JANUARY 2018

M.Sc. in Mechamcal Engineering

DEEDAR ROOF MOHAMMED

UNIVERSITY OF GAZİANTEP

GRADUATE SCHOOL OF

NATURAL & APPLIED SCIENCES

EXERGY AND EXERGECONOMIC ANALYSIS OF SOLAR ASSISTED HEAT PUMP SYSTEM

M.Sc. THESIS

IN

MECHANICAL ENGINEERING

BY

DEEDAR ROOF MOHAMMED

JANUARY 2018

Exergy and Exergeconomic Analysis of Solar Assisted Heat Pump System

M.Sc. Thesis

in

Mechanical Engineering

University of Gaziantep

Supervisor
Prof. Dr. Recep YUMRUTAŞ

by

Deedar Raoof MOHAMMED

January 2018

REPUBLIC OF TURKEY
UNIVERSITY OF GAZİANTEP
GRADUATE SCHOOL OF NATURAL & APPLIED SCIENCES
CIVIL ENGINEERING DEPARTMENT

Name of the thesis: Exergy and Exergeconomic Analysis of Solar Assisted Heat Pump System

Name of the student: Deedar Raoof MOHAMMED

Exam date: January 29, 2018

Approval of the Graduate School of Natural and Applied Sciences

Prof. Dr. Ahmet Necmeddin YAZICI
Director

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

Prof. Dr. M. Sait SÖYLEMEZ
Head of Department

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

Prof. Dr. Recep YUMRUTAŞ
Supervisor

Examining Committee Members:	Signature
Prof. Dr. Recep YUMRUTAŞ	.....
Prof. Dr. M. Sait SÖYLEMEZ	.....
Assist. Prof. Dr. Abdulkadir AYANOĞLU	.....

Ihere by declare that all information in this document has been obtained and presented in accordancewith academic rules and ethical conduct. I also declare that, as required by these rules andconduct, I havefully cited and referenced all material and results that are not original to this work.

Deedar Raoof MOHAMMED

ABSTRACT

EXERGY AND EXERGECONOMIC ANALYSIS OF SOLAR ASSISTED HEAT PUMP SYSTEM

MOHAMMED, Deedar Raoof
M.Sc. in Mechanical Engineering
Supervisor: Prof. Dr. Recep YUMRUTAŞ
January 2018
70 pages

In this thesis, exergy and exergeconomic analysis of a solar assisted heat pump (SAHP) system are studied. This system was installed in Energy Labaratory of Mechanical Engineering Department, University of Gaziantep. The SAHP system consists of a heat pump, flat plate solar collectors, energy storage tank, and a room heated. Data for all nodal points of the heating system were collected. Water-glycol solution was circulated through the cycle between the collectors and the storage tank, cycles between heat pump and the storage tank and room. The R22 was circulted in the heat pump cycle. A program in Engineering Equation Solver (EES) was prepared. Thermal and physical properties of refrigerants R744 and R34a were used in the EES program to compare results with those of R22 results. Exery and exergeconomic values are calculated by using the measurement data. The cost functions are obtained using Specific Exergy Costing method. The R134a used in the heat pump system gives the better results because of high exergy efficieny, low exergy destruction and COP. Thermoeconomic factors are low for high cost rate of exergy destruction of heating system. The Capital Investment and, Operation &Maintance and total cost of the SAHP heating system are computed as 1.451 €/kWh. The relative cost difference and exergoeconomic factor value are high for R134a that cause low exergy destrcution. It is well evaluation for cost destruction of exergy rate, relative cost difference and exergocenomic factor that R134a refrigerant has good performance and thermoeconomic values for exergy anlysis.

Key words: Solar Assisted Heat Pump, Exergy Analysis, Exergoeconomic Analysis.

ÖZET

GÜNEŞ ENERJİSİ DESTEKLİ ISI POMPASI SİSTEMİNİN EKSERJİ VE EKSERGOEKONOMİK ANALİZİ

MOHAMMED, Deedar Raoof
Yüksek Lisans Tezi, MakineMüh. Bölümü
Tez Yöneticisi: Prof. Dr. Recep YUMRUTAŞ
Ocak 2018
70 sayfa

Bu tez çalışmasında, güneş enerjisi destekli ısı pompası sisteminin exergy and exergeconomik analizi çalışılmıştır. Isıtma sistemi Gaziantep Üniversitesi Makine Mühendisliği Bölümü, Enerji Laboratuvarında kuruldu. Isıtma sistemi; ısı pompası, düz plakalı güneş kollektörleri, enerji deposu ve ısıtılan bir odadan oluşmaktadır. Sistemin tüm ölçüm noktalarından elde edilen veriler kaydedildi. Su-antifriz çözeltisi depo ve güneş kollektörleri ile ısı pompası ve enerji deposu arasında devir daim pompası ile dolaştırıldı. Isı pompası devresinde ise R22 soğutucu gazı kullanıldı. Denklem çözücü EES (Engineering Equation Solver) programlama dilinde bir program hazırlandı. R22 ile elde edilen sonuçlarla karşılaştırmak amacıyla R744 ve R134 soğutucu akışkanların ısıl ve fiziksel özellikleri kullanıldı. Ölçülen veriler kullanılarak ekserji ve ekserjoekonomik değerler hesaplanmıştır. Özel ekserji fiyat yöntemi kullanılarak fiyat fonksiyonları elde edilmiştir. Isı pompası çevriminde soğutucu akışkan R134 kullanıldığı zaman, yüksek ekserji verimi ve ısı pompası etkinlik katsayısı ile düşük ekserji kaybı vermektedir. Sistemin ilk yatırım maliyeti, bakım ve onarım maliyeti ve toplam maliyeti hesaplanmıştır. Termoekonomik factorün düşük olması, ısıtma sistemindeki ekserji yıkımının yükselmesine neden olmuştur. Bağıl fiyat farklılıgı ve ekserjiekonomik faktörün R134a için yüksek oluşu, ekserji yıkımının duüşük olmasına sebep olmuştur. Ekserji oranı, bağıl fiyat farklılığı ve ekserjiekonomik faktörün iyi hesaplanması sonucunda, R134a akışkanının hem termodinamik hem de termoekeonomik analiz için iyi performans vermiştir.

Anahtar Kelimeler: Güneş enerjisi destekli ısı pompası, Ekserji Analizi, Ekserjiekonomik Analiz.

To Prof. Dr. Recep YUMRUTAŞ

ACKNOWLEDGEMENTS

Prof. Dr. Recep YUMRUTAŞ who is the best and genius person as I have seen in my life and fully plentiful thanks to him. To hard for me to mentioned his supervisor as some words: guidance, encouragement, advice, helpful, showing the right way and also gentle by his life and many feeling cannot be described by words and sentences, only doing research and living with him can be realized.

I would like to thank **Assist. Prof.** Dr. **Abdulkadir AYANOĞLU**, for providing crucial academic and technical support and guidance throughout this thesis. Without their tireless interest and continuous guidance, their positive criticism, suggestions and motivated supports, the completion of this thesis would not have been possible.

Special thanks to **Assoc. Prof. Dr. Onder KASKA**, for his guidance, encouragement and advice. He was always available when I needed his help and advises. I owe him lots of gratitude for having me shown the ways of research. He could not realize how much I have learned from him.

I would like to thank; my parents, my husband Paraw, my daughter Sara, my son Dara, my sisters & brother, and others for their endless supports.

Finally, I would like to thank those who took part in completion of this thesis.

TABLE OF CONTENTS

Page

LIST OF TABLES

Page

LIST OF FIGURES

Page

LIST OF SYMBOLS/ABREVIATIONS

c	cost per unit of exergy (€/GJ)
$\dot{C}$	cost rate associated with exergy (€/h)
C	payment (€)
E	the amount of emission based on energy content (g/kWh)
$\dot{E}^{ph}$	exergy rate (kW)
$\dot{E}_{heat}$	rate of exergy transfer by heat (kW)
h	enthalpy (kJ/kg)
i	rate of return (%)
$\dot{m}$	mass flow rate (kg/s)
P	present value of the payment (€)
$\dot{F}$	fuel
$\dot{P}$	product
$\dot{Q}$	rate of heat transfer (kW)
s	entropy (kJ/kg K)
T	temperature (K)
T_o	environment temperature (K)
T_{in}	inlet temperature (K)
T_{out}	outlet temperature (K)
T_h	hot Source
T_c	cold Source
w	specific work (kJ/kg)
$\dot{W}$	power (kW)
$y_{dest,k}$	component exergy destruction over total exergy input
$y^*_{dest,k}$	component exergy destruction over total exergy destruction
$\dot{Z}$	rate associated with the sum of capital investment and O&M (€/h)
$\dot{Z}^{CL}$	cost rate associated with capital investment (€/h)

$\dot{Z}^{OM}$	cost rate associated with operation and maintance (€/h)
r_{i}	inflation rate
r_{n}	escalation rate
A	equivalent annuity
CELF	constant–escalation levelization factor
LIFO	Last In First Out
HP	Heat Pump
SAHP	Solar Assisted Heat Pump
SPECO	Specific Exergy Cost Method
SPDF	Single – Payment Discount Factor
USCAF	Uniform – Series Compound – Amount Factor
USSFF	Uniform – Series Sinking Fund Factor
R-22	Refrigerant
TES	Thermal Energy Storage
HFC	Hydrofluorocarbons
CFC	Chlorofluorocarbons
GCHP	Ground Coupled Heat Pump
CFFT	Complex Finite Fourier
FBT	Finite Bessel Transform
SEHPC	Solar Driven Ejector Compression Heat Pump Cycle
COP	Coefficient of Performance
GHG	Green House Gas
H_2O	Water
CRF	Capital Recovery Factor
EES	Engineering Equation Solver
EXC	Expenditure Costs
OMC	Operating and Maintenance Costs
PWF	Present Worth Factor
CRF	Capital Recovery Factor
CI	Capital Invesment

Subscripts

eff	effective
L	levelized
k	any component
n	year
dest	destruction

Greek Letters

η	energy efficiency
ε	exergy efficiency
ψ	specific flow exergy (kJ/kg)
τ	total annual operating hours of system at full load (h)

CHAPTER 1

INTRODUCTION

1.1 General View

Worldwide energy utilize continues to rise in spite of restrictions in fossil fuel energy sources. Moreover, global warming is effect due to the utilize of energy produced by fossil fuels, and it is portended to lead to considerable climate alteration and great influence to society. Therefore many countries have attempted to reduce their energy utilize. The decreasing of global warming influence and reducing total energy utilize are effective by Energy conservation, and also it assists extent the life of non-renewable energy sources [1].

In the recent decennium, heat pump models as ground source and air source which have received huge attentiveness due to energy request rising and fossil fuels exhausting at an alarming average.

The notion of heat pump has been known since the 1800s. To supply applicable, environment-friendly alternatives to traditional unitary systems, these systems have been known. They can make considerable contributions to lowering in electrical energy usage and CO_2 emissions. By charging of the solar energy to the underground storage tank, it could be utilized as the renewable energy reservoir of heat pump systems. The three main ingredients of the heat pump (HP): the heat source system, the heat pump itself, and a heat allocation and store. Conductive energy –mostly electricity/gas–require to turn on compressor and pump. For heating, ground, water, environment air is reservoir and building express as sink [2].

A cycle of heat pump was performed between cold reservoir at low temperature and hot reservoir at high temperatures by using input energy. The cycle transports from naturalistic or man-made heat sources to an implementation. The cycle could also be utilized for cooling by transporting heat from the implementation to be cooled to enviroment.

The cycle are utilized inrefrigeration, air-conditioning, heating, and other implementations. Figure 1.1 describes heat flow diagram [3].

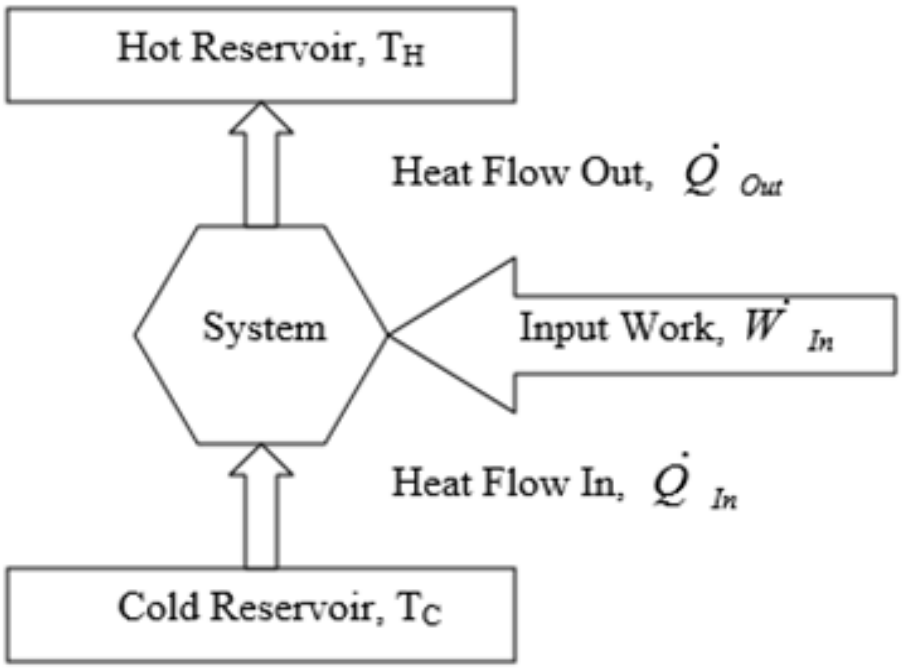

Figure 1.1 Heat flow diagram of a heat pump.

Solar assisted heat pump (SAHP) with daily energy storage for heating of a space is therefore a new alternative on the manner. The storage of solar energy in a container for heating aims by heat pump point out in literatures and easily used in the previous term. Basically, from high to low temperature reservoirs are worked by circulation which define the aim of SAHP [5].

In this study, exergy and exergeconomic analysis of a solar assisted heat pump (SAHP) system are studied. The SAHP system consists of a heat pump, flat plate solar collectors, energy storage tank, and a room heated. A program in Engineering Equation Solver (EES) was prepared. Thermal and physical properties of refrigerants R744 and R34a were used in the EES program to compare results with those of R22 results. Exery and exergeconomic values are calculated by using the measurement data. Capital investment, operation and maintanence cost and total cost are calculated for the SAHP system.

1.2 Thesis Structure

The structure of the thesis, which is exergy and exergeconomic analysis of the SAHP with a storage by methods of thermodynamic with thermoeconomic, is given as following:

Chapter 1 gives general information about worldwide energy statue, and follow with heat pump system, and finally define solar assisted heat pump system.

In Chapter 2, detail of heat pump system by their components and also energy and exergy analysis are expressed.

In Chapter 3, thermodynamic formulations by both energy and exergy methods are explained. And also, principles with formulation of exergoeconomic analysis are declared.

Energy and exergy analysis of SAHP system are performed for each component of the system in Chapter 4.

In Chapter 5, exergoeconomic analysis of the SAHP system is expressed. Specific exergy cost method (SPECO) is used for system. The results of the analyses can be used to improve to satisfy measurement of the process and decrease product costs.

In Chapter 6, results obtained from the study are discussed, and main conclusions from the study are drawn.

CHAPTER 2

LITERATURE AND BACKGROUND OF HEAT PUMP SYSTEM

2.1 Introduction

Solar energy and heat pump systems are good alternative energy for decreasing non-renewable energy resources, and also consuming cost of heating & cooling of space and water heating. Those system can be used for low temperature heating system and a solution for pollution, noise, high cost receipt and fossil fuel consumption such as CO_2, SOx, HC etc [6].

Solar energy is mostly used during summer months (about 6 months) and during winter season for energy requirement of the buildings. Solar energy are mostly applied in Turkey and flat plate solar collectors widely used for water heating at residential, air conditioning and industrial system. One of the solution is energy storage (store heat of the solar energy for high loads) which used for seasonal and diurnal discrepancy. Seasonal (long term) storage and daily (short term) storage with most potential is the thermal energy storage because the fraction of heat in the national energy balances in Turkey is considerable. Energy storage is integrated into heat pump for satisfying heat demand and decreasing cost of energy source [7].

2.2 Thermodynamic Equations of Heat Pump System

In this section, basic thermodynamic equations for a heat pump system such as Clausis statement and Carnot cycle systems will be explained shortly [8].

2.2.1 Clausis Statement

The Clausius statement work as a cyclic device where from hot source (T_h) to cold source (T_c), which shown in Figure 2.1 [2]. Lord Kelvin, in 1852, mentioned dissipation theory of energy. First law of thermodynamics always defines conversion any given quantity of mechanical energy into its equivalent heat energy.

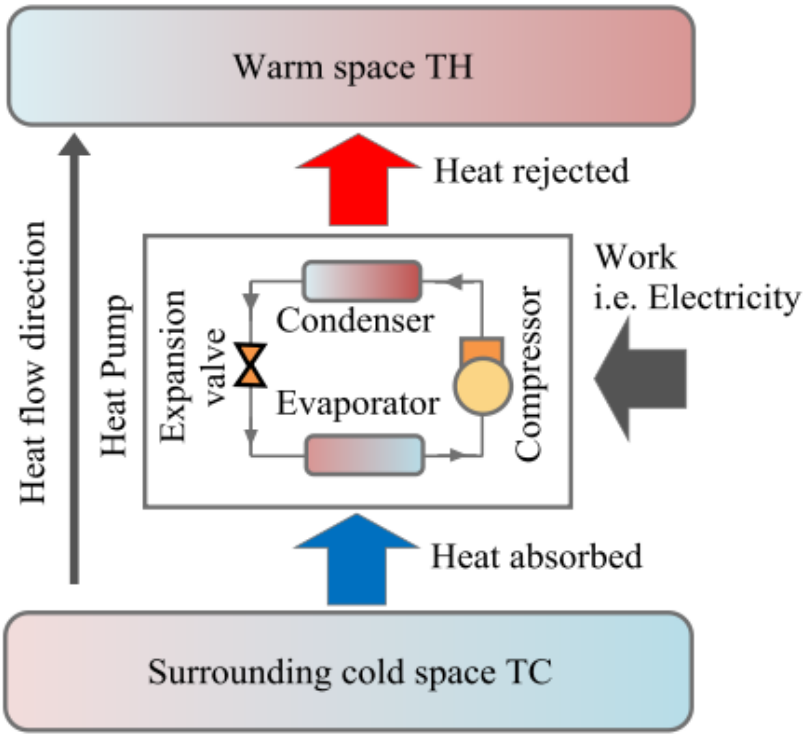

Figure 2.1 Basic heat pump.[2]

2.2.2 The Carnot Cycle

The reversed Carnot cycle (reversible) and temperature-entropy (T-s) diagram are represented as schematic in Figure 3.2(a) and (b). In reality, it is not possible to achieve this ideal reversible cycle.

Isentropical cycle expressed as follow:

Processes 1 and 2: compressor,

Processes 2 and 3: condenser,

Processes 3 and 4: expander,

Processes 4 and 1: evaporator.

