ANKARA YILDIRIM BEYAZIT UNIVERSITY

GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

EXERGOECONOMIC ANALYSIS AND LIFE CYCLE ASSESSMENT OF A COMBINED CYCLE POWER PLANT

M.Sc. Thesis by

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January, 2022

ANKARA

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A Thesis Submitted to

The Graduate School of Natural and Applied Sciences of

Ankara Yıldırım Beyazıt University

In Partial Fulfilment of the Requirements for the Degree of Master of Science

in Mechanical Engineering, Department of Mechanical Engineering

by

Ahmed Emin KILIÇ

January, 2022

ANKARA

M.SC. THESIS EXAMINATION RESULT FORM

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ACKNOWLEDGMENTS

Firstly, I would like to express my sincere and special gratitude to my supervisor, Prof. Dr. Erol ARCAKLIOĞLU for his tremendous support, motivation and patience during my study. His immense knowledge and precious recommendations constituted the milestones of this study. His guidance assisted me all the time of my research and while writing this thesis.

I also would like to thank all lecturers of the Mechanical Engineering Department of AYBU for their motivation and supports.

Finally, I must express my profound appreciation to my mother, my father, my brother and my sister for their endless encouragement and all kinds of supports during my M.Sc and throughout the period of writing my thesis. This accomplishment would not have been possible without them. Thank You.

2022, 16 January **Ahmed Emin KILIÇ**

EXERGOECONOMIC ANALYSIS AND LIFE CYCLE ASSESSMENT OF A COMBINED CYCLE POWER PLANT

ABSTRACT

Despite the increasing use of nuclear, biofuels, and renewable energy sources in energy production, the use of fossil fuels in energy production continues widely. Combined cycle power plants are among the best options that enable the more efficient use of fossil fuels. These power plants, which use a combination of the Brayton and Rankine cycles, use fossil fuels more efficiently while also causing less environmental damage. Examining the combined cycle power plant only in terms of exergy and energy or only economic or only environmental will not provide a total evaluation of the plant. For this reason, if the energy, exergy and economic analyses of the power plant are carried out and then environmental evaluation is performed, the cost of the plant is kept at the optimum level by obtaining the most appropriate thermal efficiency from the power plant, and improvement options can be obtained by determining the harmful effect of the power plant on the environment.

In this study, energy, exergy and thermoeconomic analyses and life cycle assessment of a natural gas-fired combined cycle power plant were carried out by using the actual operational data of the power plant. For the energy and exergy analysis, the balance equations of each component were constituted and the analysis was completed with the EES package program. The exergy efficiency of the power plant was determined to be 61.2 % as a result of the analyses performed under conditions where the ambient temperature and pressure were 25.4°C and 98.1 kPa, respectively, while the component with the highest exergy destruction was the combustion chamber with 89 MW. The condenser was determined to be the most exergy efficient component of the power plant, with a 96.4 % efficiency rating. While the total power obtained from the power plant is 205 MW, the power of the Brayton cycle is 134 MW and the power of the Rankine cycle is 71 MW.

For the thermoeconomic analysis, the unit exergy cost values of the pipes at the inlet and outlet of each component were calculated. These values were assigned to the exergy values obtained for each pipe in the energy and exergy analysis, and the total exergy cost values in the pipes were determined. The main cost balance equations and auxiliary equations used in all these calculations have been developed separately for each component. As a result of the thermoeconomic analysis, the value with the highest unit exergy cost was determined in pipe 2 located at the outlet of the compressor. Except for atmospheric air and water, which has no unit cost, the lowest unit exergy cost value is calculated in pipe 22, which leaves the HRSG and goes to the feed water tank. Considering the exergy cost values calculated by using the exergy values in the pipes together with the unit exergy values, it was found that the highest exergy cost value in pipe 3 located at the exit of the combustion chamber, and the lowest exergy cost was found in pipe 12, which is connected to the heat exchanger from the low-pressure steam turbine outlet. The highest relative cost difference occurred in HRSG and the lowest difference occurred in the condenser. According to the exergoeconomic factor values, the component with the highest value was the feed water tank, while the lowest exergoeconomic factor value was determined for the condenser.

The environmental impacts resulting from the production, operation and maintenance and disposal of each component as scrap were calculated and total environmental impact values were determined for the life cycle assessment of the power plant. The types of materials from which the components of the power plant are manufactured and the corresponding environmental Eco-indicator 99 scores were calculated, and then the weight values of the components and the environmental impact values resulting from production were determined. According to the results, the component with the highest environmental impact value was the heat exchanger, while the component with the lowest value was the main water pump.

Keywords: Energy analysis, exergy analysis, thermoeconomic analysis, exergoeconomic analysis, life cycle assessment, LCA, combined cycle power plant.

BİR KOMBİNE ÇEVRİM SANTRALİNİN EKSERGOEKONOMİK ANALİZİ VE YAŞAM DÖNGÜSÜ DEĞERLENDİRMESİ

ÖZ

Nükleer, biyoyakıtlar ve yenilenebilir enerji kaynaklarının enerji üretimindeki kullanımları gün geçtikçe artsa da enerji üretiminde fosil yakıt kullanımı yaygın olarak devam etmektedir. Fosil yakıtların daha verimli bir şekilde kullanılmasını sağlayan en iyi seçeneklerin başında kombine çevrim santralleri gelmektedir. Brayton ve Rankine çevrimlerinin birleştirilmesinden oluşan bu santraller bir taraftan fosil yakıtları daha verimli kullanırken, diğer yandan çevreye daha az zarar vermektedir. Kombine çevrim santralini yalnızca ekserji ve enerji ya da sadece ekonomik veya sadece çevresel açıdan incelemek santralin bütünsel olarak değerlendirilmesini sağlamayacaktır. Bu nedenle santralin enerji, ekserji ve ekonomik analizleri gerçekleştirilip çevresel değerlendirmesi yapılırsa, santralden hem en uygun ısıl verim alınarak tesisin maliyeti optimum seviyede tutulur hem de santralin çevreye zararlı etkisinin tespiti ile iyileştirme seçenekleri belirlenebilir.

Bu çalışmada doğalgazla çalışan bir kombine çevrim santralinin enerji, ekserji ve termoekonomik analizleri ile yaşam döngüsü değerlendirmesi, santralin gerçek operasyon verileri kullanılarak gerçekleştirilmiştir. Enerji ve ekserji analizi için santrali oluşturan her bir bileşenin denge denklemleri oluşturulup EES paket programı ile analiz tamamlanmıştır. Ortam sıcaklığının ve basıncının sırasıyla 25,4°C ve 98,1 kPa olduğu şartlarda gerçekleştirilen analizler sonucunda, santralin ekserji verimi %61,2 olarak bulunurken, en yüksek ekserji kaybına sahip olan bileşen 89 MW ile yanma odası olmuştur. Ekserji verimi en yüksek santral bileşeni ise %96,4 ile yoğuşturucu olarak tespit edilmiştir. Santralden elde edilen toplam güç 205 MW olarak hesaplanırken Brayton çevriminin gücü 134 MW, Rankine çevriminin gücü 71 MW olarak hesaplanmıştır.

Termoekonomik analiz için öncelikle her bir bileşenin giriş ve çıkışlarındaki boruların birim ekserji maliyet değerleri hesaplanmıştır. Bu değerler enerji ve ekserji analizinde yine her bir boru için elde edilen ekserji değerlerine atanarak, borulardaki toplam ekserji maliyet değerleri hesaplanmıştır. Tüm bu hesaplamalarda kullanılan ana

maliyet denge denklemleri ve yardımcı denklikler her bir bileşen için ayrı olarak geliştirilmiştir. Gerçekleştirilen termoekonomik analiz sonucunda birim ekserji maliyeti en yüksek olan değer kompresörün çıkışında yer alan 2 numaralı boruda tespit edilmiştir. Birim maliyeti sıfır olan atmosferik hava ile su dışında, en düşük birim ekserji maliyet değerine sahip olan boru ise HRSG'den ayrılıp besleme suyu tankına gelen 22 numaralı boruda hesaplanmıştır. Borulardaki ekserji değerleri ile birim ekserji değerlerinin birlikte kullanılmasıyla hesaplanan ekserji maliyet değerlerine bakıldığında ise, en yüksek ekserji maliyet değeri yanma odası çıkışında bulunan 3 numaralı boruda, en düşük ekserji maliyetine sahip olan boru ise düşük basınç buhar türbini çıkışından ısı değiştiricisine bağlanan 12 numaralı bulunmuştur. Ortalama birim ürün maliyeti ile ortalama birim yakıt maliyeti arasındaki izafi maliyet farkı incelendiğinde en yüksek fark HRSG'de, en düşük fark ise yoğuşturucuda meydana gelmiştir. Eksergoekonomik faktör değerlerine bakıldığında en yüksek değere sahip olan bileşen besleme suyu tankı olurken en düşük eksergoekonomik faktör değerine sahip olan bileşen ise yoğuşturucu olmuştur.

Santralin yaşam döngüsü değerlendirmesi için her bir bileşenin üretiminden, işletme ve bakımından ve hurda olarak bertaraf edilmesinden kaynaklanan çevresel etkileri hesaplanarak toplam çevresel etki değerleri bulunmuştur. Santrali oluşturan bileşenlerin üretildiği malzeme çeşitleri ile bu malzemelere karşılık gelen çevresel Eko-indikatör 99 puanları hesaplanmış, ardından bileşenlerin ağırlık değerleri ile üretimden kaynaklanan çevresel etki değerleri bulunmuştur. Hesaplanan sonuçlara göre çevresel etki değeri en yüksek olan bileşen ısı değiştiricisi olurken, en düşük değere sahip olan bileşen ana su pompası olmuştur.

Anahtar Kelimeler: Enerji analizi, ekserji analizi, termoekonomik analiz, eksergoekonomik analiz, yaşam döngüsü değerlendirmesi, LCA, kombine çevrim santrali.

CONTENTS

NOMENCLATURE

Roman Letter Symbols

$\dot{B}_j$	environmental impact ratio, mPts/h
b_j	specific environmental impact, mPts/GJ
$\dot{C}$	Total cost of exergy stream, $/h
c	Unit cost of exergy stream, $/GJ
E	Total energy, kJ
e	Total energy per unit mass, kJ/kg
$\dot{E}x$	Total exergy rate, kW
ex	Specific exergy, kJ/kg
$f_{b,k}$	Exergoenvironmental factor, %
f_k	Exergoeconomic factor, %
H	Enthalpy, kJ
h	Specific enthalpy, kJ/kg
$\dot{I}$	Irreversibility, kW
i	Nominal interest rate, %
$\dot{m}$	Mass flow rate, kg/s
N	Number of auxiliary equations
P	Amount of money
p	Times in a year
r	Escalation value
$r_{b,k}$	relative difference of specific environmental impact, %
r_k	relative cost difference, %
r_n	Nominal escalation value
r_k	relative cost difference, %
S	Entropy, kJ/K
s	Specific entropy, kJ/kg·K
T	Temperature, °C
U	Internal energy, kJ
u	Internal energy per unit mass, kJ/kg

V	Velocity, m/s
W	Work, kJ
$\dot{Y}_k$	Total environmental impact, mPts/h
$\dot{Z}$	Total cost, $/h
z	Height, m

Greek Letter Symbols

φ	Fuel exergy ratio, kW
ε^0	Standard chemical exergy, kJ/kg
∂	Partial differential
ψ	Rational efficiency, %
δ_k	Irreversibility rate
Δ	Difference
η_{II}	Exergy efficiency, %

Subscripts

CV	Control volume
ch	chemical
D	destructions
e	Exit
eff	effective
F	Fuel
gen	generation
i	Inlet
int, rev	Internal reversible
j	Any pipe number for LCA
k	Component
ke	Kinetic energy
L	Losses
pe	Potential energy
ph	Physical
Pr	Product

q	Heat transfer
s	Sulphur
sur	surrounding
w	Work
0	Reference condition

Superscripts

CI	Capital investment
n	Time period
OM	Operation and maintenance
P	Pressure
p	Times in a year
Q	Heat transfer
T	Thermal
W	Work
0	Reference condition

Acronyms

AC	Annual Payments
CELF	Constant-Escalation Levelization Factor
CRF	Capital Recovery Factor
EPDK	Enerji Piyasası Denetleme Kurumu
EES	Engineering Equation Solver
FV	Future Value
HPST	High-Pressure Steam Turbine
HRSG	Heat Recovery Steam Generator
ISO	International Organization for Standardization
KE	Kinetic Energy
LCA	Life Cycle Assessment
LCI	Life Cycle Inventory
LCIA	Life Cycle Impact Assessment
LHV	Lower Heating Value
OM	Operation and Maintenance

PEC	Purchasing Costs
PE	Potential Energy
PV	Present value
REPA	Resource and Environmental Profile Analysis
TCMB	Türkiye Cumhuriyet Merkez Bankası
SETAC	Society of Environmental Toxicology and Chemical
UNEP	United Nations Environment Program
UNWCED	United Nations World Commission on Environment and Development
US PWF	Uniform-Series Present-Worth Factor

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CHAPTER 1

INTRODUCTION

Energy is one of the most indispensable needs of human beings in order to survive. A lot of elements which are increasing world population, advances in technology, new devices that designed and manufactured to make human life easier and higher quality, machines, etc. reveal the importance of energy needs. While man meets this energy need on the one hand, it is important both for his own health and for all living creatures that it does not harm the environment and nature on the other. Thus, in the constantly developing and modernizing world, the increasing energy need of human beings and the importance of environmental factors; it has made it necessary to use energy more efficiently and more effectively. Scientist has applied to many renewable energy sources in order to meet this increasing energy need such as, wind energy, solar energy, geothermal energy, etc. However, due to the fact that these renewable energy sources still need development and more applications to fully meet the need and cannot be used when desired, compared to fossil fuels, for example; low or high wind strength, not being able to benefit from the sun at night, etc. Despite its decreasing reserves, this non-renewable energy source which obtained from fossil fuels, with a life span of more than a century, is still important since, the fact that most of its application areas are designed according to fossil fuels today.

Electrical energy, that has become a requisite need of human life, plays an important role in modern life. Population growth, technological progress etc. factors increase the need for electrical energy day by day. It is aimed to reduce the damage to the environment in electrical energy production, while increasing thermal efficiency with the use of fossil fuel in thermal power plants. Fossil fuels should be used in the most efficient and environmentally friendly way. To provide this, determination of the types of fuel burned in thermal power plants that operated with fossil fuels and the amount of these fuels can be associated with both thermal efficiency and environmental impact. For example, a natural gas fired thermal power plant causes less damage to the environment than a coal-fired or oil-fired power plant of the same capacity. According to the International Energy Agency, when the amount of CO_2 emission by energy source varying in metric tons in 2018 is analysed; 7,104 metric tons of CO_2 was emitted as a result of the use of natural gas, 11,415 metric tons of CO_2 was emitted as a result of the use of oil and 14,766 metric tons of CO_2 was emitted as a result of the use of coal [1]. On the other hand, combined cycle power plants can provide more effectiveness, with the same amount of fuel, than single cycle power plants.

The advantages of natural gas such as less damage to the environment than other sources, higher thermal efficiency, easy supply and short commissioning time show the importance of natural gas-fired thermal power plants and the need for these power plants will continue. There are a lot of thermal power plants, which use different types of energy source, to meet the electricity demand in Turkey. In the report published by the Republic of Turkey Energy Market Regulatory Authority in 2018, when the source-based distribution of licensed electricity generation is examined, licensed total electricity production is recorded as 296,003.71 GWh in Turkey. It is seen that natural gas constitutes the majority of the total electricity production amount with 30.88% in Figure 1.1 [2].

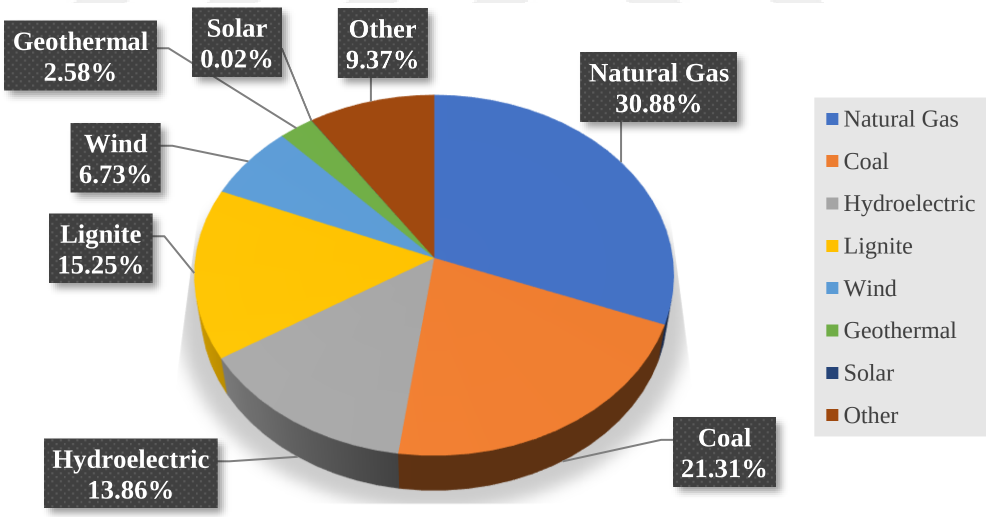

Figure 1. 1 Distribution of licensed electricity generation in 2018 by resource in percentage [2]

In addition, according to Republic of Turkey Ministry of Energy and Natural Resources report as of end-September 2019, Turkey's installed capacity is reached 90,720 MW. Distribution of this installed power according to resources is 31.4% hydroelectric, 28.8% natural gas, 22.4% coal, 8.1% wind, 6.2% solar, 1.6% geothermal and 1.7% other resources [3]. When these data with fossil fuels in the energy sector in Turkey is examined, natural gas and coal are the highest usage rates fuels.

In order to benefit from thermal power plants in the most efficient way, it is important to analyse and evaluate them thermodynamically. In these analyses, first, energy analysis that based on the first law of thermodynamics, is performed and thus, it is understood whether the components in the system work efficiently. Then, by applying the exergy analysis which is based on the second law of thermodynamics, the losses caused by the irreversibilities in the system can be determined and thus, the location and the amount of exergy destructions can be detected. Most of the thermal power plants currently active in Turkey have been established based on the first law of thermodynamics analysis. However, it has been proven by many scientific studies in recent years that, the irreversibility and useful power can be found by performing the exergy analysis which is based on the second law of thermodynamics, together with the energy analysis. The efficiency, irreversibility and thermal losses of the components that make up the thermal power plants can be found and thus, it can be determined which component needs further development or not, by doing analyses that based on both the first and the second laws of thermodynamics. In addition, by performing energy and exergy analyses together, the effects of environmental conditions on the thermal efficiency to be obtained from thermal power plants can be observed and so, it can be ensured that the power plant design is realized optimally. Also, performing energy and exergy analyses together enables the review and revision of various thermal power plants which were previously established by only using energy analysis based on the first law of thermodynamics. With the advancement of technology, these analyses can be modelled with using computer software. In many scientific studies, by using various computer software, it was possible to observe how the thermal efficiency of the power plant changed by different environmental factors such as; the outside temperature, the relative humidity of the outside air, the external air pressure etc. These observations enabled the location of the thermal power plants to be established where at the highest thermal efficiency.

Thermal power plants are required to be economically efficient while generating energy as well. With exergy analysis, thermal power generation systems can be evaluated economically as well as their thermal efficiency. When a thermal power plant is wanted to evaluate economically two factors are considered, the installation costs of the system and the fuel cost. The installation costs of the system include capital, operating and maintenance costs, while the fuel cost includes the unit price of the fuel that is consumed versus the electricity produced. In the establishment of a thermal power plant, the data obtained from only thermodynamic analysis or only traditional economic evaluation are incomplete and insufficient for correct evaluation. With the exergoeconomic (thermoeconomic) analyses which are made by using exergy analysis, the deficiencies occurring as a result of only thermodynamic analysis or only economic analysis can be eliminated. By this way, the design of thermal power generation systems can be achieved in order to obtain optimum thermal efficiency and this can present the necessary and important information about its cost and operation to make it happen effectively. In short, with applying thermoeconomic analysis, it can be enabled thermal power generation systems to be designed with both high thermal efficiency and low cost.

The thermal power plants or thermal power generation systems can be evaluated exergetically with the exergy analysis that is based on the second law of thermodynamics. Also, it is possible to evaluate the power plants or power generation systems in terms of thermoeconomic (exergoeconomic) using economic, energetic, and exergetic analyses data. In addition, by using exergy analysis, it is possible to examine the environmental impacts of the thermal systems in an exergoenvironmental manner. Emissions resulting from the burning of fossil fuels, pose a great threat to both human health and the environment. These emissions are released into the atmosphere and cause the greenhouse effect on the world and thus, causing global warming. Global warming harms vital elements such as climate, vegetation and biological life. For this reason, fossil fuels have to be used both with highest thermal efficiency and with highest economic efficiency and also in terms of the environment. It is necessary to use fossil fuels in a way that causes the least harm to nature and living things. Exergoenvironmental analysis provides information about both the thermal efficiency of a thermal power plant by exergy analysis, and examination of the environmental effects of the equipment that make up this power plant.

Exergoenvironmental analysis, for a thermal power plant or thermal power generation system, is basically based on the life cycle assessment (LCA) which starts from the production of all kinds of equipment in the plant as raw material and disposing of it after it becomes a product. Which means the determination, examination, calculation and management of the effects of all kind of component in the plant, on the environment at every stage from the cradle to the grave [4].

It will not be sufficient to examine a thermal power plant only in terms of energy or only exergy, or to only from an economic or only environmental perspective. Therefore, if the examination of the power plant covers both energy and exergy analyses together with economic and environmental analysis; it is possible to understand the effect of the power plant, while generating energy on the environment, where human beings live, by keeping the cost of the power plant at the optimum level while obtaining the most appropriate thermal efficiency from the power plant.

1.1 LITERATURE REVIEW

In this section, there are some recent studies in the literature on energy and exergy analysis, exergoeconomic and thermoeconomic analysis and life cycle assessment for thermal power plants.

