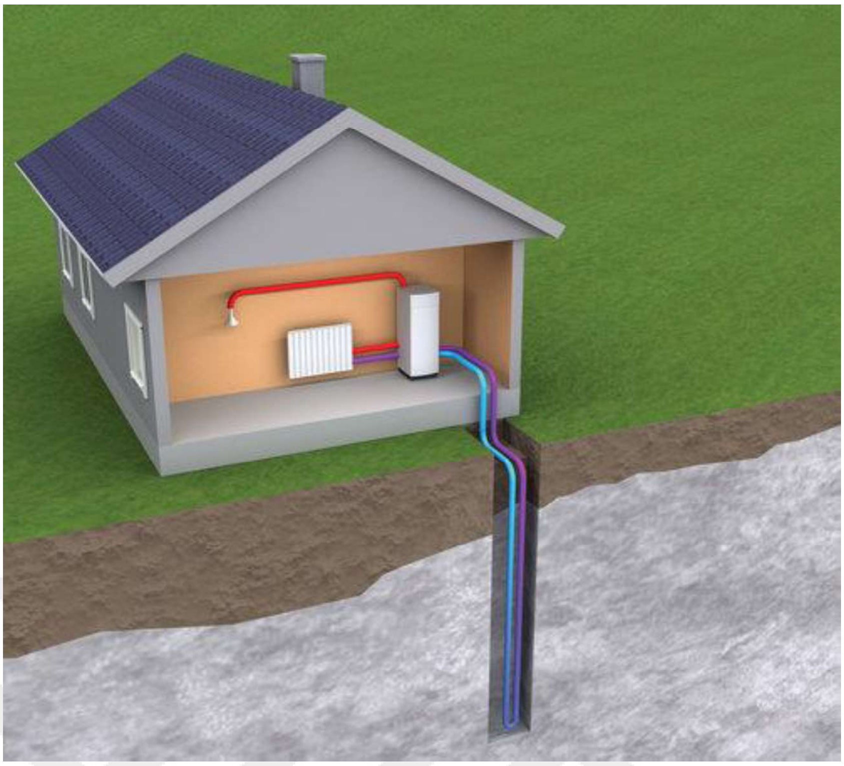

Figure 1.6 Vertical type ground source heat pumps [5]

Installation of vertical type ground source heat pump is more expensive than horizontal type ground source heat pump. The reason for this is that drilling machines are used when opening the well where the pipes will be located. The depth of the soil in which the pipes in the horizontal type ground source heat pump is placed is lower than the depth in which the pipes in the vertical type ground source heat pump are located. Thus, drilling machines are not needed in the horizontal type ground source heat pump.

The length of the pipe used in the horizontal type ground source heat pump is longer than the length of the pipe used in the vertical type ground source heat pump. This is important in terms of cost.

Comparison of air, water and ground source heat pumps described in Section 1.3 is shown in Table 1.1.

Table 1.1 Comparison of air source, water source and ground source heat pumps

Parameter	**Air source heat pumps**	**Water source heat pumps**	**Ground source heat pumps**
Additional heater requirement	Yes, due to of low air temperatures.	No	No
Availability	More easy	Not easy due to permission must be obtained from government agencies.	Easy
COP	2-4	3-5	3-5
Ease of installation	Easy	Difficult since it should be close to the water sources.	More difficult due to enough space requirement and digging.
Environment and animal friendly	Yes	Yes, but the aquatic creatures can stick to the evaporator.	Yes
Problem of icing	Yes, due to low air temperatures.	No	No
Investment cost	Cheap	Expensive due to the piping process.	More expensive due to the piping process and digging.
Stability	No stability due to variable air temperature.	Stable	More stable

1.4 Refrigerants

Refrigerants are used as an aid in transferring heat from one medium to another in a refrigeration cycle. Refrigerants generally provide heat exchange by transforming from liquid to vapor or vapor to liquid.

Refrigerants are the most important working fluids of refrigeration, air conditioning and heat pump systems. Often these refrigerants discharge the heat they draw from one medium to another, with the help of evaporation and condensation phase change processes [6].

The chemical stability of refrigerants under the conditions of use is the most important characteristic [6]. For many applications, the use of non-poisonous and non-combustible refrigerants is preferred. Easy availability and cost are other issues to consider. On the other hand, the environmental effects of the refrigerant escaping from a cooling system are also important. Halogenated components, known as chlorofluorocarbon (CFC), can remain in the atmosphere for years due to their very stability and spread over time into the stratosphere layer. CFC molecules (such as in

R11 and R12 refrigerants) contain only carbon and halogen chlorine, fluorine and/or bromine. When it reaches the upper parts of the atmosphere, the refrigerant molecules break down, releasing chlorine, which destroys the ozone layer. In the lower layers of the atmosphere, these molecules absorb infrared rays that help the earth to warm up. By replacing one or more halogens in the CFC molecules with a hydrogen atom, hydrochlorofluorocarbon (HCFC) components are formed. The presence of hydrogen in this component greatly reduces their life in the atmosphere and their negative effects on the environment [6].

Production of the components affecting the ozone layer, including chlorine and bromine refrigerants, has been taken under control by the International Montreal Protocol. In this protocol, the production of CFC type refrigerants (such as R-11, R-12, R-113 and R-114) was completely stopped on 01.01.1996. This decision was also included in the Copenhagen 1992 protocol [8]. HCFC type refrigerant production was kept constant at the production level of 01.01.1996. For developed countries, the production of this refrigerant will be reduced to 65% by 2004, 35% by 2010, 10% by 2015, 5% by 2020 and it will be stopped completely in 2030. For developing countries, it is planned to steady HCFC refrigerant production at the determined production level in 2016 and to stop HCFC refrigerant production completely in 2040 [8].

1.4.1 Global environmental characteristics of some refrigerants

Atmospheric release of CFC and HCFC refrigerants contributes to depletion of the ozone layer. The measure of a material’s ability to deplete stratospheric ozone is its ozone depletion potential (ODP), a value relative to R-11’s value of 1.0. It is the nonzero ODP of these refrigerants that led to the phase out of their production and use under the Montreal Protocol.

The global warming potential (GWP) of a greenhouse gases (GHGs) is an index describing its relative ability to trap radiant energy compared to CO_2 (R-744), which has a very long atmospheric lifetime. Measurements of climate impact of refrigerant emissions are hence often reported in CO_2 equivalents. GWP may be calculated for any particular integration time horizon (ITH). Typically, a 100 year ITH is used for calculation of GWPs for regulatory purposes and may be designated as GWP_{100}. Halocarbons (Chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs) and hydrofluorocarbons (HFCs)) and many nonhalocarbons (e.g., hydrocarbons, carbon dioxide) are GHGs. Hydrofluoro-olefins (HFOs), or unsaturated HFCs and blends using them are being developed and promoted as low-GWP alternatives to the existing halocarbon refrigerants. HFOs are also GHGs but their GWPs are drastically lower than those of HFCs. Global environmental properties of refrigerants used in this study are given in Table 1.2.

Table 1.2 Environmental properties of the refrigerants used in this study

Refrigerant	ODP	GWP_{100}
R600a (HC-600a)	0	-20
R134a (HFC-134a)	0	1300
R1234yf (HFO-1234yf)	0	<1
R407a	0	1920
R410a	0	1920

1.4.2 Required properties of the refrigerants used in the study

- It should be able to obtain more cooling with less energy consumption.
- Evaporation latent heat of the refrigerant should be high. Thus, it attracts more heat while evaporating.
- Condensing pressure should be low. High condensing pressure affects the installation elements such as compressor, condenser and pipeline.
- It should have low viscosity and low surface tension (capillarity).
- It should be environment friendly, reliable and inexpensive.
- If it escapes from the system, it should not harm the creatures and the environment.
- If it escapes from the system, it should be easily detected (such as color, odor characteristics).

- The critical temperature and pressure should be high.
- The thermal conductivity should be high.
- Its freezing temperature should be low.

In this study, it was aimed to make an approach according to the choice of refrigerant by making energy and exergy analysis for the air, water and ground source heat pump system. R134a, R600a, R410a, R407a and R1234yf refrigerants are used for this approach. In addition, R1234yf refrigerant is known to be a refrigerant similar to R134a refrigerant. Moreover looking at its global environmental characteristics, it is seen to be more environmentally friendly. As a result, when R1234yf refrigerant is used in heat pumps, heat pump efficiency has been investigated.

1.5 Literature Review

Yamankaradeniz et al. [1] explain different cycles, operating principles and thermodynamic analysis of vapor compression mechanical cooling systems, absorption cooling system and heat pumps and they are given examples for easier understanding. Furthermore, definition of refrigerants is given and brief information about the contents of refrigerants is presented.

Demir [2] examined ground heat exchangers that affect the initial investment cost and performance in ground source heat pumps and tried to be obtained optimum design conditions. The effect of each parameter on the heat exchanger size was included in the calculations.

Cengel and Boles [4] explain basic principles of thermodynamics also to present a wealth of real-world engineering applications to give us a feel for engineering practice.

Alkan [6] researched thermodynamic and thermoeconomic analysis of heat pump system with used alternative refrigerants for residential heating in Isparta. The refrigerants used are R22, R600a, R134a, R290, R410A and R404A. R600a was obtained highest COP and exergy efficiency value of the system among the refrigerants. R600a was followed by R22, R134a, R290, R410A and R404A refrigerants, respectively.

Kusuda [9] presented simplified calculation procedure for determining heat exchange between the earth and a multiplicity of buried pipes having different temperature and thermal insulation.

Kotas [10] mentions various aspects of power generation, refrigeration and cryogenic processes, distillation and chemical processes, including combustion in his book. This book gives examples of exergy analysis on thermal systems and examines thermoeconomic optimization of thermal systems in detail. Kotas uses the structural bond coefficients method in the optimization.

Yavuztürk [11] developed a numerical model for detailed analysis and accurate simulation of transient heat transfer in vertical ground loop heat exchangers in his thesis. The model was based on a two-dimensional, fully enclosed finite volume formulation and used an automatic parametric grid generation algorithm for different pipe sizes, shank spacing and borehole geometry.

Tong and Tang [12] progressed in many aspects in his book. Such as the modeling of two-phase flow, the evaluation of and experimentation on the forced-convection boiling crisis as well as heat transfer beyond the critical heat flux conditions and extended research in liquid-metal boiling. This book reexamines the accuracy of existing, generally available correlations by comparing them with updated data and thereby providing designers with more reliable information for predicting the thermal hydraulic behavior of boiling devices.

Rohsenow et al. [13] give us information about the conduction, convection, radiation, condensation and boiling in their book. At the same time, this book contains topics that thermophysical properties, heat transfer enhancement, heat exchangers, heat pipes, fluidized beds, non-Newtonian fluids and measurement techniques.

Cengel [14] explains that basic principles of heat transfer also to present a wealth of real-world engineering applications to give us a feel for engineering practice.

Dincer and Rosen [15] give us information about exergy and its applications to various energy systems and applications as a potential tool for design, analysis and optimization and its role in minimizing or eliminating environmental impacts and providing sustainable development. In this regard, several key topics ranging from the basics of the thermodynamic concepts to advanced exergy analysis techniques in a wide range of applications are covered as outlined in the contents.

Elbir [16] applied exergy analysis for ground sourced heat pumps in Lakes Region (Antalya, Burdur and Isparta). Exergy destruction of heat pump components, which are compressor, condenser, expansion valve, evaporator, fan, storage tank and circulating pump were evaluated for heating in the winter months. COP values for Antalya vary between 2.25 and 4.54, for Burdur between 5.4 and 7.81 and for Isparta between 5.64 and 7.89.

Al-Khalıdı [17] has been investigated theoretically and experimentally about the COP value of the ground source heat pump operating in heating mode at different mass flow rates. COP value was calculated based on mass flow rates. It was observed that the COP value increased as the mass flow rates increased. According to results, the maximum COP value of the system was 3.42.

Wu and Skye [18] studied on ground source heat pump system using natural refrigerant such as CO_2, NH_3, water and hydrocarbons. Furthermore compared the refrigerants' thermodynamic properties and discussed recent progress in their use in ground source heat pump system. On cooling mode COP values equal to 5.91 for CO_2, 9.07 for NH_3, 8.75 for water, 8.90 for propane and 9.20 for isobutane. On heating mode COP values equal to 3.23 for CO_2, 4.71 for NH_3, 5.93 for water, 4.60 for propane and 4.72 for isobutane.

Sagia and Rakopoulos [19] studied on ground source heat pump system using alternative refrigerants such as R32/R134a in compositions 20/80%, 30/70%, 40/60% by mass and the ternary blends R407B (10% R32, 70% R125, 20% R134a), R152a/R125/R32 (48% R152a, 18% R125, 34% R32), R-410B (45% R32, 55% R125) and R507A (50% R125, 50% R143a). COP values equal to 4.61 for R22, 4.246 for R410B, 4.062 for R507A, 4.591, 4.576, 4.524 respectively for R32/R134a in compositions respectively 20/80%, 30/70%, 40/60%, 4.608 for R152a/R125/R32, 4.201 for R407B.

Sackan [20] examined sea water source heat pump system installed at Asia Beach Resort Spa Hotel in Alanya/Antalya. For this purpose, the hotel has an area of 14.000 m^2 and this system has been operated for 1 hour at full load of cooling the rooms and a bedroom. Thus, to analyses the irreversibility and thermodynamic performance of the system components, an exergy analysis is performed according to second law of thermodynamics. The results of the study show that the exergy efficiency of the system is 66.5%.

Fei and Pingfang [21] investigated the performance characteristics of a groundwater heat pump system (GWHP) on the actual operation, using the energy and exergy analysis method. The system was installed in apartment buildings of Wuhan, China. In the duration of one year, various operating parameters of the system were monitored and the coefficient of performance (COP) values of system and chiller is determined based on a series of measurements. As a result, on cooling mode COP values equal to 5.46, exergy efficiency of system 0.19 and on heating mode COP values equal to 4.39, exergy efficiency of system 0.50.

Wang et al. [22] presented an investigation of a novel frost-free air source heat pump system that integrated with dehumidification and thermal energy storage, working with R134a and R407C as R22 alternative. Mathematical model of the system was constructed and verified by comparison with experimental data that shows the measured results were in good accordance with the numerical ones. As a result, at the ambient temperature of -10 °C and RH of 85%, the average COP for R134a was 3.3 and 8.6% higher respectively than that for R22 and R407C.

CHAPTER 2

DESIGN PROCEDURES OF HEAT PUMPS

In this theoretical study, energy and exergy analysis of the air, water and ground source heat pumps used for residential heating in Gölbaşı district of Ankara was carried out. The heat pumps examined in this study are shown schematically in Figure 2.1.

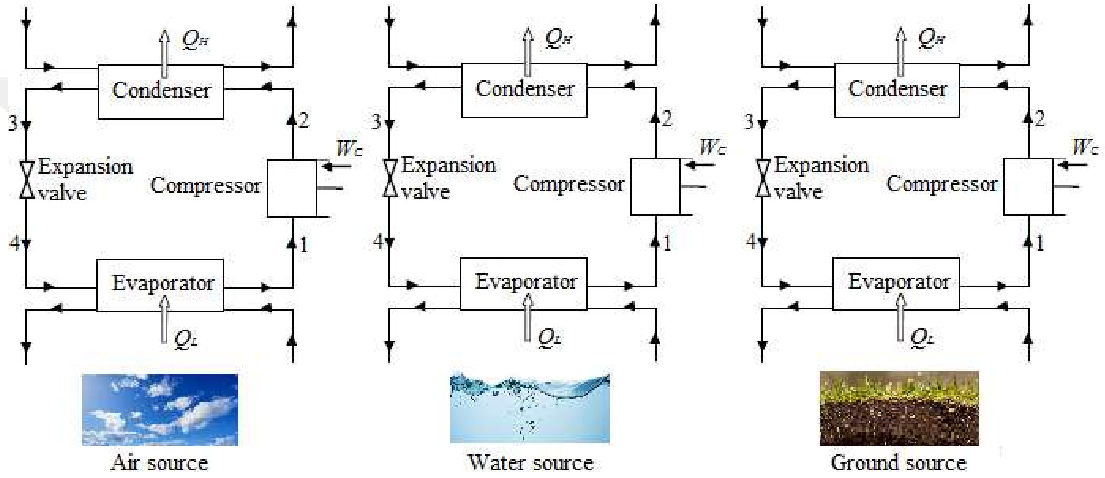

Figure 2.1 Schematic of the air, water and ground source heat pump cycles

First, thanks to the heat source, the refrigerant in the evaporator heats up and begins to evaporate. The evaporated refrigerant is drawn and compressed by the compressor. As a result of compression, refrigerants' pressure and temperature are increased. Refrigerant with high temperature and pressure is sent to the condenser. Air or water used in the area to be heated is heated by the condenser. Thus, the heat energy needed by the area to be heated is met. Then refrigerant enters the expansion valve. In the constant enthalpy of the refrigerant passing through the expansion valve, pressure and temperature are reduced and sent to the evaporator. Here, refrigerant draws heat from the source and evaporates, then enters the compressor again.

In this study, while making analysis of air, water and ground source heat pumps, some assumptions were made for the system:

- The mass flow rate of refrigerant circulating in the heat pump system is assumed to be constant.

- It is assumed that there is no heat transfer between all connection pipes and fittings in the system and the external environment.

- The pressure drop in the system has been neglected.

- It is assumed that the compression process in the compressor takes place adiabatically.

- The mechanical efficiency of the compressor is considered to be 80%.

- It is assumed that the source temperature does not change over time.

- For the ground source heat pump, it is assumed that soil properties such as moisture and soil density do not change over time.

2.1 First and Second Law Equations of Thermodynamics for Heat Pumps

Thermodynamics can be defined as the science of energy [4]. Energy can be viewed as the ability to cause changes. The first law of thermodynamics is simply an expression of the conservation of energy principle and it asserts that energy is a thermodynamic property. The second law of thermodynamics asserts that energy has quality as well as quantity and actual processes occur in the direction of decreasing quality of energy [4].

2.1.1 Conservation of mass

The conservation of mass principle is implicitly used by requiring that the mass of the system remain constant during a process for closed systems. For control volumes, however, mass can cross the boundaries and so we must keep track of the amount of mass entering and leaving the control volume [4]. Therefore, Equation 2.1 is used for conservation of mass in heat pump elements, which are control volume.

The net mass transfer to or from a control volume during a time interval Δt is equal to the net change (increase or decrease) in the total mass within the control volume during Δt [4].

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = \Delta \dot{m}_{CV} \tag{2.1}$$

where, the index CV represents the control volume,

$\Delta \dot{m}_{CV}$: Mass change in unit time in the control volume,

$\sum \dot{m}_{in}$: The total mass flow rate entering the control volume,

$\sum \dot{m}_{out}$: The total mass flow rate output from the control volume.

During a steady-flow process, the total amount of mass contained within a control volume does not change with time ($\dot{m}_{CV} \equiv$ constant). Then the conservation of mass principle requires that the total amount of mass entering a control volume equal the total amount of mass leaving it. So,

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{2.1a}$$

2.1.2 Conservation of energy

During a steady-flow process, the total energy content of a control volume remains constant ($E_{CV} \equiv$ constant) and thus the change in the total energy of the control volume is zero ($\Delta E_{CV} = 0$) [4]. Therefore, the amount of energy entering a control volume in all forms (by heat, work and mass) must be equal to the amount of energy leaving it. Then the rate form of the general energy balance reduces for a steady-flow process to,

$$\sum \dot{E}_{in} - \sum \dot{E}_{out} = \Delta \dot{E}_{CV} \tag{2.2}$$

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \tag{2.2a}$$

$$\dot{Q}_{CV} + \dot{W}_{CV} = \sum \dot{m}\left(h + \frac{V^2}{2} + gz\right)_{\text{out}} - \sum \dot{m}\left(h + \frac{V^2}{2} + gz\right)_{\text{in}} \tag{2.2b}$$

When the fluid experiences negligible changes in its kinetic and potential energies (that is, $\Delta ke = 0$, $\Delta pe = 0$), the energy balance equation is reduced further to

$$\dot{Q}_{CV} + \dot{W}_{CV} = \sum \dot{m}(h)_{\mathrm{out}} - \sum \dot{m}(h)_{\mathrm{in}} \tag{2.2c}$$

where,

$\dot{Q}_{CV}$: Net heat exchange of control volume per unit time,

$\dot{W}_{CV}$: Net power exchange of control volume per unit time,

$\dot{m}$: Mass flow rate,

h: Enthalpy.