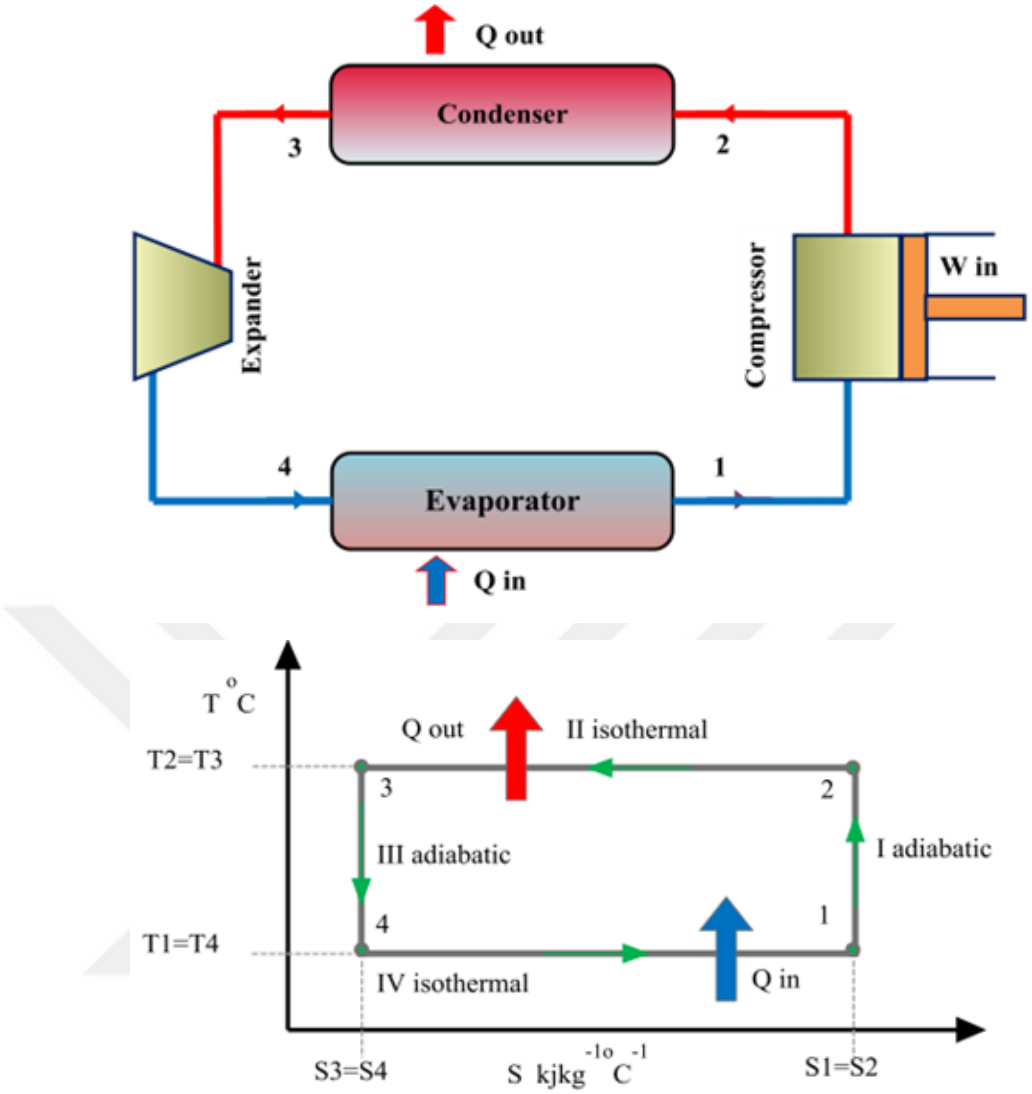

Figure 2.2 Schematic of Carnot cycle (a), and its temperature-entropy (T-s) diagram (b) [2]

2.3 Description of Solar Assisted Heat Pump

The heat pump components are compressor, evaporator, condenser, valve and energy storage with solar collector [9]. Refrigerant (R-22) flows in a closed circuit in order to realize flowing principals.

2.3.1 Solar Assisted Heat Pump

The design of a Solar Assisted Heat Pump (SAHP) with a daily storage tank and parts of the system are discussed. The SAHP is contained essentially with solar collectors (flat plate types), energy storage with heat pump to heat a room for winter season. All of the system components are explained in following subsections detail.

And also, performances of the SAHP and its system are indicated as COP_{HP} and COP_S which are fomulated to calculate system performance during operation. The basic components were explained by their properties with equations to define utilization of the system. The schematic of SAHP is shown in Figure 4.1. The system is found at Mechanical Engineering Department, heat laboratory, University of Gaziantep, Turkey at a latitude of 37.1°N. Flat plate collectors are used in the system. They absorbed solar radiation coming from sun by brine, then brine tranfers its energy to water in TES tank to store it. Water circulated in a close loop between the TES tank to heat pump for adjusting heat. Finally, heat pump move heat to room by a floor heating system. At the system, flat type collectors are used for heating of water in the tank and the stored heat is transfered to the room in the labaratory. Thermal energy storage is one of valuable parts of the system. The heating reqirements and solar radiation are time-dependent with known interval. The solar radiation varified in days and at the end of day up to morning shining, no radiotion happen. eating applications. Solar radiation is a time-dependent energy source with an intermittent character. The heating demands of residential houses are also time dependent. Thus, thermal energy storage tank store heat in the tank to transfer heat pump to final desination for heating room at requirement temperature. The refrigerant is another critical component especially for heat pump system. By changing phase, it utilizes energy tranfer between TES tank to room. R22 is used in the heat pump cycle for real data. Then, refregerants R744 and R134a are used in the EES program to compare real data with therotical computation for observing increasement-decreasement of perfonmace of heat pump system with whole system. Before starting into energy and exergy analyses, all processes are assumed as steady-state and steady-flow with negligible potential and kinetic energy effects in an adiabatic form [82-86].

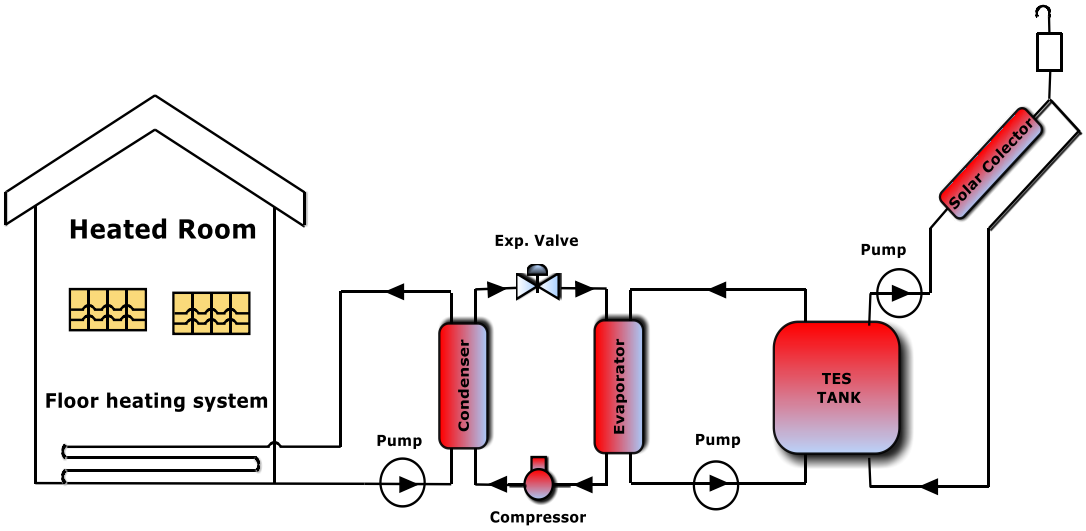

Figure 2.3 The SAHP system (Modified from ref.[21])

2.3.2 Compressor

The core ingredient of a heat pump is the compressor which circulates refrigerants through the heat pump cycle. The compressor works as from low to high pressure by a force of an electric motor fixed in compressor. Profits other than competence are therefore significant dispute, like the reality which has no require oil tank with other connection of gas (mostly cannot achieved, especially in rustic places) [10-11].

Figure 2.4 Compressor

2.3.3 Condenser

The refrigerant departs the compressor as a gas at high pressure and temperature. Heat must be takeoff from the gas in order that alter it to a liquid where this process happened in the condenser in which the refrigerant emits heat to its surroundings and

becomes a liquid. In the condenser the refrigerant flow one circuit. A water flows as a cooling fluid, at a temperature less than the refrigerant in other circuit [12-13].

Figure 2.5 Condenser

2.3.4 Evaporator

The evaporator is a fundamental ingredient of heat pump system. By circulating flowing fluid on pipes where heat transfer was occurred. The heat transfer from hot source to refrigerant by absorption of coil at evaporator. Copper with aluminum fins are used in constructed of typical evaporator when the aluminum fins surrounding the copper coils, in order to add more surface to coils and increases heat transfer more effectively [14-15].

Figure 2.6 Evaporator

2.3.5 Expansion Valve

An expansion valve is an ingredient of heat pump system which detects the quantity of fluid through evaporator that checking superheat in component. An expansion valve is often referred as “metering devices”, and it is causes a pressure fall of the working fluid. During steady state operation, the valve needle stays open and the size of the opening or the place of the needle is linked to the pressure and temperature of the evaporator [16-17].

Figure 2.7 Expansion valve

2.3.6 Flat Plate Solar Collectors

Solar collectors fundamentally work as heat exchangers which transmit incident solar radiation to flowing fluid such as air or liquids. Mostly, flat plates used for space or water heating or other applications [18-19]. In the SAHP system, flat plate solar collectors are used.

Figure 2.8 Flat plate solar collectors [20]

2.3.7 Thermal Energy Storage

The heating demands for a house is in the middle of the evening, and solar radiation is maximum in the middle of day. Thus, a equipment needs to store energy which is called as thermal energy storage (TES) tank. TES applications are the most interesting thermal systems. The TES tank was produced in cylindrical shape. The pipes fixed between storage tank to solar collectors by well insulated. TES tank categorized as hot liquids and solids of sensible heat, also latent heat in melts and vapor or chemical reaction cause heat [21].

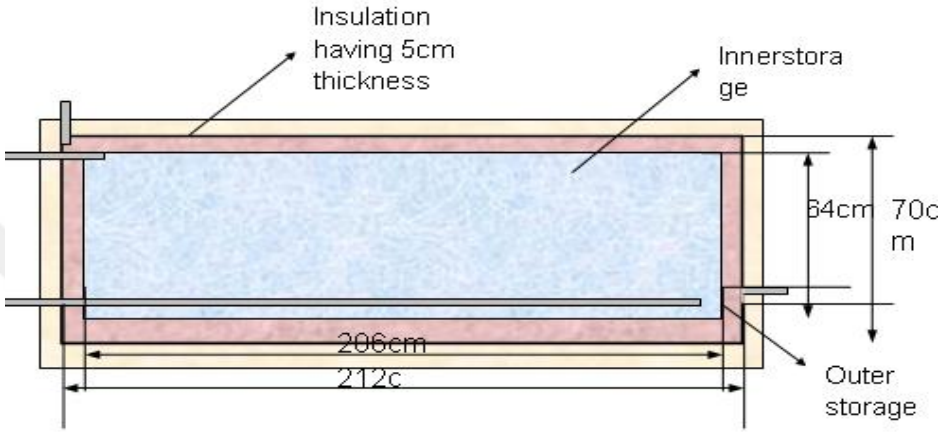

Figure 2.9 Thermal energy storage [21].

2.3.8 Refrigerants

The refrigerant has important thermodynamic properties such as is noncorrosive, safe (nontoxic and nonflammable). Ideal refrigerants are benefit for various operational pressure of heat pumps.

Chlorofluorocarbons (CFC) were found for residential usage which have chemical stability and low toxicity. The most common types were: CFC-(12, 114). Hydrofluorocarbons (HFC) have a free chlorine as R (134a,152a,32,410a, 500, 502, 507, 12, 22 and 502) [22-23].

2.4 Literature Survey

This chapter focused on literature review of SAHP with applications [24], and explained thermodynamic analysis of those studies.

Eğrican [25] studied a series SAHP with heat source as storage which figured out Figure 8. The system was consisted by water-water heat pump with solar panels and tank for heating a house. R-22 was used as flowing fluid with 10 kW heating load in

İstanbul, Turkey. The system optimized by defining collector is 30 m^2 and storage is 2.25 m^3.

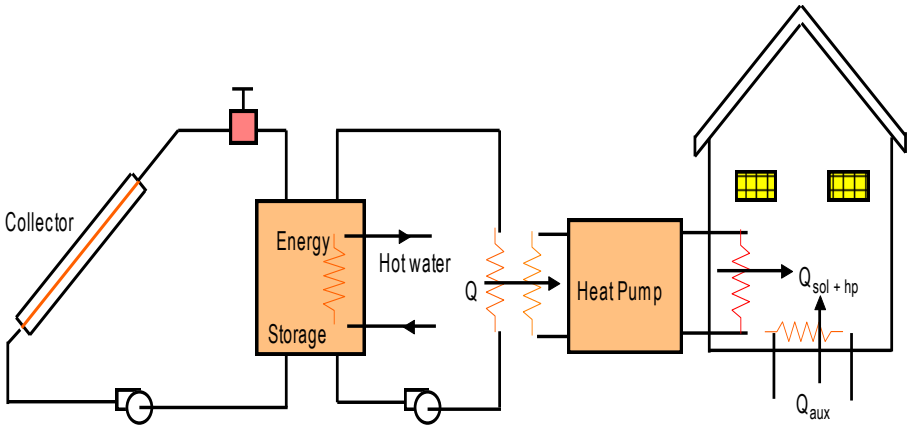

Figure 2.10 Series connected SAHP [25].

Yumrutas and Unsal [26] improved a model as analytical and computational model to obtain annual periodic performance of a Ground Couplrf Heat Pump (GCHP) and SAHP with a spherical tank. An analytical solution of the TES tank was obtained by application of Complex Finite Fourier (CFFT) and the Finite Bessel Transform (FBT) methods. And also, software program was developed for this model.

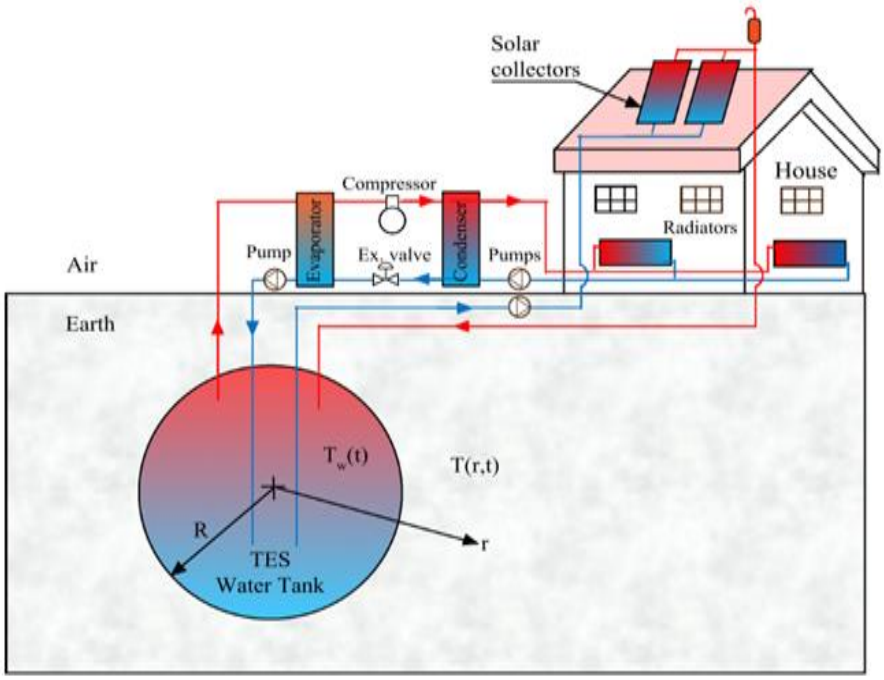

Figure 2.11 House heating with heat pump and underground TES tank [26]

Different types of material were caused to increase energy exchange of thermal conductivity between energy storage and ground. As a result of energy change; storage temperature and performance of system were increased. Finally, annual COP was increased by deep ground for energy storage [27].

Yan el at. [28] studied on thermodynamics of solar driven ejector compression heat pump cycle (SEHPC), which driving ejector by solar in order to get more performance. The SEHPC cycle utilizing R134a and R123yf as the refrigerant and contrasted to traditional heat pump cycle. Some of the important outcomes can be explained as follow: The SEHPC system produces a noteworthy amelioration heating performance, where the system coefficient of performance (COP), heating capacity and heating exergy product enhanced by 15.3%, 38.1% and 52.8% over the traditional system, respectively. It is determined that the ejector, go on by condenser and evaporator had the biggest exergy destruction. In the future research, additional examinations and empirical on working characteristics of the SECHP is demanded.
Ozgener and Hepbasli [29] studied on thermodynamics of SAGSHPGHS, where it was inaugurated at Institute of University in Turkey. The SAGSHPGHS system was consisted of three major circuit: (i) refrigerant, (ii) ground coupling and (iii) fan-coil. Some of the important outcomes which were obtained throughout the test for three months can be explained as follow:

The COP prices for system verified from 2.00 to 3.125, whilst the COP of the entire system were nearly 5-20% less than COP of the heat pump. The exergy efficiency prices for entire system and GSHP prices were as 67.7 and 71.8% for product and fuel. Finally, this study is beneficial for researchers transacting with system especially SAGSHP.
Asae el at. [30] studied on economic estimation of SAHP strengthen for stock house in Canadia. In order to consume energy and greenhouse gas (GHG) emission, the SAHP retrofit is inserted into living Canadian houses. Two thermal storage tanks were uses in the system to save extra solar energy for use later in the day. Some of the significant outcomes can be demonstrated as follow:

Energy keeping and GHG emission decreasing is mightily influenced by climate. In consequence of SAHP system retrofit annual energy consumption of a living house in Canada will minimize about 80–90 GJ. Main minimizing of GHG emission was

related with NG keeping, where around 19% of GHG emission (equivalent to 12.59 Mt of CO2e) of the CHS was decreased. In all regions the solar fraction was under 50%, fractional thermal Energy storing was higher contrasted to solar combined system whilst the solar combined system is more effective from expanded fractional energy storing perspective.

Dong el at. [31] studied on thermodynamic analysis of SIASHP with R407c for heating without water, where the system installed in the test room in Taiyvan for space heating in winter, and then contrasted it with the same sort of traditional air source heat pump (ASHP). Some of the significant outcomes can be demonstrated as follow: The COP of SASHP was 2.55 which was 5.8%larger than the 2.41 COP of SAHP. The exergy efficiency of SASHP (76.8%) was 7.9% greater than the exergy efficiency of SAHP (71.1%). The SASHP system had a best thermodynamic cycle impact than the ASHP system.