The emergence of the concept of exergy coincides with the end of the 1870s. The concept of exergy was first introduced by J.W.Gibbs first in 1878 [5]. The first exergy analysis was performed by Bodvarsson and Eggers in 1972. In this study, a geothermal power plant consisting of two different systems with single and double evaporation was handled and the exergy analysis of this power plant was conducted. As a result of the study, the overall exergy efficiency of the plant was determined [6].

Aliyu et al. examined a combined cycle power plant with triple pressure and reheat facilities. They calculated the exergy destruction according to the temperature gradients of each cycle constituting the power plant and each member of the HRSG. In this study, the component with the highest exergy destruction among the HRSG elements was determined as the high-pressure evaporator, while the lowest was determined for the intermediate-pressure superheater. The criterion in exergy destruction was found to be the effect of the magnitude of the temperature gradients on irreversibilities. The highest irreversibility occurred in the stack gas, followed by HRSG, turbine and compressor respectively in steam cycle. In terms of exergy efficiency, the component with the highest efficiency was the steam turbine, while the lowest exergy efficiency occurred in the compressor. In this study, different parametric analyses were performed by changing different parameters. The most remarkable result of these parametric studies occurred in the low-pressure steam turbine. It has been observed that the steam quality, reheat pressure and superheat pressure at the low-pressure steam turbine outlet greatly affect the turbine output data and turbine efficiency. After all the energy and exergy analyses, it was revealed which component needed more improvement and modification [7].

Ibrahim et al. made a comprehensive performance evaluation study by modelling different power plants and examining the energy and exergy analyses that were done before. The study made it possible to compare combined cycle power plants with different configurations. In the power plants where energy and exergy analyses were examined and compared with each other, beside with other components, the high level of energy destruction was observed in condenser, while the highest exergy destruction was found in the combustion chamber. The most significant reason for the exergy destruction was stated as the chemical reaction between the air and the fuel in the combustion chamber. The most important factors affecting this destruction were observed as excess air fraction at the exit of the combustion chamber and the air inlet temperature. The air-fuel ratio and preheat the air entering the compressor have been given as solutions to reduce exergy destruction. It was emphasized that energy analysis would be insufficient alone and would not reveal correct data, and the actual energy analysis should be carried out jointly with exergy analysis. With the exergy analysis method, it was stated that each component which constitutes power plants, from the design to the production of the components would be more efficient by applying exergy analysis, and it was also revealed that by taking measures to prevent exergy destruction [8].

Abuelnuor et al. performed an exergy analysis of a combined cycle power plant that produces 180 MW of power using fossil fuels in Sudan. They also calculated the exergetic efficiency of the power plant and the exergy destruction of each cycle of the power plant. It was observed that the highest exergy destruction occurred in the combustion chamber in the gas turbine cycle. Chemical reactions and large temperature differences between the fuel and the working fluid have been remarked as reasons for this result. The combustion chamber, which constitutes 63% of all exergetic destruction in the power plant, was also the component with the lowest exergetic efficiency among other components. The gas turbine has been identified as the second component with the most exergetic destruction. The highest exergy efficiency and lowest exergy destruction in the gas turbine cycle was determined for the compressor. In the steam turbine cycle, the highest exergy destruction was determined in the steam turbine, while the second highest component exergy destruction was in the HRSG. Irreversibilities and mechanical losses that occur when the power generated by the turbine and transported to the electricity generator were denoted as reason for these results. At the end of this study, thermal efficiency and exergetic efficiency were found to be 38% and 49% respectively [9].

Ahmadi et al. performed energy and exergy analyses of the one of the eight identical steam cycle units each of which has installed power capacity of 200 MW. These analyses were performed by constituting the mass, energy and exergy equilibrium equations of each component and they calculated the energy and exergy efficiencies, and irreversibility of each component by using EES (Engineering Equation Solver) software program. In the energy analysis, the condenser which constitutes 69.8% of the total heat loss and has the highest heat loss, was found the highest optimization and improvement potential component. In the exergy analysis, it was found that the condenser only corresponds to 1.53% of the total exergy loss and there is almost no improvement and optimization potential after the exergy analysis. While the thermal loss was found as 10.16% for boiler in the energy analysis however the total exergetic loss was found to be 85.66% in the exergy analysis. The component with the second highest exergetic destruction was found as the steam turbine with 7.08%. It has been stated that every 0.01 bar increase in the condenser pressure causes a decrease of approximately 0.7 MW in power generation. Specifically, the power value measured between 0.09 bar and 0.32 bar decreased from 200 MW to 183.7 MW. Based on the observations, the importance of each step to keep the condenser pressure as low as possible and to keep the pressure at the lowest level was emphasized [10].

Eryasin et al. conducted performance analysis of a natural gas fired combined cycle power plant by using real operational data. They performed energy and exergy analyses separately. The energy and exergy efficiencies of each component were calculated and parametric studies were performed with different input data. The energy and exergy efficiencies were determined as 56 % and 50.04 % respectively. The combustion chamber was obtained as the highest exergy destruction component with a rate of 63%, while the condenser was the lowest with a rate of 2%. On the other hand, the highest exergetic efficiency was determined in the compressor with a rate of 94.9% and the lowest efficiency was found to be the Combustion Chamber with 64%. It has been emphasized that the air-fuel mixture ratio should be kept at an optimum level and the excess air generated during combustion should be reduce to minimize the energy and exergy destruction rates of the combustion chamber. It has been also stated that the hot exhaust gas should be used more efficiently. By adding a preheater unit between the gas turbine exhaust outlet and the HRSG and obtaining higher heat transfer rates by lowering the pinch point temperature in the HRSG unit have been proposed to achieve this [11].

Prakash and Singh performed both thermal and economic analyses of a combined cycle power plant. In the thermal analysis for the gas turbine thermal efficiency was calculated at different cycle pressure ratios varying between 8.7-16.7 for different turbine inlet temperature ranging from 1400 K to 1475 K. According to these data, the highest efficiency was found at 1475 K temperature and 10.7 cycle pressure ratio. The effect of changing the different cycle pressure ratios at the same intervals on the efficiency of the turbine was examined by keeping the turbine inlet temperature constant at 1400 K. It was observed that the steam turbine efficiency decreased with increasing cycle pressure ratio and the reason for this given that the temperature of the gas turbine exhaust gas decreases and the HRSG produces less steam to the steam cycle. The economic analysis of the combined cycle power plant has been performed on the basis of power costs, capital and installation costs, fuel and emission costs for each unit and also both including and excluding carbon capture. The total investment cost of the power plant was higher with carbon capture compared to without carbon capture within the plant. It is also observed that the emission cost included with carbon capture situation is lower than the emission cost without carbon capture situation [12].

A. Z. Şahin et al. performed both the thermal and the economic analyses of a combined cycle power plant consisting of a gas turbine cycle and a steam turbine cycle. All analyses were done according to the following three main scenarios; exergy efficiency is considered the highest priority, levelized electricity cost is considered the highest priority, and the total investment cost is considered the highest priority. It was stated that the analyses were performed with taking into account the meteorological condition of the region where the plant is located, such as temperature and humidity. In the analyses, it has been observed that when the external temperature increases for the gas turbine, the energy and exergy efficiency of the turbine decreases. In addition, the cost of electricity increased with the increasing outside temperature. It has been determined that the most significant decreases in energy and exergy efficiencies are the situations where the external environment temperature is above 40°C. Higher humidity value also reduce energy and exergy efficiencies. The reasons for the decreasing in exergy efficiency are increasing irreversibilities and decreasing in the net power of the plant, together with the increasing ambient temperature and humidity. In the thermal analysis it has been observed that the exergy efficiency of the power plant is lower than the energy efficiency. It has been also stated that the increasing pressure ratio increases both energy efficiency and exergy efficiency. While the increasing in the size of the gas turbine increases the total cost of investment, the levelized cost of electricity reduces. The reason for this has been given as the net power generated by the larger gas turbine is higher and thus the electricity cost is lower [13].

Blumberg et al. examined two different combined cycle power plants, consisting of gas turbines with two different technologies, namely F-Class and H-Class, by analysing them from an exergoeconomic perspective. For these two technologies, the H-Class has a newer technology. Both combined cycle power plants involved in the study consist of three different subsystems gas turbine cycle, steam turbine cycle and HRSG system. The energy efficiency of the power plant, which includes the gas turbine with F-Class technology, was found to be 58.2%, while the exergy efficiency was 56%. The energy and exergy efficiency of the power plant, which consists of a gas turbine with a new technology H-Class, has been calculated as 60.6% and 58.3%, respectively. The exergy destructions of both power plants were determined as 325 MW for the F-Class and 400 MW for the H-Class. The gas turbine cycle has been the subsystem with the highest exergy destruction rate for both power plants. While the exergy destruction rate for F-Class was 76%, it was found to be 80% for H-Class. As a result of the analyses, it was observed that the component with the lowest thermal efficiency was the combustion chamber for both power plants. It was emphasized that the reason for this is the pressure drop in the combustion chamber, the formation of air and fuel mixture, and the heat transfer as a result of exothermic reactions between air and fuel. In the economic analyses it has been emphasized that the power plant with H-Class technology produces more power at the same economic values than the power plant with F-Class technology. When examined in terms of electricity costs, it was calculated that the power plant with the new technology has less cost per MWh [14].

Bolattürk et al. analysed a thermal power plant where is located in Çayırhan, Turkey and which uses coal as fuel, as an exergy and thermoeconomic perspective. In this study, energy and exergy analyses were performed by using EES. While the energy and exergy efficiencies were determined as 38% and 53% respectively. It was stated that the feed water pump has the highest exergy efficiency with a rate of 94.3%, while the high-pressure turbine has the second highest exergetic efficiency with 93.5%. The component with the lowest exergetic efficiency was determined in condenser at a rate of 51%. The component with the highest exergetic destruction was the boiler, followed by low-pressure turbine, medium-pressure turbine and high-pressure turbine, respectively. The most important reason of the exergy destructions occurring in turbines was stated as the increasing in energy transformation that occurs by converting the steam energy in the turbines into kinetic energy first and then into mechanical energy. As a result of the exergoeconomic analysis of the power plant, the highest exergetic loss cost has occurred in the boiler. It was followed by the turbine as the second and the condenser as the third. The most important reason for the highest exergetic cost in boilers was explained as the highest exergetic loss occurred in the boiler [15].

Ahmadi et al. analysed a natural gas fired combined cycle power plant in terms of energy, exergy, exergoeconomic and exergoenvironmental. In the energy and exergy, the combustion chamber was determined the highest exergy destruction component. The reason for this was given as the chemical reactions and the heat transfer as a result of high temperature differences. Also, it has been observed that the whole exergy destruction of the combined cycle power plant decreased by decreasing the compressor pressure ratio. This also reduced the usage of fuel to operate the power plant. It has been determined that decreasing the gas turbine pressure ratio increases the exergy destruction that occurs in the HRSG. While performing the Exergoeconomic analysis, the component with the highest exergy destruction cost was found to be the combustion chamber and HRSG was found as the second. The reason for the highest exergy destruction costs in these two components was shown as the highest exergy destruction rates. It is stated that the most important factor affecting both exergy destruction and exergy destruction cost for the combustion chamber and for the whole plant is the gas turbine inlet temperature. It was emphasized that the compressor pressure ratio should be decreased, the gas turbine pressure ratio should be increased and the gas turbine inlet temperature should be kept at an optimum level in order to reduce exergy destructions and costs [16].

Karlsdottir et al. studied the environmental impact of a combined geothermal heat and power generation (CHP) plant by applying the life cycle assessment method. In this study, life cycle assessment performed by using data obtained in 2012 was compared with the same analysis method using the updated data in 2017 in order to reduce the harmful effect of the power plant on the environment. The global warming potential, acidification and human toxicity factors were taken into account and the results obtained were evaluated according to these factors. In the LCA, it was seen that the applications that are currently available at the power plant and which are made to mitigate the harmful effect on the environment have yielded positive results. When the evaluation and comparison factors considered in the LCA are examined, the findings obtained in the scenario in 2017 were lower for each element compared to 2012. According to the global warming potential rate, it was observed that the year 2017 is lower than 2012, likewise acidification and human toxicity elements [17].

Martin-Gamboa et al. carried out environmental impact analyses of twenty different natural gas combined cycle power plants in Spain by using LCA and Data Envelopment Analysis (DEA) methods to cover the six-year period between 2010-2015. As a major reason for using both LCA and DEA methods instead of only LCA method in the study, it has been indicated that the analyses made by applying the specified methods together give both easier and satisfactory results and easier interpretation compared to the analyses made if each method is applied separately. It was stated that the Life Cycle Inventory (LCI) method was prepared for each plant in the specified period for each year. It was emphasized that the importance of the first step is to obtain the inputs and outputs required for the other steps. The second step of the analysis includes the analysis of the data obtained in the first step, with the LCA method. In the last step, the DEA method, besides the necessary environmental data, the calculation of both the overall environmental efficiency and the periodic environmental efficiency by using data such as the power generation and production capacity of the power plants. While performing the analysis, 1 MWh electricity production was taken as a functional unit and according to this production amount, the cradle to grow method was applied for all stages from the raw material to the generation of electricity. Life cycle profiles of all power plants were determined according to five environmental impact potentials: Global Warming (GWP), Abiotic Depletion of Fossil Fuel (ADP), Acidification (AP), Eutrophication (EP) and Human Health (HH). When all power plants were analysed according to their environmental impact potential, it was determined that the average GWP ratio is 30%, the average ADP ratio is 43%, the average AP ratio is 40%, the average EP is 38% and the average HH ratio is 40%. As a result of the DEA analysis, it was stated that only one power plant out of twenty was environmentally efficient. The reason for this was given that the plant, which has the highest environmental efficiency compared to other plants, has a low power generation capacity despite the same annual operating time, and thus, unit electricity generation [18].

Memon et al. analysed a combined cycle power plant in terms of statistical, thermo-environmental and exergoeconomic. In the thermo-environmental analysis, the performance of the power plant and its impact on the environment were analysed together. In this analysis, energy, exergy and CO_2 emission analyses were applied together. As a result of the analysis, it has been observed that the power output obtained from the gas turbine cycle decreases with the increase of the compressor inlet temperature and the compressor pressure ratio. The reason for this was given as the work input required for the compressor increases and the power output of the gas turbine decreases as a result of the decrease in the gas absorbed by the compressor at constant mass flow. It was observed that the amount of power generated by the steam turbine decreased as a result of the increasing the pressure ratio. It was emphasized that the effect of the pressure ratio factor, increasing the pressure ratio decreases the total power generation of the plant. When the effect of compressor inlet temperature and compressor pressure ratio on CO_2 emission was examined, in the parametric analysis where the pressure ratio was kept constant at 4 and the inlet temperature was increased to different levels It was observed that the CO_2 emission remained constant, and when increasing the both parameters, the CO_2 emission increased. It was observed that the CO_2 emission decreased with the increase of the pressure ratio up to the 12, and the CO_2 emission increased at the pressure ratios higher than 12. The reason for this was given as the increased pressure ratio is increased the fuel consumption. When the effect of gas turbine inlet temperature on CO_2 emission was examined, increasing the temperature between 1000 K-1300 K decreased CO_2 emission and the reason was that the fuel was used more efficiently at these temperatures [19].

Agrawal et al. examined the environmental impact of a natural gas fired combined cycle power plant using life cycle assessment method. When global warming (GWP) and climate change potentials are analysed, it has been calculated that 0.548 kg of CO_2 is emitted into the atmosphere according to the global warming potential for 1 kWh electricity generation. It has been calculated that the contribution of the combustion process to the global warming potential is 80.6%. More than 99% of this ratio is the CO_2 emission occurred during combustion, and the remaining part is occurred by other gases. It has been calculated that the biggest share of the effect of climate change on human health is CO_2 emission at the rate of 94% and the remaining 5.5% rate is realized by CH_4. Considering the acidification and eutrophication potentials analyses, it has been calculated that 63% SO_2, 34% NO_x and 3% other gases are effective on the acidification potential. On the other hand, it has been stated that the acidification effect, which was examined only during combustion, was caused by 95% SO_2 and 5% by NO_x. When the Eutrophication potential of the power plant was analysed, it was determined that, while NOx constitutes 67% of the total eutrophication potential, 30% of it was PO4 released into water, and the remaining 3% was composed of other substances affecting aquatic life. While analysing the power plant in terms of human health damage, it was stated that the human toxicity potential factor of 1 kWh electricity generation was taken into consideration. As a result of this analysis, it was calculated that 65% of the human toxicity potential element occurs during the combustion of natural gas, and the remaining 35% occurs during extraction, processing and transportation of natural gas [20].

Phumpradab et al. performed environmental analyses using the LCA method in the cradle-to-grave study for a thermal power plant using two types of fuels, natural gas and fuel oil and combined cycle power plant which uses natural gas. While the GWP rate for the thermal power plant was calculated as 89.1%, this rate was found to be 90% for the combined cycle power plant. The rate of CO_2, which is the most important air pollutant gas, for both plants was 98%, while the rate of the same gas was calculated as 99% by GWP. Acidification potential (ADP) impact factor has been evaluated according to SO_2 and NOx emissions. Since the combined cycle power plant uses only natural gas as fuel, SO_2 was not considered an important pollutant for this power plant. For this reason, the contribution to ADP by the combined cycle plant has been composed of 99% NO_x gas. While the contribution of SO2 gas to ADP was calculated as 61%, the contribution of NO_x gas was calculated as 39% for the thermal power plant. It has been stated that the contribution of the thermal power plant to ADP is two times more than the combined cycle plant's contribution to ADP, because of the usage of the two different fuels for thermal power plant. It was stated that the combined cycle power plant has higher NEP than the other power plant according to the nutrient enrichment potential (NEP) impact factor. The reason for this has been given as the combined cycle power plant emits more NO_x gas into the atmosphere by operating at higher temperatures than the thermal power plant [21].

There are many scientific studies in the literature covering energy, exergy, thermoeconomic analysis and life cycle assessment. The analyses included in these studies were generally performed separately for power plants or thermal systems. These analyses that include only energy and exergy or only thermoeconomic etc. are not enough for a comprehensive evaluation of the thermal power plants and thermal systems. In this thesis, first of all, energy and exergy analyses of the power plant were carried out, and thermal efficiencies and improvement potentials were determined. Then, the thermoeconomic analysis of the power plant was performed together with the data in the first analysis, and improvement potentials were obtained in terms of energy cost. Finally, environmental analysis of each component of the power plant was carried out by using the life cycle assessment method, and environmental improvement options were determined. Thus, the thermal power plant was examined in detail with three different analyses.

CHAPTER 2

AMBARLI NATURAL GAS FIRED COMBINED CYCLE POWER PLANT

Thermal power plants can be defined as, facilities that can convert heat energy into electrical energy. These thermal power plants take their heat energy, which presents in different fuels, from chemical energy and convert it into kinetic energy then, they convert kinetic or motion energy into electrical energy. In short, the facilities that first convert the chemical energy in fuels into heat energy, then convert heat energy into kinetic energy and convert this kinetic energy into electrical energy, are defined as thermal power plants.

Combined cycle power plants are plants that combine two power cycles so that, the energy discharged by heat transfer from one cycle is partially or completely used as heat input for the other cycle. These power plants are systems that contain the gas turbine cycle and the steam turbine cycle together. In this way, combined cycle power plants combine the high average heat addition temperature occurring in the gas turbine cycle with the low average heat rejection temperature occurring in the steam turbine cycle, resulting in a greater thermal efficiency than both cycles [22].

Ambarlı Natural Gas Combined Cycle Power Plant (NG-CCPP) is located on the coast of Avcılar Ambarlı in Beylikdüzü district of Istanbul province. In this power plant electricity is produced depending on the Republic of Turkey Electricity Generation Company (*Turkish*: EÜAŞ) since 1988, and natural gas is used as fuel, which is supplied from Petroleum Pipeline Company (BOTAS). Annual electricity generation capacity is 9,450 GWh. The power plant has a base power output of 450 MW with two gas turbines and one steam turbine in a combined block. Net 442.3 MW power is guaranteed with an efficiency of 51.37% at 15°C ambient temperature and 15°C cooling water temperature by the manufacturer company Siemens A.G. for each block. The layout plan of Ambarlı NG-CCPP is in Figure 2.1 [23].

Figure 2. 1 Ambarlı Natural Gas Fired Combined Cycle Power [23]

Control of all the equipment in Ambarlı NG-CCPP is provided by the central control, data monitoring and recording system in the control room. Operators can instantly observe the data, with the aid of conductivity measuring devices of temperature, flowrate, etc. which are located at entrance and exit of the significant devices, by means of computers and other electronic devices in the control room.

The data reaching the computers and other electronic devices in the control room are evaluated according to the predetermined value ranges of the automatic control system. If the values are in predetermined ranges, the control system maintains its current state. If the values are not in predetermined ranges, the control system automatically intervenes to certain values (such as fuel flow, generator loads, etc.) and brings the values to the desired and predetermined range. Besides, the operators working in the control room can control the system manually whenever they consider necessary; can take protective measures for the health of the employees at the facility, for the plant equipment and for the electricity generation [24].

2.1 Turbine Group

Each combined block contains two gas turbines in Ambarlı Combined Cycle Power Plant. Natural gas that is supplied by BOTAS is reduced from 50-60 bar to 17-20 bar and sent to the combustion chamber at the desired flow rate. In the combustion chamber, air which is taken from the atmosphere and passed through a multi-stage air filter to clean of dust and other pollutants and compressed, is burned with natural gas. Then, the resulting burnt gases at high pressure and temperature expand in the turbine blades and this provides the rotation of turbine blades. With the rotation of the turbine blades, the compressor blades and the generator rotor, which are connected to the same shaft, rotate and thus, the kinetic energy generated is used both in compressing the air and producing electrical energy. Figure 2.2 shows the gas turbine power generation cycle.