Equation (2.2c) is applied for the compressor, taking into account the assumptions,

$$\dot{W}_{comp} = \dot{m}_r(h_2 - h_1) \tag{2.3}$$

Equation (2.2c) is applied for the condenser, taking into account the assumptions,

$$\dot{Q}_C = \dot{m}_r(h_2 - h_3) \tag{2.4}$$

Equation (2.2c) is applied for the evaporator, taking into account the assumptions,

$$\dot{Q}_E = \dot{m}_r(h_1 - h_4) \tag{2.5}$$

$$\dot{Q}_E = \dot{m}_s c_{p,s}(\Delta T_s) \tag{2.6}$$

Equation (2.2c) is applied for the expansion valve, taking into account the assumptions,

$$h_3 = h_4 \tag{2.7}$$

Correlation of two-phase flow should be used for heat transfer calculations in the evaporator because the heat transfer coefficient of the refrigerant in the evaporator should be calculated. Some of these correlations are, in 1963, Chen proposed the flow boiling correlation for evaporation in vertical tubes to attain widespread use.

Chen's correlation includes both the heat transfer coefficients due to nucleate boiling as well as forced convective mechanisms [23]. Correlation developed by Shah in 1982 is a method prepared for vertical and horizontal pipes. Shah argued that the two-phase heat transfer coefficient was created by convective and core boiling models. Shah method predicts the convective and core boiling heat transfer coefficient as the double phase heat transfer coefficient [24]. Correlation developed by Kandlikar in 1983 is a method prepared for vertical and horizontal pipes. The Kandlikar equation predicts that the two-phase flow is formed by the coexistence of nuclear boiling and convective boiling equations. Kandlikar suggested that the larger of the core boiling or convective boiling heat transfer coefficient would be the dual phase heat transfer coefficient [24]. A literature search has been carried out on two - phase flow correlations. Deviation values of the correlations and refrigerants using the correlations were examined. As a result, Kandlikar's correlation was used in this study. This correlation and the equations associated with this correlation are given in the literature [25], as in Equation (2.8) to Equation (2.13):

$$\frac{h}{h_l} = \mathrm{c}_1 Co^{\mathrm{c}_2}(25Fr_{lo})^{c_5} + c_3 Bo^{c_4} F_{fl} \tag{2.8}$$

$$h_l = 0.023 Re_l^{0.8} Pr_l^{0.4}\left(\frac{k_l}{d_i}\right) \tag{2.9}$$

$$Re_l = \frac{G d_i (1-x)}{\mu_l} \tag{2.10}$$

$$Co = \left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_v}{\rho_l}\right)^{0.5} \tag{2.11}$$

$$Fr_{lo} = \frac{G^2}{g d_i \rho_l^2} \tag{2.12}$$

$$Bo = \frac{q''}{G h_{lv}} \tag{2.13}$$

where, *h* heat transfer coefficient (W/(m^2·°C)), *Re* Reynolds number, *G* mass flux (kg/(m^2·s)), *Co* convection number, *Pr* Prandtl number, *μ* dynamic viscosity (N·s/m^2), *Fr* Froude number, *k* thermal conductivity (W/(m^2·°C)), *Bo* boiling number, d_i inside diameter (m), F_{fl} fluid-dependent parameter, ρ density (kg/m^3), l liquid, v vapor and x quality.

The efficiency of a heat pump is expressed in terms of the coefficient of performance (COP), indicated by $\mathrm{COP_{HP}}$. The objective of a heat pump is to heat (Q_H) the field. To accomplish this objective, it requires a work input of $W_{net,in}$. Then the COP of a heat pump can be expressed as

$$COP_{HP} = \frac{Desired\ output}{Required\ input} = \frac{Q_H}{W_{net\,,in}} \tag{2.14}$$

This relation can also be expressed in rate form by replacing $Q_H$by $\dot{Q}_H$ and $W_{net\,,in}$ by $\dot{W}_{net\,,in}$ [4].

The conservation of energy principle for a heat pump cycle,

$$W_{net\,,in} = Q_H - Q_L \tag{2.15}$$

$$COP_{HP} = \frac{Q_H}{Q_H - Q_L} = \frac{1}{1 - Q_L/Q_H} \tag{2.16}$$

The maximum COP of a heat pump cycle operating between temperature limits of T_L and T_H (the COPs of reversible heat pumps) was given in Eq. 2.17 as

$$COP_{HP\,,Carnot} = \frac{1}{1 - T_L/T_H} \tag{2.17}$$

2.1.3 Exergy analysis

Exergy is defined as a useful work potential where a system or a flowing substance or energy can be produced until equilibrium with reference environmental conditions. Exergy analysis is a thermodynamic analysis technique based on the second law of thermodynamics which provides an alternative and illuminating means of assessing and comparing processes and systems rationally and meaningfully. In particular, exergy analysis yields efficiencies which provide a true measure of how nearly actual performance approaches the ideal and identifies more clearly than energy analysis the causes and locations of thermodynamic losses [15]. The main differences between energy and exergy are given in Table 2.1.

Table 2.1 The main differences between energy and exergy [26]

Energy	Exergy
1) It is dependent on the parameters of matter or energy flow only	**1)** It is dependent on the parameters of matter, energy flow and on the environment parameters.
2) It has values different from zero (equal to mc^2 in accordance with Einstein's equation).	**2)** It is equal to zero in a dead state by equilibrium with the environment.
3) It is guided by the first law of thermodynamics for all the processes.	**3)** It is guided by the first law of thermodynamics for reversible processes only (in irreversible processes it is destroyed partly or completely).
4) It is limited by the second law of thermodynamics for all processes	**4)** It is not limited for reversible processes due to the second law of thermodynamics.
5) It is motion or ability to produce motion.	**5)** It is work or ability to produce work.
6) It is always conserved in a process.	**6)** It is always conserved in a reversible process, but is always consumed in an irreversible process.
7) It is a measure of quantity.	**7)** It is a measure of quantity and quality due to entropy.

The goal of a second-law or exergy analysis of a heat pump system is to determine the components that can benefit the most by improvements. This is done identifying the locations of greatest exergy destruction and the components with the lowest exergy or second-law efficiency. Exergy destruction in a component can be determined directly from an exergy balance or indirectly by first calculating the entropy generation and then using the relation

$$\dot{X}_{dest} = T_0\dot{S}_{gen} \quad \textbf{(2.18)}$$

where T_0 is the environment (the dead state) temperature. Exergy destructions and exergy or the second-law efficiencies for major components of a heat pump system operating on the cycle shown in Figure 2.1 may be written as follows:

Compressor,

$$\dot{X}_{dest,1-2} = T_0\dot{S}_{gen,1-2} = \dot{m}T_0(s_2 - s_1) \quad \textbf{(2.19)}$$

$$\eta_{\text{II},Comp} = \frac{\dot{X}_{recovered}}{\dot{X}_{expended}} = \frac{\dot{W}_{rev}}{\dot{W}_{act,in}} = \frac{\dot{m}[h_2-h_1-T_0(s_2-s_1)]}{\dot{m}(h_2-h_1)} = \frac{\varphi_2-\varphi_1}{h_2-h_1}$$

$$= 1 - \frac{\dot{X}_{dest,1-2}}{\dot{W}_{act,in}} \quad \textbf{(2.20)}$$

Condenser,

$$\dot{X}_{dest,2-3} = T_0\dot{S}_{gen,2-3} = T_0\left[\dot{m}(s_3 - s_2) + \frac{\dot{Q}_H}{T_H}\right] \tag{2.21}$$

$$\eta_{\text{II},Cond} = \frac{\dot{X}_{recovered}}{\dot{X}_{expended}} = \frac{\dot{X}_{Q_H}}{\dot{X}_2-\dot{X}_3} = \frac{\dot{Q}_H(1-T_0/T_H)}{\dot{X}_2-\dot{X}_3} = \frac{\dot{Q}_H(1-T_0/T_H)}{\dot{m}[h_2-h_3-T_0(s_2-s_3)]}$$

$$= 1 - \frac{\dot{X}_{dest,2-3}}{\dot{X}_2-\dot{X}_3} \tag{2.22}$$

Expansion valve,

$$\dot{X}_{dest,3-4} = T_0\dot{S}_{gen,3-4} = \dot{m}T_0(s_4 - s_3) \tag{2.23}$$

$$\eta_{\text{II},ExpValve} = \frac{\dot{X}_{recovered}}{\dot{X}_{expended}} = \frac{0}{\dot{X}_3-\dot{X}_4} = 0 \tag{2.24}$$

or

$$\eta_{\text{II},ExpValve} = 1 - \frac{\dot{X}_{dest,3-4}}{\dot{X}_{expended}} = 1 - \frac{\dot{X}_3-\dot{X}_4}{\dot{X}_3-\dot{X}_4} = 0 \tag{2.24a}$$

Evaporator,

$$\dot{X}_{dest,4-1} = T_0\dot{S}_{gen,4-1} = T_0\left[\dot{m}(s_1 - s_4) - \frac{\dot{Q}_L}{T_L}\right] \tag{2.25}$$

$$\eta_{\text{II},Evap} = \frac{\dot{X}_{recovered}}{\dot{X}_{expended}} = \frac{\dot{X}_{Q_L}}{\dot{X}_4-\dot{X}_1} = \frac{\dot{Q}_L(T_0-T_L)/T_L}{\dot{X}_4-\dot{X}_1} = \frac{\dot{Q}_L(T_0-T_L)/T_L}{\dot{m}[h_4-h_1-T_0(s_4-s_1)]}$$

$$= 1 - \frac{\dot{X}_{dest,4-1}}{\dot{X}_4-\dot{X}_1} \tag{2.26}$$

Here $\dot{X}_{\dot{Q}_H}$ represents the positive of the exergy rate associated with the withdrawal of heat from the low-temperature medium at T_H at a rate $\dot{Q}_H$. Note that the directions of heat and exergy transfer become opposite when $T_H < T_0$ (that is, the exergy of the low-temperature medium increases as it loses heat). Also, $\dot{X}_{\dot{Q}_H}$ is equivalent to the power that can be produced by a Carnot heat engine receiving heat from the environment at T_0 and rejecting heat to the low temperature medium at T_H at a rate of $\dot{Q}_H$, which can be shown to be

$$\dot{X}_{\dot{Q}_H} = \dot{Q}_H\left(1-\frac{T_0}{T_H}\right) \tag{2.27}$$

From the definition of reversibility, this is equivalent to the minimum or reversible power input required to remove heat at a rate of $\dot{Q}_H$ and reject it to the environment at T_0. That is, $\dot{W}_{rev,in} = \dot{W}_{min,in} = \dot{X}_{\dot{Q}_H}$

The total exergy destruction associated with the cycle is the sum of the exergy destruction:

$$\dot{X}_{dest,total} = \dot{X}_{dest,1-2} + \dot{X}_{dest,2-3} + \dot{X}_{dest,3-4} + \dot{X}_{dest,4-1} \tag{2.28}$$

It can be shown that the total exergy destruction associated with a heat pump cycle can also be obtained by taking the difference between the exergy supplied (power input) and the exergy recovered (the exergy of the heat withdrawn from the low-temperature medium):

$$\dot{X}_{dest,total} = \dot{W}_{in} - \dot{X}_{\dot{Q}_H} \tag{2.29}$$

The second-law or exergy efficiency of the cycle can then be expressed as:

$$\eta_{\text{II},cycle} = \frac{\dot{X}_{\dot{Q}_H}}{\dot{W}_{in}} = 1 - \frac{\dot{X}_{dest,total}}{\dot{W}_{in}} \tag{2.30}$$

Substituting $\dot{W}_{in} = \frac{\dot{Q}_H}{COP_{HP}}$ and $\dot{X}_{\dot{Q}_H} = \dot{Q}_H\left(1-\frac{T_0}{T_H}\right)$ into Eq. (3.30) gives

$$\eta_{\text{II},cycle} = \frac{\dot{X}_{\dot{Q}_H}}{\dot{W}_{in}} = \frac{\dot{Q}_H(1-T_0/T_H)}{\dot{Q}_H/COP_{HP}} = \frac{COP_{HP}}{(1-T_0/T_H)} = \frac{COP_{HP}}{COP_{HP,rev}} \tag{2.30a}$$

Since $T_0 = T_L$ for a heat pump cycle. Thus, the second-law efficiency is also equal to the ratio of actual and maximum COPs for the cycle. This second-law efficiency definition accounts for all irreversibility associated within the heat pump [4].

CHAPTER 3

RESULTS AND DISCUSSIONS

3.1 Results for Heating Mode

In this theoretical study, the first and second law of thermodynamics of air, water and ground heat pumps was used for heating purposes for a house which needs 5 kW of heating in Gölbaşı district of Ankara province. The lowest air, water and ground temperature values seen in the last 10 years in Gölbaşı district of Ankara province were obtained from the necessary institutions. The temperature values required for air and ground source heat pump were obtained from the Turkish State Meteorological Service. The temperature value required for water source heat pump was taken from the General Directorate of State Hydraulic Works. According to these data, the lowest air temperature seen in the last 10 years was -18.5 °C, the lowest ground temperature was 9.8 °C and the lowest water temperature seen was 6.5 °C, based on Mogan Lake in Gölbaşı district. Copper pipes were used in all heat pump systems. R600a, R134a, R410a, R407a and R1234yf were used separately as refrigerant in each heat pump system and the mass flow rate of the fluids were calculated as 0.01674 kg/s, 0.03294 kg/s, 0.036258 kg/s, 0.03691 kg/s and 0.04095 kg/s respectively. In the calculations, some tables were used for the thermodynamic properties of fluids. Genetron Properties program was used for these tables. Temperature, pressure, specific enthalpy, entropy values for the compressor, condenser, throttling valve and evaporator, which are the basic elements of the heat pump system, were calculated using the assumptions shown in Chapter 2, Kandlikar's correlation equation and the first and second law of thermodynamics. Then, the coefficient of performance of the air, water and ground source heat pump, the exergy losses of the basic elements of the heat pump system and the second law efficiency of the system were calculated. These values were calculated and compared separately for R600a, R134a, R410a, R407a and R1234yf refrigerants in each heat pump (air, water and ground source).

The thermodynamic properties of the refrigerants used for the heat pumps examined are shown in Table 3.1 and the exergy losses of the heat pump system and its basic elements are shown in Table 3.2.

Table 3.1 Thermodynamic properties of the refrigerants used in this study

Type of heat source	Refrigerant	T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	P_1 (kPa)	P_2 (kPa)	P_3 (kPa)	P_4 (kPa)	h_1 (kJ/kg)	h_2 (kJ/kg)	h_3 (kJ/kg)	h_4 (kJ/kg)	s_1 (kJ/kg·K)	s_2 (kJ/kg·K)	s_3 (kJ/kg·K)	s_4 (kJ/kg·K)
Air	R134a	-25.00	57.70	40.00	-25.00	106.40	1016.60	1016.60	106.40	383.450	438.768	286.977	286.977	1.737	1.771	1.288	1.357
	R407a	-25.00	61.17	40.00	-25.00	244.60	1837.00	1837.00	244.60	383.870	433.248	297.784	297.784	1.817	1.847	1.421	1.474
	R410a	-25.00	65.58	40.00	-25.00	330.50	2425.60	2425.60	330.50	413.180	463.442	325.541	325.541	1.883	1.913	1.477	1.533
	R600a	-25.00	51.10	40.00	-25.00	58.40	531.20	531.20	58.40	520.990	629.218	330.532	330.532	2.320	2.388	1.436	1.537
	R1234yf	-25.00	43.50	40.00	-25.00	122.90	1018.40	1018.40	122.90	346.690	391.227	269.127	269.127	1.592	1.620	1.231	1.282
Water	R134a	1.00	62.10	40.00	1.00	303.60	1016.60	1016.60	303.60	399.200	443.452	291.661	291.661	1.759	1.785	1.303	1.334
	R407a	1.00	64.50	40.00	1.00	620.30	1837.00	1837.00	620.30	397.270	437.140	301.675	301.675	1.835	1.859	1.434	1.473
	R410a	1.00	65.30	40.00	1.00	826.50	2425.60	2425.60	826.50	422.820	463.123	325.222	325.222	1.888	1.912	1.476	1.523
	R600a	1.00	57.40	40.00	1.00	162.60	531.20	531.20	162.60	555.700	641.521	342.835	342.835	2.373	2.426	1.475	1.521
	R1234yf	1.00	51.40	40.00	1.00	326.70	1018.40	1018.40	326.70	363.940	400.242	278.142	278.142	1.626	1.648	1.259	1.285
Ground	R134a	4.00	58.60	40.00	4.00	337.70	1016.60	1016.60	337.70	400.920	439.728	287.937	287.937	1.750	1.774	1.291	1.317
	R407a	4.00	62.00	40.00	4.00	682.20	1837.00	1837.00	682.20	398.700	434.165	298.700	298.700	1.828	1.849	1.424	1.458
	R410a	4.00	62.25	40.00	4.00	907.80	2426.00	2426.00	907.80	423.700	459.224	321.324	321.324	1.879	1.900	1.464	1.504
	R600a	4.00	53.80	40.00	4.00	180.40	531.20	531.20	180.40	559.700	634.466	335.780	335.780	2.358	2.404	1.452	1.490
	R1234yf	4.00	46.60	40.00	4.00	360.90	1018.00	1018.00	360.90	365.900	394.879	275.689	275.689	1.614	1.632	1.251	1.273

Table 3.2 Exergy losses of heat pump elements

Type of heat source	Refrigerant	Loss of exergy (kW)				
		Compressor	Condenser	Expansion valve	Evaporator	System
Air	R134a	0.285198	0.218679	0.579874	0.008919	1.09267
	R407a	0.281974	0.269761	0.492608	0.048717	1.09306
	R410a	0.277916	0.246624	0.515299	0.053063	1.09290
	R600a	0.288722	0.210614	0.433574	0.149323	1.08223
	R1234yf	0.296987	0.204865	0.540583	0.051867	1.09430
Water	R134a	0.245030	0.247978	0.288417	0.365978	1.14740
	R407a	0.243596	0.304709	0.407921	0.205133	1.16136
	R410a	0.243349	0.270025	0.478384	0.159300	1.15106
	R600a	0.245302	0.239062	0.215388	0.426639	1.12639
	R1234yf	0.258808	0.234139	0.294422	0.388949	1.17632
Ground	R134a	0.220893	0.244566	0.243448	0.314530	1.02344
	R407a	0.223495	0.302247	0.351430	0.160849	1.03802
	R410a	0.218521	0.264183	0.415292	0.135132	1.03313
	R600a	0.216462	0.237620	0.178427	0.364170	0.99668
	R1234yf	0.209721	0.338402	0.249580	0.253271	1.05097

Graphical representations are given to better understand the results of the heat pump energy and exergy analysis according to the heat pump type and the type of refrigerant used.

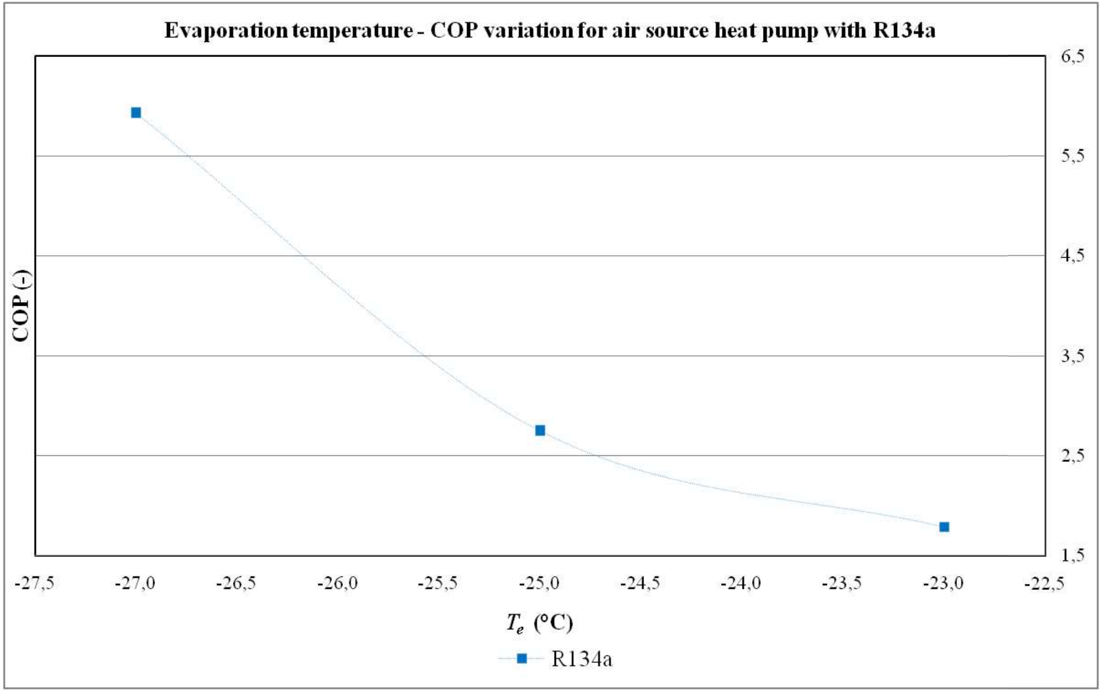

Figure 3.1 T_e - COP variation for air source heat pump with R134a

The change of coefficient of performance according to the evaporation temperature for R134a in the air source heat pump is shown in Figure 3.1.