Suleman el at. [32] studied on thermodynamic analysis of an integrated SAHP which was improved for industrial heating. For process heating water the system uses heat pump cycle and for other industrial heating process uses solar energy. The outcomes display that the energetic COP of the heat pump cycle was 3.54 while the exergy efficiency was 42.5%. Furthermore, the energetic COP of the system was 2.97 and exergy efficiency was 35.7%. In order to developing the efficiencies of energy systems more analysis that comprise supplemental parameters like the radiation and various fluids.

Lohani and Schmidt [33] studied on the compare of thermodynamic analysis of fossil plant, ground and air source types for a residential building, in order to illustrate which system has effective change and provision of thermodynamic analysis. Some of the important results can be demonstrate as follow:

The GSHP was best traditional heating system or air source heat pump, which has 50% high gross prime coefficient of performance (COP) and gross prime exergy competence and 25% less prime thermodynamic request than traditional system. In generation subsystem the exergy losses is rate of 90% whilst exergy losses system is 85-95%. Energy competence was rate of 0.5 to 1.06 while exergy competence was from 0.035-0.09.

A system design for low grade energy to observe cooling and heating cycle run by solar energy for thermodynamic analysis. The exergy destruction is 52.5% for inlet solar whereas inlet exergy destruction 25%. The efficiencies of energy and exergy system changed by 32% to 39% and 2.5%-4.0%. The energy efficiency of cooling and heating changed between 34.87% to 48.26% . Thus all of components have to be effectively work on overall system performance [34].

The R22 and R744 used in SAHP to obtain performances and exergy losses for sunny and cloudy days The COP of the R22 and R744 were 3.21 and 2.75 for cloudy day for sunny days which rise up to 20.2%. The solar exergy loss for solar collector is increased 13.5% by comparing with sunny day. The exergetic efficiency for R22 was 6.2% more than R744 as a result of R744 irreversibility. As a result of system for R744 need to improve efficiency for both of compressor and expansion [35].

The solar building was heated by vapour compression heat pump. The energy and exergy were calculated for system. The exergy losses happened spontaneously at compression and condensation. An electricity storage required to store solar arrays [36].

The SAHP was developed by a novel theoretical (30 years period) for a building and borehole spacing were discussed. By reducing minimum returning fluid temperature cause to rise the spacing among borehole decrease with length of boreholes [37].

The exergoeconomic analysis used to optimize residential solar heating buildings by using MATLAB for thermodynamic analysis of equipments, were defined. The experimental results show that exergy loss of cylindrical and trapeze tanks were 19.8% and 8.3% [38].

CHAPTER 3

THERMODYNAMIC AND THERMOECONOMIC ANALYSIS

3.1 Introduction

Energy is the essential matter in thermodynamics and energy system engineering. Energy analysis is most important sections of system. Energy like a matter that can be stored in various system, also transfered from a form to another form, or transformed among systems. Energy balances are mostly used in the design and analysis of conversion systems. Although energy balances can define energy supply needs in different types as heat, and shaft work.

Exergy expressed as a theoretical maximum useful work (shaft or electrical) used in thermal system which took into thermodynamic equilibrium with the reference temperature of surronding while heat transfer happens with system. As another definition for exergy which theoretical minimum of work requirement to specified a matter quantity's given a defined system. Thus, exergy is a definition of the access point to related point. All of real thermal systems are not reversible, and some of the exergy accommodated to the all of system is destroyed, Exergy stays stable at a reversible system constant [39].

The exergoeconomic is avaliable for economics relation with exergy analysis. It is a method for engineer through energy analysis with economic values, but it is critical to design and operate for cost-effective system [39].

Thermoeconomics analyze price of sources, bill and system irreversibilities in all over the system. The analysis is used for saving money with utilized system efficiently. By analyzing, flow streams and system cost for a thermal system asses to realize money information from the first investment up to final output.

In this chapter, general thermodynamic and thermoeconomic formulations are given containing energy and exergy methods. The formulas are appropriate for the solar assisted heat pump (SAHP) heating system.

3.2 Energy Analysis

Thermal system is defined by energy balances and with mass balances that can be mostly designing and energy systems analysis.

3.2.1 Mass Balance

The net mass transfer changes into system as increasing or decreasing in a system [39]. In the rate mass defined as:

$$\sum \dot{m}_{\mathrm{i}} - \sum \dot{m}_{\mathrm{e}} = \frac{dm_{\mathrm{system}}}{dt} \tag{3.1}$$

where *i* and *e* are due to inlet and exit moments of the any control volume, respectively. In a steady-state process, no changes happen on mass amount with time.

For a mass principle conversation can be defined as the following formula:

$$\sum \dot{m}_{\mathrm{i}} = \sum \dot{m}_{\mathrm{e}} \tag{3.2}$$

3.2.2 Energy Balance

The first law of the thermodynamics states as neither created nor destroyed according to experimental results [39-41]. The energy conservation law is written as: no different cause in a system for entering and exiting of process. The formula is expressed as [39]:

$$\dot{E}_{\mathrm{in}} - \dot{E}_{\mathrm{out}} = \Delta \dot{E}_{\mathrm{system}} \tag{3.3}$$

The total energy is zero with time and control volume of energy remains constant.

$$\dot{E}_{\mathrm{in}} = \dot{E}_{\mathrm{out}} \tag{3.4}$$

Full of the system for steady-flow system is explicitly explained as;

$$\dot{Q}_{in} + \dot{W}_{in} + \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + g z_i \right) = \dot{Q}_{out} + \dot{W}_{out} + \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + g z_e \right) \quad (3.5)$$

where h_i, h_e, V_i, V_e, z_i, z_e represent enthalpy, velocity, height for control volume of mass flow, respectively.

3.3 Exergy Analysis

For the calculation and development of systems, it is fundamental to expressed thermodynamic irreversibility and the reaction between system components. All real energy conversion processes are non-reversible according to dissociate effects as chemical reaction, temperature changes between system, blending of matter at various quantity, unrestrained expansion and friction. An exergy destruction was computed by using energy balance equations of the SAHP system. The thermodynamic irreversibility cannot be prevented by available system. Dimensionless parameters are used for performance calculations [42-45].

3.3.1 Reference Environment and Exergy Components

The reference states for temperature and pressure (T_0, p_0) are used in exergy definition for variables state and chemical properties of chemical parts during thermodynamic system, and also changed between one system to environment which is named as thermodynamic environment. The environmental temperature and pressure are generally assumed as 298.15 K and 1.013 bar [39,41,45].

Although the internal properties of the surrounding are accepted to stay constant, the extensive properties can alternate as a effect of systems change. The exergy environment is zero.

The total exergyof a system (E_{sys}) can be separated into four parts: Physical exergy E_{sys}^{PH}, kinetic exergy E^{KN},potential exergy E^{PT}, andchemical exergy E^{CH} and defined as:

$$E_{sys} = E_{sys}^{PH} + E^{KN} + E^{PT} + E^{CH} \quad (3.6)$$

The total specific exergy on a mass basis e_{sys} is;

$$e_{sys} = e_{sys}^{PH} + e^{KN} + e^{PT} + e^{CH} \tag{3.7}$$

The physical exergydefined as;

$$E_{sys}^{PH} = (U - U_0) + p_0(V - V_0) - T_0(S - S_0) \tag{3.8}$$

where U, V and S defined as internal energy, volume and entropy, respectively. The subscript "o" is expressed of the temperature T_0 and pressure p_0 of the environment for any one system. The rate of physical exergy $\dot{E}^{PH}$ explained as:

$$\dot{E}^{PH} = (H - H_0) - T_0(S - S_0) \tag{3.9}$$

where H and S define the enthalpy and entropy, respectively. The physical exergy $\dot{E}^{T}$ (due to system temperature) and mechanical exergy $\dot{E}^{M}$ (due to system pressure) expressed as:

$$\dot{E}^{PH} = \dot{E}^{T} + \dot{E}^{M} \tag{3.10}$$

A complex computation of the specific thermal and specific mechanical exergy is possible only for ideal gases and incompressible liquids:

$$e^{T} = \int_{T_0, p_0}^{T_0, p} c\left(1 - \frac{T}{T_0}\right) \mathrm{d}T \tag{3.11}$$

$$e^{M} = \int_{T_0, p_0}^{T_0, p} v \, \mathrm{d}p \tag{3.12}$$

where v defined specific volume. The specific thermal exergy of a fluid at temperature T and pressure p is explained as;

$$e^{T} = e^{PH}(T, p) - e^{PH}(T_0, p) \tag{3.13}$$

The mechanical exergy is defined as;

$$E^{\mathrm{M}} = E^{\mathrm{PH}} - E^{\mathrm{T}} \tag{3.14}$$

Kinetic and potential exergies are equal to kinetic and potential energies, respectively.

$$E^{\mathrm{KN}} = \frac{1}{2} m\vec{v}^{2} \tag{3.15}$$

$$E^{\mathrm{PT}} = mgz \tag{3.16}$$

Here $\vec{v}$ and z denote velocity and elevation relative to coordinates in the environment $\left(\vec{v}_0 = 0, z_0 = 0\right)$.

3.3.2 Exergy Balance, Exergy Destruction, and Exergy Loss

All thermal systems are controlled by the conservation laws (mass and energy). Exergy, unfortunately, cannot be conserved, but it is destroyed by inefficiency effects of a system. As a conclusion, an exergy balance must include a dissipate term, disappear in a reversible process. In addition to exergy has a lost definition from system to surrounding.

The exergy destruction is defined as $\dot{E}_{\mathrm{D}}$ due to irreversibilities (entropy generation). The irreversibilities are happend by chemical reaction, temperature difference, mixing of matter, and unrestrained expansion and friction. The exergy destruction is computed by exergy balance formulation or entropy generation, $\dot{S}_{\mathrm{gen}}$ as follow [39, 46,47]

$$\dot{E}_{\mathrm{D}} = T_0 \dot{S}_{\mathrm{gen}} \tag{3.22}$$

The exergy destruction for full of system is a summarization of the exergy destruction:

$$\dot{E}_{\mathrm{D,total}} = \sum_{\mathrm{k}=1}^{\mathrm{n_k}} \dot{E}_{\mathrm{D,k}} \tag{3.23}$$

The rate of exergy destruction in the *k*'th component of a system is given by;

$$\dot{E}_{\mathrm{D,k}} = \dot{E}_{\mathrm{F,k}} - \dot{E}_{\mathrm{P,k}} - \dot{E}_{\mathrm{L,k}} \tag{3.24}$$

where, $\dot{\mathrm{E}}_{\mathrm{F,k}}$ and $\dot{\mathrm{E}}_{\mathrm{P,k}}$ are the so-called exergetic fuel and exergetic product, respectively, and $\dot{E}_{\mathrm{L,k}}$ represents the exergy rate loss in the *k*th component, which is usually zero when the component boundaries are at T_0.

The total exergy destruction is defined as;

$$\dot{E}_{\mathrm{D,total}} = \dot{E}_{\mathrm{F,total}} - \dot{E}_{\mathrm{P,total}} - \dot{E}_{\mathrm{L,total}} \tag{3.25}$$

The general exergy balance formulation for a control volume (positive direction of work transfer) is explained as;

$$\frac{dE_{\mathrm{CV}}}{dt} = \sum\left(1-\frac{T_0}{T_{\mathrm{k}}}\right)\dot{Q}_{\mathrm{k}} - \left(\dot{W} - P_0\frac{dV_{\mathrm{CV}}}{dt}\right) + \sum\dot{E}_{\mathrm{i}} - \sum\dot{E}_{\mathrm{e}} - \dot{E}_{\mathrm{D}} \tag{3.27}$$

where $\dot{E}_{\mathrm{i}}$ and $\dot{E}_{\mathrm{e}}$ are the total exergy transfer rates at the inlet and exist for the total, physical, chemical, kinetic, and potential exergy associated with mass transfers. The term $\dot{Q}_{\mathrm{k}}$ represents the rate of heat transfer at the location on the boundary where the temperature is T_{k}. The related rate of exergy transfer $\dot{E}_{\mathrm{q,k}}$ is given by;

$$\dot{E}_{\mathrm{q,k}} = \left(1-\frac{T_0}{T_{\mathrm{k}}}\right)\dot{Q}_{\mathrm{k}} \tag{3.28}$$

For $T_{\mathrm{k}} > T_0$, the exergy rate $\dot{E}_{\mathrm{q,k}}$ related with heat transfer is always smaller than the heat transfer rate $\dot{Q}_{\mathrm{k}}$. For steady-flow systems, $\frac{dE_{\mathrm{CV}}}{dt} = 0$, and Equation (3.27) becomes as;

$$0 = \sum\left(1-\frac{T_0}{T_{\mathrm{k}}}\right)\dot{Q}_{\mathrm{k}} - \left(\dot{W} - P_0\frac{dV_{\mathrm{CV}}}{dt}\right) + \sum\dot{E}_{\mathrm{i}} - \sum\dot{E}_{\mathrm{e}} - \dot{E}_{\mathrm{D}} \tag{3.29}$$

3.3.3 Exergetic Efficiency

Dimensionless term defines available of performance. The exergetic efficiency defines the system performance or system equipment for thermodynamic analysis.

The *k'*th component ε_{k} is expressed ratio of $\dot{E}_{\mathrm{P,k}}$ with $\dot{E}_{\mathrm{F,k}}$. The exergy rates formulate as follow:

$$\varepsilon_{\mathrm{k}} = \frac{\dot{E}_{\mathrm{P,k}}}{\dot{E}_{\mathrm{F,k}}} = 1 - \frac{\dot{E}_{\mathrm{D,k}} + \dot{E}_{\mathrm{L,k}}}{\dot{E}_{\mathrm{F,k}}} \tag{3.30}$$

3.3.4 Exergy Destruction Ratio and Exergy Loss Ratio

The exergy destruction ratio $y_{\mathrm{D,k}}$ is eequal to a ratio of exergy destruction $\dot{E}_{\mathrm{D,k}}$ to total fuel exergy supplied $\dot{E}_{\mathrm{F,total}}$. This ratio is given as the formula.

$$y_{\mathrm{D,k}} = \frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{F,total}}} \tag{3.32}$$

Also the exergy destruction rate can be compared to the total exergy destruction rate $\dot{E}_{\mathrm{D,total}}$:

$$y^{*}_{\mathrm{D,k}} = \frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{D,total}}} \tag{3.33}$$

The exergy loss ratio is formulated as follow:

$$y_{\mathrm{L,total}} = \frac{\dot{E}_{\mathrm{L,total}}}{\dot{E}_{\mathrm{F,total}}} \tag{3.34}$$

As a summarization of above formulations of exergetic efficiency given as follow:

$$\varepsilon_{\mathrm{total}} = \frac{\dot{E}_{\mathrm{P,total}}}{\dot{E}_{\mathrm{F,total}}} = 1 - \frac{\dot{E}_{\mathrm{D,total}} + \dot{E}_{\mathrm{L,total}}}{\dot{E}_{\mathrm{F,total}}} = 1 - y_{\mathrm{D,total}} - y_{\mathrm{L,total}} \tag{3.35}$$

3.4 Economic Analysis

Any one thermal design units needs to define cost of related project such as capital investment, fuel, operating and maintenance (O & M) fees, and others to consider as many envision to relate with economic, technological using methods for engineering

economics [40]. The most available parameter for a system design is final product cost. So, a profit need to add for final cost by some factor as demand, competition etc.

The last cost includes fixed which can be described as producing rate and variable cost connect with output volume with others (materials, labor, etc.) [48]. A well cost analysis can be done by successful project with a good estimation for an economic analysis and computing the cost of main products.

3.4.1 Time Value of Money

To decide a good plan on capital account to earn money power, when money can be in hand, it is more available than to invest it for a year. Therefore, cost analysis needs to compare with money financial state in time, a method require to calculate money on time.

Future Value: If "*P*" any unit of money are bided in for account "*i*" percent interest per period and the interest is "*n*" periods, A future value, "*F*"

$$F = P(1+i)^{n} \tag{3.36}$$

Interest is the admission paid for using arrears. Instead of this term, rate of return used for investment and annual cost for arrears capital [49].

Compounding Frequency: Time in engineering economy unit used for year. Eq. 3.36 define as "*p*" times per year ($p \geq 1$) for a total number of "*n*" years ($n \geq 1$), and "*i*" is the annual rate of return:

$$F = P\left(1+\frac{i}{p}\right)^{np} \tag{3.37}$$

The number of periods is "np" is and "*i/p*" is the rate of return per period. In this case, the annual rate of return "*i*" is known as the nominal rate of return. The effective rate of return is define as:

$$i_{\text{eff}} = \left(1+\frac{i}{p}\right)^{p} - 1 \tag{3.38}$$

If there are a continuity on money ($p \to \infty$), the future value is as:

$$F = Pe^{\text{in}} \tag{3.39}$$

Eq's (3.38) and (3.39) become as (3.40)

$$i_{\text{eff}} = e^{\text{i}} - 1 \tag{3.40}$$

Future value becomes as:

$$F = P(1 + ni_{\text{eff}}) \tag{3.41}$$

Eq's (3.37) and (3.39) are as follows:

$$F = P(1 + i_{\text{eff}})^{\text{n}} \tag{3.42}$$

Single – payment compound amount factor (SPCAF) as $(1 + i_{\text{eff}})^{\text{n}}$.

To simplify equations, money is computed for one or more years, Eq. (3.42) is used for a non-integer exponent [49].

Present Value: A present value is a good point for future assumption. If deposited at a given rate of return and compounded would yield the actual amount received at a future time. Eq. (3.42) define as Eq. (3.43) for future value:

$$\text{P} = \text{F}\frac{1}{(1 + \text{i}_{eff})^{n}} \tag{3.43}$$

The single – payment present – worth factor or the single – payment discount factor (SPDF) as $\frac{1}{(1 + \text{i}_{\text{eff}})^{\text{n}}}$. the future and present value difference name as discount, in (i_{eff}) or effective discount rate.

Annuities: An annuity is an equal amount of money transactions happened at interval periods. Annuities are used to compute the levelized costs, fuel, and so on. This is an interval from beginning to end.

A is an account earning, i_{eff} percent per period, the future sum F required at the period ends' n^{th}:

$$F = A\frac{(1+i_{eff})^n - 1}{i_{eff}} \tag{3.44}$$

The uniform – series compound – amount factor (USCAF) is $\left[(1+i_{\text{eff}})^{n} - 1\right]/i_{\text{eff}}$, and uniform – series sinking fund factor (USSFF) is reciprocal. Eq.s 3.43 and 3.44 as:

$$\frac{P}{A} = \frac{(1+i_{\text{eff}})^n - 1}{i_{\text{eff}}(1+i_{\text{eff}})^n} \tag{3.45}$$

The capital recovery factor (CRF) is:

$$\text{CRF} = \frac{A}{P} = \frac{i_{\text{eff}}(1+i_{\text{eff}})^n}{(1+i_{\text{eff}})^n - 1} \tag{3.46}$$

Capitalized Cost: The capital cost C_{FC} have not infinite life for nyears. The economic life involve the estimate period of usage [49].

3.4.2 Inflation, Escalation, and Levelization

Inflation: For any product, its price decreases or increases by policy [40]. The cost verifies by each year. Average annual inflation rate (r_{i}) is computed by average of a year.

