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Ph.D. in Mechanical Engineering

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**UNIVERSITY OF GAZİANTEP
GRADUATE SCHOOL OF
NATURAL & APPLIED SCIENCES**

**AN INVESTIGATION OF RENEWABLE ENERGY
POTENTIAL OF MUNICIPAL BIODEGRADABLE SOLID
WASTE PLANTS IN FRAME OF THERMODYNAMIC,
THERMOECONOMIC ANALYSIS AND OPTIMIZATION
AND DEVELOPING SUSTAINABLE MODELS USING
ZERO WASTE APPROACH**

**Ph.D. THESIS
IN
MECHANICAL ENGINEERING**

**BY
ALPEREN TOZLU**

JULY 2017

**An Investigation of Renewable Energy Potential of Municipal Biodegradable
Solid Waste Plants in Frame of Thermodynamic, Thermoeconomic Analysis
and Optimization and Developing Sustainable Models Using Zero Waste
Approach**

Ph.D. Thesis

in

Mechanical Engineering

University of Gaziantep

Supervisor

Assoc. Prof. Dr. Emrah ÖZAHİ

by

Alperen TOZLU

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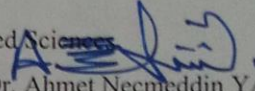
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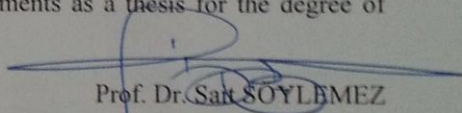
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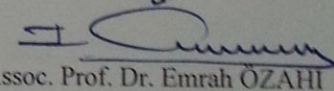
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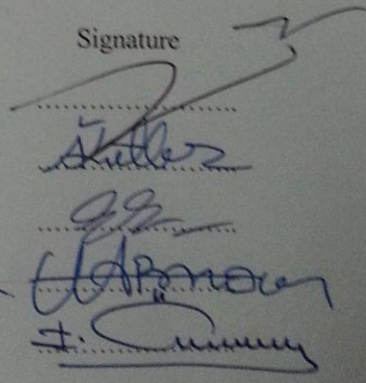
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To the memory of my grandfather

Mehmet TOZLU

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Alperen TOZLU

ABSTRACT

AN INVESTIGATION OF RENEWABLE ENERGY POTENTIAL OF MUNICIPAL BIODEGRADABLE SOLID WASTE PLANTS IN FRAME OF THERMODYNAMIC, THERMOECONOMIC ANALYSIS AND OPTIMIZATION AND DEVELOPING SUSTAINABLE MODELS USING ZERO WASTE APPROACH

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In this thesis, Gaziantep Municipal Solid Waste Power Plant (GMSWPP) is assessed and modeled in ASPEN Plus engineering software in terms of thermodynamic and thermoeconomic analyses considering the actual operating data. This thesis comprises of an extensive study when compared to other studies in this field. The novelty of the thesis can be one of the first studies, in which renewable and sustainable waste heat recovery models are designed and developed utilizing the actual operating data of an existing solid waste power plant. The models which are gas turbine (Model 1), organic Rankine (Model 2), Kalina (Model 3), gas turbine-organic Rankine (Model 4) and gas turbine-Kalina cycles (Model 5) are analyzed thermodynamically and thermoeconomically by using EES software. Moreover, originality of the thesis is also a fact that it contains an optimization study for all models, which is conducted based on the non-dominated sorting genetic algorithms (NSGA-II) method in MATLAB environment. Therefore, as a result of the thesis, it is conducted that the models can be adapted to all power plants from the point of a waste-to-energy approach for further developments. This can be expressed as another original aspect of the thesis.

Key Words: Solid waste, landfill gas, thermodynamic, thermoeconomic, optimization, waste-to-energy

ÖZET

BİYOBOZUNUR KATI ATIK TESİSLERİNİN TERMODİNAMİK, TERMOEKONOMİK ANALİZ VE OPTİMİZASYONLARI ÇERÇEVESİNDE YENİLENEBİLİR ENERJİ POTANSİYELLERİNİN ARAŞTIRILMASI VE SIFIR ATIK YAKLAŞIMIYLA SÜRDÜRÜLEBİLİR MODELLERİN GELİŞTİRİLMESİ

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Bu tezde, Gaziantep Büyükşehir Belediyesi Katı Atıktan Enerji Üretim Tesisi (GMSWPP), gerçek işletme verileri kullanılarak, ASPEN Plus engineering programında termodinamik ve termoeconomik analizler açısından değerlendirilmiş ve modellenmiştir. Bu tez, bu alandaki diğer çalışmalarla karşılaştırıldığında kapsamlı bir çalışmayı içermektedir. Mevcut bir katı atıktan elektrik üretim tesisinin gerçek işletme verilerini kullanarak yenilenebilir ve sürdürülebilir atık ısı geri kazanım modellerinin tasarlanması ve geliştirilmesi tezin özgün yanı olarak gösterilebilir. Gaz türbini (Model 1), organik Rankine (Model 2), Kalina (Model 3), gaz türbini-organik Rankine (Model 4) ve gaz türbini-Kalina (Model 5) çevrim modellerinin termodinamik ve termoeconomik analizleri EES yazılımı ile gerçekleştirilmiştir. Ayrıca, tezin orijinal bir başka yönü de, tüm modeller için MATLAB ortamında hakim olmayan sınıflandırma genetik algoritma (NSGA-II) yöntemine dayalı olarak yürütülen optimizasyon çalışmasını içermesidir. Bu nedenle, tezin bir sonucu olarak, daha ileriki gelişmeler için, modellerin atıktan enerji kapsamında, tüm enerji santrallerine adapte edilebileceği sonucuna varılmıştır. Bu da, tezin bir başka özgün yanı olarak ifade edilebilir.

Anahtar Kelimeler: Katı atık, depo gazı, termodinamik, termoeconomik, optimizasyon, atıktan-enerji.

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LIST OF SYMBOLS

A	heat transfer area, m ²
c	cost per exergy unit, \$/GJ
c_f	unit exergy cost of fuel, \$/GJ
c_p	unit exergy cost of product, \$/GJ
$\dot{C}$	cost rate, \$/h
$\dot{D}$	cost rate of exergy destruction, \$/h
$\dot{E}_x$	exergy rate, kW
f	exergoeconomic factor
h	specific enthalpy, kJ/kg
i	interest rate
$\dot{m}$	mass flow rate, kg/s
n	total life time
N	annual operation time
P	pressure, bar
PR	pressure ratio
$\dot{Q}$	heat addition, kW
r	relative cost difference
s	specific entropy, kJ/kg-K
T	temperature, °C
U	heat transfer coefficient, kW/m ² -K
V	velocity, m/s
$\dot{W}$	work flow rate-power, kW
y	mole fraction
$\dot{Z}$	capital cost rate, \$/h

Greek symbols

ε	exergy efficiency
ε_f	effectiveness
η	energy efficiency
η_C	compressor isentropic efficiency
η_{KP}	Kalina pump isentropic efficiency
η_{OP}	ORC pump isentropic efficiency
η_{GT}	gas turbine isentropic efficiency
η_{KT}	Kalina turbine isentropic efficiency
η_{OT}	ORC turbine isentropic efficiency
ϕ	maintenance factor
ψ	specific flow exergy, kJ/kg

Subscripts

0	dead state
a	actual
CH	chemical exergy
crit	critical point
D	destruction
ex	exit
exh	exhaust
in	inlet
k	component
KN	kinetic exergy
PH	physical exergy
PT	potential exergy
s	isentropic
tot	total
wat	water

Abbreviations

AD	anaerobic digestion
AFT	air fuel tank
AWR	air water radiator
CEPCI	chemical engineering plant cost index
CI	capital investment
COMP	compressor
CRF	capital recovery factor
EES	Engineering Equation Solver
EMO	evolutionary multi-objective optimization
EVAP	evaporator
GA	genetic algorithm
GE	gas engine
GEN	generator
GMSWPP	Gaziantep Municipal Solid Waste Power Plant
GT	gas turbine
GTHE	gas turbine heat exchanger
HE	heat exchanger
HPC	high pressure compressor
HTR	high temperature recuperator
INT	intercooler
KC	Kalina cycle
KCON	Kalina condenser
KE	kinetic energy
KP	Kalina pump
KT	Kalina turbine
LMTD	logarithmic mean temperature difference
LOC	lubrication oil cooler
LPC	low pressure compressor
LTR	low temperature recuperator
NSGA-II	non-dominated sorting genetic algorithm
OC	ORC condenser

OM	operating and maintenance cost
OP	ORC pump
ORC	organic Rankine cycle
OT	ORC turbine
PE	potential energy
PEC	purchased equipment cost
PR	pressure ratio
PRE	precooler
REC	recuperator
S-CO ₂	supercritical carbon dioxide
SEP	separator
SPECO	specific exergy costing
TUR	turbine
VAL	valve

CHAPTER 1

INTRODUCTION

1.1 Background and Motivation

There are incredible changes in supply-demand balances of energy production by means of energy production from solid waste and using renewable energy sources. The strategies based on the development of sustainable waste sources and energy recovery are consisted of production, collection, usage and final elimination stages. Adverse effects taken place on the environment during the life cycle of these stages can be eliminated by means of recycling technologies. The importance of waste recycling and elimination facilities is well understood in the case of approaching to the waste as both pollution and energy-raw material source. The necessity for the management of waste which creates main environmental concerns has caused the development of various types of waste management systems, worldwide. The main aim of the studies being made on elimination of wastes is to be able to choose the best method which has the least cost and risk to both human being and environment. The basic waste management strategies can be classified as waste elimination, energy recovery, recycling, waste minimization or avoiding and zero waste approach.

Zero waste approach is based on reassessment of wastes and in this way, minimization of damage of wastes on the environment by increasing energy recovery potential of current systems. The first step of zero waste management is to be able to figure out all waste constituents in waste elimination facility and also how to eliminate these wastes in order to perform an adequate environmental management. The second step is to integrate the appropriate proposed energy recovery models to current systems according to the present waste elimination techniques. At last step, recovered energy is provided for usage in waste storage plants and/or in energy dense

sectors. Zero waste approach can be evaluated as a management style which provides a favorable cost gain with low payback period and increases the current operational efficiency of the facility in long term.

As a result of a comprehensive literature survey and observation of actual municipal solid waste power plants in Turkey, it is realized that there is a need for improvement of the actual system in the frame of waste-to-energy. Because, there is a considerable amount of waste heat in such like plants. If scarce amount of energy is considered, this attempt becomes more important.

1.2 Objects

In this thesis, thermodynamic and thermoeconomic analyses of Gaziantep Municipal Solid Waste Power Plant are performed using its operational and economic data. Then, waste-to-energy models are proposed using the exhaust gas of the current plant. After the detailed simulation studies of the proposed models, a comparative thermodynamic and thermoeconomic analyses and optimization are carried out.

The main motivation behind of this thesis is to reveal renewable and sustainable energy recovery models which can be applied to all solid waste power plants, biogas and cogeneration plants in Turkey, by determining the optimum energy recovery strategies that can be adapted to actual power plants. In the scope of these renewable and sustainable models, increasing the energy recovery potentials of current solid waste power plants is aimed in order to produce additional power to the actual power plants.

1.3 Outline of the Thesis

The organization of the thesis is divided into six main chapters as follows:

In Chapter 1, a brief information and background related to waste management and waste-to-energy phenomena and the objectives of the thesis are presented.

In Chapter 2, a comprehensive literature survey on municipal solid waste management, municipal solid waste disposal methods, energy production potentials of landfill and solid waste, thermodynamic and thermoeconomic analyses on power plants, gas turbine, organic Rankine and Kalina cycles and optimization studies are presented.

In Chapter 3, general formulations of thermodynamic analysis which includes both energy and exergy methods are given. In addition to this, general principles, terminology and formulation of thermoeconomic analysis are presented. Also, the procedure of multi-objective optimization is introduced. Lastly, thermodynamic and thermoeconomic analyses of Gaziantep Municipal Solid Waste Power Plant is presented in detail.

In Chapter 4, five different renewable and sustainable models are developed to adapted in order to increase in the power production of the existing power plant. The description of the systems are given in detail. Also, the thermodynamic and thermoeconomic analyses of developed models are performed by using the Engineering Equation Solver software.

In Chapter 5, the developed models are considered with respect to parametric studies and optimization studies by using non-dominated sorting genetic algorithm (NSGA-II) method are carried out in frame of MATLAB software for them. Besides, the comparisons of models are tabulated and presented in terms of net power output, exergy efficiency, total cost rate, total capital cost rate and electricity production costs.

In chapter 6, the summary of the results and the suggestions for the further studies are pointed out.

CHAPTER 2

LITERATURE SURVEY

2.1 Introduction

Municipal solid waste (MSW) management has an important role to play in the recovery of energy from waste when the world's increasing energy needs are taken into consideration. Municipal solid waste disposal methods have been successfully implemented both in developed and developing countries where existing infrastructure networks are well organized and wastes are considered as renewable energy sources in this context. Landfilling is one of the most common and the most widely accepted disposal methods in the worldwide and landfill gas (LFG) is the main product of this method. LFG, which consists of methane, carbon dioxide and trace amounts of other compounds, is a valuable fuel because of its methane content. It can be burned in an internal combustion engine in order to produce power and at the end of the combustion process, exhaust emissions have a more environmentally friendly profile compared to the other fossil fuels.

In this chapter, a comprehensive literature survey is presented including the main topics studied in the thesis. The literature search on these topics was carried out by meticulously scanning the national and the international sources on the open literature. The topics are classified as MSW management, MSW disposal methods, LFG and electricity production from MSW power plants, thermodynamic and thermoeconomic analysis, gas turbine, organic Rankine and Kalina cycles (ORC) and optimization. The survey is presented under these titles indicating their advantageous and disadvantageous with respect to their practical suggestions in reference to mentioned literature.

2.2 Municipal Solid Waste Management

Waste products such as household food wastes, paper, plastic, glass, textile, metals, wood, leather, sanitary products, etc. constitute the content of municipal solid waste (MSW) known as garbage. In cities, the main sources of these wastes are houses, schools, restaurants, hospitals and other private and public institutions. Urbanization, population growth and technological developments directly affect the solid waste management policies of the countries and the disposal of solid wastes presents some major challenges to every country of the world, both for infrastructural and for environmental reasons. In the light of these points, the previously published works in the open literature on solid waste management and the policies developed by the countries are summarized below.

Ojeda-Benitez and Beraud-Lozano [1] conducted an analysis of a field work carried out in four cities of Mexico, which have sanitary landfill and uncontrolled dump sites, based on one stage of the municipal solid waste cycle. They emphasized that all urban solid wastes should be collected on sanitary landfills throughout the country and integrated into a system ensuring that solid wastes are processed in the most appropriate technology and an efficient way.

Metin et al. [2] presented an overview for the last decade for Turkey's solid waste data and for the management practices of urban solid waste systems having completed infrastructural works. In this study, they also noted that the compositions of the materials recovered from solid waste vary according to the season.

Fobil et al. [3] investigated the potential of urban solid waste facilities for energy generation in low-income economies and used the research experience in Ghana's Accra city for this purpose. In order to efficiently obtain energy from solid waste facilities in big cities, they first considered that the technology of these facilities should be improved; but, they have come to the conclusion that such investments are not sustainable because the investments to make energy from waste are too high and the energy to be recovered is relatively low.

Bove and Lunghi [4] analyzed and compared the different traditional and innovative systems that recover energy from the storage gas from the technoeconomic and environmental perspectives. According to the results of the work done, it was revealed that the internal combustion engines have the lowest environmental performance, but, due to their relatively low initial investment and operating costs, they are still more preferred systems. However, despite the fact that fuel cells are the most energy-efficient systems, they have yet to compete with traditional systems because of their high investment costs.

Magrinho et al. [5] identified the work that can be done by the municipal solid waste disposal facilities in Portugal around the principle of energy recovery from waste, and stated that waste management can have a real function if regular storage, energy recovery and recycling are increased.

Manaf et al. [6] reviewed the general characteristics of solid waste management in Malaysia and predicted that as a developed country towards 2020, included public health, environmental protection and efficient recovery and recycling in waste management they can construct a solid waste policy at an economically affordable cost.

Ağdağ [7] presented a general overview and compared old and new MSW management practices in Denizli using detailed data on MSW management practices including collection, transportation, disposal and recycling. It was claimed that there were still some deficiencies in the new MSW management system such as no leachate treatment and poor source separation collection.

Turan et al. [8] reviewed the amount of solid waste, its content, the investments and work performed within the scope of energy recovery from waste, and general waste legislation in a demographic study on municipal solid waste management in Turkey. They also pointed out that although some areas of the country still use primitive methods, sanitary and environmentally friendly legislation on waste management is also put into effect.

Cheng and Hu [9] discussed the main difficulties in increasing waste incineration capacity in China under the heading of initial investment and operating costs, equipment corrosion, air pollutant emissions and fly ash disposal. They also stressed that China's enactment of energy from waste for the next ten years is extremely important in terms of energy supply.

Bovea et al. [10] conducted a case study on environmental assessments of municipal solid waste management strategies in Spain using a life cycle analysis method; they proposed 24 different scenarios by studying different waste collection systems, biological processing methods, and types of approaches on energy recovery. Scenarios with biogasification and energy recovery have found to perform better than others.

Kanat [11] emphasized that solid waste management should be reorganized in Istanbul in the most appropriate way, as is the case in all the major cities in developing countries. He indicated that the old conventional waste storage techniques are not appropriate solutions for metropolitan cities like Istanbul. For this, he insisted on the necessity of defining a cleverly planned economic capacity to determine the limits of new investments and to improve existing systems.

Lino and Ismail [12] studied on the energy recycling in solid waste disposal sites in Brazil and the prevention of CO₂ emissions escaping from the atmosphere during this process. They also pointed out that wastes accumulated on landfills are very hazardous both to the environment -by creating bad smell, by polluting surface waters and ground waters- and to the public health -by increasing disease agents-.

Aydoğan et al. [13] presented a demographic study on the solid waste management policy of the provincial municipality of Gaziantep by conducting a detailed search on domestic and industrial solid waste characteristics, municipal solid waste collection methods and waste disposal strategies, the population and the growth rates.

Singh et al. [14] conducted a study to demonstrate the energy recovery opportunities in a municipal solid waste facility in India. They reported that waste to energy

concept is the key to sustainable development for integrated waste management systems.

Udomsri et al. [15] studied energy recovery from solid waste by investigating different solid waste incineration methods in Thailand. For this, they analyzed both conventional and advanced hybrid dual fuel systems, taking into account cycle performances and CO₂ emissions. They emphasized that hybrid systems would increase the efficiency of electricity generation and that burning solid wastes would significantly reduce the environmental problems related to the waste disposal in Thailand.

Adu and Lohmeueller [16] investigated the use of solid waste as an eco-efficient energy resource in Ghana. They found that approximately 60% of the solid waste is biodegradable and contains a heating value of 17 MJ per kg. Considering the energy content of the domestic solid waste per capita in the country, they also calculated the net income to be generated from the solid wastes.

Akıncı et al. [17] proposed a long-term solid waste management strategy, taking into account some cities and towns in Aegean region of Turkey. For this purpose, they tried to get an idea of the potential of the solid waste resource through some data such as the waste characteristics of the region and the household income level. In addition, they made some suggestions for processing the green wastes in these cities such like composting and biomethanization.

Johari et al. [18] addressed the economic and the environmental benefits of the methane production at solid waste facilities in the context of sustainability. For this purpose, using International Panel on Climate Change (IPCC) data, they estimated methane production quantities in the solid waste landfills globally and demonstrated that the sustainable economic and environmental benefits of methane to be produced.

Melikoğlu [19] estimated the production of landfill gas and accordingly the electricity production potential of all municipalities in Turkey, taking into account the solid waste collection sites. When Turkey's annual energy production is taken into consideration, it is stated that the municipalities have less than 1% of the amount

of energy recovery potential from solid waste. It is also estimated until 2023 that the 3226-9678 GWh electricity production from incineration and 2184-2858 GWh of electricity production from landfill gas are expected..

Oteng-Ababio et al. [20] conducted a field study on the new solid waste management technologies in the city of Accra, Africa. It was stated that all the solid waste generation potential including the informal sector should be taken into account for an effective solid waste disposal and accordingly energy recovery from waste. With a rigorous inventory work to be done, it was reported that the investment to be made considering the potential of actual solid waste generation, without any sector being excluded, would provide both effective technology selection and long-term effective solid waste disposal.

Horttanainen et al. [21] gathered different studies on the energy content of the municipal solid waste in Finland. It was found that if organic wastes, paper and cardboard are separated from the solid waste, the energy content of the remaining waste will be reduced by 50-60%. It was also stated that the heating value can be increased by decreasing the moisture content of biological wastes and the plastics with high proportions in solid wastes have an effect to increase the heating value.

Rada et al. [22] presented a comparative study between two scenarios developed to manage urban solid waste elimination and recovery much better. In these scenarios different solutions were presented for selective collection, energy recovery techniques and final disposal of solid waste. Based on the results of the mass, energy and environmental analyzes applied to the systems, it was attempted to deduce which is more efficient.

Phillips and Mondal [23] presented a mathematical model for the sustainable assessment of urban solid waste in the Varanasi region of India. They stated that the disposal of solid wastewater by gasification may be the most sustainable option and this technique would have the positive effects when considering the local and environmental conditions of the zone.

Herva et al. [24] applied the "energy and material flow analysis" and "ecological footprint" methods on the municipal solid waste facilities in Portugal and presented the results of the evaluation, taking into consideration the solid waste collection and disposal facilities in the country. They have found that the environmental benefits of combined solid waste disposal methods are considerably high. According to the first of these methods, it is found that the most important system flow is the electricity produced during solid waste disposal. According to the second method, namely the ecological footprint, the composting process and the energy production units have positive effects on the environment in terms of resource savings and electricity generation.

Hossain et al. [25] conducted a study on the potential of solid waste as a renewable energy source in the municipal solid waste facilities in Bangladesh. Upon determining that the country's solid waste resources are rich in organic content, they concluded that energy from solid waste is a serious source of renewable energy in the country. It was also stated that the total daily electricity generated by the landfill gas obtained from solid waste in the biggest six cities of the country amounts to 186,408 kWh.

Arafat et al. [26] studied the environmental impacts of different municipal solid waste disposal facilities by using life cycle assessment approach. They claimed that the recycling of paper, wood and plastics is the best solution for energy recovery. Besides, they stated that domestic and organic kitchen wastes should be used for biogas production by anaerobic digestion and textile wastes should be directly burned. According to their assessment, anaerobic digestion and gasification methods perform better in the cases of energy recovery usage and environmental aspects, while the composting provides the least has the least environmental benefit.

Gupta et al. [27] presented a comprehensive overview in order to assist government units for developing a more efficient and effective urban solid waste management strategy in India.

Ozcan et al. [28] evaluated the energy potential of wastes in Turkey using different biomass types such as municipal solid waste, wastewater treatment sludge, animal

waste and energy plants. As a result of the calculations they made, the potential amount of biogas that can be obtained from municipal solid waste was found as 4.85 TWh per year. Annual potential biogas productions were calculated as 1.88 TWh for wastewater treatment sludge, 16.19 TWh for animal waste and 165.29 TWh for agricultural waste. Thus, the primary energy value of the total annual biogas potentials of the various sources was calculated as 188.21 TWh. However, they emphasized that the investments to be made for the energy recovery from waste should be performed considering the energy recovery efficiency, environmental effects and effective waste disposal.

Scarlat et al. [29] examined the total energy potential that can be obtained from landfill gas and waste incineration, taking African circumstances into account. They underlined that African countries need to develop proper waste collection strategies to increase their energy potential from waste, and they have argued that the potential for electricity generation would increase accordingly.

Tozlu et al. [30] overviewed the recent technologies applied in the municipal solid waste management and investigated the current solid waste disposal management in Gaziantep Metropolitan Municipality by revealing the energy potential of the produced landfill gas.

As it can be understood from the detailed search above, a considerable amount of work has been done in the literature under the headings: "Municipal Solid Waste Disposal, Energy Recovery from Waste and Facilities", management strategies of solid waste facilities. Many of these studies include an appropriate disposal system selection, in which researchers concentrate on the energy potentials of their countries' waste, and searching for a feasible energy cost, investment cost and a favorable environmental impact. In the remaining studies, it is seen that the aim is to develop a local or global perspective on the concept of energy from waste by means of energy, economy and environmental impact analysis methods, which have been widely used recently. It can be said that the main reason for this interest, which is shown in both academic and strategic sense, is that the waste disposal of civilized human societies

has a vital prospect and accordingly it is regarded as one of the most sustainable energy sources of energy recovery from waste.

2.3 Municipal Solid Waste Disposal Methods

In this section, an overview about waste-to-energy (WTE) technology in the world as well as that in Turkey is presented. Some important current studies on WTE are outlined. There are three fundamental types of WTE technologies [31]:

- (i) Thermal conversion methods (incineration, pyrolysis, and gasification),
- (ii) Biochemical conversion,
- (iii) Landfill.

In thermal conversion technology, the mostly used method is incineration. “*Incineration*” method is mainly the waste destruction in a furnace by controlling combustion at high temperatures. By incineration method, approximately 70 % of total waste mass and thus 90 % of total volume can be reduced [32]. The process is carried out in three steps, such as, incineration, energy recovery and air pollution control [1, 31]. Emissions from this process contain air pollutants like SO_x , CO_x , and NO_x , which may result in air pollution and health hazards. The process is carried out in a temperature range of 750 and 1000 °C, and can be coupled with steam and electricity generation processes. “*Pyrolysis*” is another method in which thermal waste treatment is taken place in an oxygen free environment. There are three types of pyrolysis methods which are conventional pyrolysis (550- 900K), fast pyrolysis (850-1250K), and flash pyrolysis (1050-1300K). The third thermal conversion method, “*gasification*”, is a process that converts MSW into CO_2 , CO and H_2O , which occurs by the reaction of MSW at high temperatures (>700 °C), without combustion, with a controlled amount of oxygen and/or steam [31]. Depending upon the reactor design and the operational parameters, the process also generates some methane and other higher hydrocarbons (HC) [10]. Various heterogeneous reactions convert the feedstock to gas in the presence of a gasification agent [9, 11-12]. The

produced combustible gas contains CO, CO₂, CH₄, H₂, H₂O, inert gases presented in the gasification agent, trace amounts of higher HCs, and various contaminants such as small char particles, tars and ash [15].

The second main group of WTE technology is biochemical conversion. It is much more eco-friendly compared with others. Biochemical conversion is primarily based on the reaction of microorganism enzymes. Biochemical conversion method is divided into two subgroups as “*Anaerobic Digestion*” and “*Composting*” [31]. In “*Anaerobic Digestion*”, MSW is collected in an oxygen free environment, which is a combination of series of biological processes in which microorganisms break down to biodegradable material. On the other hand, “*Composting*” is a biological decomposition of biodegradable solid waste under predominantly aerobic conditions. It is a natural process of recycling decomposed organic materials into a rich soil known as compost [31].

One of the mostly used and practical technology, a third main group, for WTE today is “*Landfilling*” which is a soil-based waste disposal technique that uses engineering principles to confine solid waste to smallest area possible and reduce it to the lowest allowable volume in sanitary landfill [33]. Sanitary landfill can be defined as a scientific dumping of MSW using an engineering facility which requires detailed planning and specifications, careful construction and efficient operation [34]. The schematic representation of a typical landfilling process [32] is given in Fig. 2.1.

Although landfill is the most common waste treatment techniques in the world, the developed countries prefer to reduce their MSW by using incineration technology due to reduction of approximately 70 % of total waste mass and thus 90 % of total volume. However, it is a fact that mass burning of MSW creates major environmental problems due to pollutant discharges. Therefore the best alternative technique for energy recovery for MSW is controlled methane production at landfills.

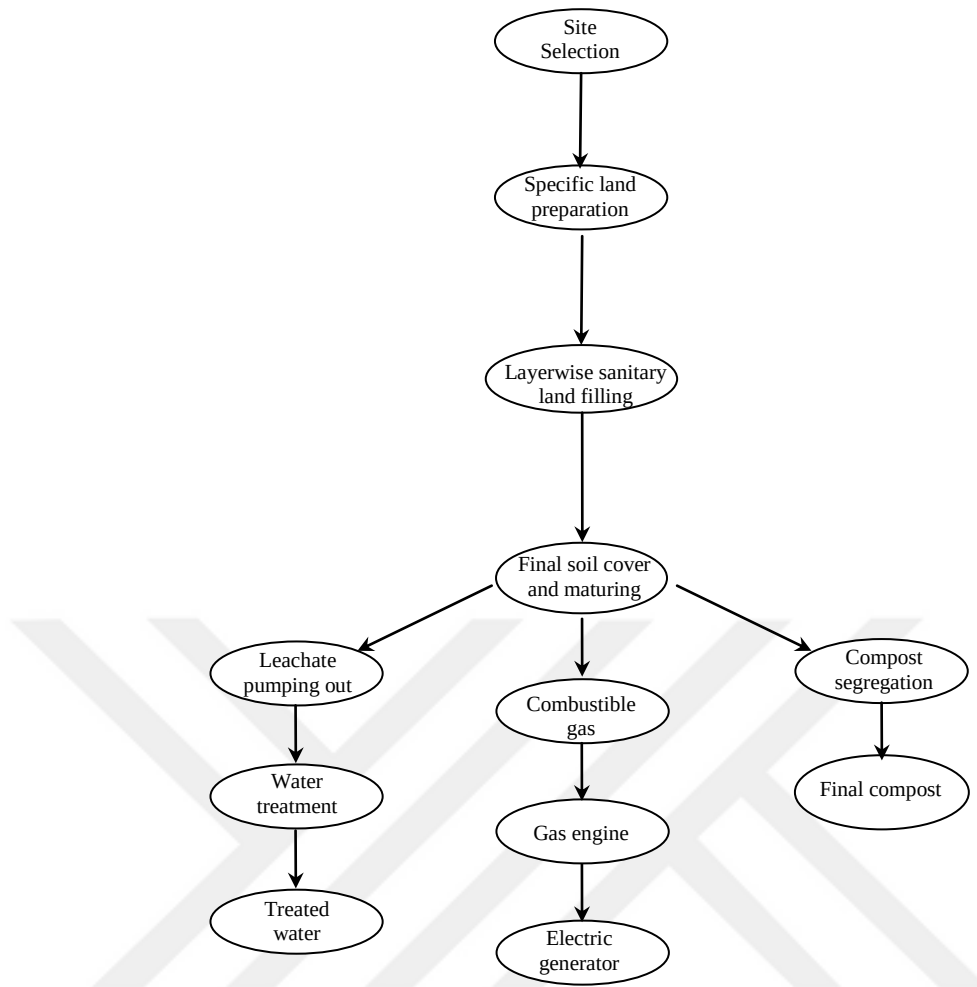


Figure 2.1. Flow diagram of MSW plant based on sanitary landfill [32].

Kumar [32] suggested that there were various options available to convert solid waste to energy which were classified in the study. The available technologies were carried out as follows: (1) sanitary landfill, (2) incineration, (3) gasification, (4) anaerobic digestion, and (5) other types. It was stated that the sanitary landfill was the scientific dumping of municipal solid waste due to which the maturity of the waste material was achieved faster and hence gas collection starts even during the landfill procedure. Incineration technology was defined as the controlled combustion of waste with the recovery of heat, to produce steam that in turn produces power through steam turbines. Kumar [32] also defined gasification technology which involves pyrolysis under limited air in the first stage, followed by higher temperature reactions of the pyrolysis products to generate low molecular weight gases with

calorific value of 1000–1200 kcal/nm³ and these gases can be used in internal combustion engines for direct power generation or in boilers for steam generation to produce power. On the other hand, in biomethanation, the putrescible fraction of waste is digested anaerobically (in absence of air), in specially designed digesters. Other available technologies were specified as pelletization, pyro-plasma, and flash pyrolysis. It was concluded that the best compromise can be chosen the technology, which had lowest life cycle cost, needed least land area, caused practically no air and land pollution, produced more power with less waste, and caused maximum volume reduction.

Kalyani and Pandey [31] presented waste to energy as a solution to solve problems about reducing the amount of waste, also produce energy from the same source. The current status, major achievements and future aspects of waste to energy in India were carried out which may help decision makers, planners and bodies involved in the management of municipal solid waste understand the current status challenges and barriers of MSWM in India for further better planning and management. There are many investigations on WTE technologies which were summarized as landfill, incineration and biochemical conversions respectively.

Chen & Lo [35] considered the greenhouse gas emissions from five different MSW treatment scenarios which includes landfilling, WTE, and material recovery and remarked that a material recovery facility or a landfill site produces less greenhouse gases than a WTE plant for handling 20000 ton/day of MSW. It was claimed that the potential of greenhouse gas mitigation from waste recycling in a material recovery facility can be considered more effective than electricity generation in a waste to energy plant. Valuable insights into the applicability of a policy framework for municipal solid waste management practices with regards to reduction of greenhouse gas emissions were proposed in the study.

Wang et al. [36] compared two different WTE approaches simple landfill incineration with energy recovery and landfill with landfill gas utilization. The effects on greenhouse gas (GHG) emission reductions in the Temperate Dry Zone and Tropical Moist Zone were investigated then it was suggested that in 2020, the

waste disposal approach will be switched from simple landfill to WTE approach in China. Landfill gas utilization and incineration with energy recovery approaches were considered and advantages were presented comparing with GHG emissions in all cases with the simple landfill. It was concluded that the landfill gas power generation system was more effective in the Tropical Moist Zone, whereas incineration with energy recovery was a better choice in the Temperate Dry Zone according to the comprehensive benefit.

Tan et al. [37] selected an actual landfill in Malaysia as the case study for consideration to adopt the advanced WTE technologies including the landfill gas recovery system, incineration, anaerobic digestion, and gasification. An interactive comparison of different WTE scenarios and further discussion on waste incineration and anaerobic digestion as the two potential WTE options were presented for Malaysia. It was reasoned that the incineration as the superior technology choice when the production of electricity and heat were considered; however, anaerobic digestion was found to be more favorable under the consideration of electricity production only.

Sevimoğlu and Tansel [38] tried to characterize the trace compounds in the LFG at the Odayeri Landfill, Istanbul. It was found out that the engine performance was significantly depended on the methane concentration of the LFG. The optimum methane concentrations and oxygen levels in LFG were also proposed in the study.

Gioannis et al. [39] investigated mechanical biological treatment (MBT) of residual municipal solid waste (RMSW) regarding to landfill gas generation comparing the raw waste with their experimental results.

Kadir et al. [40] presented the current issues and future directions as well as energy recovery initiatives associated with incineration of MSW in Malaysia where energy recovery initiatives were focused on application of WTE as well as refuse-derived fuel (RDF) technologies. An RDF pilot plant was constructed as a case study in Malaysia. It was concluded that the application of large-scale incineration technologies in Malaysia was inevitable.

Lombardi et al. [41] compared two conventional treatments which were reduced the greenhouse effect (GHE) from waste landfilling and combustion with energy recovery in reciprocating engines with three innovative possibilities: direct LFG feeding to a fuel cell (FC), the production of a hydrogen-rich gas by means of steam reforming and CO₂ capture to feed a stationary FC, the production of a hydrogen-rich gas by means of steam reforming and CO₂ capture to feed a vehicle FC. The authors looked at the issue from an environmental point of view by calculating the specific production of GHE per unit mass of waste disposed in landfill which equipped with different technologies.

Faitli et al. [42] carried out an experimental study to estimate the thermal properties of MSW samples of a selected landfill. A new apparatus was designed to measure heat flow by performing a series of systematic test on 17 different samples to calculate thermal conductivity, heat diffusivity and specific heat capacity of the landfills. Two theories were investigated for the serial and for the parallel heat flow in three phase disperse systems and also it was proposed that the measured thermal conductivity and heat diffusivity were better characterized by the parallel heat flow estimations. In addition to this, it was claimed that heat can be flow parallel in solid, liquid and gas phases with the results.

Tsai and Kuo [43] presented a comprehensive analysis of MSW-to-energy in Taiwan giving a concise summary of current status of domestic energy consumption and power generation, MSW generation and MSW incineration treatment, and electricity generation from MSW incineration plants since 2000.

Tabasova et al. [44] carried out an overview of thermal treatment methods for WTE processes technologies taking into account their performance and environmental impact and presented the possibilities of waste treatments and related legislation by the European Communities. It was resulted that 19 industrial waste incinerators in wide range of process capacities from 15 kt/y to 0.3 kt/y, and three municipal waste incinerators with nominal capacity 300, 250 and 9 kt/y were considered in Czech Republic. It was concluded that the issues related to the WTE technologies for thermal treatment of waste, heat recovery systems, flue gas issues and measures for

flue gas cleaning considering used equipment in WTE technologies were investigated.

Begum et al. [45] presented a numerical simulation model of a fixed bed gasifier for municipal solid waste (MSW) using ASPEN Plus (Advanced System for Process Engineering Plus) software. The model was developed based on Gibbs free energy minimization approach and was validated with experimental data of MSW gasification available in literature. Proximate and ultimate analysis of MSW and operation conditions used in the model development were presented and discussed. It was found that simulated and predicted composition of syngas were in good agreement with experimentally measured data (within 4 %). It was also claimed that model can be used to analyze performance of gasification process for varying operating parameters, such as air-fuel ratio, gasifier temperature and moisture content of MSW.

Nixon et al. [46] carried out a Hierarchical Analytical Network Process (HANP) model for evaluating alternative technologies for generating electricity from MSW in India. The literature and consulting experts in the field of waste management were classified for technological alternatives and evaluation criteria for the HANP in the study. It was proposed that anaerobic digestion was the preferred technology for generating electricity from MSW in India and also gasification method was preferable technology in an Analytical Network Process (AHP) model because of the exclusion of criteria dependencies and in a HANP analysis when placing a high priority on net output and retention time.

Kerdsuwan et al. [47] performed a study based on the sustainable development and eco-friendly waste management concept for these local communities using the 3R's concept (Reduced, Reuse and Recycle). It was suggested that if the separation at the source was successful, the treated waste of about 2.5 tons was divided into wet and dry fractions. Thus the wet fraction can be easily treated by conventional composting to produce soil conditioners and generate income for communities also the dry fraction should be treated by an incinerator. Therefore, the authors proposed a novel hybrid incineration-gasification system to use Refuse Derived Fuel (RDF) prepared

by dry fraction as feedstock to a downdraft gasifier. It was resulted that the producer gas generated from the gasifier can be used to substitute fossil fuel which can be used as a prototype model for other rural areas in low or low-middle income countries.

Rokni [48] carried out a MSW gasification plant integrated with Solid Oxide Fuel Cell (SOFC) which was combined with a Stirling engine to recover the energy of the off-gases from the topping SOFC cycle using detailed plant design and thermodynamic analysis. It was claimed that the gasification process integrated with SOFC and Stirling engine were increased the electricity output up of 50 % when compared with modern waste incinerators with heat recovery.

Arena [49] proposed an assessment of municipal solid waste gasification, starting from basic aspects of the process to a comparative analysis of the reactors and the environmental performances of the main commercially available gasifiers for municipal solid wastes. It was indicated that gasification was a technically viable option for the solid waste conversion, including residual waste from separate collection of municipal solid waste. It was concluded that it was able to meet existing emission limits and a remarkable effect on reduction of landfill disposal option.

Arafat and Jijakli [50] modeled a gasification method as an alternative process to incineration for energy recovery from MSW. Both processes were modeled for bulk MSW as well as segregated MSW streams in this study. Experimental gasification data from the literature was tested then reasonably match the trends of that data was found. It was resulted that the amount of energy produced per kg of mixed MSW was shown to depend on the relative composition of the MSW. It was also claimed that especially in developed countries, with smaller fractions of food and yard wastes and higher fractions of plastic and textile in the MSW, higher energy can be recovered from the waste.

Zhang et al. [51] carried out a thermodynamic analysis to evaluate the advantages and limitations of the Plasma Gasification Melting (PGM) technology. Firstly, PGM processes were divided into four sections such as drying, pyrolysis, char gasification and inorganics melting and then the energy and exergy efficiencies in each section were calculated. It was resulted that the energy and exergy efficiency of PGM

process which produces a tar-rich syngas was found as very high. It was concluded that it is beneficial to increase sensible heat input to the PGM system. In this manner, high sensible heat input or high steam injection was not suggested when considering the energy and exergy efficiency of raw syngas.

Rajasekhar et al. [52] proposed a number of varieties of waste management methods and strategies for dealing with MSW. It was stated that process of plasma gasification was a successful proven technology of producing energy from waste. The authors focused to analyze the process of plasma gasification and also to propose a critical assessment of MSW and compare the performance of the process to that of the conventional energy recovery methods.

Stehlik [53] presented a number of recent advances in technologies and improvements in units for the thermal processing of MSW and various other types of waste. The achievements including low-NO_x burners, heat exchangers, waste heat recovery systems, newly developed equipment were presented for wet scrubbing, dioxin filters and systems for the treatment of sewage sludge summarizing several examples from industrial applications.

Xue et al. [54] reviewed the current MSW collection and disposal methods in Singapore taking into account problematic allocation of incineration resources. A spatial allocation model was proposed which was tap on the mass reduction capability of incineration plants to make for a more cost efficient waste collection regime. It was concluded that the allocation results were compared with the current situation in Singapore.

Bernardi & Parente [55] investigated the financial feasibility of organic solid waste as a feedstock, which resulted in the production of biodiesel through the conversion of volatile fatty acids into lipids (VFA). Based on existing references of capital and operating costs, production and extraction yields for VFA and lipids and an internal rate of return of 15 % in real terms were resulted. It was suggested that biodiesel production was competitive compared to subsidized biodiesel traded in regions of Europe and the United States.

Adani et al. [56] monitored a mechanical–biological process for MSW treatment for one year. Two fractions were provided for the mechanical pre treatment. The oversize fraction which were greater than 50 mm was used for refuse derived fuel production because of low moisture content and high calorific value and the undersize fraction which were less than 50 mm MSW organic matter was considered for anaerobic process. The self-heating test, potential biogas production and fermentable volatile solids were investigated as parameters to describe the potential impact of treated waste, providing further useful information.

Rentizelas et al. [57] presented a model which was aimed to support decision makers in issues relating to MSW energy recovery. The model was based on optimizing the system specifications, such as the capacity of the base-load WTE facility and the peak-load capacity for biomass boiler and the location of the facility. According to the authors, an energy tri-generation case study application with mixed MSW and biomass can be provided positive financial yield of investment. In addition to this, a sensitivity analysis was performed on the effect of the most important parameters of the model which were pinpointing the parameters of interest rate, investment cost and heating oil price for the optimum solution.

Fricke et al. [58] considered selected efficiency rates and process data of aerobic and anaerobic procedures which were related to mechanical–biological treatment for the treatment of MSW. It was proposed that the anaerobic digestion can be ecologically advantageous, particularly with regard to exhaust emissions and energy balances in comparison with aerobic processes.

Yang et al. [59] carried out a methodology for application to MSW management using the eco-efficiency analysis. In the beginning the environmental impact and economic costs of each process from collection to treatment were integrated into the entire disposal system and then the contribution of each process was analyzed individually to determine the eco-effective optimization measure. It was claimed that the approach was not only reveal the best measure for maximum overall environmental improvement per unit investment cost, but the approach was identified

appropriate MSW treatment technologies with respect to definitions and some factors.

Seng et al. [60] presented insight into the benefits of organic waste recycling through composting over landfill, in terms of landfill life extension, compost product, and mitigation of greenhouse gases (GHGs). Five scenarios of organic waste recycling were carried out in the municipality of Phnom Penh (MPP), Cambodia, taking into account waste generation from 2003 to 2020.

Cano et al. [61] performed an economic assessment of thermal hydrolysis as a pretreatment to anaerobic digestion to evaluate its implementation in full-scale plants using six different solid wastes. Thermal hydrolysis was tested with batch lab-scale tests, from which an energy and economic assessment of three scenarios were carried out. It was found that the thermal hydrolysis was increased to the methane productions (up to 50 %) and kinetics parameters. It was also determined that a proper energy integration design can be lead to important economic savings and thermal hydrolysis can be enhanced up to 40 % the incomes of the digestion plant. It was concluded that in a full-scale MSW treatment plant, thermal hydrolysis can be provided almost 0.5 M€/year net benefits.

Hessami et al. [62] claimed that biogas production from household organic waste was rare because existing digesters were not suitable for small-scale applications. A digester design was carried out for waste to energy conversion. It was suggested that this proposed digester was inexpensive to construct and maintain, simple to operate and highly efficient.

Zhou et al. [63] classified 26 kinds of MSW component samples using cluster analysis method according to the proximate and ultimate analyses and heating value results. The classification were consisted of many groups included vegetables, starch food, orange peel, wood waste, printing paper, cellulose, PVC, PET, PE/PP, PS, and rubber. According to the classification, typical components can be selected for thermal conversion in waste-to-energy research.

Zhang & Huang [64] carried out a fuzzy possibilistic integer programming (FPIM) model for waste management facility expansion and waste flow allocation planning with consideration of GHG emission trading in an MSW management system. The aim was to show the interrelationships between MSW management planning and GHG emission control. The scenario of total system GHG emission control was analyzed for reflecting the feature that tradable GHG emission. It was resulted that GHG emission credit trading can be decreased total system cost through re-allocation of GHG emission credits within the entire MSW management system.

Although the methods of solid waste disposal vary according to the level of development of countries, each country resort to ways of disposing solid wastes to the extent of their own possibilities. Thus, solid waste disposal techniques are improved day by day. As a result of this developments, according to the mentioned studies above, there are still many attempts conducted on which disposal methods of MSW is more beneficial and eco-friendly one.

2.4 Landfill Gas (LFG) and Electricity Production Potentials from MSW Power Plants

In this section, a literature survey on electricity generation from the landfill gas produced in municipal solid waste facilities is given. Holanda and Balestieri [65] proposed two cogeneration schemes for the burning of municipal solid wastes, associated or not with natural gas, and examined their technical and economic feasibilities. The first proposed scheme involved a steam-based cycle using MSW combustion: the technical and economic feasibility evaluation was concluded especially because of its environmental and social benefits. Second scheme was based on a combined cycle which has a turbine burning natural gas in its topping unit and a conventional steam-generator burning MSW in its bottoming unit. Because of the longer payback periods of second proposed scheme, first proposed scheme was considered as more feasible in the study.

Verbruggen [66] proposed proper methods for measuring cogeneration or combined heat and power (CHP) activities based on design characteristics of the plants. It was investigated that the cogenerated electric output was a necessary and sufficient indicator of CHP advantage and performance. It was also remarked that regulators could be extended this indicator, but should be avoided the perverse effects of biased external benchmarking as the EU Directive entails.

Sevimoğlu and Tansel [67], investigated the performances of gas engines operated with landfill gas (LFG), which were affected by the impurities in the LFG, reducing the economic viability of energy recovery. The aim was to characterize the trace compounds in the LFG at the Odayeri Landfill, Istanbul, Turkey, which was used for energy recovery. Composite gas samples for trace compounds (hydrocarbons, siloxanes, and volatile halogenated hydrocarbons) were collected and analyzed over a 3-year period. It was resulted that the pretreatment of LFG was necessary to protect the engines at the waste-to-energy facilities with persistence levels of siloxanes and volatile halogenated hydrocarbons and it was affected to decrease the operating costs associated with engine repairs.

Raj et al. [68] presented a detailed literature survey of cogeneration technologies based on renewable energy sources like biomass, solar energy, fuel cell, and waste heat. Various designs, numerical and simulation models, key development areas, economic and environmental considerations were investigated in the study. It was suggested that the study could be useful for the researchers in cogeneration technologies to make effective decisions and generate more ideas.

Çakır et al. [69] studied the environmental benefits and sustainability aspects of cogeneration systems and importance of those systems to the use of sustainable energy. Two running modes of cogeneration system were investigated for the hospital. One of them was based on the system for supplying only hospitals own electrical demands in different running loads from 50 % to 100 %. The second mode was to run the system for 8000 h in a year at full load and sell the extra generated electricity to the electrical company. It was claimed that these systems could represent not only a saving in power and costs for the hospital, but also constitute a

safeguard against possible failures in the electrical supply, reducing the external dependency.

Carolino and Ferreira [70] emphasized that the limitations in the world sources of energy were mitigated by the exploitation of renewable forms and by increased in the efficiency of energy utilization. The study covered first and second law analyses of a cogeneration system run with the biogas produced in a MSW in Porto. The objectives were to identify locations where major irreversibilities occur, to evaluate their magnitudes and to assess the energy and exergy efficiencies of the global system and of its constituent units. It was resulted that the overall-plant first law efficiency was 37.9 % and the exergy efficiency was 36.2 %. The internal combustion engine and one of the radiators were the most inefficient units when thermodynamic perfection and exergy destruction quotient were taken into account. According to the researchers the main potential for improvement in the plant was the harnessing of the energy in the exhaust gases.

Barigozzi et al. [71] focused on a waste-to-energy plant located in Italy, that produced electric power and thermal energy from the non-recyclable fraction of municipal and industrial solid waste. The condenser system was organized with an air condenser and a water cooled condenser, coupled with a wet cooling tower in this plant. It was claimed that how the net power output can be maximized by properly regulating the combined wet and dry units of the combined cooling system. A detailed model of the steam cycle was performed by means of a commercial code (Thermoflex) and resulted that as a general rule, heat rejection was more efficient in the water cooled condenser (WC) than in the air cooled condenser (AC). Conversely, the AC was turned out better than WC when a small amount of steam was sent to condensation, in the coldest period of the year.

Bianchi et al. [72] investigated an innovative and promising strategy to improve waste conversion through integration of a conventional waste to energy (WTE) power plant with a Gas Turbine (GT). It was suggested that WTE–GT integrated plant was required to conceive a redesign and an optimization of the steam-gas cycle. The study focused on the feasibility of utilizing the hot gases leaving the GT to

superheat the steam leaving the WTE steam generator, as well as heating the feed water returning to the steam generator of the WTE condenser. Parametric analysis of the effect of the GT discharged heat on the steam mass flow production was carried out and the optimum plant match condition in terms of plants capacity ratio was identified. Also, detailed modifications to the WTE cycle and the resulting enhancement of its performance were presented. As a result of the study, guidelines to create new advanced WTE–GT integrated power plants or to repower existing low-performing WTE power plants were proposed in order to increase waste conversion into energy.

2.5 Thermodynamic and Thermoeconomic Analyses

There are a lot of parameters in power plants that should be take into account for improvements of system efficiencies. The efficiency improvement is a big problem for the companies which are not only related with technical reasons, but also it is fundamentally related with economic reasons. Because, maintenance cost and consumed fuel are directly effects to the solution of this problem. Studies of this field in the literature are outlined as follows:

Kwak et al. [73] analyzed 500-MW combined cycle plant in view of exergetic and thermoeconomic aspects. An exergy-costing method, MOPSA was used to estimate the unit costs of electricity produced from gas and steam turbines. A computer program was designed showing that the exergy and the thermoeconomic analysis could be applied to any energy systems. The unit costs of products can be estimated with accurate information of the initial investments, salvage values and maintenance costs for each component using the program. Productive structure which represented the relations between components in the energy system was shown in the exergy-costing model.

Benar et al. [74] performed life cycle assessment (LCA) methodology for determination the optimum MSW management strategy for Eskisehir city in Turkey. Five different scenarios were proposed as alternatives to the current waste

management system, which were collection and transportation of waste, a material recovery facility (MRF), recycling, composting, incineration and landfilling processes using SimaPro7 libraries for the life cycle inventory. The authors compared the abiotic depletion, global warming, human toxicity, acidification, eutrophication and photochemical ozone depletion resulting with CML 2000 method. According to the comparisons and sensitivity analysis, it was concluded that the composting scenario, S3 (15 % recycling + 77 % composting + 8 % landfilling) was found to be the more environmentally preferable alternative.

Abuşoğlu and Kanoğlu [75] performed energy, exergy, and exergoeconomic analysis for a diesel engine powered cogeneration (DEPC) plant. The authors presented the study in two parts which were the formulation developed for such a comprehensive analysis and application of the developed formulation that was considered as an actual cogeneration power plant respectively. A diesel engine powered cogeneration plant was analyzed with all related components. Mass, energy, and exergy balances were performed on the system components for determination of work and heat terms and exergy destructions. The definitions of energetic and exergetic efficiencies and the performance assessment parameters were carried out for the system components and the cogeneration plant. As a method of diesel engine powered cogeneration plant thermoeconomic analysis was given with the equations in order to perform economic analysis for this type of systems. In part 2, an application of the evaluated formulation was carried out considering an actual DEPC plant installed in Gaziantep, Turkey. Specific cost method (SPECOC) was used to perform the exergoeconomic analysis for the power plant using actual plant data [76].

Abuşoğlu and Kanoğlu [77] also reviewed a brief historical overview on the exergoeconomic analysis and optimization for combined heat and power production (CHPP). In this study, available thermoeconomic methodologies in literature were represented and their advantages and disadvantages were compared and discussed through a well-known problem, namely CGAM. Thermoeconomic analysis and optimization of combined heat and power production were listed based on the methodology used and the type of system considered.