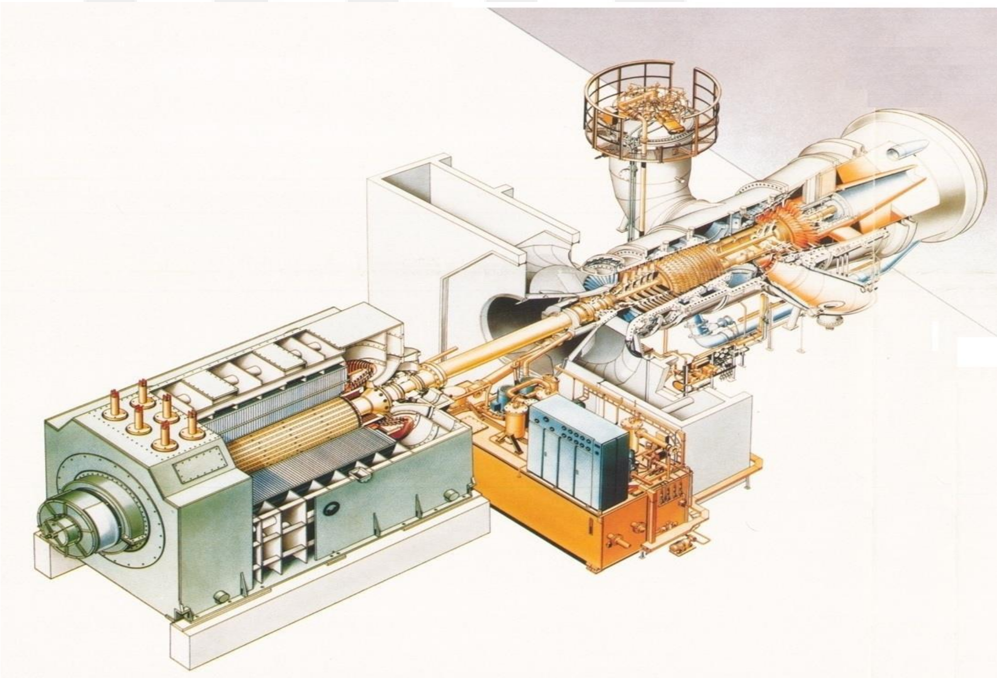

Figure 2. 2 Gas turbine power generation cycle [25]

2.1.1 Gas Turbines

Ambarlı Natural Gas Combined Cycle Power Plant consists of 3 combined blocks. The power plant contains six gas turbines totally. Each block has the same characteristics and comprises 2 gas turbines. V 94.2 type Gas turbines with gas and / or liquid fuel burning capability, which have been commissioned by Siemens AG in 1988; consist of 16-stage compressor and 4-stage turbine blades connected to the same shaft.

Each gas turbine has a double silo type combustion chamber and all gas turbines have 16 hybrid burners that provide low NO_x emission when burning with natural gas. When burning with natural gas, if all burners reach 40% of the base load, they switch from diffusion burning mode to premix burning mode. In this way, besides low NO_x emission, minimal CO emission is also provided. Gas turbine nominal load design values calculated by the manufacturer are given in Table 2.1 for 15°C outside temperature.

Table 2. 1 Gas turbine nominal load design parameters [25]

Definition	**Design Parameter**
Manufacturer	Siemens AG
Production Year	1988
Type	V94.2
Rotational Speed	3000 rev/min
Stage Number	4
Power	138,800 kW
Efficiency	32.25 %
Turbine Inlet Temperature	1050°C
Turbine Exhaust Temperature	558°C
Gas Consumption	10.15 kg/s
Compressor Stage Number	16
Pressure Ratio	10.3/1

During the operation of gas turbines within Ambarlı NG-CCPP, there are many losses in turbine blades, caused by friction, leaks and mechanical parts. The gas turbine output power and efficiency change depending on these losses, the differences in the lower calorific value of the burned fuel and the variable weather conditions in which the gas turbine operates. Gas turbines are the preferred systems in terms of supplying energy demands. The reason of this; Gas turbines' start-up time is short, their installation costs are low, there is no need for cooling water, installation of the facility is easy and fast, so they can provide early production, occupy less space, have less operating and maintenance costs, and are in compliance with environmental legislation (Figure 2.3).

Figure 2. 3 Gas Turbine of NG-CCPP [24]

Gas turbines' start-up procedure starts with the first movement and this is significant for them. For this reason, all conditions must be provided for the first move to take place. The first movement includes the following steps respectively: the rotor and the main oil pump are operated, the starter transformer drives the rotation engine of the gas turbine, the fuel valve opens, the igniters are operated and combustion occurs. The frequency of the generated electricity is ensured to be the same as the network frequency by means of the gas turbine rotation speed is controlled during operation. After the starting process is completed and rotating speed is provided to appropriate with the network frequency, the amount of electrical energy generated is increased by controlling turbine load. When the turbine outlet temperature reaches the predetermined value, it continues to operate in condition that is called gas turbine temperature control. In case of temperature control, the automatic control system intervenes to the amount of fuel, entering the combustion chamber, in order to prevent the turbine exit temperature from exceeding the predetermined limit. In order to be out of operation of the gas turbine, firstly, the amount of electricity generated is minimized by reducing it. The gas turbine rotation motor is started by turning on the transformer and the generator electricity generation is cut off by opening the generator breaker. The speed begins to drop and the fuel valve is closed. The rotor and the main oil pump continue to operate, the turbine continues to rotate without any burning, thus turbine blades are cooled [24].

2.1.2 Gas Turbine Generator

Electricity is produced from the generator on the shaft connected to the gas turbine and compressor. Air, that is compressed in the compressor connected to the gas turbine shaft and natural gas are mixed and burned in the combustion chamber and the resulting high temperature and high-pressure combustion product gases are expanded in turbine blades and rotate the turbine rotor. This enables the fuel to convert its chemical energy into mechanical energy and then, convert mechanical energy into electrical energy (Figure 2.4).

Figure 2. 4 Gas turbine generator [26]

The generator, that is connected to the gas turbine shaft converts the mechanical energy generated in the gas turbine into electrical energy. Ambarlı NG-CCPP is connected to the electrical inter-grid compound system, so the electrical energy produced must be suitable for the electrical inter-grid compound system in terms of values such as voltage and frequency. Especially, the voltage and frequency fluctuations that may occur during the operation of the gas turbine and the decommissioning of the gas turbine should not be reflected in the electrical inter-grid compound system. For this reason, the electrical energy generated in the gas turbine generator is transmitted to the system after it is arranged in the transformer and electricity transfer field. Also, when necessary, the electricity transmission can be stopped by opening the gas turbine generator breaker. Nominal load design values of gas turbine generator have been determined by the manufacturer company Siemens AG for a turbine speed of 3000 revolutions per minute (Table 2.2) [24].

Table 2. 2 Gas turbine generator nominal load design parameter [25]

Definition	**Design Parameter**
Manufacturer	Siemens AG
Manufacturing Year	1988
Type	TLRI 108 / 41
Power	160,000 kVA
Power Factor	0.8
Rated Rotational Speed	3000 rev/min
Excessive Rotational Speed	3600 rev/min
Frequency	50 Hz
Voltage	10,500 V ± 5%
Exciter Frequency	378 V
Efficiency	98.45 %
Isolation Class	F
Cooling Air Temperature	40°C
Stator Weight	147 ton
Whole Rotor	147 ton

2.1.3 Gas Turbine Stack (By-Pass Flue)

In combined cycle power plants, when the plant is in a combined cycle, that is, when both the gas turbine and the steam turbine are working together, gases that are combustion product, which their pressure and temperature decrease significantly by expanding in the gas turbine; are directed to the waste heat recovery steam generator to generate energy for the steam turbine that needs heat source to operate. When the power plant is not in a combined cycle mode and only the gas turbine is in operation, the combustion gases are directed not to the waste heat recovery steam generator, but to the gas turbine chimney or in other words the by-pass flue. In the Ambarlı NG-CCPP, there are 6 by-pass flues of the Stober & Morlock brand with a height of 47 meters and a diameter of 7.8 meters. Although it is not preferred to commission, by-pass flue is used in some periods when the waste heat boiler is not operational in order to balance the load demand. Also, by-pass flues are used to prevent overheating and damage to the elements in the heat recovery steam generator (Figure 2.5).

Figure 2. 5 Heat recovery steam generator [24]

2.1.4 Multistage Gas Turbine Air Filter

Air filters ensure the air taken from the atmosphere to purify from dust and other particles to reach it the compressor and the gas turbine without any damage. The air taken from the atmosphere may contain harmful particles such as dust. These particles are not desired to reach the gas turbine and compressor blades.

Dust and similar particles in the unfiltered air can damage the compressor and gas turbine elements by corroding or creating residues on them. These damages can be huge and costly for compressor and turbine since, compressor blades rotating at high speeds and turbine blades operating at very high temperatures and also, this is not a desired situation in any time. The air must be filtered, before directed to the compressor and turbine, to prevent this situation. Multistage air filters using in Ambarlı NG-CCPP consist of thin and thick filters. The air which is used in gas turbines and compressors, is purified from dust and similar harmful particles by these filters (Figure 2.6).

Figure 2. 6 Multistage gas turbine air filter [24]

2.2 Heat Recovery Steam Generator and Water-Steam Cycle

In Ambarlı NG-CCPP, when the combined cycle is not active and only the gas turbine cycle is active, the high temperature gases formed as a result of combustion are discharged to the atmosphere through by-pass flues. In cases, where the power plant is not operated in combined cycle mode that is, only operated with gas turbine cycle; these gases, which are release into the atmosphere and have high energy, are wasted and the efficiency remains only 30%. This situation is not preferred in power plants because of these negativities. Heat recovery steam generator (HRSG) is used in combined cycle systems in order to increase efficiency and to prevent exhaust gases with high energy from being wasted. Ambarlı NG-CCPP has 6 identical HRSGs with two same pressurized natural circulation and dome, and with condenser pre-heater (Figure 2.7).

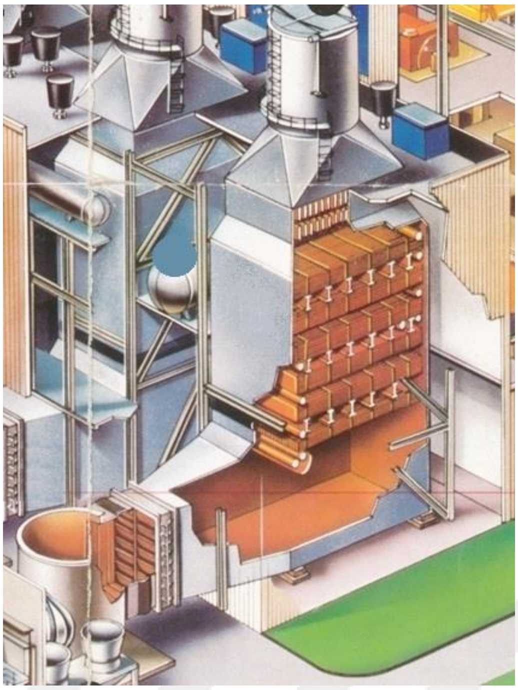

Figure 2. 7 Heat recovery steam generator [24]

These gases with high energy, which are released into the atmosphere as exhaust gas from the gas turbine, transfer the high heat to the liquid water passing through the pipe bundles in the HRSG and thus enable the water passing through the pipes to evaporate.

There are heat exchangers which arranged in bundles of pipes in HRSG and there are domes to which connected to these heat exchangers. The gas with high temperature and energy, which is exhausted from the gas turbine, rises in the HRSG. While this high-temperature gas rises, it contacts the pipes inside the HRSG and transfers heat, thus enabling the water passing through the pipes to evaporate. The exhaust gas, which gives its heat to the water in the pipes as a result of heat transfer, is then released into the atmosphere through HRSG chimneys.

Steam is produced in two different pressure lines, as low pressure (LP) and high pressure (HP) by means of domes of HRSG in Ambarlı NG-CCPP. The condensed water, which is pressurized in the main condensed water pump and heated lightly in the leakage steam condenser, is preheated by taking heat from the gas turbine exhaust gases before entering the feed water tank. Thermal insulation is applied to the HRSG wall to prevent heat loss of exhaust gases, liquid water and water vapor which of them have high temperature passing through inside the HRSG [23]. HRSG nominal load design parameters are determined by the manufacturer Simmerig-Graz-Pauker (Table 2.3).

Table 2. 3 Heat recovery steam generator nominal load design parameters [25]

Definition	Design Parameter
Manufacturer	Simmerig-Graz-Pauker
Manufacturing Year	1988
HP Steam Flowrate	231.1 – 257.9 t/h (low load / high load)
HP Steam Pressure	81 – 90.6 bar
HP Steam Temperature	526 – 530°C
LP Steam Flowrate	46.4 – 45.2 t/h (low load / high load)
LP Steam Pressure	6.3 – 6.8 bar
LP Steam Temperature	199 – 201°C
Efficiency	93%

The water leaving the feed water tank and whose pressure is increased in the high-pressure feedwater pump, enters the HRSG. First, the water passing through the high-pressure heaters is heated and enters the high-pressure dome as saturated water. The water which is circulated between the high-pressure dome and the high-pressure evaporator by means of the circulation pumps, changes phase at a constant temperature and passes into the vapor phase. Then, it leaves the high-pressure dome in the vapor phase and enters the high-pressure steam superheaters. The superheated steam obtained by heating up to high temperatures in the superheaters is sent to the steam turbines after leaving the HRSG [24].

Steam exits from the upper part of the dome as becoming separated from the boiler water and then, it goes to the first stage superheater and to the second stage superheater. Steam is overheated at both levels of superheater. Hence, as the gas turbine load increases, the boiler load also increases, and as the boiler load increases, the temperature of the gas passing over the superheaters increases. If any control device is not used, this causes the steam temperature to be higher at high loads and less at lower loads. Feed water is sprayed on the steam between the first and second superheaters at high loads in order to provide steam at a constant temperature within the boundaries of the load changes of the turbine and to keep this steam temperature at the prescribed values. Excessive increase of the steam outlet is prevented temperature during load changes by means of a steam temperature control system [24]. Figure 2.8 shows the Ambarlı NG-CCPP water-steam cycle scheme.

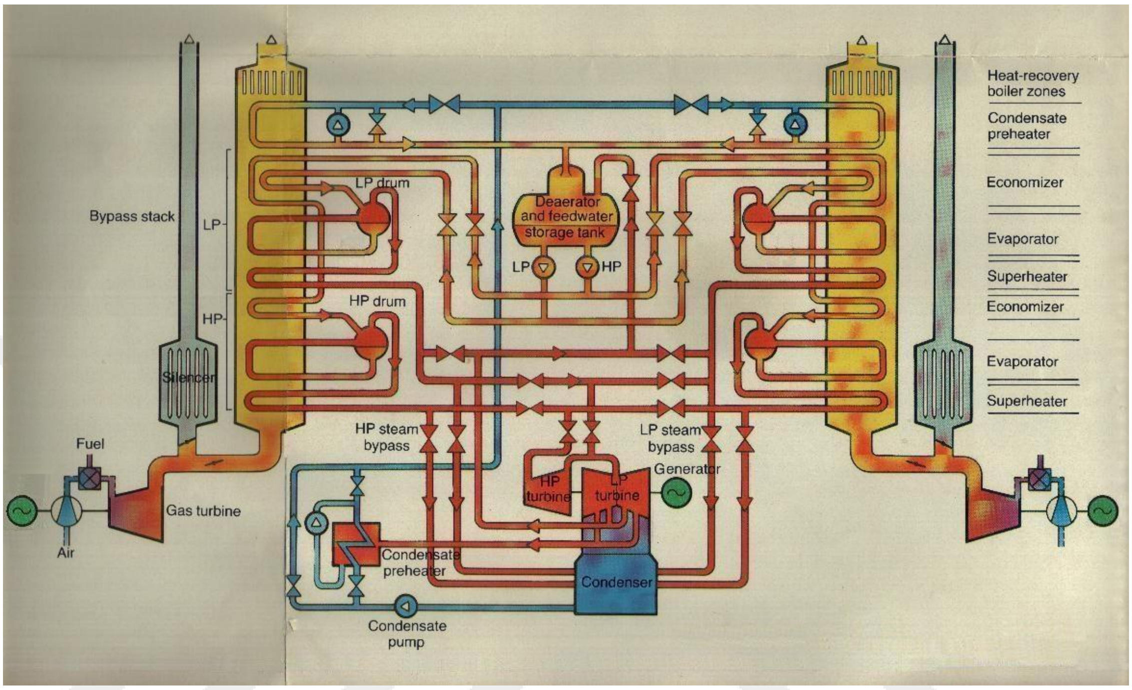

Figure 2. 8 Ambarlı NG-CCPP water-steam cycle [27]

The water, which left the feedwater tank and pressurized in the low-pressure feedwater pump, enters the HRSG. First, the water, passing through the low-pressure heater, is heated and enters the low-pressure dome as saturated water. The water, that is circulated between the low-pressure dome and the low-pressure evaporator by means of the circulation pumps, changes phase at a constant temperature and passes into the vapor phase. In the vapor phase, it leaves the low-pressure dome and enters the low-pressure steam superheater. The superheated steam, which is obtained by heating up to high temperatures, is sent to the steam turbines after leaving the HRSG [24].

The gas turbine exhaust gases finally pass through the condensed water preheater and then released into the atmosphere. The upward movement of exhaust gases, from the gas turbine exit to the HRSG exit, is caused by its low specific gravity due to high temperature. By controlling the opening ratio of the flaps at the HRSG outlet, the release rate of gas turbine exhaust gases at different outdoor air pressure and temperature values is controlled, and thus, heat absorption from the exhaust gases is increased as much as possible. The temperature control system which is similar in the high-pressure steam line, is also found in the low-pressure steam line [24].

2.3 Feedwater System

The feed water system can generally be referred to as the system that provides the necessary piping, heat exchangers to preheat the water, a recirculation system (if any), and supply control valves to meet the steam demand and control the required amount of steam. This system collects the steam turbine exhaust steam in the condensed water casing after it condenses and transports it to the boiler feed pumps. Working fluid which is pressurized and pre-heated water supplied to the boiler and to be converted into steam is commonly called feed water under normal operating conditions. In short, the feed water system includes pipes and equipment that cover the processes of collecting, transferring, heating and controlling all of these procedures [28].

Ambarlı NG-CCPP feed water system consists of condensed water tank, condensed water pump, leakage steam condenser, condensed water preheaters, feed water tank. The water in the condensed water tank is pumped with the help of the condensed water pump. There are two condensed water pumps. While the first pump pumps the water in the tank, the second pump is available as an auxiliary. The water which is pumped by the first condensed water pump, passes through the leakage vapor condenser. Leakage steam cools the leakage steam generated in the turbine. The condensed water passing through the leakage steam condenser is heated by the steam in the stages of the low-pressure steam turbine and then it comes to the condensed water preheaters. The condensed water that is heated to a certain degree here, is transmitted to the feed water tank. The condensed water in the feed water tank is heated by the steam in the stages of the low-pressure turbine. After this process, the condensed heating steam and condensed water are collected at the bottom of the feed water tank.

The operation of the feed water system is monitored from the control room. Every operation from the beginning of the system to the discharge of water is monitored and managed from the control room. The PH value and conductivity of the water are controlled by the sensors in the water discharge pumps in the feed water system, and thus operators are warned if the conductivity of the water is high [24].

2.3.1 Feedwater Tank and Pump System

Feed water tanks are containers used to degas oxygen and carbon dioxide in water and if the gases are not removed, they can cause corrosion. These types of containers are mainly used in boiler rooms where the water directed to the boiler system is used for degassing or connected to HRSG systems. There is one feed water tank in the Ambarlı NG-CCPP. The feed water tank is used in the steam cycle and operates as a gas receiver. Feed water tank has the different sort of functions; separating the oxygen and other gases in the condensed water, working as a heater when necessary, and working for the boiler feed pumps as a feed water tank.

Condensed water heated from the condensed water preheater enters the heater from above. The bleed steam feed water from the steam turbine enters the heater from the side. Entering condensed water, from different inlet directions, collides with the bleed steam feed water inside and then becomes granules. The granulated water is washed with a steam jet and so purified from gases and these gases are discharged from the top part of the feed water tank with the help of valves. Damage to HRSG, steam turbine and other equipment is prevented by these gases that is discharged from the feed water tank (Figure 2.9) [24].

The gas receiver should be continuously fed with steam above atmospheric pressure to prevent reducing the pressure at the inlets of the feed water pumps below a certain value and the water level in the feed tank should be kept under control. In order to achieve this, under operating conditions, bleed steam is taken from the second and third stages of the low-pressure turbine and sent to the feed water tank. Also, if the boilers are out of service, steam is taken from the auxiliary steam collector of the other combined group to be given to the feed water tank [24].

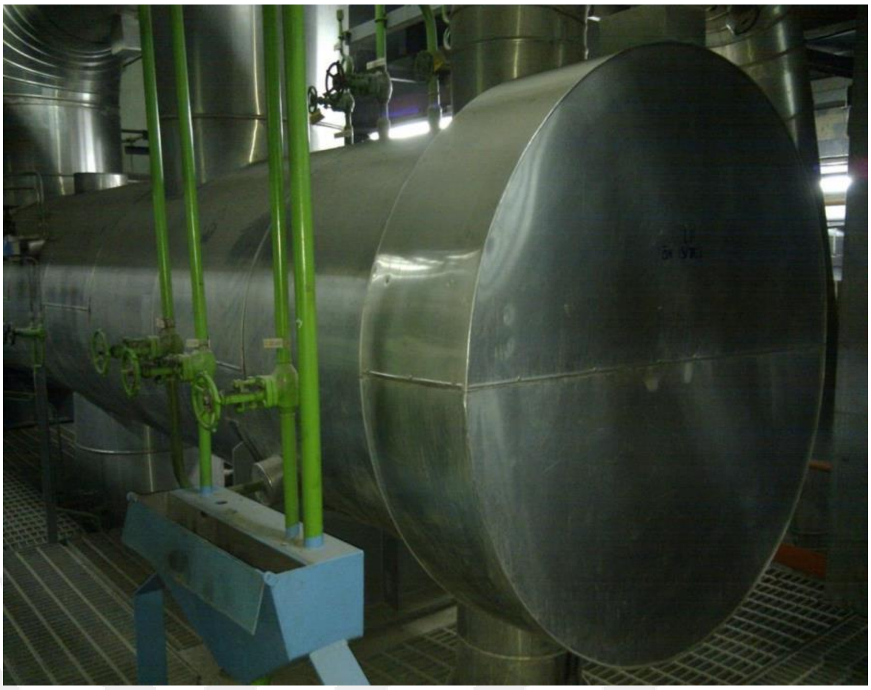

Figure 2. 9 Feedwater tank of NG-CCPP [24]

There are two pumps, the low-pressure feed water pump and the high-pressure feed water pump, which enable the water leaving the feed water tank to be directed to the required places. The water exit from the feed water tank is transmitted to the low-pressure line of the HRSG by the low-pressure feed water pump. In the same way, the water exit from the feed water tank is transmitted to the high-pressure line of the HRSG via a high-pressure feed water pump. Water which is pressurized with the aid of pumps operating in connection with the feed water system, suffers losses along the line which transmitted and then determines the pressure of the low-pressure and high-pressure domes inside the HRSG. The pressure of the feed water pumps directly affects the temperature and pressure of the steam in the HRSG. Since saturated water and saturated steam are located together in the HRSG domes, the pressure in the domes directly affects the temperature of the water-steam mixture. Intervention to the flow rates of the pumps in the feed water pump system also means to interfere with the pressure in the feed water lines. The intervention to the pressure also causes the changing flow, temperature and pressure values of the steam produced and thus the power generated from the steam turbine varies.