Escalation: The annual rate of material verifies by resource reduce, demand rise, and technological development [50]. The first and second one are positive but the third one is negative effect on escalation.

The apparent escalation rate (r_{n}) is the total annual rate of change in cost and contain inflation and escalation:

$$(1+r_{\text{n}}) = (1+r_{\text{r}})(1+r_{\text{i}}) \tag{3.49}$$

Levelization: Cost escalation applies to an material over n-year as a results in a non-uniform cot disturbance by multiplied $(1+r_{\text{n}})$, r_{n} constant rate of change. The constant–escalation levelization factor (CELF) define a relationship between the first year (P_0) and an equivalent annuity (*A*), named as levelized value.

$$\frac{A}{P_0} = \text{CELF} = \frac{k(1-k^{\text{n}})}{1-k}\text{CRF} \tag{3.50}$$

where

$$k = \frac{1 + r_{\mathrm{n}}}{1 + i_{\mathrm{eff}}} \tag{3.51}$$

The overall system required economic data as annual values of carrying charges, fuel costs, raw water costs, and operating and maintenance (O&M) cocts. Those costs change by economy. The levelized cost is defined as [51];

$$A = \mathrm{CRF} \sum_{\mathrm{m}=1}^{\mathrm{n}} P_{\mathrm{m}} = \frac{i_{\mathrm{eff}} (1 + i_{\mathrm{eff}})^{\mathrm{n}}}{(1 + i_{\mathrm{eff}})^{\mathrm{n}} - 1} \sum_{\mathrm{m}=1}^{\mathrm{n}} P_{\mathrm{m}} \tag{3.52}$$

where

$$P_{\mathrm{m}} = C_{\mathrm{m}} \frac{1}{(1 + i_{\mathrm{eff}})^{\mathrm{m}}} \tag{3.53}$$

The cost rate related with capital and O&M expenses for the *k'*th component:

$$\dot{Z}_{\mathrm{k}} = \frac{\mathrm{CC_L}}{\tau} \frac{\mathrm{PEC_k}}{\sum_{\mathrm{k}} \mathrm{PEC_k}} + \frac{\mathrm{OMC_L}}{\tau} \frac{\mathrm{PEC_k}}{\sum_{\mathrm{k}} \mathrm{PEC_k}} \tag{3.54}$$

The first one defines $\dot{Z}_{\mathrm{k}}^{\mathrm{CI}}$, and other express $\dot{Z}_{\mathrm{k}}^{\mathrm{OM}}$. The levelized cost rate of the material (fuel, raw water) can be defined for all system as:

$$\dot{\mathrm{C}}_{\mathrm{EX}} = \frac{\mathrm{EXC_L}}{\tau} \tag{3.55}$$

3.5 Thermoeconomic Analysis

Cost accounting is essentially related with:

a) define the actual cost of products or services,

b) satisfy a rational basis for pricing goods and services,

c) satisfy allocating and controlling expenditures,

d) assumption on future cost [40].

This cost balance is usually defined as Eq. (3.56) for overall system at steady state:

$$\dot{C}_{\mathrm{P,TOT}} = \dot{C}_{\mathrm{F,TOT}} + \dot{Z}_{\mathrm{TOT}}^{\mathrm{CI}} + \dot{Z}_{\mathrm{TOT}}^{\mathrm{OM}} \tag{3.56}$$

The $\dot{C}_{\mathrm{P}}$ named as total rate of product expenditures and $\dot{C}_{\mathrm{F}}$ is fuel cost rate and $\dot{Z}^{\mathrm{CI}}$ is capital investment and $\dot{Z}^{\mathrm{OM}}$ is operating and maintenance. The summarization of variables is defined by $\dot{Z}$:

$$\dot{Z} = \dot{Z}^{\mathrm{CI}} + \dot{Z}^{\mathrm{OM}} \tag{3.57}$$

3.5.1 Exergy Costing

Cost is defined as value of materials [52]. Cost is a physical property. Thus, a rule has to apply for computing from data. Cost is a property that cannot be found in the product itself [40, 53]. Thermoeconomics analysis is related with degradation, exergy, quality, cost, resource, consumption and others [54].

A true value of thermodyanmic is related with heat, work and mass transfer through system. At the reality, thermoeconomics have a notification on exergy cost analysis to define its surroundings and also inefficiencies with it.

For inlet and outlet of streams related with rates of exergy transfer is defined as:

$$\dot{C}_{\mathrm{i}} = c_{\mathrm{i}}\dot{E}_{\mathrm{i}} = c_{\mathrm{i}}(\dot{m}_{\mathrm{i}}e_{\mathrm{i}}) \tag{3.58}$$

$$\dot{C}_{\mathrm{e}} = c_{\mathrm{e}}\dot{E}_{\mathrm{e}} = c_{\mathrm{e}}(\dot{m}_{\mathrm{e}}e_{\mathrm{e}}) \tag{3.59}$$

$$\dot{C}_{\mathrm{w}} = c_{\mathrm{w}}\dot{W} \tag{3.60}$$

$$\dot{C}_{\mathrm{q}} = c_{\mathrm{q}}\dot{E}_{\mathrm{q}} \tag{3.61}$$

where c_{i}, c_{e}, c_{w}, and c_{q} denote average costs per unit of exergy of matter stream at inlet and exit, power and heat respectively, and $\dot{C}_{\mathrm{i}}$, $\dot{C}_{\mathrm{e}}$, $\dot{C}_{\mathrm{w}}$ and $\dot{C}_{\mathrm{q}}$ are the corresponding cost rates, $\dot{E}_{\mathrm{i}}$ and $\dot{E}_{\mathrm{e}}$ are exergy transfers for entering and exiting streams of matter, $\dot{W}$ is power, and $\dot{E}_{\mathrm{q}}$ is the exergy transfer rate associated with heat transfer.

Finally, they are written as [40, 55, 56]:

$$\sum_{\mathrm{e}}\left(c_{\mathrm{e}}\dot{E}_{\mathrm{e}}\right)_{\mathrm{k}} + c_{\mathrm{w,k}}\dot{W}_{\mathrm{k}} = c_{\mathrm{q,k}}\dot{E}_{\mathrm{q,k}} + \sum_{\mathrm{i}}\left(c_{\mathrm{i}}\dot{E}_{\mathrm{i}}\right)_{\mathrm{k}} + \dot{Z}_{\mathrm{k}} \tag{3.62}$$

Eq. (3.46) defines exergy streams for outlet of total cost equals to total consumption (inlet cost exergy streams add capital with others). Cost relations are always defined a positive sign for all input.

The $\dot{Z}_k$ is calculated by capital investment and operating and maintenance (O&M) costs related with the k^{th} component, and then calculated the levelized values. Cost rate $\left(\dot{Z}_i\right)$ in \$/s defined by capital investment and the maintenance costs for the k^{th} component is;

$$\dot{Z}_k = \frac{Z_k (CRF)\varphi}{(N \times 3600)} \tag{3.63}$$

where

Z_k: purchase cost component (\$)

CRF: annual capital recovery factor

N: number of hours

φ : maintenance factor.

The cost of product has to allocate in money form. In reality, fluid requires to characterize its cost by a function. The function requires to calculate in terms of sensitivity. Thus, it require many manifestation for this analysis [52,57].

3.5.2 Aggregation Level Used for Exergy Costing

Due to system, the lowest aggregation level is preferred to use [40,54,57,58]. This cost is valid for all of componenets. Exergy cost is appled for component [40].

3.6 Thermoeconomic Variables for Component Evaluation

The following equations define for thermoeconomic variables:

The average unit cost of fuel, $c_{F,k}$ (i.e. $c_{F,k} = \frac{\dot{C}_{F,k}}{\dot{E}_{F,k}}$).

The average unit cost of product, $c_{P,k}$ (i.e. $c_{P,k} = \frac{\dot{C}_{P,k}}{\dot{E}_{P,k}}$).

The cost rate of exergy destruction, $\dot{C}_{D,k}$.

The relative cost difference, r_k.

The exergoeconomic factor, f_{k}.

3.6.1 Cost of Exergy Destruction

By using the specific exergetic costs related with fuel, product, and exergy loss for the *k*th component can be written as:

$$c_{\mathrm{P,k}}\dot{E}_{\mathrm{P,k}} = c_{\mathrm{F,k}}\dot{E}_{\mathrm{F,k}} - \dot{C}_{\mathrm{L,k}} + \dot{Z}_{\mathrm{k}} \tag{3.64}$$

and $\dot{E}_{\mathrm{F,k}}$:

$$c_{\mathrm{P,k}}\dot{E}_{\mathrm{P,k}} = c_{\mathrm{F,k}}\dot{E}_{\mathrm{P,k}} + \left(c_{\mathrm{F,k}}\dot{E}_{\mathrm{L,k}} - \dot{C}_{\mathrm{L,k}}\right) + \dot{Z}_{\mathrm{k}} + c_{\mathrm{F,k}}\dot{E}_{\mathrm{D,k}} \tag{3.65}$$

also $\dot{E}_{\mathrm{P,k}}$:

$$c_{\mathrm{P,k}}\dot{E}_{\mathrm{P,k}} = c_{\mathrm{F,k}}\dot{E}_{\mathrm{F,k}} + \left(c_{\mathrm{P,k}}\dot{E}_{\mathrm{L,k}} - \dot{C}_{\mathrm{L,k}}\right) + \dot{Z}_{\mathrm{k}} + c_{\mathrm{P,k}}\dot{E}_{\mathrm{D,k}} \tag{3.66}$$

$\dot{E}_{\mathrm{P,k}}$: unit cost of fuel,

$c_{\mathrm{F,k}}$: exergy destruction

and the cost of exergy destruction as follow:

$$\dot{C}_{\mathrm{D,k}} = c_{\mathrm{F,k}}\dot{E}_{\mathrm{D,k}} \tag{3.67}$$

The cost of exergy destruction define as:

$$\dot{C}_{\mathrm{D,k}} = c_{\mathrm{P,k}}\dot{E}_{\mathrm{D,k}} \tag{3.68}$$

When $\dot{E}_{\mathrm{F,k}}$ is fixed and $\dot{E}_{\mathrm{D,k}}$ reduces for system.

3.6.2 Relative Cost Difference

The relative cost difference, r_{k} is expressed as following:

$$r_{\mathrm{k}} = \frac{c_{\mathrm{P,k}} - c_{\mathrm{F,k}}}{c_{\mathrm{F,k}}} \tag{3.69}$$

Eq. (3.35) is rewritten to show the real cost sources related with the *k*th component, by using Eq. (3.57) and (3.65) and as taking $\dot{C}_{\mathrm{L,k}} = 0$:

$$r_{\mathrm{k}} = \frac{c_{\mathrm{F,k}}\left(\dot{E}_{\mathrm{D,k}} + \dot{E}_{\mathrm{L,k}}\right) + \left(\dot{Z}_{\mathrm{k}}^{\mathrm{CI}} + \dot{Z}_{\mathrm{k}}^{\mathrm{OM}}\right)}{c_{\mathrm{F,k}}\dot{E}_{\mathrm{P,k}}} \tag{3.70}$$

also written Eq. (3.70):

$$r_{\mathrm{k}} = \frac{1-\varepsilon_{\mathrm{k}}}{\varepsilon_{\mathrm{k}}} + \frac{\dot{Z}_{\mathrm{k}}^{\mathrm{CI}} + \dot{Z}_{\mathrm{k}}^{\mathrm{OM}}}{c_{\mathrm{F,k}}\dot{E}_{\mathrm{P,k}}} \tag{3.71}$$

3.6.3 Exergoeconomic Factor

As it is seen from the Eq's (3.70) and (3.71), it is classified into two groups as non-exergy related costs (capital investment, and operating and maintenance expenses), and other is exergy destruction and exergy loss. By calculating exergoeconomic factor, f_{k} as follow:

$$f_{\mathrm{k}} = \frac{\dot{Z}_{\mathrm{k}}}{\dot{Z}_{\mathrm{k}} + c_{\mathrm{F,k}}\left(\dot{E}_{\mathrm{D,k}} + \dot{E}_{\mathrm{L,k}}\right)} \tag{3.72}$$

A exergoeconomic factor is computed to advise for low price and also improving component efficiency (reducing the exergy destruction). Otherwise, a high value gives an idea to decrease investment costs for components.

3.7 The Specific Exergy Costing (SPECO) Method

The costs is related with matter and energy stream for calculating cost balance of each components with auxiliary equations. By doing an assumption for entering and existing streams for sufficient values which is higher than one [49,56,57].

Many applications for formulation and auxiliary equations have been used in literature. They are seperated into two sections as: a) exergoeconomic methods [40,51,53,59-68] is define as product streams cost, components evaluation, with iterative optimization. b) The Lagrangian-based method [69-79] goals to optimize

overall system by computing marginal costs. The exergy were figured out auxiliary costing equations by fexplicitly formulation and doing assumptions used from experience, postulates for system analysis.

Other method is LIFO (Last In First Out) [80,81]. This method used as fuels, products, and costs are expressed by defining exergy and cost values and also removals from each material and energy stream. By the way, “local average costs” are defined because of the cost per exergy unit which computed removed exergy units cost to upstream components [81]. The SPECO (specific exergy costing) method is used for components of system as removals of exergy and cost.

The SPECO method can be used on exergy streams instead of material and energy streams directly [57]. By using systematically, fuel and product of the components with auxiliary costing equations is used to compute average costs (AVCO method) or local average costs (LIFO approach).

Lagrangian-based method is used to calculate cost for streams. The cost balances and auxiliary equations is calculated by partial derivatives in the Lagrangian-based approaches.

The SPECO is define in three sections:
Step 1- identification of exergy streams: Firstly, exergy of each components such as thermal, mechanical, and chemical exergies is defined for accuracy result of calculation.
Step 2- definition of fuel and product: The fuel is related with exergy of all components at inlet (including the exergy streams supplied to the component) adding all the exergy decreases between inlet and outlet minus all the exergy rises (between inlet and outlet) according to component purpose.
Step 3- cost equations: Exergoeconomics define exergy relation with cost for system and its surroundings with inefficient source [40]. The Equations mentioned above for using in calculation are given in next chapters.

3.7.1 The $\dot{F}$ and $\dot{P}$ Principles

The $\dot{F}$ (fuel) principle related with exergy difference between inlet and outlet is defined in the definition of the fuel. The $\dot{F}$ associated with total cost and removal exergy have to be equal to removed exergy cost.

The $\dot{P}$ (product) principle relates with exergy to an exergy stream for supplication. Aggregation level effects results, thus it should be in lower level [40].

CHAPTER 4

THERMODYNAMIC AND THERMOECONOMIC ANALYSES

4.1 Introduction

The energy usage is measurement of defining and expressing energy systems which can be misconception and bemused. Exergy is a good tool for developing and getting meaningful of energy system [42].

Energy and exergy values, exergy efficiency and exergy destructions for the each component of the SAHP system are computed and tabulated.

This chapter included thermodynamic analyses of the SAHP is related with previous chapter for using the methodologies. The results are found and discussed.

4.2 Description of Solar Assisted Heat Pump System (SAHP)

The SAHP was designed, and working conditions in detail are explained and shown in Figure 4.1. Heat pump used in the heating system is a water to water type heat pump, and assumptions are given as follow:

- Isoentropic efficiency of compressor is 80 %, works under adiabatic state.
- Evaporator and condenser circulated flowing fluid (R-22) at constant pressure.
- There is no pressure losses in system.
- The R-22 of evaporator and condenser exist are saturated vapor or liquid states.

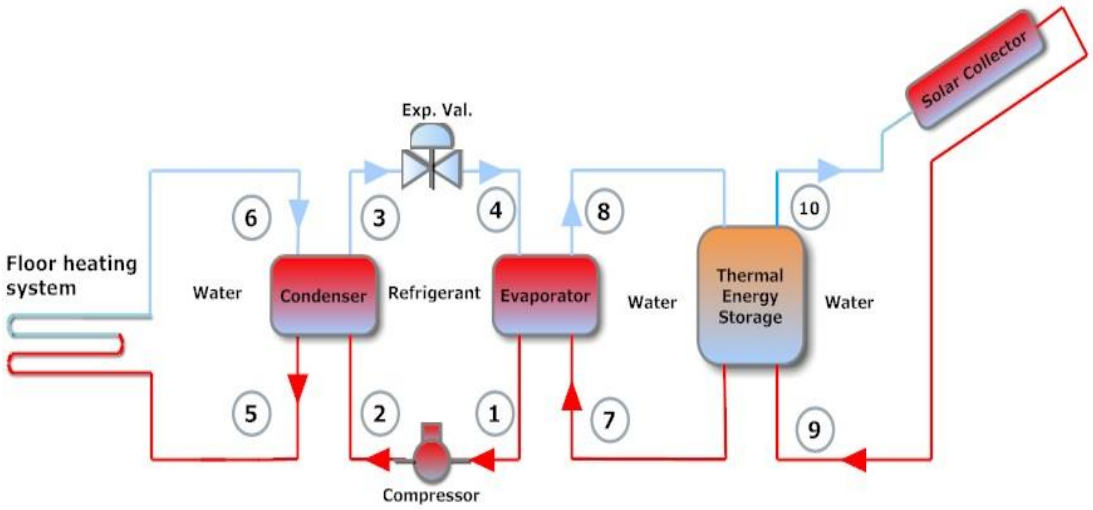

Figure 4.1. The SAHP system (Modified from ref. [21])

4.3 Exergy Relations

Exergy relations for each components of the system was satisfied in Chapter 3. The mass, energy and exergy equaitions are given as well as exergy destructions and exergy efficiencies. State numbers are shown in Fig. 4.1.

The exergy relations applied to the SAHP with daily energy storage tank (given in Fig 1) are;

$$\dot{E}x_r + \dot{W}_{comp} + \dot{E}x_{wci} - \dot{E}x_{wco} - \dot{E}x_{loss} = 0 \tag{4.1}$$

where $\dot{E}x_r$ is the exergy content of the incoming solar radiation, $\dot{W}_{comp}$ is the work consumed by the compressor and total work consumed by the pumps oriented in the system. $\dot{E}x_{loss}$ denotes the total exergy destruction rate of the system. $\dot{E}x_{wci}$ and $\dot{E}x_{wco}$ are the inlet and outlet exergy rates of the water stream through the condenser. The second law efficiency of the whole system is the ratio of the total exergy gained by the water, circulated through the condenser, to the total exergy entering the system (i.e. works and solar irradiance).