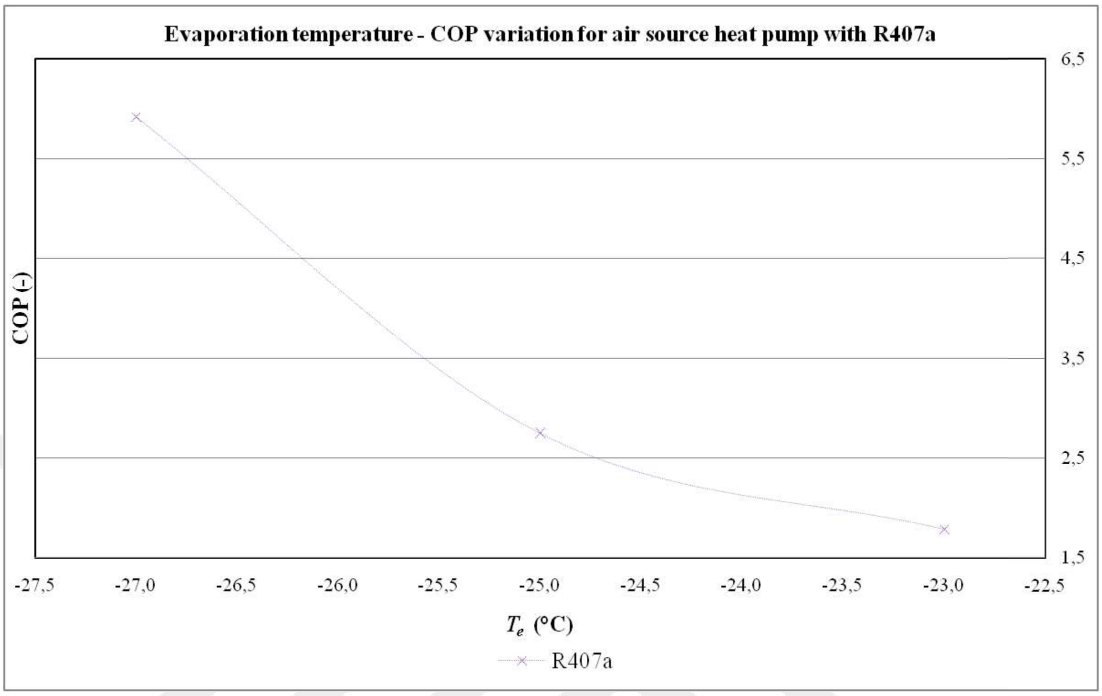

Figure 3.2 T_e - COP variation for air source heat pump with R407a

The change of coefficient of performance according to the evaporation temperature for R407a in the air source heat pump is shown in Figure 3.2.

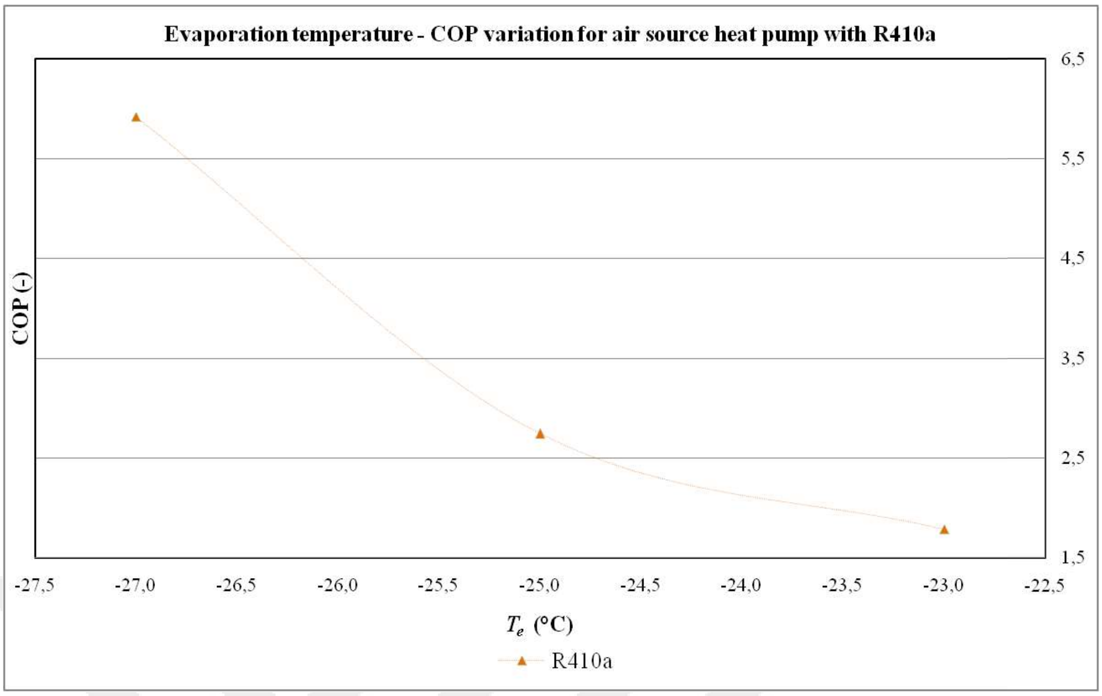

Figure 3.3 T_e - COP variation for air source heat pump with R410a

The change of coefficient of performance according to the evaporation temperature for R410a in the air source heat pump is shown in Figure 3.3.

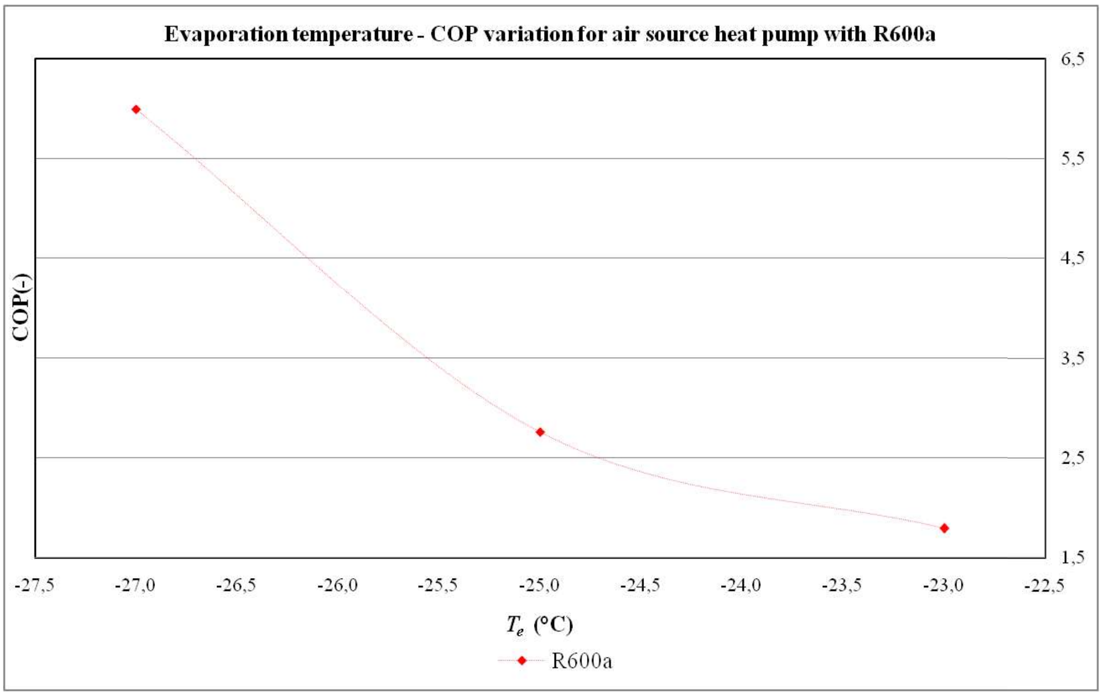

Figure 3.4 T_e - COP variation for air source heat pump with R600a

The change of coefficient of performance according to the evaporation temperature for R600a in the air source heat pump is shown in Figure 3.4.

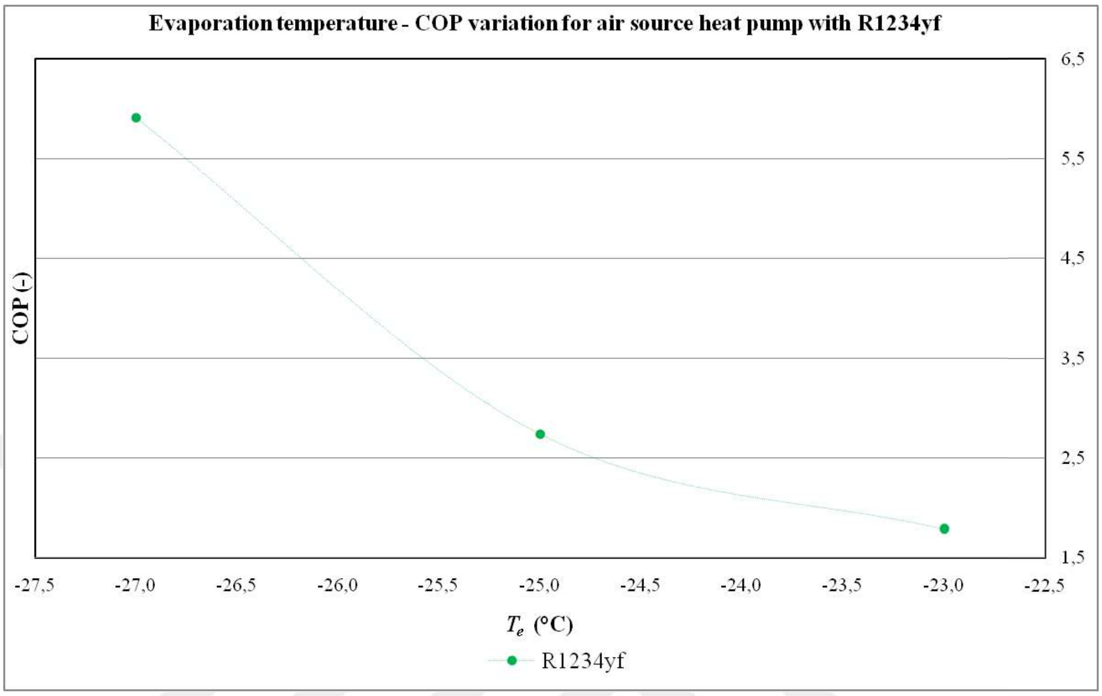

Figure 3.5 T_e - COP variation for air source heat pump with R1234yf

The change of coefficient of performance according to the evaporation temperature for R1234yf in the air source heat pump is shown in Figure 3.5.

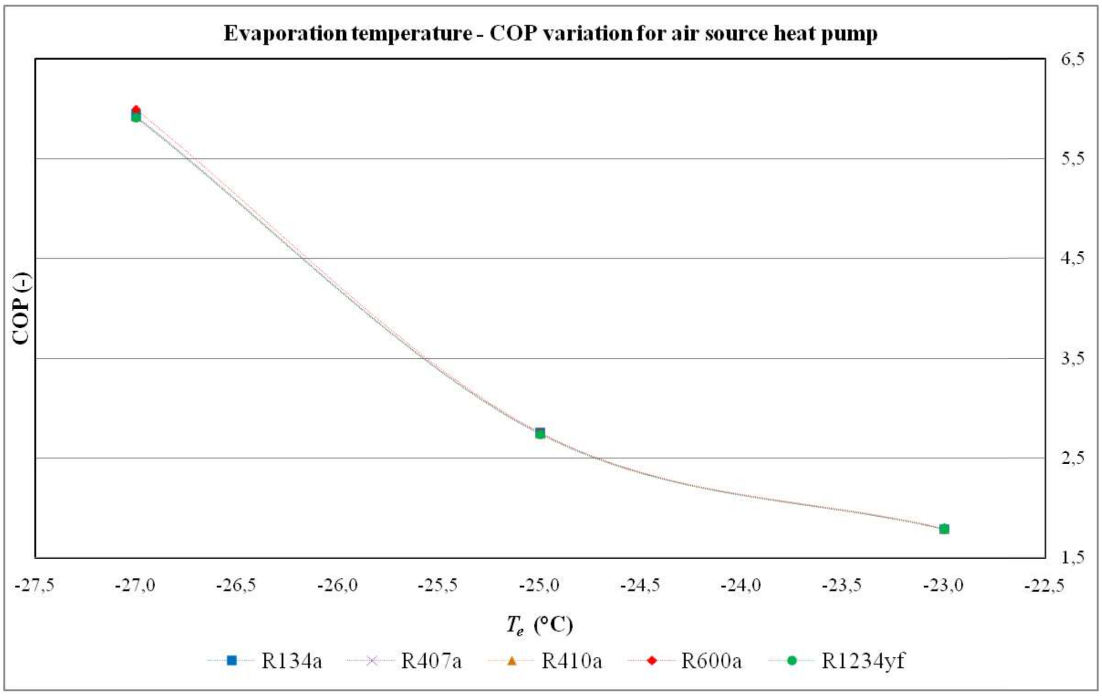

Figure 3.6 T_e - COP variation for air source heat pump

The change of coefficient of performance according to the evaporation temperature for certain refrigerants in the air source heat pump is shown in Figure 3.6. In this graph, R600a refrigerant achieved the highest COP value in the air source heat pump system at all evaporation temperatures. R600a fluid was followed by R134a, R410a, R407a and R1234yf refrigerants, respectively.

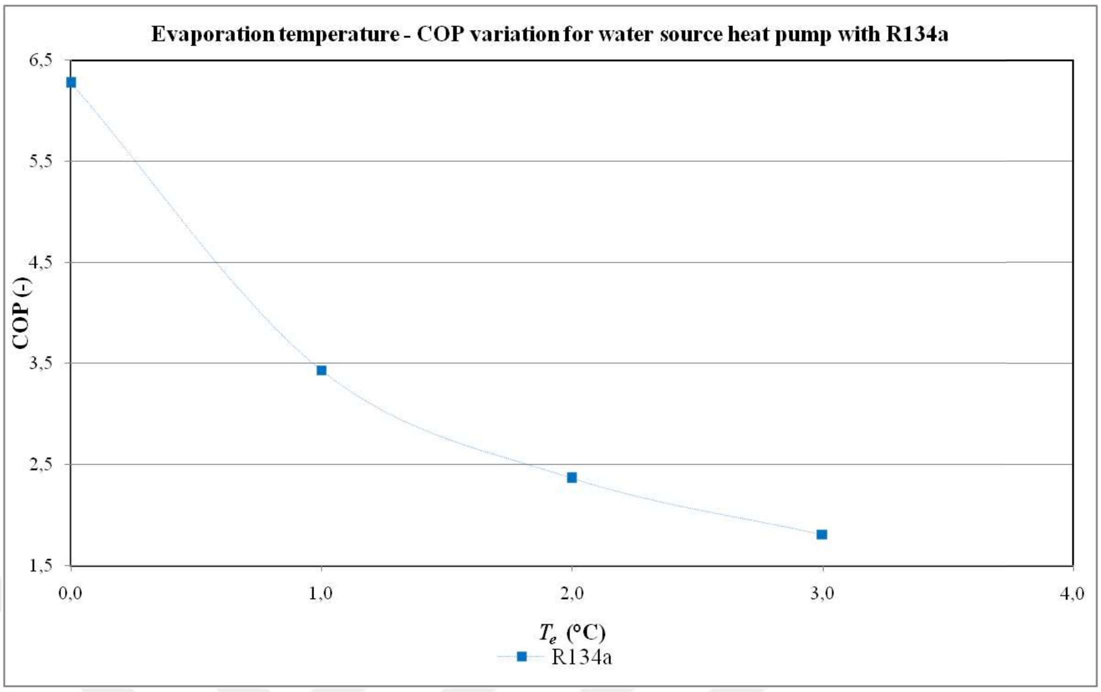

Figure 3.7 T_e - COP variation for water source heat pump with R134a

The change of coefficient of performance according to the evaporation temperature for R134a in the water source heat pump is shown in Figure 3.7.

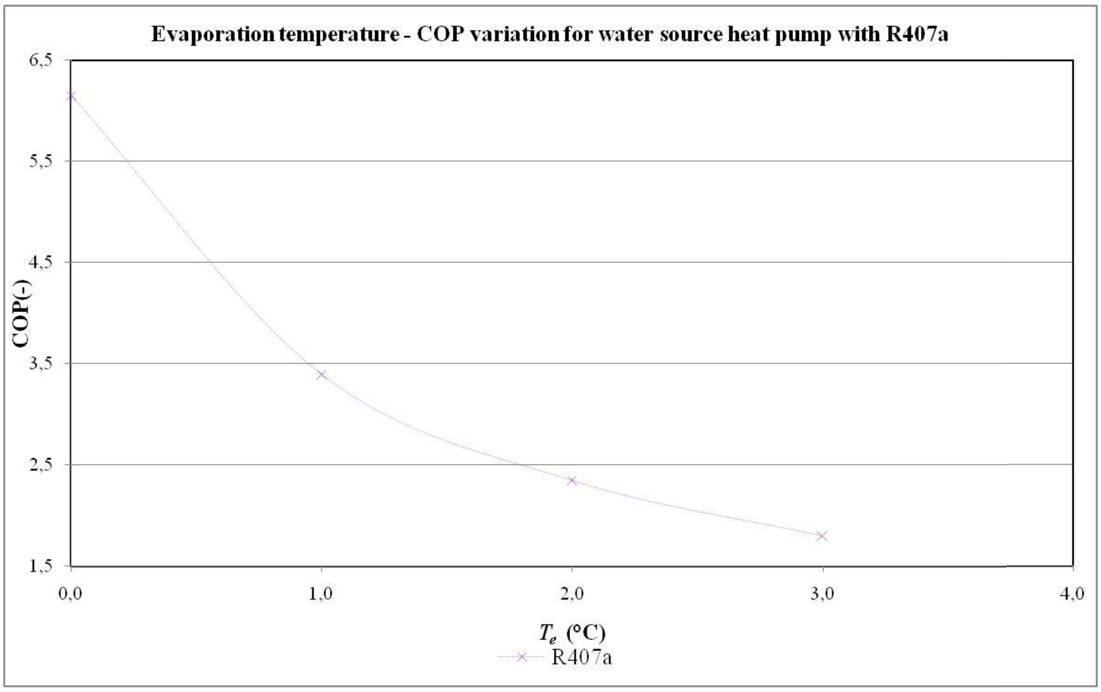

Figure 3.8 T_e - COP variation for water source heat pump with R407a

The change of coefficient of performance according to the evaporation temperature for R407a in the water source heat pump is shown in Figure 3.8.

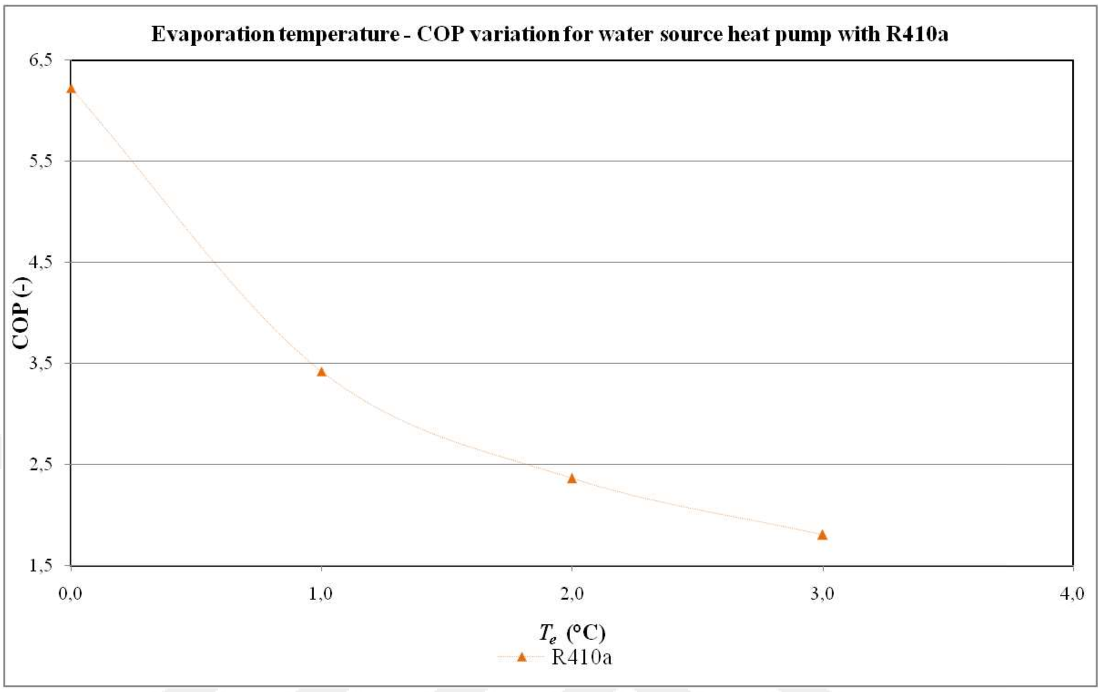

Figure 3.9 T_e - COP variation for water source heat pump with R410a

The change of coefficient of performance according to the evaporation temperature for R410a in the water source heat pump is shown in Figure 3.9.