Gungor et al. [78] performed an exergoeconomic analysis of a pilot scale gas engine driven heat pump (GEHP) drying system based on the experimental values using Exergy, Cost, Energy and Mass (EXCEM) analysis method in the study. Performance of the drying system components were considered to improve the system efficiency. Performance of the drying process was also analyzed for three different medicinal and aromatic plants from the exergoeconomic point of view. The effect of varying dead (reference) state temperatures on exergoeconomic performance parameters was investigated for both drying system components and drying process using a comprehensive parametric study. It was resulted that the dead state temperature affected the performance parameters, particularly the drying process parameters. Increase in the dead state temperature was led to an increase in the exergy efficiencies of the drying process and a decrease in the ratio of the thermodynamic loss rate to the capital cost values in a polynomial form. It was concluded that the capital cost values of the drying process was very higher when compared to those of the drying system components.

Abuşoğlu et al. [79] presented the thermoeconomic analysis and assessment of a municipal wastewater treatment system using specific exergy costing (SPECO) method which was used to allocate cost flows through subcomponents of the plant which was described in detail. Actual operational plant data were used to perform thermoeconomic analysis for Gaziantep waste water treatment plant and they concluded that, better exergetic performance and cost effectiveness can be achieved by reducing exergy destruction through better operation conditions as well as by reducing operational and exergy destruction costs through all WWTP subcomponents.

Ahrenfeldt et al. [80] presented a series of different gasification process designs to give insight to the wide range of possibilities within this process. The authors focused on state-of-the-art processes with high reliability, flexibility and efficiency within cogeneration of energy from conversion of biomass. Cogeneration processes were compared with respect to production of heat-and-power, heat-power-and fuel or heat-power-and-fertilizer which were also described in their study. It was concluded that the optimal choice of technology for a given task was dependent on many factors

including feed stock availability, knowhow, project economy, local politics, environmental concerns and LCA considerations in addition to the desired product and process characteristics and demands.

Abusoglu et al. [81] presented a thermoeconomic analysis and assessment of a municipal wastewater treatment system. Operation of an existing municipal wastewater treatment plant was described and a thermoeconomical methodology based on exergoeconomic relations and the specific exergy costing method was performed using actual operational plant data. Their results provided important information for identification of the sites with greater exergy destructions and consequently greater potential for improvements.

Fernandez et al. [82] analyzed six different scenarios for managing the MSW in order to obtain environmental impacts applying the Life Cycle Analysis methodology in Asturias (Spain). The authors comparatively carried out the effect of these scenarios based on human health, ecosystem quality, global warming and resource depletion.

Erbay and Hepbaşlı [83] performed the conventional and advanced exergoeconomic analyses for a heat pump food dryer using different drying air temperatures. The components were analyzed separately for the drying system and investigated the effects of the operating temperature on the system components. It was concluded that HRU was the most important system component with respect to cost accumulation. It was stated that the increase in the inlet drying temperature was caused to lower costs and higher improvement potentials. It was also suggested that optimization studies which were related with cost efficiencies and product quality should be considered.

Turan and Aydın [84] performed exergetic and exergo-economic analyses of an LM6000 gas turbine engine derived from CF6-80C2 aircraft turbofan engine. Balances of exergy and exergy costs were considered for each component and LM6000 gas turbine engine in order to match exergetic and economic values. In addition to this, exergy cost rate and unit exergy cost rate of the LM6000 were evaluated using exergo-economic results. It was stated that the results of the study

might be useful to define the cost flow of fuel, products, and destructions and to regulate operation conditions and maintenance of aero-derivative gas turbine fleet.

Smith et al. [85] presented a methodology which described an approach to account for costs and potential environmental impacts for industrial ecology to view residual waste as having potential to be converted into useable products and energy. The authors performed the benefits of conversion and adds process costs and potential environmental impacts from the various steps of processing.

Rajaeifar et al. [86] performed the inventory data in support of life cycle assessment of different MSW with proposing scenarios based on anaerobic digestion (AD), landfilling combined with composting, incineration, incineration combined with composting, and AD combined with incineration. In addition to this, six supplementary files were presented including inventory data on the different scenarios, data on the different damage assessment categories, normalization and single scores. As a result of the analysis, it was claimed that the most eco-friendly scenario was the combination of AD and incineration and also it could be implemented in the future.

Rajaeifar et al. [87] reviewed different MSW scenarios which were anaerobic digestion (AD), landfilling combined with composting, incineration, incineration combined with composting, and AD combined with incineration using a comparative life cycle assessment approach in Iran. It was resulted that incineration combined with composting and landfilling combined with composting scenarios were the most adverse environmental impact in the Human Health and Ecosystem Quality damage categories and landfilling combined with composting scenario has the least helpful effect on the Resources damage category.

A Rankine system consisted of solar radiation and fuel combustion which was constructed to produce electrical power was investigated from exergoeconomic point of view by Cavalcanti and Motta [88]. The exergoeconomic parameters which were exergoeconomic factor and relative cost difference for each component were evaluated in the study using specific exergy costing SPECO method. The simulation of system was performed with the use of actual data on March, June, September and

December 21st from 7 am to 4 pm for Natal region in Brazil. Daily average values of collector efficiencies, ratio of the useful solar energy, electricity produced, the specific cost per exergy unit of the produced electricity and others heat rates were examined. It was stated that the higher solar radiation could provide more advantageous for the system.

Khaljani et al. [89] performed a thermodynamic, exergo-economic and environmental assessment of a cogeneration of heat and power cycle (CHP), considering three objective functions of first and second law efficiencies and the total cost rates of the system. It was resulted that the most exergy destruction rate was taken place in the combustion chamber, and after that in heat recovery steam generator and gas turbine, respectively. The exergo-economic factor was calculated for the whole cycle as 10.59 % showing that the exergy destruction cost rate was more than capital investment cost rate. A parametric study was carried out in order to determine the effects of design parameters on the objective functions. It was claimed that the increase in pressure ratio and isentropic efficiency of air compressor and gas turbine efficiency improved thermodynamic performance of the system. On the other hand the more increase of these parameters affected the total cost rates negatively. It was suggested that the increase in air preheater exiting temperature could be useful for the system both in terms of thermodynamic and exergo-economic and the increase in the condensation temperature and pinch point temperature difference of evaporator of ORC and increase in pinch point temperature difference of heat recovery steam generator could caused the first and second law efficiency to fall down and also could increase the total cost rates of the system.

Karaali and Öztürk [90] performed a different thermoeconomic optimization method which was simple and efficient, for real complex cycles. The authors firstly suggested a thermoeconomic analysis method namely non-linear simplex direct search method which was enhanced for the aim. This method was applied to four cogeneration cycles that were simple cycle, inlet air cooling cycle, air preheated and air-fuel preheated cycles for analyzing and optimizing, which was the main purpose of the study. The four cycles were thermoeconomically optimized by using the cost equation method (method B) and the effect of size on equipment method (method A).

It was concluded that the costing method B and the costing method A had different optimum costs results. Costing method B was found to be better for real cases, but unrealistic results were attained with different excess air rates. According to the authors, the best solutions could be obtained in the case of the variable demands for electricity and steam mass production, the air and the air-fuel preheated cycles.

2.6 Gas turbine, ORC and Kalina Cycle

Closed cycle gas turbine power generation systems have a significant potential to produce power and also supply conventional coal-fired power generation systems and internal combustion (open cycle) gas turbine power generation systems. Closed cycle gas turbines which have popularity in 1950s and 1970s are lost its influences because of higher efficiency of open cycle gas turbines which is related to higher firing temperature. Nevertheless in the recent past, closed cycle gas turbines are considered again as an alternative energy conversion method because of the working fluid technologies [91]. In order to power production, all gas turbine systems are operated by means of Brayton cycle principles. Gas turbines can be classified into three groups which are open cycle in which air is the working fluid, closed cycle with air or other fluid as working fluid and semi-closed cycle. In addition to this, heat sources such as fossil, biomass, nuclear and solar are another parameter for classification of closed cycle gas turbines. Low grade heat sources of gas turbine and diesel engine exhausts are recently considered to supply global energy requirement due to its significant energy potential [91]. Power cycles using supercritical CO₂ (S-CO₂) were investigated and developed for energy recovery from low grade heat sources effectively. Moreover, Echogen Power Systems Company designed a waste heat recovery system which had 8 MW power production capacity using CO₂ and S-CO₂ as a working fluid and also this designed system was the first system in this field with its commercialization in 2014 [92]. There are also many studies in this field in the literature which are outlined with respect to heat sources as follows;

Ahn et al. [93] carried out a review of S-CO₂ power cycle technology and its current development progress and a comparative study with respect to various S-CO₂ power

cycle layouts using sodium cooled fast reactor as a heat source of the system. It was claimed that the efficiency of S-CO₂ cycle was found to be higher within the turbine inlet temperature range (450-600°C) when compared with the other power conversion systems. It was resulted that the recompression cycle layout was the most efficient cycle among the others.

Akbari and Mahmoudi [94] performed an exergoeconomic analyses of a S-CO₂ power cycle and combined power cycles S-CO₂/organic Rankine cycle (S-CO₂/ORC). Eight different types of working fluids were investigated in ORC during the thermodynamic and thermoeconomic analyses. Pinch point temperature difference in precooler and condenser, the compressor pressure ratio and the ORC turbine inlet temperature were considered using Engineering Equation Solver (EES) software in order to compare performances of both systems. It was concluded that the exergetic efficiency of combined cycle was found to be higher and the total product unit cost was found to be lower as compared to S-CO₂ cycle. It was resulted that the highest exergetic efficiency and the lowest product unit cost for combined cycle were achieved when Isobutane and RC318 were considered as an ORC working fluid, respectively.

Cha et.al. [95] carried out a set of researches which were turbine, compressor and heat exchanger designs and their flow analyses to develop a S-CO₂ cycle coupled to Kalimer-600 sodium cooled fast reactor. It was resulted that the cycle efficiency and the plant net efficiency were found as 42.8 % and 40.3 %, respectively using CFD (computational fluid dynamics) analysis.

Ishiyama et al. [96] analyzed steam, helium and S-CO₂ cycles in a prototype fusion power reactor. It was resulted that the S-CO₂ turbine cycle was found to be the most efficient one with respect to other cycles in the medium temperature regime. When the heat source temperature was at 480 °C, cycle efficiencies of the helium turbine, steam turbine and S-CO₂ turbine cycles were evaluated as 34 %, 40 % and 42 %, respectively. It was concluded that the decrease in heat source were caused to decrease in efficiency for all cycles. Moreover, helium turbine cycle was not considered when heat source temperature was at 250 °C.

Mohammadkhani et al. [97] carried out an exergoeconomic analysis and parametric study for a combined S-CO₂/ORC system. The heat source of S-CO₂ cycle was supplied by a modular helium reactor core which had the highest cost in the combined system. On the other hand the minimum cost was found for the ORC pump. The relative cost difference (r) and exergoeconomic factor (f) of ORC condenser, precooler and intercooler were found to be more than 80 % and less than 10 %, respectively. It was resulted that the effect of pinch point temperature difference in evaporator was not a significant parameter on exergoeconomic performance of system. It was claimed that the thermodynamic optimizations were not related with optimum cost values directly.

Wang and Dai [98] performed a thermodynamic and exergoeconomic analysis for a gas turbine modular helium reactor using two T-CO₂ cycles. It was found that 22 MW additional power can be provided to the system using the T-CO₂ cycles. The energy efficiency, the exergy efficiency, the total exergy destruction cost rate, the overall exergoeconomic factor and the total cost rate of the system were evaluated in order to consider the key parameters of the combined system. The energy efficiency of GT/T-CO₂ cycle was found to be higher than GT cycle when the reactor outlet temperature was at 850 °C. A specific pressure ratio of compressor was evaluated for minimization of total cost rate. It was resulted that the increase in turbine inlet temperature resulted in reduction of total cost rate and also energetic and exergetic efficiencies were also affected positively with this.

Wang and Dai [99] compared two combined cogeneration cycles which were S-CO₂/ORC and S-CO₂/T-CO₂ by means of thermodynamic and exergoeconomic analyses. The key parameters were defined as the first and second law efficiencies and total product unit cost during the study. It was concluded that the performance of S-CO₂/T-CO₂ was found to be higher than that of S-CO₂/ORC system with respect to compressor pressure ratio, turbine inlet temperature. In contrary to this, the total product unit cost of S-CO₂/ORC system was evaluated to be lower than S-CO₂/T-CO₂ when working fluid was isobutane in ORC system.

Wang et al. [100] also presented another paper in which an exergy and exergoeconomic analysis on a combined recompression S-CO₂/T-CO₂ power cycles were performed. It was resulted that the highest exergy destructions were found for the reactor and components of the T-CO₂ cycle of the system. It was also claimed that the reducing of the amount of the total cost rates of turbine and reactor resulted in an increase in outlet temperature of the reactor. It was concluded that the total cost rate of the system was decreased linearly when the cooling water temperature decreased.

Kouta et al. [101] designed a S-CO₂ cycle for power production from solar tower namely solar cogeneration system in six different cities in Saudi Arabia. A new efficiency equation was proposed as a result of the average power and water production according to monthly considerations. It was also found that the recompression solar cogeneration cycle had higher efficiency with respect to regeneration solar cogeneration cycle.

Milani et al. [102] performed an optimization for a recompression S-CO₂ cycle using solar and fossil based heat sources in Aspen Plus environment. It was proposed that the optimization was maximized solar field contribution and minimized the role of auxiliary fossil fuelled backup heating system. It was concluded that higher turbine inlet temperature was required in order to increase thermal efficiency regarding to working fluid circulation rate and compression work.

Nami et al. [103] carried out exergy, economic and environmental impact analysis and optimization study for a combined S-CO₂/ORC cogeneration system. It was resulted that about 40 % of exergy was used in order to generate electrical power output, 12 % was used for steam production and the rest of exergy was destructed. The cost of steam, the total exergy destruction cost rate, the unit cost of electricity and the average product unit cost were minimized using optimum compressor pressure ratio of 5.661 and turbine isentropic efficiency of 85.17 %.

Ahmadi et al. [104] studied a transcritical CO₂ (T-CO₂) power cycle driven by geothermal energy to generate electricity. Thermodynamic and thermoeconomic analyses were carried out in order to investigate the effect of key parameters and to

find out the important components regarding to exergoeconomic criterion on the cycle performance. As a result of exergoeconomic analysis, it was concluded that CO₂ turbine and condenser had the highest flow cost rates in comparison to other components. Exergetic efficiency and product cost rate of the system were found as 20.5 % and 263592.15 \$/year, respectively, by using ideal solution decision making technique. In addition to this, exergetic efficiency and product cost rate of the system were also evaluated with linear programming and FUZZY decision making techniques resulting better efficiency and cost rate values.

Kim et al. [105] compared nine different S-CO₂ power cycles using a 5MW landfill gas turbine waste heat as a source. Thermodynamic performance parameters were investigated for all power cycles by means of sensitivity analyses. The compressor inlet temperature variation with regarding to heat source inlet temperature was also considered to evaluate S-CO₂ cycles sensitivity. It was resulted that a recompression cycle was not suitable for the bottoming cycle applications. It was also concluded that the recuperation process was critical when compared to the intercooling process for designing the S-CO₂ cycle and it had an influence to increase the thermal efficiency and the net power production.

Besides all these studies Olumayegun et al. [91] reviewed closed-cycle gas turbines according to the historical development in chronological order and also features such as heat sources, working fluids, heat exchangers and cycle layouts/configurations. Furthermore, close-cycle gas turbine systems in open literature were also considered with respect to modeling and simulation studies, operation and control strategies. The difficulties in cycles in decision stage and potential future works were summarized in different viewpoints such as heat source technologies and power conversion system.

Organic Rankine cycle (ORC) is an important low grade thermal energy recovery technology because of its small scale feature. ORC can be used to all kinds of low-temperature heat sources including geothermal energy, solar energy, biomass energy, and especially waste heat energy. Drescher and Bruggemann [106] presented a procedure to calculate ORC efficiency with sufficient accuracy based on the DIPPR database and the Peng–Robinson–EOS. The authors aimed to find thermodynamic

suitable fluids for ORC in biomass power plants. It was resulted that the efficiency was correlated with a low minimum temperature, high vaporization temperature and a high amount of vaporization enthalpy to input enthalpy and if these parameters were optimal, low maximum and minimum process pressure leaded to a slight improvement of efficiency. The conclusion of the study showed that the best operating values for a biomass plant was a maximum process temperature of 573 K, a maximum pressure between 0.9 and 1.5 MPa. It was also concluded that the family of alkyl benzenes provided the highest efficiencies.

Hung et al. [107] investigated Rankine cycles using organic fluids as working fluids in converting low-grade energy with the main purpose of describing suitable working fluids which might yield high system efficiencies in an ORC system. The efficiencies of ORC systems were evaluated using such parameters which were turbine inlet temperature, turbine inlet pressure, condenser exit temperature, turbine exit quality, overall irreversibility, and system efficiency assuming the inlet condition of the working fluid entering turbine was in saturated vapor phase. It was indicated that wet fluids with very steep saturated vapor curves in T-s diagram had better overall performance in energy conversion efficiencies than that of dry fluids. It was also suggested that all working fluids have a similar behavior of the efficiency-condenser exit temperature relationship. Finally it was concluded that an appropriate combination of solar energy and an ORC system with a higher turbine inlet temperature and a lower condenser temperature could provide an economically feasible and environment-friendly renewable energy conversion system.

Bao and Zhao [108] reviewed the selections of working fluids and expanders for organic Rankine cycle, including an analysis of the influence of working fluids category and their thermodynamic and physical properties on the organic Rankine cycle's performance. The authors also presented a summary of pure and mixed working fluids screening researches for organic Rankine cycle, a comparison of pure and mixture working fluids applications and a discussion of all types of expansion machines operating characteristics. It was resulted that it would be beneficial to select the optimal working fluid and suitable expansion machine for an effective organic Rankine cycle system.

Kaska [109] carried out an energy and exergy analysis of a waste heat driven ORC and determined performance of the cycle and pinpoint sites of primary exergy destruction using actual operational plant data. In addition to this, variations of energy and exergy efficiencies of the system were tabulated with evaporator/condenser pressures, superheating and subcooling. It was concluded that the components which were evaporator, turbine, condenser and pump had greater exergy destructions to lower, respectively.

Tchance et al. [110] presented various Rankine cycle architectures for single fluids and other improved versions operating with ammonia/water mixture. The waste heat resources and their potential were sketched for driving ORC. It was claimed that the nature state and temperature of the heat source significantly influenced the choice of the type of ORC machine and also the temperature appeared as a critical parameter during the selection process. The maximum thermal efficiency of module was calculated as 25 %. It was concluded that the selection of a module primarily was dependent on application, heat source temperature and desired power output.

He et al. [111] categorized the low-grade heat sources coupled by ORC (organic Rankine cycle) into two groups. These groups were presented as the inlet temperature and the mass flow rate. The working mass of the heat source was directly discharged after being used and the heat release was specific and the working mass of the heat source was usually recycled after releasing heat. It was claimed that the selection of working fluids for subcritical ORC should be coupled with the types of low-grade heat sources. For the first heat source, it was resulted that the working fluids with high liquid specific heat and low latent heat of evaporation should be selected as the working fluids using theoretical analysis and numerical simulation. On the other hand, it was also concluded that the working fluids with low liquid specific heat and the high latent heat of evaporation were better for the second heat source.

Di Maria et al. [112] analyzed the energetic performance of an ORC system fueled by the heat generated from the integrated aerobic/anaerobic treatment of organic waste. The temperature and heat content of the exhaust air were increased arising

from the aerobic treatment by the combustion of the biogas produced by the anaerobic digestion of a fraction of the same waste. After the investigation, it was claimed that the best energetic utilization of the biogas was achieved for ORC compression ratios from 1.5 to 2 and for maximum air temperatures from 335 to 340 K and in these conditions, by using a micro-ORC system, it was possible to convert about 20 % of the energy content of the biogas into electrical energy.

Di Maria and Michale [113] performed the amount of heat rejected by the exhaust air generated by the aerobic treatment of organic waste (OW) with the aim of evaluating the amount of electrical energy recoverable by a micro organic Rankine cycle. The authors carried out energetic and exergetic analysis along with an evaluation of the investment costs for a full scale facility processing 32000 ton/year of OW and resulted that the average exhaust air rate was about 4000 Nm³/h with a temperature of 341 K and a relative humidity of 100 %. The net power output of the micro-ORC was evaluated in the range of 2 kW and 20 kW and the net electrical efficiency decreased from 5 % to about 2 % also the exergy efficiency decreased from 11 % to 1%. Finally it was calculated the specific investment ranging from about 2,800 €/kW to about 3,900 €/kW and the cost of the electrical energy ranging from of about 0.1 €/kWh to about 0,13 €/kWh.

Galloni et al. [114] investigated the prototype of a small ORC power plant using R245fa as working fluid at the Energy Systems Laboratory of Cassino University. The aim of the study was to assess the feasibility of small-scale ORC plants. During the investigation, hot source and cold sink temperature and R245fa vapor maximum pressure were in the range of 75-95 °C, 20-33 °C, and 6-10 bar, respectively. In this operating range, it was resulted that the best option of electric power, specific work and cycle efficiency was found as 1.2 kW, 20 kJ/kg and greater than 9 %, respectively.

Andreasen et al. [115] presented a novel organic Rankine cycle layout, named the organic split-cycle, designed for utilization of low grade heat. The authors developed this, layout by implementing a simplified version of the split evaporation concept from the Kalina split-cycle in the organic Rankine cycle in order to improve the

boiling process. Eight hydrocarbon mixtures were investigated for hot fluid inlet temperatures at 120 °C and 90 °C, using a genetic algorithm to determine the cycle conditions for which the net power output was maximized using optimizations. It was resulted that the most promising mixture was an isobutene/pentane mixture, for 90 °C hot fluid inlet temperature case, achieving 14.5 % higher net power output than an optimized organic Rankine cycle using the same mixture.

Desai and Bandyopadhyay [116] carried out thermo-economic comparisons of organic Rankine and steam Rankine cycles powered by line focusing concentrating solar collectors. A simple selection methodology was proposed based on thermoeconomic analysis, and a comparison diagram for working fluids of power generating cycles. It was concluded that concentrating solar power plants with any collector technology and any power generating cycle could be compared using the proposed methodology.

Muhammad et al. [117] performed an experimental investigation of a small scale (1 kW range) organic Rankine cycle system with R245fa as working fluid for net electrical power output ability, using low-grade waste heat from steam. A system was designed for waste steam in the range of 1–3 bar, then thermodynamic simulation was carried out. The thermal efficiency of the system, net efficiency, and expander isentropic efficiency were determined as 5.64 %, 4.66 %, and 58.3 %, respectively at maximum power output operation point. It was concluded that expander and screw pump were lost power in electric and mechanical losses presenting a need of further development of these components for better efficiency.

Ozdil et al. [118] presented thermodynamic analysis of an ORC in a local power plant which was located at southern of Turkey and analyzed system components separately using actual plant data and performance cycle taking into account of the relationship between pinch point and exergy efficiency. It was resulted that when the pinch point temperature was decreased, the exergy efficiency was increased because of the low exergy destruction rate. Energy and exergy efficiencies of the ORC were calculated as 9.96 % and 47.22 %, respectively. Exergy efficiency of the ORC for different water phases was also calculated as 41.04 % for water mixture form which

had quality of 0.3, 40.29 % for water mixture form which had quality of 0.7, 39.95 % for saturated vapor form. It was resulted that exergy destruction rates of the system were 520.01 kW for saturated liquid form, 598.39 kW for water mixture form which had quality of 0.3, 609.5 kW for water mixture form which had quality of 0.7 and 614.63 kW for saturated vapor form. It was concluded that evaporator had important effect on the system efficiency in terms of exergy rate.

Safarian and Aramoun [119] presented a theoretical framework for the energy and exergy evaluation of a basic as well as three modified ORCs which were considered as incorporating turbine bleeding, regeneration. It was resulted that evaporator had major contribution in the exergy destruction which was improved by increase in its pressure and the integrated ORC with turbine bleeding and regeneration had the highest thermal and exergy efficiencies and the lowest exergy loss due to decrease in cold utility demand and high power generation.

Li et al. [120] performed simulations based on the engineering equation solver (EES) software programme to determine the suitable working fluid for the simple ORC system in different temperature ranges. The influence of different organic working fluids were considered on the efficiency of the subcritical ORC power generation system under the condition of various temperatures and a constant thermal power of the flue gas and also its efficiency and other parameters were evaluated with those of the regenerator system. It was suggested that the efficiency of the subcritical ORC system was the best option when the parameters of the working fluid in the expander inlet were in the saturation state. It was also resulted that the R245fa was better than other working fluids when the flammability, the toxicity, the ozone depletion, the greenhouse effect and other factors of the working fluids were considered for the ORC. It was concluded that R601a working fluid could be preferred for the high-temperature heat source because of its high flammability.

Li [121] investigated many working fluid candidates for various ORC applications based on the heat source temperature domain for the thermal efficiency, exergy destruction rate and mass flow rate under different ORC configurations. It was found that the thermal efficiency could be increased when the critical temperature of the

working fluid was increased. It was also resulted that the ORC within heat exchanger had a higher thermal efficiency than the baseline ORC, the reheat ORC thermal efficiency was close to the baseline ORC, and the regenerative ORC could be achieved to have higher thermal efficiency than the baseline by reducing the addition of heat from the evaporator heat source.

Pu et al. [122] carried out a small scale ORC system capable of generating electric power using a low temperature heat source. A single stage axial turbine expander was used and R245fa and the new environmentally friendly HFE7100 were selected as the working fluids in the experimental system. It was concluded that the maximum electric output was found to be 1979 W and 1027 W from the turbine expander for R245fa and HFE7100, respectively.

The ever-increasing energy requirement forces researchers to investigate for alternative energy sources and to improve the efficiency of power generating systems. In this respect, many investigations were carried out in order to improve system efficiencies. Alexander Kalina was designed a new cycle called as the Kalina using a bottoming cycle instead of the Rankine cycle in combined cycle power plants in 1989. Usage of an ammonia-water mixture instead of water as working fluid was a possible way to improve the efficiency of steam turbine processes [123]. Power cycles with ammonia-water mixtures as working fluid utilizing different kinds of heat sources for power and heat generation were investigated with the thermo-physical properties of the mixture in Ph.D. Thesis of Eva Thorin [123]. It was stated that the mixture properties played an important role in the evaluations of the process performance and for the design of its components especially in heat exchangers. Available correlations for the thermo-physical properties were also compared and their influence on simulations and heat exchanger area predictions were investigated. As a result of the study it was found that the ammonia-water power cycles could be raised up to 32 % more power in the industrial waste heat application and up to 54 % more power in the gas engine bottoming cycle application when compared to a conventional Rankine steam cycle.

Zhang et al. [124] conducted a review study on the Kalina cycle, in order to show its advantageous as a bottoming cycle for a combined-cycle energy system as well as for generating electricity using low-temperature heat resources. Different Kalina systems and their different applications in literature were considered regarding to the heat sources and working fluid properties. It was concluded that the Kalina cycle had a better thermodynamic performance than the Rankine cycle and organic Rankine cycle with respect to both energy and exergy efficiency.

Cao et al. [125] performed a thermodynamic analysis of a Kalina cycle driven by low-grade heat source and conducted to parametric analysis selecting five key parameters which were expander inlet pressure, expander inlet temperature, concentration of ammonia-water basic solution, terminal temperature difference of high temperature recuperator (HTR) and terminal temperature difference of low temperature recuperator (LTR). In addition to the parametric analysis, an optimization was also carried out by means of Genetic Algorithm method in order to define optimum exergy efficiency of the system. It was found that, an optimum expander inlet pressure could be achieved to maximize exergy efficiency. It was also concluded that the heat recovery steam generator caused the most exergy destruction in the cycle.

Fallah et al. [126] carried out a conventional and advanced exergy analyses of a Kalina cycle using a geothermal heat source. It was found that the highest exergy destruction was occurred in the condenser, followed by the evaporator, the turbine, the LTR and the HTR with respect to conventional analysis. On the contrary to this, the condenser, the turbine and the evaporator were found to be the highest exergy destructed components of system, respectively according to advanced exergy analysis. It was also claimed that any improvements were not required for the LTR and the HTR. As a result of the advanced exergy analyses, the total exergy destruction rate of the cycle was reduced until the minimum limit of 30.63%.

Singh and Kaushik [91] performed a computer simulation of a Kalina cycle which was adapted to a coal fired steam power plant in order to generate electricity from low-temperature heat of exhaust gases. Optimum operating conditions were

evaluated for the Kalina cycle using the ammonia mass fraction in the mixture and ammonia turbine inlet pressure as the effect of key parameters. It was resulted that there was an optimum value of ammonia fraction that yielded the maximum cycle efficiency for a given turbine inlet pressure. It was also concluded that an increase in the turbine inlet pressure caused an increase in the cycle efficiency.

Singh and Kaushik [127] carried out also a study on a thermoeconomic analysis of a Kalina cycle which was coupled with an existing natural gas fired Brayton–Rankine combined cycle power plant. A well known thermoeconomic method of SPECO was used to formulate the cost balance and auxiliary equations for each component of the system and to solve these equations through a computer program in MATLAB. The largest relative cost difference was found in the steam turbine followed by heat recovery steam generator, combustion chamber and compressor in the topping cycle. However, in the Kalina bottoming cycle, the highest relative cost difference was evaluated for the recuperators followed by ammonia–water evaporator. It was resulted that the highest exergy destruction, capital cost rate and the lowest exergoeconomic factor were found in the steam turbine. It was also claimed that the recuperator and evaporator of the Kalina cycle could be redesigned for improving the capital investment due to their low values of exergoeconomic factor.

Zare et al. [128] performed an exergoeconomic analysis for a combined cogeneration cycle using the waste heat from gas turbine-modular helium reactor (GT-MHR) which was recovered by the ammonia-water power/cooling system. An optimization was carried out for the cycles using the three objective functions which were energy utilization factor, exergy efficiency and sum of the unit cost of the products. It was resulted that thermodynamic and economic performances were improved when the ammonia-water cycle was adapted to the GT-MHR. It was concluded that the energy utilization factor and exergy efficiency were increased 11.1 % and 5.9 %, respectively and the sum of unit costs of the products was decreased by 5.4 % as a result of the optimization of the cycle.

Yari et al. [129] investigated the Kalina cycle performance in comparison with TLC (trilateral Rankine cycle) and ORC systems using the low-grade heat source with a

temperature of 120 °C. It was resulted that the net power output of TLC system could be found to be higher than that of the ORC and Kalina cycle. It was claimed that the advantage of the TLC system with regarding to the ORC or Kalina cycle was dependent on the value of expander isentropic efficiency.

Rodriguez et al. [130] investigated a Kalina cycle while the mass fraction of ammonia was at 84 % with an ORC system in order to use geothermal systems as heat source. It was resulted that the Kalina cycle produced 17.8 % more net power than that of the ORC systems for the considered conditions.

Bombarda et al. [131] performed a comparison study between Kalina cycle and ORC for heat recovery from the diesel engine. It was concluded that the net power output were almost the same for both Kalina cycle and ORC systems. It was also resulted that in order to obtain optimum thermodynamic performance, very high maximum pressure was required for the Kalina cycle.

A thermodynamic optimization model was carried out for comparing the merits and demerits of ORC and Kalina cycle for waste heat recovery from CGAM cogeneration system by Nemati et al. [132]. It was suggested that the optimum pressure value for the ORC was remarkably lower than that of the Kalina. In addition to this, it was also resulted that the ORC turbine outlet was superheated vapor while the Kalina cycle turbine outlet was two-phase flow. Lower turbine size parameter of Kalina cycle was proposed presenting a positive aspect when ORC systems were considered.

2.7 Optimization

Optimization of two or more number of objectives is possible with evolutionary multi-objective optimization (EMO) contemporaneously. The popularity of EMO has become at the end of 1990's. In this method, the algorithms of evolutionary optimization (EO) use a population based approach in which more than one solution participates in an iteration and evolves a new population of solutions in each iteration [133]. EO optimization is more rational when compared to single-objective

optimization in order to solve problems. For this reason, the solution of multi-objective optimizations are preferred in this study using non-dominated sorting genetic algorithms (NSGA-II) procedure which is one of the popular EMO procedure [133]. NSGA studies on the related research area in the literature are summarized as follows;

Anderson et al. [134] developed a level optimization tool for a waste incineration plant, which was utilized a multi-objective genetic algorithm (MOGA). A Pareto-optimal solutions were carried out using MOGA for minimizing emissions and keep within operational constraints.

Boyaghchi and Molaie [135] performed an advanced exergy analysis and optimization of a real combined cycle power plant (CCPP). NSGA-II method was used to carried out the final solutions in the multi-objective optimization of the CCPP. It was stated that the optimum value of CO₂ emission and total avoidable exergy destruction rate were improved 8.3 % and 10.6 %, respectively, with respect to base case. It was also resulted that the values of turbine inlet temperature and fuel mass flow rate were decreased with a percentage of 1.7 % and 32.5 %, respectively and also compressor pressure ratio was increased with a percentage of 38 % when compared to the base case.

Feng et al. [136] carried out another study on NSGA-II method for both regenerative ORC and basic ORC in order to optimize the cycles as thermodynamically and economically. It was resulted that the optimum exergy efficiency and levelized energy cost were evaluated as 55.97 % and 0.142 \$/kWh respectively, for the Pareto-optimal solution of the regenerative ORC, which were 8.1 % and 21.1 % higher than that of the basic ORC under the considered conditions.

Hajabdollahi and Fu [137] investigated the multi-objective based configuration optimization for a SOFC-GT cogeneration plant. Genetic Algorithm for multi-objective application was applied to evaluate the maximum of exergy efficiency and minimum of total cost rate, simultaneously. It was resulted that the rate of exergy destruction of each component in optimum design was obtained.

Jain et al. [138] presented a thermo-economic and environmental analyses based on multi-objective optimization of vapor compression–absorption cascaded refrigeration system using NSGA-II method. It was resulted that the multi-objective design was found to be better with respect to two individual single-objective for the optimization of thermodynamic and total product cost designs.

Konak et al. [139] presented a review describing genetic algorithms (GA) developed specifically for problems with multiple objectives considering different methods. It was resulted that all problems required some customization of the GA approaches to properly handle the objectives, constraints, encodings and scale.

Li et al. [140] analyzed a solar-dish Brayton system thermo-economically and optimized the system using the NSGA II method. A triple-objective optimization was performed for the system and the optimum values of the power output, efficiency and dimensionless thermo-economic performance of solar-dish Brayton system which was conducted in comparison with Brayton heat engine.

Yang et al. [141] performed a thermoeconomic multi-objective optimization of an organic Rankine cycle using the exhaust gas of a diesel engine as heat source for various working fluids. The optimization was carried out maximizing net power output and minimizing the total investment cost of ORC. The most suitable working fluid was found as R245fa regarding to optimum evaporation pressure range and condensation temperature as 1.1-2.1 MPa and 25°C, respectively.

2.8 Conclusions

The literature survey is outlined in this chapter which indicates that there are many studies on municipal solid waste management, waste disposal methods, thermodynamic and thermoeconomic analysis and optimization of power plants. The large part of the studies consider solid waste management policies of countries and also solid waste disposal techniques and its improvements. In several studies, different types of power plants were analyzed in terms of thermodynamic and thermoeconomic aspects. In this chapter, some recent studies based on gas turbine, organic Rankine and Kalina cycles are also outlined. Additionally, optimization is

defined and then the method of non-dominated sorting genetic algorithm is summarized in the literature.

This thesis differs from the previously conducted studies as follows:

- a) The presented study is based on the thermodynamic and thermoeconomic performance analyses and optimization of a gas engine powered municipal solid waste power plant using actual operating power plant data in Gaziantep. The thesis is original in this scope and there is no such a study in the open literature, to the best of the author's knowledge and it is the main motivation behind this study.
- b) In literature, only a few number of studies consider MSW power plant systems. The analysis in these studies are generally limited to actual system energy analysis without thermoeconomic considerations. In this regard, the thermodynamic and thermoeconomic analysis of the system can be considered as originality of thesis.
- c) Furthermore, five different models have been designed and their thermodynamic and thermoeconomic analyses have been carried out in this thesis, which can be applied to the actual system for improvement the overall efficiency of the system.
- d) Moreover, all developed models have been considered in order to perform parametric studies and then they have been optimized by using the most preferred optimization method which is non-dominated sorting genetic algorithm.
- e) The thesis proposes theoretical foundation which includes procedure and formulation for conventional energy and economic analyses, exergoeconomic analysis and optimization of Gaziantep Municipal Solid Waste Power Plant.

CHAPTER 3

THERMODYNAMIC & THERMOECONOMIC ANALYSES AND OPTIMIZATION

3.1 Introduction

Thermodynamic and thermoeconomic analyses of all systems are performed according to the thermophysical and the chemical properties of each streams through the selected system. Thus, in order to evaluate efficiencies, exergy destructions and the costs of the component, mass, energy and exergy balances of each component are needed. The following sections provide the fundamental information on thermodynamic and thermoeconomic analyses. Then, the information about multi-objective optimization is presented. In the last section, the thermodynamic and thermoeconomic analyses of the Gaziantep Municipal Solid Waste Power Plant (GMSWPP) located in Gaziantep is carried out by using actual operating data.

3.2 Thermodynamic Analysis

The physical significance of the property E is defined as total energy of any system and it is an extensive property represents all energy of a system in a given state. This energy might be present in a variety of forms, such as the kinetic and potential energy of a system as a whole with respect to a chosen coordinate frame, energy associated with motion and position of molecules, energy associated with structure of atom, chemical energy present in a storage battery, energy present in a charged capacitor, or any of a number of other words. Shortly, energy of a system can be shown with the summation of internal, kinetic and potential energies and expressed as:

$$E = U + KE + PE \quad (3.1)$$

For a system involving a simple compressible substance, i.e., in the absence of electric, magnetic, and surface tension effects, etc. the first law of thermodynamics for a control mass can be written as [142]:

$$Q - W = \Delta U + \Delta KE + \Delta PE \quad (3.2)$$

where internal, kinetic and potential energy changes are:

$$\Delta U = m(u_2 - u_1) \quad (3.3)$$

$$\Delta KE = \frac{1}{2}m(V_2^2 - V_1^2) \quad (3.4)$$

$$\Delta PE = mg(z_2 - z_1) \quad (3.5)$$

where the changes of kinetic and potential energies are zero for stationary systems (that is $(\Delta KE = \Delta PE = 0)$). Therefore, the total energy change can simply be taken as $\Delta E = \Delta U$ for such systems.

The first law of thermodynamics stresses the quantity of energy and claims that energy cannot be created or destroyed. On the other hand, the second law of thermodynamics deals with the quality of energy. Throughout this thesis, a control volume in which a steady flow process is taking place chosen as a model for the analyses. Mass, energy and exergy balances for a control volume under steady state conditions, with negligible kinetic and potential energy changes can be expressed, respectively, as [75]:

$$\Sigma \dot{m}_i = \Sigma \dot{m}_e \quad (3.6)$$

$$\dot{Q} - \dot{W} = \Sigma \dot{m}_e h_e - \Sigma \dot{m}_i h_i \quad (3.7)$$

$$\dot{E}x_{Heat} - \dot{W} = \Sigma \dot{m}_e \psi_e - \Sigma \dot{m}_i \psi_i + \dot{E}x_D \quad (3.8)$$

where the subscripts e and i represent the exit and inlet states, $\dot{Q}$ and $\dot{W}$ are the net heat and work transfers, $\dot{m}$ is the mass flow rate through the system boundary, h is

the specific enthalpy, $\dot{E}x_D$ is the exergy destruction and $\dot{E}x_{Heat}$ is the net exergy transfer by heat transfer at temperature T , which are given by:

$$\dot{E}x_D = T_0 \dot{S}_{gen} \quad (3.9)$$

$$\dot{E}x_{Heat} = \Sigma \left(1 - \frac{T_0}{T} \right) \dot{Q} \quad (3.10)$$

where the subscript 0 represents the reference state, taken most of the times as the environmental condition. Exergy can be defined as the energy that is available to use, or in other words, work potential that can be utilized. Exergy is the summation of kinetic, potential and chemical exergies. The definition of the exergy and specific flow exergy is shown in following equation:

$$\dot{E}x = \dot{E}x_{PH} + \dot{E}x_{KN} + \dot{E}x_{PT} + \dot{E}x_{CH} \quad (3.11)$$

$$\psi = \psi_{PH} + \psi_{KN} + \psi_{PT} + \psi_{CH} \quad (3.12)$$

where the superscripts PH, KN, PT and CH indicate physical, kinetic, potential and chemical exergies. Kinetic and potential exergies are equal to kinetic and potential energies, respectively. For a stationary system, the kinetic and the potential energies are equal to zero. The definitions of the subcomponent of exergy are given as follows:

$$\dot{E}x_{PH} = \dot{m} \psi_{PH} \quad (3.13)$$

$$\dot{E}x_{KN} = \frac{1}{2} \dot{m} (V_2^2 - V_1^2) \quad (3.14)$$

$$\dot{E}x_{PT} = \dot{m} g (z_2 - z_1) \quad (3.15)$$

$$\dot{E}x_{CH} = \dot{m} \psi_{CH} \quad (3.16)$$

The physical and chemical exergies can be determined from the flow exergy relation as:

$$\psi_{PH} = (h - h_0) - T_0 (s - s_0) \quad (3.17)$$

$$\psi_{CH} = \sum_i y_i \psi_i^{CH} + \bar{R} T_0 \sum_i y_i \ln y_i \quad (3.18)$$

where s , y_i and $\bar{R}$ denote the entropy, mole fraction and universal gas constant, respectively.

Efficiency of a system is defined as the desired output divided by the required input for a system to produce that desired output. The definition of energy efficiency is related to the first law of thermodynamics. Thus, the efficiency is [143]:

$$\eta = \left(\frac{\text{energy in products}}{\text{total energy input}} \right) \quad (3.19)$$

The expression of exergy efficiency is related to the second law of thermodynamics. Sometimes it is called as the second law efficiency. Exergy efficiency can take different forms depending on to the type of a system under consideration. But, generally it is defined as [143]:

$$\varepsilon = \left(\frac{\text{exergy in products}}{\text{total exergy input}} \right) \quad (3.20)$$

In addition to this, efficiencies of sub-components of thermodynamic systems are also important. Isentropic efficiencies of turbine and compressor/pump can be defined as:

$$\eta_t = \frac{w_a}{w_s} = \frac{h_i - h_e}{h_i - h_{es}} \quad (3.21)$$

$$\eta_{comp/pump} = \frac{w_s}{w_a} = \frac{h_{es} - h_i}{h_e - h_i} \quad (3.22)$$

where w_a is the actual specific work and w_s is the isentropic specific work through turbine, the subscript es denotes isentropic condition at exit state. The exergy (second law) efficiencies of turbine and compressor/pump are given as:

$$\varepsilon_t = \frac{w_a}{w_{rev}} = \frac{h_i - h_e}{h_i - h_e - T_0(s_i - s_e)} \quad (3.23)$$

$$\mathcal{E}_{comp/pump} = \frac{w_{rev}}{w_a} = \frac{h_e - h_i - T_0(s_e - s_i)}{h_e - h_i} \quad (3.24)$$

where w_{rev} , reversible specific work is equal to the sum of specific exergy destruction and actual work. The exergy efficiency of a heat exchanger in a power plant is calculated by the increase in the exergy of the cold stream divided by the decrease in the exergy of the hot stream as follows:

$$\mathcal{E}_{he} = \frac{(\dot{E}x_e - \dot{E}x_i)_{cold}}{(\dot{E}x_i - \dot{E}x_e)_{hot}} = \frac{\dot{m}_{cold}[h_e - h_i - T_0(s_e - s_i)]_{cold}}{\dot{m}_{hot}[h_i - h_e - T_0(s_i - s_e)]_{hot}} \quad (3.25)$$

where $\dot{m}_{cold}$ and $\dot{m}_{hot}$ are the mass flow rates of the cold and hot streams, respectively.

In order to analyze a typical heat, the effective heat transfer area of a heat exchanger should be well defined. This can be achieved by using the logarithmic mean temperature difference (*LMTD*) method which is expressed as follows:

$$LMTD = \frac{(T_{H,i} - T_{C,e}) - (T_{H,e} - T_{C,i})}{\ln\left(\frac{T_{H,i} - T_{C,e}}{T_{H,e} - T_{C,i}}\right)} \quad (3.26)$$

where the subscripts H , C , i and e denote hot, cold, inlet and exit, respectively. Then the area of heat exchanger becomes a function of *LMTD* and it is determined as follows

$$A_k = \frac{\dot{Q}_k}{U_k LMTD} \quad (3.27)$$

where U_k and $\dot{Q}_k$ represent to heat transfer coefficient and the amount of heat transfer, respectively.

3.3 Thermoeconomic Analysis

Thermoeconomics of a power plant is associated with second law of thermodynamics which is a favorable and certain way to consider costs related to thermodynamics of this power plant. Cost analysis may not be always an adequate and good option to show the economy of a power plant. For this reason, Tsatsaronis and Moran proposed a new methodology in 1997 which is called exergoeconomy [144]. In this thesis, specific exergy costing (SPECO) method is used for the thermoeconomic analysis of the power plant system. In SPECO method, firstly, all energetic and exergetic flows in all states of system are determined. Then, all sub-components of the system are categorized with respect to fuel and product approach. All exergy additions and removals to a sub-component are considered as the fuel (F) and product (P), respectively. As a result, the cost balance and auxiliary equations for each component are written as [75]:

$$\sum_e \dot{C}_{e,k} + \dot{C}_{w,k} = \dot{C}_{q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k \quad (3.28)$$

$$\dot{C}_j = c_j \dot{E}_j \quad (3.29)$$

where $\dot{C}$ represents the cost rate (\$/h) and e and i represent inlet and exit flow rate of a sub-component k . $\dot{Z}_k$ represents the entire cost rate related to capital investment (CI) and operation and maintenance costs (OM) of the sub-component k . The capital cost rate of components is calculated as:

$$\dot{Z} = (PEC * CRF * \varphi) / (3600 * N) \quad (3.30)$$

where PEC , CRF , φ and N represent the purchased equipment cost, capital recovery factor, maintenance factor (~ 1.06) and annual operation hours, respectively. Herein, PEC values can be computed by using various correlations or actual economic data. PEC correlations usually associated with chemical engineering plant cost index (CEPCI) which should be considered. CEPCI is a vital parameter to consider the total capital cost of a plant which is calculated using the ratio between the cost index of reference year and original year [145]. The correlations suggested

by Turton et al. [146] are used which are then updated to 2014 values using the following definition:

$$C_1 = C_2 \frac{(CEPCI_1)}{(CEPCI_2)} \quad (3.31)$$

In order to perform a reliable thermoeconomic analysis, the plant cost index was considered between 2011 and 2016, which are 585.7 and 556.8, respectively. The reference year was selected to be 2011 in which year the power plant was started up to produce electricity. Then, the value of CEPCI was evaluated as 1.052 and the components were updated with respect to this value [147]. The CRF can be calculated depending on the interest rate (i) and total life time (n) of the system. It can be expressed as:

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (3.32)$$

and the general cost rate balance is

$$\dot{C}_p = \dot{C}_f + \dot{Z} \quad (3.33)$$

where the subscripts p and f represent the product and fuel, respectively.

The performance of a sub-component can be assessed by using SPECO method and for this reason, the cost flow rates through any sub-components related with the exergy loss are evaluated by means of the cost history of the power plant. This is supplied by using the exergoeconomic factor f_k , and it is defined for a sub-component k as [75]:

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{f,k} \dot{E}x_{D,k}} \quad (3.34)$$

where $c_{f,k}$ is the unit exergetic cost of the fuel of any sub-component k and $\dot{E}x_{D,k}$ is the corresponding exergy destruction of the same component. Relative cost difference, r_k is another important parameter in thermoeconomic evaluations. It is

the relative increase in the average cost per exergy unit between fuel and product of a component. For a sub-component k, it is defined as:

$$r_k = \frac{c_{p,k} - c_{f,k}}{c_{f,k}} \quad (3.35)$$

where $c_{p,k}$ is the unit exergetic cost of a product of any sub-component k. The relative cost difference is very significant parameter for considering and optimizing system components. The cost rate of exergy destruction, $\dot{D}_{D,k}$ is defined as:

$$\dot{D}_{D,k} = c_{f,k} \dot{E}x_{D,k} \quad (3.36)$$

3.4 Optimization

Optimization having two or more number of objectives is possible with evolutionary multi-objective optimization (EMO) simultaneously. EMO is more rational when compared to single-objective optimization [133]. Due to this reason, the multi-objective optimization method was performed in this thesis by using a popular EMO procedure, namely, the non-dominated sorting genetic algorithms (NSGA-II) procedure in MATLAB environment. Multi-objective optimization formulation is outlined by Konak et al. [139]. In order to optimize a system with multiple (K) objectives, minimization of objectives should be defined. For an n -dimensional decision variable, in a solution space of x :

$$x = \{x_1, x_2, x_3, \dots, x_n\} \quad (3.37)$$

It is necessary to determine an x^* vector to minimize:

$$z(x^*) = \{z_1(x^*), z_2(x^*), \dots, z_K(x^*)\} \quad (3.38)$$

The solution space x is generally restricted by a series of constraints, such as $g_j(x^*) = b_j$ for $j = 1, \dots, m$ and bounds on the decision variables.

In order to perform an optimization, a decision has to be made for both minimization and maximization. Herein, the critical point is that the selection should be at the

same direction for all objective functions. Due to this reason, if a minimization study is going to be carried out and one of the objectives is to be maximized, this objective functions can be taken as negative (-1) one. Among all feasible solutions, the most dominant and preferable solution is accepted as the Pareto optimal solution. For a determined Pareto optimal set, all corresponding objective function values are called as the Pareto front.

There are too many optimization methods in the open literature, which can be adapted to figure out multi-objective optimization problems. Genetic Algorithms (GA) are one of the most promising optimization methods [148]. In the GA, the solution vector (x) is defined as a chromosome and collection of chromosomes form a population. Generation of new solutions is accomplished by crossover and mutation, where crossover is forming of new solutions from two chromosomes called 'parents'. It is expected that the offspring inherits good genes from the parents to optimize the population. Mutation is generally applied at gene level, which helps the population remain in the local optima by setting a mutation rate between zero and one, so that the offspring will not be very different than their parents. The GA optimization can be performed with the following steps which are given in Table 3.1.

Table 3.1. The GA optimization steps [148]

1- Initiate a population	4- Mutation
2- Select random parents from the population	5- Reproduction of new population from the children population
3- Crossover	6- Stop evolution

3.5 Thermodynamic and Thermoeconomic Analyses of Gaziantep Municipal Solid Waste Power Plant

3.5.1 System Description

Gaziantep is located in the southeast region of Turkey, which is the 6th biggest city population wise (2000000). Total land area of Gaziantep is 7642 km². With the total consumption, Gaziantep is on the 11th rank in Turkey [30]. Annual electricity

consumption per capita is 2560 kWh in Gaziantep. The reason of this high electricity consumption is clearly due to dense population and industrial facilities in the city. Over the last decades, industry in Gaziantep has developed very rapidly, especially in the areas of plastic, textile, food, carpet and packaging industries. In Gaziantep, there are some private companies which produce their own electricity using natural gas, fuel oil, LFG and also hydro-power. The installed power capacities of these companies are given in Fig. 3.1 [30].

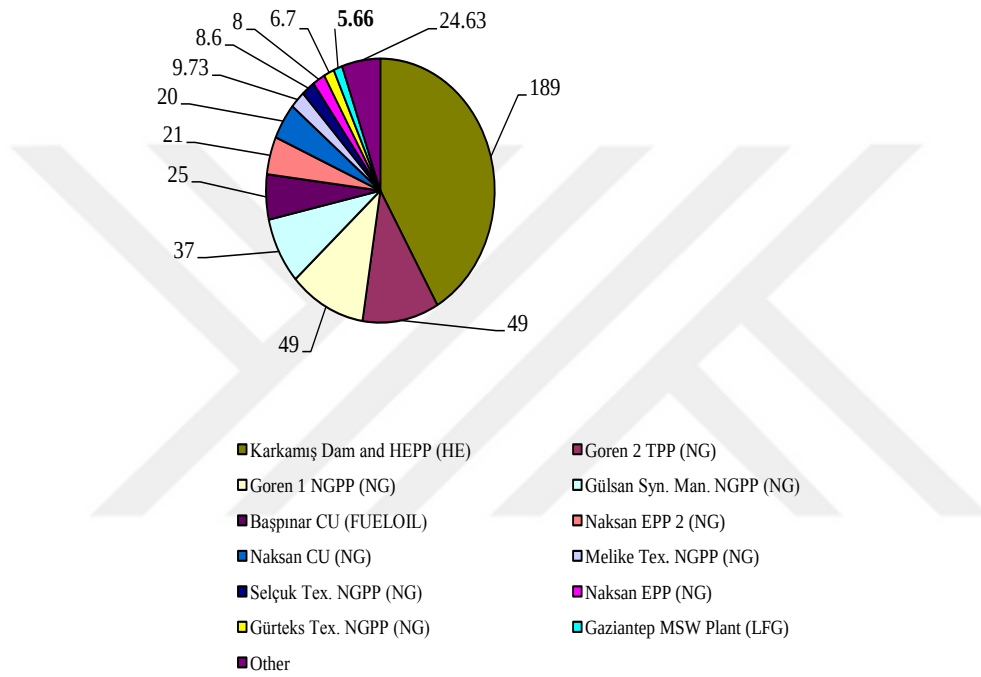


Figure 3.1. Distribution of energy production in MW in Gaziantep

In this total, LFG produced from municipal solid waste (MSW) occupies a major part in for energy production of Gaziantep, in terms of waste to energy concept. Historically, when MSW management as Gaziantep is considered, MSW was collected by Gaziantep Metropolitan Municipality in an unsanitary landfill until 1996, for only disposal purpose at Beylerbeyi location which was too close to the city center. After 1993, Gaziantep Metropolitan Municipality has started to construct

a sanitary landfill plant with an installed power capacity of 5.66 MW for both disposal of MSW and energy production from produced LFG in Mazmahor, Uzundere under the grant of Southeastern Anatolia Project (GAP). This constructed sanitary landfill has 32.3 ha solid waste storage area and also 10000000 m³ solid waste capacity which will fulfill the need until 2046 [30]. This plant produces 1.25% of total power demand of Gaziantep. Average percentage compositions of MSW in Gaziantep and its comparison with other metropolitan cities is given in Table 3.2 [30].