The total exergy gained by the system is;

$$\Delta\dot{E}x = \dot{E}x_{wco} - \dot{E}x_{wci} \tag{4.2}$$

The total entering exergy rate is;

$$\dot{E}x_{in} = \dot{E}x_r + \dot{W}_{pump} \tag{4.3}$$

Therefore, second law of efficiency becomes as;

$$\eta_{II} = \frac{\Delta \dot{E}}{\dot{E}_r + \dot{W}_{pump}} \tag{4.4}$$

Combining Equations (4.1) and (4.4), the second law efficiency may also be expressed as;

$$\eta_{II} = 1 - \left[\frac{\dot{E}x_{loss}}{\dot{E}x_r + \dot{W}_{pump}} \right] \tag{4.5}$$

4.4 Exergy Relations for System Components

Irreversibitlity or the exergy destruction of the each individual component (i.e. Compressor, TES, Solar collectors) is computed by applying exergy balance over them so that their acts on the total exergy destruction of the SAHP can be expressed. The exergy balance relation for any steady state systems is;

$$\dot{E}x_{in} - \dot{E}\,x_{out} - \dot{E}x_{loss} = 0 \tag{4.6}$$

Since exergy, like energy, can be transferred to or from a system in three forms: heat, work, and mass flow, first two terms of the Equation (4.6) includes them if they exist. By undertaking the Eequation (4.6) exergy balance for insulated condenser is;

$$\left(\dot{E}\,x_2 + \dot{E}\,x_6 \right) - \left(\dot{E}x_3 + \dot{E}x_5 \right) - \dot{E}x_{dest} = 0 \tag{4.7}$$

where $\dot{E}\,x_2$ and $\dot{E}x_3$ are entering and leaving exergies to/from the condenser due to mass flow of the R22, and $\dot{E}\,x_5, \dot{E}x_6$ are entering and leaving exergies to/from the condenser due to mass flow of the water, respectively. Equation (7) can be written in detail as;

$$\dot{m}_{wc}.\left[\left(h_5 - h_6 \right) - T_o.\left(s_5 - s_6 \right) \right] + \dot{m}_r.\left[\left(h_2 - h_3 \right) - T_o.\left(s_2 - s_3 \right) \right] - \dot{E}x_{dest} = 0 \tag{4.8}$$

Here, $\dot{m}_{wc}$ and $\dot{m}_r$ are mass flow rate of the water and mass flow rate of the refrigerant flowing through condenser, respectively. Mass flow rate of the water through the condenser are measured, and the mass flow rate of the refrigerant is found by applying first law of the thermodynamic.

Exergy balance for evaporator is;

$$\left(\dot{E}x_4 + \dot{E}x_7\right) - \left(\dot{E}x_1 + \dot{E}x_8\right) - \dot{E}x_{dest} = 0 \tag{4.9}$$

The two terms at the right hand side of the Equation (4.9) denote the exergy that enters the evaporator and leaves the evaporator, respectively, due to mass flow of R22 and water. And this equation is extracted as,

$$\dot{m}_{we}.\left[\left(h_7 - h_8\right) - T_o.\left(s_7 - s_8\right)\right] + \dot{m}_r.\left[\left(h_2 - h_3\right) - T_o.\left(s_2 - s_3\right)\right] - \dot{E}x_{dest} = 0 \tag{4.10}$$

Here, $\dot{m}_{we}$ and $\dot{m}_r$ are mass flow rate of the water and mass flow rate of the refrigerant flowing through evaporator, respectively. Mass flow rate of the water through the evaporator was measured.

Exergy balance for the compressor is,

$$\left(\dot{E}x_1 + W_{comp}\right) - \left(\dot{E}x_2 + \left(1 - \frac{T_o}{T_b}\right)\dot{Q}_{Loss}\right) - \dot{E}x_{dest} = 0 \tag{4.11}$$

and the clear form of Equation (4.10) is

$$\dot{m}_r.\left[\left(h_1 - h_2\right) - T_o.\left(s_2 - s_1\right)\right] + \dot{W}_{comp} - \left(1 - \frac{T_o}{T_b}\right).\dot{Q}_{Loss} - Ex_{dest} = 0 \tag{4.12}$$

Q_{Loss} is the heat loss from the compressor, and it is computed from the energy balance or the first law of the thermodynamic. First law relation for the compressor is,

$$\dot{E}_{in} - \dot{E}_{out} = 0 \tag{4.13}$$

$$\dot{m}_r.\left(h_2 - h_1\right) + \dot{W}_{comp} - \dot{Q}_{Loss} = 0 \tag{4.14}$$

and the exergy balance for the expansion valve is

$$\dot{E}x_3 - \dot{E}x_4 - \dot{E}x_{loss} = 0 \tag{4.15}$$

and clear form is

$$\dot{m}_r.\left[\left(h_3 - h_4\right) - T_o.\left(s_3 - s_4\right)\right] - \dot{E}x_{dest} = 0 \tag{4.16}$$

Inner energy storage having larger volume is used as heat sink for the evaporator. And the energy gained from the solar collector is stored in it. Thus exergy analysis of the thermal energy storage (given in fig 1) is done based on it. General expression of the exergy balance for it is;

$$\left(\dot{E}\,x_8 - \dot{E}\,x_7\right) + \left(1 - \frac{T_o}{T_b}\right).\dot{Q}_{st} - \dot{E}\,x_{dest} = E\,x_2 - E\,x_1 \tag{4.17}$$

where:

$$\dot{\mathrm{E}}\,\mathrm{x}_8 - \dot{\mathrm{E}}\,\mathrm{x}_7 = \dot{\mathrm{m}}_{\mathrm{we}}.\left[\left(\mathrm{h}_8 - \mathrm{h}_7\right) - \mathrm{T}_{\mathrm{o}}.\left(\mathrm{s}_8 - \mathrm{s}_7\right)\right] \tag{4.18}$$

and

$$E\,x_2 - E\,x_1 = m_{st.}\left[\left(u_2 - u_1\right) - T_o\left(s_2 - s_1\right)\right] \tag{4.19}$$

Equation (4.17) is valid if the heat pump consume energy from storage and the solar collector charged the storage simultaneously. $\dot{Q}_{st}$ denotes the heat transfer from the outer storage to the inner storage. $m_{st.}$ is the mass of water in the inner storage. In the event that the solar collector charges the energy into storage, and heat pump is not running, Equation (4.14) reduces to the following equation.

$$\left(1 - \frac{\mathrm{T_o}}{\mathrm{T_b}}\right).\dot{\mathrm{Q}}_{\mathrm{st}} - \dot{\mathrm{E}}\,\mathrm{x}_{\mathrm{dest}} = \mathrm{E}\,\mathrm{x}_2 - \mathrm{E}\,\mathrm{x}_1 \tag{4.20}$$

During nights, there is no solar radiation means that there is no energy charging into the storage, and because of the cold whether, heat pump runs. In these situations, Equation (4.14) becomes as;

$$\left(\dot{\mathrm{E}}\,\mathrm{x}_8 - \dot{\mathrm{E}}\,\mathrm{x}_7\right) - \dot{\mathrm{E}}\,\mathrm{x}_{\mathrm{dest}} = \mathrm{E}\,\mathrm{x}_2 - \mathrm{E}\,\mathrm{x}_1 \tag{4.21}$$

The exergy balance for the solar collectors is;

$$Ex_r + Ex_{10} - Ex_9 - Ex_L = 0 \tag{4.22}$$

Where:

$$\dot{E}\,x_r = I_t.A_c.\tau\alpha.\left(1 - \frac{T_o}{T_p}\right) \tag{4.23}$$

$$\dot{E}\,x_L = \dot{Q}_{Loss}.\left(1 - \frac{T_o}{T_p}\right) \tag{4.24}$$

$$\dot{E}_{10} - \dot{E}_9 = \dot{m}_b .[cp_b .(T_9 - T_{10})] - T_a .cp_b .\ln\left(\frac{T_9}{T_{10}}\right) \quad (4.25)$$

T_p is the plate temperature, and can be evaluated by the first law of the thermodynamic.

$$Q_s = Q_u + Q_l \quad (4.26)$$

$$Q_s = I_t .A_c .\tau\alpha \quad (4.27)$$

$$Q_l = U_L .A_c .(T_p - T_0) \quad (4.28)$$

$$Q_u = \dot{m}_b .cp_b .(T_9 - T_{10}) \quad (4.29)$$

$$T_p = \frac{Q_s - Q_u}{U_L .A_c} + T_o \quad (4.30)$$

4.5 Exergy Analysis Results

The SAHP heating stem given in Figure 1 was run during the heating period of February 14 through April 30, 2002. Results are used to determine exergy and the second law analysis of the SAHP. An illustrative example will have given for March 1, 2012 and March 29,2012. Detailed energy and first law analysis were given in Yumrutaş and Kaska [87].

Variation of daily average values of outside air temperature, storage temperature (water temperature in storage tank) and solar radiation data on horizontal surface are seen from Figure 2, 3 and 4 for the heating period of February 14 through April 30, 2002. Daily averaged minimum and maximum values of solar insulation incident on horizontal surface are 2.7 and 21.2 MJ/m^2. Several sunny days are observed in February, however, oscillation in solar insolation is seen due to cloud movements during March and April. Monthly average values of solar radiation on horizontal surface are 10.8, 11, and 13.2 MJ/m^2 for February, March and April months, respectively. In cloudy days, the solar insolation is usually less than 6 MJ/m^2.

Daily averaged minimum and maximum ambient air temperatures are calculated from measured values as 7.8 and 16.1^{o}C. While the minimum and maximum storage temperatures are 11.3 and 39.7 ^{o}C during experimental period. Monthly averages of the ambient air temperature, the storage temperature obtained to be 8.86, 10.28, and

12.47°C, 26.9, 25.8, and 23.8°C respectively. Since the storage temperature is generally greater than the ambient air temperature, heat extraction from the storage is more efficient operation for the heat pump cycle. The storage temperature is function of solar energy charged to the storage tank and energy extraction from the tank for use in heat pump cycle.

Coefficient of performance of the heat pump (COP_{HP}) and that of the overall system (COP_S) for the months of February, March, and April are shown graphically in Yumrutaş and Kaska [87]. Minimum and maximum COP_{HP} values are 2.5 and 3.1 in February, 2.4 and 3.6 in March, and 3.2 and 2.6 in April. The ranges for COP_S are 2.0 - 2.5, 1.9 - 3.0, and 2.1 - 2.7, respectively. Monthly average values are 2.9, 3.2, and 3.0 for COP_{HP}, 2.3, 2.6, and 2.5 for COP_S, respectively.

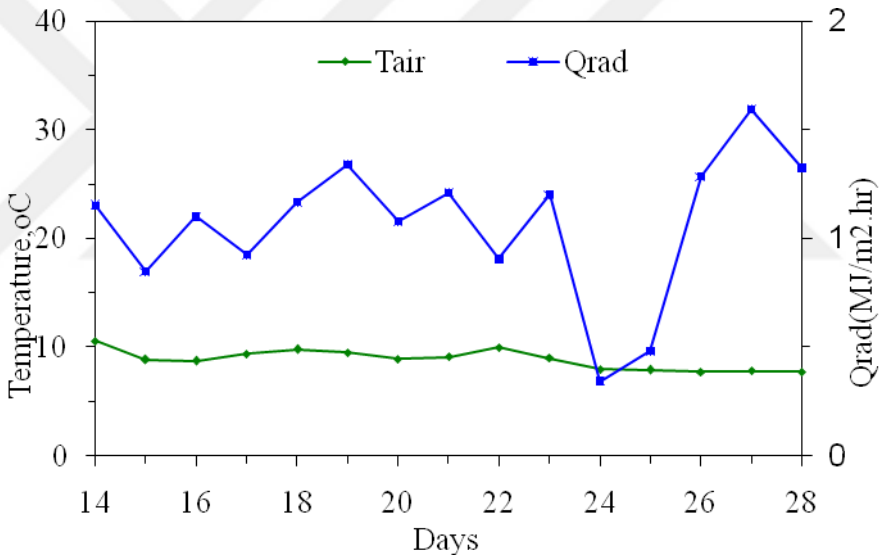

Figure 4.2 Daily average values of outside air and solar radiation data on the collector surface for February [87]

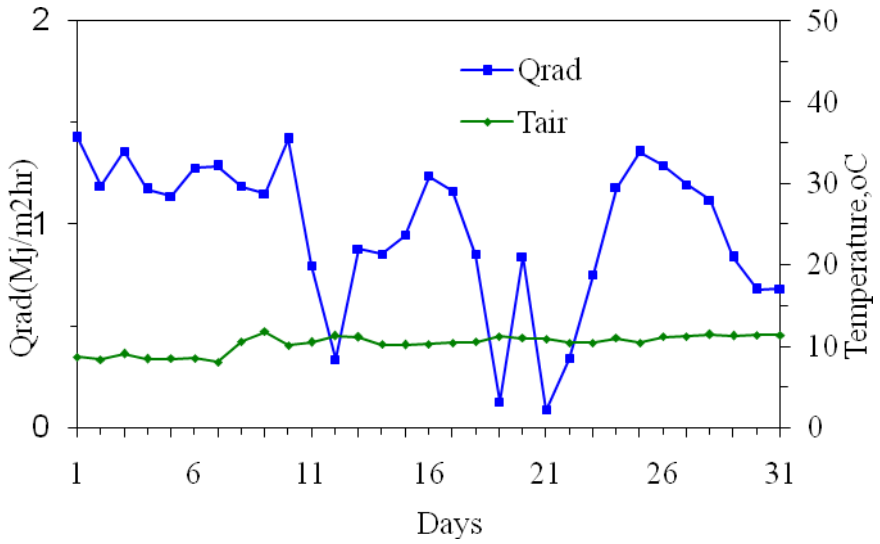

Figure 4.3 Daily average values of outside air and solar radiation data on the collector surface for March [87]

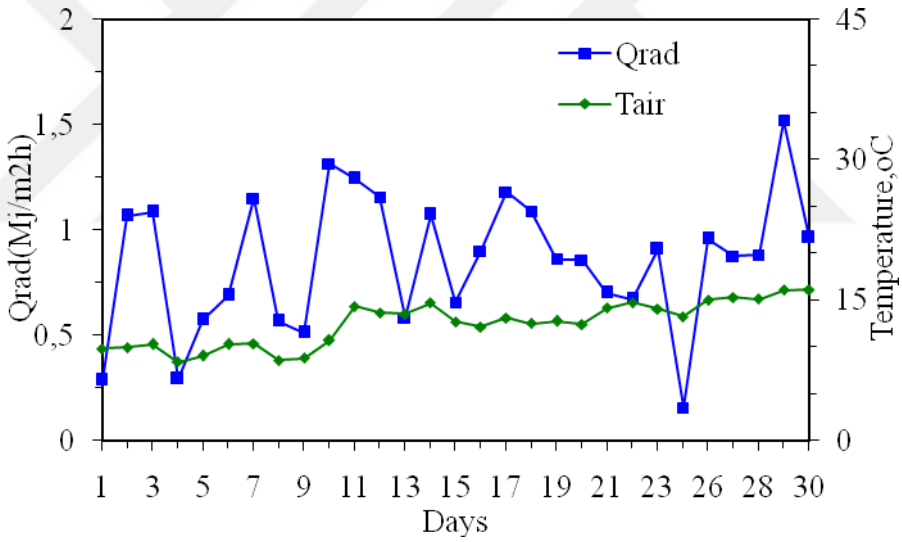

Figure 4.4 Daily average values of outside air and solar radiation data on the collector surface for April [87]

An illustrative example is given in Table 4.3. Exergy analysis of the system is done by applying model given previously. Table 4.1 includes measured data of pressure, temperature and mass flow rate of the water loops in the system. While temperatures and pressures of refrigerant loop are measured, mass flow rate of the refrigerant given in the table was calculated from energy balance for the evaporator and the condenser. Corresponding specific enthalpy, specific entropy, specific energy, specific exergy, energy rate, exergy rate are also included in the table. State entropy

values of the water-antifreeze solution were calculated by using constant specific heat values for %20 ethylene-glycol in water. Three different types of refrigerants are used for comparing exergy rate and other calculation. For R22, minimum and maximum values at inlet of condenser and compressor are as 0,376 and 0,713 kW, given in Table 4.1. For R744, minimum and maximum values at inlet of condenser and compressor are as 1,879 and 2,57 kW, presented in Table 4.2. For R134-a, minimum and maximum values at inlet of condenser and compressor are as 1,063 and 1,403 kW, given in Table 4.3. As a result of thermodynmic tables, each of the refrigerant verification effect on verifiraction of enthalpy and entropy that cause verification of exergy rate. The comprosser and condenser are importnat components of the heat pump. Additionally, condenser causes to transfer heat from TES tank to the heat pump, and the comprosser causes to run heat pump system by auxilary work. The highest rate of exergy rate founds at Table 4.1 for heat pump system. This is an idea for following calculation that R134a has the high utilization between other refrigerant.

Table 4.1 Thermodynamic properties used in SAHP system for R-22

State	Fluid	Pressure P (kPa)	Temper-ature, T (°C)	Mass flowrate, $\dot{m}$ (kg/s)	Enthalpy, h (kJ/kg)	Entropy, s (kJ/kgK)	Exergy rate, $\dot{E}$ (kW)
0	R22	101,3	19	---	425,3	1,969	---
0	Water	101,3	19	---	79,75	0,2819	---
1	R22	531	27,4	0,0097	424,4	1,813	0,403
2	R22	1544	92	0,0097	462,1	1,834	0,713
3	R22	1544	40,25	0,0097	250,1	1,168	0,4133
4	R22	531	1,979	0,0097	250,1	1,182	0,376
5	Water	150	21	0,027	88,02	0,3104	0,01387
6	Water	150	39,17	0,027	164,2	0,5611	0,2284
7	Water	150	28,2	0,036	118,3	0,4116	0,1173
8	Water	150	17,01	0,036	71,33	0,2533	0,02303
9	Water	150	31,62	0,1	115,1	0,4588	0,1793
10	Water	150	56,31	0,1	235,8	0,7845	0,2711

Table 4.2 Thermodynamic properties used in SAHP system for R744

State	Fluid	Pressure, P (kPa)	Tempe-rature, T (°C)	Mass flow rate, $\dot{m}$	Enthalpy, h (kJ/kg)	Entropy, s kJ/kgK)	Exergy rate, $\dot{E}$ (kW)
0	R744	101,3	19	---	-6,026	-0,01939	---
0	Water	101,3	19	---	79,75	0,2819	---
1	R22	531	27,4	0,009	-2,892	-0,3176	1,879
2	R22	1544	92	0,009	80,01	-0,2749	2,57
3	R22	1544	40,25	0,009	-0,5477	-0,5028	2,39
4	R22	531	1,979	0,009	-0,5477	-0,3099	1,881
5	Water	150	21	0,027	88,02	0,3104	0,01387
6	Water	150	39,17	0,027	117	0,4072	0,08445
7	Water	150	28,2	0,036	70,69	0,2506	0,0197
8	Water	150	17,01	0,036	71,33	0,2533	0,02303
9	Water	150	31,62	0,1	115,1	0,4588	0,1793
10	Water	150	56,31	0,1	113,4	0,3955	0,03908

Table 4.3 Thermodynamic properties used in SAHP systemforR134-a

State	Fluid	Pressure P (kPa)	Tempe-rature T (°C)	Massflowrate, $\dot{m}$ (kg/s)	Enthalpy, h (kJ/kg)	Entropy, s (kJ/kgK	Exergy rate, $\dot{E}$ (kW)
0	R134a	101,3	19	---	271,3	1,088	---
0	Water	101,3	19	---	79,75	0,2819	---
1	R22	531	27,4	0,0097	269,8	0,9555	1,165
2	R22	1544	92	0,0097	299	0,9724	1,403
3	R22	1544	40,25	0,0097	108,5	0,3943	1,081
4	R22	531	1,979	0,0097	108,5	0,4013	1,063
5	Water	150	21	0,027	88,02	0,3104	0,01387
6	Water	150	39,17	0,027	156,4	0,5363	0,202
7	Water	150	28,2	0,036	114,8	0,4	0,1051
8	Water	150	17,01	0,036	71,33	0,2533	0,02303
9	Water	150	31,62	0,1	115,1	0,4588	0,1793
10	Water	150	56,31	0,1	226,8	0,7571	0,2494

Figures 5-8 show variation of exergy destructions and temperatures of the heat pump systems run during March 1,2001. Discontinuity in the graphs shows the hours at which heat pump does not works. Figure 5 shows variation of exergy destruction and variations of T1, T4, T7, T8 temperatures which are inlet and outlet temperatures of refrigerant and water respectively. Average exergy destruction occurred in evaporator is calculated as 0.1445 kW. Averages values of the temperatures T1, T4,

T7 and T8 are 33.3 ^{o}C,3.7 ^{o}C,33,9 ^{o}C and 22,5 ^{o}C respectively. Variations of inlet and outlet water temperature, inlet and outlet refrigerant temperatures and exergy destruction occurred in the condenser are given in Figure 4.6. Avrage exergy destruction in the condenser is 0.1033kW. Averages values of the temperatures T2, T3, T5 and T6 are 101.1^{o}C, 40.8 ^{o}C,21.45 ^{o}C and 41,57 ^{o}C respectively. Average exergy destruction rate of the compressor and expansion valves are 0.32kW and 0.041 kW respectively. Depend on the results, exergy destruction rate occurred in the heat pump components can be set in order from highest to lowest as compressor, evaporator, condenser and expansion valve. This result is convenient the result given in [88]. As a conclusion of Figure 5-8, 50 measurements were done for a day. All of taken data from real system close to each other that shows accuracy and sensivity of the heat pump with minimum error. Fault of the system or when the desired room temperature satisfied caused to stop system. Thus, discontinuity was seen from Figures 5.8.