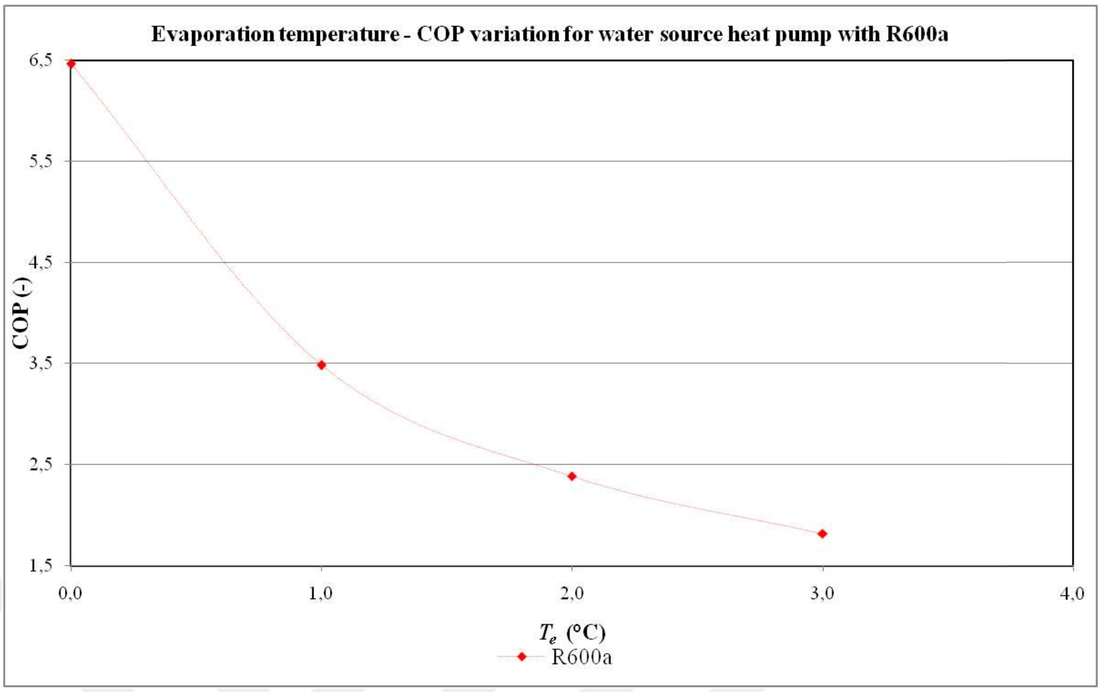

Figure 3.10 T_e - COP variation for water source heat pump with R600a

The change of coefficient of performance according to the evaporation temperature for R600a in the water source heat pump is shown in Figure 3.10.

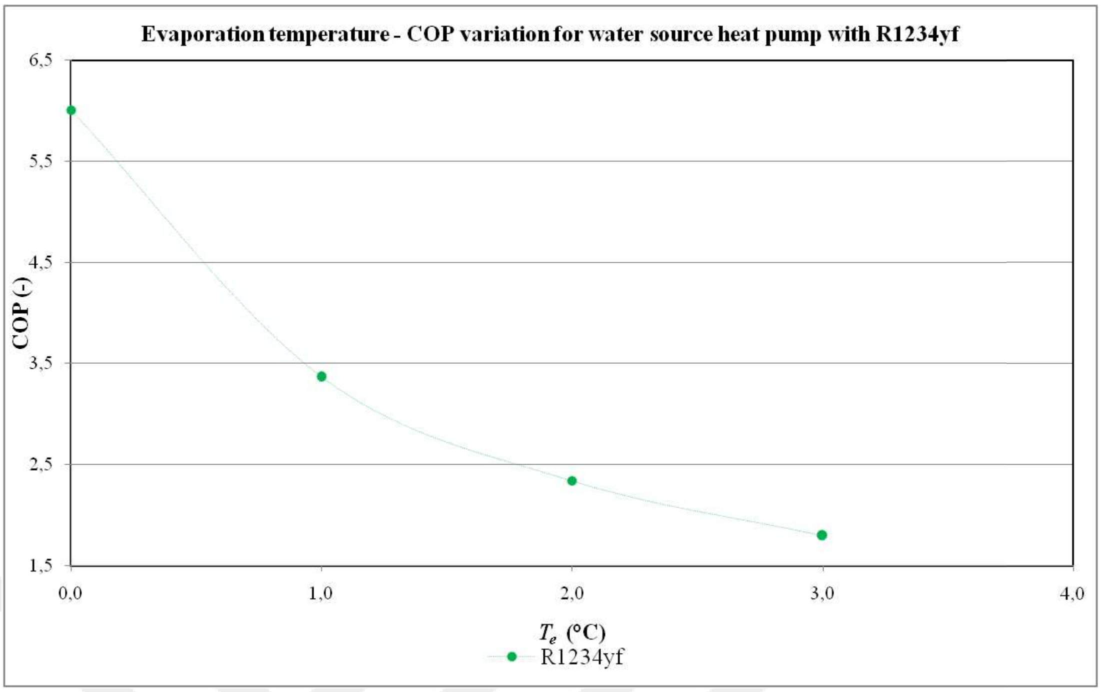

Figure 3.11 T_e - COP variation for water source heat pump with R1234yf

The change of coefficient of performance according to the evaporation temperature for R1234yf in the water source heat pump is shown in Figure 3.11.

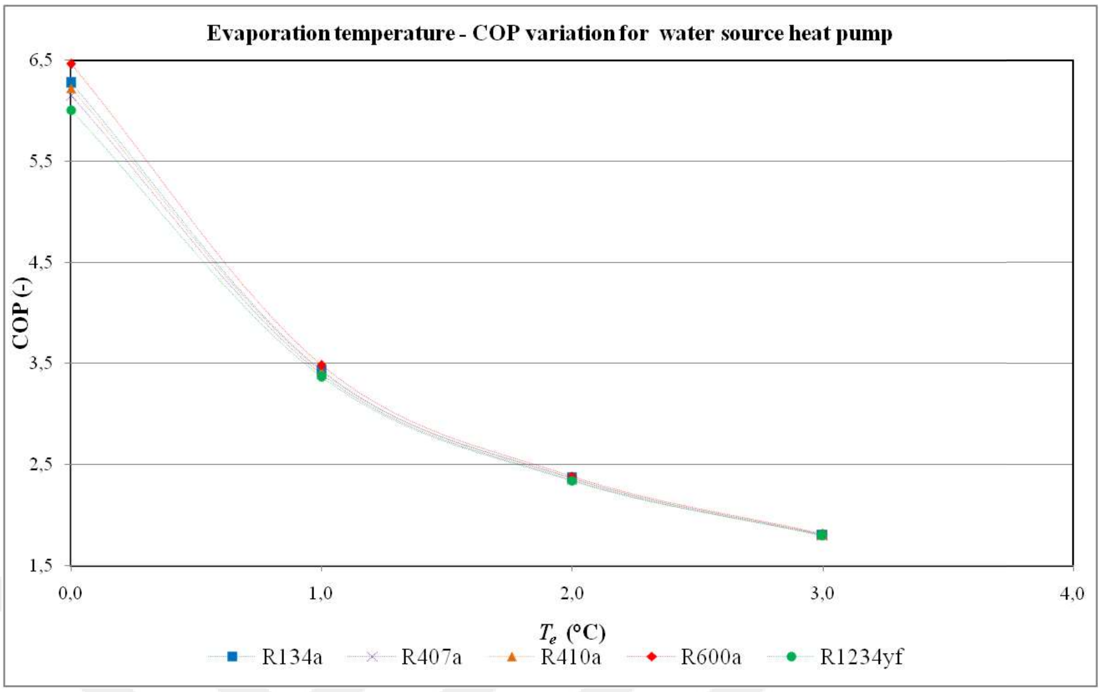

Figure 3.12 T_e - COP variation for water source heat pump

The change of coefficient of performance according to the evaporation temperature for certain refrigerants in the water source heat pump is shown in Figure 3.12. In this graph, R600a refrigerant achieved the highest COP value in the water source heat pump system at all evaporation temperatures. R600a fluid was followed by R134a, R410a, R407a and R1234yf refrigerants, respectively.

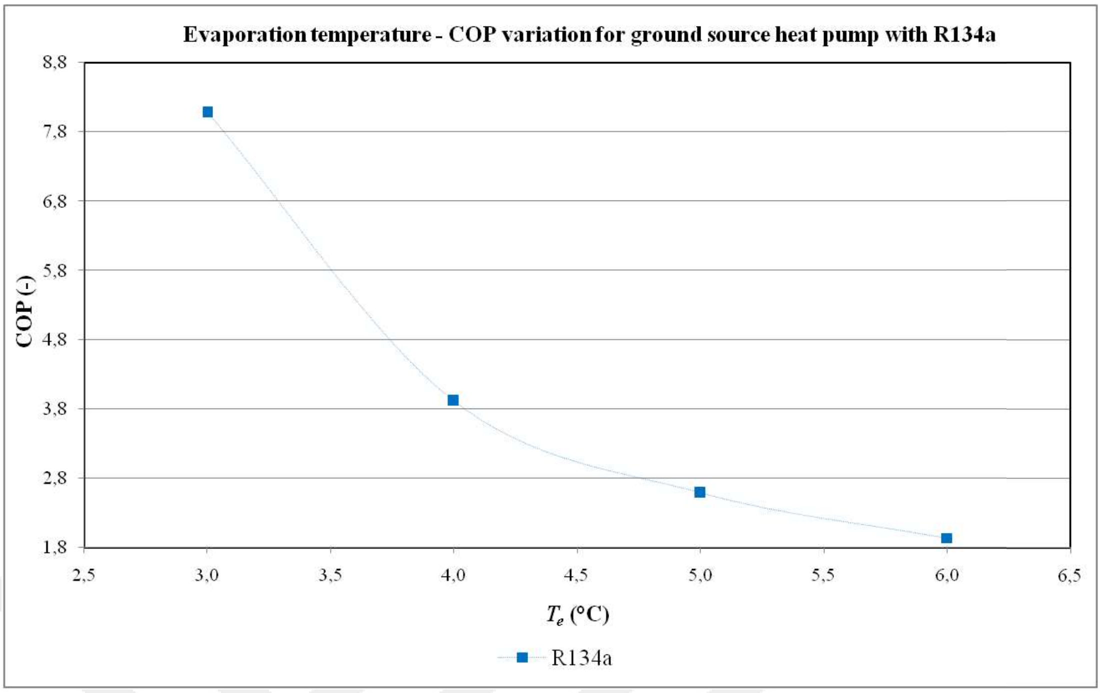

Figure 3.13 T_e - COP variation for ground source heat pump with R134a

The change of coefficient of performance according to the evaporation temperature for R134a in the ground source heat pump is shown in Figure 3.13.

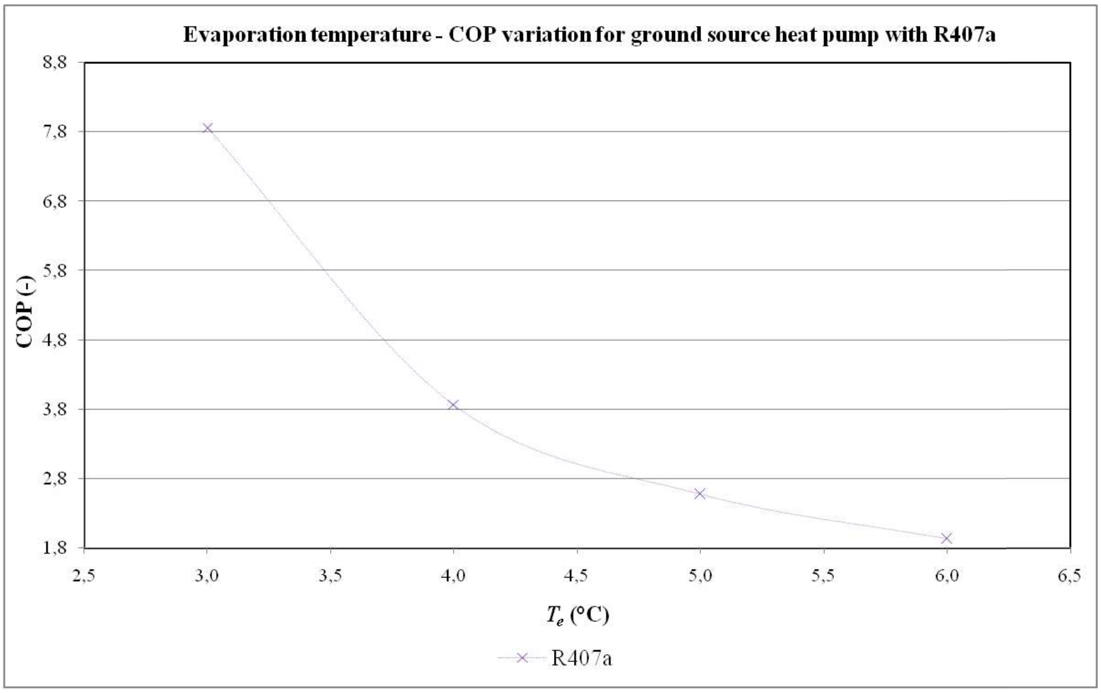

Figure 3.14 T_e - COP variation for ground source heat pump with R407a

The change of coefficient of performance according to the evaporation temperature for R407a in the ground source heat pump is shown in Figure 3.14.

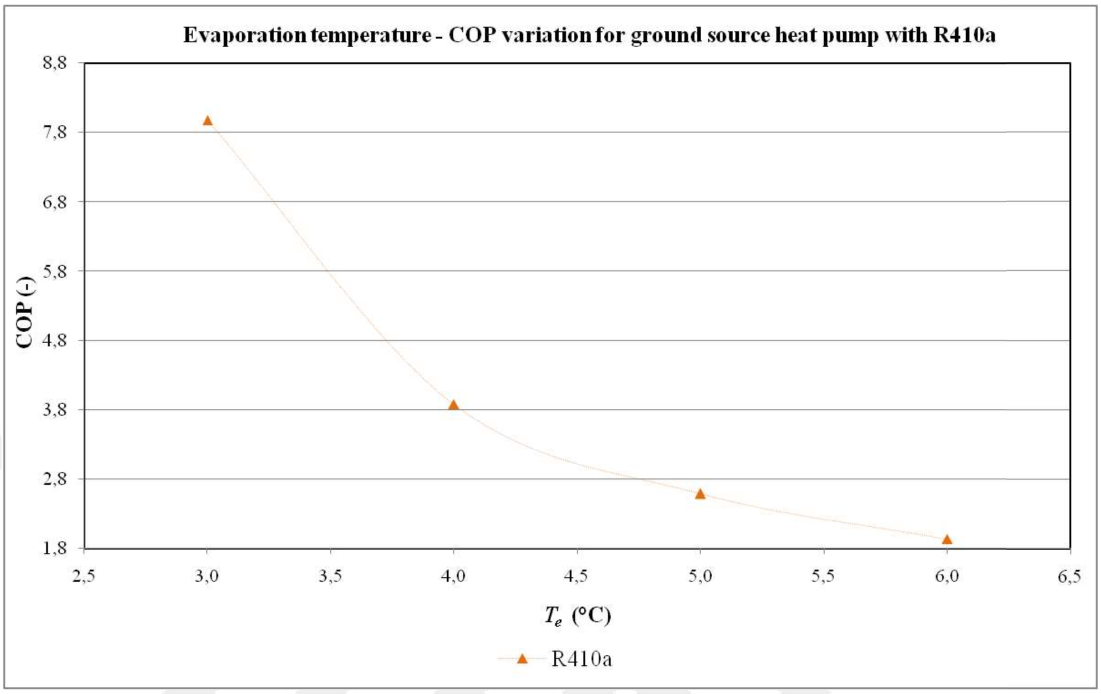

Figure 3.15 T_e - COP variation for ground source heat pump with R410a

The change of coefficient of performance according to the evaporation temperature for R410a in the ground source heat pump is shown in Figure 3.15.

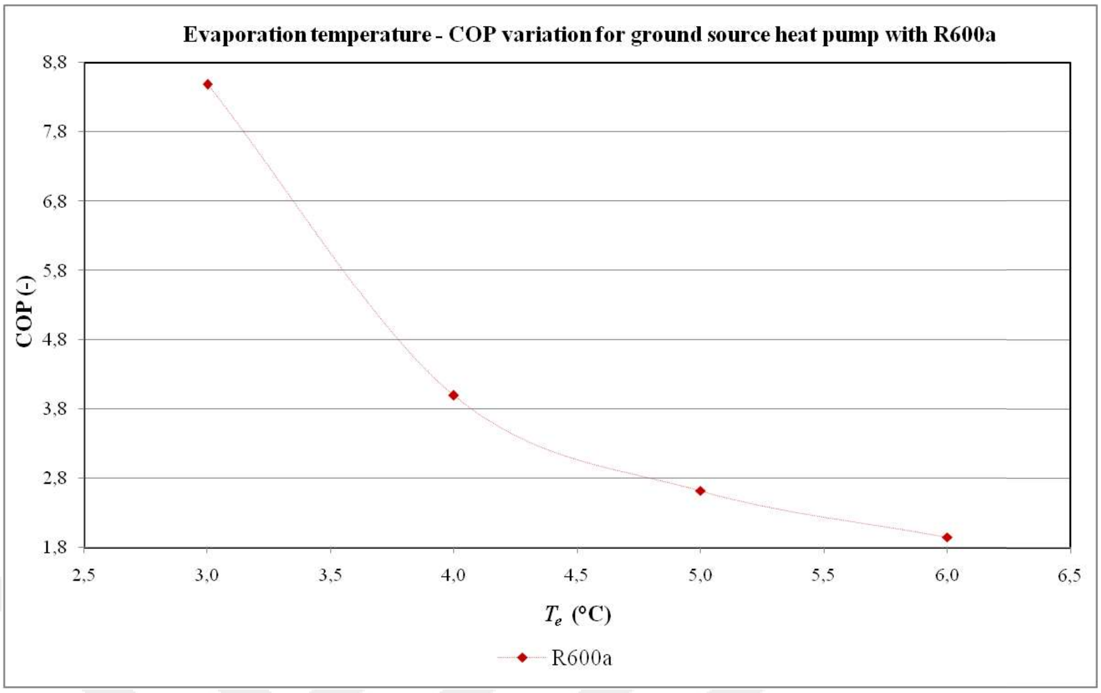

Figure 3.16 T_e - COP variation for ground source heat pump with R600a

The change of coefficient of performance according to the evaporation temperature for R600a in the ground source heat pump is shown in Figure 3.16.

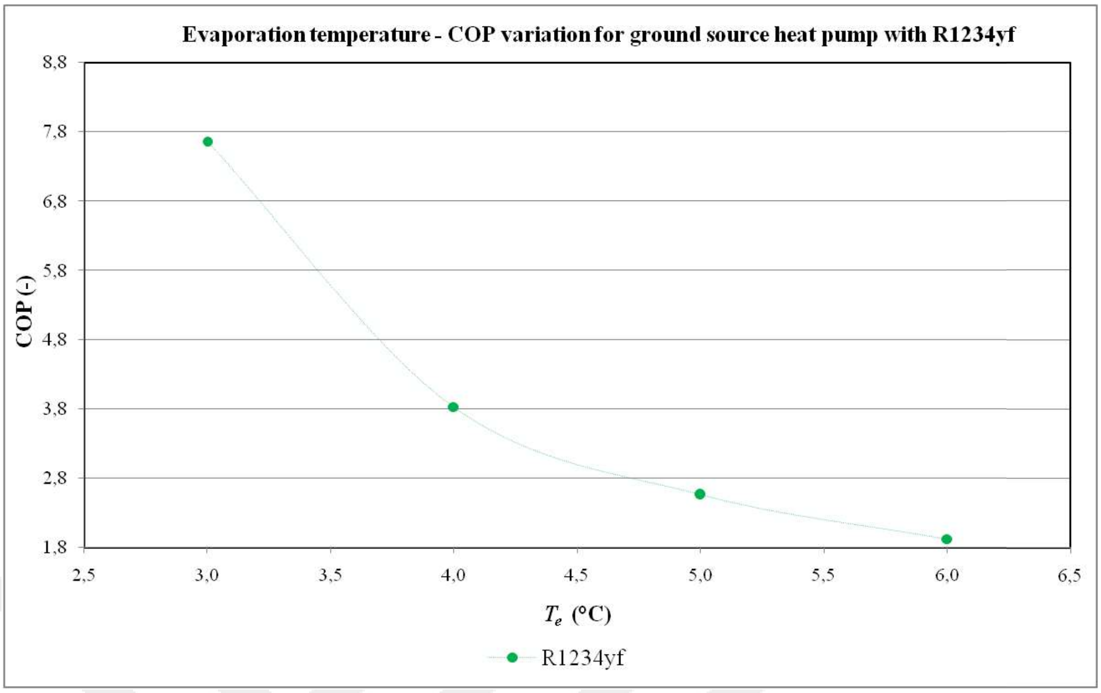

Figure 3.17 T_e - COP variation for ground source heat pump with R1234yf

The change of coefficient of performance according to the evaporation temperature for R1234yf in the ground source heat pump is shown in Figure 3.17.