Table 3.2. Compositions of MSW in six metropolitan cities in Turkey [30]

Cities	(%) Compositions					
	Organic	Paper/ Cardboard	Plastic	Metal	Glass	Others
Istanbul	43	7.8	14.2	5.8	6.2	23.1
Izmir	46	12	12	3	4	23
Bursa	53.1	18.4	11.6	3	3.4	10.5
Gaziantep	41.6	6.1	4.8	1	7.1	39.5
Adana	64.4	14.8	5.9	1.4	3.1	11.4
Mersin	63	18.4	6.7	1.3	3.1	7.6

Figure 3.2 is the schematic layout of Gaziantep Municipal Solid Waste Power Plant (GMSWPP). In GMSWPP, LFG is obtained from the anaerobic decomposition of organic substances in domestic solid waste, industrial, and medical wastes. The total MSW carried to GMSWPP is 1500 tons daily. All wastes collected in GMSWPP are subjected to mechanical segregation of plastic, metal and glass, and then rest of MSW is sent to sanitary landfilling area. On the other hand, medical waste is sterilized first by a pretreatment and then sent into landfilling area. MSWs which are buried underground in landfilling area are led to produce LFG for months. The produced LFG from the storage area is collected from 114 high dense polyethylene (HDPE) funnels with a diameter of 800 mm and depths changing between 8 – 41 meters. The collected LFG is then transferred to 6 manifold stations. If the temperature of the LFG is higher than 40-45 °C, it is cooled through the heat

exchangers by means of chilled water which is denoted by blue line in Fig. 3.3. The LFG which is under 40-45 °C is sucked into tandem demisters/filters eliminate moisture and particulates such as aluminum, ash, etc. by using blowers. There is one flare stack mounted at the exit of blowers in order to drain excess LFG in an emergency case. The collected water is sent back to drain tank. The filtered LFG sucked through four pipelines is collected in the A/C (active carbon) tanks for desulphurization process. At the exit of the A/C tanks, if the operating pressure exceeds 14.5 kPa gage, excess LFG is sent back to the beginning of the process pipeline by a pressure control valve. At the same time, the volumetric content of LFG, are measured and also recorded on the control panel. At any instant, the average volumetric values of the components of LFG are given in Table 3.3. The percentage of methane in LFG according to the monthly distribution in 2014 was also given in Fig. 3.3. As it can be seen from the figure, the average volumetric percentage of methane is nearly 45. Moreover, the operation is stopped automatically when methane content decreases below 38% or oxygen is increased above 6%. The flow rate, the operating pressure and the temperature data throughout the process are measured at 7, 15 and 5 stations, respectively. The volumetric flow rate of LFG is measured 17.5 m³/min, through the main pipeline. After the reduction of H₂S content of the gas to 100-120 ppm, it is then pumped into three 1.13 MW Janbacher-4 type gas engines coupled with generators to produce electricity. At the end of the process the exhaust gas temperature reaches 550-600 °C degrees. Due to the increasing amount of MSW in Gaziantep, two 1.131 MW Janbacher-4 type gas engines were constructed next to the existing.

Table 3.3. Average Volumetric Composition of LFG Produced in Gaziantep Municipal Solid Waste Power Plant

Components	Chemical Formula	(%) Dry Volume
Methane	CH ₄	45-53
Carbon dioxide	CO ₂	30-32
Carbon monoxide	CO	1-5
Nitrogen	N ₂	2-6
Oxygen	O ₂	3-5
Hydrogen	H ₂	3-7
Hydrogen Sulfur	H ₂ S	0-2

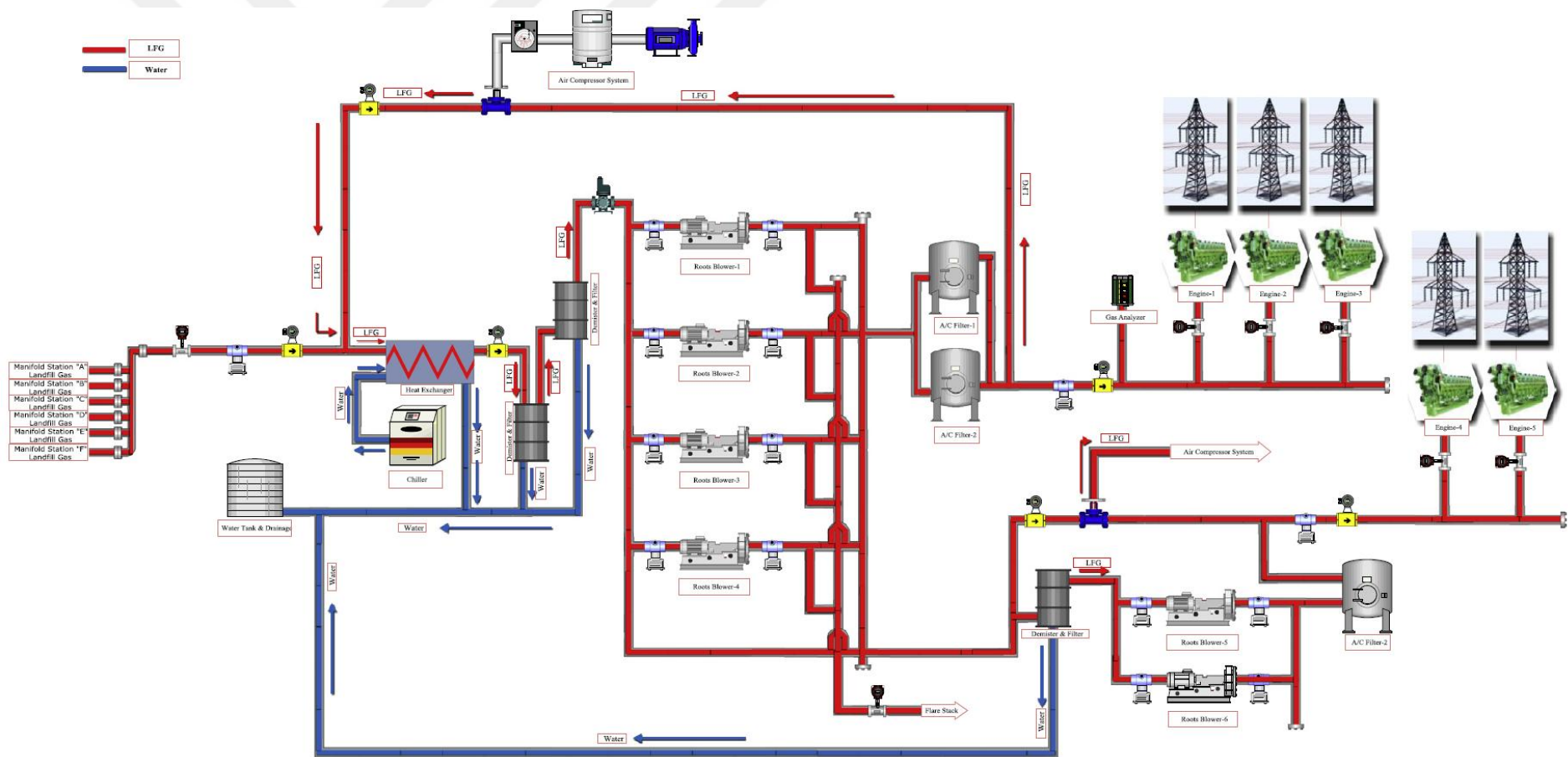


Figure 3.2. Schematic layout of the Gaziantep Municipal Solid Waste Power Plant

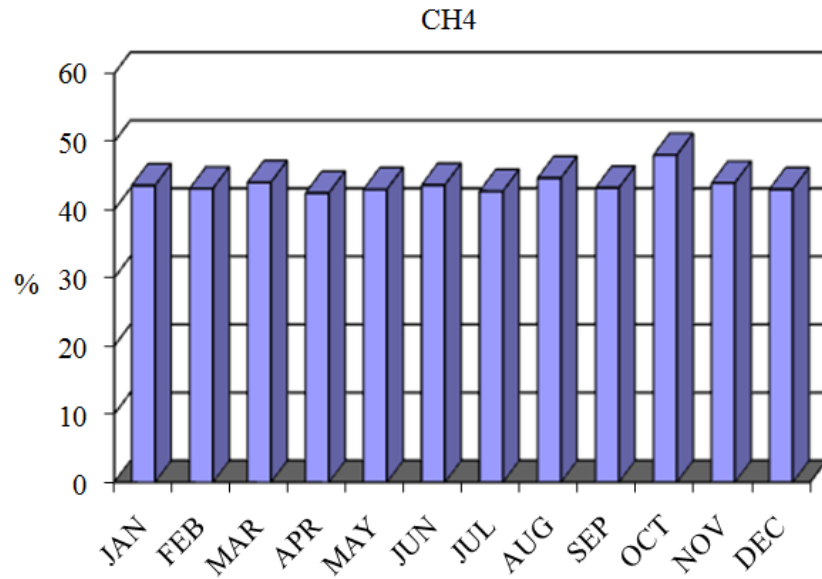


Figure 3.3. Monthly methane content in LFG produced in Gaziantep Municipal Solid Waste Power Plant

3.5.2 Performance Analysis of Gas Engine

Jenbacher 416 GS type gas engine which has 145 mm cylinder diameter, 185 mm piston stroke and 1500 rpm engine speed is used in Gaziantep Municipal Solid Waste Power Plant for electricity production. The photographic view of Jenbacher 416 GS gas engine is given in Fig. 3.4.



Figure 3.4. Photographic view of Jenbacher 416 GS gas engine

The relations used for the analysis of engine performance are as follows: The crank offset and average piston speed are calculated by:

$$a = S / 2 \quad (3.39)$$

$$U_p = 2SN \quad (3.40)$$

where a , S , U_p and N represent crank offset, stroke, piston mean speed and engine rotational speed, respectively. Average piston speed will normally be in the range of 5 - 15 m/s for large diesel engines and also for high-performance automobile engines [149]. Displacement volume (V_d) is calculated by taking the difference between bottom dead center (BDC) and top dead center (TDC):

$$V_d = V_{BDC} - V_{TDC} \quad (3.41)$$

Its value for any number of cylinders can be calculated as:

$$V_d = (\pi B^2 S / 4) N_c \quad (3.42)$$

where B and N_c represent cylinder bore and number of cylinders, respectively. Minimum cylinder volume occurs when the piston is at TDC and it is called clearance volume. Clearance volume (V_c) and compression ratio (r_c) are defined as follows: is expressed as:

$$V_c = V_{TDC} = V_d - V_{BDC} \quad (3.43)$$

$$r_c = (V_d + V_c) / V_c \quad (3.44)$$

Brake power or net power output ($\dot{W}_b$) is directly taken from the technical specification catalogue of engine and it is used in the calculations of brake mean effective pressure and brake power for one cylinder:

$$bmep = 2\dot{W}_b / V_d N \quad (3.45)$$

Other characteristic parameters are specific power (S_p), output per displacement (OPD), specific volume (S_v) and specific weight (S_w) for an engine as follows:

$$S_p = \dot{W}_b / A_p \quad (3.46)$$

$$OPD = \dot{W}_{b,cyl} / V_d \quad (3.47)$$

$$S_v = V_d / \dot{W}_{b,cyl} \quad (3.48)$$

$$S_w = m_{engine} / \dot{W}_b \quad (3.49)$$

A_p , m_{engine} and $\dot{W}_{b,cyl}$ represent the piston cross section area, weight of engine and brake power per cylinder, respectively. These parameters are taken from the catalogue. The specific fuel consumption (*bsfc*) for brake power is given below:

$$bsfc = \dot{m}_{fuel} / \dot{W}_b \quad (3.50)$$

The combustion efficiency (η_{cbs}) is defined as the fraction of fuel which is burnt and it is in the range of 0.95 and 0.98.

$$\eta_{cbs} = \dot{Q}_{in} / q_{LHV} \dot{m}_{fuel} \quad (3.51)$$

where $\dot{Q}_{in}$ is the rate of net heat input to engine and q_{LHV} is lower heating value of fuel which is taken average value of the range given in the technical specification catalogue. Also, thermal efficiency of engine is defined as:

$$\eta_{th} = \dot{W}_b / q_{LHV} \dot{m}_{fuel} \eta_{cbs} \quad (3.52)$$

Jenbacher 416 gas engine performance characteristics are given in Table 3.4.

Table 3.4. Jenbacher 416 GS gas engine performance characteristics

Cylinder Diameter, D	145	mm
Cylinder Bore, B	145	mm
Stroke, S	185	mm
Crank Offset, a	92.5	mm
Number of Cylinders, N_c	16	-
Piston Cross Sectional Area, A_p	0.0165	m ²
Compression Ratio, r_c	13.5	-
Displacement Volume, V_d	0.00305	m ³
Clearance Volume, V_c	0.000244	m ³
Cylinder Volume, V_d	0.003299	m ³
Air Flow Rate, $\dot{m}_a$	3.048	kg/s
Fuel Flow Rate, $\dot{m}_{fuel}$	0.152	kg/s
Air-Fuel Ratio, AF	20.05	-
Lubrication Oil Flow Rate	148.0	kg/s
Exhaust Flow Rate	3.2	kg/s
Piston Mean Speed, U_p	9.25	m/s
Engine Speed, N	1500	rpm
Heating Value, q_{LHV}	14,000/20000	kJ/kg
Brake Power, $\dot{W}_b$	1131	kW
Brake Power per Cylinder, $\dot{W}_{b,cyl}$	70.68	kW
Brake Mean Effective Pressure, $b MEP$	1851	kPa
Torque τ	7200	Nm
Brake Specific Fuel Consumption, $bsfc$	483.81	gr/kWh
Specific Power, S_p	4283.63	kW/m ²
Specific Volume, S_v	0.043	L/kW
Specific Weight, S_w	4.77	ton/MW
Output per Displacement, OPD	23.17	kW/L
Combustion Efficiency, η_c	0.98	%
Volumetric Efficiency, η_v	0.95	%
Thermal Efficiency, η_t	43.3	%

3.5.3 Thermodynamic Analysis of Gaziantep Municipal Solid Waste Power Plant

In Gaziantep Municipal Solid Waste Power Plant (GMSWPP), five identical Jenbacher 416 type gas engines are used in order to produce electrical power. Thermodynamic analysis is performed for one engine and then applied for whole power plant. The engine room of the GMSWPP is modeled by using ASPEN Plus Engineering software and analyzed in Engineering Equation Solver (EES) software environment by making use of the actual power plant data. The engine room consists of an air-fuel tank (AFT), a compressor (COMP), an intercooler (INT), a lubrication oil cooler (LOC), a gas engine (GE), an air-water radiator (AWR) and a turbine (TUR). Herein the turbine and compressor are used as a part of the turbocharger unit. The exhaust gases as well as the air are assumed to be handled with sufficient accuracy by the ideal gas model at all states considered in the analyses. The schematic representation of the engine room is given in Fig. 3.5. Thermodynamic properties of the substances at typical states visited by the actual system are given in tabular form, see Table 3.5.

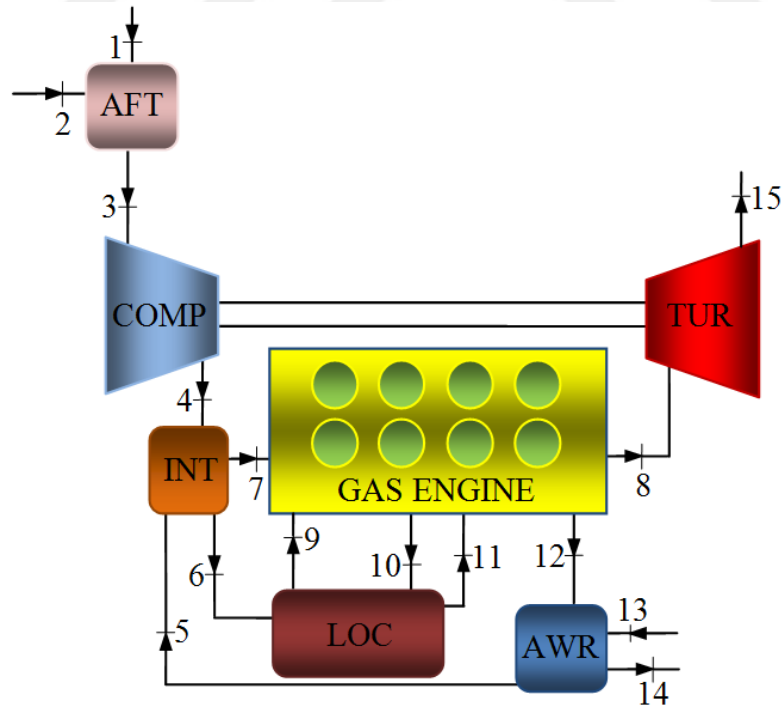


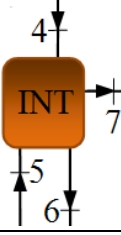
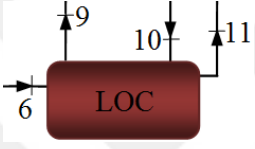
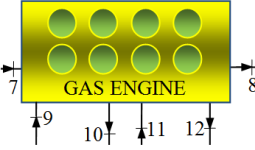
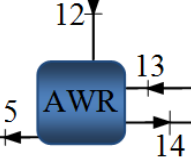
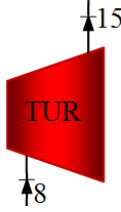
Figure 3.5. Schematic layout of the gas engine in Gaziantep Municipal Solid Waste Power Plant

Table 3.5. Thermodynamic properties of the states of the power plant

State	Fluid	T (°C)	P (bar)	$\dot{m}$ (kg/s)	h (kJ/kg)	s (kJ/kg.K)	ψ (kJ/kg)	$\dot{E}_x$ (kW)
0	LFG	20	1	-	-4895	8.473	-	-
0'	LFG+Air	20	1	-	50.36	5.813	-	-
0''	Water	20	1	-	83.93	0.2962	-	-
0'''	Air	20	1	-	293.6	5.682	-	-
1	Air	20	1.08	3.05	293.6	5.66	6.476	19.75
2	LFG	20	1.08	0.15	-4895	8.445	26203	3930
3	LFG+Air	20	1.08	3.2	50.36	5.791	39.24	125.6
4	LFG+Air	102	1.98	3.2	135.3	5.87	101	323.1
5	Water	40	1.35	7.33	167.6	0.5722	2.765	20.27
6	Water	46.12	1.35	7.33	193.2	0.6532	4.631	33.94
7	LFG+Air	42.2	1.98	3.2	37.24	5.635	71.89	230
8	Exhaust	674.3	4.24	3.2	986.6	6.495	454.9	1456
9	Luboil	79.91	3.33	148.1	200.6	0.672	11.06	1638
10	Luboil	84.1	3.33	148.1	209.5	0.6969	12.61	1867
11	Water	82.57	1.35	7.33	345.8	1.106	24.55	180
12	Water	88.43	1.35	7.33	370.4	1.174	29.05	212.9
13	Air	20	1	21.83	293.6	5.682	0	0
14	Air	80.58	1	21.83	354.5	5.872	5.514	120.4
15	Exhaust	566.6	1.9	3.2	866.1	6.59	306.4	980.3

All heat transfer, work and exergy destructions rates together with exergy efficiencies of all sub-components were calculated by means of the governing equations given in Table 3.6.

Table 3.6. Energy and exergy equations of the subcomponents of the power plant

COMP		$\dot{m}_3 = \dot{m}_4$ $\dot{W}_{Comp,a} = \dot{m}_3(h_4 - h_3)$ $\dot{W}_{Comp,s} = \dot{m}_3(\psi_4 - \psi_3)$ $\dot{E}x_{D,Comp} = \dot{W}_{Comp,a} - \dot{W}_{Comp,s}, \epsilon_{Comp} = \frac{\dot{W}_{Comp,s}}{\dot{W}_{Comp,a}}$
INT		$\dot{m}_4 = \dot{m}_7, \dot{m}_5 = \dot{m}_6$ $\dot{m}_4(h_4 - h_7) = \dot{m}_5(h_6 - h_5)$ $\dot{E}x_{D,INT} = \dot{m}_4(\psi_4 - \psi_7) - \dot{m}_5(\psi_6 - \psi_5)$ $\epsilon_{INT} = \frac{\dot{m}_4(\psi_4 - \psi_7)}{\dot{m}_5(\psi_6 - \psi_5)}$
LOC		$\dot{m}_6 = \dot{m}_{11}, \dot{m}_9 = \dot{m}_{10}$ $\dot{m}_9(h_{10} - h_9) = \dot{m}_6(h_{11} - h_6)$ $\dot{E}x_{D,LOC} = \dot{m}_9(\psi_{10} - \psi_9) - \dot{m}_6(\psi_{11} - \psi_6)$ $\epsilon_{LOC} = \frac{\dot{m}_9(\psi_{10} - \psi_9)}{\dot{m}_6(\psi_{11} - \psi_6)}$
GE		$\dot{m}_7 = \dot{m}_8$ $\dot{E}x_{F,GE} = 1131kW, \dot{E}x_{P,GE} = \dot{E}x_8 - \dot{E}x_7$ $\dot{E}x_{D,GE} = \dot{E}x_{F,GE} - \dot{E}x_{P,GE}$ $\epsilon_{GE} = \frac{\dot{E}x_{F,GE}}{\dot{E}x_{P,GE}}$
AWR		$\dot{m}_5 = \dot{m}_{12}, \dot{m}_{13} = \dot{m}_{14}$ $\dot{m}_5(h_{12} - h_5) = \dot{m}_{13}(h_{14} - h_{13})$ $\dot{E}x_{D,AWR} = \dot{m}_5(\psi_{12} - \psi_5) - \dot{m}_{13}(\psi_{14} - \psi_{13})$ $\epsilon_{LTR} = \frac{\dot{m}_5(\psi_{12} - \psi_5)}{\dot{m}_{13}(\psi_{14} - \psi_{13})}$
TUR		$\dot{m}_8 = \dot{m}_{15}$ $\dot{W}_{TUR,a} = \dot{m}_8(h_8 - h_{15})$ $\dot{W}_{TUR,s} = \dot{m}_8(\psi_8 - \psi_{15})$ $\dot{E}x_{D,TUR} = \dot{W}_{TUR,s} - \dot{W}_{TUR,a}, \epsilon_{TUR} = \frac{\dot{W}_{TUR,a}}{\dot{W}_{TUR,s}}$

Summary of the results obtained from the for all subcomponents are given in Table 3.7. The thermodynamic analyses of have shown that the exergy efficiency of the Gaziantep Municipal Solid Waste Power Plant is 47.84%. Beside this, the thermal

efficiency of the gas engine is found to be 38.87%, which is compatible with the technical specifications of the Jenbacher 416 type gas engine. The exergy efficiencies of the compressor and the turbine of the turbocharger are 72.68% and 81.16%, respectively. This represents that a remarkable exergy losses exist at the turbocharger unit. The highest and lowest exergy efficiencies are found at the gas engine and intercooler, respectively. The exergy efficiencies of the INT, LOC and AWR units are reckoned to be 14.7 %, 63.97 % and 62.49 %, respectively. It is clearly seen that heat exchangers have lowest exergy efficiencies when compared with the other components.

Table 3.7. Energy and exergy analyses of the sub-components

Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}_F$ (kW)	$\dot{E}_P$ (kW)	$\dot{E}_D$ (kW)	ε (%)
COMP	-	271.7	271.7	197.5	74.23	72.68
INT	313.7	-	93.02	13.68	79.35	14.70
LOC	1309	-	228.3	146	82.25	63.97
GE	-	1131	1225	1131	94.48	92.29
AWR	1486	-	192.6	120.4	72.26	62.49
TUR	-	385.7	475.2	385.7	89.51	81.16
Energy efficiency of the plant					38.87	
Exergy efficiency of the plant					47.84	

3.5.4 Thermoeconomic Analysis of Gaziantep Municipal Solid Waste Power Plant

In order to be able to do a thoroughly complete thermoeconomic analysis, it was necessary to have the operational and economical data belonging to LFG system and all sub-components of the plant which are provided by the Gaziantep Municipal Solid Waste Power Plant. Exergy cost rate balance and corresponding auxiliary equations belonging to each sub-component of the gas engine are tabulated Table 3.8. Auxiliary equations are carried out using fuel and product principles for sub-components of the plant.

Table 3.8. Exergy cost rate balance and corresponding auxiliary equations for all sub-components of the gas engine

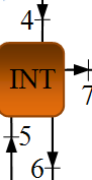
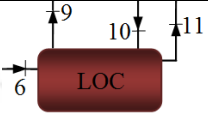
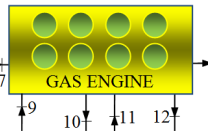
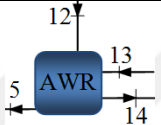
COMP		$c_e \dot{W}_{COMP} + \dot{Z}_{COMP} = c_4 \dot{E}x_4 - c_3 \dot{E}x_3$	$c_3 = c_4$
INT		$c_4 \dot{E}x_4 + c_5 \dot{E}x_5 + \dot{Z}_{INT} = c_7 \dot{E}x_7 + c_6 \dot{E}x_6$	$c_5 = c_6$
LOC		$c_6 \dot{E}x_6 + c_{10} \dot{E}x_{10} + \dot{Z}_{LOC} = c_{11} \dot{E}x_{11} + c_9 \dot{E}x_9$	$c_6 = c_{11}$ $c_9 = c_{10}$
GE		$c_7 \dot{E}x_7 + c_9 \dot{E}x_9 + c_{11} \dot{E}x_{11} + \dot{Z}_{GE} = c_8 \dot{E}x_8 + c_{10} \dot{E}x_{10} +$	$c_7 = c_8$
AWR		$c_{12} \dot{E}x_{12} + c_{13} \dot{E}x_{13} + \dot{Z}_{AWR} = c_5 \dot{E}x_5 + c_{14} \dot{E}x_{14}$	$c_{13} = 0$
TUR		$c_8 \dot{E}x_8 + \dot{Z}_{TUR} = c_e \dot{W}_{TUR} + c_{15} \dot{E}x_{15}$	$c_8 = c_{15}$

Table 3.9 shows the actual investment costs and other equipment costs which are taken from the CEV Energy Group who, the contractor company of Gaziantep Municipal Solid Waste Power Plant until 2046. The operating cost of each sub-component is also taken as 20 % of the capital investment cost according to the statement of the contractor company.

Table 3.9. Capital investment of the Gaziantep Municipal Solid Waste Power Plant

I. Fixed Capital Investment	x 10³ \$
A. Direct Costs	
1. Onsite Costs	
Purchased equipment costs	
Chiller	113.89
Blower	1.18
DeSO _x	97.62
Compressor	54.67
Intercooler	20.83
Lubrication oil cooler	52.06
Gas engine	283.15
Air water radiator	52.06
Turbine	54.67
Other plant equipment (radiator, booster and flare)	351.43
Total Purchased Equipment Cost (PEC)	1079.56
Purchased equipment installation	520.63
Piping	488.1
Instrumentation and controls	569.44
Electrical equipment and materials	488.1
Total Onsite Costs	2066.27
2. Offsite Costs	
Civil, structural, and architectural work	162.7
Service facilities	455.56
Total Offsite Costs	618.26
TOTAL DIRECT COSTS	3764.09
B. Indirect Costs	
Engineering and supervision	146.43
Construction costs	244.05
Total Indirect Costs	390.48
FIXED CAPITAL INVESTMENT	4154.57
II. Other Outlays	146.43
TOTAL CAPITAL INVESTMENT	4301

It is predicted by the company that the process with full load will be carried out at approximately 8040 h per year for 10 years. The total purchased equipment costs (PEC) of all sub-components in the plant are presented in Table 3.10 evaluating the link between first capital investment (CI) cost and operating & maintenance (OM) costs using operation time. The exergy flow rate $\dot{E}_x$, cost flow rate $\dot{C}$ and the unit exergy cost c of each stream in the plant (Fig. 3.6) are evaluated by means of the exergy cost rate balance and by the corresponding auxiliary equations (see Table 3.8) and the results are shown in Table 3.11.

Table 3.10. Cost rate of first CI and OM cost for the sub-components of the plant

Component	PEC (x 10 ³ \$)	$\dot{Z}_k^{CI}$ (\$/h)	$\dot{Z}_k^{OM}$ (\$/h)	$\dot{Z}_k^T$ (\$/h)
Chiller	113.89	1.417	0.283	1.700
Blower	1.18	0.015	0.003	0.018
DeSO _x	97.62	1.214	0.243	1.457
Compressor	54.67	0.680	0.136	0.816
Intercooler	20.83	0.259	0.052	0.311
Lubrication oil cooler	52.06	0.648	0.130	0.778
Gas engine	283.15	3.497	0.700	4.197
Air water radiator	26.03	0.648	0.130	0.778
Turbine	54.67	0.680	0.136	0.816
Other plant equipment	351.43	4.371	0.874	5.245
Total Purchased Equipment Cost (PEC)	1079.56	13.427	2.685	16.113

Table 3.11. The exergy flow rates, cost flow rates and the unit exergy costs of each stream in the plant

State	Fluid	$\dot{E}_x$ (kW)	$\dot{C}$ (\$/h)	c (\$/GJ)
1	Air	19.75	0	0
2	LFG	3930	2.491	35.25
3	LFG+Air	125.6	78.01	35.27
4	LFG+Air	323.1	78.01	90.73
5	Water	20.27	1567	114.3
6	Water	33.94	1567	191.4
7	LFG+Air	230	16.81	13.92
8	Exhaust	1456	16.81	88.08
9	Luboil	1638	1001	5905
10	Luboil	1867	1001	6728
11	Water	180	1567	1015
12	Water	212.9	159.5	122.2
13	Air	0	0	0
14	Air	120.4	20.11	8.714
15	Exhaust	980.3	3.21	11.33

The unit exergy cost of fuels and products, the relative exergy cost difference, the exergoeconomic factor, the cost rate of exergy destruction and the total investment cost rate of the sub-components in the plant are tabulated in Table 3.12 considering fuel (F) and product (P) costs of each sub-component and also CI and OM costs which are given in Table 3.10 and 3.11.

Table 3.12. Thermoeconomic results of the sub-components of the plant

Component	$c_{f,k}$ (\$/GJ)	$c_{p,k}$ (\$/GJ)	f (%)	r (%)	$\dot{D}_D$ (\$/h)	$\dot{Z}^T$ (\$/h)
COMP	55.87	78.01	5.182	39.64	14.93	0.816
INT	78.01	16.81	1.376	78.45	22.28	0.311
LOC	1001	1567	0.2618	56.47	296.4	0.778
GE	16.81	58.7	42.33	71.36	5.717	4.197
AWR	159.5	20.11	1.841	87.39	41.49	0.778
TUR	16.81	55.87	13.09	232.3	5.417	0.816

As a result of the thermoeconomic analysis on GMSWPP, the capital investment cost rate, the operating and maintenance costs rate and the total cost rate of the Jenbacher 416 type gas engine are found to be 3.497 \$/h, 0.699 \$/h and 4.197 \$/h, respectively. The total cost rate of power plant is found to be 18.44 \$/h. The net electrical output of one engine is found to be 1131 kW. The total cost rate of the power plant is found to be 18.44 \$/h. Marketing price of 1 kWh electricity is set to 13.3 ¢ throughout 10 years regarding to the agreement between the government and CEV Energy Group. The price of 1 kWh electricity production in GMSWPP is found to be 1.631 ¢ which is less than marketing price. There are 5 identical gas engines in GMSWPP which have electricity production capacity of 5655 kWh. As a result of the thermoeconomic analyses, the total investment cost and annual gain of GMSWPP are found to be 21,505,000 \$ and 5,305,450 \$, respectively. The payback period of Gaziantep Municipal Solid Waste Power Plant is found to be 4.05 years as the thermoeconomic analysis predicts, which is acceptable for energy production power plants.

3.6 Conclusion

In this chapter, the general formulations for mass, energy and exergy analyses of any energy systems as well as general principles, terminology and formulation of thermoeconomic analysis, which is also called exergoeconomic analysis are given. In addition to this, the NSGA-II optimization method is introduced and explained. Thereafter, the thermodynamic and the thermoeconomic analyses of Gaziantep Municipal Solid Waste Power Plant is performed by using the given thermodynamic and thermoeconomic relations. The actual operating conditions of GMSWPP are displayed by utilizing both thermodynamic and thermoeconomic analyses.

CHAPTER 4

RENEWABLE & SUSTAINABLE MODELS ADAPTED TO THE ACTUAL SYSTEM

4.1 Introduction

In this chapter, renewable and sustainable models adapted to the actual system are considered from the point of view of their thermodynamic and thermoeconomic analyses. All adapted models are designed so as to produce more electrical power in addition to the existing power production capacity of the Gaziantep Municipal Solid Waste Power Plant by utilizing the exhaust gases of the plant as a heat source. The temperature, pressure and mass flow rate of the exhaust gases of the plant are 566.7 °C, 1.9 bar and 16 kg/s, respectively. System description of all models are given and then thermodynamic and thermoeconomic analyses of all models are performed with respect to their design parameters. The thermodynamic analyses of the designed models are carried out by using Engineering Equation Solver (EES) program. In order to achieve thermoeconomic analyses of the models, specific exergy costing (SPECOC) method is used.

4.2 Model 1: Gas Turbine Cycle (GT)

4.2.1 System Description

The first designed model is a re-compression S-CO₂ gas turbine cycle. The developed system consists of a gas turbine heat exchanger (GTHE), a gas turbine (GT), a recuperator (REC), a low pressure compressor (LPC), a high pressure compressor (HPC), a precooler (PRE) and an intercooler (INT). The schematic layout of the gas turbine cycle is given in Fig. 4.1.

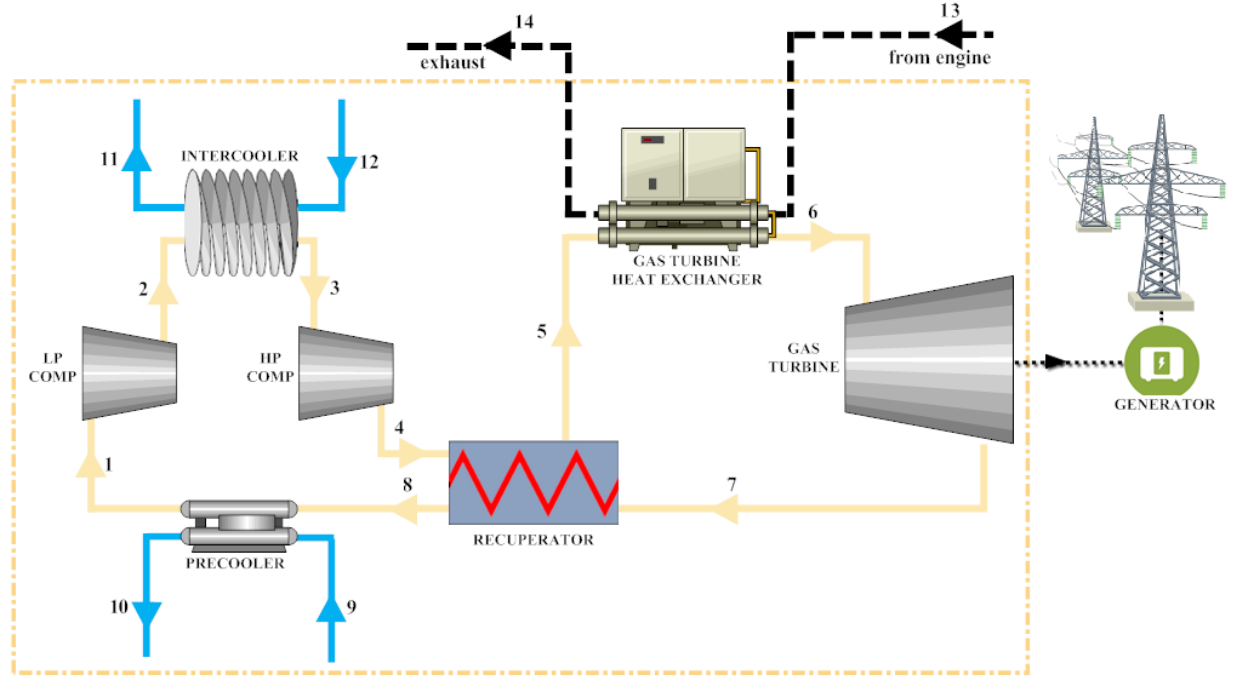


Figure 4.1. Schematic layout of the Gas turbine cycle

The design of the re-compression S-CO₂ gas turbine cycle is initiated utilizing the inlet parameters of the low pressure compressor in order to maintain the working fluid, CO₂ in the supercritical region. Therefore, the temperature and the pressure of S-CO₂ ($T_{crit}=30.98^{\circ}\text{C}$ and $P_{crit}= 73.8$ bar) are taken as 40°C and 74 bar, respectively for the inlet state of the low pressure compressor. The pressure of CO₂ is safeguarded in the range of $74\text{ bar} < P < 200\text{ bar}$ in order to mould a rational power cycle model [105]. The pressure ratio (PR_{GT}) relating the low pressure compressor and the turbine pressures is evaluated as 2.703. The effectiveness of all heat exchangers ($\mathcal{E}_{HE}$) and the isentropic efficiencies of the compressors (η_C) and the gas turbine (η_{GT}) are taken to be 0.85, 0.85 and 0.92, respectively, relying upon the vales quoted in in the literature [93-105]. The working fluid CO₂ is heated in GTHE by means of the waste heat from MSW plant before entering the GT for heat recovery. The logarithmic mean temperature difference of GTHE ($LMTD_{GTHE}$) is taken to be as 11.05°C in this model. In addition to these assumptions, the model is completed by making use of the following general assumptions: the system operates steadily; the kinetic and potential energy changes, the pressure losses in the pipes and at the heat

exchangers are neglected; the state at the exit of the precooler and intercooler are taken to be as saturated liquid for the working fluid.

4.2.2 Thermodynamic Analysis

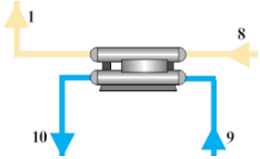
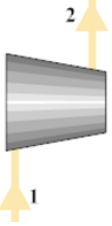
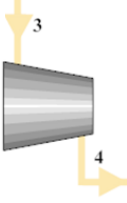
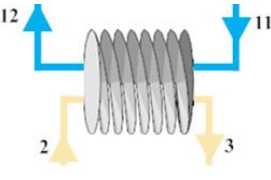
The heat transfer rates, the work, the exergy destructions and the exergy efficiencies of all sub-components are evaluated by means of the governing equations related with continuity, energy and exergy equations, which are briefly given in Table 4.1 for each component. Besides, the energy and exergy efficiencies of the gas turbine cycle are calculated by using the following equations given below:

$$\eta_{GT} = (\dot{W}_{GT} - (\dot{W}_{LPC} + \dot{W}_{HPC})) / \dot{m}_{exh} (h_{13} - h_{14}) \quad (4.1)$$

$$\varepsilon_{GT} = (\dot{W}_{GT} - (\dot{W}_{LPC} + \dot{W}_{HPC})) / \dot{m}_{exh} (\psi_{13} - \psi_{14}) \quad (4.2)$$

Table 4.1. Energy and exergy equations of the subcomponents of the GT cycle

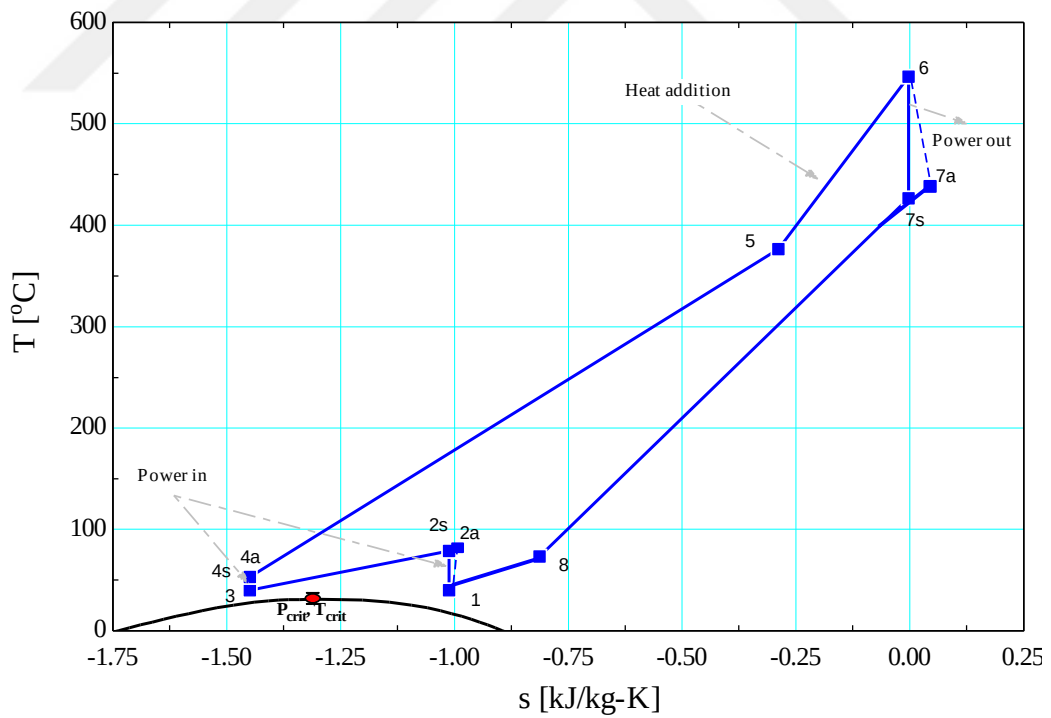
GTHE		$\dot{m}_{13} = \dot{m}_{14} = \dot{m}_{exh}, \dot{m}_5 = \dot{m}_6 = \dot{m}_{CO_2}$ $\dot{m}_{exh} (h_{13} - h_{14}) = \dot{m}_{CO_2} (h_6 - h_5)$ $\dot{E}x_{D,GTHE} = \dot{m}_{exh} (\psi_{13} - \psi_{14}) - \dot{m}_{CO_2} (\psi_6 - \psi_5)$ $\varepsilon_{GTHE} = \frac{\dot{m}_{exh} (\psi_{13} - \psi_{14})}{\dot{m}_{CO_2} (\psi_6 - \psi_5)}$
GT		$\dot{m}_{115} = \dot{m}_{116} = \dot{m}_{CO_2}$ $\dot{W}_{GT,a} = \dot{m}_{CO_2} (h_6 - h_7)$ $\dot{W}_{GT,s} = \dot{m}_{CO_2} (\psi_6 - \psi_7)$ $\dot{E}x_{D,GT} = \dot{W}_{GT,s} - \dot{W}_{GT,a}, \varepsilon_{GT} = \frac{\dot{W}_{GT,a}}{\dot{W}_{GT,s}}$
REC		$\dot{m}_4 = \dot{m}_5 = \dot{m}_7 = \dot{m}_8 = \dot{m}_{CO_2}$ $(h_4 - h_5) = (h_8 - h_7)$ $\dot{E}x_{D,REC} = \dot{m}_{CO_2} ((\psi_4 - \psi_5) - (\psi_8 - \psi_7))$ $\varepsilon_{REC} = \frac{(\psi_4 - \psi_5)}{(\psi_8 - \psi_7)}$

PRE		$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{wat}, \dot{m}_1 = \dot{m}_8 = \dot{m}_{CO_2}$ $\dot{m}_{CO_2} (h_8 - h_1) = \dot{m}_{wat} (h_{10} - h_9)$ $\dot{E}x_{D,PRE} = \dot{m}_{CO_2} (\psi_8 - \psi_1) - \dot{m}_{wat} (\psi_{10} - \psi_9)$ $\varepsilon_{PRE} = \frac{\dot{m}_{CO_2} (\psi_8 - \psi_1)}{\dot{m}_{wat} (\psi_{10} - \psi_9)}$
LPC		$\dot{m}_1 = \dot{m}_2 = \dot{m}_{CO_2}$ $\dot{W}_{LPC,a} = \dot{m}_{CO_2} (h_2 - h_1)$ $\dot{W}_{LPC,s} = \dot{m}_{CO_2} (\psi_2 - \psi_1)$ $\dot{E}x_{D,LPC} = \dot{W}_{LPC,a} - \dot{W}_{LPC,s}, \varepsilon_{LPC} = \frac{\dot{W}_{LPC,s}}{\dot{W}_{LPC,a}}$
HPC		$\dot{m}_3 = \dot{m}_4 = \dot{m}_{CO_2}$ $\dot{W}_{HPC,a} = \dot{m}_{CO_2} (h_4 - h_3)$ $\dot{W}_{HPC,s} = \dot{m}_{CO_2} (\psi_4 - \psi_3)$ $\dot{E}x_{D,HPC} = \dot{W}_{HPC,a} - \dot{W}_{HPC,s}, \varepsilon_{HPC} = \frac{\dot{W}_{HPC,s}}{\dot{W}_{HPC,a}}$
INT		$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_{wat}, \dot{m}_1 = \dot{m}_8 = \dot{m}_{CO_2}$ $\dot{m}_{CO_2} (h_2 - h_3) = \dot{m}_{wat} (h_{12} - h_{11})$ $\dot{E}x_{D,INT} = \dot{m}_{CO_2} (\psi_2 - \psi_3) - \dot{m}_{wat} (\psi_{12} - \psi_{11})$ $\varepsilon_{INT} = \frac{\dot{m}_{CO_2} (\psi_2 - \psi_3)}{\dot{m}_{wat} (\psi_{11} - \psi_{12})}$

Thermodynamic properties of each substance at the considered states in the system are tabulated and given in Table 4.2. In addition to this, T-s diagram of the GT cycle is shown in Fig. 4.2.

Table 4.2. Thermodynamic properties of states of the GT cycle

State	Fluid	T (°C)	P (bar)	$\dot{m}$ (kg/s)	h (kJ/kg)	s (kJ/kg.K)	ψ (kJ/kg)	$\dot{E}_x$ (kW)
0	CO ₂	20	1.0	-	-5.168	-0.01403	-	-
0'	Water	20	1.0	-	83.93	0.2962	-	-
0''	Air	20	1.0	-	293.6	5.682	-	-
1	CO ₂	40	74	11.09	-84.6	-1.012	213.1	2364
2	CO ₂	78.93	121.7	11.09	-63.72	-1.003	231.3	2566
3	CO ₂	40	121.7	11.09	-211.6	-1.45	214.5	2380
4	CO ₂	53.1	200	11.09	-199.2	-1.444	225.2	2498
5	CO ₂	376.6	200	11.09	315.1	-0.2889	400.8	4446
6	CO ₂	546.7	200	11.09	524.2	-0.00295	526.2	5837
7	CO ₂	426.6	74	11.09	392.3	0.01355	389.4	4320
8	CO ₂	73.1	74	11.09	-19.39	-0.8128	219.9	2440
9	Water	20	1.0	10.50	83.93	0.2962	0	0
10	Water	34	1.0	10.50	142.5	0.4913	1.355	14.23
11	Water	20	1.0	15.15	83.93	0.2962	0	0
12	Water	42	1.0	15.15	175.9	0.5989	3.289	49.82
13	Exhaust	566.7	1.9	16.00	866.2	6.591	306.4	4903
14	Exhaust	381.9	1.9	16.00	665.5	6.321	184.8	2957

**Figure 4.2.** T-s diagram of the GT cycle

Thermodynamic analysis of the GT cycle is carried out via the equations given in Table 4.1. The results of the calculations of the GT cycle are given in Table 4.3.

Table 4.3. Thermodynamic results of the GT cycle

Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}_{x_F}$ (kW)	$\dot{E}_{x_P}$ (kW)	$\dot{E}_{x_D}$ (kW)	ε (%)
GTHE	2730	-	1946	1391	555.6	71.5
GT	-	1463	1517	1463	53.69	96.5
REC	5726	-	1948	1880	68.12	96.5
PRE	723.3	-	75.87	14.23	61.64	18.8
LPC	-	231.5	231.5	202.6	28.97	87.5
HPC	-	136.7	136.7	118.3	18.45	86.5
INT	1394	-	186.9	49.82	137.1	26.7
GT System	Energy efficiency (%)				34.1	
	Exergy efficiency (%)				56.3	

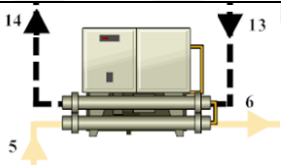
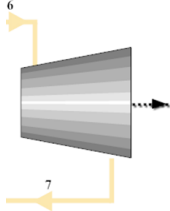
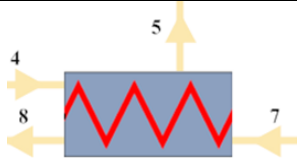
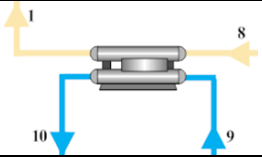
As a result of the thermodynamic analysis of the GT cycle, the net power output, the energy and the exergy efficiencies were found to be 1095 kW, 34.1 % and 56.3 %, respectively. This power output puts another 19.33 % power on top of the existing power output of the system. The efficiencies of the GT cycle are found to be reasonable when compared with the other S-CO₂ gas turbine cycles in the open literature [93-105]. However, the performance of the GT cycle can be increased by decreasing the exergy destructions. It is clearly seen from the Table 4.3 that the maximum and the minimum exergy destructions were observed at GTHE and HPC units, respectively. The high amount of the exergy destruction of GTHE is estimated due to usage of different working fluids simultaneously, heat transfer through finite temperature difference and fluid-structure interaction. However, it should be mentioned here that maximum exergy destruction does not always mean minimum exergy efficiency or vice versa. As it can be seen from the above analyses that the GTHE has the highest exergy destruction amounting to 555.6 kW while it has the exergy efficiency of 71.5 % which is not the lowest among others. Starting in different terms, the precooler has the exergy efficiency of 18.8% (the lowest) has the exergy destruction of only 61.64 kW which is almost nine folds lower than that of

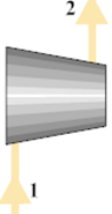
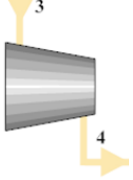
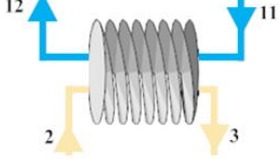
GTHE. In this system, the recuperator and the gas turbine have the highest exergy efficiency of 96.5 %. Despite of being a typical heat exchanger, the reason of the highest exergy efficiency of the recuperator can be explained by the fact that only CO₂ streams in the same phase were used in both streams of the recuperator.

4.2.3 Thermoeconomic Analysis

To perform a thermoeconomic analysis on GT cycle model, at first, all energetic and exergetic flows in all states of system are to be determined. Then, all sub-components of the system are designated by following the fuel and product approach. All exergy additions to and removals from a sub-component are considered as the fuel and product, respectively. As a result of the application of the SPECO procedure, the cost balance and the auxiliary equations are obtained for each sub-component of the system. The relevant equations are given in Table 4.4.

Table 4.4. Exergetic cost rate balances and auxiliary equations for the sub-components of the GT cycle

GTHE		$c_{13}\dot{E}x_{13} + c_5\dot{E}x_5 + \dot{Z}_{GTHE} = c_{14}\dot{E}x_{14} + c_6\dot{E}x_6$ $c_{13} = \text{known}, \frac{\dot{C}_{13}}{\dot{E}x_{13}} = \frac{\dot{C}_{14}}{\dot{E}x_{14}}$
GT		$c_6\dot{E}x_6 + \dot{Z}_{GT} = c_{GT}\dot{W}_{GT} + c_7\dot{E}x_7$ $\frac{\dot{C}_6}{\dot{E}x_6} = \frac{\dot{C}_7}{\dot{E}x_7}$
REC		$c_4\dot{E}x_4 + c_7\dot{E}x_7 + \dot{Z}_{REC} = c_5\dot{E}x_5 + c_8\dot{E}x_8$ $\frac{\dot{C}_7}{\dot{E}x_7} = \frac{\dot{C}_8}{\dot{E}x_8}$
PRE		$c_8\dot{E}x_8 + c_9\dot{E}x_9 + \dot{Z}_{PRE} = c_1\dot{E}x_1 + c_{10}\dot{E}x_{10}$ $c_9 = 0, \frac{\dot{C}_1}{\dot{E}x_1} = \frac{\dot{C}_8}{\dot{E}x_8}$

LPC		$c_{LPC}\dot{W}_{LPC} + \dot{Z}_{LPC} = c_2\dot{E}x_2 - c_1\dot{E}x_1$ $\frac{\dot{C}_1}{\dot{E}x_1} = \frac{\dot{C}_2}{\dot{E}x_2}$
HPC		$c_{HPC}\dot{W}_{HPC} + \dot{Z}_{HPC} = c_4\dot{E}x_4 - c_3\dot{E}x_3$
INT		$c_2\dot{E}x_2 + c_{11}\dot{E}x_{11} + \dot{Z}_{INT} = c_3\dot{E}x_3 + c_{12}\dot{E}x_{12}$ $c_{11} = 0, \frac{\dot{C}_2}{\dot{E}x_2} = \frac{\dot{C}_3}{\dot{E}x_3}$

$\dot{Z}$ stands for the entire cost rate related with the capital investment (CI) and operation and maintenance costs (OM) of each sub-component of the system. The capital cost rate of the components is calculated by using the Eq. (3.30). In the thermoeconomic analysis of a system, total lifetime (n), annual operation time (N), interest rate (i), maintenance factor (φ) and capital recovery factor (CRF) play important role. In this thesis, the economic constants are selected for a rational thermoeconomic analysis taken into account existing power plant in Gaziantep, literature and the reality of the economic aspect of Turkey [134-141]. The existing power plant in Gaziantep can be operated until 2046 according to the agreement between the municipality and the contractor company and also it can be produced 8040 h annual power regarding to actual operating hours. Due to this reason, the total life time and annual operation time are selected as 30 years and 8040 hours, respectively. Although Turkey is a promising country, it can be experienced economic difficulties due to its geographical location and the internal disturbance such as coup attempt, terrorist attacks it is experienced in the country. For this reason, the interest rate was determined to be higher than its real value and it was accepted as 15%. All the theoretical and practical thermoeconomic studies performed in the literature shows the maintenance factor is selected to be as the overall acceptance of 1.06 [134-141]. Selected economic constants are given in Table 4.5.