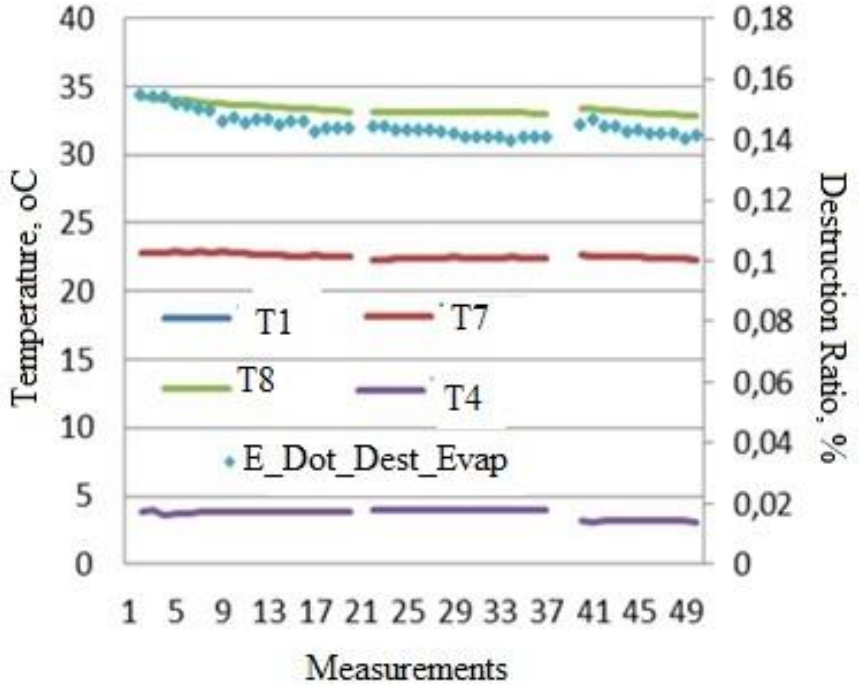

Figure 4.5 Exergy destructions and temperatures of evaporator

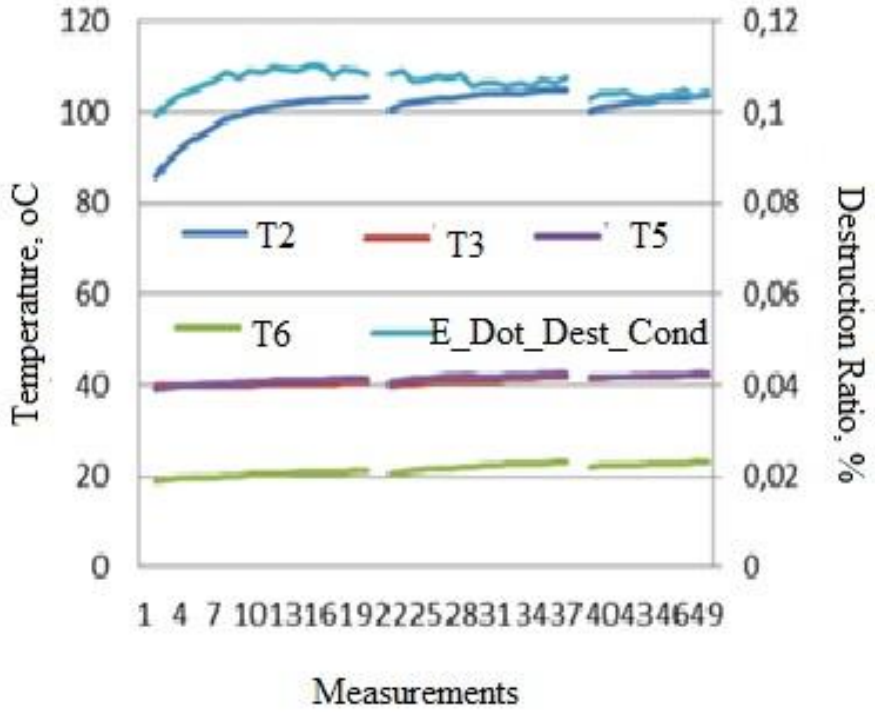

Figure 4.6 Exergy destructions and temperatures of condenser

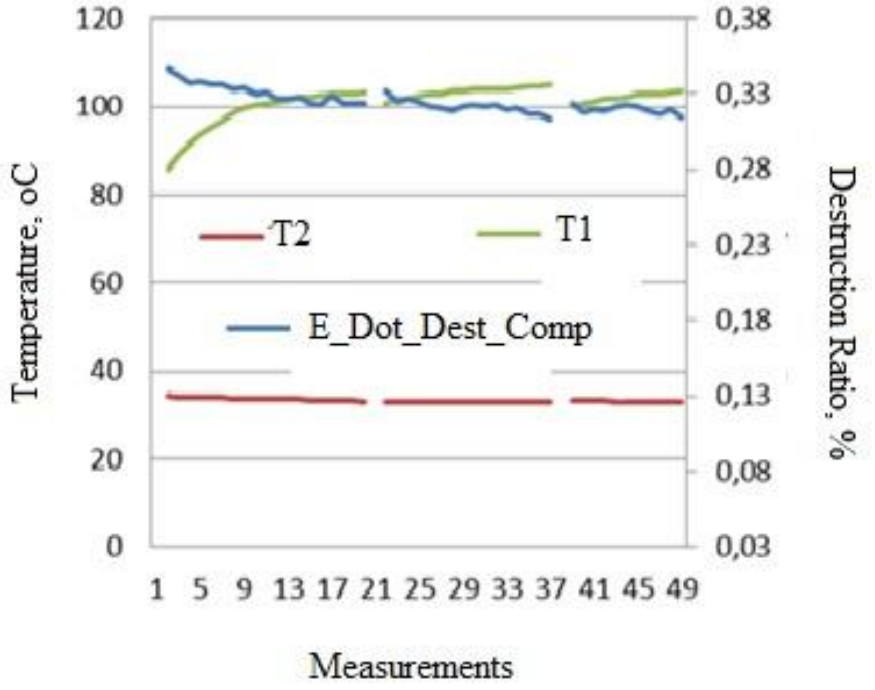

Figure 4.7 Exergy destructions and temperatures of compressor

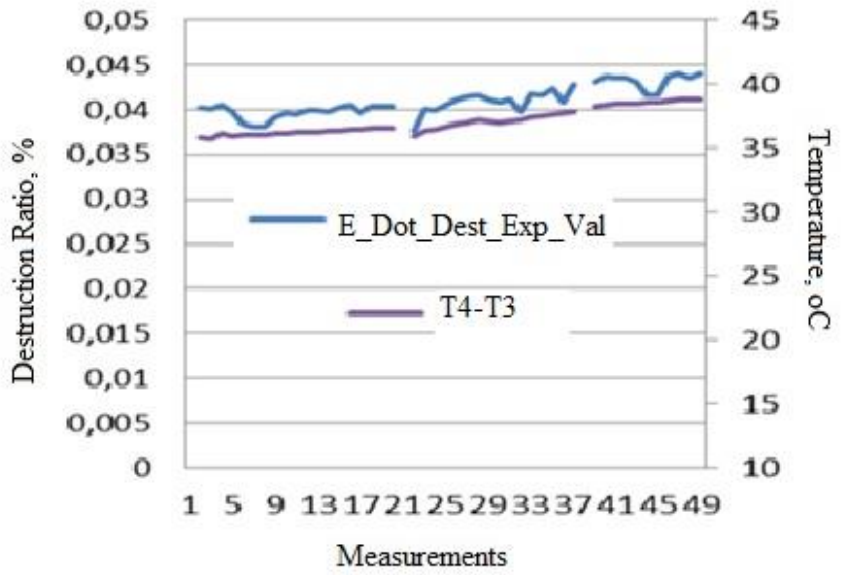

Figure 4.8 Exergy destructions and temperatures of expansion valve

The rates of exergy destructions with fuel and products rates of exergy declared in Tables 4.4-4.6. For R22, the highest exergetic fuel and product rate are 0,9414 and 0,4272 kW at compressor because of energy consumption. The lowest exergy destruction rate is obtained for the compressor as 0,05506 kW. The highest exergy destruction rate is seen at evaparator because of insulation. The highest exergetic efficiency is obtained at expansion valve at a rate of 90,98. For R744, the highest exergetic fuel rate is 1,593 for condenser, because of well heat transfer and highest exergetic product rate is 1,509 kW at compressor, because of energy consumption. The lowest exergy destruction rate is obtained for the expansion valve as 0,5089 kW. The highest exergy destruction rate is seen at expansion valve because of pressure drop. The highest exergetic efficiency is obtained at evaparator at a rate of 99,30. For R134a, the highest exergetic fuel rate is 1,605 for condenser because of well heat transfer and highest exergetic product rate is 1,403 kW at compressor because of energy consumption. The lowest exergy destruction rate is obtained from expansion valve as 0,01844 kW. The highest exergy destruction rate is seen at condenser (0,1338%) because of high heat loss. The highest exergetic efficiency is obtained at expansion valve at a rate of 98,29.

Table 4.4 Thermodynamic analysis of SAHP system for (fuel&product of R-22)

Components	$\dot{E}_F$ (kW)	$\dot{E}_P$ (kW)	$\dot{E}_D$ (kW)	$\dot{E}_L$ (kW)	y^* (%)	y (%)	ε (%)
Compressor	0,403	0,713	0,05506	0,365	0,8592	0,0236	56,29
Condenser	0,9414	0,4272	0,08512	0,4291	1,328	0,03647	45,37
Evaporator	0,4933	0,38	0,1133	0	1,768	0,04855	77,03
Expansion Valve	0,4133	0,376	0,03729	0	0,5819	0,01598	90,98
Thermal Storage	0,01387	0,2284	0,5907	0	3,348	0,09195	16,47
Solar Collector	0,2711	0,1793	0,4504	0	7,028	0,193	66,14
Heating System	0,2023	0,3884	0,2146	0	9,217	0,2531	1,919

Table 4.5 Thermodynamic analysis of SAHP system for (fuel&product of R744)

Components	$\dot{E}_F$ (kw)	$\dot{E}_P$ (kW)	$\dot{E}_D$ (kW)	$\dot{E}_L$ (kW)	y^* (%)	y (%)	ε (%)
Compressor	0,8173	1,509	0,1127	0,8041	0,1699	0,02567	54,71
Condenser	1,593	1,342	0,1097	0,1412	0,1655	0,025	84,25
Evaporator	0,7999	0,7943	0,005583	1,016E-20	0,008421	0,001272	99,30
Expansion Valve	1,328	0,8196	0,5089	0	0,7676	0,1159	61,69
Thermal Storage	0,01387	0,08445	0,07058	0	0,1065	0,01608	6,087
Solar Collector	0,03908	0,1793	0,2184	0	0,3294	0,04975	4,588
Heating System	0,2023	0,01939	0,2217	1,355E-20	0,3345	0,05051	0,0958

Table 4.6 Thermodynamic analysis of SAHP system for (fuel&product of R134a)

Components	$\dot{E}_F$ (kw	$\dot{E}_P$ (kW)	$\dot{E}_D$ (kW)	$\dot{E}_L$ (kW)	y^* (%)	y (%)	ε (%)
Compressor	1,165	1,403	0,04461	0,2826	0,4759	0,008781	83,05
Condenser	1,605	1,095	0,1338	0,3763	1,428	0,02634	68,22
Evaporator	1,168	1,142	0,02568	6,776E-20	0,274	0,005055	97,80
Expansion Valve	1,081	1,063	0,01844	0	0,1967	0,003629	98,29
Thermal Storage	0,01387	0,202	0,1882	0	2,007	0,03703	14,56
Solar Collector	0,2494	0,1793	0,4287	0	4,573	0,08438	0,7189
Heating System	0,2023	0,3546	0,5569	0	5,94	0,1096	1,752

CHAPTER 5

THERMOECONOMIC ANALYSIS

5.1 Introduction

Thermoeconomic analysis fort he SAHP heating system are given in this section. Thermoeconomic is a kind of engineering and economic analysis also, optimizing system [89-91]. The objectives of thermoeconomic analysis contain below:

- Computing individually costs of each parts,
- Figure out cost analysis of whole system
- Optimizing specific points in an any part,
- Optimizing the whole system.

This study presents specific exergy cost (SPECO) method to define design parameters for each components. The method defines in three steps which are: a) exergy exergy streams, b) fuel and product for each equipment, c) accommodate cost equation [92-95].

5.2 Thermoeconomic Analysis

The SAHP system life is designed for 15 years. The system works at 3930 for full operation, interest 0.06, salvage value 0.2, return factor 0.05. Present Worth Factor (PWF) with Capital Recovery Factor (CRF) are calculated for each equipment of the system[96-98].

The SPECO is used to figure out cost and exergy of SAHP system. Exergetic cost relation and other equations are written from Eq's. (5.1) through (5.9) referred to system [99-100]. The thermoeconomic parameters are given in Tables 5.3.

Compressor

$$\dot{C}_{W\,COMP} + \dot{Z}_{COMP} = \dot{C}_2 - \dot{C}_1 \tag{5.1}$$

Condenser

$$(\dot{C}_6 - \dot{C}_5) + \dot{Z}_{CON} = \dot{C}_2 - \dot{C}_3 \tag{5.2}$$

$$\frac{\dot{C}_2}{E_2} = \frac{\dot{C}_3}{E_3}(P) \tag{5.3}$$

Expansion Valve

$$\dot{C}_3 + \dot{Z}_{EXP} = \dot{C}_4 \tag{5.4}$$

Evaporator

$$(\dot{C}_8 - \dot{C}_7) + \dot{Z}_{CON} = \dot{C}_4 - \dot{C}_1 \tag{5.5}$$

$$\frac{\dot{C}_1}{E_1} = \frac{\dot{C}_4}{E_4}(P) \tag{5.6}$$

Thermal Storage

$$(\dot{C}_8 - \dot{C}_7) + \dot{Z}_{THER} = \dot{C}_{10} - \dot{C}_9 \tag{5.7}$$

$$\frac{\dot{C}_9}{E_9} = \frac{\dot{C}_{10}}{E_{10}}(P) \tag{5.8}$$

Solar Collector

$$\dot{C}_{10} + \dot{Z}_{THER} = \dot{C}_9 \tag{5.9}$$

$$\frac{\dot{C}_9}{E_9} = \frac{\dot{C}_{10}}{E_{10}}(P) \tag{5.10}$$

Heating System

$$\dot{C}_6 + \dot{Z}_{THER} = \dot{C}_5 \tag{5.11}$$

$$\frac{\dot{C}_6}{E_6} = \frac{\dot{C}_5}{E_5}(P) \tag{5.12}$$

Formulations are presented in previous chapters, also related with CI and O&M costs for all equipments of system shown in Table 5.1. The O&M costs are assumed as non, because of results are closer to zero. The highest CI is for condenser, and then comprosser. Both of them are main components of heat pump and whole system. The lowest one is expansion valve.

Table 5.1 The capital investment and O&M costs for SAHP equipments [101-104].

Components	PEC (€/h)	Z_k^{CI} (€/h)	Z_k^{OM} (€/h)	Z_k^{T} (€/h)
Compressor	1142	0.256	---	0.256
Condenser	2088	0.485	---	0.485
Expansion	15	0.003	---	0.003
Evaporator	2088	0.148	---	0.148
Thermal	650	0.151	---	0.151
Solar	657	0.152	---	0.152
Heating	1000	0.232	---	0.232
Total Cost	7640	1.454	---	1.454

The rates of exergy flow, cost flow and the unit exergy costs are shown in Tables 5.2-5.4. For R22, the highest unit exergy cost and cost flow rate are found as 0,0059 €/kWh for exit of compressor and 0,0156 €/kWh for exit of evaparator. For R744, the highest unit exergy cost and cost flow rate are found as 0,0275 €/kWh for and 0,01464 €/kWh for inlet of compressor. For R134a, the highest unit exergy cost and cost flow rate are found as 0,017 €/kWh for exit of compressor and 0,016 €/kWh for exit of expansion valve. Because of high capital invesment.

Table 5.2 The exergy flow rates, cost flow rates and unit exergy costs of SAHP for R22.

State No	$\dot{E}(kW)$	$\dot{c}(€/KWh)$	$\dot{C}(€/h)$
1	0,403	0,01464	0,0058838
2	0,713	0,008283	0,005905779
3	0,4133	0,01427	0,005897791
4	0,376	0,01569	0,00589944
5	0,01387	0,00003024	0,0000004194288
6	0,2284	0,00002877	0,000006571068
7	0,1173	0,00003024	0,000003547152
8	0,02303	0,000154	0,00000354662
9	0,1793	0,00003024	0,000005422032
10	0,2711	0,00002	0,000005422

Table 5.3 The exergy flow rates, cost flow rates and unit exergy costs of SAHP for R744.