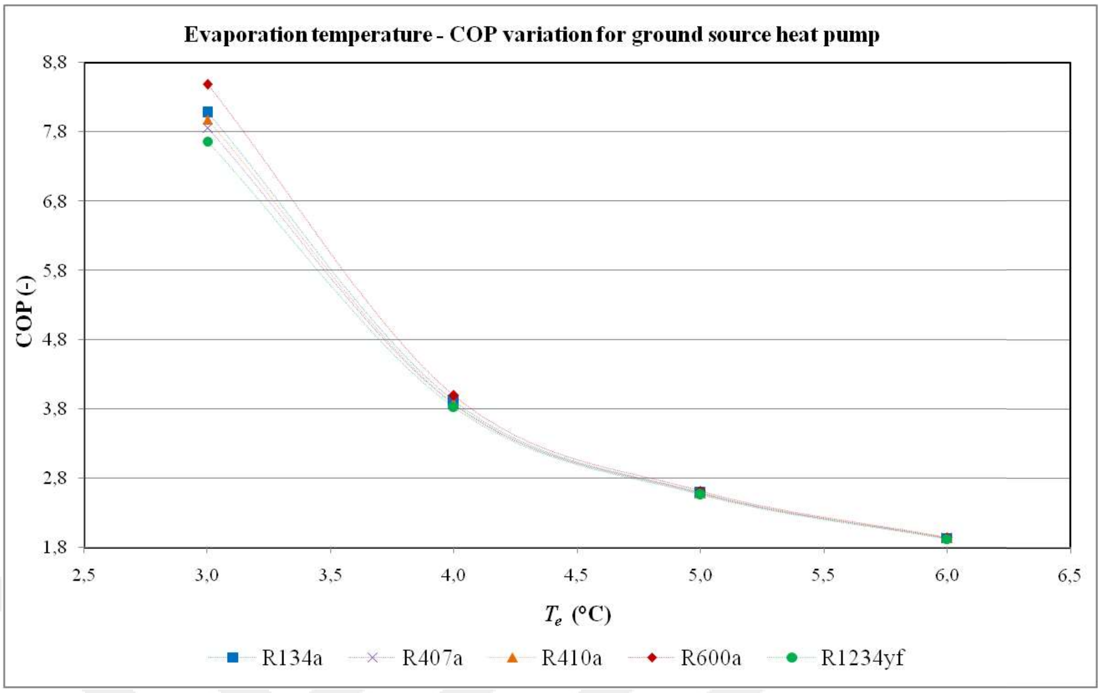

Figure 3.18 T_e - COP variation for ground source heat pump

The change of coefficient of performance according to the evaporation temperature for certain refrigerants in the ground source heat pump is shown in Figure 3.18. In this graph, R600a refrigerant achieved the highest COP value in the ground source heat pump system at all evaporation temperatures. R600a fluid was followed by R134a, R410a, R407a and R1234yf refrigerants, respectively.

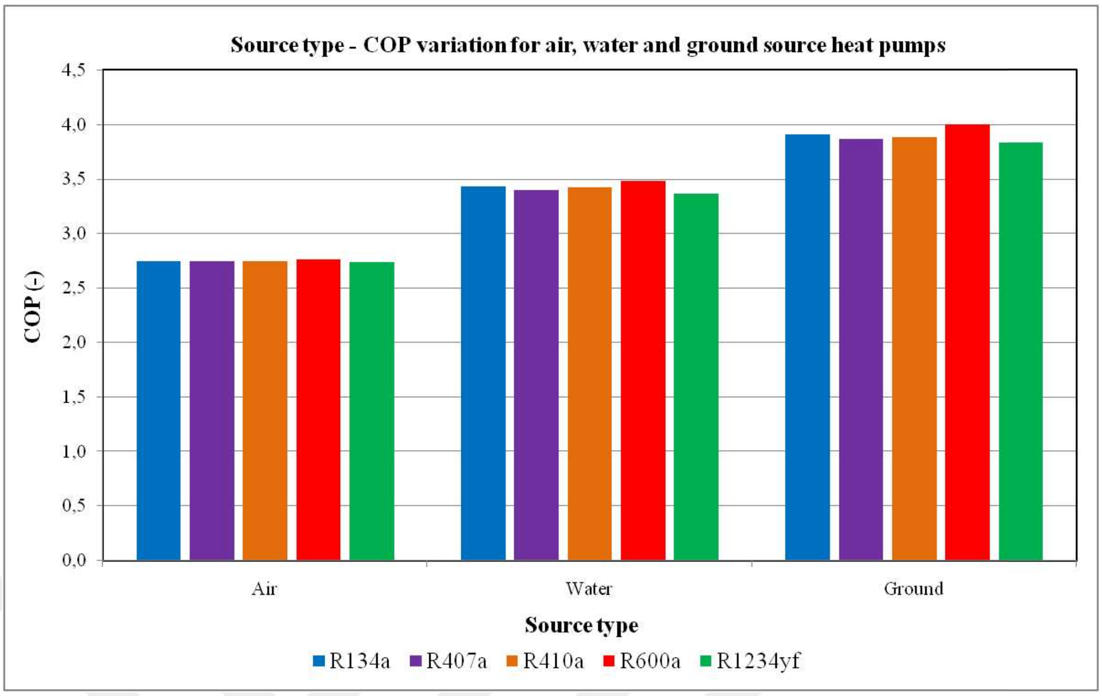

Figure 3.19 Source type - COP variation for air, water and ground source heat pumps

The comparison chart for the air, water and ground source heat pump Gölbaşı district is shown in Figure 3.19. According to this chart, the ground source heat pump achieved the highest COP value for Gölbaşı district.

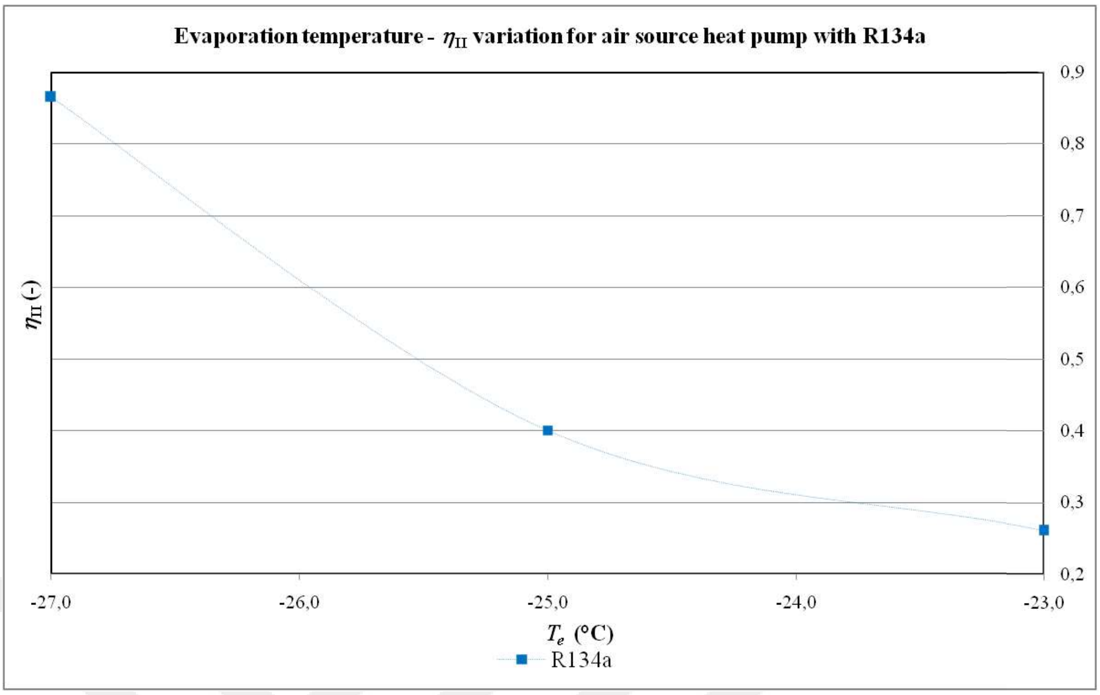

Figure 3.20 T_e - η_{II} variation for air source heat pump with R134a

The variation of exergy efficiency according to the evaporation temperature for R134a in the air source heat pump is shown in Figure 3.20.

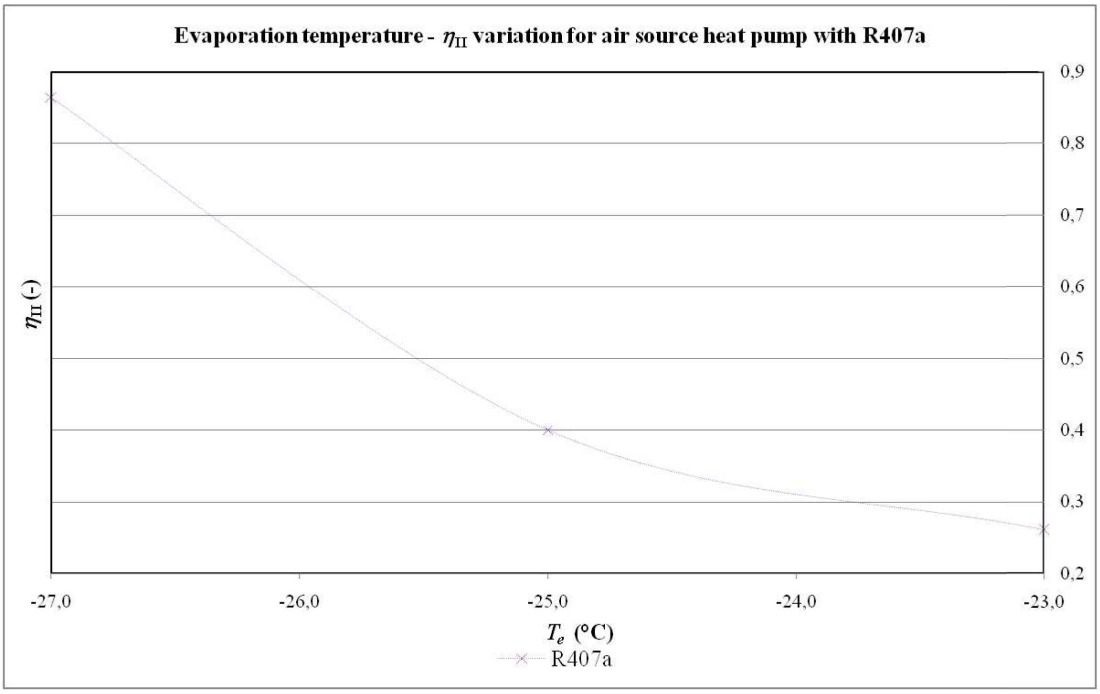

Figure 3.21 T_e - η_{II} variation for air source heat pump with R407a

The variation of exergy efficiency according to the evaporation temperature for R407a in the air source heat pump is shown in Figure 3.21.

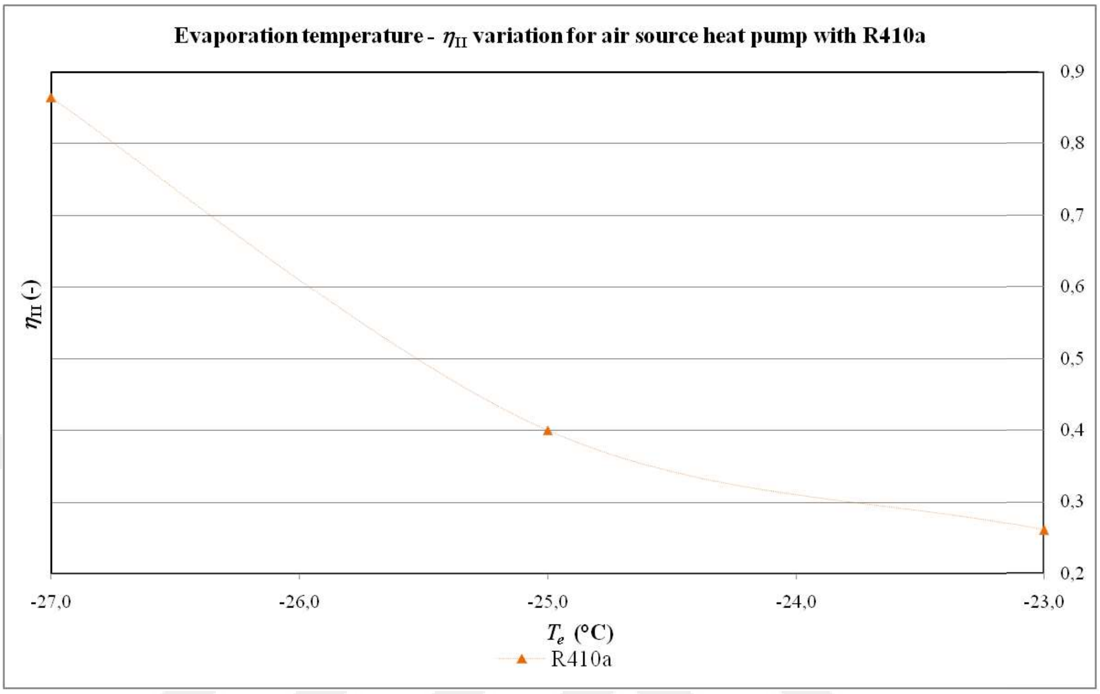

Figure 3.22 T_e - η_{II} variation for air source heat pump with R410a

The variation of exergy efficiency according to the evaporation temperature for R410a in the air source heat pump is shown in Figure 3.22.

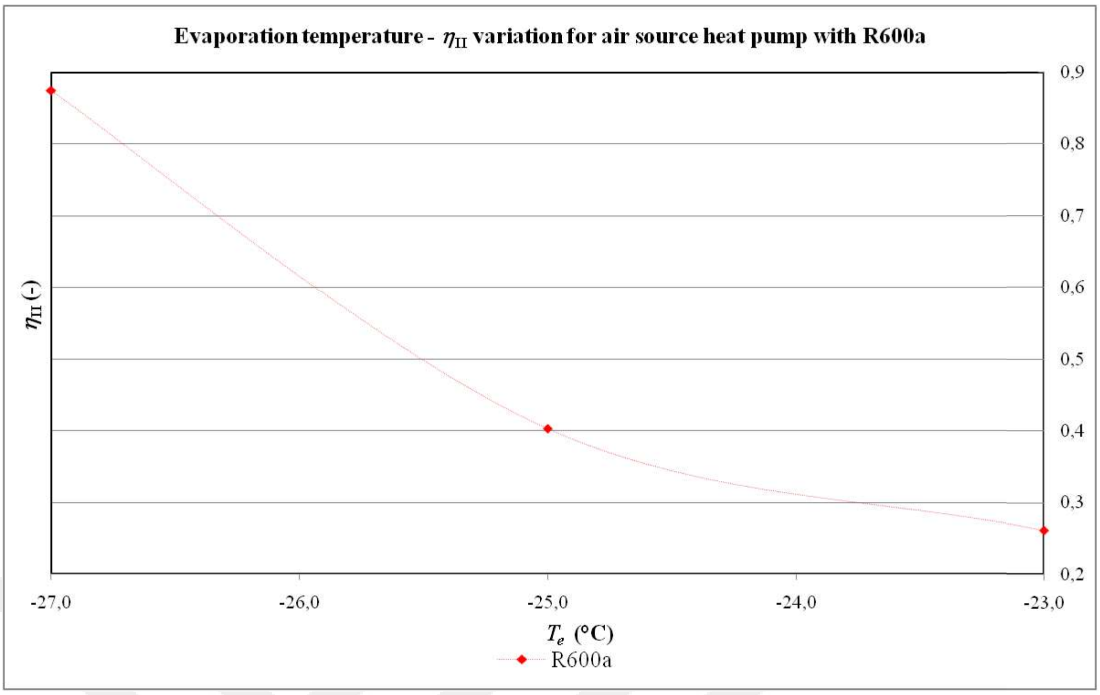

Figure 3.23 T_e - η_{II} variation for air source heat pump with R600a

The variation of exergy efficiency according to the evaporation temperature for R600a in the air source heat pump is shown in Figure 3.23.

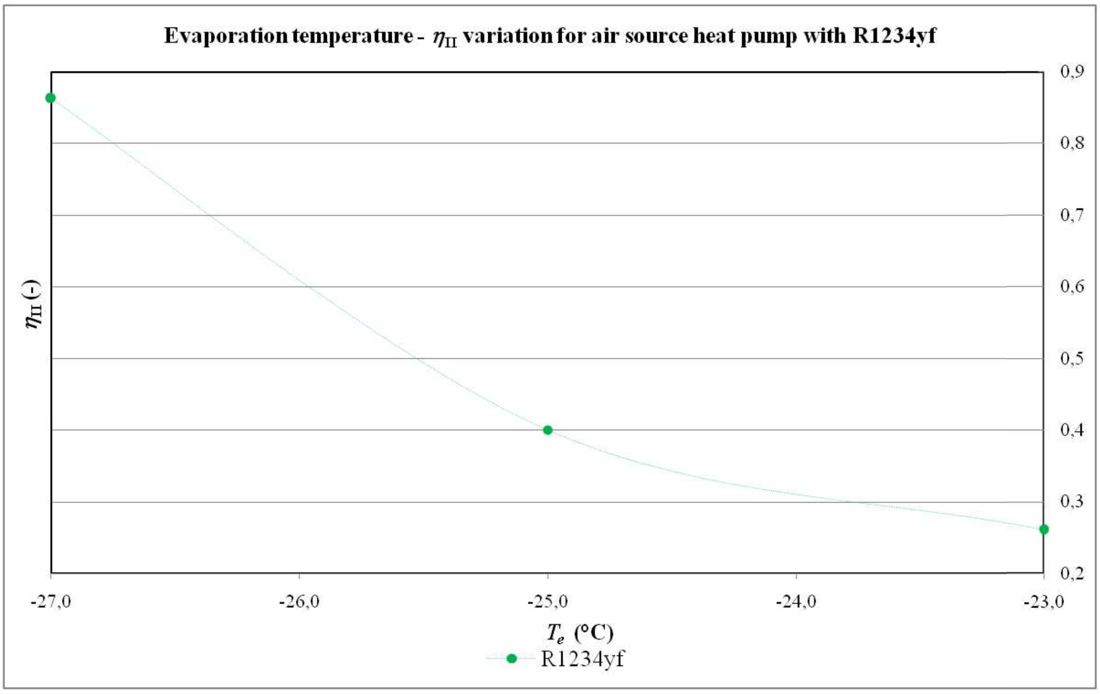

Figure 3.24 T_e - η_{II} variation for air source heat pump with R1234yf

The variation of exergy efficiency according to the evaporation temperature for R1234yf in the air source heat pump is shown in Figure 3.24.