Table 4.5. Economic constants

Total life time (n)	30
Annual operation time (N)	8040
Interest rate (i)	15%
Maintenance factor (φ)	1.06
Capital recovery factor (CRF)	$CRF = \frac{i(1+i)^n}{(1+i)^n - 1}$

In order to perform the exergoeconomic analyses, the cost relations of the sub-components are required for the capital costs of the system. The capital costs are calculated using the thermodynamic relations of each component. The capital investment cost function of each component is given in Table 4.6 [129].

Table 4.6. Capital investment cost functions of sub-components of the GT cycle

GTHE	$PEC_{GTHE} = 2681(A_{GTHE})^{0.59}$
GT	$PEC_{GT} = 4405(\dot{W}_{GT})^{0.7}$
REC	$PEC_{REC} = 2681(A_{REC})^{0.59}$
PRE	$PEC_{PRE} = 2143(A_{PRE})^{0.514}$
LPC and HPC	$PEC_{COMP} = 71.1 \left[\frac{\dot{m}_{CO_2}}{0.92 - \eta_c} \right] (PR_{GT})(\ln(PR_{GT}))$
INT	$PEC_{INT} = 2143(A_{INT})^{0.514}$

All heat exchangers in the GT cycle are considered as shell and tube types and modeled by using the Eq. (3.27). The heat transfer coefficients are assumed to be as 1.1, 0.7 and 2 for gas turbine heat exchanger, recuperator and both precooler and intercooler, respectively [129]. Herein, it should be pointed out that the chemical engineering plant cost index (CEPCI) is a vital parameter to consider in the calculation of the total cost of the plant and it is calculated using by taking the ratio of the cost indexes of reference the year and the original year [145]. In this model,

the value of CEPCI is calculated as 1.052 and the components are updated with respect to this value. The total cost rate $\dot{C}_{tot}$ (\$/h) is calculated by summation of the cost rate difference of the inlet and the exit streams of the exhaust gas and the total capital cost rate of the system.

The equations given in Table 4.6 are used in the calculation of the capital investment of the GT cycle. Then, the operating and the maintenance costs are calculated by using the chemical engineering plant cost index and the constants given in Table 4.5. The exergy flow rates, the cost flow rates and the unit exergy costs of the GT cycle are tabulated as shown in Table 4.7.

Table 4.7. Exergy flow rates, unit exergy costs and cost flow rates of the GT cycle

State	$\dot{E}_x$ (kW)	c (\$/GJ)	$\dot{C}$ (\$/h)
1	2364	4.851	41.28
2	2566	4.851	44.82
3	2380	4.851	41.55
4	2498	4.881	43.89
5	4446	4.881	78.12
6	5837	4.851	101.9
7	4320	4.851	75.44
8	2440	4.851	42.6
9	0	0	0
10	14.23	28.96	1.484
11	0	0	0
12	49.82	19.46	3.49
13	4903	3.21	56.66
14	2957	3.21	34.17

The calculations resulting from thermoeconomic analysis of the GT cycle are shown in Table 4.8. The total capital cost rate and the total cost rate of the GT cycle are found to be 20.47 \$/h and 44.57 \$/h, respectively. The lower the summation of the cost rate of exergy destruction and the capital cost rate, the higher the exergoeconomic factor is observed. Validity of this statement can be seen from the fact that, the work-producing devices have higher exergoeconomic factor among the others such that the exergoeconomic factors of GT, HPC and LPC are 93.94 %, 93.94 %, 93.94 %, respectively.

72.27 % and 62.39 %, respectively. In other words, the work-producing devices operating in this system are more effective from the exergoeconomic standpoint. The GTHE has a relatively higher relative cost difference of 51.11 % and also a relatively lower exergoeconomic factor of 16.98 % which may partially be related with the high amount of the exergy destruction taking place. Finally, all heat exchangers have lower exergoeconomic factors values stemming from their high exergy destruction costs when compared with the other components of operating in the system. A special care should be paid on these components in order to increase the total system performance.

Table 4.8. Thermoeconomic results of the GT cycle

Component	$\dot{E}x_D$ (kW)	c_f (\$/GJ)	c_p (\$/GJ)	$\dot{D}_D$ (\$/h)	$\dot{Z}$ (\$/h)	r (%)	f (%)
GTHE	555.6	3.21	4.85	6.42	1.313	51.11	16.98
GT	53.69	4.85	7.78	0.94	14.54	60.55	93.94
REC	68.12	4.85	4.88	1.19	1.398	0.62	21.20
PRE	61.64	4.85	4.85	1.08	0.159	2.7×10^{-9}	12.86
LPC	28.97	3.51	4.85	0.37	0.608	38.03	62.39
HPC	18.45	3.51	4.88	0.23	0.608	38.89	72.27
INT	172.8	4.93	4.93	3.07	0.335	3.4×10^{-8}	9.842

4.3 Model 2: Organic Rankine Cycle (ORC)

4.3.1 System Description

Organic Rankine cycle is the devised second model to analyze the additional electricity production in the actual power plant. The ORC system consists of an evaporator (EVAP), an ORC turbine (OT), an ORC condenser (OC) and an ORC pump (OP). The schematic representation of the ORC system is given in Fig. 4.3. As it can be seen from the schematic representation of the ORC system, the working fluid is pumped from the condenser to the evaporator (state 6-3). Then the pressurized working fluid is passed through the evaporator to the ORC turbine (state

3-4). In the evaporator, the high pressure working fluid is transferred into superheated vapor state by using the heat addition from the exhaust gases (state 1-2). After that, shaft power is generated by the expansion of high energy fluid in the ORC turbine (state 3-4). Finally, low pressure vapor from the turbine exit is passed through the condenser and while passing its state changes from vapor to the saturated liquid state (state 5-6) by rejecting its heat to the condensation water (state 7-8). The inlet and exit temperatures of the exhaust gas are kept at constant throughout the system cycle. The inlet temperature of the turbine is also kept constant according to temperature difference (ΔT) which is defined as the temperature difference between the critical temperature of the working fluid and the inlet temperature of the turbine. The effectiveness of the evaporator and the condenser (ϵ_{HE}) and the isentropic efficiencies of the ORC pump (η_{OP}) and the ORC turbine (η_{OT}) were taken as 0.85 for all components by using the data available in the literature [106-122]. Additional assumptions made related with the system are as follows: the system operates steadily; the kinetic and potential energy changes are negligible; the pressure losses in the pipes and the heat exchangers can be neglected. The design parameters of the ORC system are given in Table 4.9.

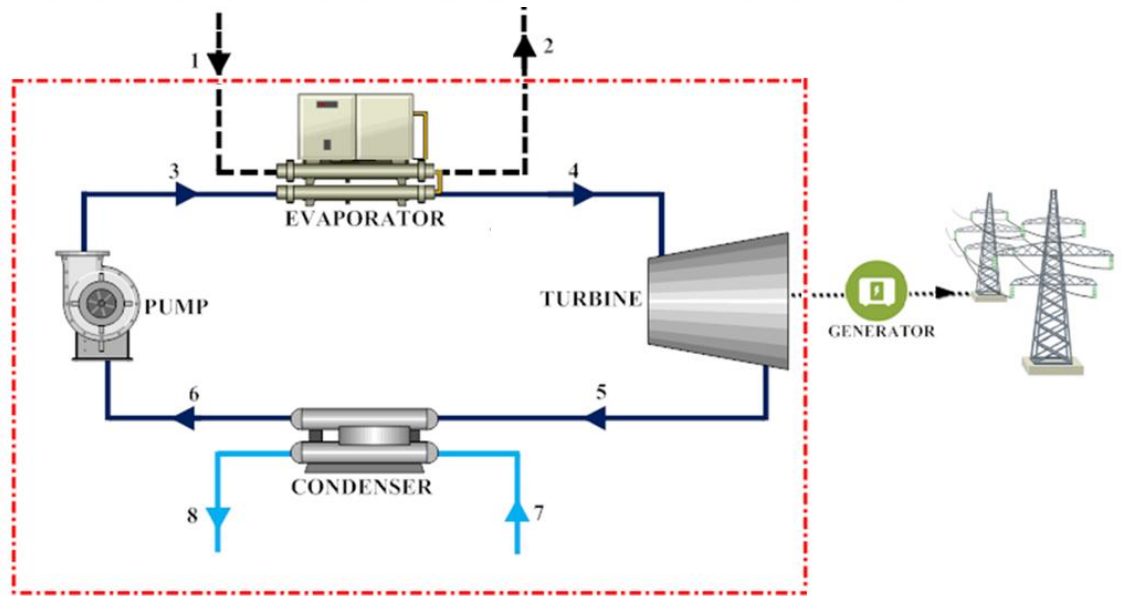


Figure 4.3. Schematic layout of the ORC

Table 4.9. Design parameters of the ORC

Parameters	Symbol	Value
Inlet temperature of heat source (°C)	T_1	566.7
Outlet temperature of heat source (°C)	T_2	140
Mass flow rate of heat source (kg/s)	$\dot{m}_{exh}$	16
Pressure ratio of ORC	PR_{ORC}	5
Temperature difference	ΔT	15
Inlet temperature of ORC turbine (°C)	T_4	Ranging from 275.9 to 329.6
Exit temperature of ORC turbine (°C)	T_5	Ranging from 262.1 to 312.4
Isentropic efficiency of pump	η_{OP}	0.85
Isentropic efficiency of ORC turbine	η_{OT}	0.85
Effectiveness of evaporator and condenser	ϵ_{HE}	0.85
Inlet temperature of cooling water (°C)	T_7	20
Outlet temperature of cooling water (°C)	T_8	42
Ambient temperature (°C)	T_0	20

In ORC systems generally low temperature ranges (80-120 °C) are used in order to generate electricity. In the current model, the temperature of the exhaust gas is higher than the low temperature range in ORC systems. Due to this reason, high decomposition temperature of organic fluids are selected for the ORC system which are toluene, octamethyltrisiloxane (MDM), octamethyl cyclotetrasiloxane (D₄) and n-decane. The properties of working fluids are given in Table 4.10.

Table 4.10. Properties of the organic fluids

Fluid	Mol.Weight (kg/kmol)	P_{crit} (bar)	T_{crit} (°C)	T_{dec} (°C)	Fluid type
Toluene	92.14	41.3	318.6	≈426.85	Dry
MDM	236.53	14.2	290.9	≈400	Dry
D ₄	296.62	13.3	313.3	≈400	Dry
n-decane	142.3	21.0	344.6	≈401	Dry

4.3.2 Thermodynamic Analysis

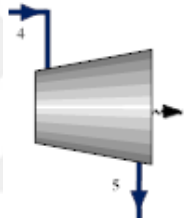
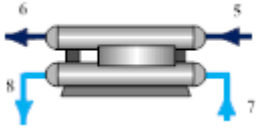
The heat transfer rates, the work, the exergy destructions and the exergy efficiencies of all sub-components of ORC system are evaluated by means of the governing equations related with continually, energy and exergy equations, which are given in

Table 4.11. In addition to this, the energy and the exergy efficiencies of the organic Rankine cycle are achieved using the equations given below;

$$\eta_{ORC} = (\dot{W}_{OT} - \dot{W}_{OP}) / \dot{m}_{exh} (h_1 - h_2) \quad (4.3)$$

$$\varepsilon_{ORC} = (\dot{W}_{OT} - \dot{W}_{OP}) / \dot{m}_{exh} (\psi_1 - \psi_2) \quad (4.4)$$

Table 4.11. Energy and exergy equations of the subcomponents of the ORC

EVAP		$\dot{m}_1 = \dot{m}_2 = \dot{m}_{exh}, \dot{m}_5 = \dot{m}_6 = \dot{m}_{wf}$ $\dot{m}_{exh} (h_1 - h_2) = \dot{m}_{wf} (h_4 - h_3)$ $\dot{E}x_{D,EVAP} = \dot{m}_{exh} (\psi_1 - \psi_2) - \dot{m}_{wf} (\psi_4 - \psi_3)$ $\varepsilon_{EVAP} = \frac{\dot{m}_{exh} (\psi_1 - \psi_2)}{\dot{m}_{wf} (\psi_4 - \psi_3)}$
OT		$\dot{m}_4 = \dot{m}_5 = \dot{m}_{wf}$ $\dot{W}_{OT,a} = \dot{m}_{wf} (h_4 - h_5)$ $\dot{W}_{OT,s} = \dot{m}_{wf} (\psi_4 - \psi_5)$ $\dot{E}x_{D,OT} = \dot{W}_{OT,s} - \dot{W}_{OT,a}, \varepsilon_{OT} = \frac{\dot{W}_{OT,a}}{\dot{W}_{OT,s}}$
OC		$\dot{m}_5 = \dot{m}_6 = \dot{m}_{wf}, \dot{m}_7 = \dot{m}_8 = \dot{m}_{wat}$ $\dot{m}_{wf} (h_5 - h_6) = \dot{m}_{wat} (h_8 - h_7)$ $\dot{E}x_{D,OC} = \dot{m}_{wf} (\psi_5 - \psi_6) - \dot{m}_{wat} (\psi_8 - \psi_7)$ $\varepsilon_{OC} = \frac{(\psi_5 - \psi_6)}{(\psi_8 - \psi_7)}$
OP		$\dot{m}_3 = \dot{m}_6 = \dot{m}_{wf}$ $\dot{W}_{OP,a} = \dot{m}_{wf} (h_3 - h_6)$ $\dot{W}_{OP,s} = \dot{m}_{wf} (\psi_3 - \psi_6)$ $\dot{E}x_{D,OP} = \dot{W}_{OP,a} - \dot{W}_{OP,s}, \varepsilon_{OP} = \frac{\dot{W}_{OP,s}}{\dot{W}_{OP,a}}$

The thermodynamic analysis of the selected working fluids in the ORC system is carried out by using the equations already given in Table 4.11. The results obtained from the analyses for each working fluid in the ORC system are given through the tables from Table 4.12 to 4.15.

Table 4.12. Thermodynamic results for toluene in ORC

Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}x_F$ (kW)	$\dot{E}x_P$ (kW)	$\dot{E}x_D$ (kW)	ε (%)
EVAP	7226	-	3725	2294	1430	61.6
OT	-	590.8	647.7	590.8	56.91	91.21
OC	5558	-	1652	168.9	1483	10.22
OP	-	6.237	6.237	5.53	0.7062	88.68
ORC System	Energy efficiency (%)					8.09
	Exergy efficiency (%)					15.69

Table 4.13. Thermodynamic results for MDM in ORC

Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}x_F$ (kW)	$\dot{E}x_P$ (kW)	$\dot{E}x_D$ (kW)	ε (%)
EVAP	7226	-	3725	2210	1515	59.33
OT	-	335.8	368.4	335.8	32.53	91.17
OC	5809	-	1844	176.5	1668	9.571
OP	-	2.907	2.907	2.578	0.3295	88.67
ORC System	Energy efficiency (%)					4.61
	Exergy efficiency (%)					8.94

Table 4.14. Thermodynamic results for D₄ in ORC

Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}x_F$ (kW)	$\dot{E}x_P$ (kW)	$\dot{E}x_D$ (kW)	ε (%)
EVAP	7226	-	3725	2250	1474	60.42
OT	-	295.7	323.2	295.7	27.43	91.51
OC	5849	-	1929	177.7	1752	9.21
OP	-	2.488	2.488	2.206	0.282	88.67
ORC System	Energy efficiency (%)					4.06
	Exergy efficiency (%)					7.87

Table 4.15. Thermodynamic results for n-decane in ORC

Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}x_F$ (kW)	$\dot{E}x_p$ (kW)	$\dot{E}x_D$ (kW)	ε (%)
EVAP	7226	-	3725	2325	1400	62.42
OT	-	342.6	373	342.6	30.35	91.86
OC	5800	-	1953	176.2	1776	9.025
OP	-	0.8865	0.8865	0.7861	0.1004	88.67
ORC System	Energy efficiency (%)					4.73
	Exergy efficiency (%)					9.18

Thermodynamic analyses have shown that, the net power outputs obtained from the ORC system in using toluene, MDM, D4 and n-decane are 584.6 kW, 332.9 kW, 293.3 kW and 341.8 kW, respectively. The exergy efficiencies of the ORC system **in** using the same working fluids spelled above in the same sequence are calculated as 15.69 %, 8.94 %, 7.87 % and 9.18 %, respectively. As it can be seen from the tables showing the thermodynamic results, toluene is found to be the working fluid performing better among all with respect to the net power output and the exergy efficiency of the ORC system. However, the efficiencies of the cycles can be improved by decreasing the exergy destructions in ORC system. The exergy destruction of evaporator and condenser can be expected higher when compared to ORC turbine and pump. The highest exergy destruction is found to be 1776 kW in the condenser when the working fluid is n-decane in ORC system.

4.3.3 Thermoeconomic Analysis

When performing thermoeconomic analysis on ORC model, the same procedure is followed as in the case of GT cycle in section 4.2.3. By applying of the SPECO procedure, the cost balance and the auxiliary equations are obtained for each sub-component of the system. The relevant equations are given in tabular form Table 4.16.

Table 4.16. Exergy cost rate balances and auxiliary equations for sub-components of the ORC

EVAP		$c_1 \dot{E}x_1 + c_3 \dot{E}x_3 + \dot{Z}_{EVAP} = c_2 \dot{E}x_2 + c_4 \dot{E}x_4$ $c_1 = \text{known}, \frac{\dot{C}_1}{\dot{E}x_1} = \frac{\dot{C}_2}{\dot{E}x_2}$
OT		$c_4 \dot{E}x_4 + \dot{Z}_{OT} = c_{OT} \dot{W}_{OT} + c_5 \dot{E}x_5$ $\frac{\dot{C}_4}{\dot{E}x_4} = \frac{\dot{C}_5}{\dot{E}x_5}$
OC		$c_5 \dot{E}x_5 + c_7 \dot{E}x_7 + \dot{Z}_{OC} = c_6 \dot{E}x_6 + c_8 \dot{E}x_8$ $c_7 = 0, \frac{\dot{C}_5}{\dot{E}x_5} = \frac{\dot{C}_6}{\dot{E}x_6}$
OP		$c_{OP} \dot{W}_{OP} + \dot{Z}_{OP} = c_3 \dot{E}x_3 - c_6 \dot{E}x_6$ $\frac{\dot{C}_3}{\dot{E}x_3} = \frac{\dot{C}_6}{\dot{E}x_6}$

The capital cost rate of the components are calculated by using the Eq. (3.30). In order to perform thermoeconomic analysis of ORC system, the economic constants are taken to be the same as in GT cycle and are given in Table 4.5. The capital costs are calculated by using the thermodynamic relations belonging to each component. The capital investment cost function of each component is given in Table 4.17 [97, 129].

Table 4.17. Capital investment costs function of sub-components of the ORC

EVAP	$PEC_{EVAP} = 130(A_{EVAP} / 0.093)^{0.78}$
OT	$PEC_{OT} = (\frac{1536 * \dot{m}_{wf}}{0.92 - \eta_{OT}}) * \ln(PR_{ORC}) * (1 + \exp(0.036 * T_4 - 54.4))$
OC	$PEC_{OC} = 1773(\dot{m}_{wf})$
OP	$PEC_{OP} = 3540(\dot{W}_{OP})^{0.71}$

The heat transfer coefficient of the evaporator is assumed to be as 0.3 [97, 129]. As a result of thermoeconomic analyses, the total cost rate of the ORC with toluene, MDM, D4 and n-decane are 52.74 (\$/h), 55.89 (\$/h), 56.43 (\$/h) and 51.4 (\$/h), respectively. The total capital cost rates of toluene, MDM, D4 and n-decane are also found to be 8.50 (\$/h), 11.71 (\$/h), 12.21 (\$/h) and 7.79 (\$/h), respectively. Although the n-decane has the lowest total cost rate compared with the other working fluids, the produced power of the n-decane is significantly less than the toluene. Due to this reason, n-decane cannot be considered as the optimum working fluid under the condition of the design parameters of the ORC. By means of the thermoeconomic analyses, the components of the ORC system are also investigated by taking into account of the unit exergy cost of the product and fuel, the relative cost difference, the exergoeconomic factor, the cost rate of exergy destruction, the capital cost rate and the exergy destruction rate. The results are given in Table 4.18. For all working fluids, the exergoeconomic factor for ORC turbine and pump are found to be higher than that of the evaporator and the condenser. This situation may be directly attributed to the high exergy destruction taking place at these the components. As a result of this, the cost rates of the exergy destruction of the evaporator and the condenser are remarkably higher than that of the ORC turbine and the pump. It is clearly seen that an optimization procedure should be realized in order to select a proper working fluid by also taking into account the thermoeconomic results at the same time.

Table 4.18. Thermoeconomic performance of the components of the ORC

Working fluid	Component	$\dot{E}x_D$ (kW)	c_f (\$/GJ)	c_p (\$/GJ)	$\dot{D}_D$ (\$/h)	$\dot{Z}$ (\$/h)	r (%)	f (%)
Toluene	EVAP	1430	3.21	5.942	16.53	1.253	45.98	6.723
	OT	56.91	5.942	9.482	1.217	6.642	37.34	83.84
	OC	1483	5.942	51.28	31.73	0.3335	88.41	0.9891
	OP	0.7062	9.482	0.3022	0.02411	0.2743	96.81	91.54
MDM	EVAP	1515	3.21	7.022	17.5	1.187	54.28	6.054
	OT	32.53	7.022	15.46	0.8223	9.874	54.6	91.94
	OC	1668	7.022	56.1	42.16	0.4957	87.48	1.105
	OP	0.3295	15.46	0.1337	0.0183	0.1595	99.14	89.21
D4	EVAP	1474	3.21	6.868	17.04	1.238	53.26	6.46
	OT	27.43	6.868	16.71	0.6783	10.31	58.91	93.53
	OC	1752	6.868	57.85	43.31	0.5178	88.13	1.124
	OP	0.282	16.71	0.1402	0.0169	0.1428	99.16	88.89
n-decane	EVAP	1034	3.21	6.553	11.95	1.168	51.02	8.50
	OT	31.78	6.553	11.59	0.7496	6.012	43.44	88.4
	OC	1792	6.553	59.26	42.28	0.3018	88.94	0.674
	OP	0.732	11.59	0.1245	0.0305	0.3144	98.93	90.73

4.4 Model 3: Kalina Cycle (KC)

4.4.1 System Description

Here a Kalina cycle model is investigated for the additional electricity production at the actual power plant as a third alternative model. The Kalina cycle consists of a generator (GEN), a separator (SEP), a turbine (KT), a valve (VAL), a low temperature recuperator (LTR), a condenser (KCON), a pump (KP) and a high temperature recuperator (HTR). The schematic layout of the Kalina cycle is given in Fig. 4.4.

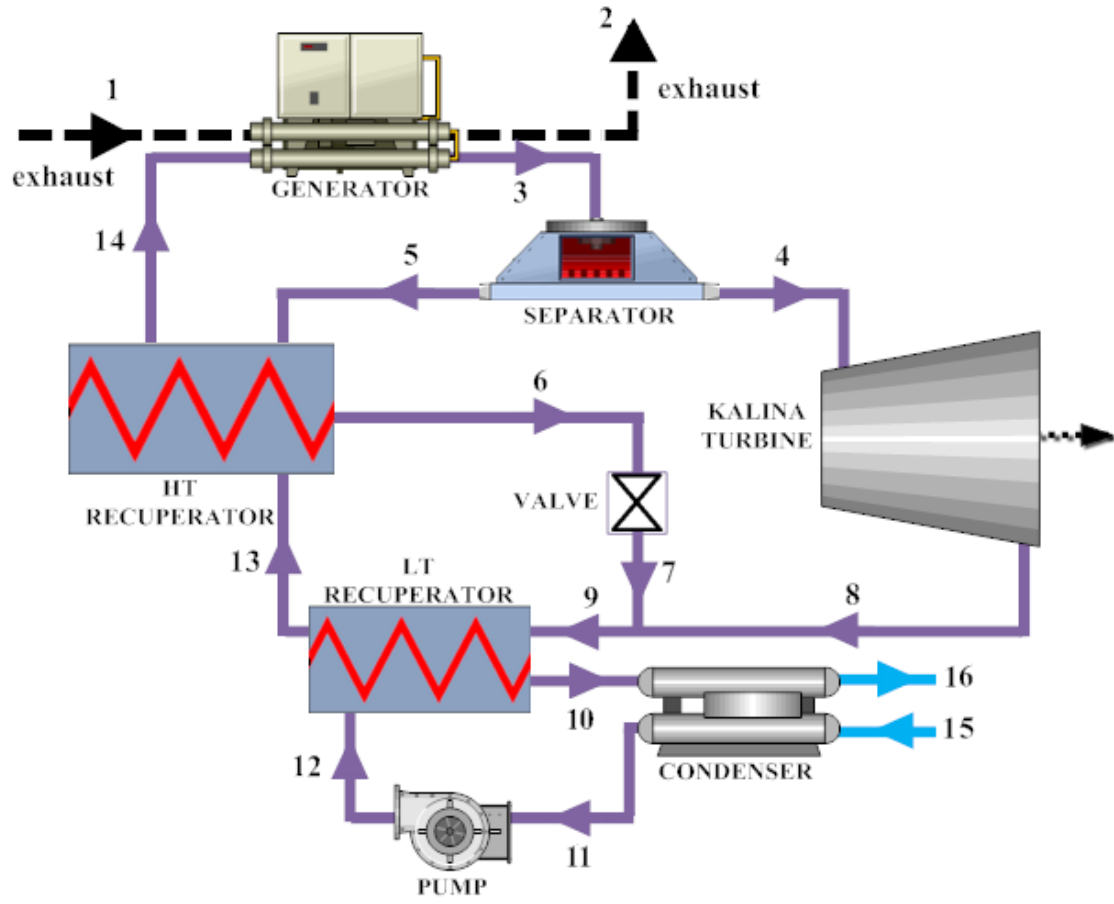


Figure 4.4. Schematic layout of the Kalina cycle

As it can be seen from the schematic representation of the Kalina cycle, the two-phase ammonia-water (aw) mixture is separated into ammonia-rich vapor and ammonia-poor solution in the separator after the heat addition in the generator (state 14-3-4-5). The ammonia-rich vapor passes through the turbine and power is generated (state 4-8). The temperature and the pressure of ammonia-poor solution are decreased in HTR and valve, respectively (state 5-6-7). Then the ammonia-poor solution is mixed with ammonia-rich solution before entering to the LTR (state 7-8-9). After that, it is cooled until the saturated phase is obtained by using the LTR and condenser (state 9-10-11). Then, the liquid ammonia-water mixture is pressurized by means of a pump. Upon the completion of the pumping process, the pressurized liquid ammonia-water mixture is then heated while in passing through LTR and HTR, respectively. Finally, the ammonia-water mixture enters the generator to

complete the cycle. The effectiveness of the generator, HTR, LTR and KCON (ε_{HE}^f) and the isentropic efficiencies of the pump (η_{KP}) and the turbine (η_{KT}) are taken to be as 0.85, 0.85 and 0.92, respectively by following the common usage in the literature [123-132]. Additional assumptions made with regard to the system operation are as follows: the system operates steadily, the kinetic and potential energy changes are negligible, and finally, the pressure losses taking place in the flow of substance through the pipes and at the instance of the flow through the heat exchangers are negligible.

4.4.2 Thermodynamic Analysis

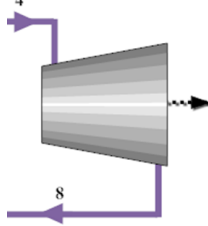
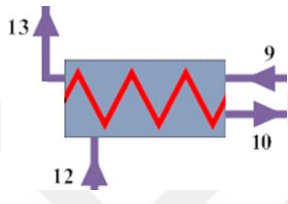
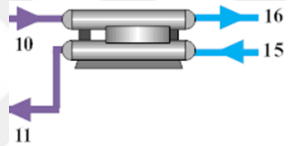
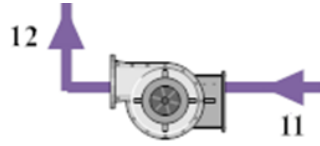
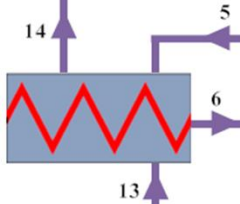
The heat transfer rates, the work, the exergy destructions and the exergy efficiencies of all sub-components are computed by means of the relevant governing equations obtained from the application of continuity, energy and exergy equations, which are summarized in Table 4.19. Energy and exergy efficiencies of the Kalina cycle are calculated by means of the following equations expressed below:

$$\eta_{KC} = (\dot{W}_{KT} - \dot{W}_{KP}) / \dot{m}_{exh} (h_1 - h_2) \quad (4.5)$$

$$\varepsilon_{KC} = (\dot{W}_{KT} - \dot{W}_{KP}) / \dot{m}_{exh} (\psi_1 - \psi_2) \quad (4.6)$$

Table 4.19. Energy and exergy equations of the subcomponents of the Kalina cycle

	$\begin{aligned} \dot{m}_1 &= \dot{m}_2, \dot{m}_{14} = \dot{m}_3 \\ \dot{m}_1(h_1 - h_2) &= \dot{m}_{14}(h_3 - h_{14}) \\ \dot{E}x_{D,GEN} &= \dot{m}_1(\psi_1 - \psi_2) - \dot{m}_{14}(\psi_3 - \psi_{14}) \\ \varepsilon_{GEN} &= \frac{\dot{m}_1(\psi_1 - \psi_2)}{\dot{m}_{14}(\psi_3 - \psi_{14})} \end{aligned}$
	$\begin{aligned} \dot{m}_3 &= \dot{m}_4 + \dot{m}_5 \\ \dot{m}_3 h_3 &= \dot{m}_4 h_4 + \dot{m}_5 h_5 \\ \dot{m}_3 X_3 &= \dot{m}_4 X_4 + \dot{m}_5 X_5 \\ \dot{E}x_{D,SEP} &= \dot{E}x_3 - (\dot{E}x_4 + \dot{E}x_5), \varepsilon_{SEP} = \frac{\dot{E}x_3}{\dot{E}x_4 + \dot{E}x_5} \end{aligned}$

KT		$\dot{m}_4 = \dot{m}_8$ $\dot{W}_{KT,a} = \dot{m}_4(h_4 - h_8)$ $\dot{W}_{KT,s} = \dot{m}_4(\psi_4 - \psi_8)$ $\dot{E}x_{D,KT} = \dot{W}_{KT,s} - \dot{W}_{KT,a}, \varepsilon_{KT} = \frac{\dot{W}_{KT,a}}{\dot{W}_{KT,s}}$
VAL		$\dot{m}_6 = \dot{m}_7$ $\dot{m}_6 h_6 = \dot{m}_7 h_7$ $\dot{E}x_{D,VAL} = \dot{E}x_6 - \dot{E}x_7$ $\varepsilon_{VAL} = \frac{\dot{E}x_6}{\dot{E}x_7}$
LTR		$\dot{m}_9 = \dot{m}_{10}, \dot{m}_{12} = \dot{m}_{13}$ $\dot{m}_9(h_9 - h_{10}) = \dot{m}_{12}(h_{13} - h_{12})$ $\dot{E}x_{D,LTR} = \dot{m}_9(\psi_9 - \psi_{10}) - \dot{m}_{12}(\psi_{13} - \psi_{12})$ $\varepsilon_{LTR} = \frac{\dot{m}_9(\psi_9 - \psi_{10})}{\dot{m}_{12}(\psi_{13} - \psi_{12})}$
KCON		$\dot{m}_{10} = \dot{m}_{11}, \dot{m}_{15} = \dot{m}_{16}$ $\dot{m}_{10}(h_{10} - h_{11}) = \dot{m}_{15}(h_{16} - h_{15})$ $\dot{E}x_{D,KCON} = \dot{m}_{10}(\psi_{10} - \psi_{11}) - \dot{m}_{15}(\psi_{16} - \psi_{15})$ $\varepsilon_{KCON} = \frac{\dot{m}_{10}(\psi_{10} - \psi_{11})}{\dot{m}_{15}(\psi_{16} - \psi_{15})}$
KP		$\dot{m}_{11} = \dot{m}_{12}$ $\dot{W}_{KP,a} = \dot{m}_{11}(h_{12} - h_{11})$ $\dot{W}_{KP,s} = \dot{m}_{11}(\psi_{12} - \psi_{11})$ $\dot{E}x_{D,KP} = \dot{W}_{KP,a} - \dot{W}_{KP,s}, \varepsilon_{KP} = \frac{\dot{W}_{KP,s}}{\dot{W}_{KP,a}}$
HTR		$\dot{m}_5 = \dot{m}_6, \dot{m}_{13} = \dot{m}_{14}$ $\dot{m}_5(h_5 - h_6) = \dot{m}_{13}(h_{14} - h_{13})$ $\dot{E}x_{D,HTR} = \dot{m}_5(\psi_5 - \psi_6) - \dot{m}_{13}(\psi_{14} - \psi_{13})$ $\varepsilon_{HTR} = \frac{\dot{m}_5(\psi_5 - \psi_6)}{\dot{m}_{13}(\psi_{14} - \psi_{13})}$

The thermodynamic properties of the substances needed for calculations at certain states are presented in Table 4.20. The specific flow exergy values of ammonia-water are much greater than those of other substances in Kalina cycle due to their chemical

exergy values. The relevant equations are already given in the previous chapter, see Eqs. (3.11-3.18).

Table 4.20. Thermodynamic properties of the states in the Kalina cycle

State	Fluid	T (°C)	P (bar)	$\dot{m}$ (kg/s)	X	h (kJ/kg)	s (kJ/kgK)	ψ (kJ/kg)	$\dot{E}_x$ (kW)
1	Exhaust	566.7	1.9	16	-	866.2	6.591	306.4	4903
2	Exhaust	80	1.9	16	-	353.9	5.686	59.43	950.8
3	NH ₃ H ₂ O	400	50	4.385	0.8	2442	6.09	17079	74887
4	NH ₃ H ₂ O	400	50	3.067	0.93	2320	5.845	19726	60503
5	NH ₃ H ₂ O	400	50	1.318	0.4974	2726	6.48	11058	14569
6	NH ₃ H ₂ O	62.17	50	1.318	0.4974	42.73	0.7395	10058	13252
7	NH ₃ H ₂ O	62.64	16.67	1.318	0.4974	42.73	0.752	10054	13247
8	NH ₃ H ₂ O	286.9	16.67	3.067	0.93	2002	5.845	19382	59447
9	NH ₃ H ₂ O	123.8	16.67	4.385	0.80	1413	4.459	16528	72471
10	NH ₃ H ₂ O	47.17	16.67	4.385	0.80	82.14	0.6613	16311	71517
11	NH ₃ H ₂ O	36.34	16.67	4.385	0.80	30.38	0.4969	16309	71510
12	NH ₃ H ₂ O	37.17	50	4.385	0.80	35.25	0.4969	16312	71523
13	NH ₃ H ₂ O	52.17	16.67	4.385	0.80	217.2	1.077	16324	71576
14	NH ₃ H ₂ O	77.17	16.67	4.385	0.80	852.9	2.969	16405	71931
15	Water	20	1	9.223	-	83.93	0.2962	0	0
16	Water	25	1	9.223	-	104.8	0.3669	0.1764	1.627

The thermodynamic analysis of the Kalina cycle is accomplished by using the equations given in Table 4.19. The results obtained from the thermodynamic analysis of the each component of the Kalina cycle are shown in Table 4.21.

Table 4.21. Thermodynamic results of the Kalina cycle

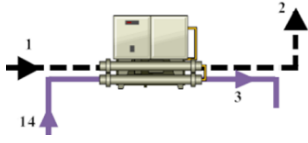
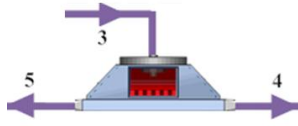
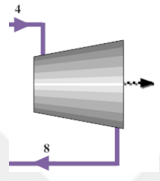
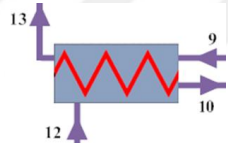
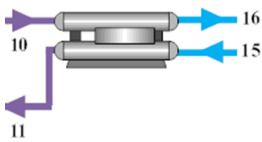
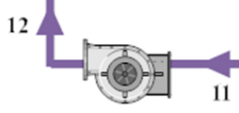
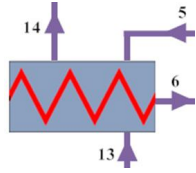
Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}_{x_F}$ (kW)	$\dot{E}_{x_P}$ (kW)	$\dot{E}_{x_D}$ (kW)	ε (%)
GEN	8197	-	3952	2956	996.4	74.79
KT	-	976	1056	976	80.17	92.41
LTR	5836	-	954.2	52.72	901.5	5.525
KCON	227	-	7.749	1.627	6.123	20.99
KP	-	21.4	21.36	13.65	7.711	63.89
HTR	3535	-	1049	355	693.6	33.85
KC	Energy efficiency (%)					11.65
	Exergy efficiency (%)					24.15

The thermodynamic analysis of Kalina cycle has shown that, the net power output, the energy and the exergy efficiencies are 954.6 kW, 11.65 % and 24.15 %, respectively. Efficiencies obtained from the analysis of the Kalina cycle are found to be compatible when compared with the values quoted in other studies on Kalina cycles available in the literature [123-132]. However, the energy and the exergy efficiencies of the Kalina cycle can be improved by reducing the exergy destructions. It is clearly seen from the Table 4.21 that the maximum exergy destructions are taking place at the GEN, LTR and HTR, respectively. The reason for that may be the high thermal losses taking place in the heat exchangers which can potentially cause high levels of exergy destruction to occur in all heat exchangers. It is possible to decrease the exergy destruction in heat exchangers by using optimum heat exchanger design. In contrary to this, the exergy destruction of the condenser seems to be lower when compared to other heat exchangers. But, at the same time the condenser has also very low exergy efficiency. This means that low exergy destruction does not show always higher exergy efficiency. Exergy efficiencies at the separator and valve are seen to be relatively higher which may be tied to the use of the adiabatic process assumption in their analyses. Finally, in the cycle, the exergy efficiencies of the KT and the KP are seen to be very high due to the small amount of the exergy destruction of them.

4.4.3 Thermoeconomic Analysis

The same procedure applied to the GT and ORC systems is utilized here too, to perform thermoeconomic analysis on the Kalina cycle. The cost balance and the auxiliary equations are obtained for each sub-component of the system by applying the SPECO procedure. The resulting equations are given in Table 4.22.

Table 4.22. Exergy cost rate balances and auxiliary equations for sub-components of the Kalina cycle

GEN		$c_1 \dot{E}x_1 + c_{14} \dot{E}x_{14} + \dot{Z}_{GEN} = c_2 \dot{E}x_2 + c_3 \dot{E}x_3$ $c_1 = \text{known}, \frac{\dot{C}_1}{\dot{E}x_1} = \frac{\dot{C}_2}{\dot{E}x_2}, \frac{\dot{C}_{14}}{\dot{E}x_{14}} = \frac{\dot{C}_3}{\dot{E}x_3}$
SEP		$c_3 \dot{E}x_3 + \dot{Z}_{SEP} = c_4 \dot{E}x_4 + c_5 \dot{E}x_5$ $\frac{\dot{C}_4}{\dot{E}x_4} = \frac{\dot{C}_5}{\dot{E}x_5}$
KT		$c_4 \dot{E}x_4 + \dot{Z}_{KT} = c_{KT} \dot{W}_{KT} + c_8 \dot{E}x_8$ $\frac{\dot{C}_4}{\dot{E}x_4} = \frac{\dot{C}_8}{\dot{E}x_8}$
VAL		$c_6 \dot{E}x_6 + \dot{Z}_{VAL} = c_7 \dot{E}x_7$
LTR		$c_9 \dot{E}x_9 + c_{12} \dot{E}x_{12} + \dot{Z}_{LTR} = c_{10} \dot{E}x_{10} + c_{13} \dot{E}x_{13}$ $\frac{\dot{C}_9}{\dot{E}x_9} = \frac{\dot{C}_{10}}{\dot{E}x_{10}}$
KCON		$c_{10} \dot{E}x_{10} + c_{15} \dot{E}x_{15} + \dot{Z}_{KCON} = c_{11} \dot{E}x_{11} + c_{16} \dot{E}x_{16}$ $c_{15} = 0, \frac{\dot{C}_{10}}{\dot{E}x_{10}} = \frac{\dot{C}_{11}}{\dot{E}x_{11}}$
KP		$c_{KP} \dot{W}_{KP} + \dot{Z}_{KP} = c_{12} \dot{E}x_{12} - c_{11} \dot{E}x_{11}$ $\frac{\dot{C}_{11}}{\dot{E}x_{11}} = \frac{\dot{C}_{12}}{\dot{E}x_{12}}$
HTR		$c_5 \dot{E}x_5 + c_{13} \dot{E}x_{13} + \dot{Z}_{HTR} = c_6 \dot{E}x_6 + c_{14} \dot{E}x_{14}$ $\frac{\dot{C}_{13}}{\dot{E}x_{13}} = \frac{\dot{C}_{14}}{\dot{E}x_{14}}$

The capital cost rates of the components are calculated by using the Eq. (3.30). In order to perform thermoeconomic analysis on Kalina cycle, the economic constants are taken to be the same as in the case of GT cycle, see Table 4.5. The capital costs are calculated by using the thermodynamic relations belonging to each component. The capital investment cost function of each component is given in Table 4.23 [129].

Table 4.23. Capital investment cost functions of the sub-components of the Kalina cycle

GEN	$PEC_{GEN} = 2143(A_{GEN})^{0.514}$
KT	$PEC_{KT} = 4405(\dot{W}_{KT})^{0.7}$
LTR	$PEC_{LTR} = 2143(A_{LTR})^{0.514}$
KCON	$PEC_{KCON} = 2143(A_{KCON})^{0.514}$
KP	$PEC_{KP} = 1120(\dot{W}_{KP})^{0.8}$
HTR	$PEC_{HTR} = 2143(A_{HTR})^{0.514}$

The heat transfer coefficient of GEN, LTR, HTR and KCON are assumed to be 0.3, 0.7, 0.7 and 0.8, respectively [129]. The total capital investment costs of the valve and separator are so small that they are not taken into account in the analyses [129]. The equations given in Table 4.23 are used to calculate the capital investment of the Kalina cycle. Then, the operating and the maintenance costs are calculated by using the chemical engineering plant cost index by using the equations which are given Table 4.5. The exergy flow rates, the cost flow rates and the unit exergy costs of the Kalina cycle are tabulated as shown in Table 4.24.

Table 4.24. Exergy flow rates, cost flow rates and unit exergy costs of the GT cycle

State	$\dot{E}_x$ (kW)	c (\$/GJ)	$\dot{C}$ (\$/h)
1	4903	3.21	56.66
2	950.8	3.21	10.99
3	74887	4.457	1202
4	60503	4.462	971.9
5	14301	4.462	229.7
6	13252	4.711	224.8
7	13247	4.713	224.8
8	59447	4.462	954.9
9	72471	4.395	1147
10	71517	4.395	1132
11	71510	4.395	1131
12	71523	4.395	1132
13	71576	4.457	1148
14	71931	4.457	1154
15	0	0	0
16	1.627	90.45	0.5297

The total capital cost rate and the total cost rate of the Kalina cycle are found to be 15.36 \$/h and 62.79 \$/h, respectively. Although the low exergy destruction is taking place at KP, it has the highest exergoeconomic factor and the relative cost difference values in comparison with other components existing in the system, see Table 4.25. This situation can be best explained by the high difference between the unit exergy cost of the product and fuel of KP. The result of exergoeconomic factor of KP being higher may be attributed to the fact that it has the lowest total capital cost rate among the others. The KCON has also higher relative cost difference and low exergoeconomic factor which is related with high product and fuel difference and low total capital cost rate like as in the case of KP. Excluding the case for KCON, It is observed that all heat exchangers have lower exergoeconomic factors because of their high exergy destruction costs when compared with other components of the system. It may be clear from the foregoing discussions that a special attention should be paid on these components to increase the total system performance. Other high exergoeconomic factor belongs to KT due to its low exergy destruction when compared to other components

Table 4.25. Thermoeconomic results of the Kalina cycle

Component	$\dot{E}x_D$ (kW)	c_f (\$/GJ)	c_p (\$/GJ)	$\dot{D}_D$ (\$/h)	$\dot{Z}$ (\$/h)	r (%)	f (%)
GEN	996.4	3.21	4.457	11.51	1.755	38.85	13.23
KT	80.17	4.462	7.944	1.288	10.95	43.83	89.47
LTR	901.5	4.395	4.457	14.26	1.676	1.407	10.52
KCON	6.123	4.395	90.45	0.09687	0.4071	95.14	80.78
KP	7.711	0.5784	4.395	0.01606	0.2604	86.84	94.19
HTR	693.6	4.457	4.711	11.13	0.7526	5.4	6.334

4.5 Model 4: Combined Cycle of GT and ORC (GT-ORC)

4.5.1 System Description

The 4th model is designed by combining the GT and ORC cycles. Herein, the waste heat used for gas turbine cycle from the actual power plant is used as the heat source for the ORC. Thus, the exhaust gas can be used efficiently as much as it is possible. The combined cycle consists of a gas turbine heat exchanger (GTHE), a gas turbine (GT), a recuperator (REC), a low pressure compressor (LPC), a high pressure compressor (HPC), a precooler (PRE), an intercooler (INT), an evaporator (EVAP), an ORC turbine (OT), a condenser (CON) and an ORC pump (OP). The schematic view of the GT-ORC system is shown in Fig. 4.5.

The GT cycle and ORC are explained in the Models 1 and 2, respectively. $LMTD_{GTHE}$ and PR_{GT} are taken differently as 9.6 and 2.8, respectively than Model 1 in order to achieve better thermodynamic performance. Also the working fluid used in the ORC is changed to R245fa. The working fluids used in Model 2 are ones with high decomposition temperatures due to the high temperature of the exhaust gas. But, in Model 4 it is not required to have high decomposition temperature working fluid. A different working fluid is used because the temperature of the exhaust gas is less than the Model 2 at the ORC inlet. R245fa is the most preferred working fluid in the

current literature and is used because of the efficient one among others [118, 120, 122]. The properties of R245fa is given in Table 4.26.