2.4 Water Preparation and Purification System

Unprocessed raw water to be used in water-steam cycle power plants may contain harmful substances that could damage the power plant and adversely affect the thermal efficiency of the power plant. These substances; dissolved gases such as oxygen, carbon dioxide, nitrogen; salts such as magnesium, sodium, potassium, sulphate and silica compounds; also, there may be solid substances such as moss, clay, and mud. All these substances can damage components such as turbines and pumps used in steam cycle power plants, and so which can negatively affect the overall thermal efficiency of the system. The water that is used in the system must be purified, demineralised and softened in order to prevent damage to these components used in the plants.

The water recirculating in the Ambarlı NG-CCPP system must be purified, softened, and filtering to prevent from damaging to the important and main equipment such as HRSG, condenser, turbines and pumps as well as all components and equipment which are in contact with water or water vapor.

In the water purification system in the Ambarlı NG-CCPP, the raw water drawn from the Azatlı spring water located in Istanbul is used and used raw water is collected in two 800 m^3 concrete tanks in the power plant for purification. These tanks are also used as feed water tanks for fire in emergency situations. After the samples taken from the tanks are passed through the sand and carbon filters, they become ready for the utility water requirement with the pre-treatment process by flocculation method. Water that is subjected to pre-treatment process, feeds two separate demineralization systems of 50 m^3 [26].

2.5 Steam Turbine Group

The steam turbine electricity producer group in the Ambarlı NG-CCPP consists of two steam turbines as low pressure and high pressure and a generator on the same shaft. The steam turbines in the power plant are double exhaust, two-cylinder condensation type turbines. Steam turbines are located in a case that provides thermal insulation to minimize heat loss and increase the efficiency of the steam turbine. There is also a condenser at the bottom of the steam turbine casing.

Steam produced at two different pressures as high pressure and low pressure with HRSG within Ambarlı NG-CCPP is transmitted to the steam turbine electricity producer group via HP and LP steam lines. Steam from the HP steam line directly enters the high-pressure turbine. Steam, which expands and loses pressure and decreases in temperature, generates power by rotating the turbine blades. The steam from the HP steam turbine enters the LP turbine by combining with the steam coming from the LP steam line. Steam, which loses pressure by expanding and decreases in temperature, generates power by rotating the LP turbine blades. Also, bleed steam is taken from the LP steam turbine and used for heating the feed water tank and meeting the leakage steam requirement (Figure 2.10) [24].

Figure 2. 10 Steam turbine group [26]

2.5.1 Steam Turbine

There are three steam turbines in the Ambarlı NG-CCPP. Steam turbines have two stages, low pressure and high pressure. These two stages, located inside the steam turbine casing and on the same shaft, are called high pressure (HP) steam turbine and low pressure (LP) steam turbine. The high-pressure part is single flow and consists of

26-stage reaction type blades. The low-pressure part is double flow and the last stage blades are 1.05 meters long (Figure 2.11). The design parameters and nominal load values of the steam turbines have been determined by the manufacturer, Siemens AG (Table 2.4).

Table 2. 4 Steam turbine nominal load and design parameters [25]

Definition	Design Parameter
Manufacturer	Siemens AG, UB KWU
Manufacturing Year	1988
HP Type	M 30 – 25
LP Type	N 30 – 2 x 10 - A
Rotational Speed	3000 rev/min
HP Stage Number	26
LP Stage Number	2 x 8
Power	172,700 kVA
HP Steam Flow Rate	462 t/h
LP Steam Flow Rate	93 t/h
HP Steam Flow Pressure	76 bar
LP Steam Flow Pressure	6 bar
Exhaust Steam Pressure	0.04 bar
HP Steam Temperature	524°C
LP Steam Temperature	197°C

Figure 2. 11 Steam turbine of NG-CCPP [24]

All necessary and sufficient conditions must be provided for the commissioning and operation of the steam turbines within Ambarlı NG-CCPP. After the necessary and sufficient conditions are provided, the steam turbines operate by the steam that is generated from the high temperature gases discharged from the gas turbine cycle as exhaust gas which is used for heating the water circulating in the HRSG. After the HP steam produced by HRSG reaches certain temperature (524°C), pressure (32 bar) and flow rate (16 kg/s) conditions, the steam is taken as a sample from the HRSG outputs of the HP and LP steam lines. Then, silica and conductivity values are checked, so that the steam does not damage the turbine blades and does not reduce the thermal efficiency of the steam turbine. After the all specified conditions are met, the heating system is activated and the steam turbines are heated before the vapours produced in the HRSG enter the turbine. The turbine heating system is activated at a certain temperature and is deactivated after reaching a certain temperature. Thus, homogeneous heating of turbine equipment is ensured and thermal stresses are prevented. After the heating process is completed, the turbine discharge valves are closed, allowing the steam entering the turbines to rotate the blades. With the rotation of the turbine shaft, the generator, which starts to generate electrical energy, regulates the voltage and current of the electricity produced by the regulator located at the outlet and is transmitted to the inter-grid composite system.

2.5.2 Steam Turbine Generator and Gland Steam System

Ambarlı NG-CCPP is connected to the inter-network composite system. Therefore, the generated electrical energy must be suitable for the inter-grid composite system in terms of values such as voltage and frequency (Figure 2.12).

Figure 2. 12 Steam turbine generator [24]

Steam turbine generator nominal load design values have been determined by the manufacturer company Siemens AG for the turbine speed of 3000 revolutions per minute (Table 2.5).

Table 2. 5 Steam turbine nominal load and design parameters [25]

Definition	Design Parameter
Manufacturer	Ganz Electric Works
Manufacturing Year	1988
Type	ORG 587 (hydrogen cooling)
Power	216,000 kVA (max 35°C cooling water inlet temp.) 248,000 kVA (12°C cooling water inlet temp.)
Power Factor	0.8
Nominal Rotational Speed	3000 rev/min
Overspeed	3800 rev/min
Voltage	15,750 V ± 5%
Exciter Voltage	465 V
Stator Current	7918 A
Exciter Current	1315 A
Efficiency	98.80% (guaranteed)
Isolation Class	F
Stator Weight	135 ton
Whole Rotor	42 ton
Cooling System Code	ICW37H71
Hydrogen Pressure	3.0 bar (g)
Hydrogen Purity	97% (min 95%)

The steam in the steam turbine bearings and supplied to provide leaking is called gland steam. During the operation of the steam turbines, gland steam is used to prevent the high pressure and temperature steam passing through the turbine by leaking from the turbine shaft bearings and adversely affecting the steam turbine efficiency. By means of the gland steam, some steam is held in a way to apply pressure to the steam turbine beds from outside in order to increase the pressure inside the turbine casing [24].

2.6 Condenser Group

After the work produced by the steam turbine, the steam is used up. The exhausted steam is cooled in the condenser. The condenser, cooling water pumps and pipes that connecting each of them in the condenser system (Figure 2.13).

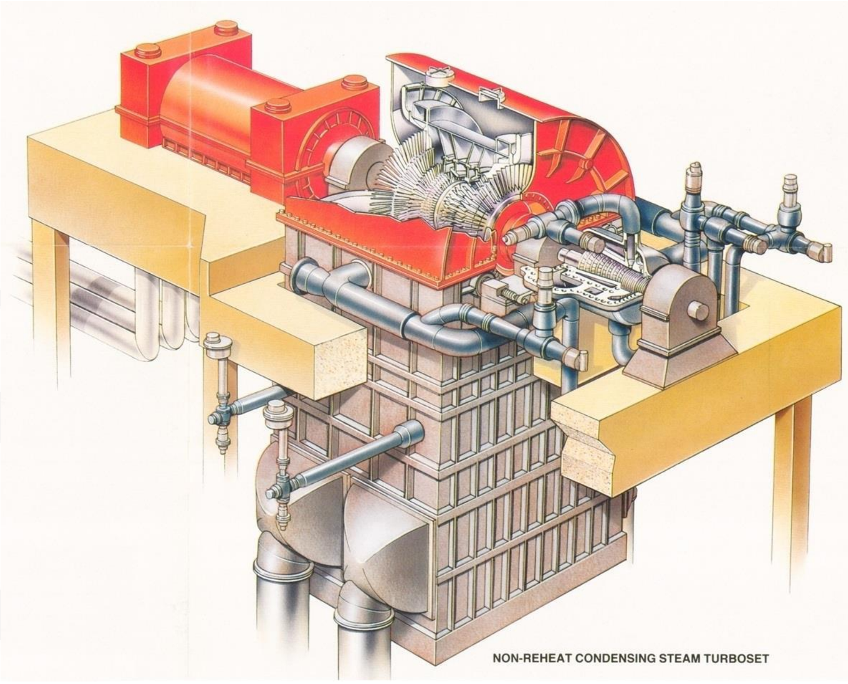

Figure 2. 13 Condenser and steam turbine [25]

2.6.1 Condenser

Steam loses its pressure and temperature by expanding in steam turbines and enters the condenser located just below the steam turbine. The mixture of saturated steam and water which flowing down in the condenser, condenses completely by giving heat to the pipe bundles through which cooling water passes, and is transferred to the main condensed water pump by leaving the condenser in the liquid water phase. Sea water is used as cooling water in the condenser at the Ambarlı NG-CCPP. Seawater entering the condenser draws heat to allow the steam to condense, and its temperature rises approximately 4ºC to 6ºC and leaves the condenser. Because of the foreign materials (algae, sand, etc.) that constantly accumulate on the seawater line, heat transmission decreases and as a result of the condenser performance is affected negatively. This negative situation can be prevented by regular cleaning. Nominal load design values of the condenser are presented in Table 2.6.

Table 2. 6 Condenser nominal load design parameters [25]

Definition	**Design Parameter**
Manufacturer	Siemens AG, UB KWU
Manufacturing Year	1988
Type	Two Chamber, single pass, 1 KNE 73x34-1 TSB90
Inlet temperature of the condenser's water vapor	35-40°C
Condensate water temperature	33-38°C
Condenser steam flow rate	555 ton/h
Inlet temperature of cooling water (design)	15°C
Inlet temperature of cooling water (summer)	26-27°C
Inlet temperature of cooling water (winter)	8-915°C
Flow rate of the cooling water (single pump)	18,000 m^3/h
Flow rate of the cooling water (double pump)	36,000 m^3/h
Cooling water pump	780 kV, 105 A
Condenser material	Titanium
Cooling area	11,025 m^2
Cooling pipe number	20,524 pieces
Cooling pipe diameter	19 mm
Cooling pipe wall thickness	0.5 mm
Maximum working pressure (steam condensation)	1.15 bar
Maximum working temperature (cooling water)	85°C
Maximum working temperature (steam condensation)	38°C
Volume (cooling water)	279 m^3
Volume (steam condensation)	113 m^3

2.6.2 Cooling Water Systems

In the Ambarlı NG-CCPP, sea water drawn from the Marmara Sea is used as cooling water. The cooling water system provides cleaned cooling water to the turbine condensers and the installation cooling water systems within the power plant. The required pressure in the pipelines supplying sea water to the condensation water and auxiliary installation cooling water coolers is provided by cooling water pumps. The saturated steam leaving out of the low-pressure steam turbine is condensed by giving heat to the cooling water in the condenser and reused in the system. The heat taken during the condensation of the steam is discharged through sea water.

The cooling water required for the three combined blocks is taken from the Marmara Sea where is 1 km away from the power plant, through 335-meter pipes. The system consists of three pairs (6 piece) of half-capacity circulation pumps and 3 pieces of one-kilometre concrete pipelines with a diameter of 2.5 m. Cooling water returns to the Sea of Marmara through three separate concrete pipelines and a 235-meter discharge pipe. Cooling water enters the power plant through three water intake heads and the circulation water flows to the control area with the difference in level. There are two circulation pumps with a capacity of 16,000 tons / hour for each steam turbine electricity generating group at the control site. While one of the cooling water pumps is in service, the other is held as an auxiliary. In summer days, the need for cooling to the power plant increases. Two cooling water pumps can be used together in the cooling water system for a short time to meet this need.

The cooling water system consists of the following main parts;

- 335 meters long sea cooling water pump installation inlet lines
- Approximately one km long concrete culvert cooling water intake and return lines
- Mussel filter system
- Two partitions condenser and condenser pipes cleaning system
- Condenser air deaeration and air intake system
- 235 meters long line laid in the Marmara Sea

Cooling water pumps can be operated individually or together from the main control room. The pumps have non-return valves operated by a hydraulic system and there is an electronic lock between these valves and the pump. When the pump is operated, the non-return valve is opened, and when the pump stops, the non-return valves are closed by means of electronic lock. Also, the opening ratio of the non-return valve can be seen in the main control room. It is important to filter the seawater from the coolant pumps before it enters the condenser. With the filtration of the sea water, heat exchange during its movement in the pipes becomes more efficient. Technical features of cooling water pumps are presented in Table 2.7.

Table 2. 7 Cooling water pumps technical data [25]

Definition	Technical Parameter
Flow Rate	18,000 m^3
Pump Current	106 A
Pump RPM	485 rpm
Pump Density	1.03 g/cm^3
Pressure	3 kg/cm^2
Voltage	6 kV 66%
Pump Head	21,9 m
Capacity	2 x 50%

2.7 Electrical System

Ambarlı NG-CCPP has an electrical system in order to use the electricity generated by the gas turbine and steam turbine in the power plant.

2.7.1 Switchyard

After the electrical energy that is produced in Ambarlı NG-CCPP, leaves the transformers, it must be connected to the high voltage line of the inter-grid composite system. The system that allows the generated electricity to be distributed with the help of networks and allows disconnection when it's necessary is electricity transfer area, also known as the switchyard (Figure 2.14).

Figure 2. 14 Ambarlı NG-CCPP switchyard [23]

There are two different voltage outputs, 154 kV and 380 kV, in the switchyard. There are six generators in the first two blocks are connected to the 154 kV switchgear, while there are three generators where at the two gas turbines outlet and at one steam turbine outlet, are connected to the 380 kV switchgear.

2.7.2 Main Transformers

There are 9 main transformers in the power plant. Six of these have a conversion ratio of 10.5 / 154 kV, while the remaining three have a conversion ratio of 10.5 / 400 kV. Six of the main transformers have 180 MVA and three of them have 200 MVA.

2.7.3 Start-up Transformers

There are two AEG ETİ brand transformers in power plant. These transformers are also called starting transformers. Units can be initiated by means of starting transformers and thus the internal demand of the plant can be provided.

2.7.4 Internal Demand Transformers

There are 9 internal demand transformers within the plant. Six of these feeds the gas turbine section, the remaining three feed the steam turbine. The values of the internal demand transformers feeding the gas turbine section and steam turbine section are 10.5 / 6.3 kV, 2000 kVA and 15.75 / 6.3 kV, 10000 kVA respectively.

2.7.5 Low Voltage Distribution System

While the low voltage system is fed by 10.5 / 0.4 kV transformers in gas turbines, it is fed by 3.5 / 0.4 kV transformers in steam turbines. The nominal voltage of the low voltage system is 0.4 kV.

2.7.6 Medium Voltage Distribution System

Medium voltage distribution system has metal protected internal type cells and its voltage value is 6.3 kV. The energy requirement of the medium voltage distribution system in the Ambarlı NG-CCPP is provided by the steam turbine internal need transformers and the starting transformer.

2.7.7 Direct Current System

Each unit and common system have its own direct current system in the power plant. The nominal voltage of the direct current system is 220 Volts.

2.7.8 Urgent Feeding System

The emergency supply system in the Ambarlı NG-CCPP is used to feed the required places in the plant. In the system, there is an emergency diesel generator that undertakes the task of starting any gas turbine with a power of 2070 kVA with gas or diesel oil.

2.8 Supporting System

Used in the power generation of the Ambarlı NG-CCPP; apart from the main components such as gas turbines, steam turbines and HRSG, there are equipment and systems that ensure the healthy operation of these main components and support the long-term operation of them. While auxiliary cooling system, lubrication system, air compressor system, fire extinguishing system are among support systems; auxiliary steam collectors and spare oil-fuel tanks are in the support equipment.

2.8.1 Auxiliary Cooling System

Auxiliary cooling system is used to cool certain equipment and components in the Ambarlı NG-CCPP. HRSG feed water pumps and main condensed water recirculation pump gaskets, high pressure HRSG feed water pump oil cooler, turbine lubricating oil coolers, hydrogen leakage oil coolers, condenser vacuum pump circulation water coolers and generator hydrogen coolers are among these equipments.

2.8.2 Lubrication System

With the lubrication system in the Ambarlı NG-CCPP, all equipment that needs lubrication, especially the main equipment such as turbines, compressors, generators and pumps, are lubricated by this system, thus efficient and healthy operation of all these components are provided (Figure 2.15).

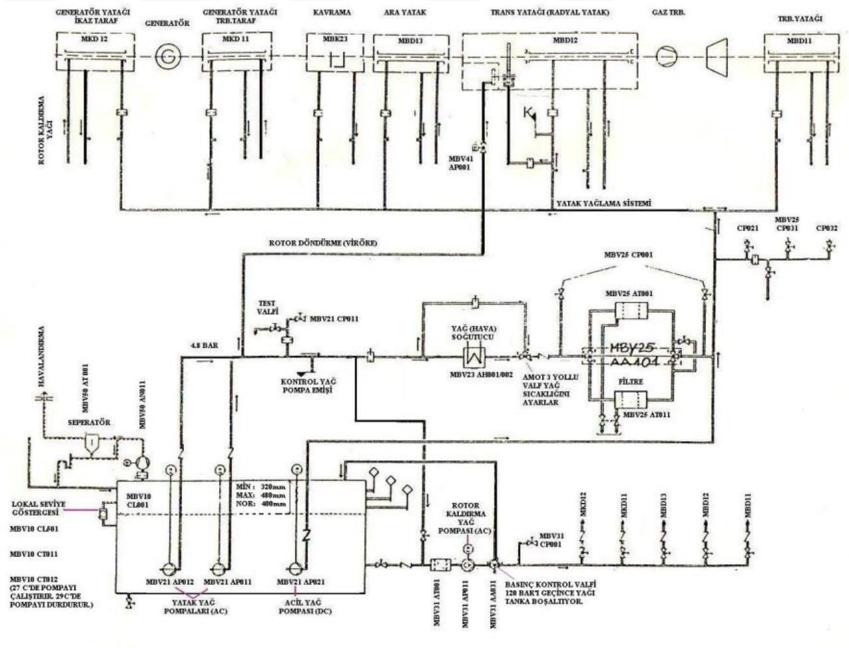

Figure 2. 15 Schematic view of lubrication system [24]

Main oil tank in the lubrication system provides the oil required for the generator control system, the hydraulic rotor rotation system, lubrication and cooling of the bearings. The main oil tank serves as an oil reservoir for the specified equipment, and it also serves as deaeration and ventilation of the oil. The oil in the oil tank must be at a certain level. These levels are between 1515-1590 mm for the Ambarlı NG-CCPP. If the oil level falls below 1515 mm or rises above 1590 mm, the system warns of low and high oil levels respectively. When necessary, oil can be drained with the valve at the bottom of the oil tank.

The absence of residues and air bubbles in the oil in the oil tank is important for the systems to be lubricated to work properly. The oil tank capacity is designed to prevent the oil in the system from circulating more than eight times per hour to provide this. Thus, the oil entering the oil tank is absorbed for circulation after approximately eight minutes. During this time, the sediment contained in the oil sinks to the bottom and air bubbles separate from the oil. The main oil tank can be evacuated to the desired level or completely for maintenance. In addition, all oil in the oil system can be completely evacuated into the main oil tank if desired.

2.8.3 Air Compressor System

The compressors in the Ambarlı NG-CCPP absorb the air from the atmosphere by filtering it in multi-stage air filters. The compressors are run idle for a certain time (10 seconds) at the first start. The purpose of operating the compressor idle is to degrease the sucked air. The air pumped by the idle compressor goes to the separator together with the oil. By means of filters located here, the oil stays under the separator and thus the air is supplied to the compressor suction again through the non-return valve, as a free from oil.

2.8.4 Fire Extinguishing Water System

Ambarlı NG-CCPP has two 840 tons of raw water tanks. Raw water tanks feed the fire extinguishing water pumps and provide water for the fire extinguishing system. The fire extinguishing water taken from the raw water tanks feeds the three fire extinguishing water pumps in the water discharge system. 10 bar pressure is provided in the fire-fighting water line with the fire extinguishing water pumps.

There is a fire extinguishing system that automatically controls the pressure in the fire extinguishing water line. The fire extinguishing system is automatically activated when the pressure in the fire extinguishing water line drops to 9 bar and is automatically deactivated when it rises to 10 bar. Diesel fire extinguishing water pump automatically enters service when the line pressure drops to 7 bar and can be manually switched off if necessary.

2.8.5 Auxiliary Steam Header

The auxiliary steam line at the Ambarlı NG-CCPP is a common steam line between units. The auxiliary steam line fed from the units' own steam systems or auxiliary steam boilers; It provides steam to the heating system in the lodgings, the wall and heater heating system, the steam turbine sealing steam system, the heating system of the feed tank (when working only with fuel oil) and the water treatment (from mineral) system.