State No	$\dot{E}(kW)$	$\dot{c}(€/KWh)$	$\dot{C}(€/h)$
1	1,879	0,01464	0,02750856
2	2,57	0,00793	0,02040323
3	2,39	0,009004	0,02151956
4	1,881	0,0146	0,0274626
5	0,01387	0,000004	0,0000000605
6	0,08445	0,001133	0,00009568
7	0,0197	0,0000043	0,0000000859
8	0,02303	0,0000036	0,00000008465
9	0,1793	0,0000043	0,0000007822
10	0,03908	0,00002	0,0000007816

Table 5. 4 The exergy flow rates, cost flow rates and unit exergy costs of SAHP for R134a.

State No	$\dot{E}(kW)$	$\dot{c}(€/KWh)$	$\dot{C}(€/h)$
1	1,165	0,01464	0,0170556
2	1,403	0,01216	0,01706048
3	1,081	0,01577	0,01704737
4	1,063	0,01605	0,01706115
5	0,01387	0,00002783	0,000000386
6	0,202	0,000002627	0,00000053
7	0,1051	0,00002783	0,00000292
8	0,02303	0,0001271	0,00000292
9	0,1793	0,00002783	0,00000498
10	0,2494	0,00002	0,00000498

The thermoeconomic parameters of the SAHP for R22, R744 and R134a are shown in Tables 5.5-5.10. The CI, O&M and total cost of SAHP are computed as 1.451 €/kWh. Thermoeconomic factors are low as 2,766E-08% at evaparator, and high cost rate of exergy destruction at expansion valve is 0,00053 % for R22 . And also, relative cost difference is minimum and maximum values are 0,713 % and 0,055 % for condenser and expansion valve. For R744, relative cost difference is maximum at 0,620 % for condeser and minimum at 0,0028 % for expansion valve. The maximum destruction occured at expansion valve as 0,0045 % and exergoeconomic has the

highest value at expansion valve (0,000025 %). Finally, for R134a, the exergoeconomic factor is defined for low and high values as 0,0000049 % and % 5,062E-08 for expansion valve and evaparator. The unit cost of exergy destruction is lowest as a value of 0,00065 % for compressor.

Table 5.5 The thermoeconomic parameters for SAHP of R22.

Component	$c_{f,k}$ (€/kWh)	$c_{p,k}$ (€/kWh)	r (%)	f (%)	$\dot{D}_D$ (€/h)
Compressor	0,01464	0,008283	0,4341	2,470E-07	0,0008059
Condenser	0,008311	0,01424	0,7134	4,794E-07	0,0007074
Expansion	0,01566	0,01479	0,05533	0,000001155	0,001774
Evaporator	0,01427	0,01569	0,09918	2,766E-08	0,0005322
Thermal	0,00003024	0,00002877	1,951	0,0001512	0,000006489
Solar	0,00002	0,00003024	2,512	0,00007157	0,000009008
Heating	0,0001238	0,00001024	1,083	0,000008723	0,00007313
Total	0,05299	0,05299	5,223	0,0002155	0,003762

Table 5.6 Thethermoeconomicparametersfor SAHP of R774.

Component	$c_{f,k}$ (€/kWh)	$c_{p,k}$ (€/kWh)	r (%)	f (%)	$\dot{D}_D$ (€/h)
Compressor	0,01464	0,007939	0,4576	1,107E-07	0,001649
Condenser	0,009072	0,009	0,007947	9,004E-07	0,0009954
Expansion	0,01459	0,01463	0,002802	0,00002515	0,00008146
Evaporator	0,009004	0,0146	0,6209	3,213E-09	0,004582
Thermal	0,000004363	0,001133	260,7	0,003177	3,079E-07
Solar Collector	0,00002	0,000004363	1,218	0,0001476	0,000004368
HeatingSystem	0,000008039	0,00001564	2,945	0,0003578	0,000001782
Total	0,04731	0,04731	264,7	0,003708	0,007314

Table 5.7 Thethermoeconomicparametersfor SAHP of R134a.

Component	$c_{f,k}$ (€/kWh)	$c_{p,k}$ (€/kWh)	r (%)	f (%)	$\dot{D}_D$ (€/h)
Compressor	0,01464	0,01216	0,1693	3,218E-07	0,0006529
Condenser	0,01216	0,01575	0,2948	3,303E-07	0,001628
Expansion	0,01602	0,01476	0,07842	0,00000498	0,0004114
Evaporator	0,01577	0,01605	0,01735	5,062E-08	0,0002908
Thermal	0,00002783	0,000002627	1,094	0,0001874	0,000005236
Solar Collector	0,00002	0,00002783	2,391	0,00007519	0,000008575
HeatingSystem	0,00009925	0,000007825	1,079	0,00001154	0,00005528
Total	0,05868	0,05868	4,5	0,0002561	0,002941

The COP for heat pump changed between 2.9 % to 5.3 % (shown in Figure 5.1). The maximum value is obtatined for R134a, more effiecient than other refrigerants. The exergy efficient of heat pump components are compared in Figure 5.2. The evaparator has the highest efficieny for three refrigerant but for the highest average values of heat pump refrigerant obtained from R134a. The exergy destruction is minimum for R22 refrigerant as observed from Figure 5.3. And highest value is seen for R134a. The high exergy efficieny with COP were caused high exergy destruction. By comparing exergy and COP relations from Figures 5.1-5.3; R134a is good to used in heat pump system for more utilization.

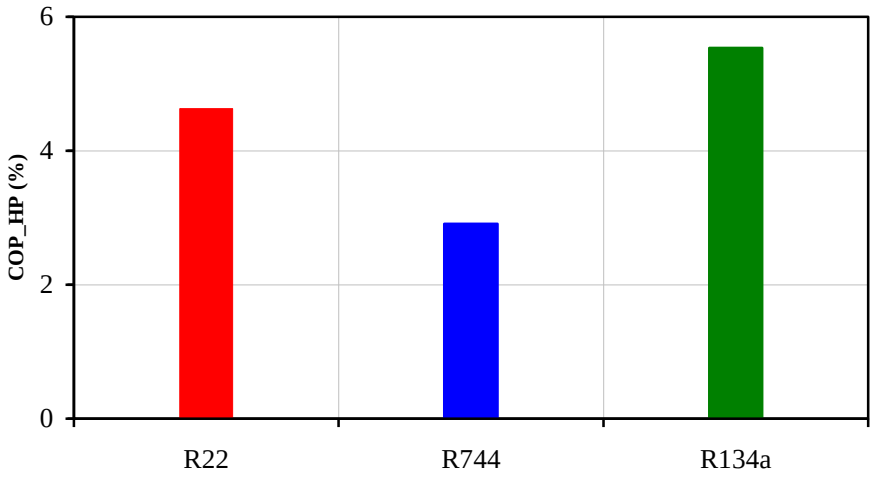

Figure 5.1 COP_{HP} between Refrigerants

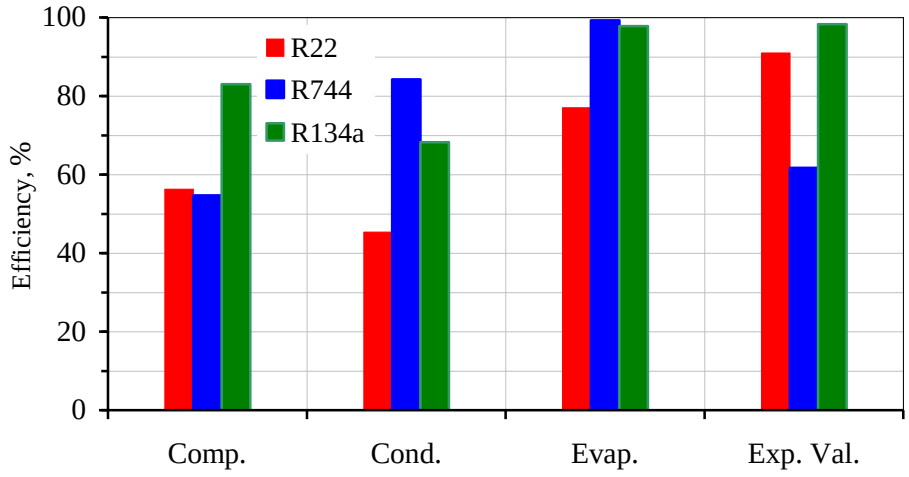

Figure 5.2 Exergy Efficiency between heat pump components

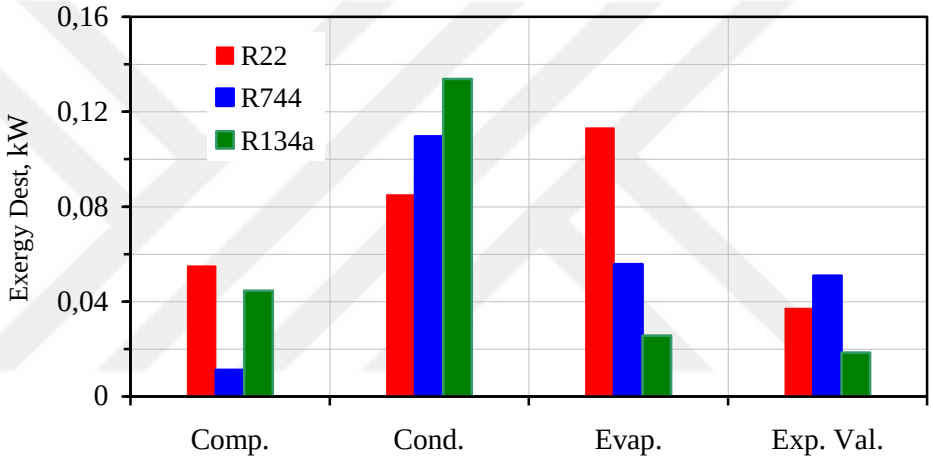

Figure 5.3 Exergy Destruction between heat pump components

The thermoeconomic parameters which are known as cost destruction of exergy rate, relative cost difference and exergocenomic factor. They are depicted in Figures 5.4-5.6. At Figure 5.4, the highest exergy rate cost destruction is high at evaparator for R22 refrigerant, but minimum avarega value is for R134a for the SAHP system. It is seen from the previous figures for exergy relations that R134a has maximum efficiency. The relative cost difference is high for R22 and low for R134a as seen from Figure 5.5. R134a gives a good results. The exergoeconomic factor is minimum for R22 and maximum for R134a refrigerant observed from Figure 5.6. It is well evulation for cost destruction of exergy rate, relative cost difference and exergocenomic factor that R134a refrigerant has good perfonmace for exergy anlysis

also good at thermoeconomic calculation. It can be used as efficient refrigerant in heat pump system instead of R22 for future real study.

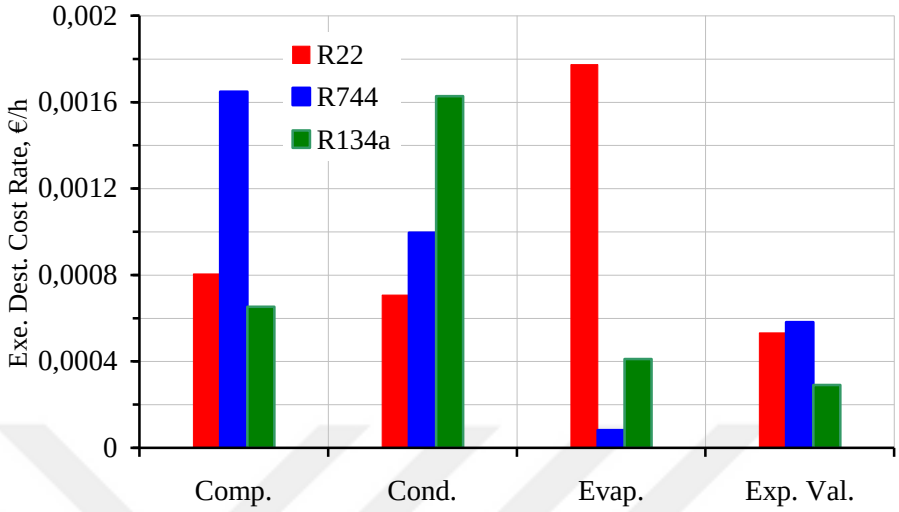

Figure 5.4 Exergy Rate Cost Destruction between heat pump components

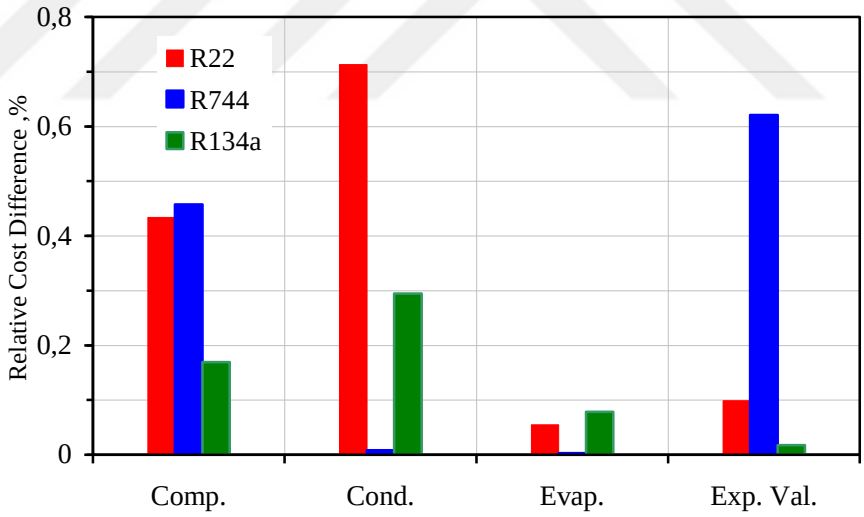

Figure 5.5 Relative Cost Difference between heat pump components

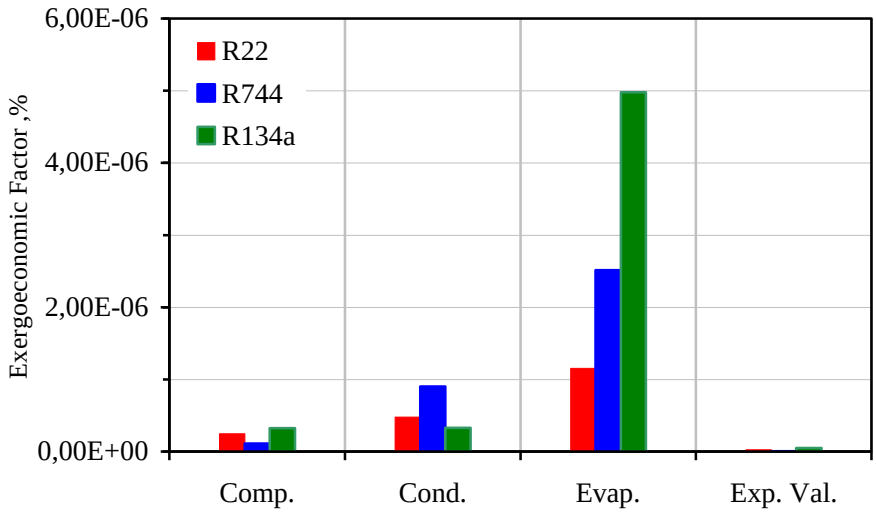

Figure 5.6 Exergoeconomic factor between heat pump components

CHAPTER 6

CONCLUSIONS AND FUTURE WORKS

6.1 Conclusions

The data were collected for solar assisted heat pump (SAHP) system installed at Campus of University of Gaziantep for collecting measurement between half month of February 15, to end month of April 30. Daily clear sunny and cloudy day were taken care for measurements for collected data. Particularly, the SAHP system has an example for solar assisted heat pump heating of building. As known, higher solar radiation cause to increase performance of solar assisted heat pump space heating system. The data used to calculate exergy and economical values of the SAHP system for residential system. The SAHP was designed as the followings: isoentropic efficiency of compressor is 80 %, works under adiabatic state, without pressure losses and evaporator and condenser exist are saturated vapor or liquid states. The R22 is real fluid used in the heat pump. Also data for R744 and R34a were used in program prepared in EES to compare results obtained with those of R22. More important results are given as the following.

- The energy and exergy computaions were done for three types of refrigerants. For R134-a, minimum and maximum values at inlet of condenser and compressor are as 1,063 and 1,403 kW because of high heat loss, obtained from Table 3. The high efficiency of heat pump is obtained for R134a.
- Variation of exergy destructions with temperatures were discussed. Average exergy destruction in evaporator, condenser, compressor and expansion valves were obtained as 0.1445, 0.1033, 0.32kW and 0.041 kW, respectively.
- The SAHP system life were designed for 15 years. From the system, the following values are obtained for interest of 0.06, salvage value of 0.2, return factor of 0.05.

- The Capital Invesment (CI) and Opearating and Maintance (OM) Cost were obtained and given in Table 5.1. The highest CI is obtained for condenser, and then comprosser. Both of them are main components of heat pump and whole system. The lowest one is expansion valve.

- The rates of exergy flow, cost flow and the unit exergy costs are shown in Tables 5.2-5.4. For R134a, the highest unit exergy cost and cost flow rate are found as 0,017 €/kWh for exit of compressor and 0,016 €/kWh for exit of expansion valve. Because of high capital invesment.

- The termoeconomic parameters of SAHP for R22, R744 and R134a are shown in Tables 5.5-5.10. The R134a, the exergoeconomic factor is defined for low and high values as 0,00000498 % and % 5,062E-08 for expansion valve and evaparator. The unit cost of exergy destruction is lowest as a value of 0,0006529 % for compressor.

- The evaparator has the highest efficieny for three refrigerants, but for the highest average values of heat pump refrigerant was obtained for R134a. The high exergy efficieny with COP were caused high exergy destruction. As a result, R134a is good in heat pump system for more utilization.

- The thermoeconomic parameters which were known as cost destruction of exergy rate, relative cost difference and exergocenomic factor were shown in Figures 5.4-5.6. The minimum average value is R134a for the system, related with maximum efficiency. The relative cost difference is low for R134a. The exergoeconomic factor is maximum for R134a refrigerant.

- It is well evulation for cost destruction of exergy rate, relative cost difference and exergocenomic factor that R134a refrigerant has good perfonmace for exergy anlysis also good at thermoeconomic calculation. It can be used as efficient refrigerant in heat pump system instead of R22 for future real study.

6.2 Future Works

- For increasing system efficiency, new equipments should be installed by decreasing power requirements related with this type of the system. Thus, new challenges have to investigate.

- Solar radiation coming to collector surface should be measured by inserting automatic sensors, and fluid at collector inlet and exist.

- To measurement more sensitivity results, thermocouples have to be insulated during working conditions.

- The operation and maintenance costs of the SAHP system should be decreased by working in accurate situation.

- The SAHP heating system should be optimized by using different types of algorithms.

REFERENCES

[1] Arunwattana W. (2010). 'A model for improvement of water heating heat exchanger designs for residential heat pump water heaters.' , Ms. Thesis, Massey University: Palmerston North/New Zealand.

[2] Salman A. (2012). 'Experimental investigation on performance of a heating and coolingsystem with ground coupled heat pump.', Ms. Thesis, Gaziantep University:
Gaziantep/Turkey.

[3] Schaefer L. (2000). 'Single pressure absorption heat pump analysis.', PhD Thesis,Institute of Technology: Georgia.

[4] Othman M. (2013). 'Modelling the performance of horizontal heat exchanger of ground-coupled heat pump systems with Egyptian conditions.', Ms. Thesis, Al-Minia University: Egypt.