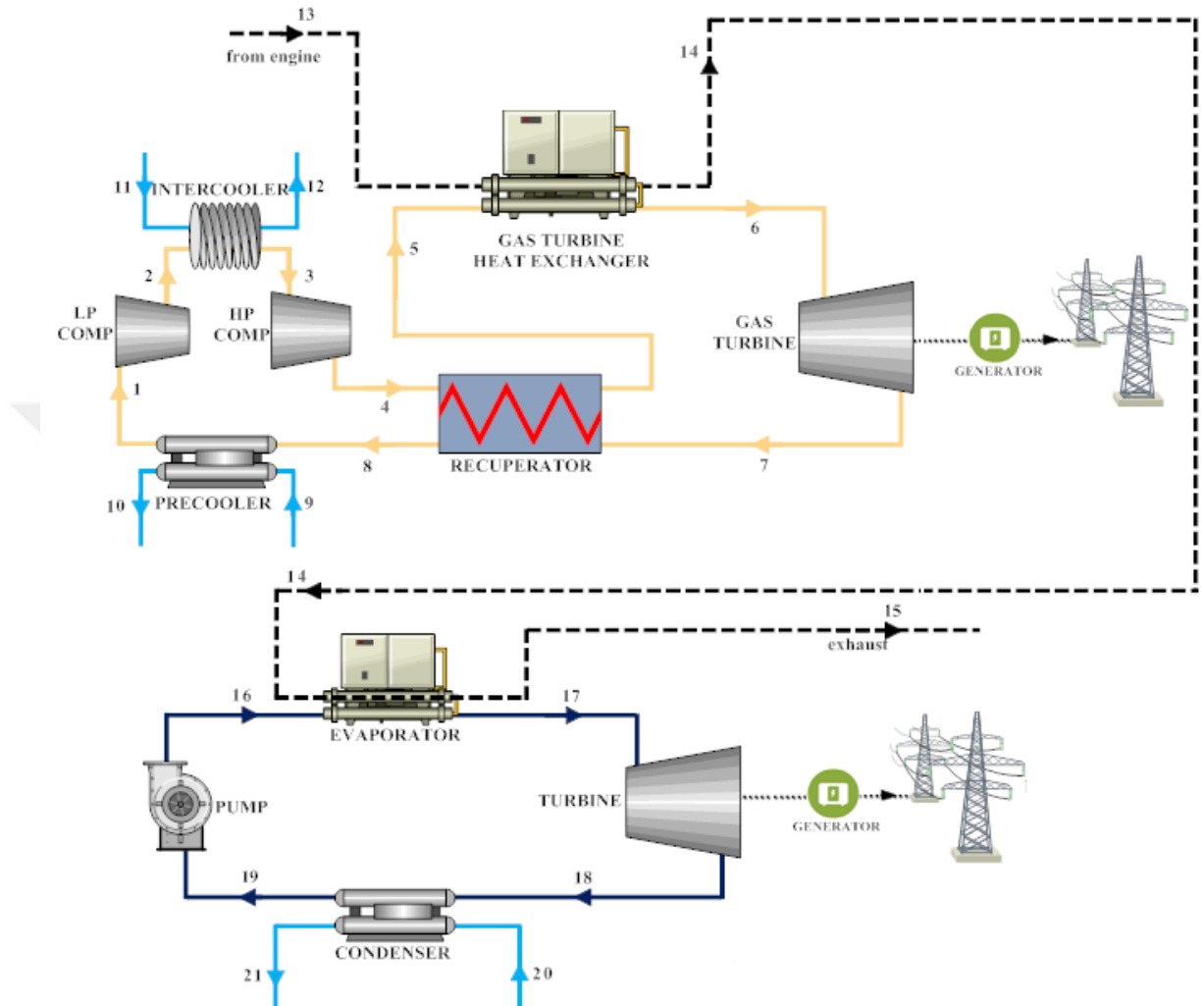


Figure 4.5. Schematic layout of the GT-ORC

Table 4.26. Properties of the R245fa

Fluid	Mol.Weight (kg/kmol)	P_{crit} (bar)	T_{crit} (°C)	Fluid type
R245fa	134.05	36.4	154.1	Isentropic

4.5.2 Thermodynamic Analysis

The heat transfer rates, the work, the exergy destructions and the exergy efficiencies of all sub-components are evaluated by means of the governing equations related with continuity, energy and exergy equations are given in Table 4.27. Thermodynamic properties at each state in the system are tabulated and given in Table 4.28. Energy and exergy efficiencies of the GT-ORC are calculated by means of the equations given below:

$$\eta_{GT-ORC} = ((\dot{W}_{OT} - \dot{W}_{OP}) + (\dot{W}_{GT} - (\dot{W}_{LPC} + \dot{W}_{HPC}))) / \dot{m}_{exh} (h_{13} - h_{15}) \quad (4.7)$$

$$\varepsilon_{GT-ORC} = ((\dot{W}_{OT} - \dot{W}_{OP}) + (\dot{W}_{GT} - (\dot{W}_{LPC} + \dot{W}_{HPC}))) / \dot{m}_{exh} (\psi_{13} - \psi_{15}) \quad (4.8)$$

Table 4.27. Energy and exergy equations of the subcomponents of the GT-ORC

GTHE		$\dot{m}_{13} = \dot{m}_{14} = \dot{m}_{exh}, \dot{m}_5 = \dot{m}_6 = \dot{m}_{CO_2}$ $\dot{m}_{exh} (h_{13} - h_{14}) = \dot{m}_{CO_2} (h_6 - h_5)$ $\dot{E}x_{D,GTHE} = \dot{m}_{exh} (\psi_{13} - \psi_{14}) - \dot{m}_{CO_2} (\psi_6 - \psi_5)$ $\varepsilon_{GTHE} = \frac{\dot{m}_{exh} (\psi_{13} - \psi_{14})}{\dot{m}_{CO_2} (\psi_6 - \psi_5)}$
GT		$\dot{m}_{115} = \dot{m}_{116} = \dot{m}_{CO_2}$ $\dot{W}_{GT,a} = \dot{m}_{CO_2} (h_6 - h_7)$ $\dot{W}_{GT,s} = \dot{m}_{CO_2} (\psi_6 - \psi_7)$ $\dot{E}x_{D,GT} = \dot{W}_{GT,s} - \dot{W}_{GT,a}, \varepsilon_{GT} = \frac{\dot{W}_{GT,a}}{\dot{W}_{GT,s}}$
REC		$\dot{m}_4 = \dot{m}_5 = \dot{m}_7 = \dot{m}_8 = \dot{m}_{CO_2}$ $(h_4 - h_5) = (h_8 - h_7)$ $\dot{E}x_{D,REC} = \dot{m}_{CO_2} ((\psi_4 - \psi_5) - (\psi_8 - \psi_7))$ $\varepsilon_{REC} = \frac{(\psi_4 - \psi_5)}{(\psi_8 - \psi_7)}$
PRE		$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{wat}, \dot{m}_1 = \dot{m}_8 = \dot{m}_{CO_2}$ $\dot{m}_{CO_2} (h_8 - h_1) = \dot{m}_{wat} (h_{10} - h_9)$ $\dot{E}x_{D,PRE} = \dot{m}_{CO_2} (\psi_8 - \psi_1) - \dot{m}_{wat} (\psi_{10} - \psi_9)$ $\varepsilon_{PRE} = \frac{\dot{m}_{CO_2} (\psi_8 - \psi_1)}{\dot{m}_{wat} (\psi_{10} - \psi_9)}$

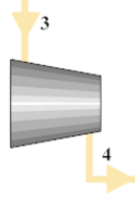
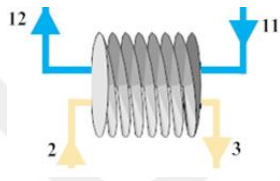
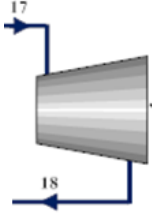
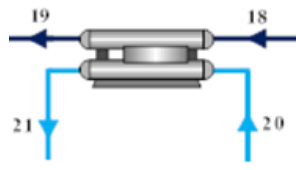
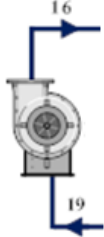
LPC		$\dot{m}_1 = \dot{m}_2 = \dot{m}_{CO_2}$ $\dot{W}_{LPC,a} = \dot{m}_{CO_2} (h_2 - h_1)$ $\dot{W}_{LPC,s} = \dot{m}_{CO_2} (\psi_2 - \psi_1)$ $\dot{E}x_{D,LPC} = \dot{W}_{LPC,a} - \dot{W}_{LPC,s}, \varepsilon_{LPC} = \frac{\dot{W}_{LPC,s}}{\dot{W}_{LPC,a}}$
HPC		$\dot{m}_3 = \dot{m}_4 = \dot{m}_{CO_2}$ $\dot{W}_{HPC,a} = \dot{m}_{CO_2} (h_4 - h_3)$ $\dot{W}_{HPC,s} = \dot{m}_{CO_2} (\psi_4 - \psi_3)$ $\dot{E}x_{D,HPC} = \dot{W}_{HPC,a} - \dot{W}_{HPC,s}, \varepsilon_{HPC} = \frac{\dot{W}_{HPC,s}}{\dot{W}_{HPC,a}}$
INT		$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_{wat}, \dot{m}_1 = \dot{m}_8 = \dot{m}_{CO_2}$ $\dot{m}_{CO_2} (h_2 - h_3) = \dot{m}_{wat} (h_{12} - h_{11})$ $\dot{E}x_{D,INT} = \dot{m}_{CO_2} (\psi_2 - \psi_3) - \dot{m}_{wat} (\psi_{12} - \psi_{11})$ $\varepsilon_{INT} = \frac{\dot{m}_{CO_2} (\psi_2 - \psi_3)}{\dot{m}_{wat} (\psi_{11} - \psi_{12})}$
EVAP		$\dot{m}_{14} = \dot{m}_{15} = \dot{m}_{exh}, \dot{m}_{16} = \dot{m}_{17} = \dot{m}_{wf}$ $\dot{m}_{exh} (h_{14} - h_{15}) = \dot{m}_{wf} (h_{17} - h_{16})$ $\dot{E}x_{D,EVAP} = \dot{m}_{exh} (\psi_{14} - \psi_{15}) - \dot{m}_{wf} (\psi_{17} - \psi_{16})$ $\varepsilon_{EVAP} = \frac{\dot{m}_{exh} (\psi_{14} - \psi_{15})}{\dot{m}_{wf} (\psi_{17} - \psi_{16})}$
OT		$\dot{m}_{17} = \dot{m}_{18} = \dot{m}_{wf}$ $\dot{W}_{OT,a} = \dot{m}_{wf} (h_{17} - h_{18})$ $\dot{W}_{OT,s} = \dot{m}_{wf} (\psi_{17} - \psi_{18})$ $\dot{E}x_{D,OT} = \dot{W}_{OT,s} - \dot{W}_{OT,a}, \varepsilon_{OT} = \frac{\dot{W}_{OT,a}}{\dot{W}_{OT,s}}$
CON		$\dot{m}_{18} = \dot{m}_{19} = \dot{m}_{wf}, \dot{m}_{20} = \dot{m}_{21} = \dot{m}_{wat}$ $\dot{m}_{wf} (h_{18} - h_{19}) = \dot{m}_{wat} (h_{21} - h_{20})$ $\dot{E}x_{D,CON} = \dot{m}_{wf} (\psi_{18} - \psi_{19}) - \dot{m}_{wat} (\psi_{21} - \psi_{20})$ $\varepsilon_{CON} = \frac{(\psi_{18} - \psi_{19})}{(\psi_{21} - \psi_{20})}$
OP		$\dot{m}_{19} = \dot{m}_{16} = \dot{m}_{wf}$ $\dot{W}_{OP,a} = \dot{m}_{wf} (h_{16} - h_{19})$ $\dot{W}_{OP,s} = \dot{m}_{wf} (\psi_{16} - \psi_{19})$ $\dot{E}x_{D,OP} = \dot{W}_{OP,a} - \dot{W}_{OP,s}, \varepsilon_{pump} = \frac{\dot{W}_{OP,s}}{\dot{W}_{OP,a}}$

Table 4.28. Thermodynamic properties of states of the GT-ORC

State	Fluid	T (°C)	P (bar)	$\dot{m}$ (kg/s)	h (kJ/kg)	s (kJ/kg.K)	ψ (kJ/kg)	$\dot{E}_x$ (kW)
1	CO ₂	39	74	11.07	-87.87	-1.022	212.9	2356
2	CO ₂	78.82	123.8	11.07	-69.84	-1.022	231.4	2561
3	CO ₂	39	123.8	11.07	-215.8	-1.464	214.5	2374
4	CO ₂	52.1	207.2	11.07	-204.9	-1.464	225.6	2497
5	CO ₂	375.9	207.2	11.07	313.2	-0.2984	401.7	4446
6	CO ₂	547.7	207.2	11.07	525	-0.00895	528.7	5851
7	CO ₂	423.4	74	11.07	376.7	-0.00895	387.3	4286
8	CO ₂	71.1	74	11.07	-22.42	-0.8215	219.5	2429
9	Water	20	1	10.51	83.93	0.2962	0	0
10	Water	34	1	10.51	142.5	0.4913	1.355	14.25
11	Water	20	1	15.25	83.93	0.2962	0	0
12	Water	42	1	15.25	175.9	0.5989	3.289	50.15
13	Exhaust	566.7	1.9	16	866.2	6.591	306.4	4903
14	Exhaust	380	1.9	16	663.5	6.318	183.7	2939
15	Exhaust	90	1.9	16	364	5.714	61.26	980.2
16	R245fa	84.96	42.09	27.7	316.3	1.36	15.33	424.6
17	R245fa	139	42.09	27.7	489.3	1.798	59.83	1658
18	R245fa	90.01	8.417	27.7	472.4	1.806	40.48	1121
19	R245fa	82.5	8.417	27.7	313	1.358	12.35	342.3
20	Water	20	1	40.79	83.93	0.2962	0	0
21	Water	42	1	40.79	175.9	0.5989	3.289	134.2

Thermodynamic analysis of the GT-ORC system is carried out by making use of the governing equations quoted in Table 4.27 and the thermodynamic properties listed in Table 4.28. The results emerging from the application of the thermodynamic analyses on all components of the GT-ORC are given in Table 4.29.

Table 4.29. Thermodynamic results of the GT-ORC

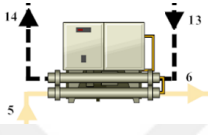
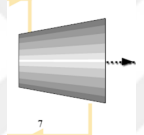
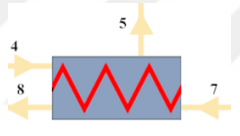
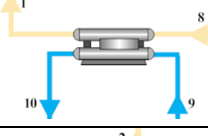
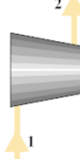
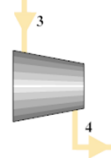
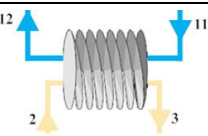
Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}x_F$ (kW)	$\dot{E}x_P$ (kW)	$\dot{E}x_D$ (kW)	ε (%)
GTHE	2758	-	1964	1405	559.3	71.52
GT	-	1509	1565	1509	55.65	96.44
REC	5734	-	1949	1857	92.05	95.28
PRE	724.3	-	72.89	14.25	58.64	19.55
LPC	-	234.7	234.7	205.3	29.37	87.48
HPC	-	142.2	142.2	123	19.25	86.46
INT	1403	-	187.3	50.15	137.1	26.78
EVAP	4791	-	1959	1233	725.8	62.94
OT	-	469.1	536.2	469.1	67.06	87.49
CON	4406	-	779.1	134.2	644.9	17.22
OP	-	93.83	93.83	82.3	11.53	87.72
GT-ORC	Energy efficiency (%)				18.76	
	Exergy efficiency (%)				38.43	

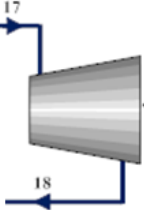
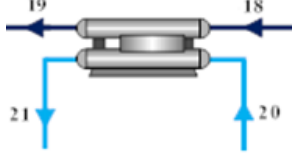
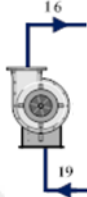
As a result of the thermodynamic analysis carried out for GT-ORC, the net power output, the energy and the exergy efficiencies are found to be 1508 kW, 18.76 % and 38.43 %, respectively. However, the energy and the exergy efficiencies of the GT-ORC can be improved by decreasing the exergy destructions at the components in which the destructions are relatively higher. It is clearly seen that from Table 4.29 that the maximum exergy destructions occur at the EVAP, CON and GTHE, respectively. Due to the thermal losses in heat exchangers, high amount of exergy destruction is expected for all kinds of heat exchangers. It is possible to decrease the exergy destruction in heat exchangers by using optimum heat exchanger design. The highest exergy efficiency is seen for the GT because of its negligibly small exergy destruction rate. In addition to this, in GT-ORC, the exergy efficiency of the REC is also seen to be very high due to its small amount of exergy destruction rate.

4.5.3 Thermoeconomic Analysis

Thermoeconomic analysis on GT-ORC is performed by using the same procedures as applied to the other models analyzed before and the details of which are already weel-outlined. Cost balance and auxiliary equations are obtained for each sub-component of the system. The related equations are given in Table 4.30.

Table 4.30. Exergy cost rate balances and auxiliary equations for sub-components of GT-ORC

GTHE		$c_{13}\dot{E}x_{13} + c_5\dot{E}x_5 + \dot{Z}_{GTHE} = c_{14}\dot{E}x_{14} + c_6\dot{E}x_6$ $c_{13} = \text{known}, \frac{\dot{C}_{13}}{\dot{E}x_{13}} = \frac{\dot{C}_{14}}{\dot{E}x_{14}}$
GT		$c_6\dot{E}x_6 + \dot{Z}_{GT} = c_{GT}\dot{W}_{GT} + c_7\dot{E}x_7$ $\frac{\dot{C}_6}{\dot{E}x_6} = \frac{\dot{C}_7}{\dot{E}x_7}$
REC		$c_4\dot{E}x_4 + c_7\dot{E}x_7 + \dot{Z}_{REC} = c_5\dot{E}x_5 + c_8\dot{E}x_8$ $\frac{\dot{C}_7}{\dot{E}x_7} = \frac{\dot{C}_8}{\dot{E}x_8}$
PRE		$c_8\dot{E}x_8 + c_9\dot{E}x_9 + \dot{Z}_{PRE} = c_1\dot{E}x_1 + c_{10}\dot{E}x_{10}$ $c_9 = 0, \frac{\dot{C}_1}{\dot{E}x_1} = \frac{\dot{C}_8}{\dot{E}x_8}$
LPC		$c_{LPC}\dot{W}_{LPC} + \dot{Z}_{LPC} = c_2\dot{E}x_2 - c_1\dot{E}x_1$ $\frac{\dot{C}_1}{\dot{E}x_1} = \frac{\dot{C}_2}{\dot{E}x_2}$
HPC		$c_{HPC}\dot{W}_{HPC} + \dot{Z}_{HPC} = c_4\dot{E}x_4 - c_3\dot{E}x_3$
INT		$c_2\dot{E}x_2 + c_{11}\dot{E}x_{11} + \dot{Z}_{INT} = c_3\dot{E}x_3 + c_{12}\dot{E}x_{12}$ $c_{11} = 0, \frac{\dot{C}_2}{\dot{E}x_2} = \frac{\dot{C}_3}{\dot{E}x_3}$

EVAP		$c_{14}\dot{E}x_{14} + c_{16}\dot{E}x_{16} + \dot{Z}_{EVAP} = c_{15}\dot{E}x_{15} + c_{17}\dot{E}x_{17}$ $\frac{\dot{C}_{14}}{\dot{E}x_{14}} = \frac{\dot{C}_{15}}{\dot{E}x_{15}}$
OT		$c_{17}\dot{E}x_{17} + \dot{Z}_{OT} = c_{OT}\dot{W}_{OT} + c_{18}\dot{E}x_{18}$ $\frac{\dot{C}_{17}}{\dot{E}x_{17}} = \frac{\dot{C}_{18}}{\dot{E}x_{18}}$
CON		$c_{18}\dot{E}x_{18} + c_{20}\dot{E}x_{20} + \dot{Z}_{CON} = c_{19}\dot{E}x_{19} + c_{21}\dot{E}x_{21}$ $c_{20} = 0, \frac{\dot{C}_{18}}{\dot{E}x_{18}} = \frac{\dot{C}_{19}}{\dot{E}x_{19}}$
OP		$c_{OP}\dot{W}_{OP} + \dot{Z}_{OP} = c_{16}\dot{E}x_{16} - c_{19}\dot{E}x_{19}$ $\frac{\dot{C}_{16}}{\dot{E}x_{16}} = \frac{\dot{C}_{19}}{\dot{E}x_{19}}$

The capital cost rate of the components are calculated by using the Eq. (3.30) in GT-ORC system. For performing the thermoeconomic analysis of GT-ORC, the economic constants are taken to be same as in GT and ORC which are given in Table 4.5. The capital costs are calculated by using the thermodynamic relations of the each component. The capital investment cost function of each component of GT and ORC is given in Table 4.6 and Table 4.17, respectively [97, 129]. The heat transfer coefficients of the heat exchangers are already given in the related sections. The exergy flow rates, the cost flow rates and the unit exergy costs of the GT-ORC are given in Table 4.31.

Table 4.31. Exergy flow rates, cost flow rates and unit exergy costs of the GT-ORC

State	$\dot{E}_x$ (kW)	c (\$/GJ)	$\dot{C}$ (\$/h)
1	2356	4.797	40.68
2	2561	4.797	44.23
3	2374	4.797	41
4	2497	4.828	43.4
5	4446	4.805	76.91
6	5851	4.797	101
7	4286	4.797	74.01
8	2429	4.797	41.94
9	0	0	0
10	14.25	27.72	1.422
11	0	0	0
12	50.15	19.18	3.464
13	4903	3.21	56.66
14	2939	3.21	33.96
15	980.2	3.21	11.33
16	424.6	7.287	11.14
17	1658	5.875	35.05
18	1121	5.875	23.72
19	342.3	5.875	7.239
20	0	0	0
21	134.2	36.15	17.46

The total capital cost rate and the total cost rate of the GT-ORC are found to be 45.36 \$/h and 90.69 \$/h, respectively. The thermoeconomic results of GT-ORC are shown in Table 4.32. GT and OT have the highest exergoeconomic factors due to their higher thermodynamic performance. But all heat exchangers have lower exergoeconomic factor values because of their high exergy destruction costs when compared to other components of the system. It can be seen from Table 4.32 that CON, EVAP and GTHE have the highest cost rates of destruction because of their high exergy destruction rate and capital cost rates among the other heat exchangers. Special attention can be focused on these components in order to increase the total system performance. Although the REC has higher capital cost rate, it has an average exergoeconomic factor value which can be explained with its lower exergy destruction and higher capital cost rate. The lowest exergoeconomic factor is found to be 6.744 for the condenser. This result can be explained by the fact that the condenser has both high mass flow rate and exergy destruction.

Table 4.32. Thermoeconomic results of the GT-ORC

Component	$\dot{E}x_D$ (kW)	c_f (\$/GJ)	c_p (\$/GJ)	$\dot{D}_D$ (\$/h)	$\dot{Z}$ (\$/h)	r (%)	f (%)
GTHE	559.3	3.21	4.797	6.464	1.426	49.43	18.07
GT	55.65	4.797	4.797	0.9609	14.85	6.17×10^{-8}	93.92
REC	92.05	4.797	4.805	1.59	1.443	0.1753	47.58
PRE	58.64	4.797	4.797	1.013	0.1633	7.80×10^{-8}	13.89
LPC	29.37	3.426	4.797	0.3622	0.6507	40.01	64.24
HPC	19.25	3.426	4.828	0.2374	0.6507	40.92	73.27
INT	137.1	4.797	4.797	2.368	0.2299	6.42×10^{-8}	8.849
EVAP	725.8	3.21	5.875	8.388	1.28	83.01	13.24
OT	67.06	5.875	5.875	1.418	19.65	9.12×10^{-8}	93.27
CON	644.9	5.875	5.875	13.64	0.9863	2.85×10^{-8}	6.744
OP	11.53	5.875	7.287	0.2438	1.787	24.04	88

4.6 Model 5: Combined Cycle of GT and KAL (GT-KAL)

4.6.1 System Description

The last model is constructed from the combination of GT and Kalina cycle. The waste heat used for gas turbine cycle from the actual power plant is used as the heat source for the Kalina cycle. It can be said that the temperature of the exhaust gas is reduced as much as possible. The GT-KAL cycle consists of a gas turbine heat exchanger (GTHE), a gas turbine (GT), a recuperator (REC), a low pressure compressor (LPC), a high pressure compressor (HPC), a precooler (PRE), an intercooler (INT), a generator (GEN), a separator (SEP), a turbine (KT), a valve (VAL), a low temperature recuperator (LTR), a condenser (KCON), a pump (KP) and a high temperature recuperator (HTR). The schematic layout of the GT-KAL cycle is given in Fig. 4.6. $LMTD_{GTHE}$ and PR_{GT} are taken differently as 9.6 and 2.8, respectively than Model 1 in order to achieve better thermodynamic performance.

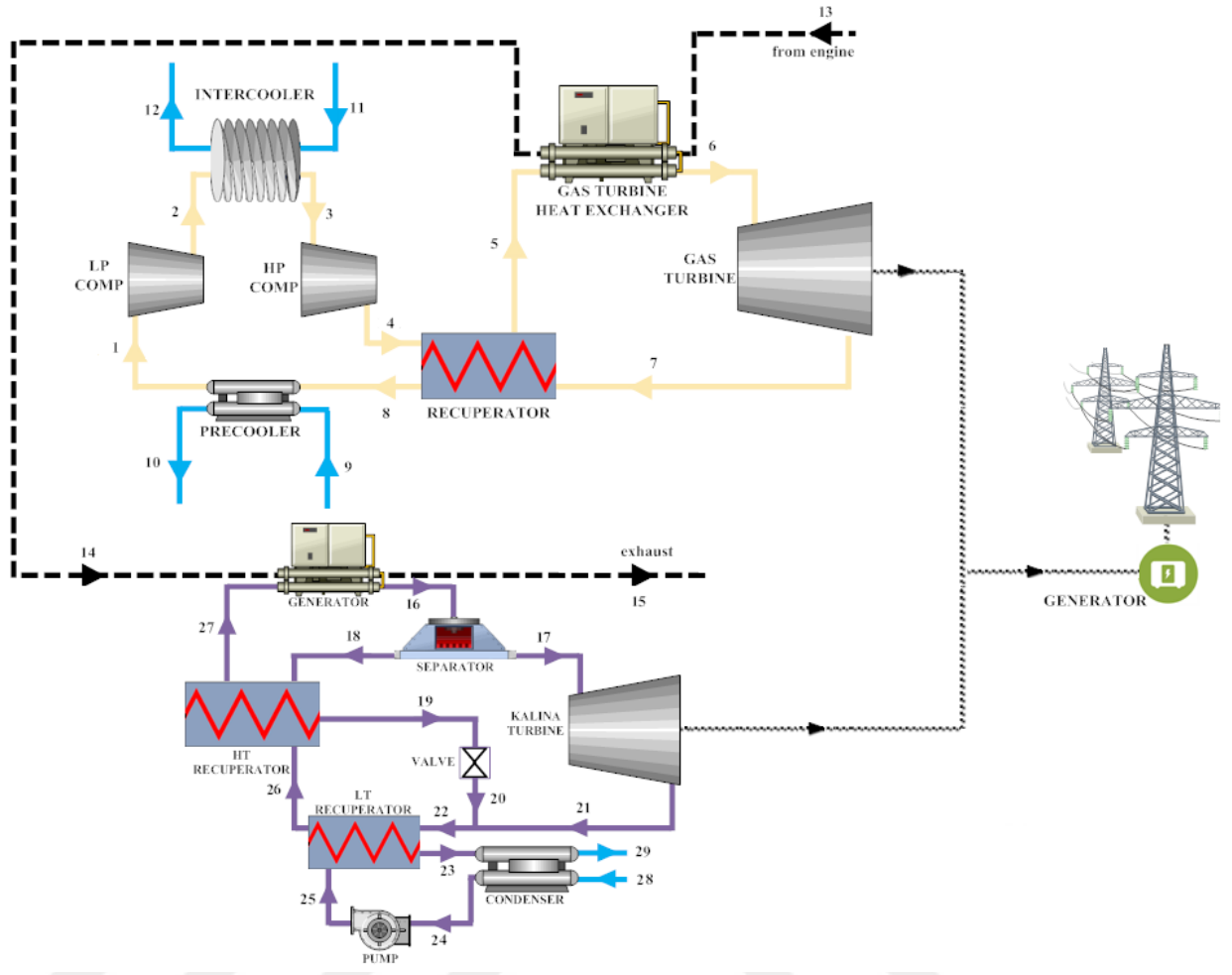


Figure 4.6. Schematic layout of the GT-KAL cycle

4.6.2 Thermodynamic Analysis

The heat transfer rates, the work, the exergy destructions and the exergy efficiencies of all sub-components are evaluated by means of the governing equations related with continuity, energy and exergy equations, which are given in Table 4.33. The thermodynamic properties at each state in the system are tabulated and given in Table 4.34. The energy and the exergy efficiencies of the GT-ORC are calculated by means of the equations given below:

$$\eta_{GT-KAL} = ((\dot{W}_{KT} - \dot{W}_{KP}) + (\dot{W}_{GT} - (\dot{W}_{LPC} + \dot{W}_{HPC}))) / \dot{m}_{exh} (h_{13} - h_{15}) \quad (4.9)$$

$$\varepsilon_{GT-KAL} = ((\dot{W}_{KT} - \dot{W}_{KP}) + (\dot{W}_{GT} - (\dot{W}_{LPC} + \dot{W}_{HPC}))) / \dot{m}_{exh} (\psi_{13} - \psi_{15}) \quad (4.10)$$

Table 4.33. Energy and exergy equations of the subcomponents of the GT-KAL cycle

GTHE		$\dot{m}_{13} = \dot{m}_{14} = \dot{m}_{exh}, \dot{m}_5 = \dot{m}_6 = \dot{m}_{CO_2}$ $\dot{m}_{exh} (h_{13} - h_{14}) = \dot{m}_{CO_2} (h_6 - h_5)$ $\dot{E}x_{D,GTHE} = \dot{m}_{exh} (\psi_{13} - \psi_{14}) - \dot{m}_{CO_2} (\psi_6 - \psi_5)$ $\varepsilon_{GTHE} = \frac{\dot{m}_{exh} (\psi_{13} - \psi_{14})}{\dot{m}_{CO_2} (\psi_6 - \psi_5)}$
GT		$\dot{m}_{115} = \dot{m}_{116} = \dot{m}_{CO_2}$ $\dot{W}_{GT,a} = \dot{m}_{CO_2} (h_6 - h_7)$ $\dot{W}_{GT,s} = \dot{m}_{CO_2} (\psi_6 - \psi_7)$ $\dot{E}x_{D,GT} = \dot{W}_{GT,s} - \dot{W}_{GT,a}, \varepsilon_{GT} = \frac{\dot{W}_{GT,a}}{\dot{W}_{GT,s}}$
REC		$\dot{m}_4 = \dot{m}_5 = \dot{m}_7 = \dot{m}_8 = \dot{m}_{CO_2}$ $(h_4 - h_5) = (h_8 - h_7)$ $\dot{E}x_{D,REC} = \dot{m}_{CO_2} ((\psi_4 - \psi_5) - (\psi_8 - \psi_7))$ $\varepsilon_{REC} = \frac{(\psi_4 - \psi_5)}{(\psi_8 - \psi_7)}$
PRE		$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{wat}, \dot{m}_1 = \dot{m}_8 = \dot{m}_{CO_2}$ $\dot{m}_{CO_2} (h_8 - h_1) = \dot{m}_{wat} (h_{10} - h_9)$ $\dot{E}x_{D,PRE} = \dot{m}_{CO_2} (\psi_8 - \psi_1) - \dot{m}_{wat} (\psi_{10} - \psi_9)$ $\varepsilon_{PRE} = \frac{\dot{m}_{CO_2} (\psi_8 - \psi_1)}{\dot{m}_{wat} (\psi_{10} - \psi_9)}$
LPC		$\dot{m}_1 = \dot{m}_2 = \dot{m}_{CO_2}$ $\dot{W}_{LPC,a} = \dot{m}_{CO_2} (h_2 - h_1)$ $\dot{W}_{LPC,s} = \dot{m}_{CO_2} (\psi_2 - \psi_1)$ $\dot{E}x_{D,LPC} = \dot{W}_{LPC,a} - \dot{W}_{LPC,s}, \varepsilon_{LPC} = \frac{\dot{W}_{LPC,s}}{\dot{W}_{LPC,a}}$
HPC		$\dot{m}_3 = \dot{m}_4 = \dot{m}_{CO_2}$ $\dot{W}_{HPC,a} = \dot{m}_{CO_2} (h_4 - h_3)$ $\dot{W}_{HPC,s} = \dot{m}_{CO_2} (\psi_4 - \psi_3)$ $\dot{E}x_{D,HPC} = \dot{W}_{HPC,a} - \dot{W}_{HPC,s}, \varepsilon_{HPC} = \frac{\dot{W}_{HPC,s}}{\dot{W}_{HPC,a}}$

INT		$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_{wat}, \dot{m}_1 = \dot{m}_8 = \dot{m}_{CO_2}$ $\dot{m}_{CO_2}(h_2 - h_3) = \dot{m}_{wat}(h_{12} - h_{11})$ $\dot{E}x_{D,INT} = \dot{m}_{CO_2}(\psi_2 - \psi_3) - \dot{m}_{wat}(\psi_{12} - \psi_{11})$ $\varepsilon_{INT} = \frac{\dot{m}_{CO_2}(\psi_2 - \psi_3)}{\dot{m}_{wat}(\psi_{11} - \psi_{12})}$
GEN		$\dot{m}_{14} = \dot{m}_{15}, \dot{m}_{27} = \dot{m}_{16}$ $\dot{m}_{14}(h_{14} - h_{15}) = \dot{m}_{27}(h_{16} - h_{27})$ $\dot{E}x_{D,GEN} = \dot{m}_{14}(\psi_{14} - \psi_{15}) - \dot{m}_{27}(\psi_{16} - \psi_{27})$ $\varepsilon_{GEN} = \frac{\dot{m}_{14}(\psi_{14} - \psi_{15})}{\dot{m}_{27}(\psi_{16} - \psi_{27})}$
SEP		$\dot{m}_{16} = \dot{m}_{17} + \dot{m}_{18}$ $\dot{m}_{16}h_{16} = \dot{m}_{18}h_{18} + \dot{m}_{17}h_{17}$ $\dot{m}_{16}X_{16} = \dot{m}_{17}X_{17} + \dot{m}_{18}X_{18}$ $\dot{E}x_{D,SEP} = \dot{E}x_{16} - (\dot{E}x_{17} + \dot{E}x_{18}), \varepsilon_{SEP} = \frac{\dot{E}x_{16}}{\dot{E}x_{17} + \dot{E}x_{18}}$
KT		$\dot{m}_{17} = \dot{m}_{21}$ $\dot{W}_{KT,a} = \dot{m}_{17}(h_{17} - h_{21})$ $\dot{W}_{KT,s} = \dot{m}_{17}(\psi_{17} - \psi_{21})$ $\dot{E}x_{D,KT} = \dot{W}_{KT,s} - \dot{W}_{KT,a}, \varepsilon_{KT} = \frac{\dot{W}_{KT,a}}{\dot{W}_{KT,s}}$
VAL		$\dot{m}_{19} = \dot{m}_{20}$ $\dot{m}_{19}h_{19} = \dot{m}_{20}h_{20}$ $\dot{E}x_{D,VAL} = \dot{E}x_{19} - \dot{E}x_{20}$ $\varepsilon_{VAL} = \frac{\dot{E}x_{19}}{\dot{E}x_{20}}$
LTR		$\dot{m}_{22} = \dot{m}_{23}, \dot{m}_{25} = \dot{m}_{26}$ $\dot{m}_{22}(h_{22} - h_{23}) = \dot{m}_{25}(h_{26} - h_{25})$ $\dot{E}x_{D,LTR} = \dot{m}_{22}(\psi_{22} - \psi_{23}) - \dot{m}_{25}(\psi_{26} - \psi_{25})$ $\varepsilon_{LTR} = \frac{\dot{m}_{22}(\psi_{22} - \psi_{23})}{\dot{m}_{25}(\psi_{26} - \psi_{25})}$
KCON		$\dot{m}_{23} = \dot{m}_{24}, \dot{m}_{28} = \dot{m}_{29}$ $\dot{m}_{23}(h_{23} - h_{24}) = \dot{m}_{28}(h_{29} - h_{28})$ $\dot{E}x_{D,KCON} = \dot{m}_{23}(\psi_{23} - \psi_{24}) - \dot{m}_{28}(\psi_{29} - \psi_{28})$ $\varepsilon_{KCON} = \frac{\dot{m}_{23}(\psi_{23} - \psi_{24})}{\dot{m}_{28}(\psi_{29} - \psi_{28})}$

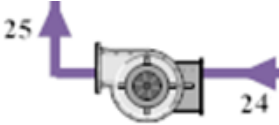
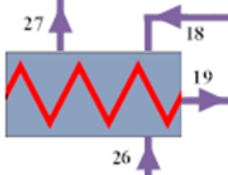
KP		$\dot{m}_{23} = \dot{m}_{24}$ $\dot{W}_{KP,a} = \dot{m}_{24}(h_{25} - h_{24})$ $\dot{W}_{KP,s} = \dot{m}_{24}(\psi_{25} - \psi_{24})$ $\dot{E}x_{D,KP} = \dot{W}_{KP,a} - \dot{W}_{KP,s}, \varepsilon_{KP} = \frac{\dot{W}_{KP,s}}{\dot{W}_{KP,a}}$
HTR		$\dot{m}_{18} = \dot{m}_{19}, \dot{m}_{26} = \dot{m}_{27}$ $\dot{m}_{18}(h_{18} - h_{19}) = \dot{m}_{26}(h_{27} - h_{26})$ $\dot{E}x_{D,HTR} = \dot{m}_{18}(\psi_{18} - \psi_{19}) - \dot{m}_{26}(\psi_{27} - \psi_{26})$ $\varepsilon_{HTR} = \frac{\dot{m}_{18}(\psi_{18} - \psi_{19})}{\dot{m}_{26}(\psi_{27} - \psi_{26})}$

Table 4.34. Thermodynamic properties of the states of the GT-KAL cycle

State	Fluid	T (°C)	P (bar)	$\dot{m}$ (kg/s)	X	h (kJ/kg)	s (kJ/kgK)	ψ (kJ/kg)	$\dot{E}x$ (kW)
1	CO ₂	39	74	11.07	-	-87.87	-1.022	212.9	2356
2	CO ₂	78.82	123.8	11.07	-	-69.84	-1.022	231.4	2561
3	CO ₂	39	123.8	11.07	-	-215.8	-1.464	214.5	2374
4	CO ₂	52.1	207.2	11.07	-	-204.9	-1.464	225.6	2497
5	CO ₂	375.9	207.2	11.07	-	313.2	-0.2984	401.7	4446
6	CO ₂	547.7	207.2	11.07	-	525	-0.00895	528.7	5851
7	CO ₂	423.4	74	11.07	-	376.7	-0.00895	387.3	4286
8	CO ₂	71.1	74	11.07	-	-22.42	-0.8215	219.5	2429
9	Water	20	1	10.51	-	83.93	0.2962	0	0
10	Water	34	1	10.51	-	142.5	0.4913	1.355	14.25
11	Water	20	1	15.25	-	83.93	0.2962	0	0
12	Water	42	1	15.25	-	175.9	0.5989	3.289	50.15
13	Exhaust	566.7	1.9	16	-	866.2	6.591	306.4	4903
14	Exhaust	380	1.9	16	-	663.5	6.318	183.7	2939
15	Exhaust	86.98	1.9	16	-	361	5.705	60.68	970.9
16	NH ₃ H ₂ O	372.5	40	2.2	0.80	2371	6.086	17173	37781
17	NH ₃ H ₂ O	372.5	40	1.54	0.93	2246	5.837	19654	30267
18	NH ₃ H ₂ O	372.5	40	0.66	0.4974	2663	6.487	10979	7246
19	NH ₃ H ₂ O	70.48	40	0.66	0.4974	80.06	0.8533	10047	6631
20	NH ₃ H ₂ O	70.81	13.33	0.66	0.4974	80.06	0.8632	10044	6629
21	NH ₃ H ₂ O	269.9	13.33	1.54	0.93	1941	5.837	19331	29770
22	NH ₃ H ₂ O	115	13.33	2.2	0.80	1382	4.464	16496	36291
23	NH ₃ H ₂ O	47.98	13.33	2.2	0.80	420.1	1.721	16338	35943
24	NH ₃ H ₂ O	32.34	13.33	2.2	0.80	11.27	0.4363	16306	35873
25	NH ₃ H ₂ O	32.98	40	2.2	0.80	15.14	0.4363	16310	35881
26	NH ₃ H ₂ O	55.48	13.33	2.2	0.80	629	2.361	16359	35990
27	NH ₃ H ₂ O	82.48	13.33	2.2	0.80	990.9	3.418	16411	36105
28	Water	20	1	36.55	-	83.93	0.2962	0	0
29	Water	25	1	36.55	-	104.8	0.3669	0.1764	6.446

The thermodynamic results of the components of the GT-KAL cycle are given in Table 4.35. As a result of the thermodynamic analysis of GT-KAL cycle, the net power output, the energy and the exergy efficiencies are calculated as 1594 kW, 19.71 % and 40.53 %, respectively. The energy and the exergy efficiencies of the GT-KAL cycle can be improved by decreasing the exergy destructions. It is clearly seen from Table 4.35 that the maximum exergy destructions occur for the GTHE and HTR, respectively. Due to the thermal losses in heat exchangers high amount of exergy destruction are expected for all kinds of heat exchangers. It is possible to decrease the exergy destruction in heat exchangers by using optimum heat exchanger design. The highest exergy efficiency is seen for the GT because of its negligibly small exergy destruction rate. In addition to this in GT-KAL cycle, the exergy efficiencies of the REC, KT and KP are also seen very high due to their small amount of exergy destruction rates.

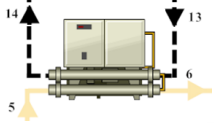
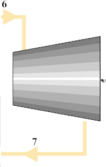
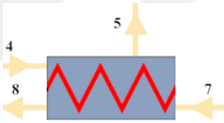
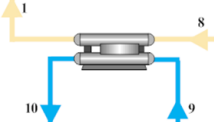
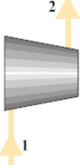
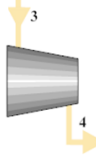
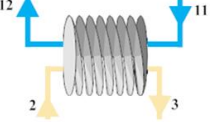
Table 4.35. Thermodynamic results of the GT-KAL cycle

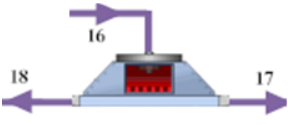
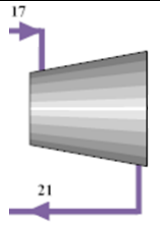
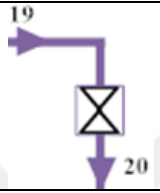
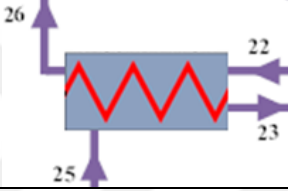
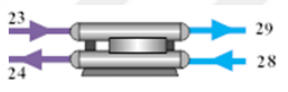
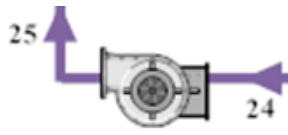
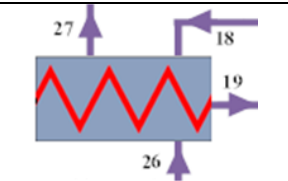
Component	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}x_F$ (kW)	$\dot{E}x_P$ (kW)	$\dot{E}x_D$ (kW)	ε (%)
GTHE	2758	-	1964	1405	559.3	71.52
GT	-	1509	1565	1509	55.65	96.44
REC	5734	-	1949	1857	92.05	95.28
PRE	724.3	-	72.89	14.25	58.64	19.55
LPC	-	234.7	234.7	205.3	29.37	87.48
HPC	-	142.2	142.2	123	19.25	86.46
INT	1403	-	187.3	50.15	137.1	26.78
GEN	4840	-	1968	1676	291.8	85.17
KT	-	469.7	496.7	469.7	26.96	94.57
HTR	1705	-	614.9	114.9	500.1	18.68
LTR	2177	-	347.9	109	238.8	31.34
KP	-	8.506	8.506	7.847	0.6593	92.25
KCON	899.3	-	70.19	6.446	63.74	9.184
GT-KAL Cycle	Energy efficiency (%)				19.71	
	Exergy efficiency (%)				40.53	

4.6.3 Thermoeconomic Analysis

Thermoeconomic analysis on GT-KAL is carried out by using the same procedures applied on the other models which are already explained in full detail. The cost balance and the auxiliary equations are obtained for each sub-component of the system. The related equations are given in Table 4.36.

Table 4.36. Exergy cost rate balances and auxiliary equations for sub-components of GT-KAL cycle

GTHE		$c_{13}\dot{E}x_{13} + c_5\dot{E}x_5 + \dot{Z}_{GTHE} = c_{14}\dot{E}x_{14} + c_6\dot{E}x_6$ $c_{13} = \text{known}, \frac{\dot{C}_{13}}{\dot{E}x_{13}} = \frac{\dot{C}_{14}}{\dot{E}x_{14}}$
GT		$c_6\dot{E}x_6 + \dot{Z}_{GT} = c_{GT}\dot{W}_{GT} + c_7\dot{E}x_7$ $\frac{\dot{C}_6}{\dot{E}x_6} = \frac{\dot{C}_7}{\dot{E}x_7}$
REC		$c_4\dot{E}x_4 + c_7\dot{E}x_7 + \dot{Z}_{REC} = c_5\dot{E}x_5 + c_8\dot{E}x_8$ $\frac{\dot{C}_7}{\dot{E}x_7} = \frac{\dot{C}_8}{\dot{E}x_8}$
PRE		$c_8\dot{E}x_8 + c_9\dot{E}x_9 + \dot{Z}_{PRE} = c_1\dot{E}x_1 + c_{10}\dot{E}x_{10}$ $c_9 = 0, \frac{\dot{C}_1}{\dot{E}x_1} = \frac{\dot{C}_8}{\dot{E}x_8}$
LPC		$c_{LPC}\dot{W}_{LPC} + \dot{Z}_{LPC} = c_2\dot{E}x_2 - c_1\dot{E}x_1$ $\frac{\dot{C}_1}{\dot{E}x_1} = \frac{\dot{C}_2}{\dot{E}x_2}$
HPC		$c_{HPC}\dot{W}_{HPC} + \dot{Z}_{HPC} = c_4\dot{E}x_4 - c_3\dot{E}x_3$
INT		$c_2\dot{E}x_2 + c_{11}\dot{E}x_{11} + \dot{Z}_{INT} = c_3\dot{E}x_3 + c_{12}\dot{E}x_{12}$ $c_{11} = 0, \frac{\dot{C}_2}{\dot{E}x_2} = \frac{\dot{C}_3}{\dot{E}x_3}$

GEN		$c_{14}\dot{E}x_{14} + c_{27}\dot{E}x_{27} + \dot{Z}_{GEN} = c_{15}\dot{E}x_{15} + c_{16}\dot{E}x_{16}$ $\frac{\dot{C}_{27}}{\dot{E}x_{27}} = \frac{\dot{C}_{16}}{\dot{E}x_{16}}$
SEP		$c_{16}\dot{E}x_{16} + \dot{Z}_{SEP} = c_{17}\dot{E}x_{17} + c_{18}\dot{E}x_{18}$ $\frac{\dot{C}_{17}}{\dot{E}x_{17}} = \frac{\dot{C}_{18}}{\dot{E}x_{18}}$
KT		$c_{17}\dot{E}x_{17} + \dot{Z}_{KT} = c_{KT}\dot{W}_{KT} + c_{21}\dot{E}x_{21}$ $\frac{\dot{C}_{17}}{\dot{E}x_{17}} = \frac{\dot{C}_{21}}{\dot{E}x_{21}}$
VAL		$c_{19}\dot{E}x_{19} + \dot{Z}_{VAL} = c_{20}\dot{E}x_{20}$
LTR		$c_{22}\dot{E}x_{22} + c_{25}\dot{E}x_{25} + \dot{Z}_{LTR} = c_{23}\dot{E}x_{23} + c_{26}\dot{E}x_{26}$ $\frac{\dot{C}_{22}}{\dot{E}x_{22}} = \frac{\dot{C}_{23}}{\dot{E}x_{23}}$
KCON		$c_{23}\dot{E}x_{23} + c_{28}\dot{E}x_{28} + \dot{Z}_{KCON} = c_{24}\dot{E}x_{24} + c_{29}\dot{E}x_{29}$ $c_{28} = 0, \frac{\dot{C}_{23}}{\dot{E}x_{23}} = \frac{\dot{C}_{24}}{\dot{E}x_{24}}$
KP		$c_{KP}\dot{W}_{KP} + \dot{Z}_{KP} = c_{25}\dot{E}x_{25} - c_{24}\dot{E}x_{24}$ $\frac{\dot{C}_{24}}{\dot{E}x_{24}} = \frac{\dot{C}_{25}}{\dot{E}x_{25}}$
HTR		$c_{18}\dot{E}x_{18} + c_{26}\dot{E}x_{26} + \dot{Z}_{HTR} = c_{19}\dot{E}x_{19} + c_{27}\dot{E}x_{27}$ $\frac{\dot{C}_{26}}{\dot{E}x_{26}} = \frac{\dot{C}_{27}}{\dot{E}x_{27}}$

The capital cost rate of the components is calculated by using the Eq. (3.30) in GT-KAL cycle. In order to perform the thermoeconomic analysis of GT-KAL cycle, the economic constants are taken as the same in GT and Kalina cycle, which are given in Table 4.5. The capital costs are calculated using the thermodynamic relations of each component. The capital cost function of each component of GT and Kalina cycles is given in Table 4.6 and Table 4.23, respectively [129]. The heat transfer coefficient of

heat exchangers is already given in related section. The exergy flow rates, the cost flow rates and the unit exergy costs of the GT-KAL cycle are given in Table 37.

Table 4.37. Exergy flow rates, cost flow rates and unit exergy costs of the GT-KAL cycle

State	$\dot{E}_x$ (kW)	c (\$/GJ)	$\dot{C}$ (\$/h)
1	2356	4.797	40.68
2	2561	4.797	44.23
3	2374	4.797	41
4	2497	4.828	43.4
5	4446	4.805	76.91
6	5851	4.797	101
7	4286	4.797	74.01
8	2429	4.797	41.94
9	0	0	0
10	14.25	27.72	1.422
11	0	0	0
12	50.15	19.18	3.464
13	4903	3.21	56.66
14	2939	3.21	33.96
15	970.9	3.21	11.22
16	37781	4.365	593.7
17	30267	4.396	479
18	7246	4.396	114.7
19	6631	4.75	113.4
20	6629	4.751	113.4
21	29770	4.396	471.2
22	36291	4.345	567.7
23	35943	13.66	1767
24	35873	13.66	1764
25	35881	13.66	1764
26	35990	4.365	565.6
27	36105	4.365	567.4
28	0	0	0
29	6.446	186.4	4.325

The total capital cost rate and the total cost rate of the GT-Kalina cycle are found to be 33.73 \$/h and 79.17 \$/h, respectively. Thermoeconomic results of GT-KAL cycle is shown in Table 4.38. GT and KT have the highest exergoeconomic factor values due to their higher thermodynamic performance. But all heat exchangers have lower

exergoeconomic factor values because of their high exergy destruction costs when compared to other components of the system. It can be seen from Table 4.38, LTR, HTR, GTHE and GEN have the highest cost rate of destruction because of their high exergy destruction rate. Special attention can be focused on these components in order to increase the total system performance. The lowest exergoeconomic factor is found to be 6.086 for the HTR. This result can be explained by the fact that the HTR has worst thermodynamic performance when compared with other components. On the other hand, the highest relative cost difference is found for KCON. It is clearly shown that a thermodynamic improvement is required for the Kalina condenser.

Table 4.38. Thermoeconomic results of the GT-KAL cycle

Component	$\dot{E}x_D$ (kW)	c_f (\$/GJ)	c_p (\$/GJ)	$\dot{D}_D$ (\$/h)	$\dot{Z}$ (\$/h)	r (%)	f (%)
GTHE	559.3	3.21	4.797	6.464	1.426	49.43	18.07
GT	55.65	4.797	4.797	0.9609	14.85	6.17×10^{-8}	93.92
REC	92.05	4.797	4.805	1.59	1.443	0.1753	47.58
PRE	58.64	4.797	4.797	1.013	0.1633	7.80×10^{-8}	13.89
LPC	29.37	3.426	4.797	0.3622	0.6507	40.01	64.24
HPC	19.25	3.426	4.828	0.2374	0.6507	40.92	73.27
INT	137.1	4.797	4.797	2.368	0.2299	6.42×10^{-8}	8.849
GEN	291.8	3.21	4.365	5.039	3.598	35.99	41.66
KT	26.96	4.396	8.529	0.4266	6.561	48.45	93.89
HTR	500.1	4.365	4.75	7.858	0.5093	8.097	6.086
LTR	238.8	13.66	13.66	11.74	0.9793	1.70×10^{-8}	7.697
KP	0.6593	8.529	13.66	0.02024	0.1247	37.56	86.03
KCON	63.74	13.66	186.4	3.134	0.8743	92.67	21.81

4.7 Conclusion

In this chapter, the renewable and the sustainable models that can be adapted to the actual system in order to utilize the exhaust gas of the Gaziantep Municipal Solid Waste Power Plant as an energy source have been analyzed in terms of thermodynamic and thermoeconomic aspects. Based on the thermodynamic analyses of all models; GT-KAL cycle (Model 5) is found to be best one among others with respect to power production capacity. 28.18 % additional power can be supplemented to the actual power plant output by using the developed model. However the GT-KAL cycle model is highly expensive when compared to GT, Kalina and ORC. The total cost rate of GT-KAL cycle is found to be 79.17 \$/h which is significantly more than other models. From the point of thermoeconomic analyses, the best option is evaluated as GT cycle with the total cost rate of 44.57 \$/h. The results of the thermodynamic and thermoeconomic analyses show the importance of an optimization study which is required for the developed models.

CHAPTER 5

RESULTS AND DISCUSSIONS

5.1 Introduction

In this chapter, five renewable and sustainable models which are described in the previous chapter are considered with respect to the thermodynamic and thermoeconomic analyses in the frame of various parametric studies. The performance of the models are evaluated in terms of the pressure, the temperature and the mass boundaries. The boundaries of the each parameter are arranged taking into account the selected parameter in the design stage of the given cycle. That is, the upper and lower boundaries of the parameter are selected according to the selected parameter in the design stage. It is not rational to operate the system with values other than these boundaries. In addition to this, all models are optimized by using non-dominated sorting genetic algorithm (NSGA-II) method in MATLAB software with respect to selected parameter boundaries. The optimal solutions of the each cycle is illustrated utilizing the Pareto frontier schematically. Then, the optimal solutions are compared to the design stage of each cycle and the net power output, the energy efficiency, the exergy efficiency, the total cost rate of cycle and the total capital cost rate of the cycle are given with their deviations. Lastly, the comparison of all models are performed depended on the electricity cost production.

5.2 Effect of the Pressure Ratio on Models

Pressure ratio is a dominant parameter on all kinds of cycles during the evaluating the performance of the system. For this reason the effect of pressure ratio of GT, KC and ORC are considered with the given boundaries for all models except for Model 2 due to the second model design is based on temperature changes.

The effect of the pressure ratio of GT cycle and KC on the energy and the exergy efficiencies as well as the net power output of the GT and KC are shown in Fig. 5.1 and Fig. 5.2. Also, Figures from 5.3 to 5.6 show the effect of the pressure ratio of GT, ORC and KC on same performance parameters of the GT-ORC and GT-KAL cycles, respectively. It is clearly seen that the increase in the pressure ratio value of cycles caused an increase in the energy efficiency, the exergy efficiency and the net power output of all systems except for GT-ORC related to PR_{ORC} . The increase in the pressure difference between the inlet and outlet streams of the turbines which required for power producing directly depends on the increase in the pressure ratio. Therefore, the increase in the net power output of the systems by increasing pressure ratio can be shown as a result of this situation. Moreover, it can be explained such that the heat transfer through GTHE and GEN begins to increase after a certain time while pressure ratio of cycles are increasing. On contrary to this, the increase in PR_{ORC} greatly increases the power required for the operation of the ORC pump in the ORC system. As a result of this, the net power output of the GT-ORC system severely decreases and the energy and the exergy efficiencies decrease accordingly.