2.8.6 Auxiliary Fuel Oil Tanks

Fuel oil tanks within Ambarlı NG-CCPP are systems that store fuel to the power plant in case of a possible natural gas supply problem, so that electricity generation is not interrupted. Also, if there is no problem in natural gas supply but its flow rate or pressure decreases, fuel oil can be burned with natural gas in the same way and so, deficiency is prevented. Besides, electricity can be produced by burning only fuel oil in the power plant by stopping completely natural gas flow.

CHAPTER 3

ENERGY, EXERGY AND THERMOECONOMICS & LIFE CYCLE ASSESSMENT

3.1 Energy

Energy can be defined as the capacity of a system to do work or its ability to do work. In other words, the ability to cause any change is called “energy” [29]. While solar, wind, geothermal, hydro and biomass energies are among renewable energies; oil, natural gas, coal and nuclear are non-renewable energies.

3.1.1 Thermodynamics Laws

There are four laws in thermodynamics: zeroth, first, second and third laws. The zeroth law of thermodynamics states that two objects in contact with each other and in thermal equilibrium and the third object in thermal equilibrium with one of these two objects will also be in thermal equilibrium with the other object. In short, if one of the two objects in thermal equilibrium is also in thermal equilibrium with a third object, it indicates that all three will be in thermal equilibrium with each other.

3.1.2 Entropy

"Entropy" is defined as a thermodynamic property expressed by the second law of thermodynamics, which reveals the direction and result of disorders and changes occurring in a thermodynamic system or any substance. According to the Clausius inequality, named after R. J. E. Clausius, who lived in the middle of the nineteenth century and is regarded as one of the founders of thermodynamics; for reversible or non-reversible cyclic processes, the entropy change is expressed by the following equation [29].

$$\oint \frac{\delta Q}{T} \leq 0 \tag{3.1}$$

For the internal reversible processes, the entropy is expressed by the following equation.

$$dS = \left(\frac{\delta Q}{T}\right)_{int,rev} \tag{3.2}$$

While T represents the temperature at the system boundary in this equation, δQ represents the heat transfer between the system and the environment.

3.1.3 Energy Analysis

The thermodynamic systems are classified as closed or open based on whether they have a fixed volume or a fixed mass. While performing energy and exergy analyses on thermal power plants, they are considered as open systems (control volume).

The energy balance per unit time for steady-flow open systems is written as follows according to the principle of energy conservation expressed by the First Law of Thermodynamics:

$$\dot{Q}_{CV} - \dot{W}_{CV} = \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) - \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) \tag{3.3}$$

In open systems, the entropy change per unit time is calculated over three main factors. These can be specified as heat transfer, mass transfer and entropy generation in the system. Entropy change per unit time in open systems is calculated by the sum of these: the entropy transfer that occurs as a result of heat transfer in the system, the entropy transferred as a result of the net mass transfer occurring in the system per unit time, and the entropy generation that occurs as a result of irreversibilities in the system. In steady-flow open systems, there is no change in entropy per unit time and thus the entropy balance is possible in these systems. This equilibrium per unit time between steady-flow open systems and the surrounding of the system at T_0 temperature is written as follows:

$$\dot{S}_{gen,CV} = \sum_{e} \dot{m}_e s_e - \sum_{i} \dot{m}_i s_i + \sum \frac{\dot{Q}_{sur}}{T_0} \geq 0 \tag{3.4}$$

3.2 Exergy

Most of the problems in thermodynamics involve systems that can be modelled in equilibrium. But these systems are not in balance with their surroundings. For example, the temperature of a system that is below or above ambient temperature is not in equilibrium with its surroundings. In this case, the thermal balance needs between a system and its surrounding cannot be met. Any lack of balance between the system and its surroundings can be used to produce shaft work. The second law of thermodynamics is used to calculate the maximum work that a system can produce [32].

The first term meaning "exergy" was used by Carnot in 1824. The first studies on exergy analysis started with Gouy and Stodola. Many different scientists contributed significantly to the development of the exergy concept after Gouy and Stodola. In the thermodynamic analysis of systems, the concept of exergy was used by Bosnjakovic in 1935. After Z. Rant used the term exergy for the first time and was accepted and entered the literature worldwide, the use of the term exergy is very advanced with the contributions of many scientists such as Tsatsaronis, Gibert, Kotas, Bejan, Moran [34].

The exergy of a system is defined as the maximum shaft work that can be done between the system and its environment. The exergy is also defined as a measure between the system state and the system environment. So, exergy means a common feature of the system and the environment. When the environment is specifically defined, exergy can only take values in terms of the system's properties. Thus, exergy can be accepted as an extensive property, and it, like other extensive properties such as entropy and mass, can be transferred from system to system. Exergy is generally destroyed and cannot be conserved. In the boundary condition, the system suddenly returns to equilibrium with its environment and no work can be produced due to the balance between the system and its environment. So, exergy is completely destroyed. In such sudden situations, the system, which has the capacity to do work in the beginning, completely loses its capacity to do work. Exergy value is never negative, and the lowest value is zero, because no work is required to achieve a similar abrupt change with the system's environment [35].

When there are differences in pressure, temperature, height, and velocity between the system and its environment, the work potential is occurred. Another definition of exergy is the portion of energy that can be transformed into other types of energy i.e., the useful portion of energy. According to this definition, "anergy" refers to the portion of energy that cannot be transformed into other types of energy and thus cannot be used. As a result, as the sum of total energy, exergy, and anergy, it can be expressed as follows [36]:

$$Energy = Exergy + Anergy \tag{3.5}$$

The internal energy of the environment can be given as an example of anergy. Different types of energy, for example, the unusable portion of mechanical energy is anergy and equals zero. Since all the internal energy of the environment is unusable energy, that is, anergy, it becomes equal to zero. In other words, the exergy of the internal energy of the environment is zero. Table 3.1 shows the comparison between energy and exergy.

Table 3. 1 Comparison of energy and exergy [32]

ENERGY	EXERGY
Only depends on the energy or matter flow parameters. It is independent from environmental parameters.	It depends on both energy or material flow parameters and environmental parameters.
It has different values than zero when it is in equilibrium with the environment.	It can be equal to zero when in equilibrium with the environment and in a dead state.
According to the first law of thermodynamics, it is conserved in all processes.	It is conserved for reversible processes but for irreversible (real) processes it is either partially or completely destroyed according to the second law of thermodynamics.
It cannot be created nor destroyed	It is always conserved for reversible processes. But it is never conserved and always destroyed for irreversible processes.
It occurs in many different types (such as KE, PE, heat and work etc.) and it is the capacity of movement.	It occurs in many different types (kinetic, potential, work and thermal exergy) and it is the capacity to do work.
It is only a measure of quantity.	It is a measure of both quantity and quality.

While energy and entropy cannot be correctly compared, energy and exergy can, as shown in the table above, because both terms have the same unit. Also, analysis and efficiency factors can be mentioned for both terms. Energy efficiency can be achieved with energy analysis, and in the same way, exergy efficiency can be achieved with exergy analysis. For this reason, energy and exergy concepts can be compared with each other under the headings of design, analysis and evaluation [30].

3.2.1 Exergy Components

The total exergy ($\dot{E}x$) of a system consists of four components, neglecting magnetic, nuclear, electric and surface tension effects [35]:

- Kinetic Exergy ($\dot{E}x_{ke}$)
- Potential Exergy ($\dot{E}x_{pe}$)
- Physical Exergy ($\dot{E}x_{ph}$)
- Chemical Exergy ($\dot{E}x_{ch}$)

$$\dot{E}x = \dot{E}x_{ke} + \dot{E}x_{pe} + \dot{E}x_{ph} + \dot{E}x_{ch} \tag{3.6}$$

3.2.1.1 Chemical Exergy

Chemical exergy is defined as the maximum amount of work that can be obtained from the restricted dead state to the dead state by the heat or material transfer occurring between the system and its environment [31].

When performing the exergy analysis of thermal power plants, it is generally assumed that the working fluid in the power plant does not undergo any chemical reaction during the cycle and thus does not expose any chemical change, so chemical exergy is ignored. Calculating the exergy of the fuel used in the power plant, on the other hand, is critical for exergy balance.

The standard chemical exergy of the substances has been calculated by reference to the properties of some proper environmental substances. Reference substances are divided into three groups in general; gas components in the atmosphere, solids in the lithosphere, and ionic and non-ionic substances in the hydrosphere.

3.2.1.2 The Exergy of Work

The maximum work potential of a certain amount of energy is defined as exergy and work can be transformed into other types of energy. Therefore, the entire work is equal to exergy directly.

$$\dot{E}x^{W} = \dot{W} \tag{3.7}$$

3.2.1.3 The Exergy of Heat Transfer

The general definition of a heat machine can be made as follows: A heat machine works by taking a certain amount of heat from the heat source at a certain temperature and gives a certain amount of heat to the environment at a certain temperature. The work resulting from this heat transfer is also equal to the exergy generated as a result of heat transfer. Based on this, the exergy equation resulting from heat transfer is expressed as follows for the system with the parameters of temperature (T), ambient temperature (T_0) and heat transfer per unit time ($\dot{Q}$):

$$\dot{E}x^{Q} = \dot{Q}\left[1 - \frac{T_0}{T}\right] \tag{3.8}$$

3.2.2 Exergy Analysis

Energy analysis is a method for analysing energy conversion systems that is based on the first law of thermodynamics. The entering and releasing energies are the two most important factors here. The released energy is divided into two categories: products and waste. The efficiencies calculated as a result of energy analysis are generally evaluated as energy amount ratios and are used to compare and assess various systems. The energy efficiencies obtained as a result of the energy analysis, however, do not indicate how close they are to the ideal. Furthermore, it fails to reveal the thermodynamic losses that occur in a system. Exergy analysis, on the other hand, eliminates many of the shortcomings of energy analysis. Exergy analysis is based on the second law of thermodynamics, so it can answer questions such as where the inefficiencies occur during the process, what is the magnitude, and what is the cause of the inefficiencies. While exergy analysis indicates the energy that can be used in the process occurring in a system, that is useful work. It also indicates the unavailable energy [32].

Energy and energy analysis of a system are generally carried out by following steps [32]:

a) First of all, the examined system should be detailed according to the depth of energy and exergy analysis.
b) Mass and energy balances should be written for each element/component that makes up the system to be used in energy and exergy analysis. Also, each input data (such as the amount of work, heat transfer value, etc.) of the system should be determined. Apart from these data, technical properties such as temperature and pressure of the system should also be determined.
c) A proper reference environment for the system where energy and exergy analysis will be performed should be determined.
d) Energy and exergy values should be calculated according to the properties of the specified reference environment and results should be evaluated.
e) Based on the calculated exergy values, proper exergy balance equations should be written for each element/component of the system. While writing the exergy balance equations, the irreversibilities that occur in the exergy analysis should also be specified.
f) Both energy and exergy efficiencies should be determined by applying the energy and exergy values in the defined efficiency equations.
g) Finally, all results (irreversibilities, losses, efficiencies, etc.) obtained as a result of energy and exergy analysis should be interpreted and, development and improvement options for the system should be evaluated.

Specific exergy can be written as follows,

$$ex = (h - h_0) - T_0(s - s_0) \tag{3.9}$$

Equation 3.9 is applied for the control volume having one inlet and one outlet where kinetic, potential and chemical exergies are neglected. For a control volume with more than one inlet and outlet, the exergy balance is written as follows;

$$\sum Ex^Q + \sum Ex^W + \sum_i Ex_i - \sum_o Ex_o - \dot{I} = 0 \tag{3.10}$$

In Equation 3.10, the first and second member indicate the exergy resulting from heat transfer and work exergy respectively. The third and fourth member of the equation express the inlet exergy (i) and outlet exergy (o) of the control volume. The last member ($\dot{I}$) in the equation denotes exergy destruction, also known as irreversibility. In real processes, the exergy inlet the control volume is greater than the exergy outlet and the difference between two of them makes the irreversibility.

3.2.2.1 Exergy Equations for Common Components

The exergy balance for a equipment k and the whole system are written as follows [35]:

$$\dot{E}x_{F,k} = \dot{E}x_{Pr,k} + \dot{E}x_{L,k} + \dot{E}x_{D,k} \tag{3.11}$$

$$\dot{E}x_{F,tot} = \dot{E}x_{Pr,tot} + \dot{E}x_{L,tot} + \sum_k \dot{E}x_{D,k} \tag{3.12}$$

where, subscript F refers to general fuel exergy, Pr refers to product exergy, exergy losses L and D refers to exergy destructions regardless of the type of fuel. In Equation 3.12, the total fuel exergy is the exergy of the streams inlet to the system, the total product exergy is the exergy of the streams outlet from the system, the total exergy loss is the total exergy lost by heat transfer between the system and the environment due to the temperature difference. The last term refers to the total exergy destruction caused by irreversibilities in the system. All losses resulting from heat transfer are included in exergy destruction with the assumption that there is no temperature difference between the system and its environment. Thus, $\dot{E}x_{L,k}$ becomes equal to zero and the exergy destruction is written as follows [34]:

$$\dot{E}x_{D,k} = T_0\dot{S}_{gen,k} \tag{3.13}$$

- **Compressor, Fan or Pump**

Product exergy and fuel exergy for compressor, fan or pump (Figure 3.4) can be written as follows:

$$\dot{E}x_{Pr} = \dot{E}x_2 - \dot{E}x_1 \tag{3.14}$$

$$\dot{E}x_{Pr} = \dot{E}x_2 - \dot{E}x_1 \tag{3.15}$$

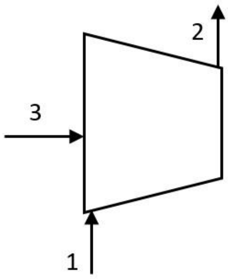

Figure 3. 1 Inlet and exit streams of compressor, fan or pump

- **Turbine or Expander**

Fuel exergy and product exergy for the turbine or expander (Figure 3.5) can be written as follow:

$$\dot{E}x_{Pr} = \dot{E}x_4 \tag{3.16}$$

$$\dot{E}x_F = \dot{E}x_1 - \dot{E}x_2 - \dot{E}x_3 \tag{3.17}$$

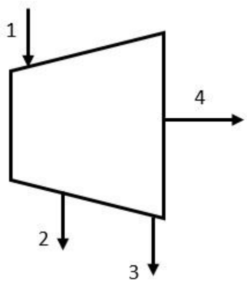

Figure 3. 2 Inlet and exit streams of turbine or expander

- **Heat Exchanger**

For the heat exchanger (Figure 3.6), whose purpose is to heat the cold flow, fuel and product exergy can be written as follows:

$$\dot{E}x_{Pr} = \dot{E}x_2 - \dot{E}x_1 \tag{3.18}$$

$$\dot{E}x_F = \dot{E}x_3 - \dot{E}x_4 \tag{3.19}$$

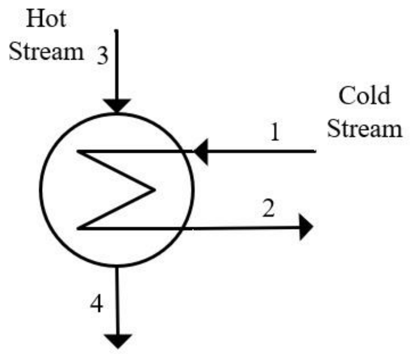

Figure 3. 3 Inlet and exit streams of heat exchanger

- **Mixing Chamber**

For the mixing chamber (Figure 3.7) where hot and cold streams are mixed, the product and fuel exergy can be written as follows:

$$\dot{E}x_{Pr} = \dot{E}x_3 \tag{3.20}$$

$$\dot{E}x_F = \dot{E}x_1 + \dot{E}x_2 \tag{3.21}$$

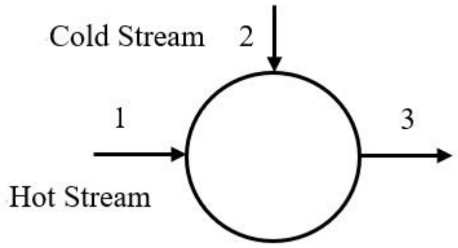

Figure 3. 4 Inlet and exit stream of mixing chamber

- **Combustion Chamber or Gasifier**

Product and fuel exergy for the combustion chamber or gasifier (Figure 3.8) can be written as follows:

$$\dot{E}x_{Pr} = \dot{E}x_3 \tag{3.22}$$

$$\dot{E}x_F = \dot{E}x_1 + \dot{E}x_2 \tag{3.23}$$

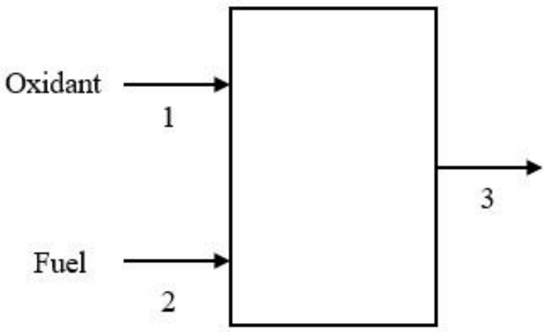

Figure 3. 5 Inlet and exit stream of combustion chamber or gasifier

- **Boiler**

The product and fuel exergy for the boiler (Figure 3.9) can be written as follows:

$$\dot{E}x_{Pr} = \left(\dot{E}x_6 - \dot{E}x_5\right) + \left(\dot{E}x_8 - \dot{E}x_7\right) \tag{3.24}$$

$$\dot{E}x_F = \left(\dot{E}x_1 + \dot{E}x_2\right) - \left(\dot{E}x_3 + \dot{E}x_4\right) \tag{3.25}$$

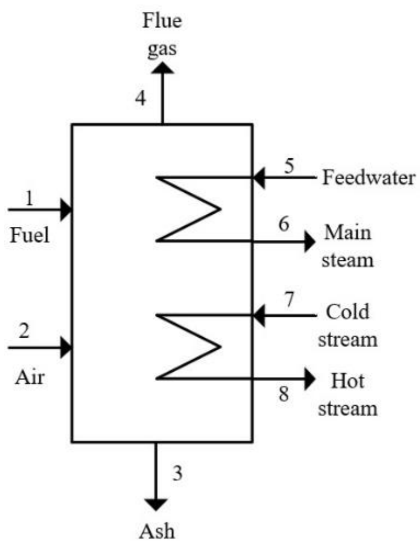

Figure 3. 6 Inlet and exit stream of boiler

3.2.2.2 Performance Criteria in Exergy Analysis

Unlike the energy basis performance criteria, Exergy-based performance criteria are based on the second law of thermodynamics. In exergy-based performance criteria, situations such as heat transfer, work, material flow occurring in the control volume of the examined system mean exergy input and output. In reversible processes, this exergy input and output are equal to each other and no exergy destruction occurs. In real (irreversible) processes, on the other hand, exergy input and output are never equal, and at the end of the process, exergy destruction always occurs due to irreversibilities. Exergy-based performance criteria also measure how close a system approaches the ideal state in which there are no irreversibilities.

Different exergy-based performance criteria such as rational efficiency and irreversibility ratio have been developed along with the widespread use of exergy analysis for thermal systems [34].

3.2.2.2.1 Rational Efficiency

Rational efficiency is defined as the ratio of exergy inlet a system to exergy outlet the system. Based on the exergy balance in Equation 3.10, the equation that contains all exergy and irreversibilities of the system is written as follows:

$$\sum \Delta \dot{E}x_i = \sum \Delta \dot{E}x_o + \dot{I} \tag{3.26}$$

Where, $\sum \Delta \dot{E}x_i$ and $\sum \Delta \dot{E}x_o$ refer to the total exergy inlet and outlet (or received) respectively, $\dot{I}$ refers to the total irreversibilities occurred in the system. Based on the exergy balance in Equation 3.26, the rational efficiency (ψ) is determined as follows:

$$\psi = \frac{\sum \Delta \dot{E}x_o}{\sum \Delta \dot{E}x_i} = 1 - \frac{\dot{I}}{\sum \Delta \dot{E}x_i} \tag{3.27}$$

Rational efficiency can be expressed by the Equation 3.27, as well as by the exergy of the products and fuels in the system. The rational efficiency of a component k can be written as the ratio of exergy of products to exergy of fuels as follows:

$$\psi_k = \frac{\dot{E}x_{Pr,k}}{\dot{E}x_{F,k}} = 1 - \frac{\dot{E}x_{D,k}}{\dot{E}x_{F,k}} \tag{3.28}$$

3.2.2.2.2 Irreversibility Ratio

The total exergy entering rate of a component in a system is called the irreversibility rate. This definition applies only to one component. The irreversibility ratio for the whole system is defined as the ratio of the irreversibilities occurring in the whole system or the exergy destruction occurring in the whole system to the total exergy entering the system [34]. The irreversibility rate for the component *k* is expressed as follows:

$$\delta_k = \frac{\dot{I}_k}{\sum \Delta\dot{E}x_i} \tag{3.29}$$

Similarly, the irreversibility ratio where the exergy destruction occurred for the component *k* and the total energy entering the system is expressed as total fuel exergy, is written as follows:

$$\delta_k = \frac{\sum_k \dot{E}x_{D,k}}{\sum \Delta\dot{E}x_F} \tag{3.30}$$

3.3 Thermoeconomics

Before moving on to thermoeconomics and thermoeconomic analysis, which is one of the important elements of engineering and includes both energy and economic analysis, combines both disciplines, and develops the equations to be used in the analysis, it is necessary to define the basic economic elements.

3.3.1 Economic Analysis and Economic Components

In order to analyse a thermal system comprehensively, it is necessary to perform energy and exergy analysis as well as economic analysis. The economic analysis provides various economic predictions and potential improvements for a thermal power plant that is being built or is already in operation.

The goals of economic analysis are can be stated like that, correctly estimating the investment costs of the thermal system or power plant, calculations of the product's cost on the basis of realistic and correct assumptions based on inflation, price increase, price balancing and taxes, and making profit analysis of many different investments [38].

3.3.1.1 Time Value of Money

The time value of money is a concept which states that a certain amount of money owned in the present is worth more than the same amount of money owned in the future.