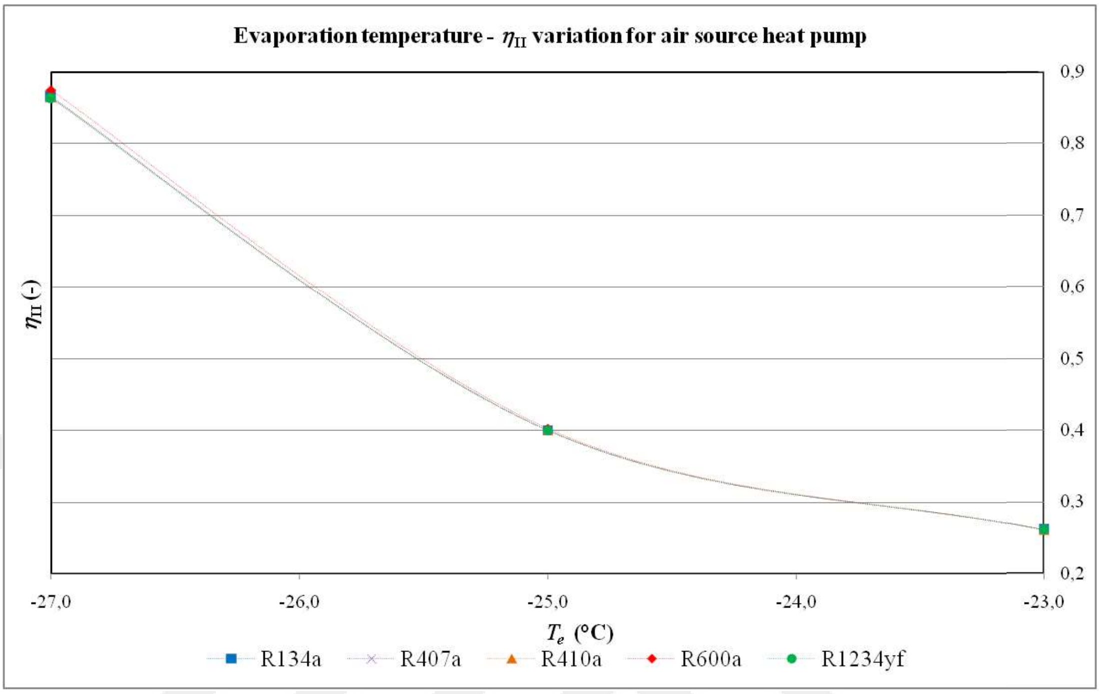

Figure 3.25 T_e - η_{II} variation for air source heat pump

The variation of exergy efficiency according to the evaporation temperature for certain refrigerants in the air source heat pump is shown in Figure 3.25. In this graph, R600a refrigerant achieved the highest η_{II} value in the air source heat pump system at all evaporation temperatures. R600a fluid was followed by R134a, R410a, R407a and R1234yf refrigerants, respectively.

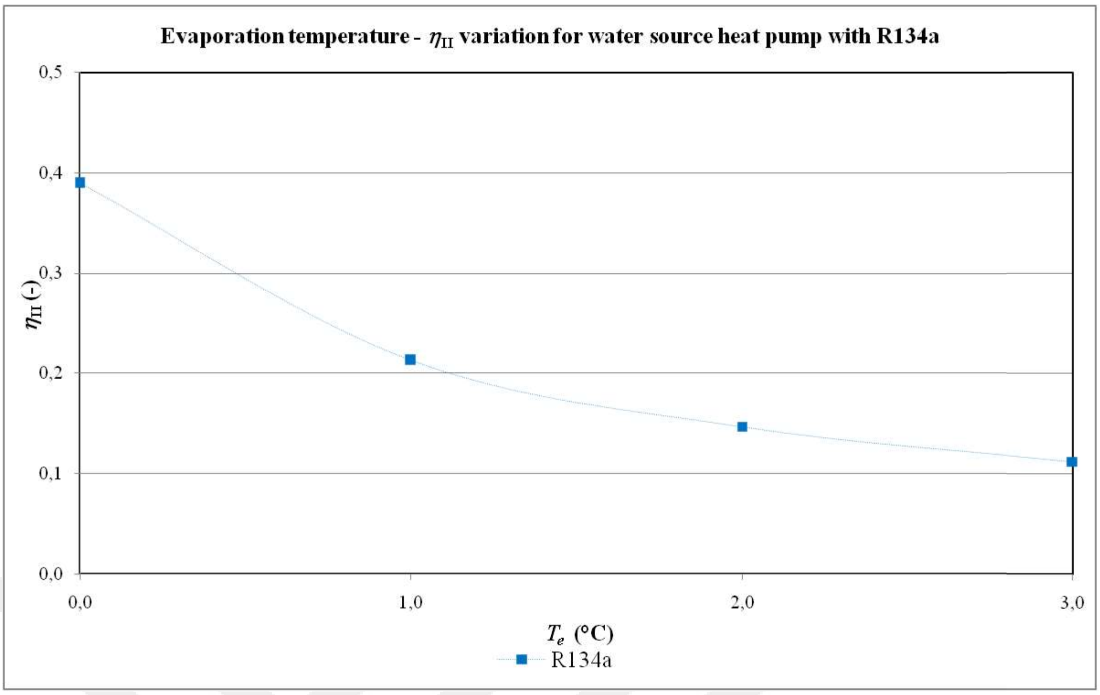

Figure 3.26 T_e - η_{II} variation for water source heat pump with R134a

The variation of exergy efficiency according to the evaporation temperature for R134a in the water source heat pump is shown in Figure 3.26.

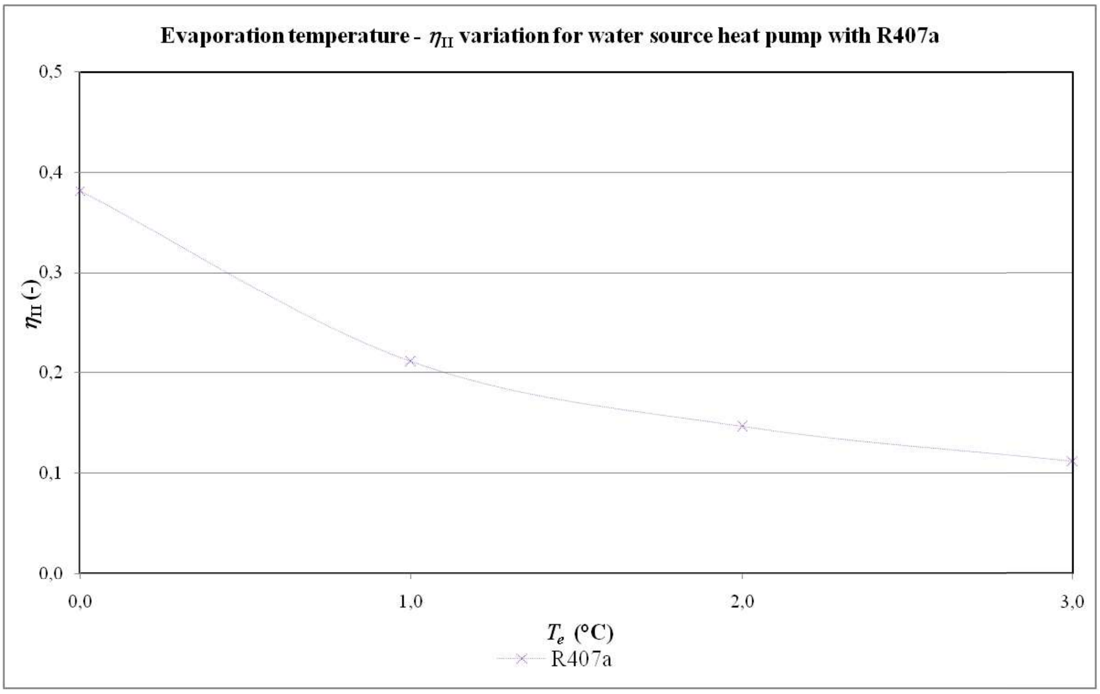

Figure 3.27 T_e - η_{II} variation for water source heat pump with R407a

The variation of exergy efficiency according to the evaporation temperature for R407a in the water source heat pump is shown in Figure 3.27.

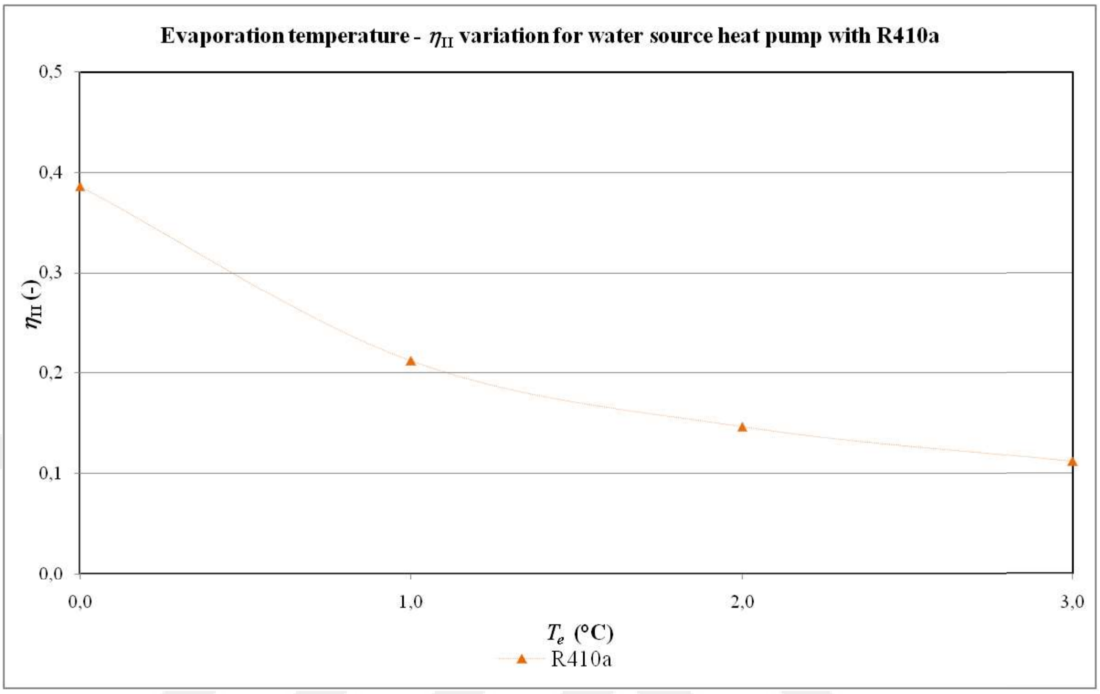

Figure 3.28 T_e - η_{II} variation for water source heat pump with R410a

The variation of exergy efficiency according to the evaporation temperature for R410a in the water source heat pump is shown in Figure 3.28.

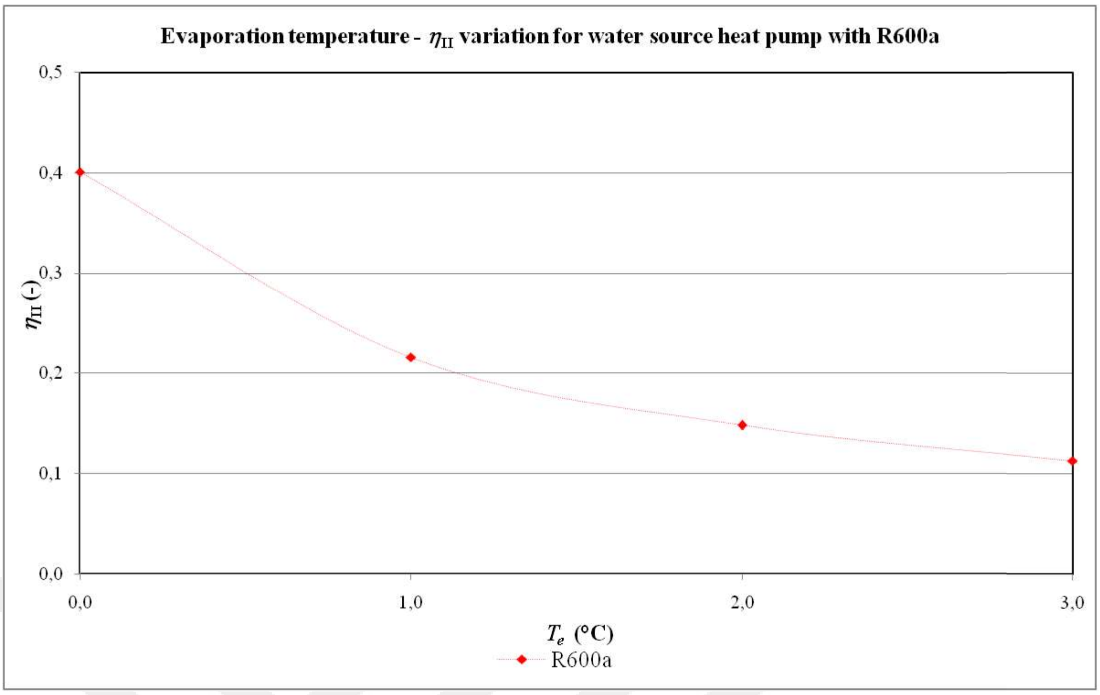

Figure 3.29 T_e - η_{II} variation for water source heat pump with R600a

The variation of exergy efficiency according to the evaporation temperature for R600a in the water source heat pump is shown in Figure 3.29.

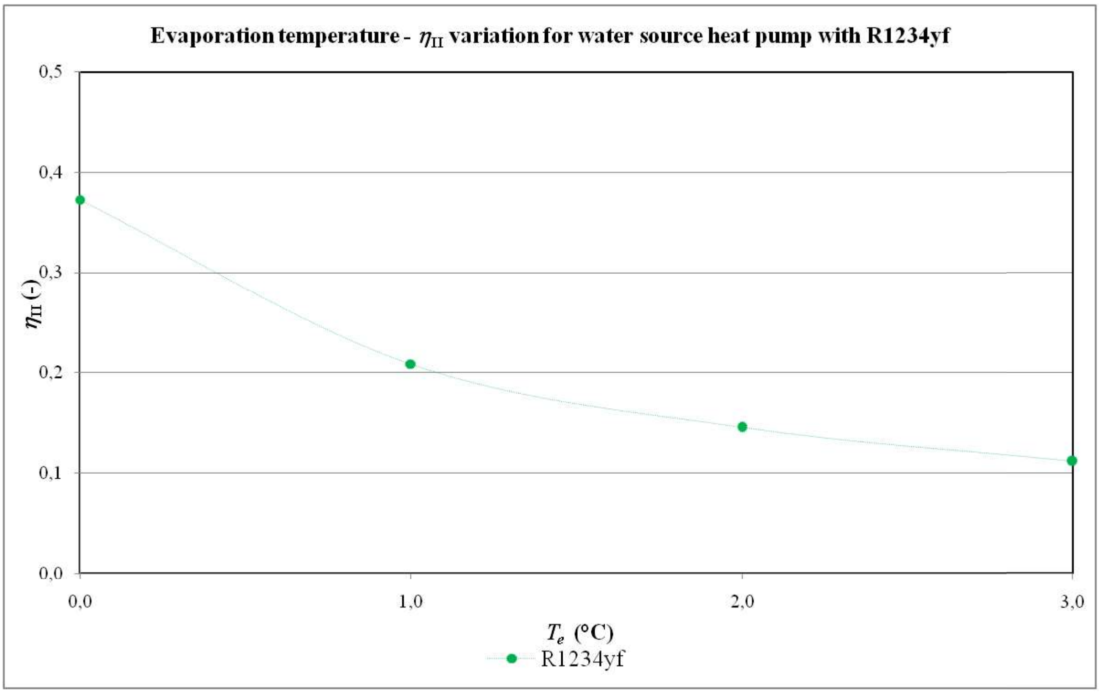

Figure 3.30 T_e - η_{II} variation for water source heat pump with R1234yf

The variation of exergy efficiency according to the evaporation temperature for R1234yf in the water source heat pump is shown in Figure 3.30.

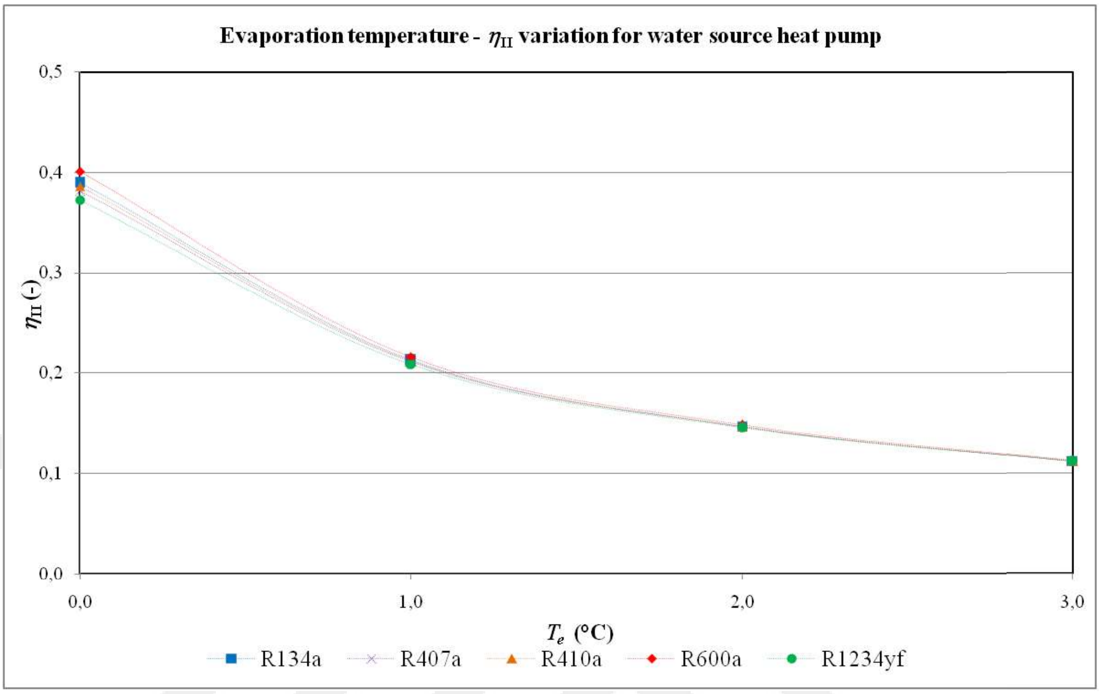

Figure 3.31 T_e - η_{II} variation for water source heat pump

The variation of exergy efficiency according to the evaporation temperature for certain refrigerants in the water source heat pump is shown in Figure 3.31. In this graph, R600a refrigerant achieved the highest η_{II} value in the water source heat pump system at all evaporation temperatures. R600a fluid was followed by R134a, R410a, R407a and R1234yf refrigerants, respectively.

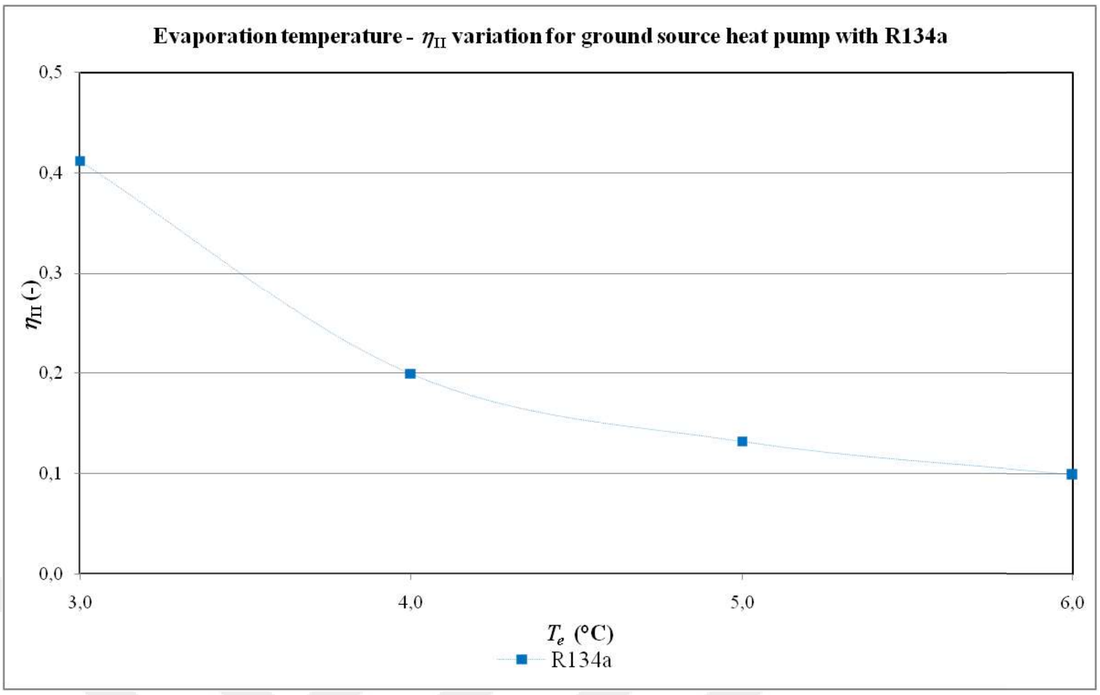

Figure 3.32 T_e - η_{II} variation for ground source heat pump with R134a

The variation of exergy efficiency according to the evaporation temperature for R134a in the ground source heat pump is shown in Figure 3.32.