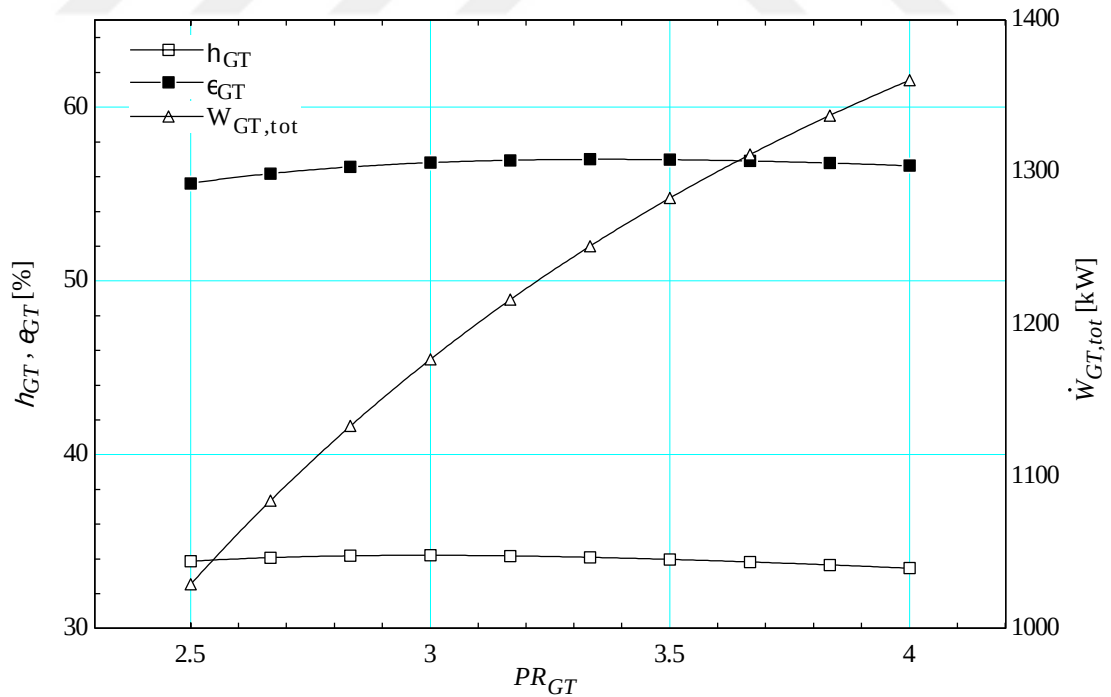


Figure 5.1. Effect of the PR_{GT} on the energy and exergy efficiencies and the net power output of the GT

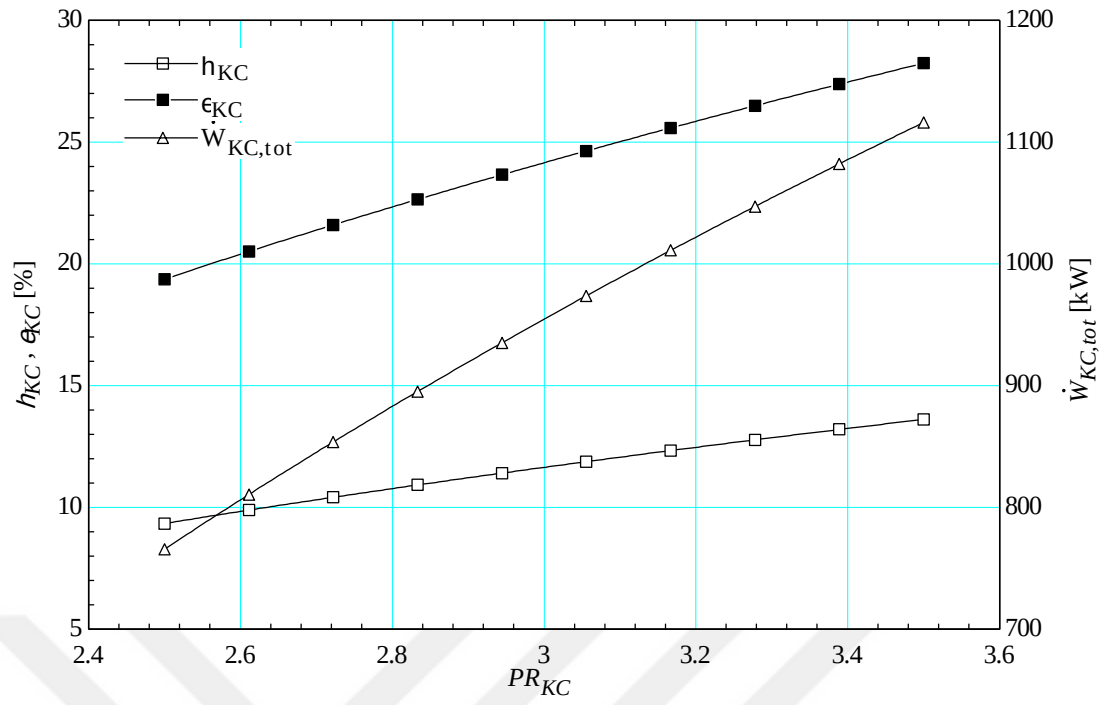


Figure 5.2. Effect of the PR_{KC} on the energy and exergy efficiencies and the net power output of the KC

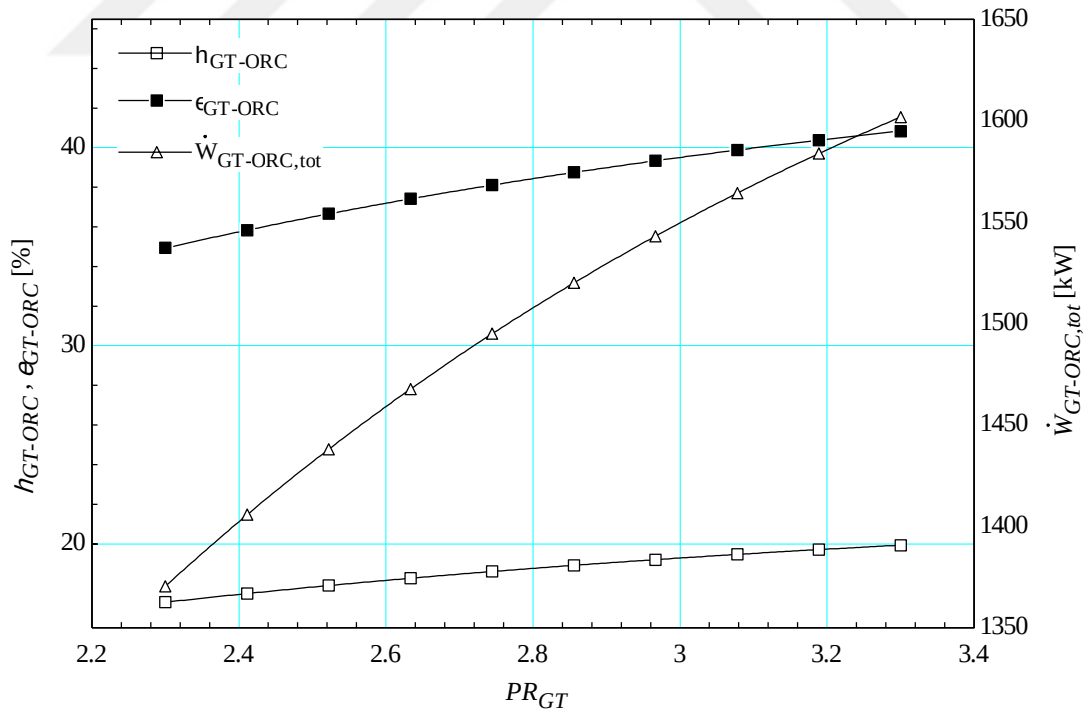


Figure 5.3. Effect of the PR_{GT} on the energy and exergy efficiencies and the net power output of GT-ORC

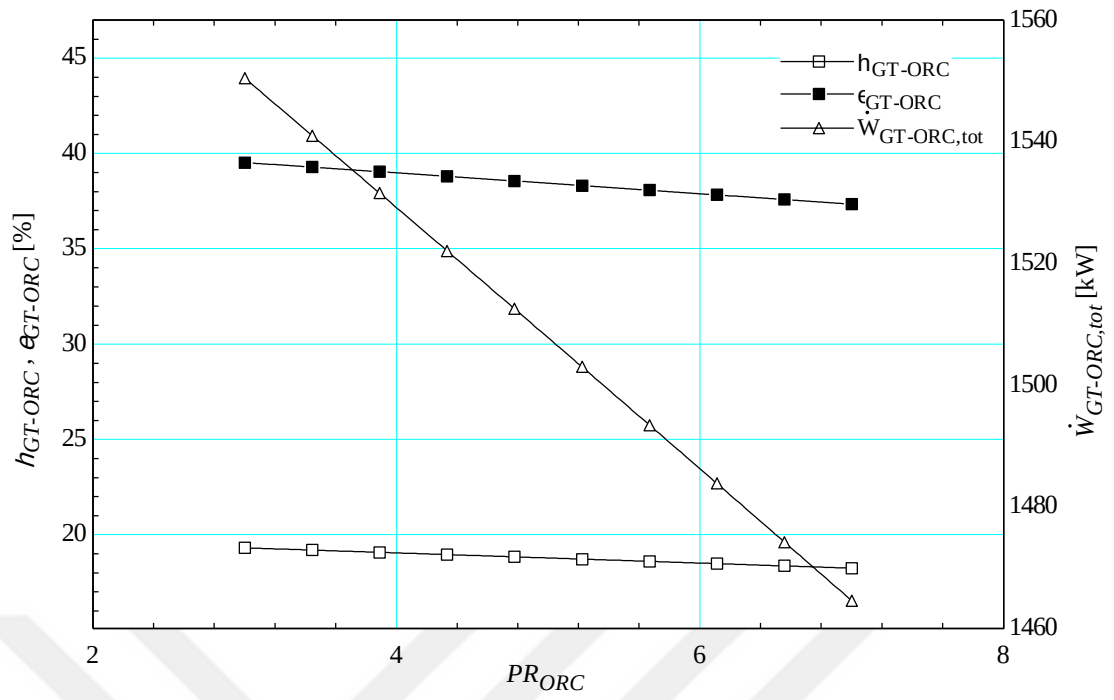


Figure 5.4. Effect of the PR_{ORC} on the energy and exergy efficiencies and the net power output of GT-ORC

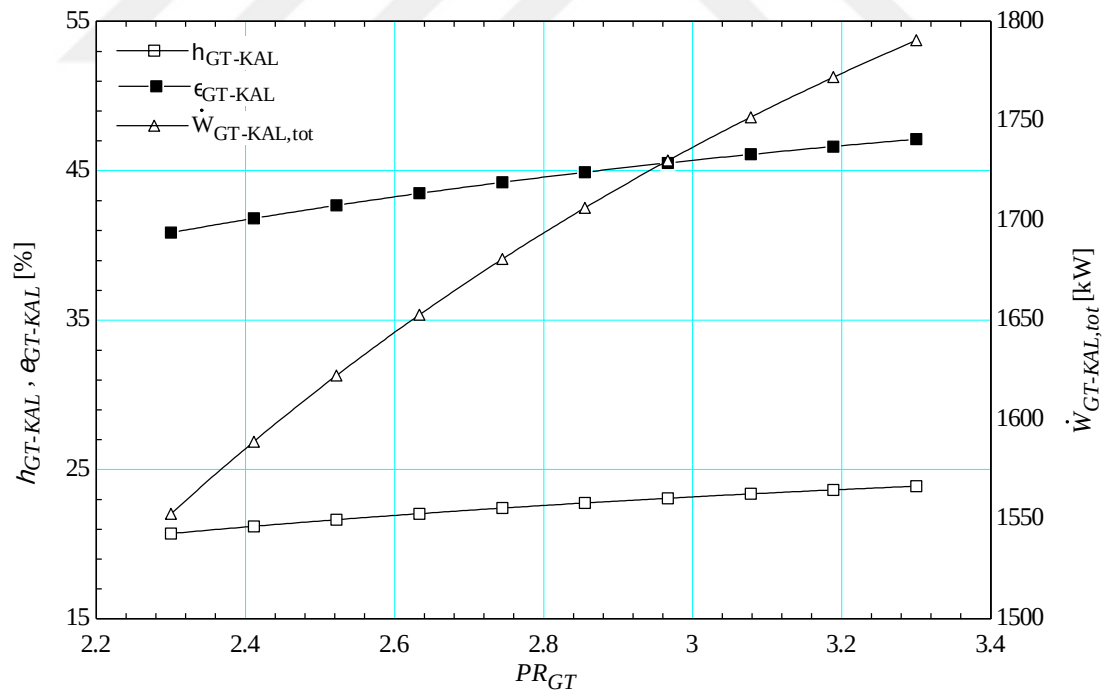


Figure 5.5. Effect of the PR_{GT} on the energy and exergy efficiencies and the net power output of GT-KAL

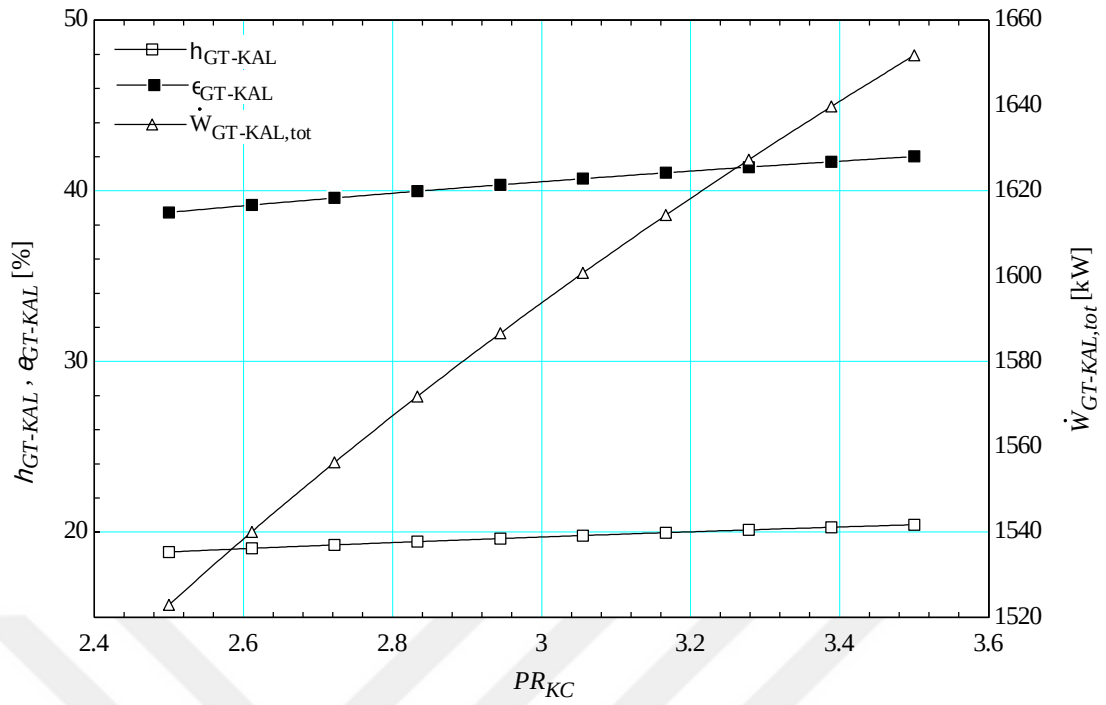


Figure 5.6. Effect of the PR_{KC} on the energy and exergy efficiencies and the net power output of GT-KAL

Figures from 5.7 to 5.12 show the effect of the pressure ratio of GT, ORC and KC on the total cost rate and the net power output of the systems. It can be concluded that the total cost rate of all cycles increase by increasing the pressure ratio value of cycles due to the capital cost increases in the sub-components of the systems. This can be regarded as one of the clearest indicators of the contrast between the thermodynamic performance of systems and their thermoeconomic performance. Due to this reason, it is obviously shown that the optimization is vitally important to see the boundaries of the thermodynamic and thermoeconomic performances of systems before designing stage and at the same time to determine optimum operating values.

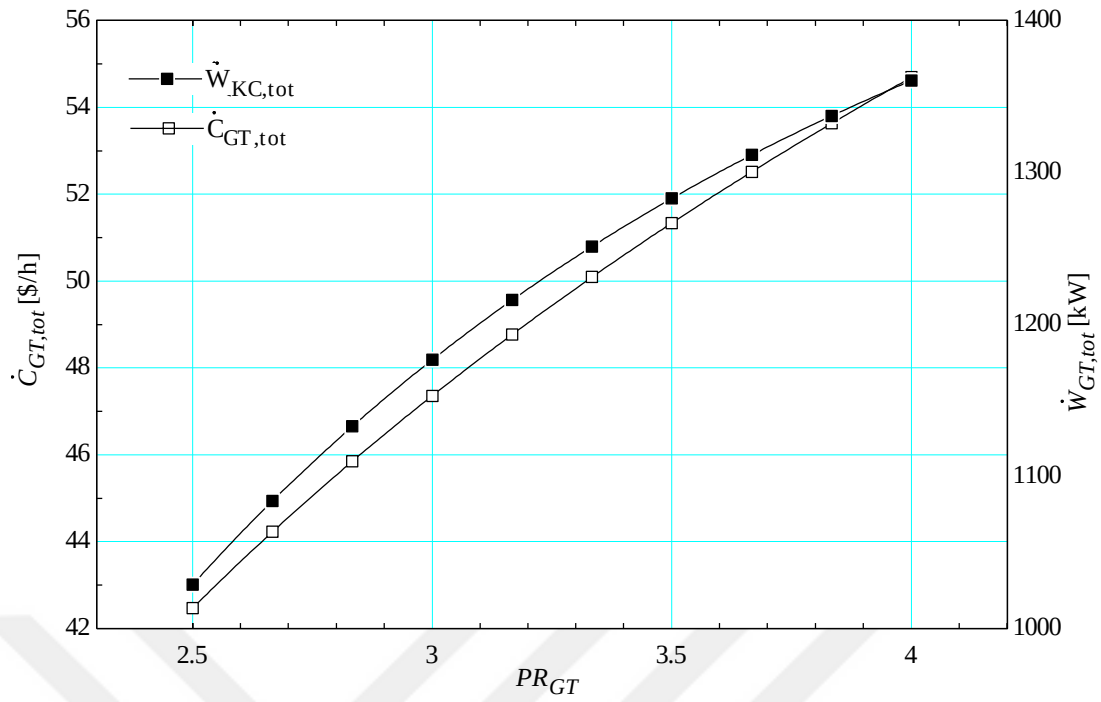


Figure 5.7. Effect of the PR_{GT} on the net power output and the total cost rate of GT

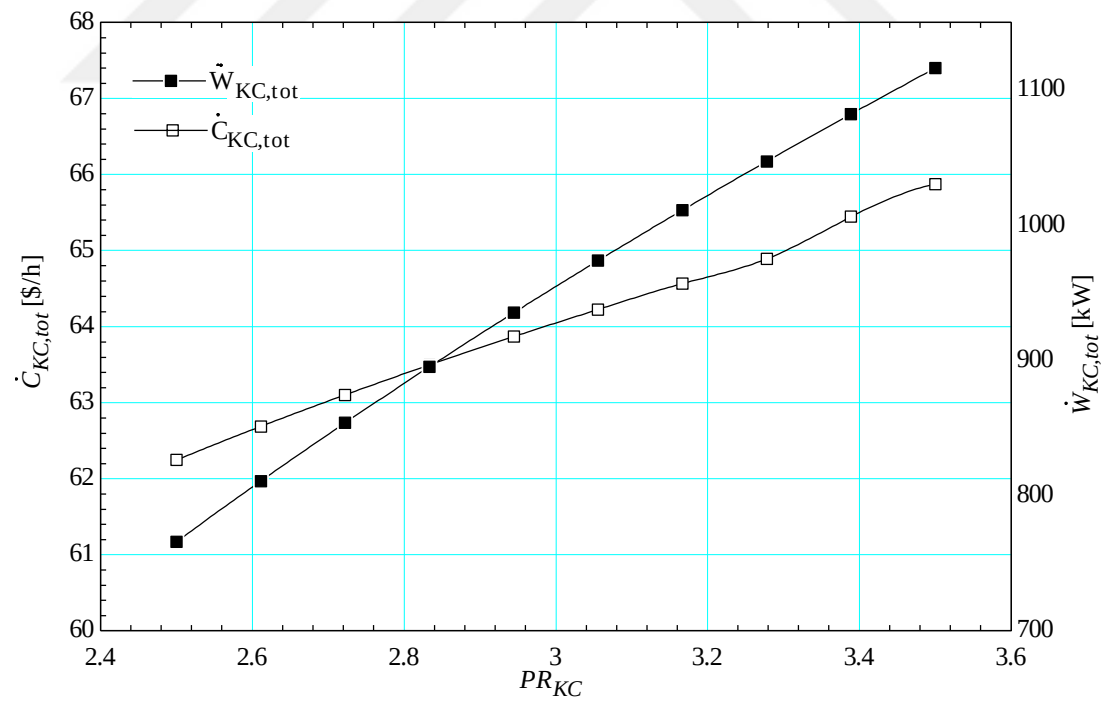


Figure 5.8. Effect of the PR_{KC} on the net power output and the total cost rate of KC

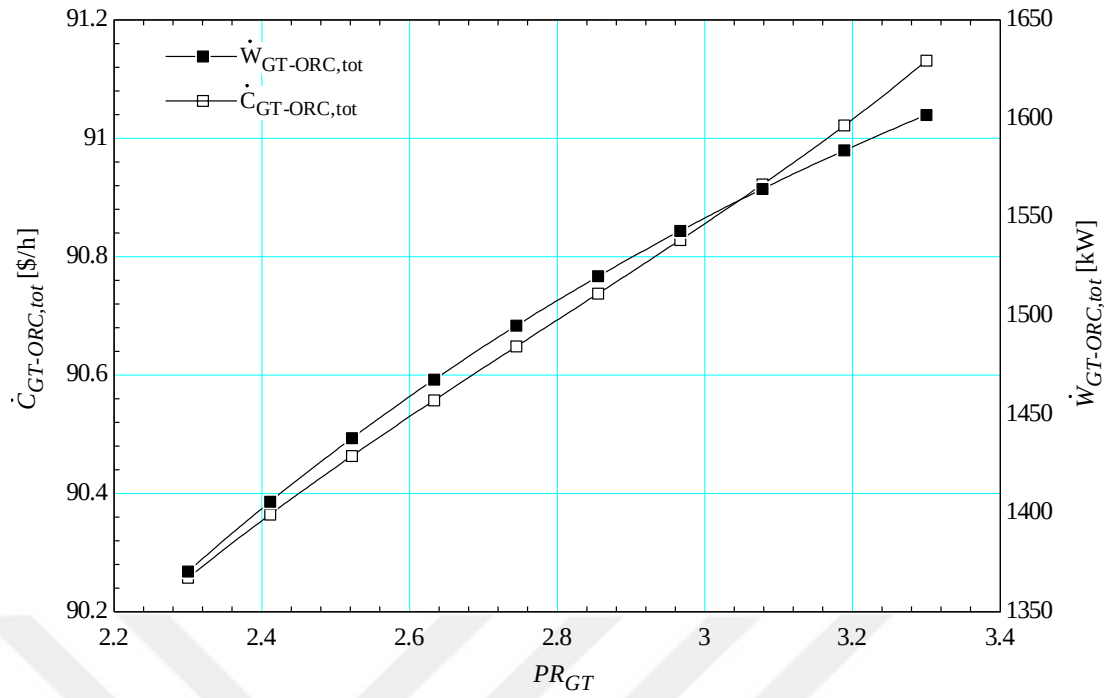


Figure 5.9. Effect of the PR_{GT} on the net power output and the total cost rate of GT-ORC

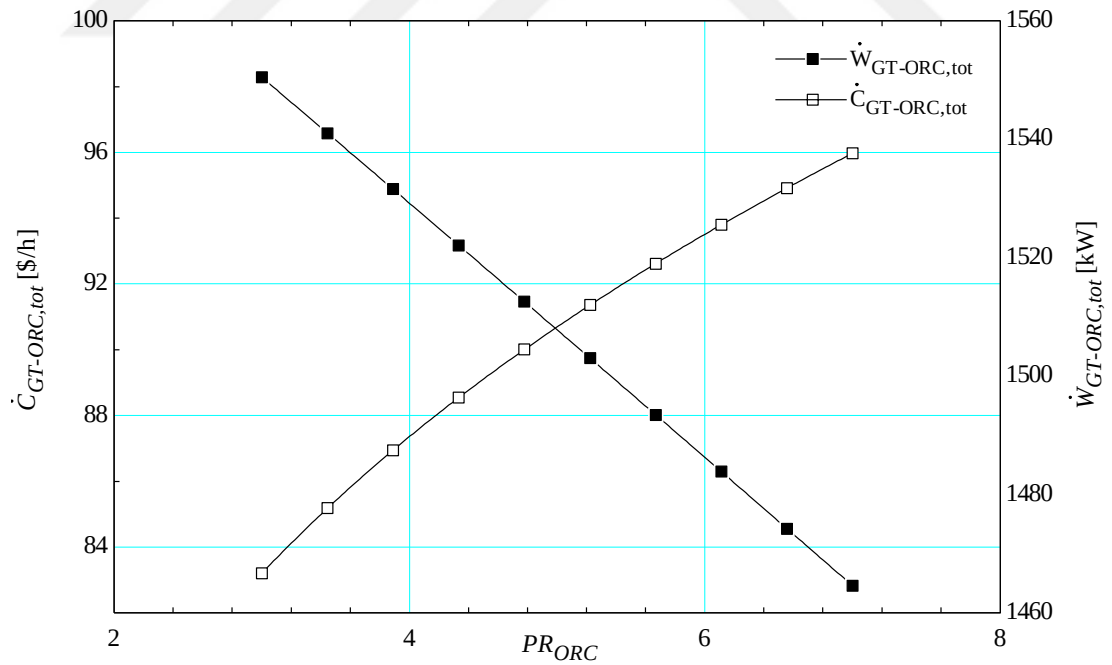


Figure 5.10. Effect of the PR_{ORC} on the net power output and the total cost rate of GT-ORC

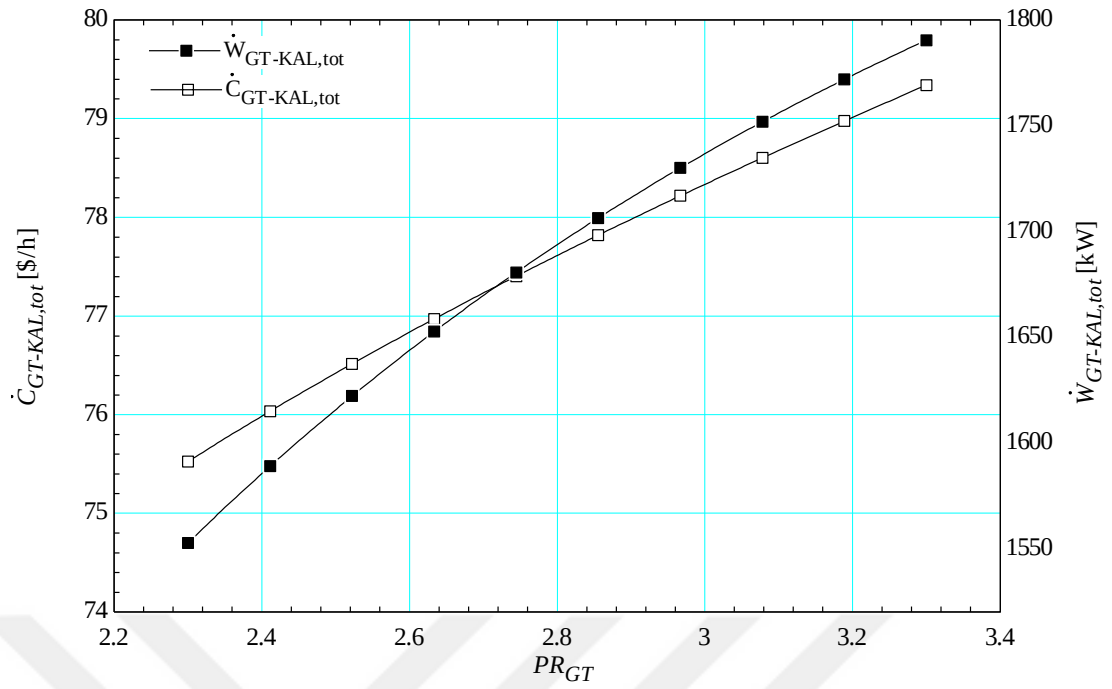


Figure 5.11. Effect of the PR_{GT} on the net power output and the total cost rate of GT-KAL

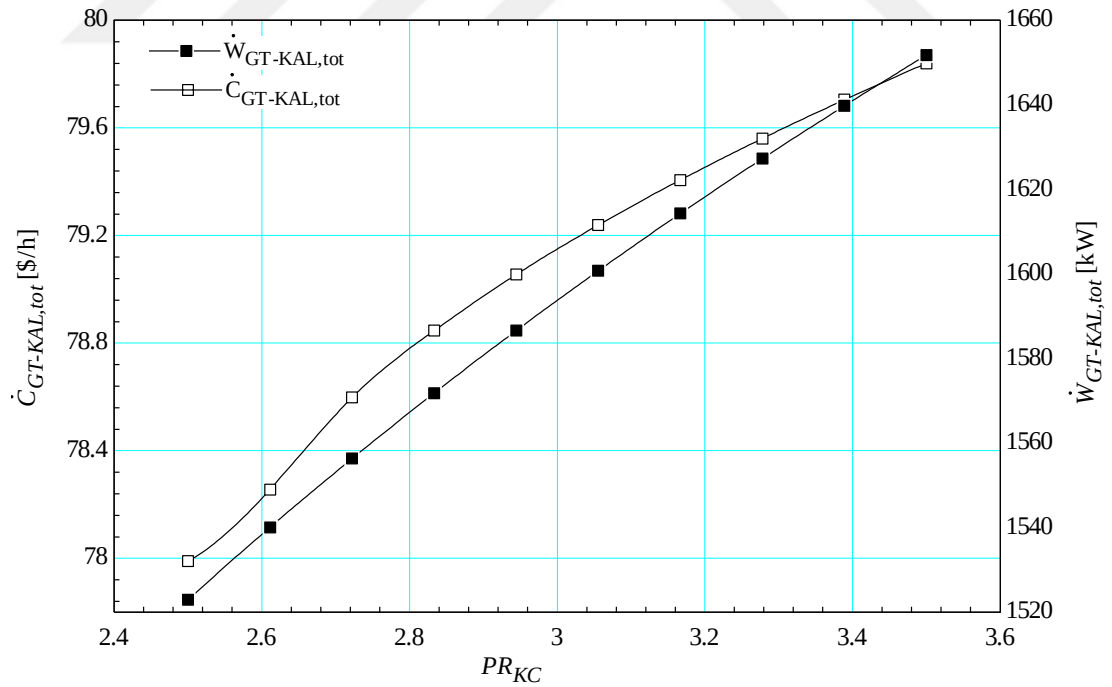


Figure 5.12. Effect of the PR_{KC} on the net power output and the total cost rate of GT-KAL

5.3 Effect of the Temperature on Models

Temperature is another important parameter on power production systems in order to consider the thermodynamic and thermoeconomic performances of system. Due to this, the effect of $LMTD_{GTHE}$ and $LMTD_{GEN}$ are evaluated for Model 1, 4 and 5. But the LMTD are not considered for Model 2 and 3 because of the working fluids which are used in related cycles are maintained in sub-critical region during the power production process. This can be signified with the temperature difference are found to be very high between the exhaust gas of actual system and working fluids in cycles. The effect of the LMTD of GTHE and GEN on the energy efficiency, the exergy efficiency and the net power output of GT, GT-ORC and GT-KAL is presented in Figures from 5.13 to 5.16. The increase in LMTD decreases the net power output and consequently the energy and the exergy efficiencies. Because the increase in LMTD causes an increase in the amount of heat losses from the GTHE and GEN in related cycles. As a result of this the energy and the exergy efficiencies and also the net power output reduction in GT-ORC system can be expected.

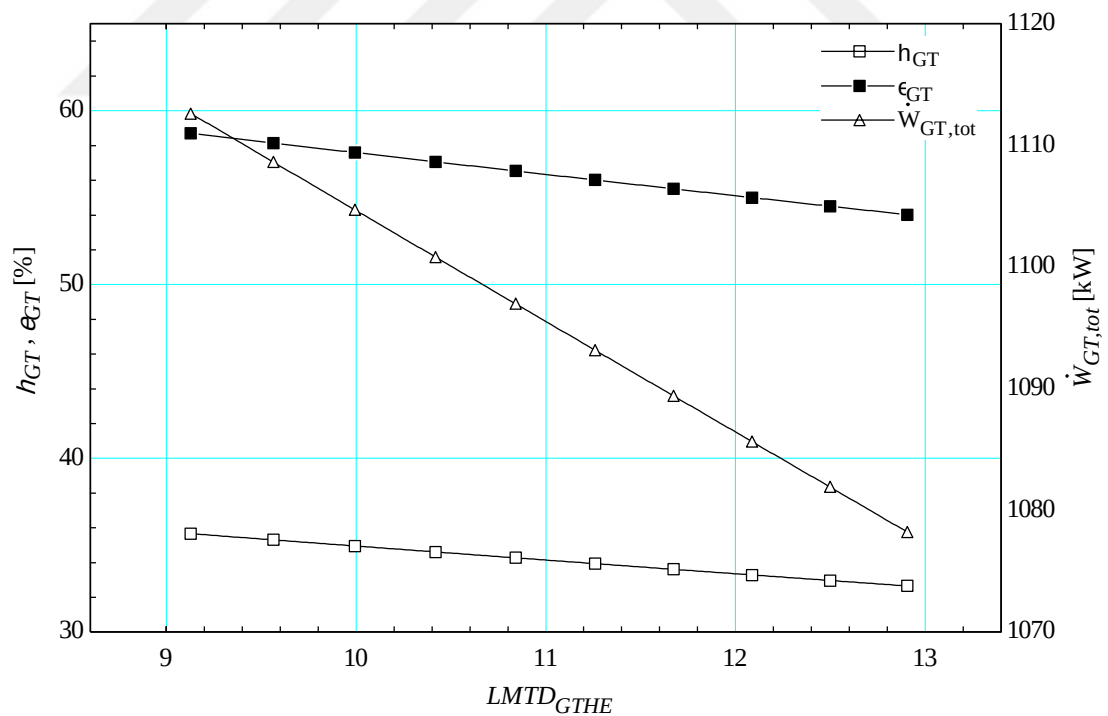


Figure 5.13. Effect of the $LMTD_{GTHE}$ on the energy and exergy efficiencies and the net power output of GT

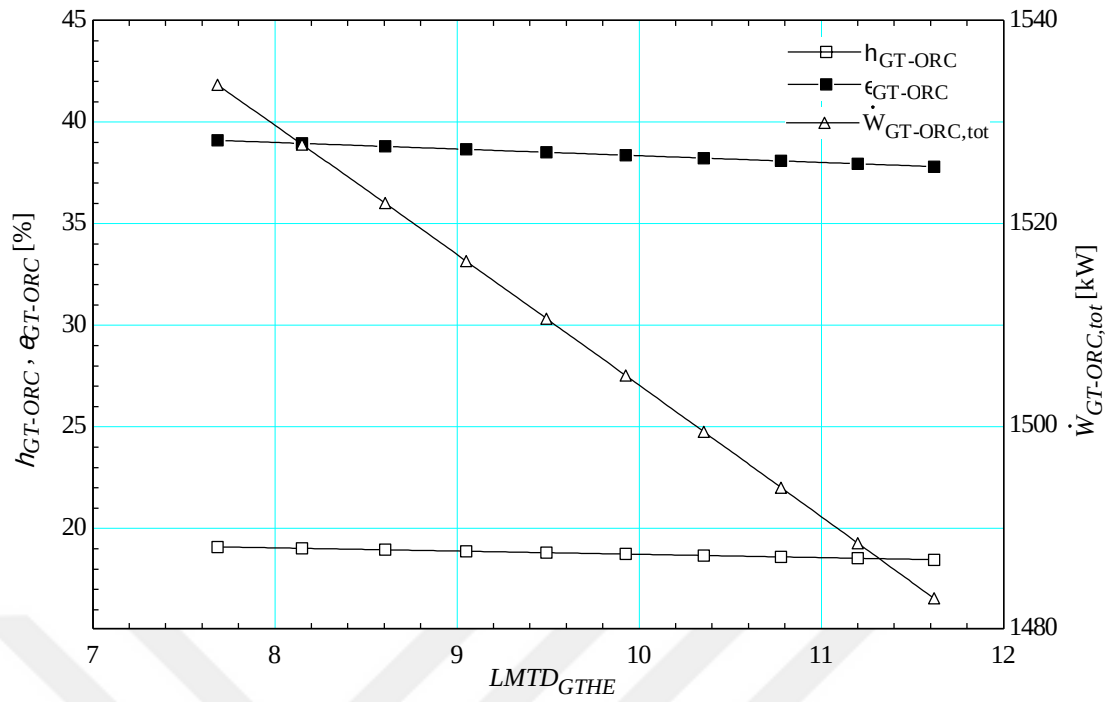


Figure 5.14. Effect of the $LMTD_{GTHE}$ on the energy and exergy efficiencies and the net power output of GT-ORC

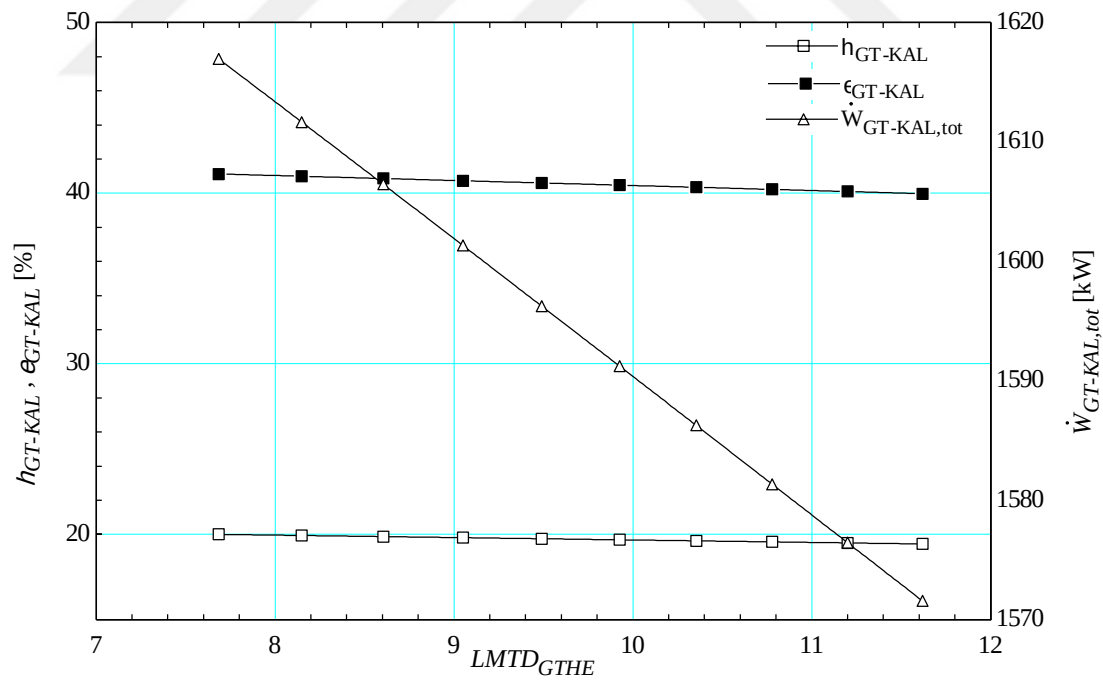


Figure 5.15. Effect of the $LMTD_{GTHE}$ on the energy and exergy efficiencies and the net power output of GT-KAL

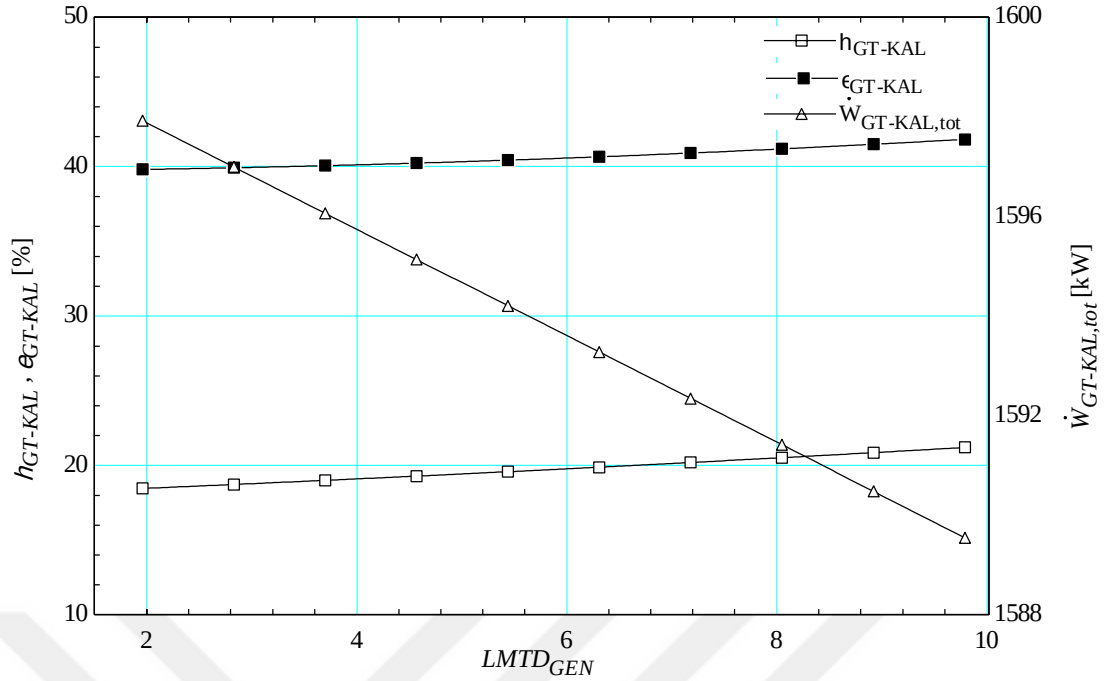


Figure 5.16. Effect of the $LMTD_{GEN}$ on the energy and exergy efficiencies and the net power output of GT-KAL

As can be seen from Figures 5.18 to 5.20, the total cost rate of the GT-ORC and GT-KAL combined systems are decreased by increasing the LMTD values. It can be said that the decrease in the net power output causes a decrease in the total cost rate of the GT-ORC and GT-KAL combined systems. The total capital costs of GTHE, GEN and GT remarkably decrease by increasing of the $LMTD_{GTHE}$. But, despite a decrease in the net power output of cycle by increasing the $LMTD_{GTHE}$ also causes an increases of the total cost rate of GT cycle which is shown in Fig. 5.17. The reason of that can be explained with the increment of cost rate of inlet stream of the GTHE in GT is greater than that of the increment of cost rate of exit stream in same component. Thus, the total cost rate of the cycle is increased even if the total capital cost rate decrease. This matter is not appeared for the GT-ORC and GT-KAL due to the usage of exit stream of the GTHE in these cycles. Lastly, it can be stated that although the increase in $LMTD_{GTHE}$ causes to decrease in the total cost of the system, the cost difference of the inlet and exit stream of GTHE increased greatly, thereby increasing the total cost rate of the system. However, it can be resulted that the LMTD has a dominant effect on the total cost rate of the systems but not for all.

Therefore, it can be deduced that the increase in $LMTD$ affected to combined cycles positively.

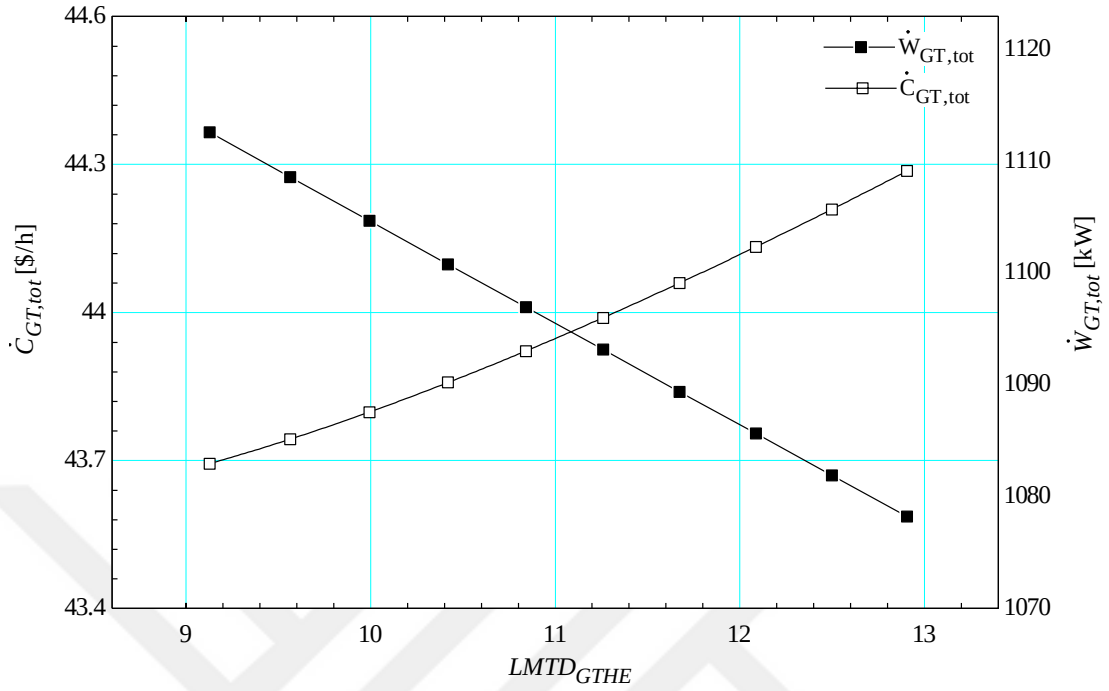


Figure 5.17. Effect of the $LMTD_{GTHE}$ on the net power output and the total cost rate of GT

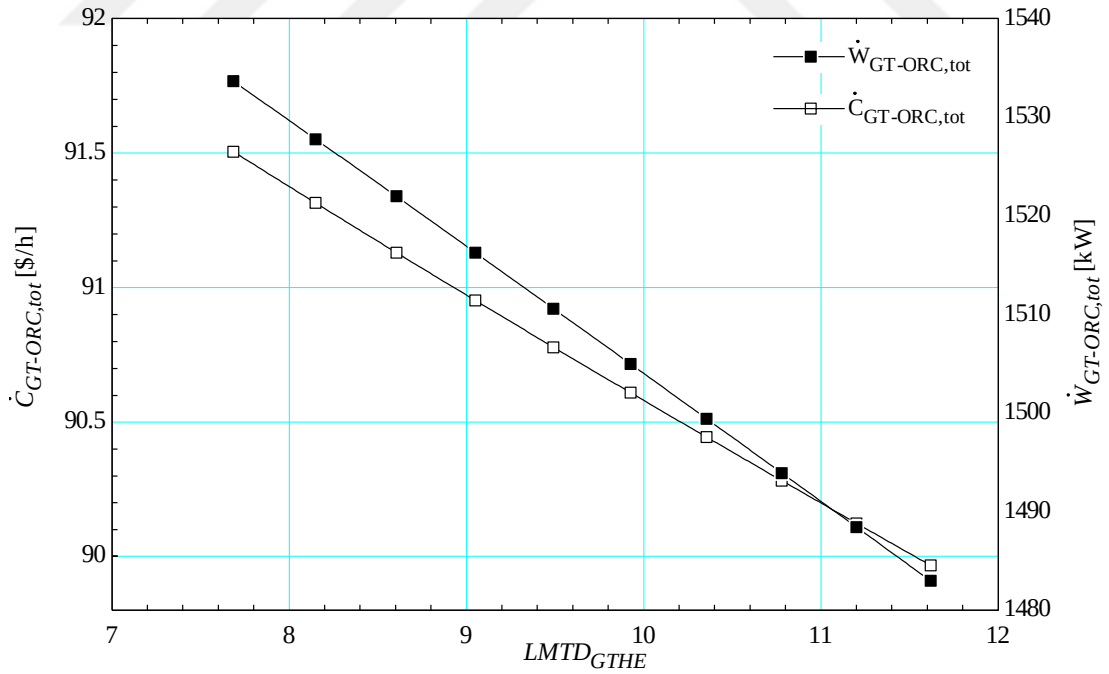


Figure 5.18. Effect of the $LMTD_{GTHE}$ on the net power output and the total cost rate of GT-ORC

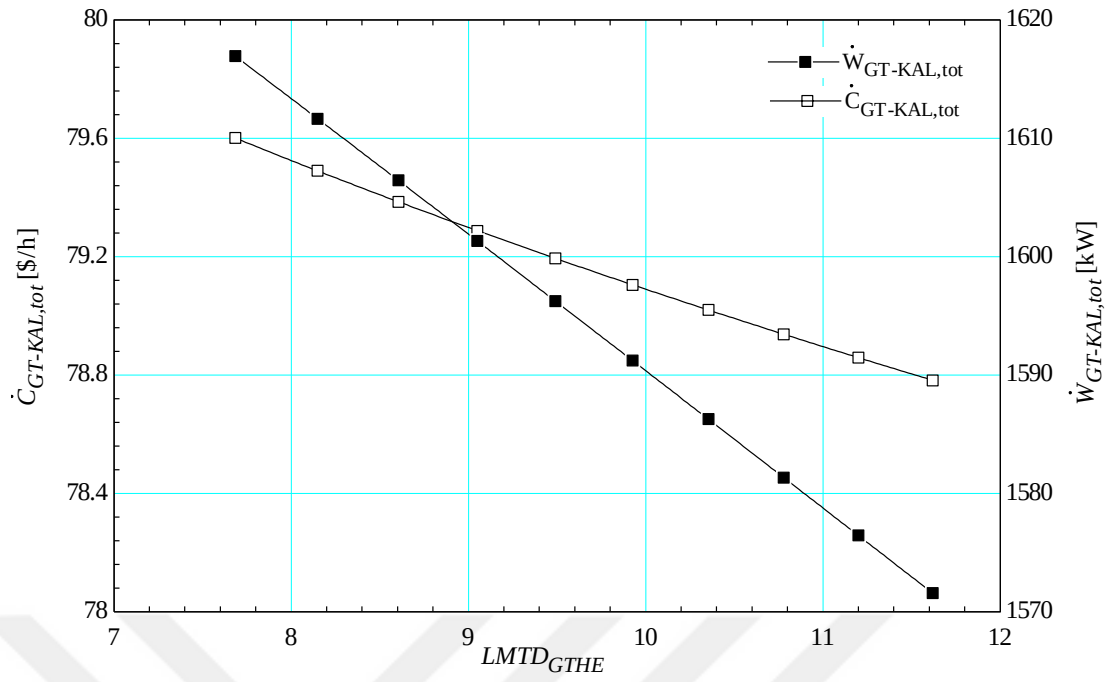


Figure 5.19. Effect of the $LMTD_{GTHE}$ on the net power output and the total cost rate of GT-KAL cycle

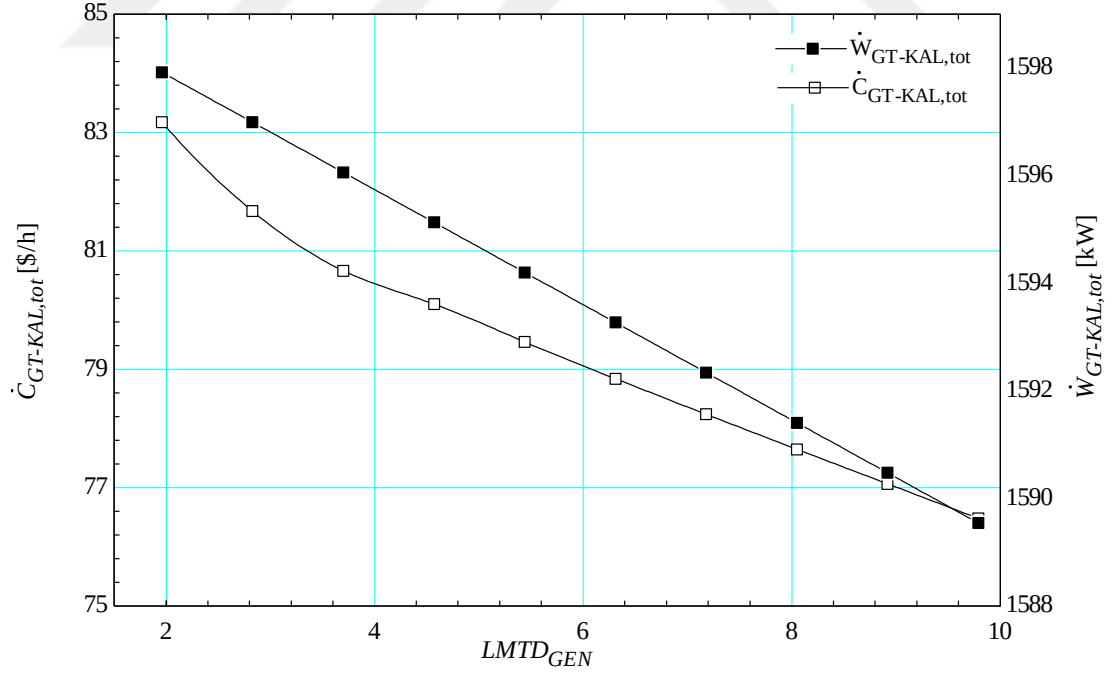


Figure 5.20. Effect of the $LMTD_{GEN}$ on the net power output and the total cost rate of GT-KAL cycle

Figure 5.21 shows the effect of the temperature difference which is defined as the temperature difference between the critical temperature of the working fluid and the inlet temperature of the turbine on the exergy efficiency, the net power output, the total cost rate respectively.