3.3.1.2 Future Value of Money

When the *P* amount of money is deposited into an account to operate at a compound interest rate *i*, for *n* time periods, the future *FV* value of that deposit after n periods is calculated by the following equation [35]:

$$FV = P(1 + i)^n \tag{3.31}$$

3.3.1.3 Compound Interest Frequency

The year is generally used as a unit of time when analysing engineering systems economically. If the compound interest happens p times in a year, the future FV value of a P amount of money after *n* time is calculated with the following equation:

$$FV = P\left(1 + \frac{i}{p}\right)^{np} \tag{3.32}$$

where *i/p* is the interest rate per period, *np* is the number of periods and *i* is the nominal interest rate. The situation where compound interest occurs only once a year instead of p times in a year is called "effective interest rate" (i_{eff}) and is calculated as follows:

$$i_{eff} = \left(1 + \frac{i}{p}\right)^{p} - 1 \tag{3.33}$$

The effective interest rate is higher than the nominal interest rate. When Equation 3.62 is rearranged with the effective interest rate, it becomes as follows [35]:

$$FV = P\left(1 + i_{eff}\right)^n \tag{3.34}$$

3.3.1.4 Present Value of Money and Present Value Factor

It is important to know the present values of the expenditures in certain periods in the future and the obtained revenues for the projects whose economic analysis will be carried out. The present value PV is calculated by the following equation as a result of depositing a certain FV amount of money in the future into an account with a certain interest rate [35]:

$$PV = FV \frac{1}{\left(1 + i_{eff}\right)^n} \tag{3.35}$$

Discount rate is defined as the difference between the present value of money and its future value. Based on this definition, when the i_{eff} value is accepted as the discount rate, the present value factor of money (PVF) is calculated as follows [35]:

$$PVF = \frac{1}{\left(1 + i_{eff}\right)^n} \tag{3.36}$$

3.3.1.5 Annuity

An annuity is defined as the movements of equal amounts of money that take place at specified equal time intervals. The time interval is generally accepted as one year. These types of money movements are used to pay off a particular debt or to accumulate capital. Annual payments (AC) of expenses such as fuel, product, employee payments are calculated with the following equation [35]:

$$AC = FV \frac{i_{eff}}{(1 + i_{eff})^n - 1} \tag{3.37}$$

3.3.1.6 Capital Recovery Factor (CRF)

The present value of an annual payment is expressed as the value of the total annual payment at the end of a specified period when it is deposited at the beginning of the annual payment at the effective interest rate and is calculated by the following equation [35]:

$$\frac{PV}{AC} = \frac{(1 + i_{eff})^n - 1}{i_{eff}(1 + i_{eff})^n} \tag{3.38}$$

While the right side of Equation 3.38 is called the Uniform-Series Present-Worth Factor (US PWF), the left-hand side is called the Capital-Recovery Factor (CRF) and is expressed as follows [35]:

$$CRF = \frac{AC}{PV} = \frac{i_{eff}(1 + i_{eff})^n}{(1 + i_{eff})^n - 1} \tag{3.39}$$

3.3.2 Thermoeconomic Analysis

In order to achieve the most appropriate and efficient design, techniques that combine different scientific disciplines are used in the analysis and design of thermal systems. Among these scientific disciplines, thermodynamics-based exergy analysis and cost-based economic analysis for thermal systems stand out as the two most commonly used and best suited for optimum design. In thermal systems with energy-converting components, energy-based unit costs are generally taken into consideration for costs. Many researchers recommend that costs for thermal systems should be energy-based. What is meant by the energy here is the exergy term used in real systems. Since the unit costs realized on the basis of exergy are based on the second law of thermodynamics, it is more consistent than the energy-based costs. Since exergy-based unit cost analysis includes both exergy analysis and economic analysis, it has been named thermoeconomics by the researchers [39].

A correct and complete thermoeconomic analysis consists of the following steps [40];

- Energy analysis
- Exergy analysis
- Exergy based costing
- Evaluation of each system component in terms of thermoeconomic

3.3.3 Exergoeconomics

The exergoeconomics, which is derived from the words “exergo” which means work generation and work potential in Greek, and the economy, is an analysis method that includes both exergy analysis and economic analysis and is widely used by researchers [41].

Exergoeconomics is an engineering branch that combines exergy analysis and economic analysis appropriately for each component of the thermal system and the whole thermal system. It provides information that cannot be obtained with traditional energy or exergy analysis and economic analysis in system design where cost is important and in the operation of that system. Exergoeconomics interacts with different engineering fields such as process synthesis, thermodynamic simulation, economic analysis, process analysis and evaluation, mathematical optimization methods, expert and fuzzy systems and optimization of energy conversion systems and it offers different optimization methods [42].

3.3.4 Thermoeconomic Analysis Methods

In thermoeconomic and exergoeconomic analyses, the costs of exergy at the input and output of each component of a thermal system are determined. Product cost obtains with these determined values and auxiliary equations. Costs of mass and energy streams for each component of the thermal system are calculated with the help of additional cost equations and cost balance written separately for each component [43].

There are many different methods in the literature for thermoeconomic analysis [44]:

- Exergy Economic Approach-EEA [45]
- Thermoeconomics Functional analysis-TFA [46]
- First Exergoeconomic Approach-FEA [47]
- Exergetic Cost Theory-ECT [48]
- Engineering Functional Analysis-EFA [49]
- Last-In-First-Out Approach-LIFOA [50]
- Structural Analysis Approach-SAA [51]
- Specific Exergy Costing-SPECO [44]
- Exergy-Cost-Energy-Mass Analysis-EXCEM [52]
- Modified Productive Structure-MOPSA [53]

In this thesis, exergy economic analysis will be conducted by using the specific exergy costing (SPECO) method.

3.3.4.1 Specific Exergy Costing (SPECO) Method

SPECO method is an exergoeconomic analysis method developed by Lazzaretto and Tsatsaronis [44] and consists of three steps:

- Determination of exergy streams: In the first stage, the exergy streams at the inlets and outlets of all components of the system are determined.
- Identification of fuel and product: In the second stage, the determined exergy flows are expressed as fuel and product and are determined separately for each system.
- Calculation of costs: In the last stage, the costs of the calculated exergy stream values are determined.

The applications included in the first and second steps represent the exergy analysis of the two fundamentals of the exergoeconomic analysis. After performing exergy analysis for the first and second step, the last step, which is cost calculation, is started. The last step includes the economic analysis of exergoeconomics. The mass and energy streams entering and leaving the system, heat and work exergies values, are transformed into the cost of the inputs, the cost of the outputs, the cost of work and the cost of heat [34].

3.3.4.2 Thermoeconomic Components

Cost accounting for a company is usually accomplished to determine the true cost of a product or service, to create a rational basis for the product or service and to control expenses [34]. The cost balance for a thermal system in steady state can be written as follows. While $\dot{C}$ in this equation expresses the cost of exergy streams such as mass, heat transfer, work exergy, the term $\dot{Z}$ refers to all costs other than these costs:

$$\dot{C}_{Pr,tot} = \dot{C}_{F,tot} + \dot{Z}_k \tag{3.40}$$

where, $\dot{C}_{Pr,tot}$ represents the total exergy cost of the products, $\dot{C}_{F,tot}$ refers to the total exergy cost of fuels and $\dot{Z}_k$ represents the operating and maintenance costs and initial investment costs for a component k. $\dot{Z}_k$ can be written with its components as follows:

$$\dot{Z}_k = \dot{Z}_k^{CI} + \dot{Z}_k^{OM} \tag{3.41}$$

$\dot{Z}_k^{CI}$ refers to the initial investment cost and is calculated as the ratio of the annual initial investment cost to the annual working hours. $\dot{Z}_k^{OM}$ refers to the operating and maintenance cost and is calculated as the ratio of the annual operating and maintenance cost to the annual working hours:

$$\dot{Z}_k^{CI} = \frac{C_k^{CI}.CRF}{8600} \tag{3.42}$$

$$\dot{Z}_k^{OM} = \frac{C_k^{OM}}{8600} \tag{3.43}$$

In this thesis, the annual working hour has been accepted as 8600 hours.

3.3.4.2.1 Exergy Costing

In exergy costing, one cost correlates with each of exergy stream. Exergy cost balances are written as follows for exergy transfer terms resulting from the mass flows inlet ($\dot{E}_i$) and outlet the system ($\dot{E}_o$), work ($\dot{E}_w$) and heat transfer ($\dot{E}_q$) occurring in the system:

$$\dot{C}_i = c_i \dot{E}_i = c_i (\dot{m}_i . \dot{e}_i) \tag{3.44}$$

$$\dot{C}_o = c_o \dot{E}_o = c_o (\dot{m}_o . \dot{e}_o) \tag{3.45}$$

$$\dot{C}_w = c_w \dot{E}_w = c_w \dot{W} \tag{3.46}$$

$$\dot{C}_q = c_q \dot{E}_q \tag{3.47}$$

The terms c_i, c_o, c_w and c_q refer to the average unit cost of the exergy streams of the inlet and exit mass flows and the work and heat transfer terms in the system, respectively, and their units are *\$/GJ*. The terms $\dot{C}_i$, $\dot{C}_o$, $\dot{C}_w$ and $\dot{C}_q$ refer to the total costs of these average unit costs of exergy streams, respectively, and their units are *\$/h* or *\$/s*.

Exergy costing includes cost balances formulated separately for each component of the system. In the exergy cost balance constituted for a component k of the system, the total cost of the products is equal to the total exergy cost of the flows inlet the system, the initial investment cost and the sum of the operating and maintenance costs. The exergy cost balance for a component k that receives heat from its environment and produces work is written as follows:

$$\sum_o \dot{C}_{o,k} + \dot{C}_{w,k} = \dot{C}_{q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k \tag{3.48}$$

Equation 3.80 states that the overall exergy costs of exergy flows are equal to the total expense spent to obtain them. This equation can be written more clearly as follows:

$$\sum_{o}(c_o\dot{E}_o)_k + (c_w\dot{E}_w)_k = (c_q\dot{E}_q)_k + \sum_{i}(c_i\dot{E}_i)_k + \dot{Z}_k \tag{3.49}$$

3.3.4.2.2 Exergy Destruction Cost

Exergy balance for a component k and the entire system was written in Equations 3.40 and 3.41 in the energy and exergy analysis chapter. There, it was accepted that there was no temperature difference between the system and its environment, and thus all losses due to heat transfer were included in exergy destructions. As a result, it was stated that the term resulting from exergy losses $\dot{E}x_{L,k}$ for a component k is equal to zero and the exergy destruction $\dot{E}x_{D,k}$ is equal to the multiplying of $T_0\dot{S}_{gen,k}$.

The following equation expresses that the exergy destruction is compensated by the addition of fuel and the fuel exergy unit cost does not change in the exergy destruction cost calculation:

$$\dot{C}_{D,k} = c_{F,k}\dot{E}_{D,k} \text{ (constant } \dot{E}_{Pr,k}) \tag{3.50}$$

Similarly, the following equation expresses that less product is obtained with the same amount of fuel and it also expresses the assumption that, the exergy destruction cost increases the total cost of the product is as follows:

$$\dot{C}_{D,k} = c_{Pr,k}\dot{E}_{D,k} \text{ (constant } \dot{E}_{F,k}) \tag{3.51}$$

3.3.4.2.3 Thermoeconomic Equations

Some of the thermal system components have multiple product exergy streams. The main cost equations alone are insufficient to form thermoeconomic equations for such components. Auxiliary equations are used to compensate for this shortcoming. A sufficient number of auxiliary equations are required for thermoeconomic analysis of thermal system components. Some methods known in the literature as the F and P principle are used to write these auxiliary equations [43].

- The *F (Fuel) principle* states that when the exergy difference occurring at the inlet and exit of the component is evaluated as fuel, the exergy cost ($\dot{C}_F$) is the same as the exergy value at the component where the difference occurs. Using this method, an additional auxiliary equation can be written for each exergy stream that occurs at the component's output [44].
- The *P (product) principle*, on the other hand, refers to the situation of adding exergy stream to the system component in addition to the exergy stream already it has. According to the P principle, each exergy stream added to the streams which are defined as a product for a system component is at the same average cost ($\dot{C}_{Pr}$)). This principle provides an additional auxiliary equation one less than the number of product streams the equipment has. The number of exergy streams of the system component is equal to the number of exergy streams obtained by both methods. In summary, by applying both principles, the N_e number of auxiliary equations needed for the total number of exergies can be calculated as N_{e-1} [44].

In thermoeconomic analysis, as in energy and exergy analysis, thermoeconomic main cost equations and auxiliary cost equations can be obtained for system components in steady-state. These equations are included below for compressor, fan or pump group, turbine or expander group, heat exchanger, mixing unit, Combustion Chamber and boiler [34].

- **Compressor, Fan or Pump**

Product cost and fuel cost equations for compressor, fan or pump (Figure 3.10) can be written as follows:

$$\dot{C}_{Pr} = \dot{C}_2 - \dot{C}_1 \tag{3.52}$$

$$\dot{C}_F = \dot{C}_3 \tag{3.53}$$

There is no auxiliary equation for this equipment group. The value calculated from the cost equation is c_2.

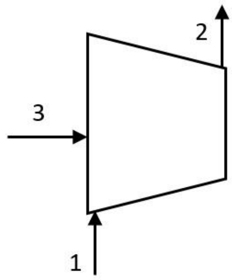

Figure 3. 7 Inlet and exit cost streams of compressor, fan or pump

- **Turbine or Expander**

Product cost and fuel cost equations for turbine or expander (Figure 3.11) can be written as follows:

$$\dot{C}_{Pr} = \dot{C}_4 \tag{3.54}$$

$$\dot{C}_F = \dot{C}_1 - \dot{C}_2 - \dot{C}_3 \tag{3.55}$$

Auxiliary equations for the turbine or expander group can be written as follows:

$$c_2 = c_3 = c_1 \tag{3.56}$$

The value calculated from the cost equation is c_4 for the turbine group.

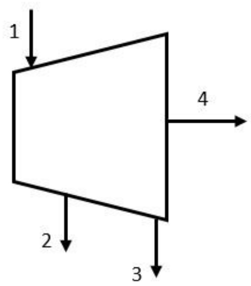

Figure 3. 8 Inlet and exit cost streams of turbine or expander

- **Heat Exchanger**

The product cost and fuel cost equations for the heat exchanger (Figure 3.12), whose purpose is to heat the cold stream, can be written as follows:

$$\dot{C}_{Pr} = \dot{C}_2 - \dot{C}_1 \tag{3.57}$$

$$\dot{C}_F = \dot{C}_3 - \dot{C}_4 \tag{3.58}$$

Auxiliary equations for the heat exchanger can be written as follows:

$$c_3 = c_4 \tag{3.59}$$

The value calculated from the cost equation is c_2 for the heat exchanger.

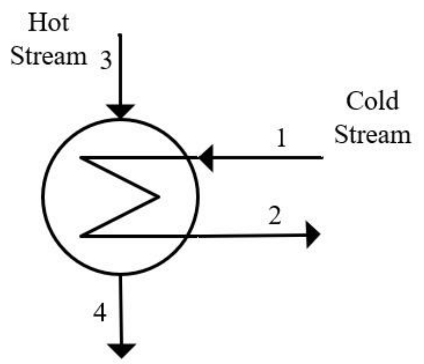

Figure 3. 9 Inlet and exit cost streams of heat exchanger

- **Mixing Chamber**

The product cost and fuel cost equations for the mixing chamber (Figure 3.13) can be written as follows:

$$\dot{C}_{Pr} = \dot{C}_3 \tag{3.60}$$

$$\dot{C}_F = \dot{C}_1 + \dot{C}_2 \tag{3.61}$$

There is no auxiliary equation for the mixing chamber. The value calculated from the cost equation is c_3.

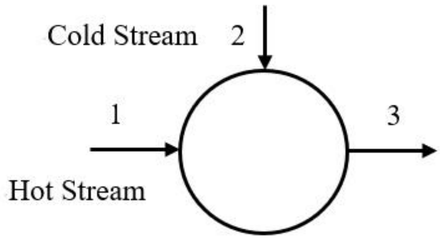

Figure 3. 10 Inlet and exit cost streams of mixing chamber

- **Combustion Chamber or Gasifier**

The product cost and fuel cost equations for the combustion chamber or gasifier (Figure 3.14) can be written as follows:

$$\dot{C}_{Pr} = \dot{C}_3 \quad (3.62)$$

$$\dot{C}_F = \dot{C}_1 + \dot{C}_2 \quad (3.63)$$

There is no auxiliary equation for the combustion chamber or gasifier. The value calculated from the cost equation is c_3.

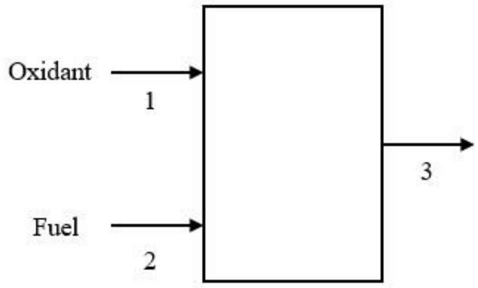

Figure 3. 11 Inlet and exit cost streams of combustion chamber or gasifier

- **Boiler**

The product cost and fuel cost equations for the boiler (Figure 3.15) can be written as follows:

$$\dot{C}_{Pr} = (\dot{C}_6 - \dot{C}_5) + (\dot{C}_8 - \dot{C}_7) \tag{3.64}$$

$$\dot{C}_F = (\dot{C}_1 + \dot{C}_2) + (\dot{C}_3 + \dot{C}_4) \tag{3.65}$$

The auxiliary equations for the boiler can be written as follows:

$$\frac{(\dot{C}_6 - \dot{C}_5)}{\dot{E}_6 - \dot{E}_5} = \frac{(\dot{C}_8 - \dot{C}_7)}{\dot{E}_8 - \dot{E}_7} \tag{3.66}$$

The value calculated from the cost equation is c_6 or c_8 for the boiler.

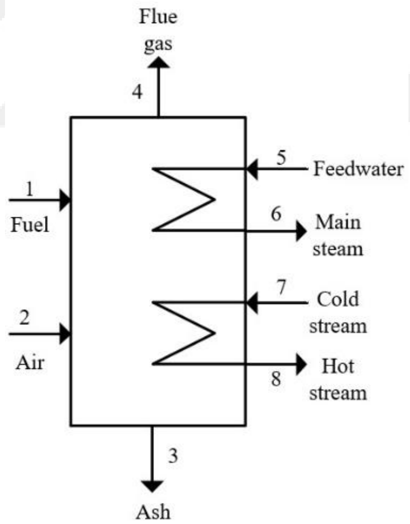

Figure 3. 12 Inlet and exit cost streams of boiler

3.3.4.3 Performance Criteria in Thermoeconomic Analysis

A system in the design phase or a system that is already in operation can be evaluated in thermoeconomics way. Both systems' thermoeconomic evaluations are carried out in parallel. Because a system in the design phase is not yet available, the initial investment cost is only included in the assessment for systems that are already installed. For systems in the design phase, operation and maintenance costs are also ignored [37].

The following evaluation criteria are taken into consideration while evaluating a thermal system that is in the design phase from a thermoeconomics way. These factors for a component k of a thermal system can be listed as follows [36]:

- Exergetic efficiency
- Exergy destruction and exergy loss
- Exergy destruction ratio and exergy loss ratio
- Capital investment cost, operating and maintenance cost, and their sum
- Exergy destruction cost and exergy loss cost
- Relative cost difference, r_k
- Exergoeconomic factor, f_k

The relative cost difference (r_k) is calculated as the ratio of the difference between the exergy costs of the product and the fuel and the exergy cost per unit using the following formula:

$$r_k = \frac{c_{Pr,k} - c_{F,k}}{c_{F,k}} \tag{3.67}$$

Exergoeconomic factor (f_k) gives information about the sources of the main costs and is calculated with the following equation.

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{F,k}(\dot{E}x_{D,k} + \dot{E}x_{L,k})} \tag{3.68}$$

The following methods are recommended to be followed in order for a thermal system consisting of multiple components to be cost-effective [35]:

a. All components of the thermal system are sorted in order of decreasing cost, using the sum of $\dot{Z}_k + \dot{C}_{D,k}$.
b. The designs of the components are reviewed, especially those for which this total is high.

c. The relative cost difference of components with high $\dot{Z}_k$ and $\dot{C}_{D,k}$ values is given special consideration.
d. The main cost resources (capital investment or cost of exergy destruction) are evaluated by considering the exergoeconomic factor.
    - If the exergoeconomic factor is high, the option to reduce the initial investment cost is considered by taking into account the changes in the efficiency of the components.
    - If the exergoeconomic factor is low, the efficiency of the components is tried to be increased by increasing the initial investment cost.
e. Sub-components with high exergy destructions or exergetic losses and that do not contribute to the initial investment cost or fuel cost are removed from the system.
f. It is tried to increase the exergy efficiency of the components whose exergy destruction and exergetic losses are high and whose exergy efficiency is low compared to others.

It is emphasized that when using these methods, thermoeconomic variables should be applied specifically for each component, and each component should be evaluated within its own group. Compressors and pumps, for example, should be evaluated among themselves, with the same manner heat exchangers should be evaluated in themselves [35].

3.4 LIFE CYCLE ASSESSMENT

The increase in the world population, industrial, technological and commercial developments increase the needs of human being day by day. More production, more raw materials and energy usage are required to meet the growing needs. Nowadays, concerns are increasing not only about how to meet these needs but also about the depletion of natural resources and environmental degradation. With the increasing environmental awareness, new methods have been developed to alleviate these concerns and to minimize the negative effects of systems, products and services on the environment. Life cycle assessment (LCA), one of these methods, emerges as a result of environmental impact assessment awareness.

In the periods after the 2000s known as the "Decade of Elaboration", LCA studies gained momentum. In these years, an International Life Cycle Partnership was realized between The United Nations Environment Program (UNEP) and The Society of Environmental Toxicology and Chemical (SETAC) for the purposes such as more effective use of data and indicators in the LCA method and dissemination of LCA [55].