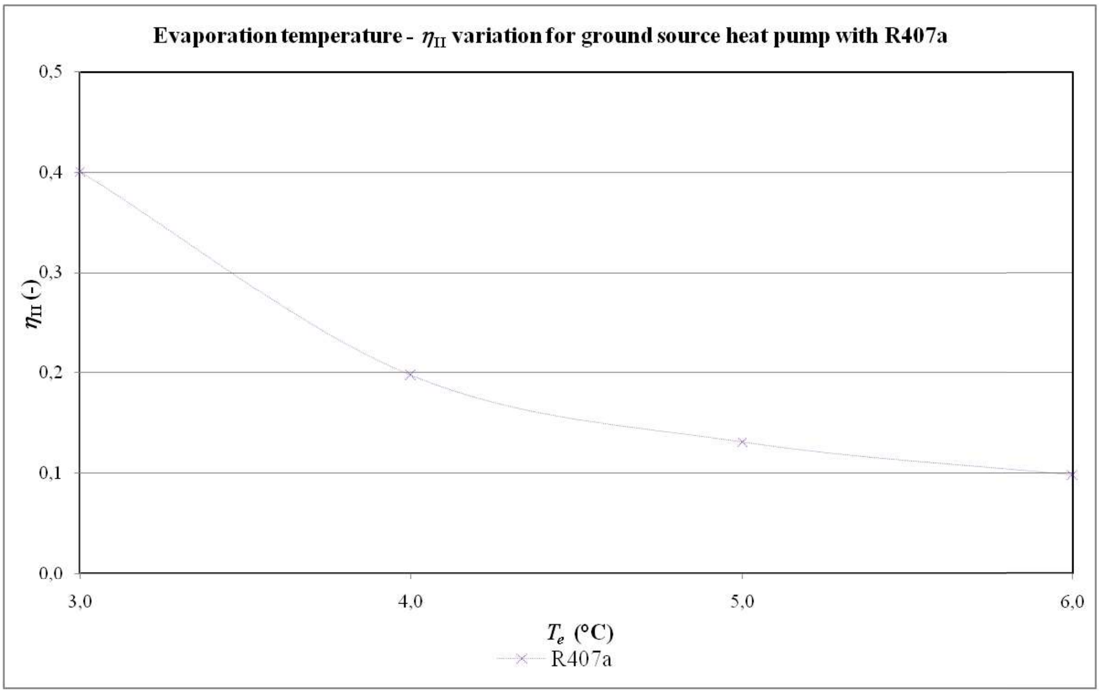

Figure 3.33 T_e - η_{II} variation for ground source heat pump with R407a

The variation of exergy efficiency according to the evaporation temperature for R407a in the ground source heat pump is shown in Figure 3.33.

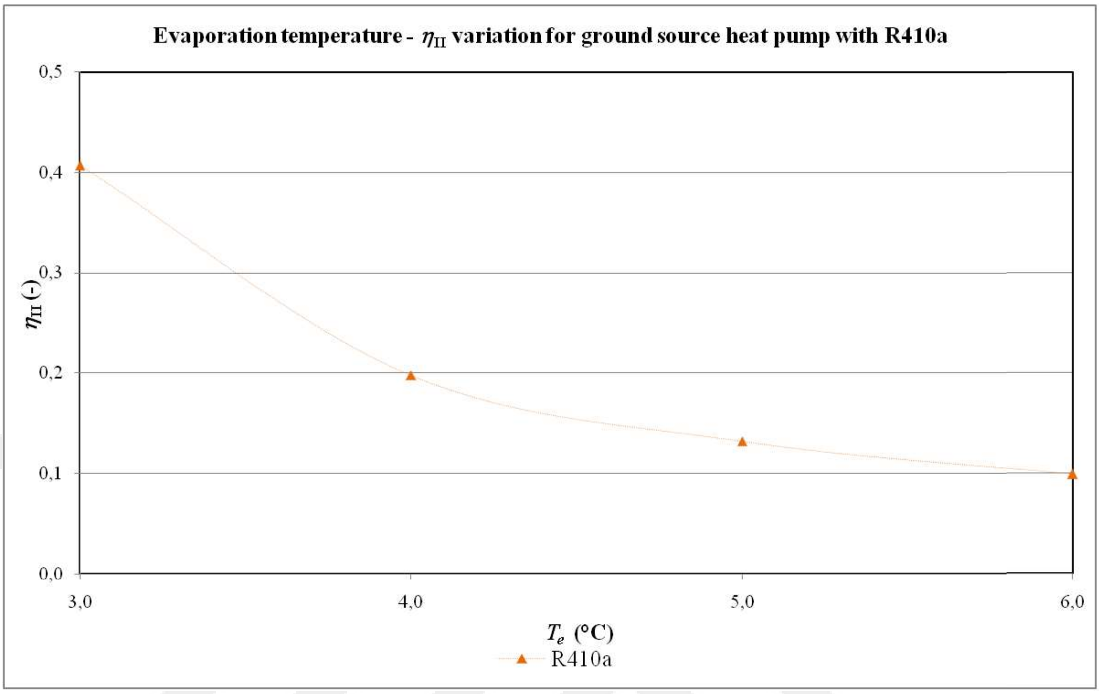

Figure 3.34 T_e - η_{II} variation for ground source heat pump with R410a

The variation of exergy efficiency according to the evaporation temperature for R410a in the ground source heat pump is shown in Figure 3.34.

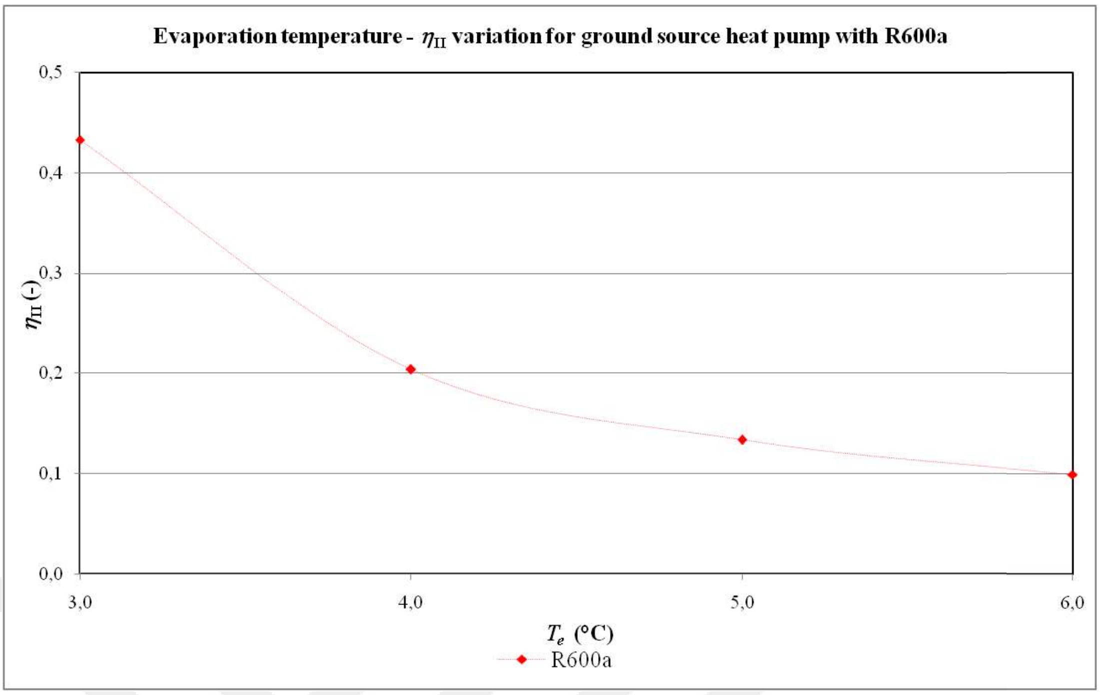

Figure 3.35 T_e - η_{II} variation for ground source heat pump with R600a

The variation of exergy efficiency according to the evaporation temperature for R600a in the ground source heat pump is shown in Figure 3.35.

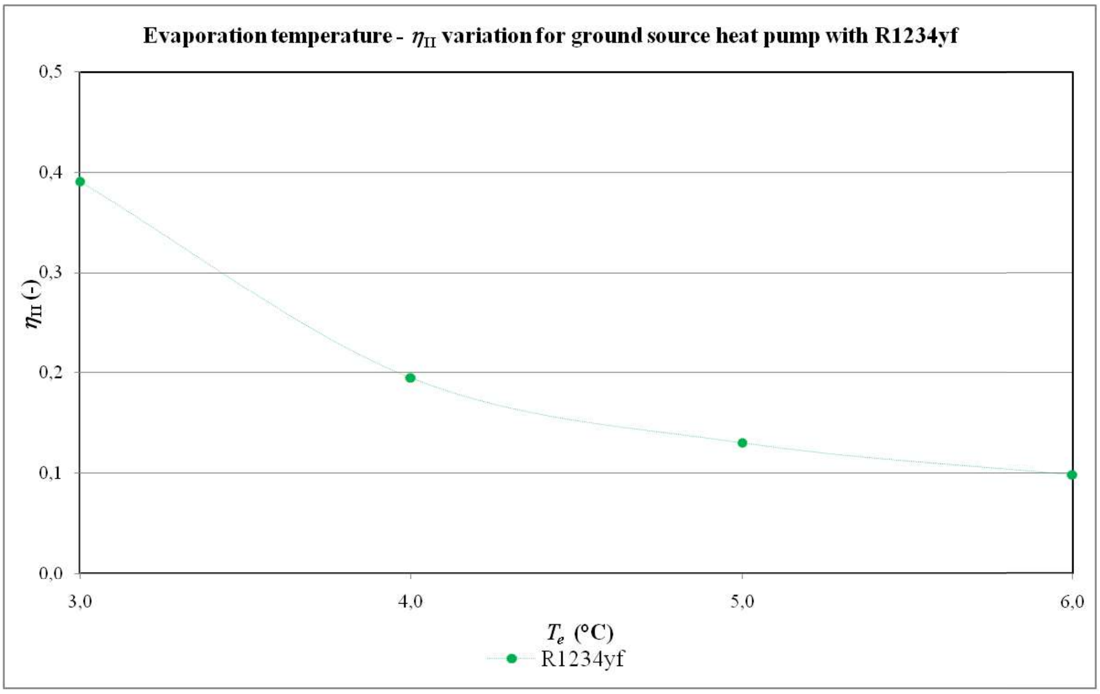

Figure 3.36 T_e - η_{II} variation for ground source heat pump with R1234yf

The variation of exergy efficiency according to the evaporation temperature for R1234yf in the ground source heat pump is shown in Figure 3.36.

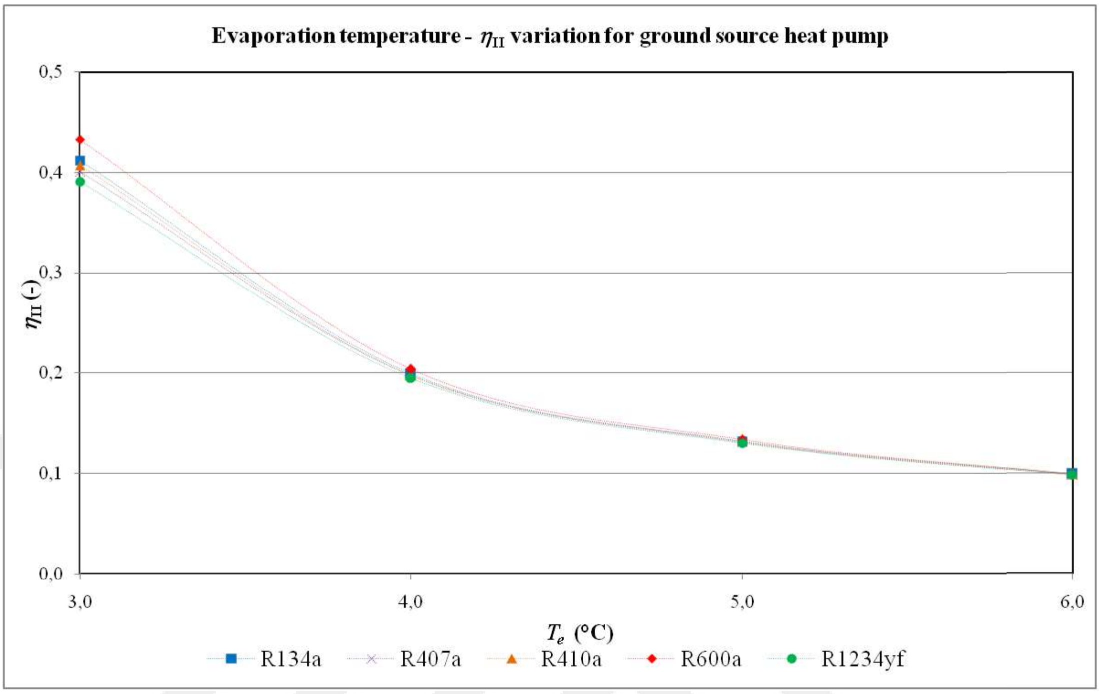

Figure 3.37 T_e - η_{II} variation for ground source heat pump

The variation of exergy efficiency according to the evaporation temperature for certain refrigerants in the ground source heat pump is shown in Figure 3.37. In this graph, R600a refrigerant achieved the highest η_{II} value in the ground source heat pump system at all evaporation temperatures. R600a fluid was followed by R134a, R410a, R407a and R1234yf refrigerants, respectively.

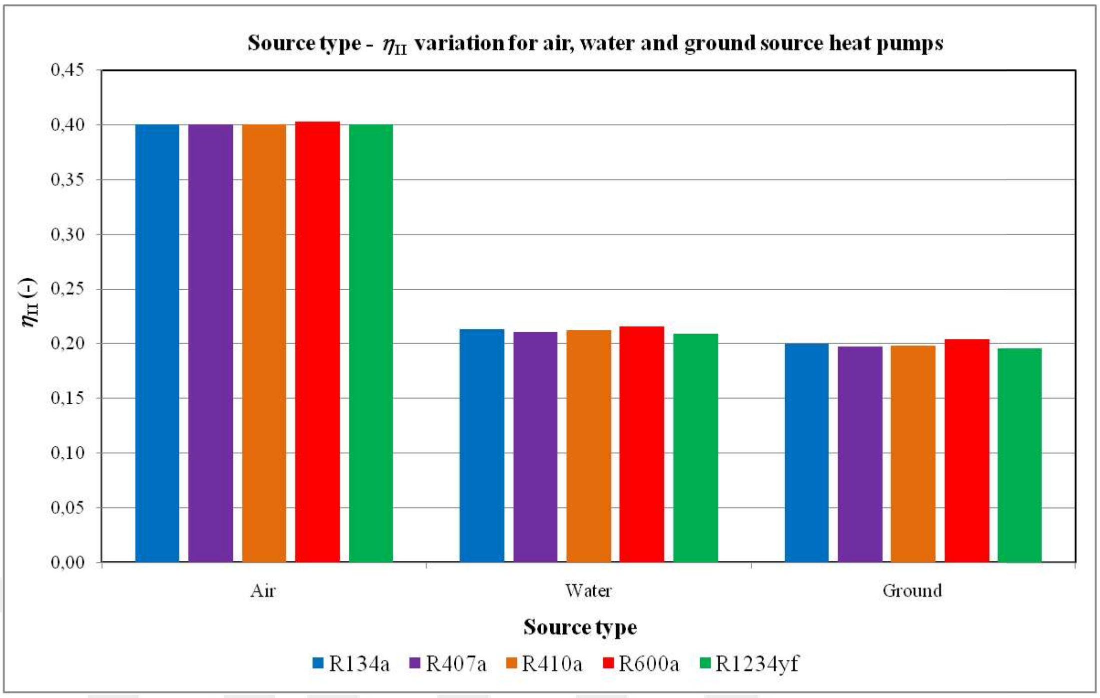

Figure 3.38 Source type - η_{II} variation for air, water and ground source

The comparison chart for the air, water and ground source heat pump Gölbaşı district is shown in Figure 3.38. According to this chart, the air source heat pump achieved the highest η_{II} value for Gölbaşı district.

3.2 Results for Cooling Mode

In addition in this theoretical study, the first and second law of thermodynamics of air, water and ground heat pumps was used for cooling purposes for a house which needs 4 kW of cooling in Gölbaşı district of Ankara province. The highest air, water and ground temperature values seen in the last 10 years in Gölbaşı district of Ankara province were obtained from the necessary institutions. The temperature values required for air and ground source heat pump were obtained from the Turkish State Meteorological Service. The temperature value required for water source heat pump was taken from the General Directorate of State Hydraulic Works. According to these data, the highest air temperature seen in the last 10 years was 39 °C, the highest water temperature was 24°C and the highest ground temperature seen was 27.6 °C, based on Mogan Lake in Gölbaşı district. R600a, R134a, R410a, R407a and R1234yf were used separately as refrigerant in each heat pump system and the mass flow rate of the fluids were calculated as 0.01674 kg/s, 0.03294 kg/s, 0.036258 kg/s, 0.03691 kg/s and 0.04095 kg/s respectively. In the calculations, some tables were used for the thermodynamic properties of fluids. Genetron Properties program was used for these tables. Temperature, pressure, specific enthalpy, entropy values for the compressor, condenser, throttling valve and evaporator, which are the basic elements of the heat pump system, were calculated using the assumptions shown in Chapter 2, Kandlikar's correlation equation and the first and second law of thermodynamics. Then, the coefficient of performance of the air, water and ground source heat pump and the second law efficiency of the system were calculated for cooling mode. These values were calculated and compared separately for R600a, R134a, R410a, R407a and R1234yf refrigerants in each heat pump (air, water and ground source).

Graphical representations are given to better understand the results of the heat pump energy and exergy analysis for cooling mode according to the heat pump type and the type of refrigerant used.

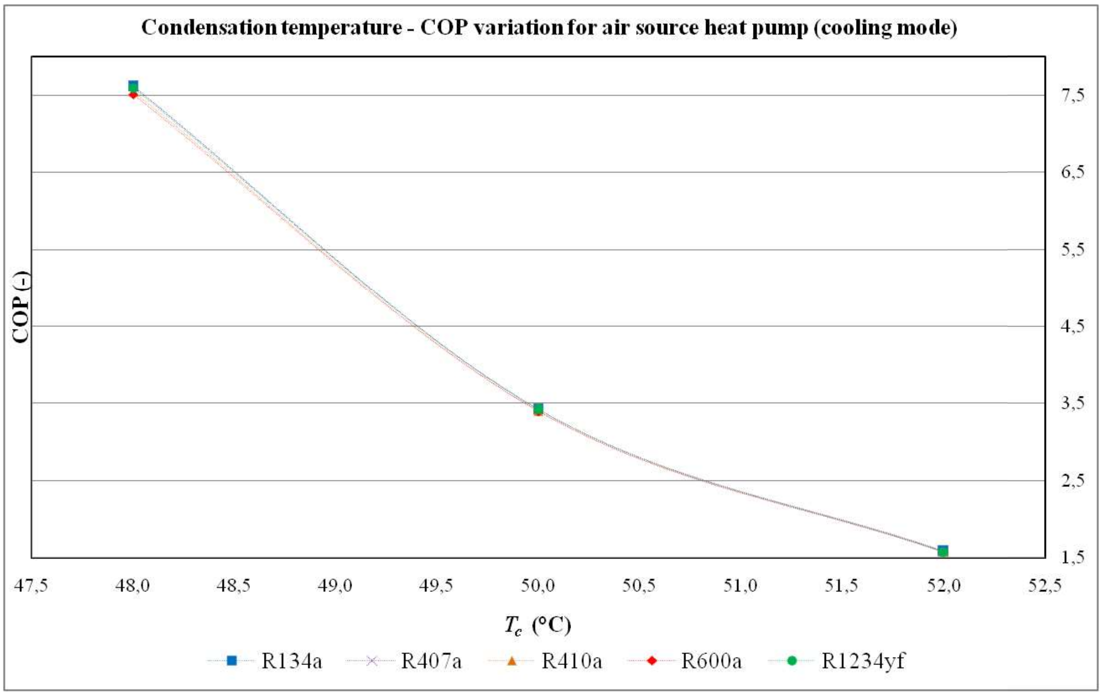

Figure 3.39 T_c - COP variation for air source heat pump (for cooling mode)

The change of coefficient of performance for cooling mode according to the condensation temperature for certain refrigerants in the air source heat pump is shown in Figure 3.39. In the graph, R407a refrigerant achieved the highest COP value in the air source heat pump system at all condensation temperatures. R407a fluid was followed by R134a, R1234yf, R410a and R600a refrigerants, respectively.