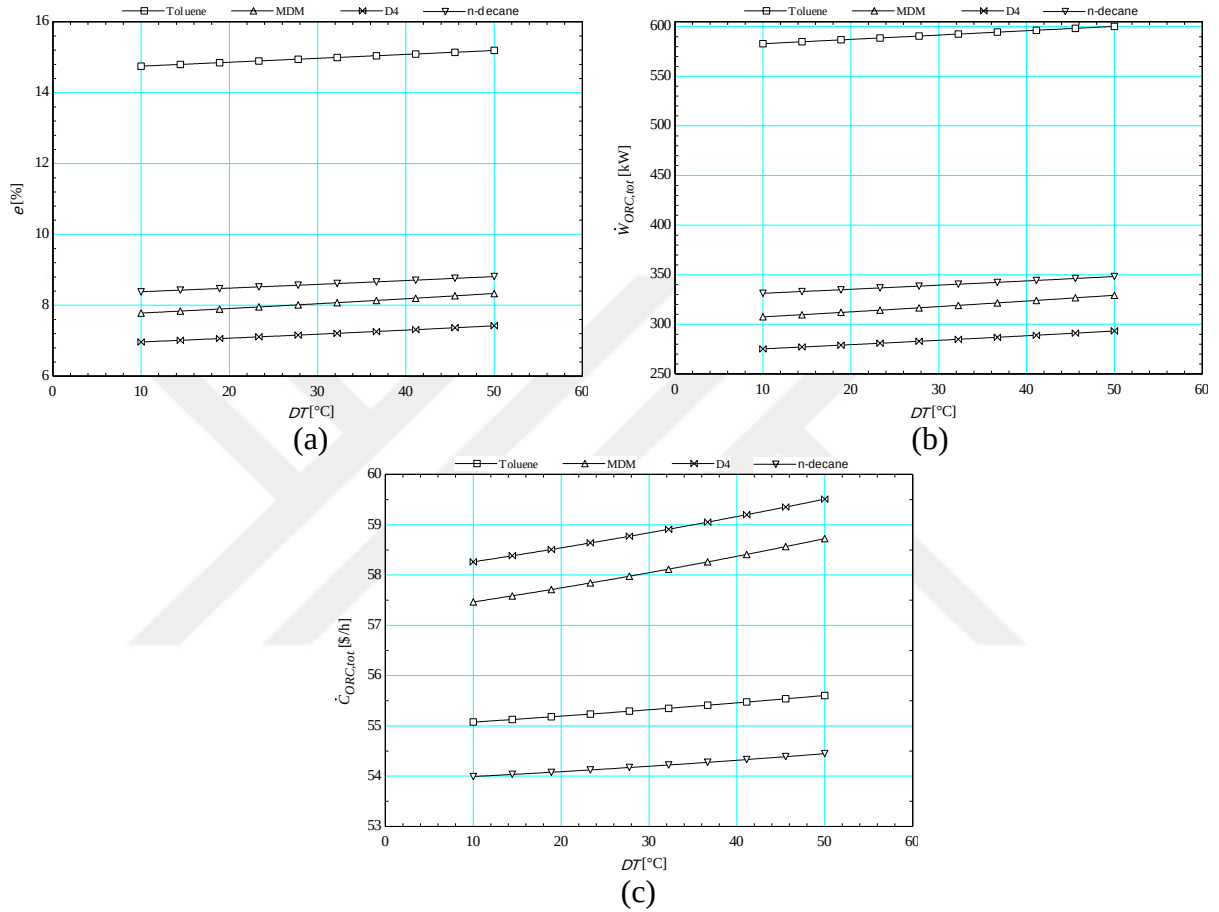


Figure 5.21. Effect of the temperature difference on (a) the exergy efficiency, (b) the net power output and (c) the total cost rate

It can be seen from Fig. 5.21(a) that the increase in the temperature difference causes in the increase of the exergy efficiency as well as the increase in the net power output for all working fluids which is shown in Fig. 5.21(b). The inlet and the exit temperature of the exhaust gas are kept constant which means that the heat capacity of the evaporator is fixed throughout the loop of ORC. Hence the higher the inlet temperature of the turbine causes the lower the mass flow rate of the working fluid

during the process. Then the lower mass flow rate of the working fluid produces the lower the net power output. In addition to this, the total cost rate is also increased by increasing of the temperature difference which is given in Fig. 5.21(c). This is mainly related with the total cost of the ORC system with respect to working fluids. The highest and lowest value of total cost are evaluated for the D4 and n-decane, respectively.

Figure 5.22 shows the effect of the temperature difference on the exergoeconomic factor of the evaporator, the condenser, the turbine and the pump of the ORC system. The higher the exergoeconomic factor provides the better performance for the system which is directly related to the exergy destruction and the total cost of the system. The total cost of the system is associated with the purchased equipment cost (PEC) of the components. It is observed that the exergoeconomic factor of evaporator, the ORC turbine and the pump decreases by increasing of the temperature difference for all working fluids which is given in Fig. 5.22(a), 5.22(c) and 5.22(d), respectively. This is related with the increasing of exergy destruction of these components. Contrary to this, the increase in the temperature difference causes in the increase of exergoeconomic factor of the condenser which is shown in Fig. 5.22(b). The outlet temperature of the ORC turbine is decreased by increasing of the temperature difference. Due to this reason, the exergy destruction in the condenser is decreased when the outlet temperature of the ORC turbine is decreased. Herein the exergoeconomic factor of the evaporator and the condenser are found to be significantly lower than the exergoeconomic factor of ORC turbine and the pump. This is due to the higher amount of the exergy destructions of the evaporator and the condenser. The lowest exergoeconomic factor is found for the condenser among the other components. It can be explained with the exergy efficiency of this component which is also the lowest one when compared to the others. Even if the maximum exergy efficiency is found for the ORC turbine, the highest exergoeconomic factor is found for the ORC pump. Because the total cost of the ORC pump is remarkably less than the ORC turbine.

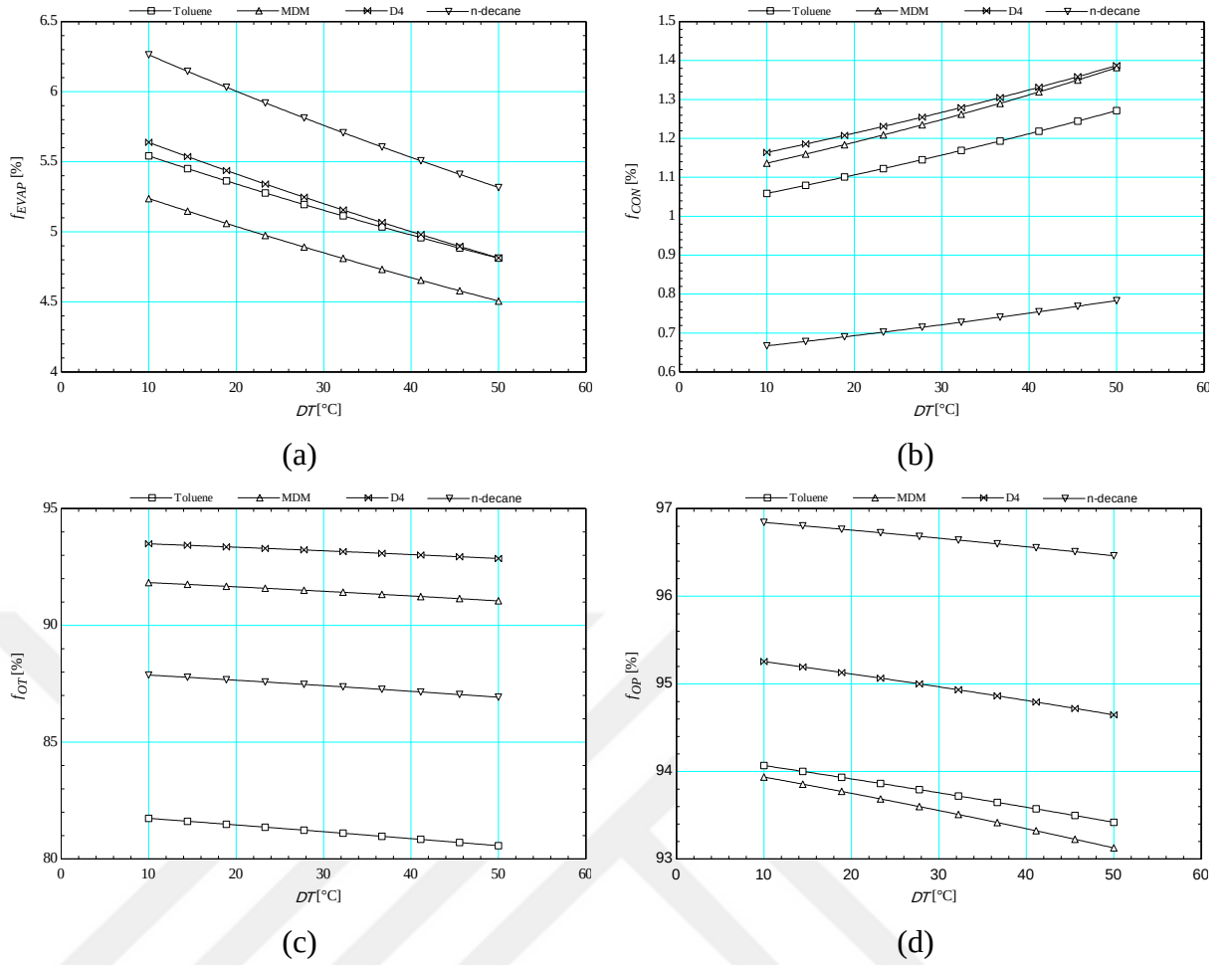


Figure 5.22. Effect of the temperature difference on the exergoeconomic factor of (a) the evaporator, (b) the condenser, (c) the ORC turbine (d) the ORC pump

The inlet temperature of the ORC turbine in evaporator, the mass flow rate of the working fluid in the condenser and the ORC turbine and the required power in the ORC pump are significant parameters for evaluating the exergoeconomic factor of the working fluids in the ORC system. In evaporator, the highest and the lowest exergoeconomic factors are evaluated for the n-decane and MDM, respectively which is shown in Fig. 5.22 (a). It can be explained that the critical temperature of n-decane and MDM are the highest and the lowest one, respectively among the other working fluids in the recent study. The same characteristic is shown in Fig. 5.22 (d) for the ORC pump. This case is related to the total cost of the component which is required to produce desired output from the ORC pump. The maximum and the minimum total cost of the ORC pump is evaluated for the n-decane and MDM,

respectively. The produced power is also an important parameter to evaluate the PEC of ORC turbine. The maximum power is provided by using the working fluid of toluene from the ORC system. Furthermore, the lowest exergy destruction in the ORC turbine is obtained by using the toluene in the system. Therefore the minimum exergoeconomic factor is evaluated for toluene which is shown in Fig. 5.22(c). Fig. 5.22(b) shows the increases of exergoeconomic factor of each working fluid contrarily to the other figures. Herein the lowest exergoeconomic factor is occurred for n-decane because of the highest exergy destruction.

Figures from 5.23 to 5.25 show the effects of the exhaust temperature of the heat source on the exergy efficiency as well as the net power output, the total cost rate and the exergoeconomic factor of components of the ORC system, respectively.

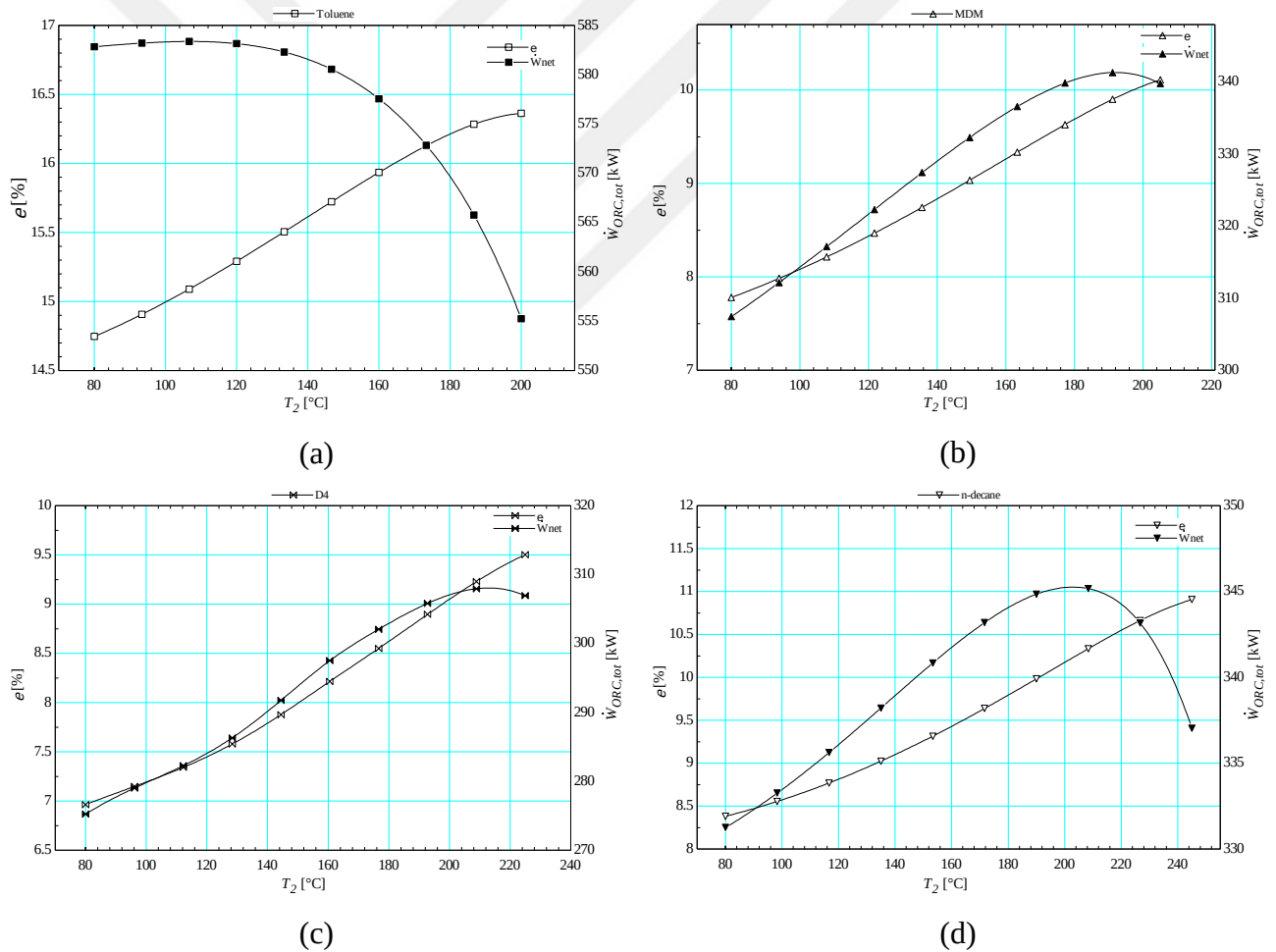


Figure 5.23. Effect of the exit temperature of the heat source on exergy efficiency and net power output for (a) toluene, (b) MDM, (c) D4 and (d) n-decane

In Figure 5.23, the effect of the exhaust temperature of the heat source on the exergy efficiency and the net power output is shown for each working fluid separately. While the exergy efficiencies increase continuously, the net power output increases firstly until producing the maximum net power and then begins to decrease with the increase in the exhaust temperature of the heat source. The optimum exhaust temperature of the heat source for toluene, MDM, D4 and n-decane are evaluated as 102 °C, 192 °C, 214 °C and 204 °C, respectively.

Figure 5.24 demonstrates the total cost rate changes with respect to the exhaust temperature of the heat source. As can be seen from the Fig. 5.24, the maximum and minimum total cost rate of the system are found to be D4 and n-decane, respectively at same available heat source. The reason for this can be explained with the inlet temperature of the turbine which is related with critical temperature. D4 and n-decane have the lowest and the highest critical temperature values among the working fluids.

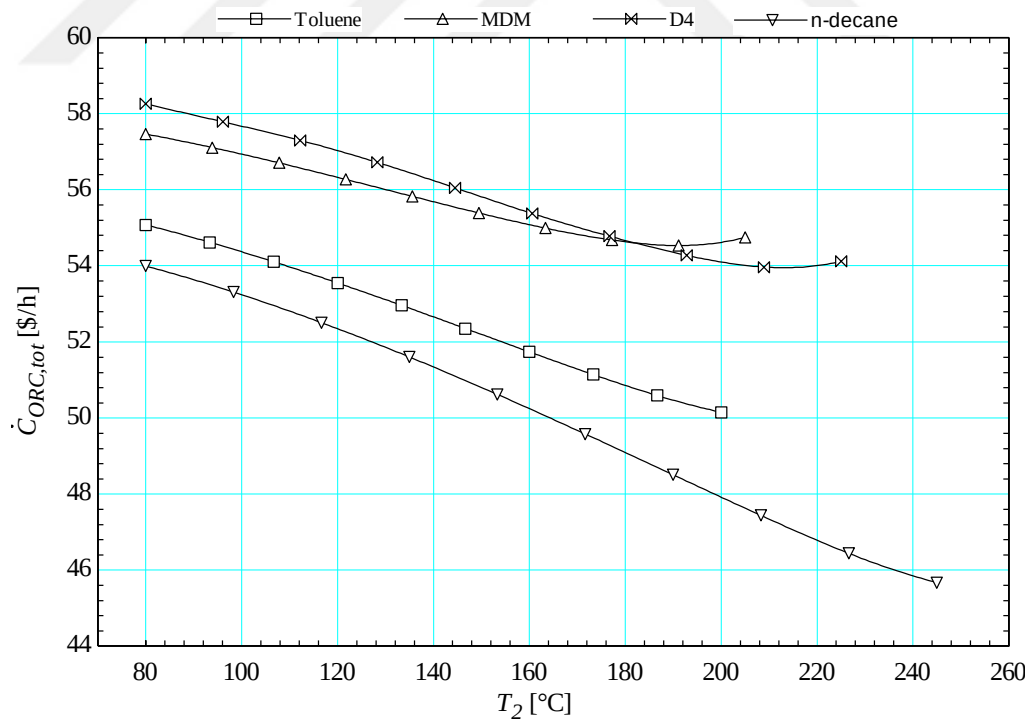


Figure 5.24. Effect of the exit temperature of the heat source on the total cost rate

Figure 5.25 shows the effect of the exit temperature of the heat source on the exergoeconomic factor of each component.

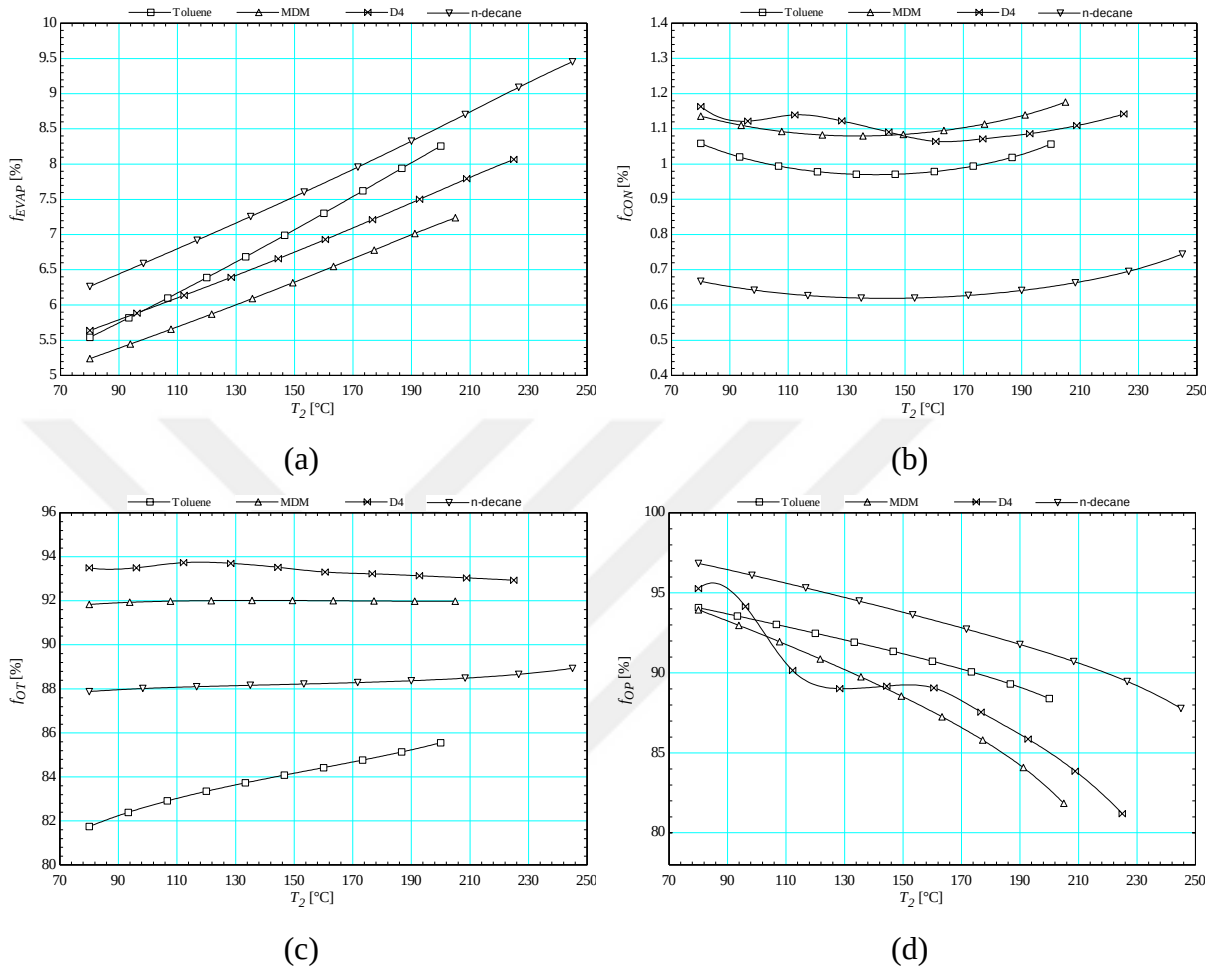


Figure 5.25. Effect of the exit temperature of the heat source on exergoeconomic factor of (a) the evaporator, (b) the condenser, (c) the ORC turbine (d) the ORC pump

It can be seen from Fig. 5.25(a), the exergoeconomic factor increases with the increase in the exit temperature of the heat source. The slope of toluene is the higher one than the other working fluids due to the significant reduction of the exergy destruction in the evaporator regarding to the increase in the exit temperature of the heat source. The exergoeconomic factor of the condenser first decreases then begins to increase as can be seen from Fig. 5.25(b). It is clearly shown in Fig. 5.25(c), the

exergoeconomic factor of the ORC turbine can be improved by using toluene while the exergoeconomic factor of the ORC turbine almost remains constant for the other working fluids. The exergoeconomic factor of the ORC pump decreases sharply with the increase in the exit temperature of the heat source as can be seen in Fig. 5.25(d). The increase in the exit temperature of the heat source caused the increase in required power for the pump. Due to this fact, the increase in PEC of the ORC pump affects the exergoeconomic factor of the ORC pump negatively.

It is obviously seen from Fig. 5.26, that the increase in the inlet temperature of the Kalina turbine causes a decrease in the net power output. This result can be explained with the mass flow rate of the ammonia-water mixture. Since the amount of heat transfer from the Kalina generator is the same, the increase in the inlet temperature of the Kalina turbine causes the decrease in the mass flow rate of ammonia-water mixture which also affect the net power output negatively. This result also causes the energy and the exergy efficiencies to remain almost constant. Figure 5.27 shows the relation between the net power output and the total cost rate when the turbine inlet temperature increases. It is clearly seen that the total cost rate increases by increasing the turbine inlet temperature. As a result of this, it can be said that the increase in the inlet temperature of Kalina turbine causes increase in the total capital cost of the components.

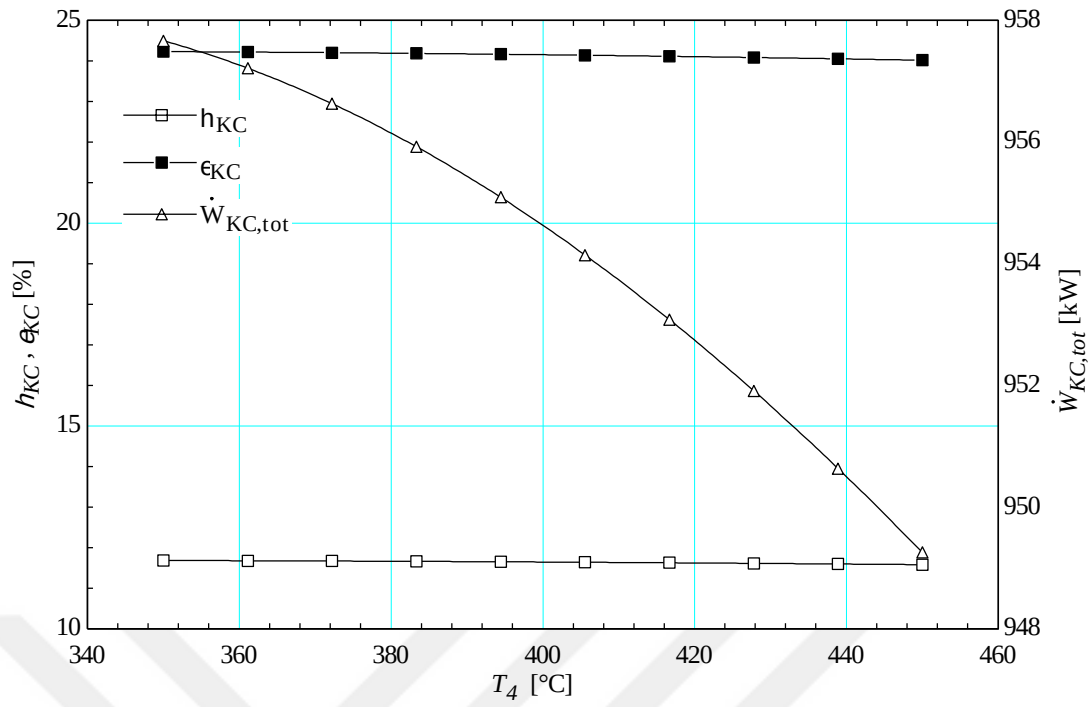


Figure 5.26. Effect of the inlet temperature of the Kalina turbine on the energy and exergy efficiencies and the net power output

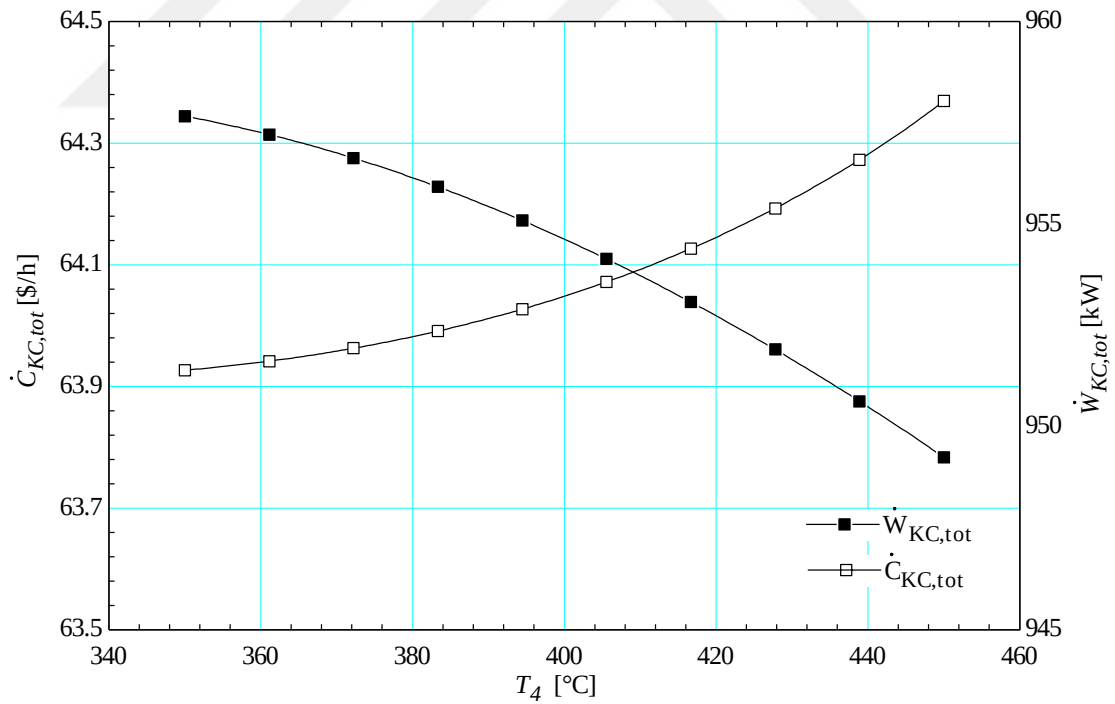


Figure 5.27. Effect of the inlet temperature of the Kalina turbine on the net power output and the total cost rate

The effect of the exit temperature of the exhaust gas from the evaporator on the energy and the exergy efficiencies and the net power output of GT-ORC is presented in Fig. 5.28. It can be said that the increase in the exit temperature of the exhaust gas causes the decrease in the energy and exergy efficiencies as well as the net power output. As the exit temperature of the exhaust gas increases, the amount of the heat transfer from the evaporator decreases and the net power output decreases. As a result of this, the energy and the exergy efficiencies decrease. Herein, the decrease in the total cost rate can be explained with the increasing of the total capital cost of the ORC system. Figure 5.29 shows the effect of the exit temperature of the exhaust gas from the evaporator on the net power output and the total cost rate of the GT-ORC system.

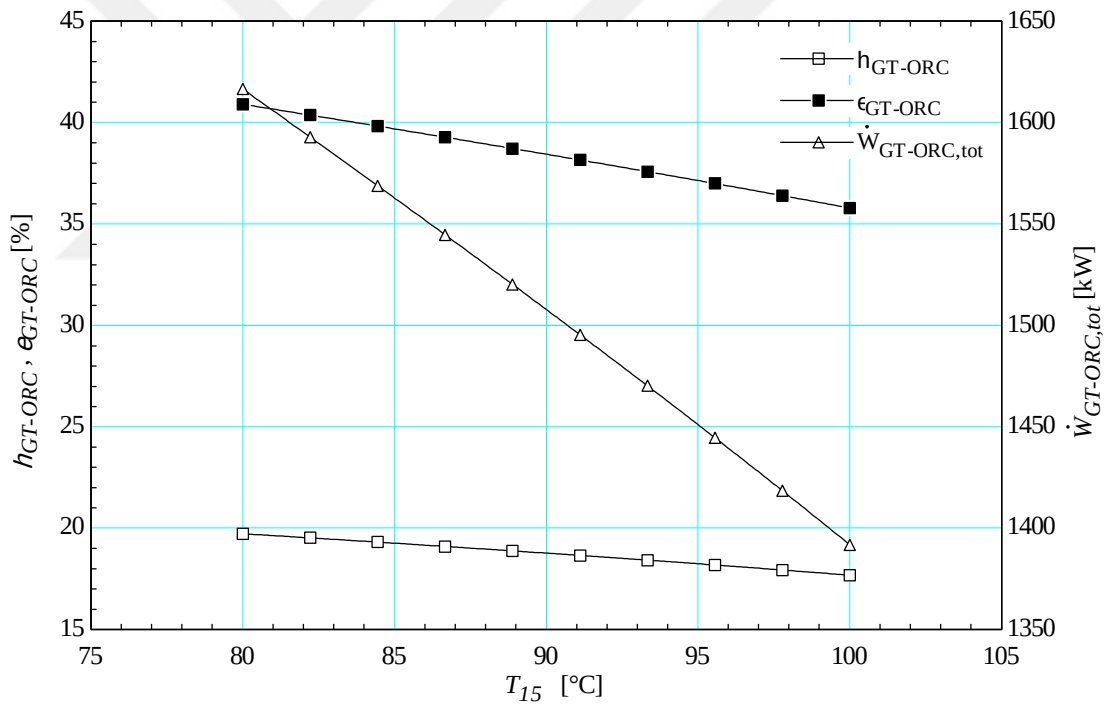


Figure 5.28. Effect of the T_{15} on the energy and exergy efficiencies and the net power output of GT-ORC

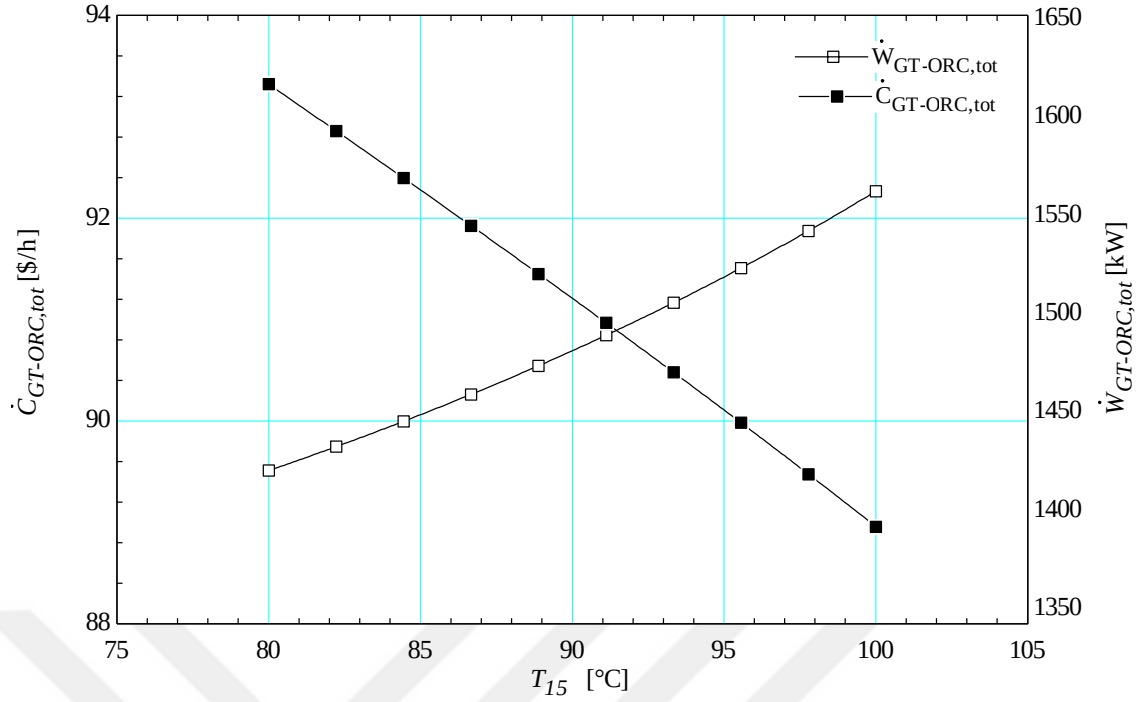


Figure 5.29. Effect of the T_{15} on the net power output and the total cost rate of GT-ORC

Figures 5.30 and 5.31 show the effect of the dead state temperature on the energy efficiency, the exergy efficiency, the net power output and the total cost rate of GT cycle, respectively. The energy and exergy efficiencies and the net power output of the GT cycle have the same behavior with respect to the dead state temperature. The energy and exergy efficiencies and net power output of the GT cycle decrease when the dead state temperature increases due to the performance reduction of subcomponents during the process. It is also seen that the total cost rate of GT cycle decreased almost linearly with increasing of the dead state temperature because of the total capital cost of the sub-components.

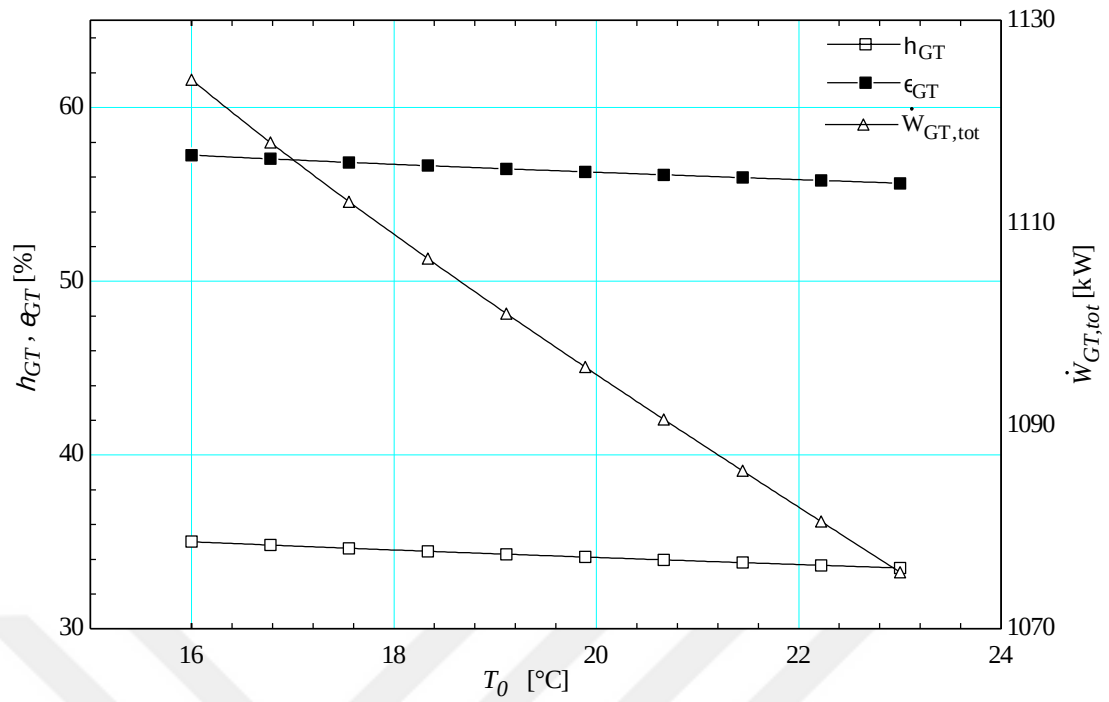


Figure 5.30. Effect of the T_0 on the energy and exergy efficiencies and the net power output of the GT

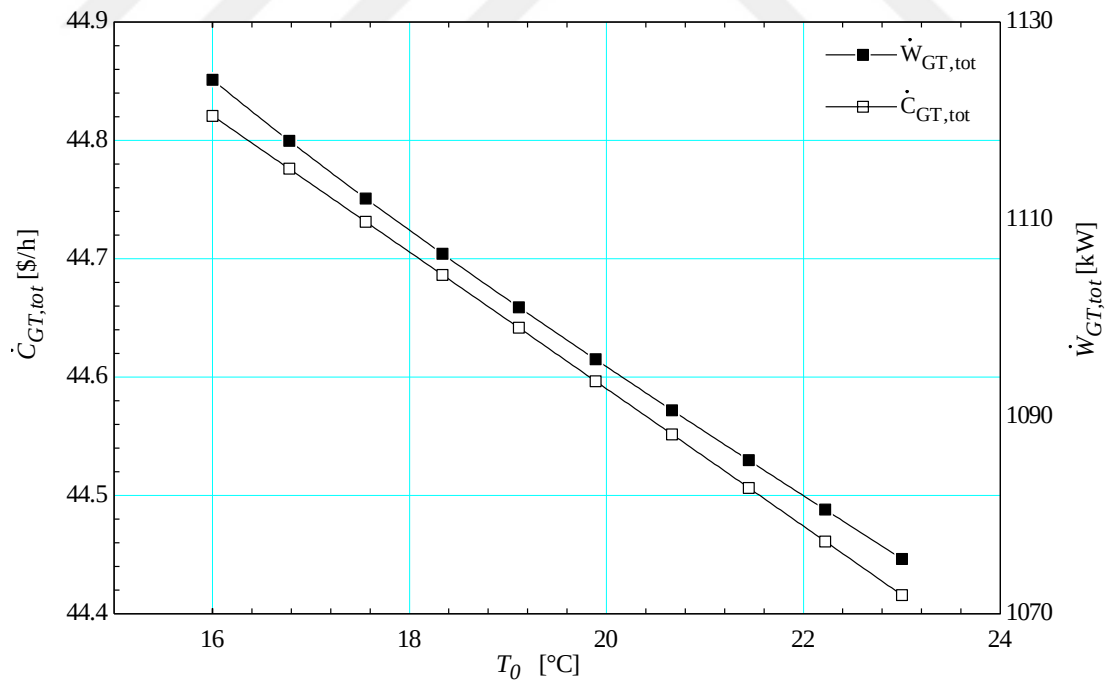


Figure 5.31. Effect of the T_0 on the net power output and the total cost rate of the GT

5.4 Effect of the Pressure on Models

The effect of the inlet pressure of turbine for GT and KAL are also taken into consideration as a part of parametric study. The effect of the inlet pressure of the turbine on the energy and exergy efficiencies as well as the net power output of the GT and KC are given in Fig. 5.32 and 5.33. The energy and exergy efficiencies are increased with regarding to increase in the net power output of the GT cycle which are occurred when the inlet pressure of gas turbine is raised. But the increment of the net power output can be evaluated as very low with a value of almost 15 kW. On the other hand, the increase in the inlet pressure of the KC causes a decrease in the mass flow rate of ammonia-water mixture and consequently the net power output. Because the amount of heat transfer from the Kalina generator is kept constant when the pressure increases. Thus, it can be deduced that the increase in the inlet pressure of the turbines does not affect the energy and exergy efficiencies and also net power output significantly.

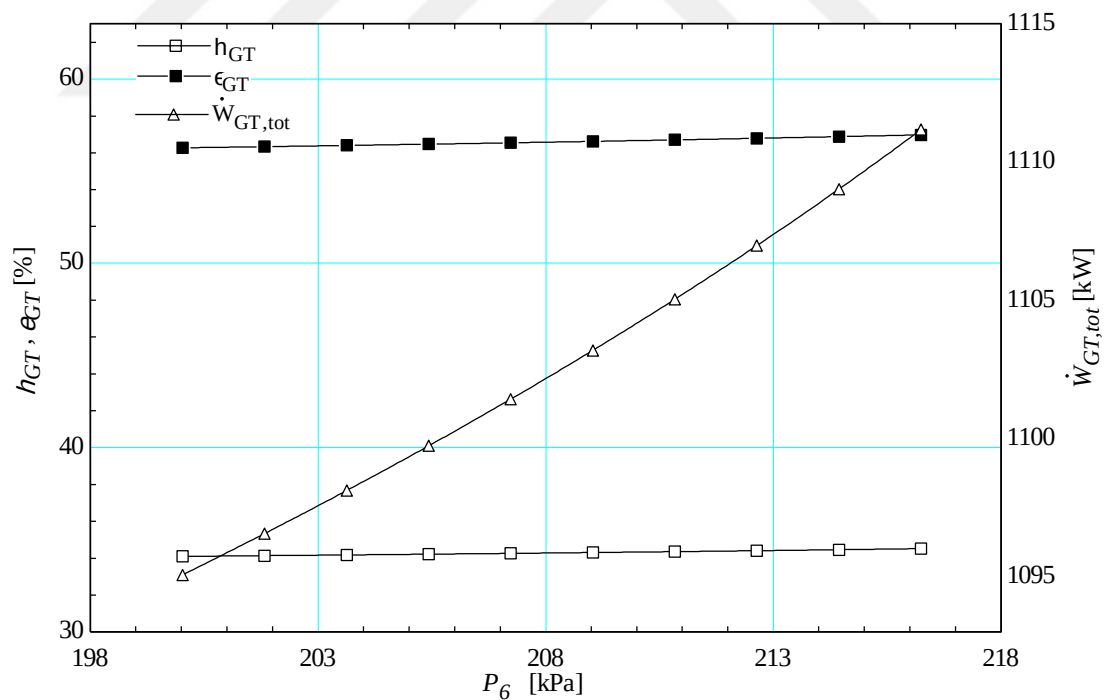


Figure 5.32. Effect of the P_6 on the energy and exergy efficiencies and the net power output of the GT

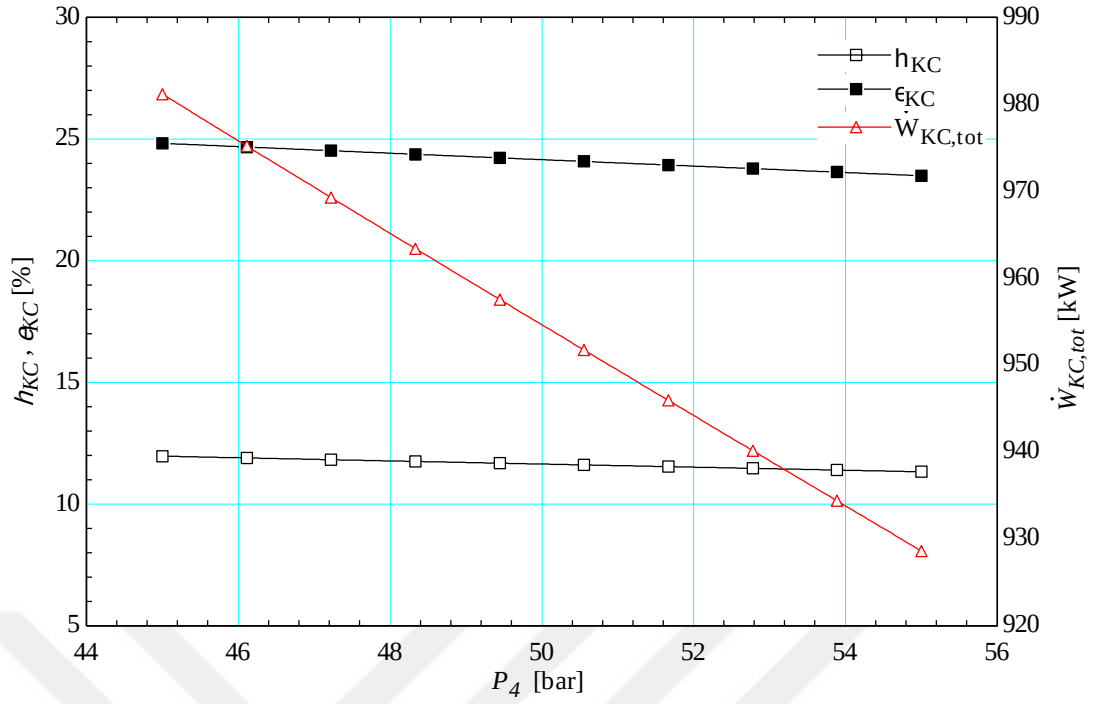


Figure 5.33. Effect of the P_4 on the energy and the exergy efficiencies and the net power output of KC

It is clearly seen that from Fig. 5.34, the total cost rate of the GT is almost constant during the parametric study due to the slight increase of the net power output. In other words, this rate of increase is very small such that it can be said to be no more influence of the inlet pressure on the total cost rate of GT cycle. Besides of this, increase in the inlet pressure of Kalina turbine caused a decrease in the total cost rate of the KC which is given in Fig. 5.35. This result can be attributed that the decrease in the net power output causes a decrease in the total capital cost of the subcomponents of the Kalina cycle.

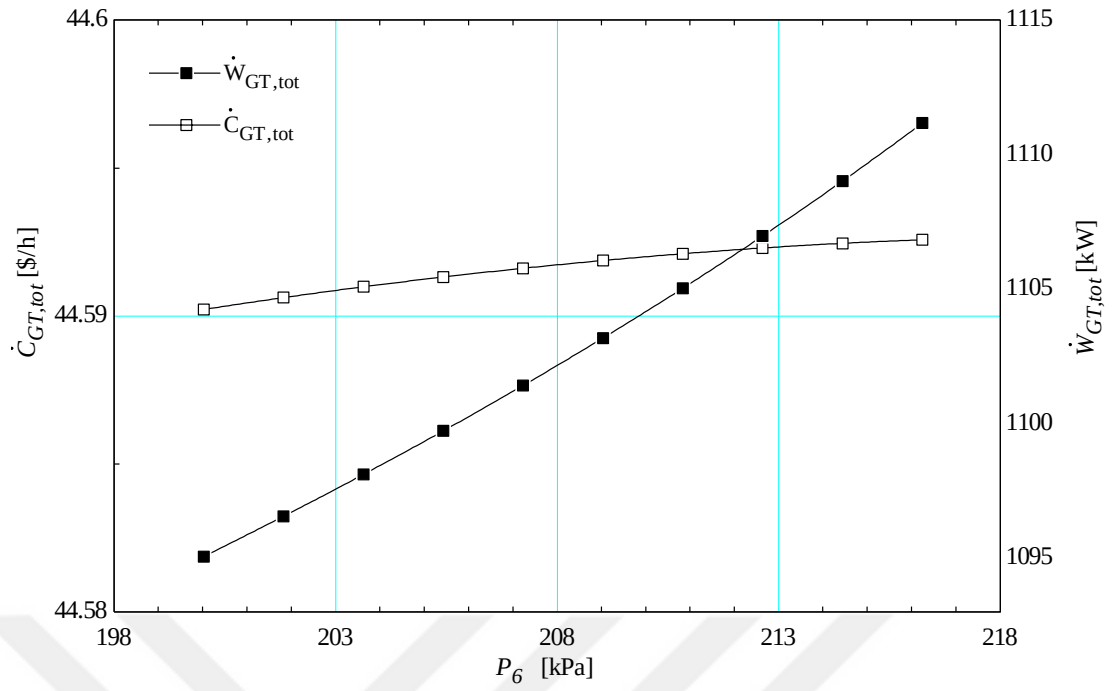


Figure 5.34. Effect of the P_6 on the total cost rate and the net power output of the GT

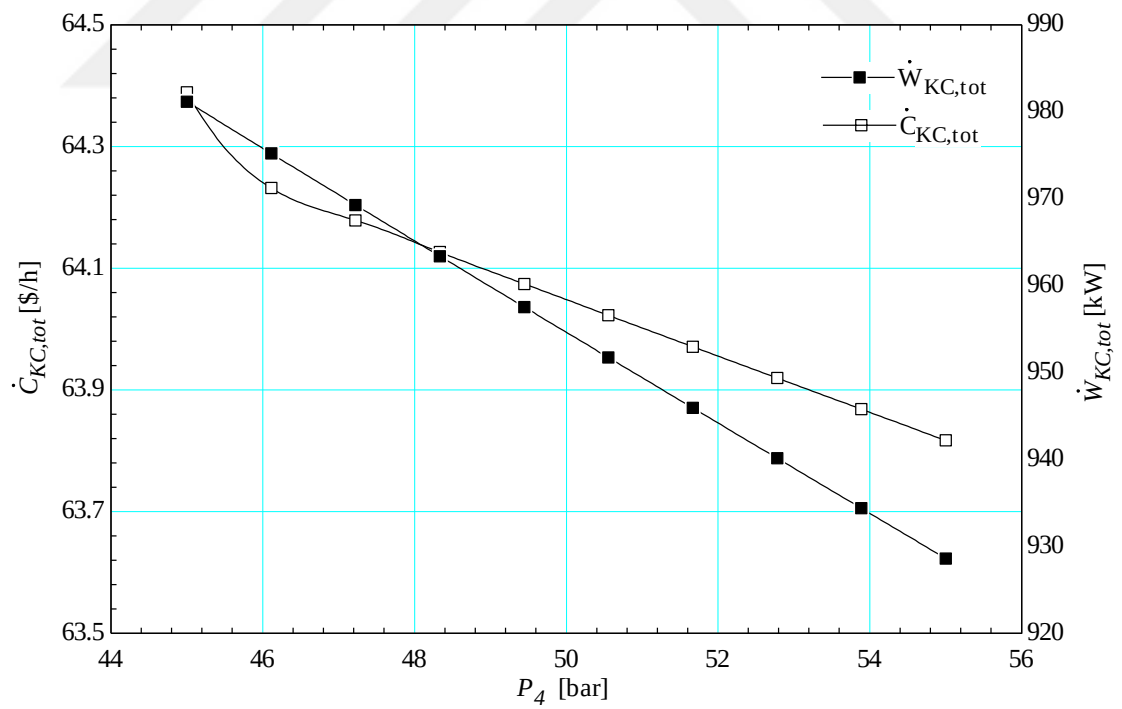


Figure 5.35. Effect of the P_4 on the net power output and the total cost rate of KC

5.5 Effect of the Mass on Models

The energy efficiency, exergy efficiency and the net power output for the KC during the change of the ammonia concentration in range between 0.7 to 0.9 is given in Fig 5.36. In addition to this, the same performance parameters of the GT-KAL cycle is performed by taking into consideration the mass flow rate of ammonia-water solution which is shown in Fig. 5.37. It is obviously seen that the increase in the ammonia concentration affects positively the thermodynamic performance of KC because of the NH_3 has high productivity in comparison with water at same thermophysical conditions. The increase in the mass flow rate of ammonia-water solution causes an increase in the thermodynamic performance of the GT-KAL. Because the working fluid used in the cycle is directly related to the power production for the system. Due to this reason, the increase in the mass flow rate of the working fluid is an advantage for systems. Thermodynamically, it is possible to increase the net power production depending on the ammonia concentration or mass flow rate of working fluid, but the thermoeconomic indicators in Fig. 5.38 and Fig. 5.39 show that the total cost ratio also increases rapidly. Therefore, if the figures from 5.36 to 5.39 are taken into consideration, it is obviously seen that a thermodynamic and thermoeconomic optimization are simultaneously needed for a cycle to achieve the best solution.