3.4.1 Methodological Framework of LCA

LCA methodology consists of four stages according to ISO standards [58, 59]:

- Goal and scope definition
- Inventory analysis (Life Cycle Inventory analysis, LCI)
- Impact assessment (Life Cycle Impact Assessment, LCIA)
- Interpretation

These four stages are related to one another in the LCA methodology. Figure 3.16 depicts the relationship of the stages to one another and the overall framework of the LCA methodology.

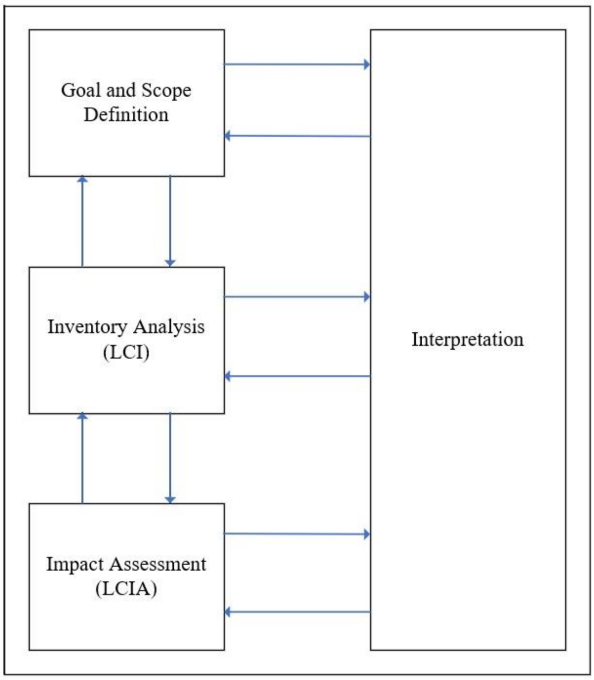

Figure 3. 13 Stages of the LCA [58, 59]

3.4.1.1 Goal and Scope Definition

The first step of the LCA study is to define the purpose and scope. The first stage is crucial as it has a significant impact on the outcome of the LCA study. For this reason, it is necessary to clearly define the purpose and scope of an LCA study to be carried out.

The scope definition states the limitations of the LCA study. The detail, span and depth of the study should be compatible with the purpose for which it is stated and sufficient for the study. The system and its boundary, functional unit, impact categories, allocation procedures, impact assessment methods and assumptions should be defined for the scope definition. Since the LCA study includes the data collection procedure and is an iterative evaluation method, the scope can be changed during the study [58, 59].

3.4.1.1.1 Functional Unit and System Boundary

The functional unit is defined as a reference that allows the different results to be compared with each other. The main goal of specifying the functional unit is normalizing the data and comparing different systems with each other. The functional unit is selected from three different aspects: stability, efficiency and performance quality of the product. In the inventory analysis stage, which is the next stage after the definition of purpose and scope, all data obtained are expressed in functional units. The reference point for the functional unit must be defined. It is used to select the functional unit, and the units to be compared in LCA studies must have the same reference point. Examples of functional units are 1-ton feedstock, 1-L fuel, 1-km driving [56]. When the functional unit for electricity generation of a natural gas-fired combined cycle power plant is selected as 1 kWh, the reference flow becomes 10 gCO_2/kWh [59].

The system boundary includes the unit processes involved in the LCA study and shows what is included in the study. The purpose and scope should be consistent with the system boundary. In holistic LCA studies, the system boundary covers the entire cradle-to-grave process and includes all stages from the production of the substance to waste disposal [60].

3.4.1.1.2 Allocation Procedure

The separation of inlet and exit streams of a system, between the worked product system and other product systems, is defined as allocation [59, 60]. In allocation, only the considered environment is important for products. In other words, for each product, a system has, only the environmental effects of that system are important. The allocation procedure is applied if there is more than one product in the system, such as common products, recycling and reuse systems. The processes of the systems examined in LCA applications produce more than one product and the application of allocation directly affects the results of the LCA studies. For this reason, the allocation procedure must be applied correctly according to ISO standards. Allocation procedures for different systems can be made according to the needs of that system. System extension and partitioning methods are depicted among the most used allocation procedures [57, 61].

3.4.1.1.3 Impact Categories

Impact categories are defined as an element that aims to reveal the results of the LCI analysis, to present the results of the environmental categories and the obtained results in environmental hazard categories. Impact categories of LCA studies include the following [54]:

- Global Warming
- Ozone Depletion
- Acidification
- Eutrophication
- Toxicity
- Human Health
- Resource Depletion
- Land Use
- Water Use

3.4.1.2 Inventory Analysis

Life cycle inventory analysis (LCI) is the stage that includes the gathering of data that will be used in the LCA study for a system. The necessary input and output data are measured at the system boundaries at this stage. The obtained data from the LCI are presented in a tabular form. Since collecting and organizing these data is a very difficult and time-consuming process, LCI is considered the most difficult phase in LCA studies. Data can usually be collected from databases of LCA software, from the facilities and power plants, and previous studies. Since each phase of the LCA study is interrelated, the data collected under the LCI affect other phases as well. For this reason, the accuracy of the collected data is one of the most important factors in the success of the LCA study [57, 60]. Inventory analysis stages according to ISO standards are as in Figure 3.17.

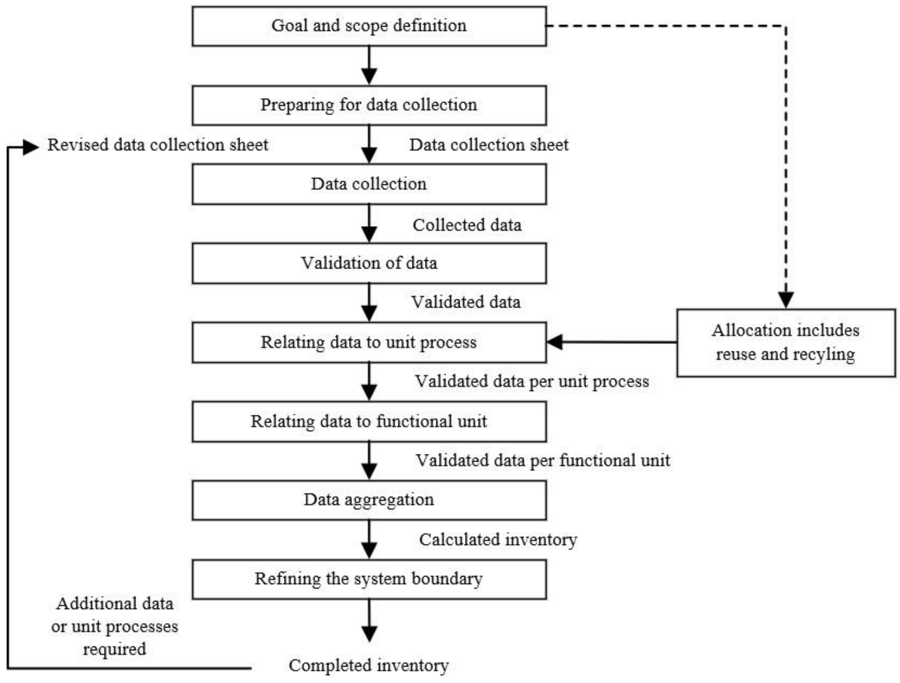

Figure 3. 14 Steps of Life Cycle Inventory analysis [58, 59]

3.4.1.3 Impact Assessment

In the life cycle impact assessment (LCIA) phase, the amount and significance of potential environmental impacts of a system or product are evaluated using the results obtained from the inventory analysis phase for the same system or product. In other words, the results obtained in the LCI stage are transformed into potential environmental impacts at this stage. The difference of the LCIA method from other assessment methods such as environmental performance assessment, risk assessment and environmental impact assessment is that it is based on a functional unit and has a relative approach [58, 59].

The LCIA phase has two elements, mandatory elements and optional elements. Mandatory elements of the LCIA phase include the selection of impact categories, impact category indicators and LCIA characterization models, classification and characterization. Optional elements of the LCIA phase are normalization, grouping, and weighting. LCIA elements are shown in Figure 3.18 [58, 59].

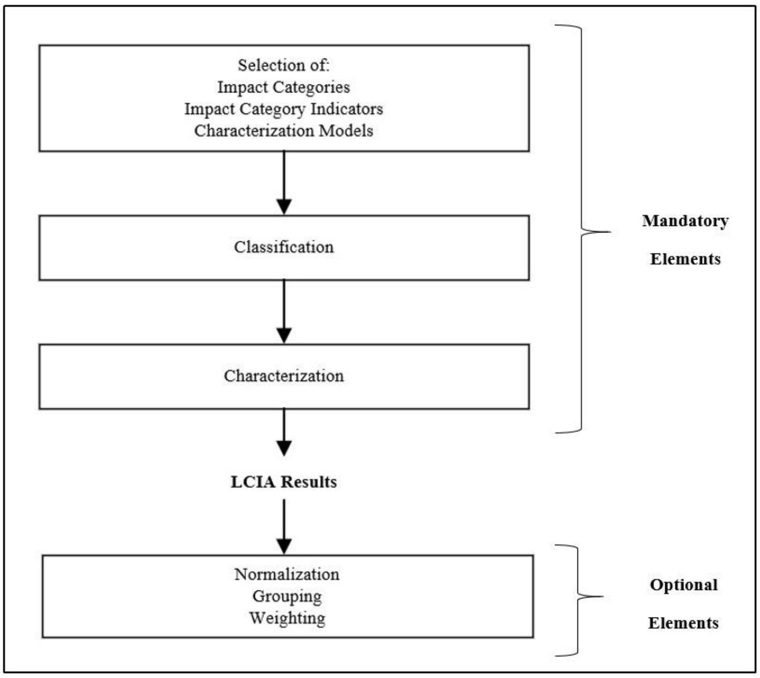

Figure 3. 15 The elements of LCIA [58, 59]

Table 3. 2 Examples of LCI categories of emissions

Impact Category	**Scale**	**Examples of LCI Data (i.e., classification)**
Global Warming	Global	Carbon Dioxide (CO_2), Carbon Monoxide (CO), Nitrogen Dioxide (NO_2), Methane (CH_4)
Stratospheric Ozone Depletion	Global	Chlorofluorocarbons (CFCs), Hydrochlorofluorocarbons (HCFCs), Halons, Methyl Bromide (CH_3Br)
Acidification	Regional, Local	Sulphur Oxides (SO_x), Nitrogen Oxides (NO_x), Hydrochloric Acid (HCL), Ammonia (BH_4)
Eutrophication	Local	Phosphate (PO_4), Nitrogen Oxide (NO), Nitrogen Dioxide (NO_2)
Terrestrial and Aquatic Toxicity	Local	Organic pollutants, Pesticides, Terrestrial livers, Aquatic livers
Human Health	Global, Regional, Local	Total releases to air, Water, Soil
Resource Depletion	Global, Regional, Local	Amount of minerals used Amount of fossil used
Land Use	Global, Regional, Local	Disposal in a landfill
Water Use	Regional, Local	Quantity of consumed water

3.4.1.4 Interpretation

The interpretation phase is the final stage of the LCA. The results obtained from the LCI and LCIA stages are evaluated and interpreted at this stage. The purpose at the interpretation stage is to provide understandable results according to the definition of goal and scope defined in the first stage of the LCA, and the qualifications specified in the first stage. In the interpretation phase, all of the LCI and LCIA results are evaluated, the main impacts in the life cycle are determined, and the final results are obtained. Based on the results, comments and suggestions are made. It is aimed that these comments and suggestions will aid decision-making mechanisms. Figure 3.19 depicts the relationship between the interpretation phase, which is the final phase of the LCA study, and the other phases [58, 59].

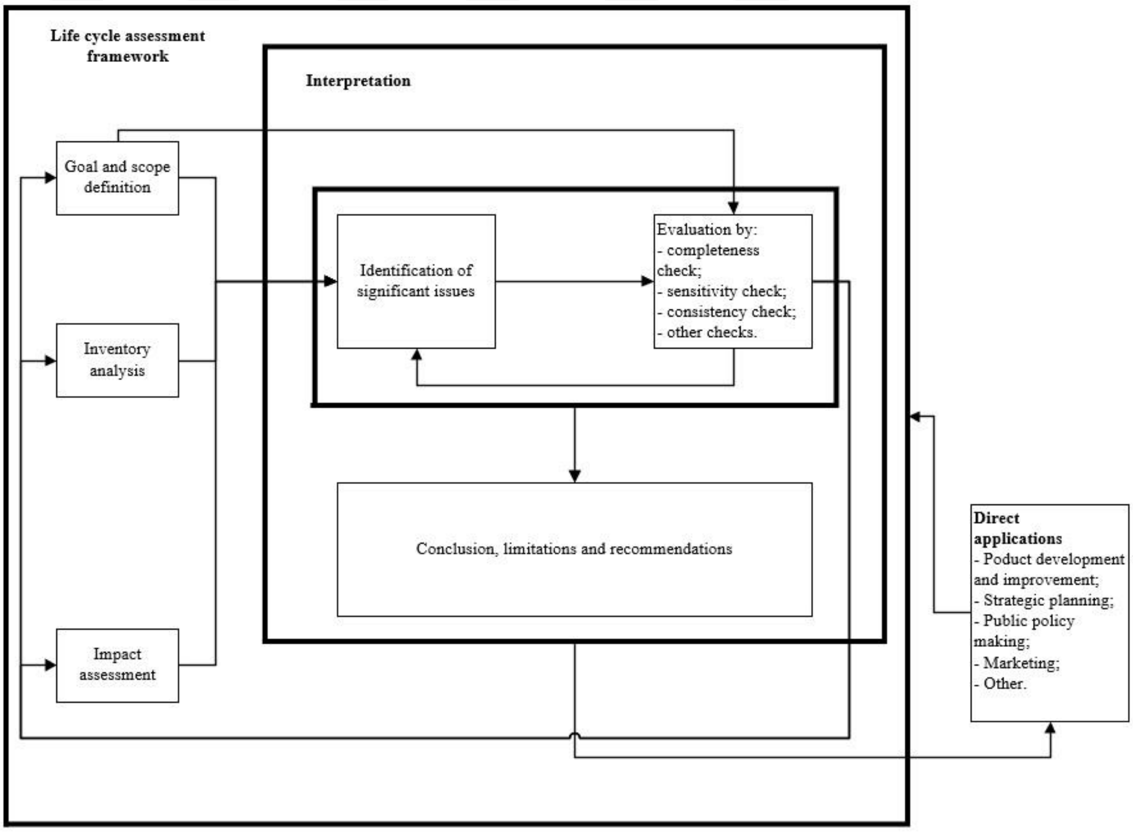

Figure 3. 16 Relationships between the interpretation phase with the other phases of LCA [58]

3.4.2 Life Cycle Assessment Software

Since LCA requires a large number of data processing, software have great importance for LCA studies. Many procedures such as data, calculations and processes are carried out by software that has an LCA database developed by different companies. LCA software enables researchers to comment on the study by performing the necessary procedures and processes. GaBi Software [61], SimaPro [62], Umberto [63], opanLCA [64], REGIS [65] and GREET [66] are some of the software used in LCA studies.

3.4.3 Exergoenvironmental Analysis

When thoroughly analysing a thermal system, environmental analysis should be performed in addition to energy, exergy, and exergoeconomic analyses. With these analyses, the thermodynamic and economic inefficiencies, as well as the environmental effects of the systems, can be determined, and thus optimal solutions for the system, which is in the design phase or is already operational, can be proposed. Thermodynamic inefficiencies of thermal systems consist of exergy destructions caused by the irreversibilities in the system and exergy losses caused by the discharge of exergy into the environment. Detection of all inefficiencies in thermal systems is possible by performing exergy analysis in each analysis method. Over the years, exergetic and environmental analysis have been combined with different approaches. Some of these approaches are as follows [67]:

- Cumulative exergy consumption [68],
- Exergoecological analysis [69],
- Extended exergy accounting [70],
- Environomic analysis [71], and
- Exergoenvironmental analysis [72]

The analysis method in which environmental impacts are evaluated together with exergy is called "exergoenvironmental" analysis. As in exergoeconomic analysis, in exergoenvironmental analysis, the location, magnitude and environmental effects of exergy destructions caused by irreversibilities in the system are determined. Exergy analysis is combined with life cycle assessment in the exergoenvironmental analysis [73]. The exergoenvironmental analysis is performed for both a system and each component of the system. The options are obtained to reduce these effects by determining the environmental impact of both the component and the system [67].

Exergoenvironmental analysis includes the LCA study. Therefore, each stage of the LCA and each step of these stages are included in the exergoenvironmental analysis. In the LCA study performed for a thermal system, the life cycle assessment is performed for the input streams (including the supply of fuels) and the components of the system. Inventory analysis is carried out with factors such as consumption of natural resources and energy, emissions etc. In addition, the environmental impact assessment is performed using a quantitative indicator [72]. There are many different impact assessment methods in the literature. Some of these methods are Environmental Priority Strategies (EPS), Environmental Development of Industrial Products (EDIP), ReCiPE, Eco-indicator 95 and Eco-indicator 99 [73]. One of these methods, Eco-indicator 99, is widely used because it is compatible with LCA studies. Eco-indicator 99 is a method developed to determine the possible effects of a system in the design phase. Figure 3.20 shows the general structure of the Eco-indicator 99 method [74].

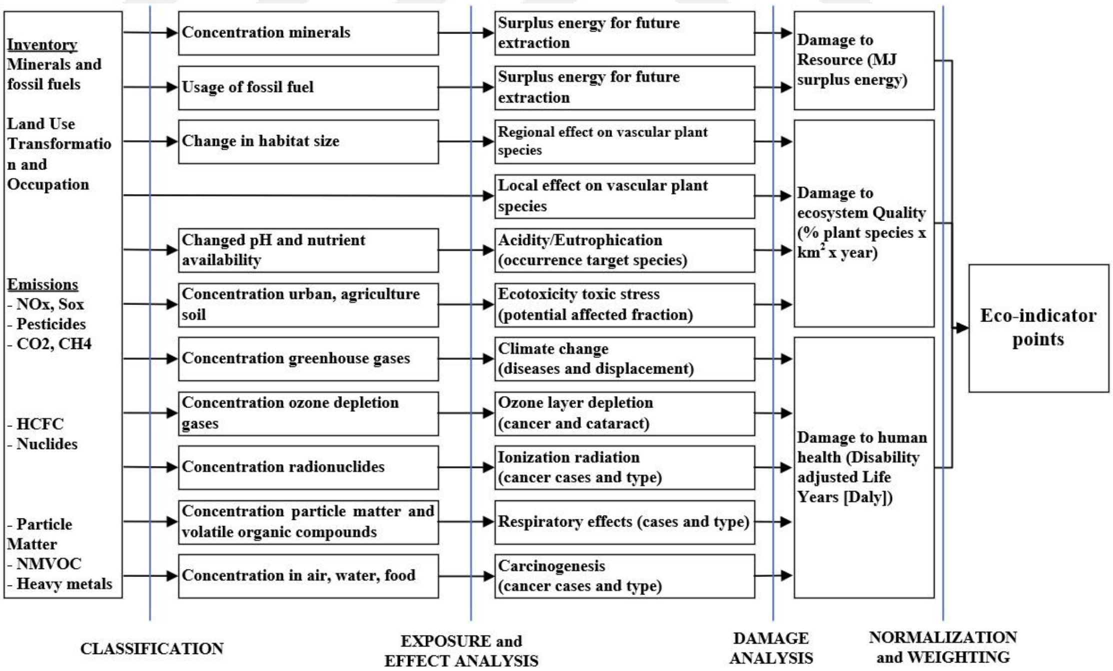

Figure 3. 17 General structure of Eco-indicator 99 [72]

The Eco-indicator 99 method assigns source analysis, soil use analysis, or fate analysis to each component based on the water-soil-air sections where problems may arise. Thus, the classification of potential environmental problems is realized. Following the identification of environmental problems, environmental impacts are examined in three different damage categories: human health, ecosystem quality, and impact on natural resources. These three damage categories are standardized and Eco-indicator points are assigned at the last stage. The high eco-indicator score means high damage and the scores obtained can be calculated from the database of computer programs [74]. Table 3.5 shows sample Eco-indicator 99 scores [72].

Table 3. 3 Examples of Eco-indicator 99 points of some emissions [74]

Emission or product	**Eco-indicator 99 (Pts)**
1 kg CO_2 emitted	0.0054545
1 kg CH_4 emitted	0.1146225
1 kg N_2O emitted	1.7922000
1 kg NO_x emitted	2.7493600
1 kg SO_x emitted	1.4993700
1 kg CO emitted	0.0083636
1 kWh electricity in Germany	0.01302

It is important to know the weight of the component and the required raw material to produce one of the components for the purpose of dimension the system. However, since this information is not usually found in the literature, calculations are made roughly [73, 75].

Exergoenvironmental analysis consists of three stages. These are as follows, respectively [57]:

- Exergy analysis for each component of the system
- Performing of LCA for all components in the system and each stream entering the system
- The assignment of environmental effects obtained by LCA to exergy streams.