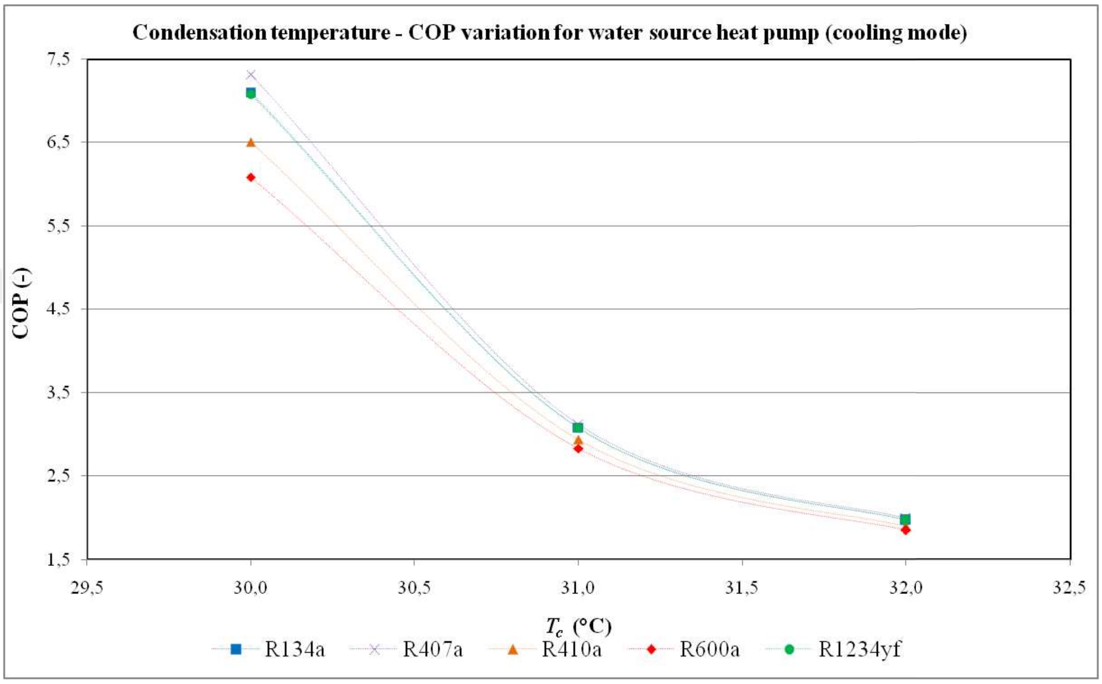

Figure 3.40 T_c - COP variation for water source heat pump (for cooling mode)

The change of coefficient of performance for cooling mode according to the condensation temperature for certain refrigerants in the water source heat pump is shown in Figure 3.40. In the graph, R407a refrigerant achieved the highest COP value in the water source heat pump system at all condensation temperatures. R407a fluid was followed by R134a, R1234yf, R410a and R600a refrigerants, respectively.

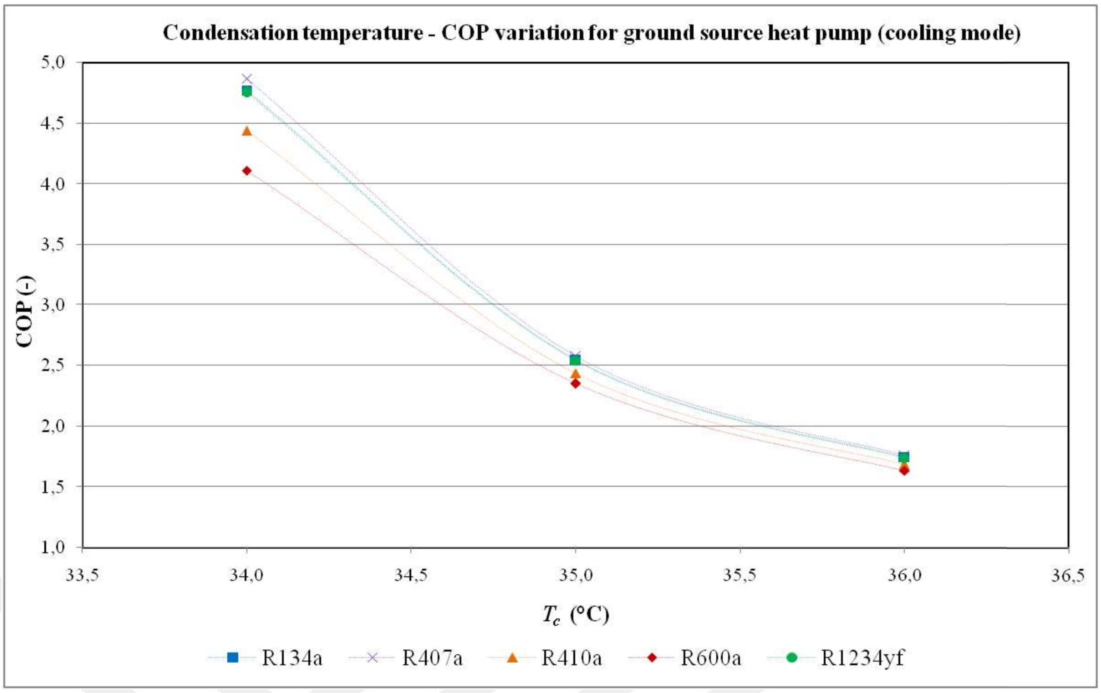

Figure 3.41 T_c - COP variation for ground source heat pump (for cooling mode)

The change of coefficient of performance for cooling mode according to the condensation temperature for certain refrigerants in the ground source heat pump is shown in Figure 3.41. In the graph, R407a refrigerant achieved the highest COP value in the ground source heat pump system at all condensation temperatures. R407a fluid was followed by R134a, R1234yf, R410a and R600a refrigerants, respectively.

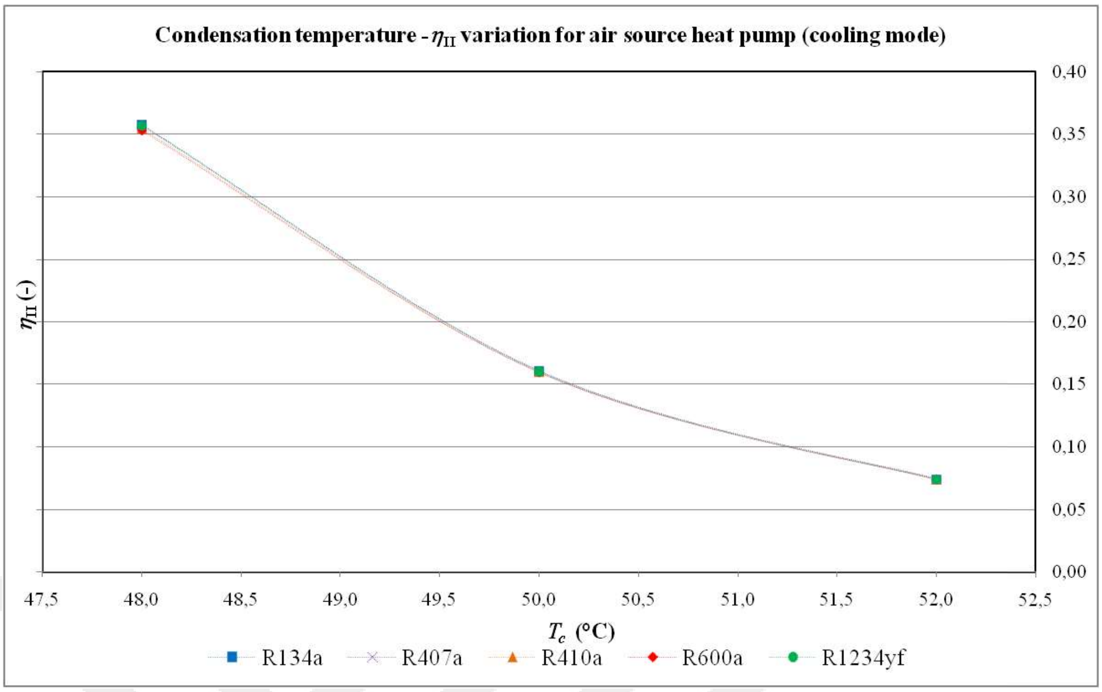

Figure 3.42 T_c - η_{II} variation for air source heat pump (for cooling mode)

The variation of exergy efficiency for cooling mode according to the condensation temperature for certain refrigerants in the air source heat pump is shown in Figure 3.42. In this graph, R407a refrigerant achieved the highest η_{II} value in the air source heat pump system at all condensation temperatures. R407a fluid was followed by R134a, R1234yf, R410a and R600a refrigerants, respectively.

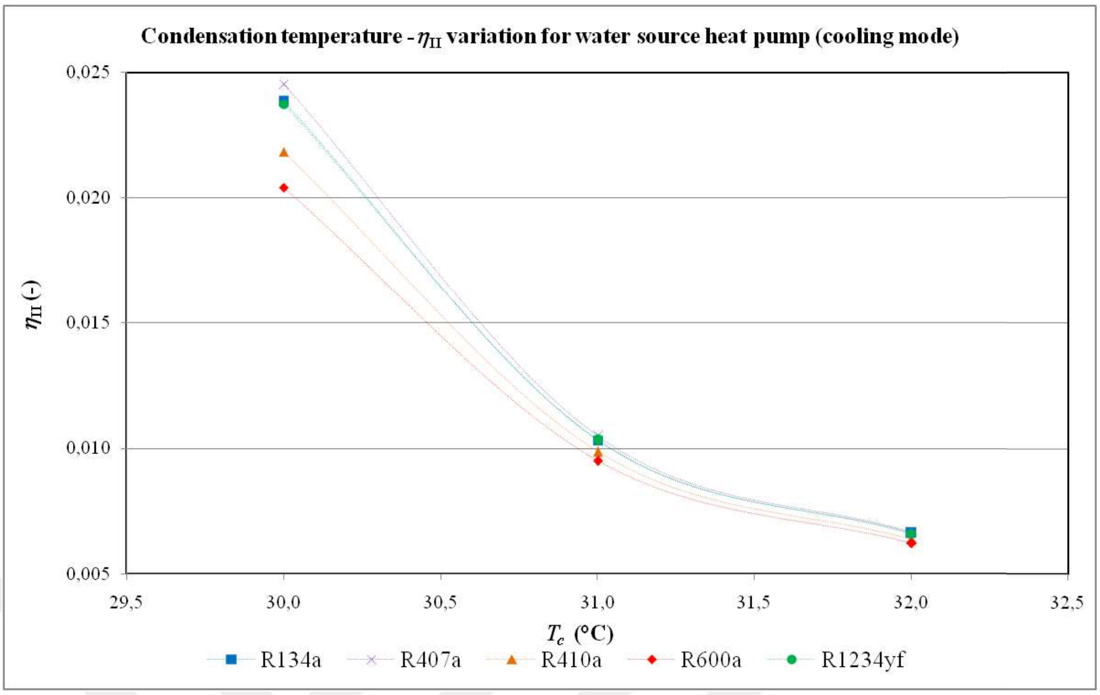

Figure 3.43 T_c - η_{II} variation for water source heat pump (for cooling mode)

The variation of exergy efficiency for cooling mode according to the condensation temperature for certain refrigerants in the water source heat pump is shown in Figure 3.43. In this graph, R407a refrigerant achieved the highest η_{II} value in the water source heat pump system at all condensation temperatures. R407a fluid was followed by R134a, R1234yf, R410a and R600a refrigerants, respectively.

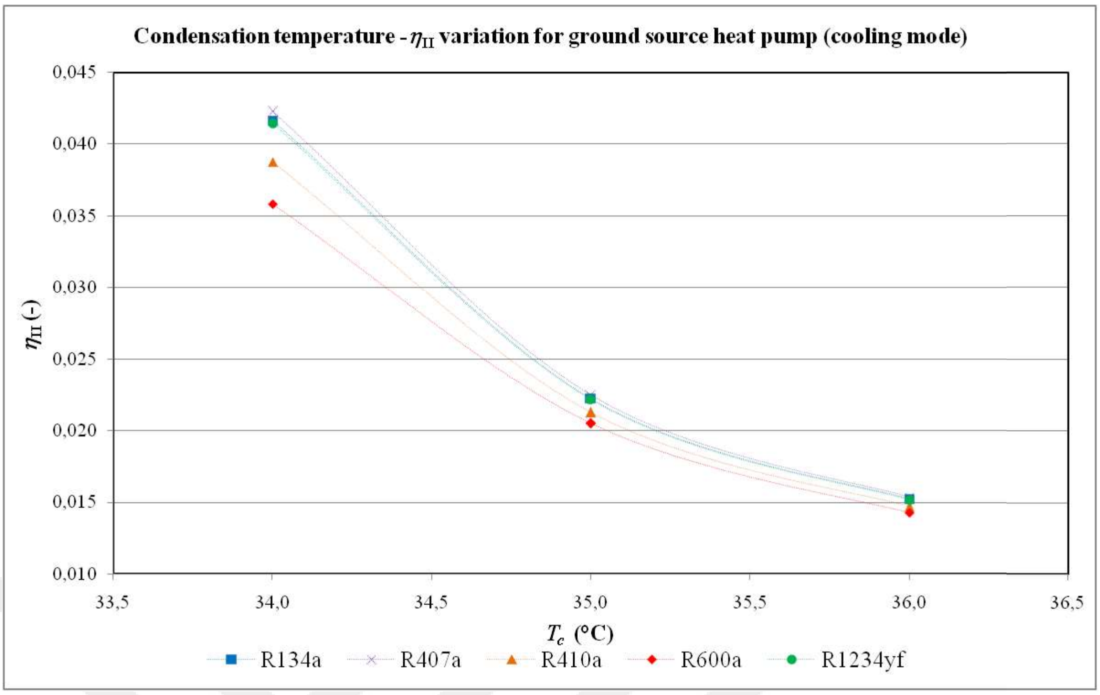

Figure 3.44 T_c - η_{II} variation for ground source heat pump (for cooling mode)

The variation of exergy efficiency for cooling mode according to the condensation temperature for certain refrigerants in the ground source heat pump is shown in Figure 3.44. In this graph, R407a refrigerant achieved the highest η_{II} value in the ground source heat pump system at all condensation temperatures. R407a fluid was followed by R134a, R1234yf, R410a and R600a refrigerants, respectively.

When air, water and ground source heat pumps were examined for the cooling mode, the following results were obtained.

- When R134a, R407a, R410a, R600a and R1234yf refrigerants are used separately as working fluid in a heat pump to be employed in Gölbaşı district, R407a is more effective than the others in terms of COP and η_{II} values in air, water and ground source heat pumps operating at the same conditions.

- When the air, water and ground source heat pumps that can be used in Gölbaşı district of Ankara are compared, it is seen that the water source heat pump is more efficient than the others in terms of energy efficiency under the same conditions.

- When the air, water and ground source heat pumps that can be used in Gölbaşı district of Ankara are compared, it is seen that the air source heat pump is more effective than the others in terms of exergy efficiency under the same conditions.

3.3 Economic Analysis

In this theoretical study, economic analysis has also been made for air, water and ground source heat pumps for heating mode. In this economic analysis for air, water and ground source heat pumps, only the power consumed by the heat pump in the compressor was taken into account. For comparison, the heating power (heating capacity) of the electric heater used is considered to be the same as the heating power of the heat pump. While making the calculations, the time needed for the heater has been accepted as 180 days (6 months). In addition, it has been assumed that both systems work 10 hours per a day. The unit price of electricity used in the calculations was taken as 0.79 ₺/(kW·h) [27]. In the market researching, it was learned that the labor and piping cost of the air source heat pump was between 1,000 and 3,000 ₺, and the labor and piping cost of the water and ground source heat pump was between 150 000 and 400,000 ₺. It has been observed that affect the labor and piping costs of the properties of water or soil where the water and ground source heat pump will be installed (such as density, specific heat, heat transmission coefficient) and horizontal or vertical placement of the heat exchangers. Here, since the project area is not known, a clear price of labor and piping could not be made, and an approximate cost was given. The information obtained and the results of the calculations are presented in Table 3.3.

Table 3.3 Cost and payback periods of air, water and ground source heat pumps compared to electric heaters [28]

Heat pump type	**Heating capacity (kW)**	**Machine and equipment cost (₺)**	**Labor and piping cost (₺)**	**Annual energy consumption cost of the heat pump (₺)**	**Annual energy consumption cost of an electric heater (₺)**	**Payback period (Year)**
Air source heat pump	8	62767	1000-3000	2774.32	11376	7.42-7.65
	12	66683	1000-3000	4162.19	17064	5.25-5.40
	16	79833	1000-3000	5291.26	22752	4.63-4.74
Water and ground source heat pumps	6	80567	150000-400000	1551.40	8532	33.02-68.84
	12	88399	150000-400000	3159.68	17064	17.14-35.12
	16	92604	150000-400000	5881.39	22752	13.03-26.46

When Table 3.3 is examined, it is determined that as the heating power of each heat pump increases, the payback period decreases. In addition, it has been observed that the labor and piping costs of the air source heat pump are very low compared to the labor and piping costs of the water or ground source heat pumps, so the payback period is shorter.

CHAPTER 4

CONCLUSIONS

The increase in the world's population also increases energy needs. It is predicted that the fossil fuels used today to meet this need will run out in a short time. Therefore, the demand for renewable energy sources is increasing. When renewable energy sources are examined, heat pumps are notable for being environmentally friendly and economical. For this reason, in this study, the use of air, water and ground source heat pumps for residential heating was studied theoretically for the Gölbaşı district of Ankara. In this study five different refrigerants were used, namely R134a, R407a, R410a, R600a and R1234yf. Here, it is considered that R1234yf fluid will gain popularity in the near future and whether it is possible to use it in heat pumps is examined. Energy and exergy analysis of each heat pump was performed using five refrigerants and the results were compared. In theoretical analysis, for air source heat pump, the COP values of R134a, R407a, R410a, R600a and R1234yf refrigerants were found to be 2.743983, 2.743395, 2.743633, 2.75979 and 2.741527, respectively. In addition, η_{II} values were found as 0.400346, 0.400261, 0.400295, 0.402653 and 0.399988, respectively. For water source heat pump, the COP values of R134a, R407a, R410a, R600a and R1234yf refrigerants were found to be 3.430179, 3.397651, 3.421599, 3.480346 and 3.363461, respectively. In addition, η_{II} values were found as 0.212840, 0.210822, 0.212308, 0.215953 and 0.2087, respectively. For ground source heat pump, the COP values of R134a, R407a, R410a, R600a and R1234yf refrigerants were found to be 3.911314, 3.867198, 3.881885, 3.994935 and 3.828839, respectively. In addition, η_{II} values were found as 0.199403, 0.197154, 0.197903, 0.2036660 and 0.195198, respectively. By comparing these results the following inferences can be made:

- When R134a, R407a, R410a, R600a and R1234yf refrigerants are used separately as working fluid in a heat pump to be employed in Gölbaşı district, R600a is more effective than the others in terms of COP and η_{II} values in air, water and ground source heat pumps operating at the same conditions.

- When the air, water and ground source heat pumps that can be used in Gölbaşı district of Ankara are compared, it is seen that the ground source heat pump is more efficient than the others in terms of energy efficiency under the same conditions.

- When the air, water and ground source heat pumps that can be used in Gölbaşı district of Ankara are compared, it is seen that the air source heat pump is more effective than the others in terms of exergy efficiency under the same conditions.

- It is seen that the heat pump source temperature, i.e. air, water or ground temperature, directly affects the efficiency of the heat pump. Based on this result, great care should be taken in choosing the right source for the region where any heat pump will be installed.

- The expected performance could not be achieved if the R1234yf refrigerant was used in heat pumps. However, today, R1234yf refrigerant is preferred because it is more environmentally friendly than other fluids. Therefore, it is predicted that this refrigerant will be more efficient if used with nanoparticles. Thus, the COP and η_{II} values of a new refrigerant, which will be obtained by adding various nanoparticles to the R1234yf refrigerant, will be more effective than other refrigerants.

- In the economic analysis, it is seen that the payback period decreases as the heating power increases for each heat pump. In addition, it is observed that the payback period of the air source heat pump for each heating power is shorter than the water and ground source heat pumps.

When air, water and ground source heat pumps were examined for the cooling mode, the following results were obtained.

- When R134a, R407a, R410a, R600a and R1234yf refrigerants are used separately as working fluid in a heat pump to be employed in Gölbaşı district, R407a is more effective than the others in terms of COP and η_{II} values in air, water and ground source heat pumps operating at the same conditions.

- When the air, water and ground source heat pumps that can be used in Gölbaşı district of Ankara are compared, it is seen that the water source heat pump is more efficient than the others in terms of energy efficiency under the same conditions.
- When the air, water and ground source heat pumps that can be used in Gölbaşı district of Ankara are compared, it is seen that the air source heat pump is more effective than the others in terms of exergy efficiency under the same conditions.

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