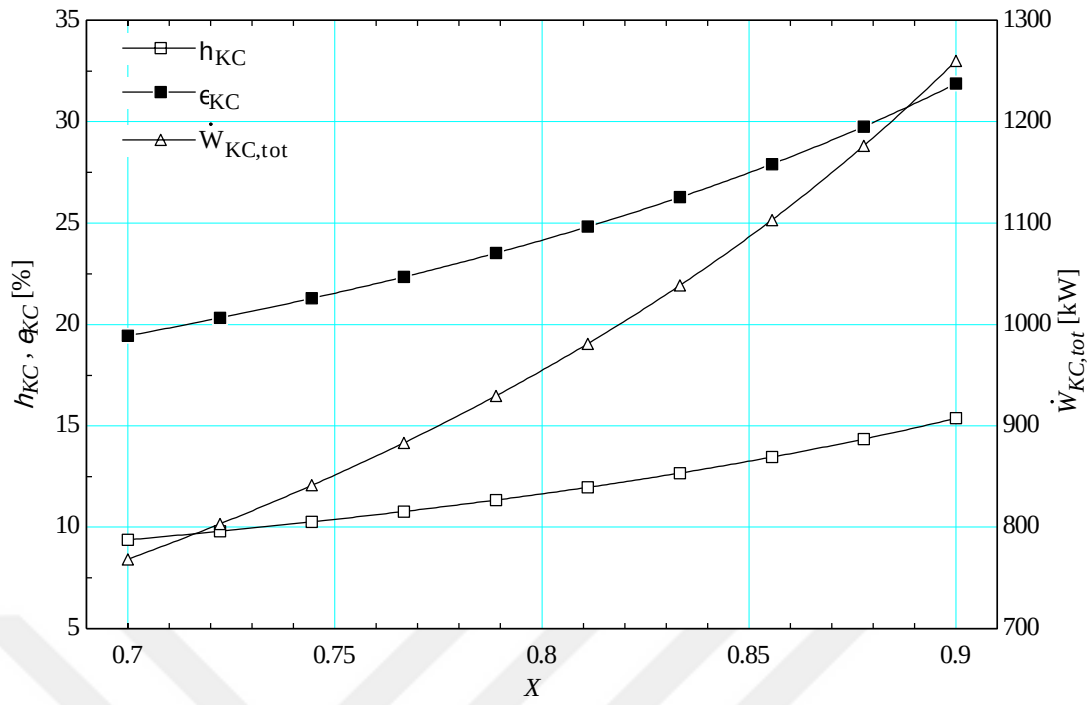


Figure 5.36. Effect of the X on the energy and exergy efficiencies and the net power output of KC

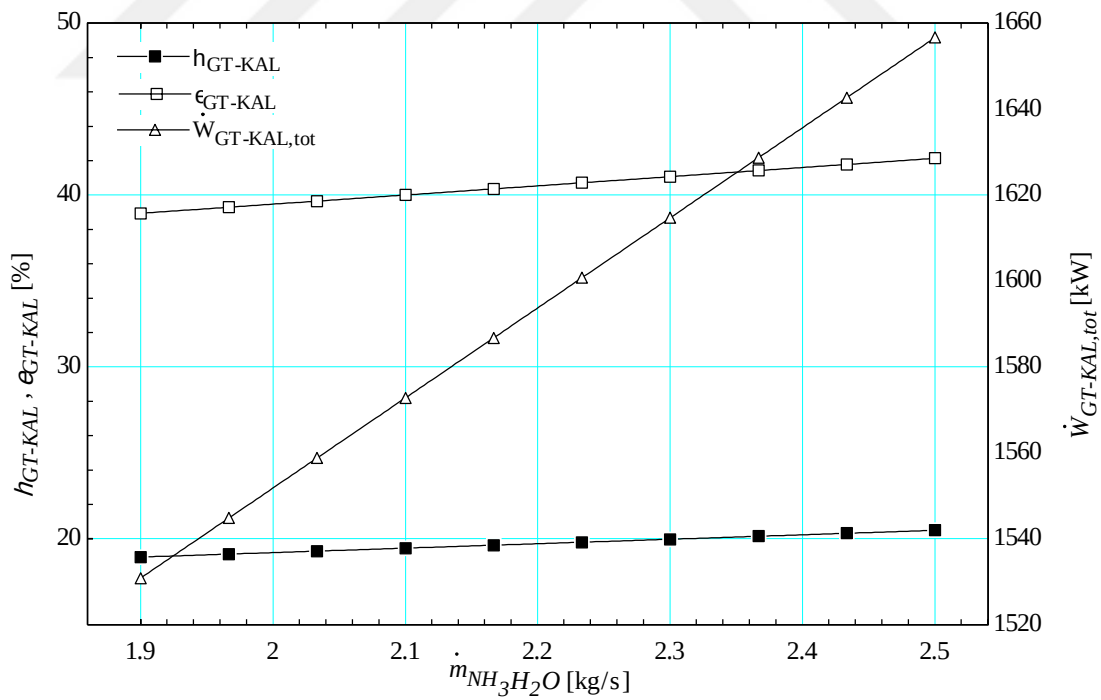


Figure 5.37. Effect of the $\dot{m}_{NH_3H_2O}$ on energy and exergy efficiencies and net power output of GT-KAL

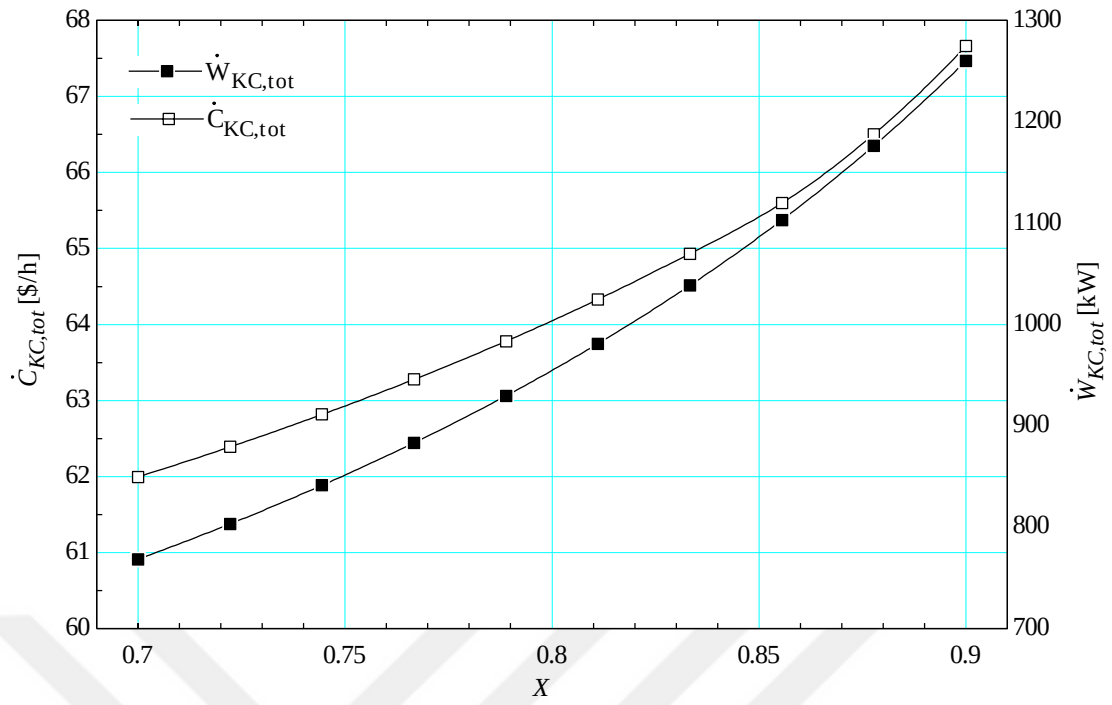


Figure 5.38. Effect of the X on the net power output and the total cost rate of KC

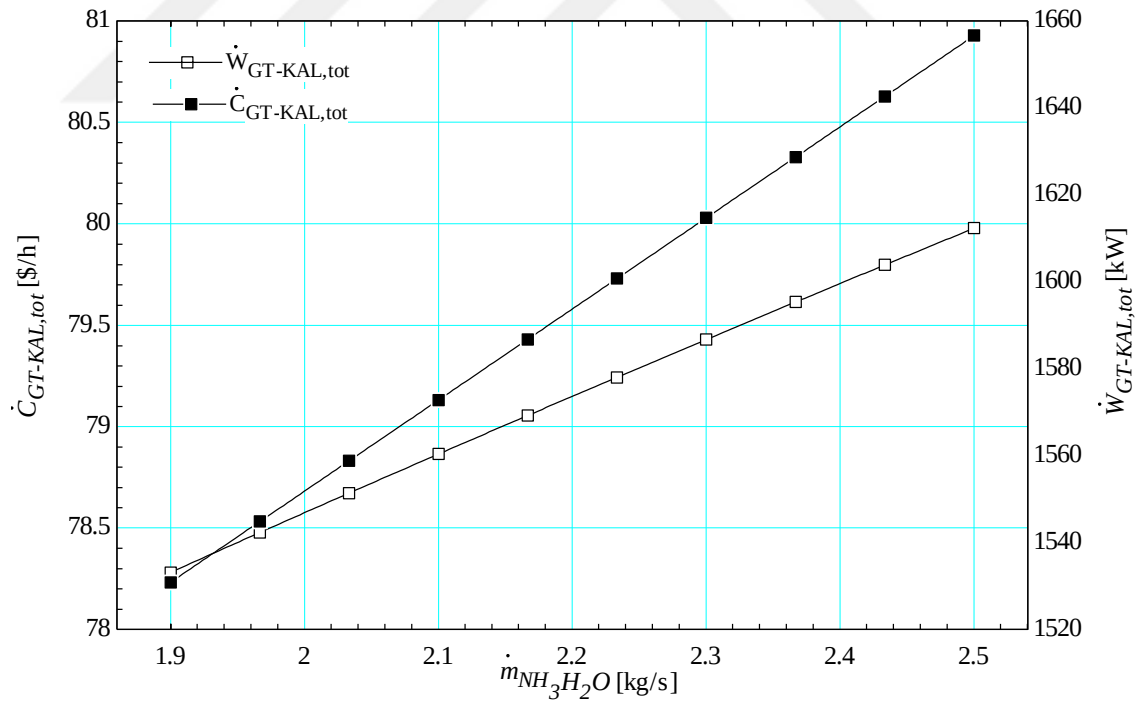


Figure 5.39. Effect of the $\dot{m}_{NH_3H_2O}$ on the net power output and the total cost rate of GT-KAL

5.6 Optimization

In all models except for ORC, the optimization of the systems are carried out by considering two objective functions which are the total cost rate and the exergy efficiency of the model. In ORC, the objective functions are selected as total cost rate and the net power output of the cycle in order to compare the selected working fluid performances clearly. Because the exergy efficiency of ORC is generally found as very low compared to other cycles. The net power output is selected as a more suitable parameter for benchmarking of four different working fluids used in the ORC. The optimization flow charts of the models are represented from Fig. 5.40 to 5.44. In addition to this, throughout all models, the population size, the Pareto fraction and the number of generations are set to 120, 0.8 and 1000, respectively due to the default option in MATLAB software [148].

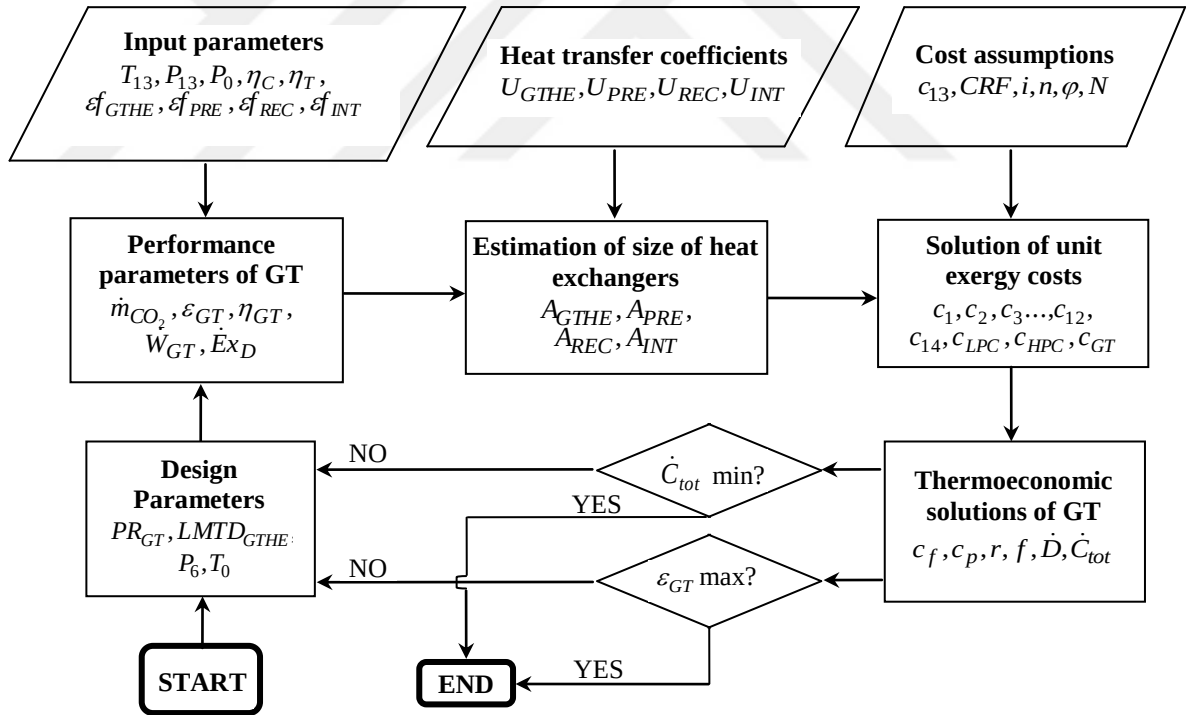


Figure 5.40. Optimization flow chart of the GT

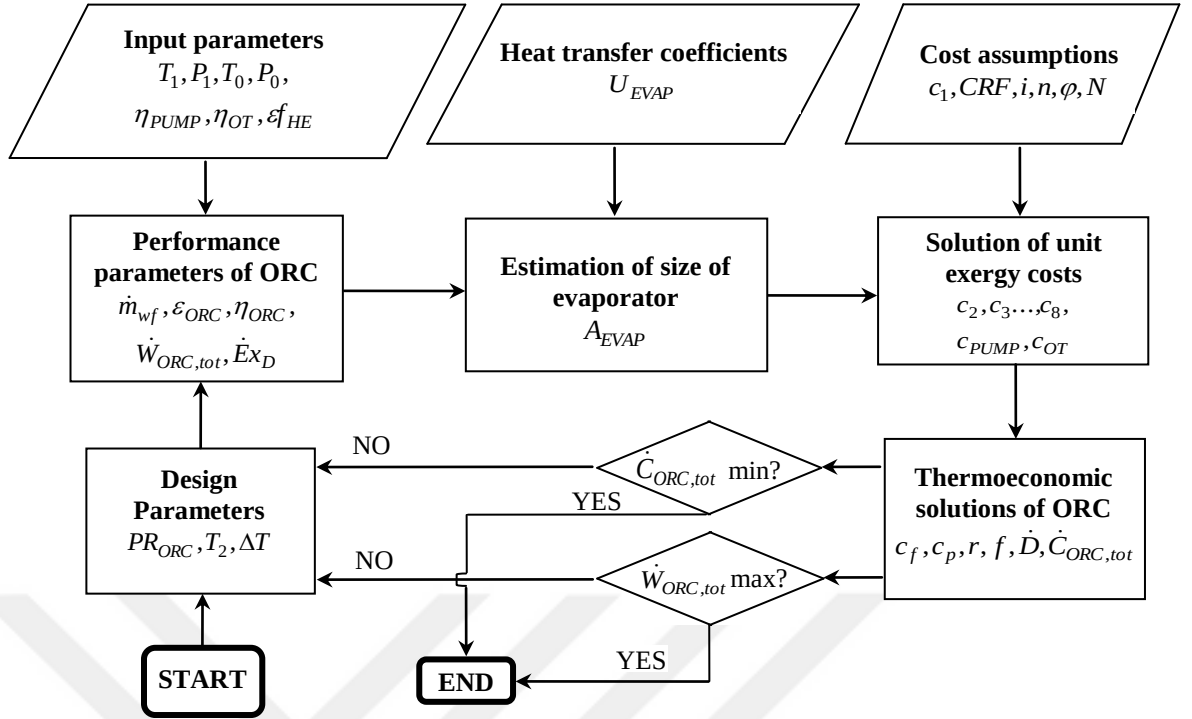


Figure 5.41. Optimization flow chart of the ORC

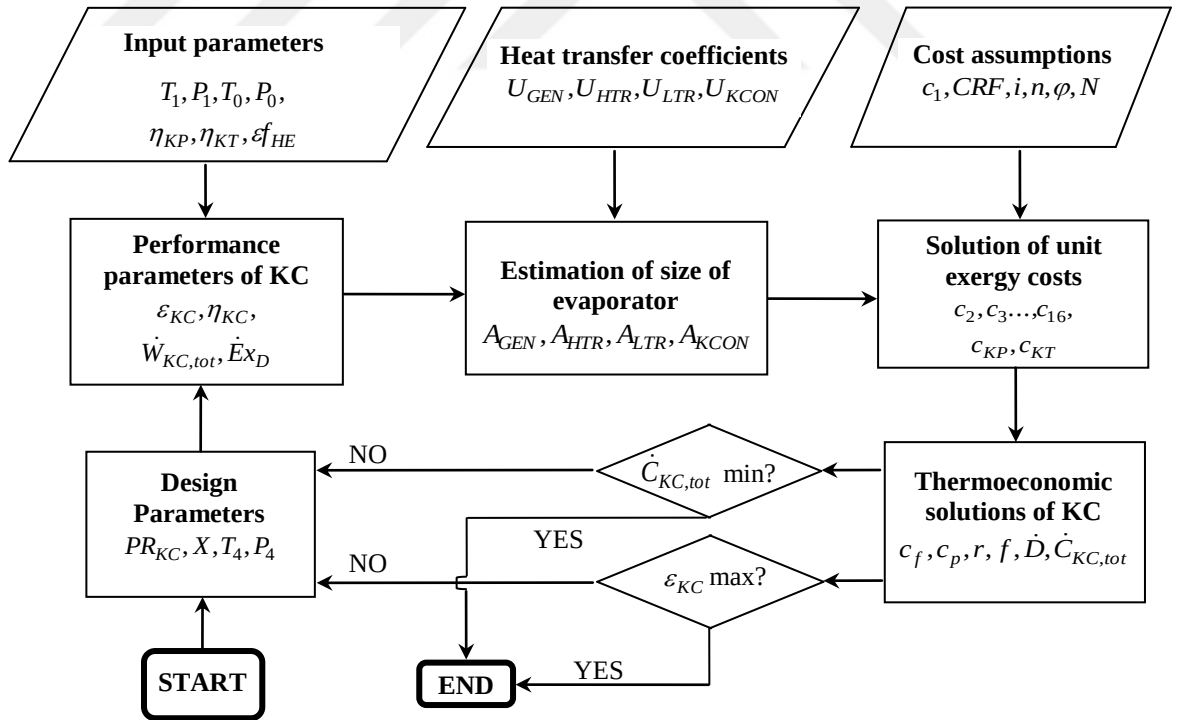


Figure 5.42. Optimization flow chart of the KC

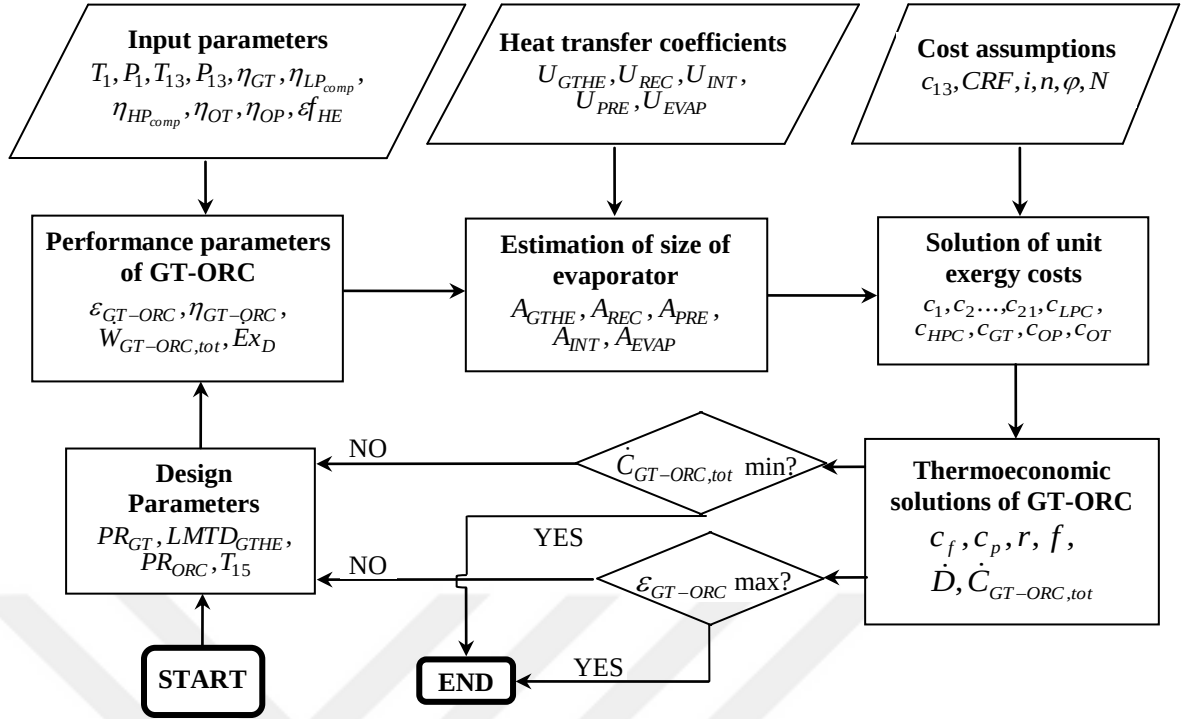


Figure 5.43. Optimization flow chart of the GT-ORC

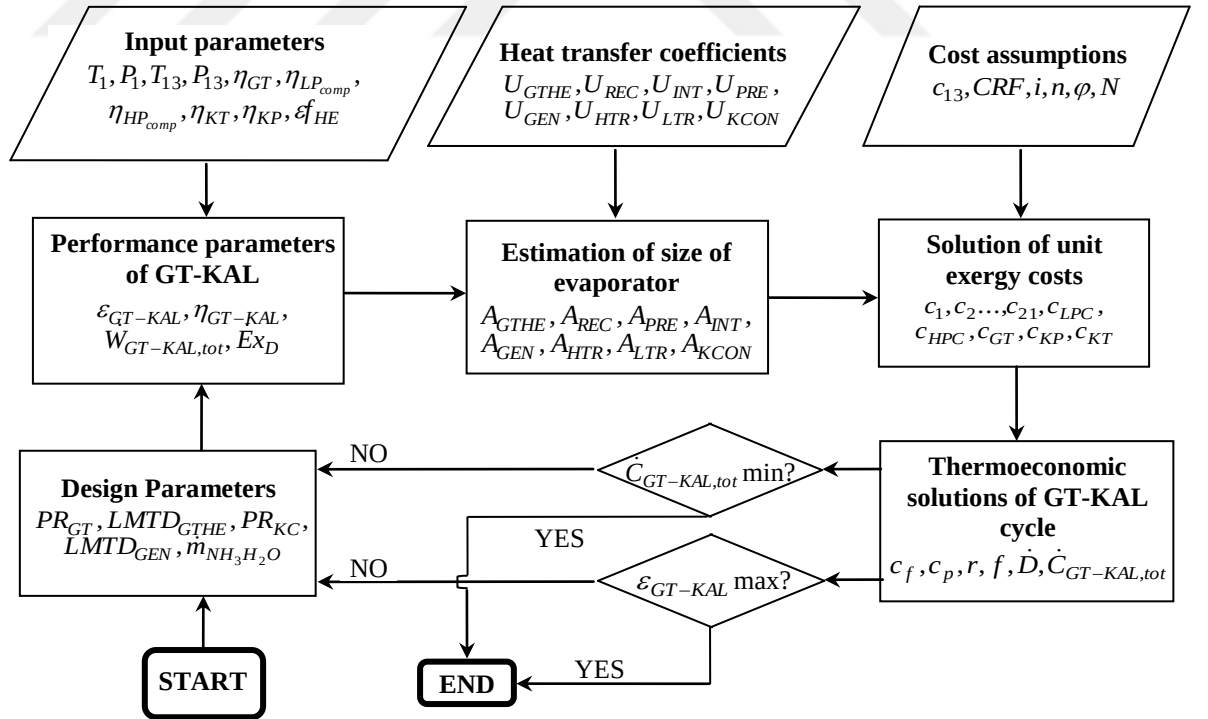


Figure 5.44. Optimization flow chart of the GT-KAL

The total cost rate $\dot{C}_{tot}$ (\$/h) is calculated by the summation of the cost rate difference between inlet and exit stream of the exhaust gas and the total capital cost rate of the system. The exergy efficiency of the system, ε is calculated from the Eqn. 4.2. The optimization of the objective functions is evaluated using selected parameters for each model in the specified range which is given in Table 5.1 to evaluate the optimal solution of the objective functions.

Table 5.1. The ranges of the decision parameters of models

GT	PR_{GT}	$2.5 \leq PR_{GT} \leq 4$	
	$P_6(\text{bar})$	$185 \leq P_6 \leq 320$	
	$T_0(^{\circ}\text{C})$	$16 \leq T_0 \leq 23$	
	$LMTD_{GTHE}$	$9 \leq LMTD_{GTHE} \leq 13$	
ORC	PR_{ORC}	$3 \leq PR_{ORC} \leq 7$	
	$\Delta T(^{\circ}\text{C})$	$10 \leq \Delta T \leq 50$	
	$T_2(^{\circ}\text{C})$	Toluene	$80 \leq T_2 \leq 200$
		MDM	$80 \leq T_2 \leq 205$
		D4	$80 \leq T_2 \leq 225$
		n-decane	$80 \leq T_2 \leq 240$
KC	PR_{KC}	$2.5 \leq PR_{KC} \leq 3.5$	
	X	$0.7 \leq X \leq 0.9$	
	$T_4(^{\circ}\text{C})$	$350 \leq T_4 \leq 450$	
	$P_4(\text{bar})$	$45 \leq P_4 \leq 55$	
GT-ORC	PR_{GT}	$2.3 \leq PR_{GT} \leq 3.3$	
	$LMTD_{GTHE}$	$7.6 \leq LMTD_{GTHE} \leq 11.6$	
	PR_{ORC}	$3 \leq PR_{ORC} \leq 7$	
	$T_{15}(^{\circ}\text{C})$	$80 \leq T_{15} \leq 100$	
GT-KAL	PR_{GT}	$2.3 \leq PR_{GT} \leq 3.3$	
	$LMTD_{GTHE}$	$7.6 \leq LMTD_{GTHE} \leq 11.6$	
	PR_{KAL}	$2.5 \leq PR_{GT} \leq 3.5$	
	$LMTD_{GEN}$	$2 \leq LMTD_{GEN} \leq 9.7$	
	$\dot{m}_{NH_3H_2O}$	$1.9 \leq \dot{m}_{NH_3H_2O} \leq 2.5$	

Figure 5.45, 5.48, 5.49 and 5.50 show the Pareto frontier of the GT, KC, GT-ORC and GT-KAL for multi-objective function solution with respect to the best exergy efficiency and the total cost rate. It is clearly seen that from the figures, there is an inverse proportion between the thermodynamic and economic performance of the systems. The Pareto frontier of for each working fluid in ORC by using the multi-objective function solution regarding to the best net power output and the total cost rate is represented in Fig 5.46(a-d) and 5.47.

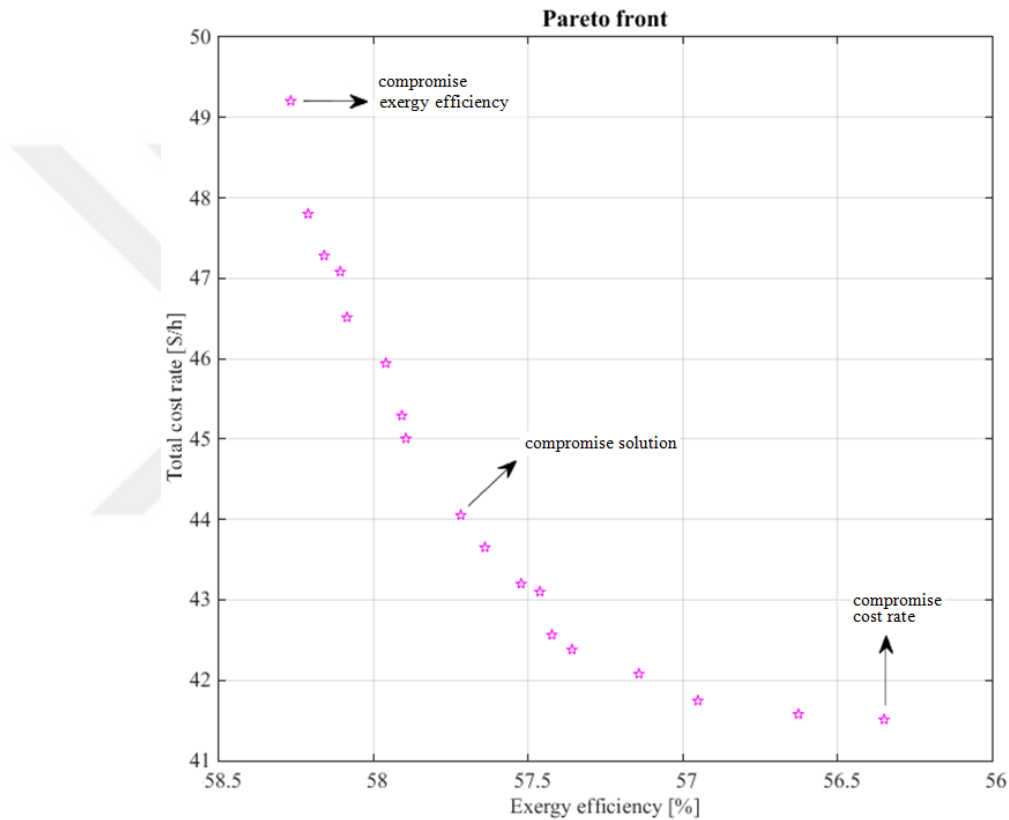


Figure 5.45. Pareto frontier for optimum solution of the GT

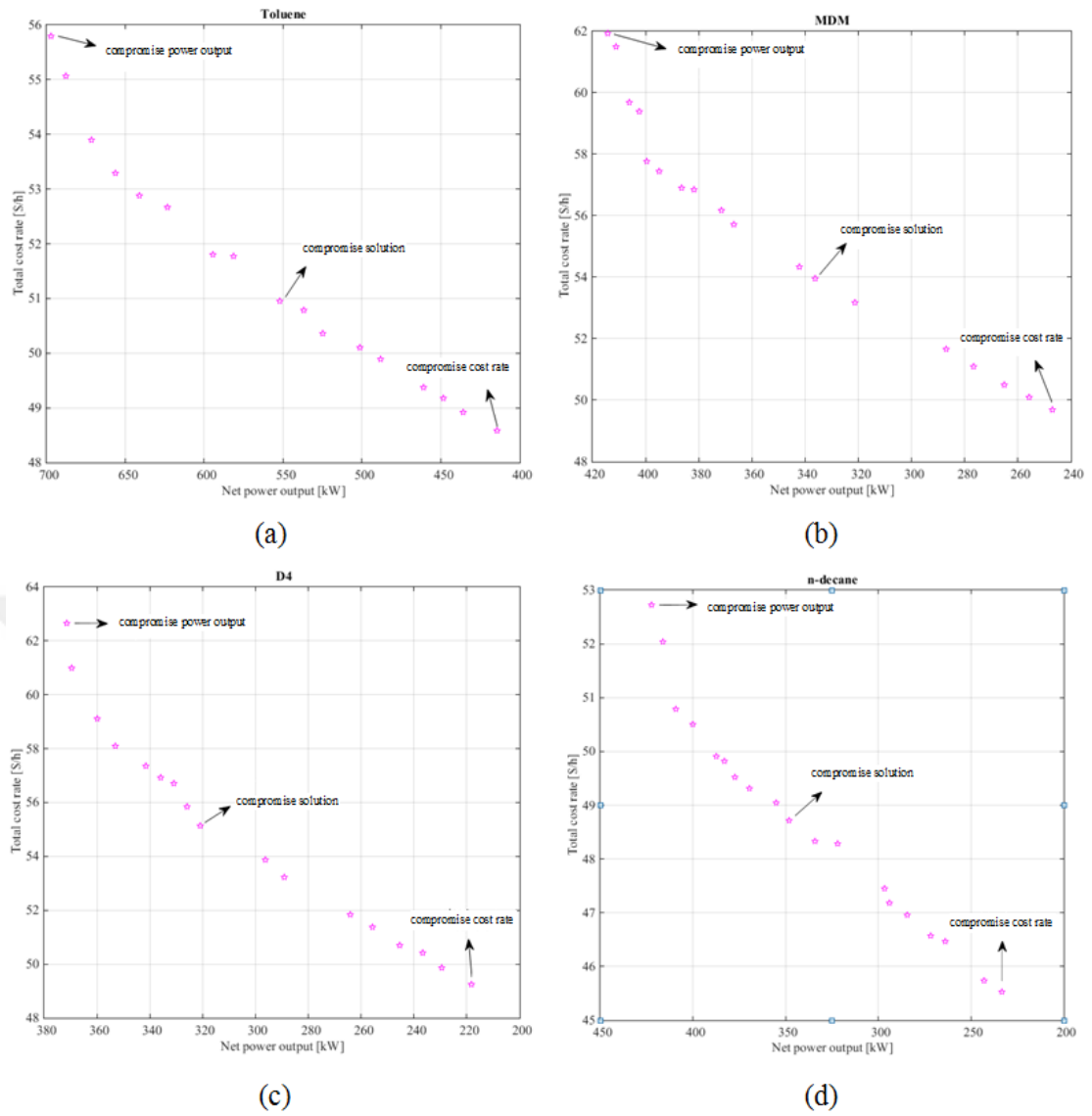


Figure 5.46. Pareto frontier for optimum solution of (a) toluene, (b) MDM, (c) D4 (d) n-decane in the ORC

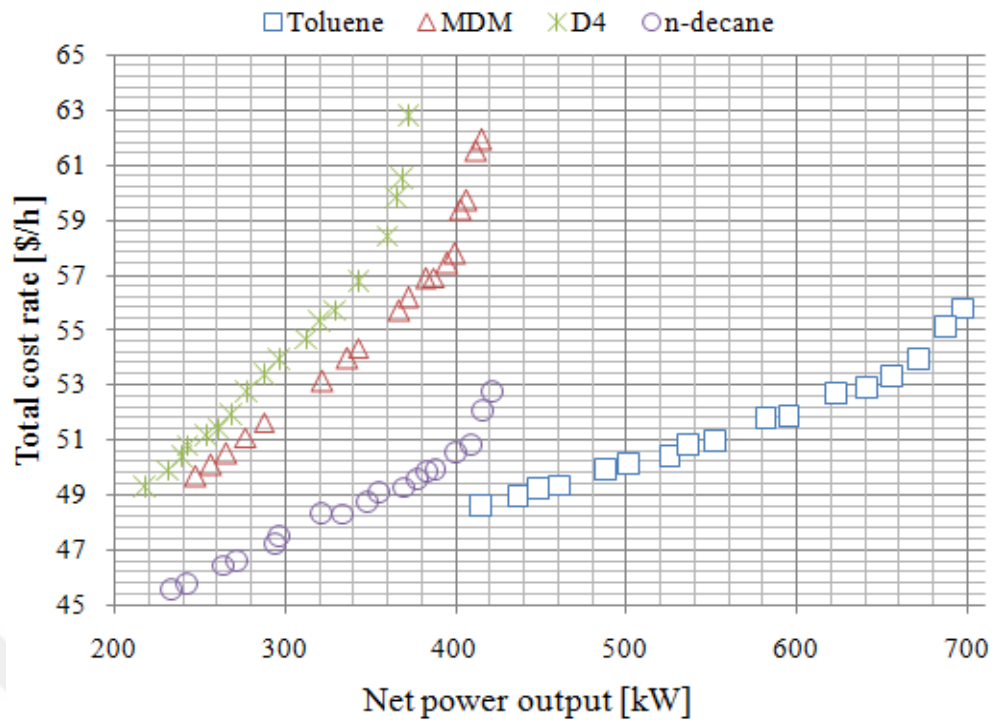


Figure 5.47. Pareto frontier results of all working fluids

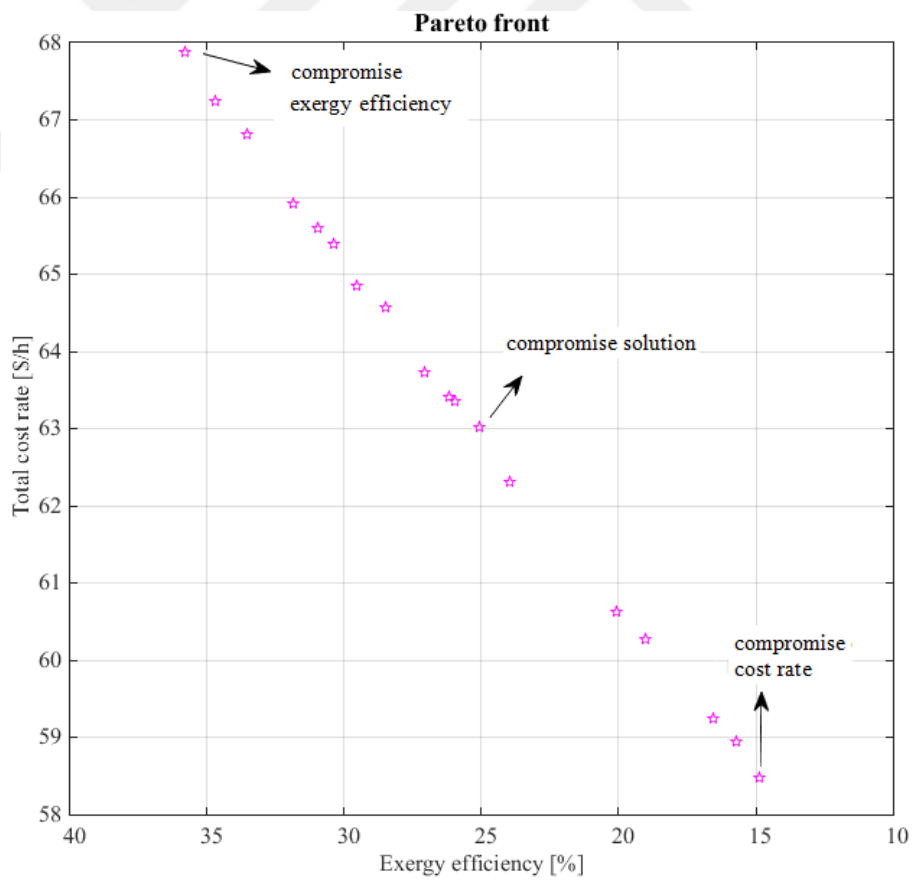


Figure 5.48. Pareto frontier for optimum solution of the KC

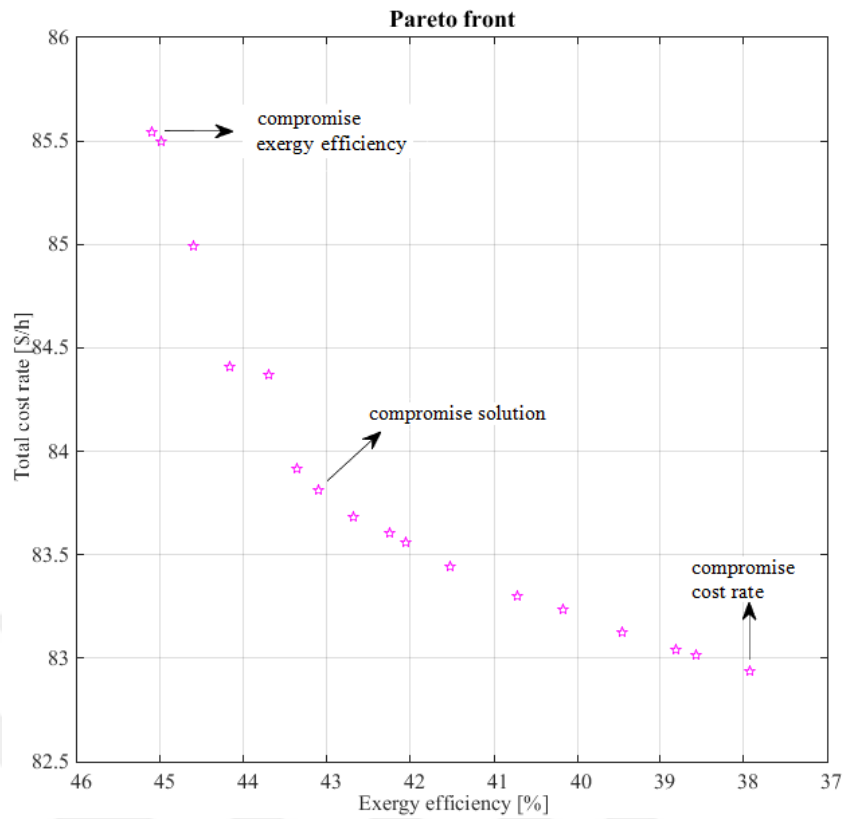


Figure 5.49. Pareto frontier for optimum solution of GT-ORC

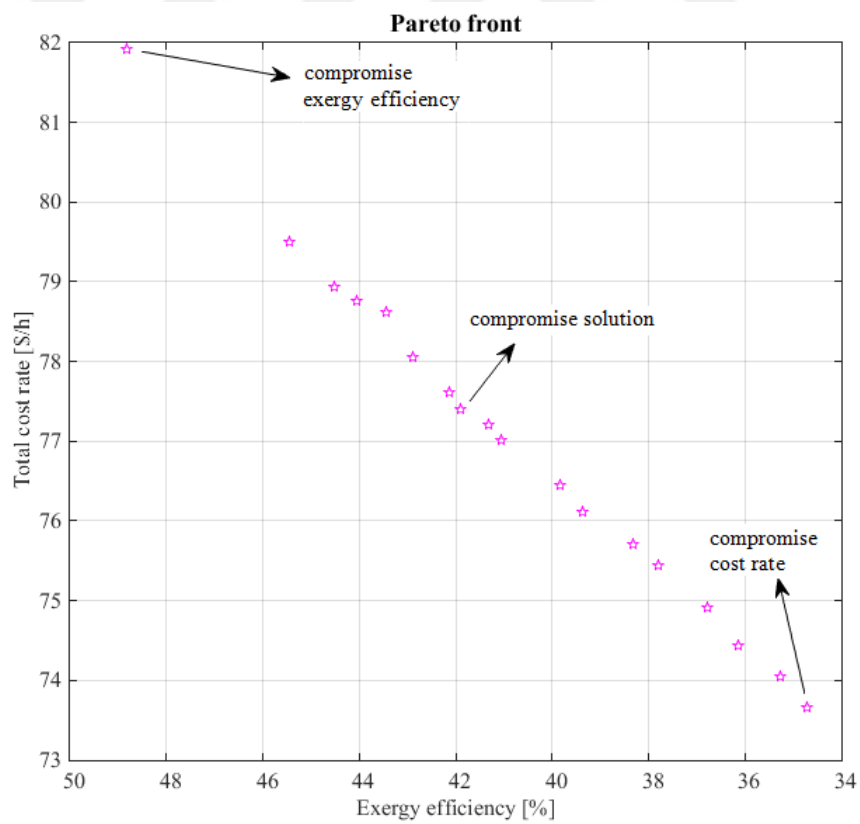


Figure 5.50. Pareto frontier for optimum solution of the GT-KAL

Decision stage and optimum design of the decision parameters of the models are given in Table 5.2. The optimization results for all models are carried out by using the decision parameters regarding to their optimum values. The NSGA-II results of the models are tabulated in Table 5.3.

Table 5.2. Decision stage and optimum design of the decision parameters of models

	Decision parameters	Decision stage	Optimum design
GT	PR_{GT}	2.703	2.633
	P_6 (bar)	200	207.7
	T_0 (°C)	20	16.91
	$LMTD_{GTHE}$	11.05	11.34
ORC (Toluene)	PR_{ORC}	5	4.30
	ΔT (°C)	15	13.73
	T_2 (°C)	140	141.61
ORC (MDM)	PR_{ORC}	5	4.76
	ΔT (°C)	15	14.83
	T_2 (°C)	140	134.37
ORC (D4)	PR_{ORC}	5	5.28
	ΔT (°C)	15	19.20
	T_2 (°C)	140	148.99
ORC (n-decane)	PR_{ORC}	5	5.04
	ΔT (°C)	15	20.70
	T_2 (°C)	140	205.81
KC	PR_{KC}	3	3.076
	X	0.8	0.808
	T_4 (°C)	400	401.14
	P_4 (bar)	50	51.76
GT-ORC	PR_{GT}	2.8	3.25
	$LMTD_{GTHE}$	9.6	10.46
	PR_{ORC}	5	3.11
	T_{15} (°C)	90	81.17
GT-KAL	PR_{GT}	2.8	3.06
	$LMTD_{GTHE}$	9.6	10.57
	PR_{KAL}	3	2.94
	$LMTD_{GEN}$	5.85	8.78
	$\dot{m}_{NH_3H_2O}$	2.2	2.09

Table 5.3. NSGA-II results of the models

	Properties	Developed model	Optimum Solution	Deviation (%)
GT	$\dot{W}_{net}$ (kW)	1095	1114	(+) 1.71
	η (%)	34.09	35.22	(+) 3.21
	ε (%)	56.26	57.64	(+) 2.39
	$\dot{C}_{tot}$ (\$/h)	44.57	44.08	(-) 1.11
	$\dot{Z}_{tot}$ (\$/h)	20.47	20.14	(-) 1.64
ORC (Toluene)	$\dot{W}_{net}$ (kW)	584.6	552.1	(-) 5.89
	ε (%)	15.69	14.32	(-) 9.57
	$\dot{C}_{tot}$ (\$/h)	52.74	50.95	(-) 3.51
	$\dot{Z}_{tot}$ (\$/h)	8.50	7.824	(-) 8.64
ORC (MDM)	$\dot{W}_{net}$ (kW)	332.9	336.11	(+) 0.96
	ε (%)	8.93	8.554	(-) 4.39
	$\dot{C}_{tot}$ (\$/h)	55.89	53.95	(-) 3.60
	$\dot{Z}_{tot}$ (\$/h)	11.71	11.26	(-) 4.00
ORC (D4)	$\dot{W}_{net}$ (kW)	293.3	320.36	(+) 8.45
	ε (%)	7.87	8.41	(+) 6.42
	$\dot{C}_{tot}$ (\$/h)	56.43	55.30	(-) 2.04
	$\dot{Z}_{tot}$ (\$/h)	12.21	12.91	(+) 5.42
ORC (n-decane)	$\dot{W}_{net}$ (kW)	341.8	348.77	(+) 1.99
	ε (%)	9.17	10.59	(+) 13.40
	$\dot{C}_{tot}$ (\$/h)	51.4	48.70	(-) 5.54
	$\dot{Z}_{tot}$ (\$/h)	7.10	7.988	(+) 11.11
KC	$\dot{W}_{net}$ (kW)	954.6	990.5	(+) 3.62
	η (%)	11.65	12.08	(+) 3.56
	ε (%)	24.15	25.06	(+) 3.63
	$\dot{C}_{tot}$ (\$/h)	64.05	63.12	(-) 1.47
	$\dot{Z}_{tot}$ (\$/h)	15.73	15.68	(-) 0.31
GT-ORC	$\dot{W}_{net}$ (kW)	1508	1701	(+) 11.34
	η (%)	18.76	20.79	(+) 9.76
	ε (%)	38.43	43.06	(+) 10.75
	$\dot{C}_{tot}$ (\$/h)	90.69	83.79	(-) 8.24
	$\dot{Z}_{tot}$ (\$/h)	45.36	38.16	(-) 18.86
GT-KAL	$\dot{W}_{net}$ (kW)	1594	1610	(+) 0.99
	η (%)	19.71	21.03	(+) 6.28
	ε (%)	40.53	41.93	(+) 3.39
	$\dot{C}_{tot}$ (\$/h)	79.17	77.59	(-) 2.04
	$\dot{Z}_{tot}$ (\$/h)	33.73	33.22	(-) 1.54

In all models except for toluene used in ORC, the net power output, the energy and exergy efficiencies and the total cost rate are improved as a result of the optimization. Toluene used in ORC can be improved economically but as a result of the optimization a remarkable reduction is seen to be for the net power output also. Nevertheless, toluene is considered as the most suitable fluid used in the ORC accordingly the power production. In addition to this, the best improvement of the net power output and the total capital cost rate are achieved with a deviation of 11.34 % and 18.86 %, respectively for the GT-ORC.

5.7 Conclusion

In this chapter, the parametric studies of the renewable and the sustainable models are investigated as well as the optimization of all models by using the non-dominated sorting genetic algorithms (NSGA-II) method. According to the parametric studies and the optimization of models, the thermodynamic and the thermoeconomic performance parameters are tabulated in Table 5.4.

Table 5.4. Thermodynamic and thermoeconomic results of the models by using NSGA-II method

Models	$\dot{W}_{net}$ (kW)	ε (%)	$\dot{C}_{tot}$ (\$/h)	$\dot{Z}_{tot}$ (\$/h)	Electricity production cost (¢/kWh)	Payback Period (years/30 years)
Model 1	1114	57.64	44.08	20.14	3.957	5.80
Model 2	552	14.32	50.95	7.82	9.228	10.44
Model 3	990	25.06	63.12	15.68	6.372	6.85
Model 4	1701	43.06	83.79	38.16	4.925	8.03
Model 5	1610	41.93	77.59	33.22	4.819	7.29

The results of the parametric studies and the optimization showed that the Model 1 is found to be the best option to produce energy to the actual power plant when the electricity production cost and payback period are taken into account. On the other

hand, the lowest capital cost rate is found for the Model 2. Since the amount of the energy that can be produced with the Model 2 is less than the others, it seems that it is not a feasible model according to its high electricity production cost. It can only be offered as an alternative to low-budget investors. The highest net power output can be provided by using the Model 4 with the highest total capital cost rate. Due to this reason, the Model 4 is not seen as advantageous one among others. It seems that the Model 3 has an advantageous when compared to the Model 2, but, on the other hand, the Model 3 has a higher electricity production cost than the Models 1-4-5. However, another model (Model 5) which includes the Kalina cycle can produce electricity which has almost 35 % less electricity production cost than Model 3. The same results are seen for Model 2 and Model 4 comparisons involving ORC. Accordingly, the clearest conclusion to be drawn from the table can be started such that, the ORC and the KC models are not suitable for the waste heat recovery if the exhaust gas is at high temperature.

CHAPTER 6

CONCLUSION AND RECOMMENDATIONS

6.1 Conclusions

In recent years, energy production from solid waste significantly is developed by means of the improving the disposal technologies. The importance of renewable energy sources are remarkably increased in order to improve the performance of thermodynamic systems. The investigations of the renewable energy potential of the solid waste power plants in terms of thermodynamic, thermoeconomic and optimization are performed. The sustainable models are designed and developed in this thesis with the following steps of; (i) the examination of the literature related with the subject in detail, (ii) the modeling of the existing solid waste plant then thermodynamic and thermoeconomic assessment using the actual operating data, (iii) the development of renewable and sustainable models that use waste heat as a heat source, (iv) the thermodynamic and the thermoeconomic analyses of the developed models and the realization of the optimization studies using the results obtained from analyses and (v) the comparison of the performance of the models with each other.

The main conclusions of the thesis can be summarized as follows:

- The energy efficiency, the exergy efficiency and the electricity production cost of Gaziantep Municipal Solid Waste Power Plant are found to be 38.87 %, 47.84 % and 1.63 ¢/kWh, respectively. Accordingly, the payback period of the which has 5655 kW installed power capacity actual system is calculated as 4.05 years.

- Five developed models are designed and assessed as beneficial for power production additional to the actual system. Each model has the possibility to generate electricity at a lower cost than the selling cost when the electricity production cost is taken into account.
- In the GT model, 1095 kW electric generating capacity that can be produced according to the design of the model is improved to 1114 kW by using the optimization procedure named as NSGA-II method in MATLAB software. The electricity production cost of the GT is found to be 3.957 ¢/kWh which is also deduced as the best result among the other models. It is possible to generate electricity at a lower cost than the other models by using the GT model.
- Although the power production of the ORC by using high-temperature exhaust gas, seems to be achieved, in it can be concluded that it is not very feasible. The power output of the ORC system using toluene as a working fluid is found to be 552.1 kW which is the best results obtained from the ORC model which has the 4 different working fluids. The thermoeconomic assessment for the ORC reveals once again that the most suitable fluid at the cost of electricity production is toluene at 9.22 ¢/kWh. However, it is clearly seen that this model is less advantageous when compared to the GT model.
- Although the Kalina cycle gives approximate results with the GT model in terms of power generation potential, it is presented as a model that can generate electricity at a higher cost. However, it should be said that the system is feasible and it is possible to achieve 17.5 % additional power to the existing power plant.
- The GT-ORC model which is the combination of GT and ORC systems has the highest power production capacity with 1701 kW among the other ones. The amount of this power production is 30% of the total power production of the existing system. The model is also feasible with the electricity production cost of 4.925 ¢/kWh in frame of thermoeconomics.

- As a result of the NSGA-II optimization, the net power output and the electricity production cost of the GT-KAL model are found to be 1610 kW and 4.819 ¢/kWh, respectively. The thermodynamic and the thermoeconomic results of the GT-KAL system are close to the results of the GT-ORC system. However, it can be said that the GT-KAL system is seen to be more advantageous than GT-ORC in frame of thermoeconomic aspects.
- As a result of the systematic thermodynamic and thermoeconomic analyses, GT cycle is found to be the best model among others for utilizing of waste energy of a typical MSW power plant.
- The payback period of models from Model 1 to Model 5 are found to be 5.80, 10.44, 6.85, 8.03 and 7.29 years, respectively. As a result of this, GT is also seen to be the best model when compared to other models.

6.2 Recommendations

Various recommendations can be proposed for future research on the related topics as follows:

- The selection of the best model should be also assessed by utilizing other thermoeconomic analyze methods and also optimization methods. The results should be compared with each other.
- The number of the developed models and the used working fluids in cycles should be increased and a wider comparison should be performed.

The developed models should be compared with each other considering the environmental aspects and the negative environmental effects should be minimized.

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PUBLICATIONS

Number of Published Papers

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Sum of the Times Cited

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