[5] Zhang Y. (2012). 'Investigation on the application of heat pump systems in residential buildings in the UK.', Ms. Thesis, Conventry University: Conventry.

[6] Bakirci, K. (2009). 'Models of solar radiation with hours of bright sunshine: a review', *Renewable and Sustainable Energy Reviews.*, **13**, 2580-2588.

[7] Bakirci, K. (2015). 'Models for the estimation of diffuse solar radiation for typical cities in Turkey', Energy., **82**, 827-838.

[8] Van Wylen, G.J. and Sonntag, R.E. (1985). 'Fundamentals of Classical Thermodynamics'. 3rd Edition. New York: John Wiley & Sons.

[9] Buker, M.S., and Riffat, S.B. (2016). 'Solar assisted heat pump systems for low temperature water heating applications: A systematic review', *Renewable and Sustainable Energy Reviews.*,**55**, 399–413.

[10] Cerit E.,and Erbay, B.L. (2013). 'Investigation of the effect of rollbond evaporator design on the performance of direct expansion heat pump experimentally', *Energy ConversionManagement.*, **72**, 163–70.

[11] Chow, T.T., Peib, G., Fong, K.F., Lin, Z., Chan, ALS., and He, M. (2010). 'Modeling and application of direct-expansion solarassisted heat pump for water heating in subtropical Hong Kong', *Applied Energy.*,**87**, 643–9

[12] Xu, G., Deng, S., Zhang, X., Yang, L., and Zhang, Y. (2009). 'Simulation of a photovoltaic/thermal heat pump system having a modified collector/evaporator', *Solar Energy.*, **83**, 1967–76.

[13] Qin, W., Liu, Y., Liang, G., Li, J., Sun, S., and Chen, G. (2011). 'Development and experimental validation of a novel indirect-expansion solar-assisted multifunctional heat pump', *Energy Building.*, **43**, 300–304.

[14] Fu, HD., Pei, G., Long, H., Zhang, T., and Chow. (2012). 'Experimental study of a photovoltaic solar-assisted heat-pump/heat-pipe system', *Applied Thermal Engineering.*, **40**, 343–50.

[15] Koroneos, C., and Tsarouhis, M. (2012). 'Exergy analysis and life cycle assessment of solar heating and cooling systems in the building environment', *Journal of Clean Production.*,**32**, 52–60.

[16] Axaopoulos, P., Panagakis, P., and Kyritsis, S. (1998). 'Experimental comparison of a solar assisted heat pump vs. a conventional thermosyphon solar system', *International Journal of Energy Resource.*,**22**, 1107–20.

[17] Ito, S., Miura, N., and Wang, K. (1999). 'Performance of a heat pump using direct expansion solar collectors', *Solar Energy.*,**65**, 189–96.

[18] Chaturvedi, S.K., Chen, D.T., and Kheireddine, A. (1998). 'Thermal performance of a variable capacity direct expansion solar-assisted heat pump', *Energy Conversion Managament.*, **39**, 181–91.

[19] García, A., Herrero-Martin, R., Solano, J.P., and Pérez-García, J. (2018). 'The role of insert devices on enhancing heat transfer in a flat-plate solar water collector', *Applied Thermal Engineering.*, **132**, 479-489.

[20] www.visualdictionaryonline.com/energy/solar-energy/flat-plate-Solar

[21] Kaska O. (2002). 'Experimental investigation of performance of solar assisted heat pump space heating system with an energy storage.', Ms. Thesis, Gaziantep University: Gaziantep/Turkey.

[22] Chata, G., Benjamín, F., Chaturvedi, S.K., and Almogbel A. (2005). 'Analysis of a direct expansion solar assisted heat pump using different refrigerants', *Energy Conversion Managament.*, **46**, 2614–24.

[23] Mohanraj, M., Jayaraj, S., and Muraleedharan, C. (2009). 'A comparison of the performance of a direct expansion solar assisted heat pump working with R22 and a mixture of R407C–liquefied petroleum gas', *Process Institue Mechanical Engineering Part A and Journal Power Energy.*, **223**, 821–33.

[24] Buker M.S.,and Riffat, S.B. (2015). 'Building integrated solar thermal collectors a review', *Renewable Sustainable Energy Reviews.*,**51**, 327–46.

[25] Egrican, A. (1991). 'Performance of a solar assisted heat pump system', *Applied Energy.*, **51**, 93-109.

[26] Yumrutas, R., and Unsal, M. (2000). 'A computational model of a heat pump system with a hemispherical surface tank as the ground heat source', *Energy.*,**25**, 371-388.

[27] Yumrutas, R., Kaska, O., and Carpinlioglu, M. (2003). 'An experimental study on a solar assisted heat pump system with an energy storage', First International Exergy, Energy and Environment Symposium.

[28] Yan, G., Bai, T., and Yu, J. (2016). 'Energy and exergy efficiency analysis of solar driven ejector-compressor heat pump cycle', *Solar Energy.*,**125**, 243-255.

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

[30] Asaee, S., Ugursal, V., and Beausoleil-Morrison, I. (2017). 'Techno-economic assessment of solar assisted heat pump system retrofitin the Canadian housing stock', *Applied Energy.*,**190**,439–452.

[31] Dong, X., Tian, Q., and Li, Z. (2017). 'Energy and exergy analysis of solar integrated air source heat pumpfor radiant floor heating without water', *Energy and Buildings.*,**142**,128–138.

[32] Suleman, F., Dincer, I., and Agelin-Chaab, M. (2014). 'Energy and exergy analyses of an integrated solar heat pump system', *Applied Thermal Engineering.*,**73**,559-566.

[33] Lohani, S.P., and Schmidt, D. (2010). 'Comprasion of energy and exergy analysis of fossil plan, ground air source heat pump building heating system', *RenewableEnergy.*, **35**, 1275-1282.

[34] Khaliq A. (2015). 'A theoretical study on a novel solar based integrated system for simultaneous production ocooling and heating', *International Journal of Refrigeration.*, **52**, 66-82.

[35] Cho H. (2015). 'Comparative study on the performance and exergy efficiency of a solar hybrid heat pump using R22 and R744', *Energy.*, **93**, 1267-1276.

[36] Badescu V. (2002). 'First and second law analysis of a solar assisted heat pump based heating system', *Energy Conversion and Management.*, **43**, 2539–2552.

[37] Cimmino M., and Eslami-Nejad P. (2006). 'A simulation model for solar assisted shallow ground heat exchangers in series arrangement', *Energy and Buildings.*, **157**,227–246.

[38] Ucar A., and Inalli M. (2006). 'Exergoeconomic analysis and optimization of a solar-assisted heatingsystem for residential buildings', *Building and Environment.*, **41**, 1551–1556.

[39] Çengel, Y. A and Boles, M. A. (2008). 'Thermodynamics: An Engineering Approach'. 6th Edition. New York: Mc-Graw Hill.

[40] Bejan, A., Tsatsaronis, G., and Moran, M. (1996). '*Thermal Design and Optimization'*. 1st Edition. New York: Wiley.

[41] Van Wylen, G. J. And Sonntag, R.E. (1985). 'Fundamentals of Classical Thermodynamics'. 3rd Edition. New York: John Wiley & Sons.

[42] Wark, K. Jr. (1995). 'Advanced Thermodynamics for Engineers'. 1st Edition. New York: Mc-Graw Hill.

[43] Wall, G. and Gong, M. (2001). 'On exergy and sustainable development – Part 1: Conditions and concepts', *Exergy, An International Journal.*, **1**, 128-145.

[44] Wall, G. And Gong, M. (2001). 'On exergy and sustainable development – Part 2: Indicators and methods', *Exergy, An International Journal.*, **1**, 217-233.

[45] Nikulshin, V., Wu, C., and Nikulshina V. (2002). 'Exergy efficiency of energy intensive systems', *Exergy, An International Journal.*, **2**, 78-86.

[46] Tsatsaronis, G. and Ho-Park, M. (2002). 'On avoidable and unavoidable exergy destructions and investment costs in thermal systems', *Energy Conversion and Management.*, **43**, 1259-1270.

[47] Wall G. (1986). '*Exergy – A useful concept*.', PhD Thesis, Physical Resource Theory Group, Chalmers University of Technology and University of Goteborg: Goteborg/Sweden.

[48] Dunbar, W.R., Lior, N. And Gaggioli, R.A. (1992). 'The components equations of energy and exergy', *Journal of Energy Resources Technology.*, **114**, 75-83.

[49] Vajpayee, S. K. (2001). 'Fundamentals of Economics for Enginering Technologists and Engineers'. London: Prentice Hall.

[50] Park, C. S. (1990). 'Advanced Engineering Economics'. New York: John Wiley& Sons.

[51] Gönen, T. (1990). 'Engineering Economy for Engineering Managers'. NewYork: John Wiley & Sons.

[52] Tsatsaronis, G. and Moran, M.J. (1997). 'Exergy-aided cost minimization', *Energy Conversion and Management.*, **38**, 1535-1542.

[53] Lazzaretto A. and Tsatsaronis G. (1997). 'On the quest for objective equations in exergy costing', In *Proceedings of the ASME Advanced Energy Systems Division.*, **37**,197-209.

[54] Lozano, M.A. and Valero A. (1993). 'Theory of the exergetic cost', *Energy.*, **18**, 939-960.

[55] Reini, M., Lazzaretto A., Macor A. (1995). Average structural and marginal costs as a result of a unified formulation of the thermoeconomic problem. In *Proceedings of Second Law Analysis of Energy Systems: Towards the 21st Century*, Rome.

[56] Oh, S.D., Pang, H.S., Kim, S.M., and Kwak, H.Y. (1996). 'Exergy analysis for a gas turbine cogeneration system', *Transactions of the ASME.*, **118**, 782-791.

[57] Kwon, Y.H., Kwak, H.Y., and Oh, S.D. (2001). 'Exergoeconomic analysis of gas turbine cogeneration systems', *Exergy, An International Journal.*, **1**, 31-40.

[58] Lazzaretto, A. and Tsatsaronis, G. (2001). 'Comparison Between SPECO-Based and Functional Exergoeconomic Approaches', In *Proceedings of the 2001 ASME. International Mechanical Engineering Congress and Exposition in New York*, 1-16.

[59] Ahern, J. E. (1980). 'The Exergy Method of Energy System Analysis'. 1st Edition. New York: John Wiley & Sons.

[60] Gaggioli, R. A. and Wepfer, W. J. (1980). 'Exergy economics', *Energy, The International Journal.*, **5**, 823-837.

[61] Tsatsaronis, G., Lin, L., and Pisa, J. (1993). 'Exergy costing in Exergoeconomics', *Journal of Energy Resources Technology.*, **115**, 9-16.

[62] Tsatsaronis, G. and Ho-Park, M. (2002). 'On avoidable and unavoidable exergy destructions and investment costs in thermal systems', *Energy Conversion and Management.*, **43**, 1259-270.

[63] Sciubba, E. (2001). 'The concept of Extended Exergy Accounting and its application to the analysis and design of thermal systems', *Exergy,An International Journal.*, **1**, 68-84.

[64] Kim, S.M., Oh, S.D., Kwon, Y.H., and Kwak, H.Y. (1998). 'Exergoeconomic analysis of thermal systems', *Energy.*, **23**, 393-406.

[65] Valero, A., Lozano, M.A., Serra, L., and Torres, C. (1994). 'Application of the exergetic cost theory to the CGAM problem', *Energy.*, **19**, 365-381.

[66] Kwon, Y.H., Kwak, H.Y., and Oh, S.D. (2001). 'Exergoeconomic analysis of gas turbine cogeneration systems', *Exergy, An International Journal.*, **1**, 31-40.

[67] Tsatsaronis, G. and Ho-Park, M. (2002). 'On avoidable and unavoidable exergy destructions and investment costs in thermal systems', *Energy Conversion and Management.*, **43**, 1259-270.

[68] Kwak, H. Y., Byun, G. T., Kwon, Y. H., and Yang, H. (2004). 'Cost structure of CGAM cogeneration system', *International Journal of Energy Research.*, **28**, 1145-1158.

[69] El-Sayed, Y. and Evans, R. L. (1970). 'Thermoeconomics and the design of heat systems', *Journal of Engineering and Power.*, **92**, 27-35.

[70] El-Sayed, Y.M. and Gaggioli, R.A. (1989). 'A critical review of Second Law costing methods -1: Background and Algebraic Procedures', *Journal of Energy Resources Technology.*, **111**, 1-7.

[71] Frangopoulos, C.A. (June 1991). 'Comparison of thermoeconomic and thermodynamic optimal designs of a combined-cycle plant', *International Conference on the Analysis of Thermal and Energy Systems, Athens, Greece.*, 305-318.

[72] Frangopoulos, C.A. (1994). 'Application of the thermoeconomic functional approach to the CGAM problem', *Energy.*, **19**, 323-342.

[73] Frangopoulos, C.A. (1991). 'Intelligent Functional Approach: A method for analysis and optimal synthesis – design – operation of complex systems', *International Journal of Energy, Environment, Economics.*, **1**, 267-274.

[74] Frangopoulos, C.A. (1991). 'Optimization of synthesis – design – operation of a cogeneration system by The Intelligent Functional Approach', *International Journal of Energy, Environment, Economics.*, **1**, 275-287.

[75] Frangopoulos, C.A. (1987). 'Thermoeconomic functional analysis and optimization', *Energy.*, **12**, 563-571.

[76] Manolas, D.A., Frangopoulos, C.A., Gialamas, T.P., and Tsahalis, D.T. (1997). 'Operation optimization of an industrial cogeneration system by a genetic algorithm', *Energy Conversion and Management.*, **38**, 1625-1636.

[77] Von Spakovsky, M.R. and Evans R.B. (1990). 'The design and performance optimization of thermal systems', *Journal of Engineering for Gas Turbines and Power.*, **112**, 86-93.

[78] Tsatsaronis, G., Pisa, J., Valero, A., Lozano, M.A., Serra, L., Frangopoulos, and C.,Von Spakovsky, M. (1994). 'CGAM problem: Definition and conventional solution',*Energy.*, **19**, 279-286.

[79] Tsatsaronis, G. and Pisa, J. (1994). 'Exergoeconomic evaluation and optimization of energy systems – Application to the CGAM problem', *Energy.*, **19**, 287-321.

[80] Toffolo, A. and Lazzaretto, A. (2002). 'Evolutionary algorithms for multi-objective energetic and economic optimization in thermal system design', *Energy.*, **27**, 549-567.

[81] Marc A. Rosen., Ibrahim Dincer., and Mehmet Kanoglu (2008). 'Role of exergy in increasing efficiency and sustainability and reducing environmental impact', *Energy Policy.*,**36,** 128–137.

[82] Zainudin, K. (2010). 'Thermodynamic analysis of ground source heat pump.', Ms. Thesis, Malaysia Pahang University.

[83] Yilmaz, C. (2011). 'Thermodynamic and economic analysis of geothermal energy use in hydrogen production and liquefaction.', Ms. Thesis, Gaziantep University.

[84] Yilmaz, C. (2015). 'Thermoeconomic design and optimization of geothermal energy use in hydrogen production and liquefaction.', PhD. Thesis, Gaziantep University.

[85] Abusoglu, A. (2008). 'Exergetic and Thermoeconomic Performance Analysis and Optimization of Diesel Engine Powered Cogeneration Plants.', PhD. Thesis, Gaziantep University.

[86] Ayanoglu, A. (2010). 'Exergy And Exergoeconomic Analysis of Geothermal Assisted High Temperature Electrolysis.', Ms. Thesis, Gaziantep University.

[87] Yumrutaş, R. and Kaşka, Ö. (2004). 'Experimental investigation of thermal performance of a solar assisted heat pump system with an energy storage', *International Journal of Energy Research.*, **28**, 163–175.

[88] Dikici, A. and Akbulut, A. (2008). 'Performance characteristics and energy–exergy analysis of solar-assisted heat pump system', *Building and Environment.*,**43**, 1961– 1972.

[89] Abusoglu, A. and Kanoglu, M. (2009). 'Exergoeconomic analysis and optimization of combinedheat and power production: a review', *Renewable Sustainable Energy Review*,**13**,2295–2308.

[90] Kanoglu, M., Ayanoglu, A., and Abusoglu, A. (2011). 'Exergoeconomic assessment of a geothermal assisted high temperature steam electrolysis system', *Energy.*,**36**,4422–4433.

[91] Abusoglu, A. and Kanoglu, M. (2009). 'Exergetic and thermoeconomic analyses of diesel engine powered cogeneration: Part 1 – Formulations', *Applied Thermal Engineering.*,**29**, 234-241.

[92] Abusoglu, A. and Kanoglu, M. (2009). 'Exergetic and thermoeconomic analyses of diesel engine powered cogeneration: Part 2 – Application', *Applied Thermal Engineering.*, **29**, 242-249.

[93] Lazzaretto, A. and Tsatsaronis, G. (2006). 'SPECO: A systematic and general methodologyfor calculating efficiencies andcosts in thermal systems', *Energy.*, **31**,1257–1289.

[94] Yücer, C. T. and Hepbasli, A. (2014). 'Exergoeconomic and enviroeconomic analyses of a building heating system using SPECO and Lowex methods', *Energy and Buildings.*, **73**, 1-6.

[95] Yücer, C. T. and Hepbasli, A. (2013). 'Exergoeconomic analyses of an energy supply chain for space heating in a building', *Energy and Buildings.*, **62**, 343-349.

[96] Akbulut, U., Utlu, Z., and Kincay O. (2016). 'Exergy,exergoenvironmental and exergoeconomic evaluation of a heat pump-integrated wall heating system', *Energy.*, **107**, 502-522.

[97] Arat, H. and Arslan, O. (2017). 'Exergoeconomic analysis of district heating system boosted by the geothermal heat pump', *Energy.*, **119**, 1159-1170.

[98] Ucar, A. and Inalli M. (2006). 'Exergoeconomic analysis and optimization of a solar-assisted heating system for residential buildings', *Building and Environment.*, **41**, 1551-1556.

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

[100] Li, Y. and Xiong Y. (2018). 'Thermo-economic analysis of a novel cascade integrated solar combined cycle system', *Energy.*,**145**,116-127.

[101] Gungor, A., Tsatsaronis, G., Gunerhan, H., and Hepbasli, A. (2015). 'Advanced exergoeconomic analysis of a gas engine heat pump (GEHP) for food drying processes', *Energy Conversion and Management*, **91**, 132–139.

[102] Akbulut, U., Utlu, Z., and Kincay O. (2016) 'Exergy, exergoenvironmental and exergoeconomic evaluation of a heat pump-integrated wall heating system', *Energy*, **107**, 502-522.

[103] Ozel-Erol, G., and Hepbasli, A. (2017). 'Performance assessment of an ice rink refrigeration system through advanced exergoeconomic analysis method', *Energy and Buildings*, **138**, 118– 126

[104] Zogou, O, and Stamatelos, A. (2007). 'Optimization of thermal performance of a building with ground source heat pump system', *Energy Conversion and Management*, **48**, 2853–2863.