As in the Exergoeconomic analysis, in the exergoenvironmental analysis, the environmental effect of the component of the system is assigned to exergy streams. The environmental impact ratio ($\dot{B}_j$) of any mass flow of j is calculated as follows [72]:

$$\dot{B}_j = b_j \dot{E}x_j \tag{3.69}$$

where, $\dot{B}_j$ represents the environmental impact rate and its unit is Eco-indicator points per unit time (Pts/s). b_j is a specific environmental impact, and $\dot{E}x_j$ is the exergy ratio. The relation of environmental effects with the transfer of work ($\dot{B}_w$) and heat ($\dot{B}_q$) is expressed by the following equations, respectively [72]:

$$\dot{B}_w = \dot{b}_w \dot{W} \tag{3.70}$$

$$\dot{B}_q = \dot{b}_q \dot{E}x_q \tag{3.71}$$

The environmental impact ($\dot{Y}_k$) of any k component of the system is calculated with the Equation 3.72:

$$\dot{Y}_k = \dot{Y}_k^{CO} + \dot{Y}_k^{OM} + \dot{Y}_k^{DI} \tag{3.72}$$

The term $\dot{Y}_k^{CO}$ refers to the environmental impacts in the manufacturing process, $\dot{Y}_k^{OM}$ refers to the environmental impacts associated with operation and maintenance, and the term $\dot{Y}_k^{DI}$ refers to the environmental impacts associated with waste. The environmental impact balance for a component k is written as follows [72]:

$$\sum_{j=1}^{n} \dot{B}_{j,k,in} + \dot{Y}_k = \sum_{j=1}^{m} \dot{B}_{j,k,out} \tag{3.73}$$

$$\sum_{j=1}^{n} (b_j \dot{E}_j)_{k,in} + \dot{Y}_k = \sum_{j=1}^{n} (b_j \dot{E}_j)_{k,out} \tag{3.74}$$

The average specific environmental effect ($b_{P,k}$) of the product of a k component and the average specific environmental effect ($b_{F,k}$) of the fuel are determined by the following correlations [73]:

$$b_{P,k} = \frac{\dot{B}_{P,k}}{\dot{E}_{P,k}} \tag{3.75}$$

$$b_{F,k} = \frac{\dot{B}_{F,k}}{\dot{E}_{F,k}} \tag{3.76}$$

The relative difference of specific environmental impacts ($r_{b,k}$) value is calculated by the relationship between the average specific environmental impact of the product and the average specific environmental impact of the fuel [72]:

$$r_{b,k} = \frac{b_{P,k} - b_{F,k}}{b_{F,k}} \tag{3.77}$$

The environmental impact sources of any component of the thermal system are determined by the exergoenvironmental factor and compared with other components. Exergoenvironmental factor is expressed by the following relation [72]:

$$f_{b,k} = \frac{\dot{Y}_k}{\dot{Y}_k + \dot{B}_{D,k}} \tag{3.78}$$

where, $\dot{B}_{D,k}$ represents the effect of exergy destruction on the environment and is expressed by the following equation;

$$\dot{B}_{D,k} = b_{F,k}\dot{E}x_{D,k} \tag{3.79}$$

CHAPTER 4

RESULTS

4.1 Energy and Exergy Analyses Results

The combined cycle power plant is comprised as a result of the combining of the gas turbine cycle (Brayton cycle) and the steam turbine cycle (Rankine cycle) with the heat recovery steam generator (HRSG). Since it is comprised of combining two different cycles, the efficiency of the combined cycle power plant is higher than the efficiency of each power plant separately. Natural gas is used as fuel for the Brayton cycle in Ambarlı NG-CCPP. The air, whose pressure and temperature are increased by being compressed in the compressor, is burned with natural gas in the combustion chamber and transfers its energy to the gas turbine blades, enabling the turbine blades to rotate. The mechanical energy obtained by the rotation of the turbine blades generates electrical energy by rotating the shaft to which the generator is connected. The waste exhaust gas from the gas turbine is sent to the HRSG and is used to heat the high-temperature steam to be used in the Rankine cycle. Exhaust gas, which transfers its heat in HRSG, is thrown into the atmosphere from the chimney. The superheated steam entering the high-pressure steam turbine in the Rankine cycle generates mechanical energy by rotating the turbine blades. The remaining energy potential of the steam is evaluated in the low-pressure turbine and converted into mechanical energy by rotating the low-pressure turbine blades. The total mechanical energy obtained from the high-pressure turbine and the low-pressure turbine generates electrical energy by rotating the generator shaft connected to both turbines.

4.1.1 Balance Equation of the Plant Components

Ambarlı NG-CCPP consists of a compressor, combustion chamber and gas turbine components, HRSG, low-pressure steam turbine, condenser, main water pump, heat exchanger, high-pressure pump, low-pressure pump and feed water tank components (Figure 4.5). In this section, mass, energy, entropy and exergy balance equations are obtained for each component.

4.1.1.1 Compressor

Atmospheric air is taken by pipe 1, compressed in the compressor and transmitted to the combustion chamber by pipe 2 (Figure 4.1). The mass, energy, entropy and exergy balance equations for the compressor are written as follows, respectively.

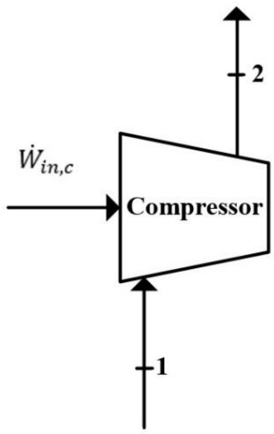

Figure 4. 1 Compressor

$$\dot{m}_1 = \dot{m}_2 \tag{4.1}$$

$$\dot{m}_1 h_1 + \dot{W}_{in,c} = \dot{m}_2 h_2 \tag{4.2}$$

$$\dot{m}_1 s_1 + \dot{S}_{gen,c} = \dot{m}_2 s_2 \tag{4.3}$$

$$\dot{m}_1 ex_1 + \dot{W}_{in,c} = \dot{m}_2 ex_2 + \dot{E}x_{d,c} \tag{4.4}$$

4.1.1.2 Combustion Chamber

After the air compressed in the compressor and transmitted to the combustion chamber with the pipe 2, it burns by mixing with the natural gas coming from the main natural gas network (Figure 4.2). The mass, energy, entropy and exergy balance equations for the combustion chamber are given as follows, respectively.

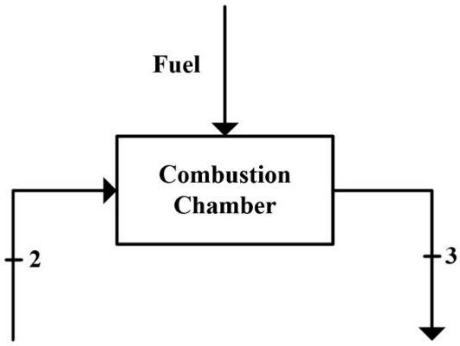

Figure 4. 2 Combustion Chamber

$$\dot{m}_2 + \dot{m}_{fuel} = \dot{m}_3 \tag{4.5}$$

$$\dot{m}_2 h_2 + \dot{Q}_{cc} = \dot{m}_3 h_3 \tag{4.6}$$

In the case where the control volume is accepted as the combustion chamber, when the combustion chamber is evaluated in terms of energetic and exergetic, the energy of the fuel ($\dot{Q}_{cc}$) is found by multiplying the flow rate and the lower heating value (LHV) of the natural gas as in Equation 4.7 [29].

$$\dot{Q}_{cc} = \dot{m}_{fuel} LHV \tag{4.7}$$

The entropy and exergy balance equations for the combustion chamber are as follows, respectively.

$$\dot{m}_2 s_2 + \dot{S}_{gen,cc} + \frac{\dot{Q}_{cc}}{T_{s,i}} = \dot{m}_3 s_3 \tag{4.8}$$

$$\dot{m}_2 ex_2 + \dot{m}_{fuel} ex_{fuel} = \dot{m}_3 ex_3 + \dot{E}x_{d,cc} \tag{4.9}$$

Where, $T_{s,i}$ represents the heat source temperature.

4.1.1.3 Gas Turbine

The gases with high pressure and temperature, which are released as a result of the combustion occurring in the combustion chamber, are transmitted to the gas turbine through the pipe 3. The expanded gases in the gas turbine are sent to HRSG with the pipe 4 located at the turbine outlet (Figure 4.3). The mass, energy, entropy and exergy balance equations for the gas turbine are given as follows, respectively.

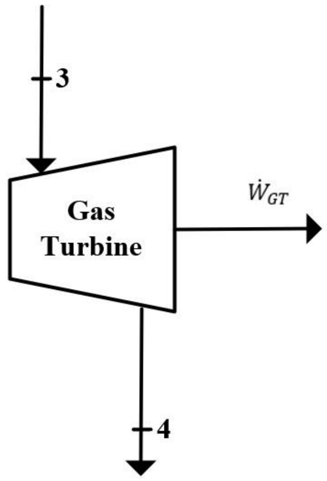

Figure 4. 3 Gas Turbine

$$\dot{m}_3 = \dot{m}_4 \tag{4.10}$$

$$\dot{m}_3 h_3 = \dot{m}_4 h_4 + \dot{W}_{GT} \tag{4.11}$$

$$\dot{m}_3 s_3 + \dot{S}_{gen,GT} = \dot{m}_4 s_4 \tag{4.12}$$

$$\dot{m}_3 ex_3 = \dot{m}_4 ex_4 + \dot{W}_{GT} + \dot{E}x_{d,GT} \tag{4.13}$$

4.1.1.4 Heat Recovery Steam Generator (HRSG)

The exhaust gases from the gas turbine are transmitted to HRSG with the pipe 4, and after giving their heat to the pipe bundles there, they are thrown into the atmosphere with the chimney pipe number 5. After the water passing through the pipe bundles is brought to the desired temperature in HRSG, it is sent to the steam turbine cycle (Figure 4.4). The mass, energy, entropy and exergy balance equations for HRSG are given as follows, respectively.

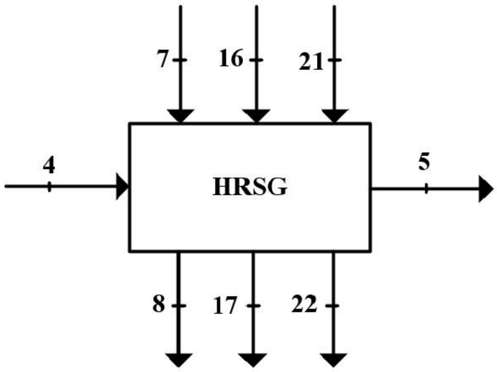

Figure 4. 4 Heat recovery steam generator

$$\dot{m}_4 = \dot{m}_5, \quad \dot{m}_7 = \dot{m}_8, \quad \dot{m}_{16} = \dot{m}_{17}, \quad \dot{m}_{21} = \dot{m}_{22} \tag{4.14}$$

$$\dot{m}_4 h_4 + \dot{m}_7 h_7 + \dot{m}_{16} h_{16} + \dot{m}_{21} h_{21} = \dot{m}_5 h_5 + \dot{m}_8 h_8 + \dot{m}_{17} h_{17} + \dot{m}_{22} h_{22} + \dot{Q}_{HRSG} \tag{4.15}$$

$$\dot{m}_4 s_4 + \dot{m}_7 s_7 + \dot{m}_{16} s_{16} + \dot{m}_{21} s_{21} + \dot{S}_{gen,HRSG} = \dot{m}_5 s_5 + \dot{m}_8 s_8 + \dot{m}_{17} s_{17} + \dot{m}_{22} s_{22} + \frac{\dot{Q}_{HRSG}}{T_{s,i}} \tag{4.16}$$

$$\dot{m}_4 ex_4 + \dot{m}_7 ex_7 + \dot{m}_{16} ex_{16} + \dot{m}_{21} ex_{21} = \dot{m}_5 ex_5 + \dot{m}_8 ex_8 + \dot{m}_{17} ex_{17} + \dot{m}_{22} ex_{22} + \dot{Q}_{HRSG}\left(1 - \frac{T_0}{T_s}\right) + \dot{E}x_{d,HRSG} \tag{4.17}$$

4.1.1.5 Pumps

The energy and exergy relations of the main water pump, low pressure pump and high-pressure pump in the power plant are included in this section as subheadings.

4.1.1.5.1 Main Water Pump

The main water pump in the power plant increases the pressure of the condensed water sent from the condenser with the pipe 19 and transmits it to the heat exchanger with the pipe 20 (Figure 4.6). The mass, energy, entropy and exergy balance equations for the main water pump are given as follows, respectively.

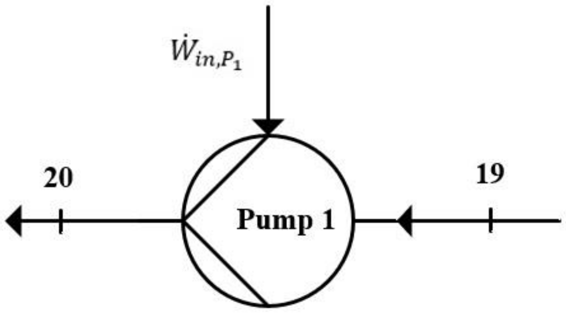

Figure 4. 5 Main water pump

$$\dot{m}_{19} = \dot{m}_{20} \tag{4.18}$$

$$\dot{m}_{19}h_{19} + \dot{W}_{in,P_1} = \dot{m}_{20}h_{20} \tag{4.19}$$

$$\dot{m}_{19}s_{19} + \dot{S}_{gen,P_1} = \dot{m}_{20}s_{20} \tag{4.20}$$

$$\dot{m}_{19}ex_{19} + \dot{W}_{in,P_1} = \dot{m}_{20}ex_{20} + \dot{E}x_{d,P_1} \tag{4.21}$$

4.1.1.5.2 Low-Pressure Pump

The low-pressure pump increases the pressure of the water coming from the feed water tank with the pipe 15 and sends it to the HRSG with the pipe 16 (Figure 4.7). The mass, energy, entropy and exergy balance equations for the low-pressure pump are given as follows, respectively.

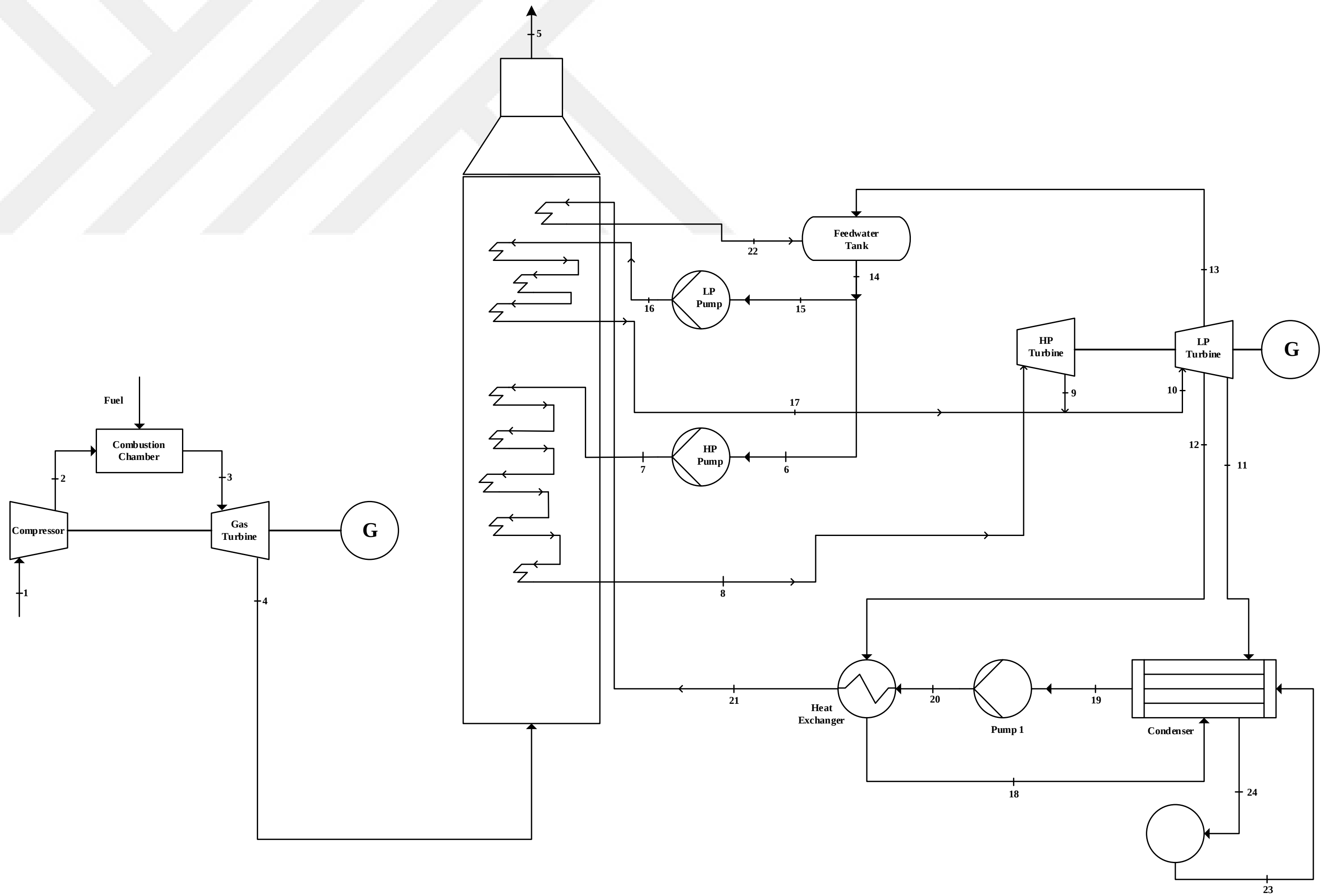

Figure 4. 6 Flow diagram of Ambarlı NG-CCPP

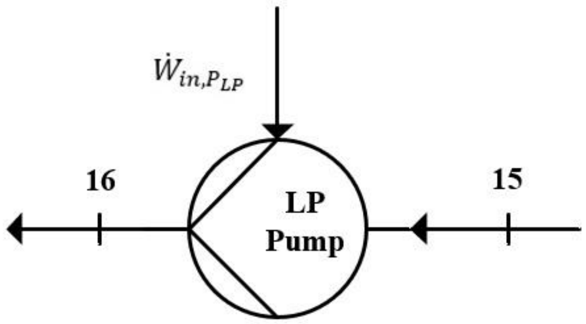

Figure 4. 7 Low-pressure pump

$$\dot{m}_{15} = \dot{m}_{16} \tag{4.22}$$

$$\dot{m}_{15}h_{15} + \dot{W}_{in,P_{LP}} = \dot{m}_{16}h_{16} \tag{4.23}$$

$$\dot{m}_{15}s_{15} + \dot{S}_{gen,P_{LP}} = \dot{m}_{16}s_{16} \tag{4.24}$$

$$\dot{m}_{15}ex_{15} + \dot{W}_{in,P_{LP}} = \dot{m}_{16}ex_{16} + \dot{E}x_{d,P_{LP}} \tag{4.25}$$

4.1.1.5.3 High-Pressure Pump

The high-pressure pump increases the pressure of the water coming from the feed water tank with the pipe 6 and sends it to the HRSG with the pipe 7 (Figure 4.8). The mass, energy, entropy and exergy balance equations for the high-pressure pump are given as follows, respectively.

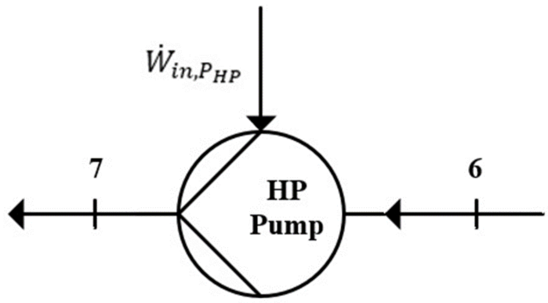

Figure 4. 8 High-pressure pump

$$\dot{m}_6 = \dot{m}_7 \tag{4.26}$$

$$\dot{m}_6 h_6 + \dot{W}_{in,P_{HP}} = \dot{m}_7 h_7 \tag{4.27}$$

$$\dot{m}_6 s_6 + \dot{S}_{gen,P_{HP}} = \dot{m}_7 s_7 \tag{4.28}$$

$$\dot{m}_6 ex_6 + \dot{W}_{in,P_{HP}} = \dot{m}_7 ex_7 + \dot{E}x_{d,P_{LP}} \tag{4.29}$$

4.1.1.6 High-Pressure Steam Turbine

The superheated steam produced in HRSG by utilizing the heat of the gas turbine exhaust gas is firstly transmitted to the high-pressure steam turbine (HPST) located in the steam turbine group through the pipe 8 (Figure 4.9). The produced mechanical energy is transferred to the generator with the aid of the shaft to which both steam turbines are connected. The mass, energy, entropy and exergy balance equations for HPST are given as follows, respectively.

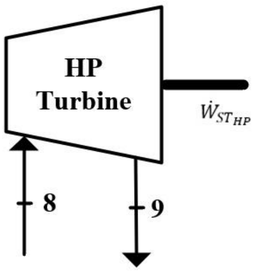

Figure 4. 9 High-pressure steam turbine

$$\dot{m}_8 = \dot{m}_9 \tag{4.30}$$

$$\dot{m}_8 h_8 = \dot{m}_9 h_9 + \dot{W}_{ST_{HP}} \tag{4.31}$$

$$\dot{m}_8 s_8 + \dot{S}_{gen,ST_{HP}} = \dot{m}_9 s_9 \tag{4.32}$$

$$\dot{m}_8 ex_8 = \dot{m}_9 ex_9 + \dot{W}_{ST_{HP}} + \dot{E}x_{d,ST_{HP}} \tag{4.33}$$

4.1.1.7 Low-Pressure Steam Turbine

Although the pressure of the superheated steam which performs mechanical work in the high-pressure turbine drops, there is still work potential. The work potential of the steam, whose pressure decreases and the specific volume increases, is evaluated in the turbine blades located in the low-pressure steam turbine (LPST) and having a larger diameter than the high-pressure steam turbine.

Bleed steam is taken from the low-pressure steam turbine through pipe 13 and sent to the feed water tank, thus purifying the unwanted gases in the steam turbine. The bleed steam is also used for heating the feed water tank [24]. The steam, which is carried from the high-pressure steam turbine to the low-pressure steam turbine with the pipe 10, expands in the turbine blades here and then it is sent to the condenser by the pipe 11 and to the heat exchanger by the pipe 12 (Figure 4.10). The mass, energy, entropy and exergy balance equations for the LPST are given as follows, respectively.

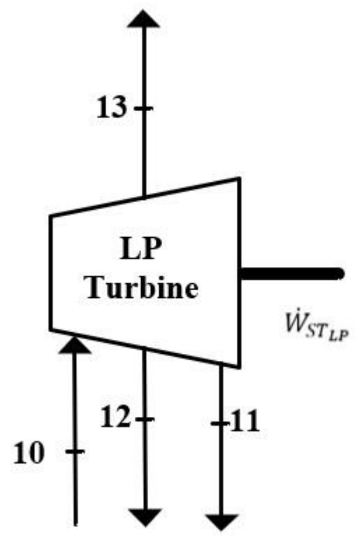

Figure 4. 10 Low-pressure steam turbine

$$\dot{m}_{10} = \dot{m}_{11} + \dot{m}_{12} + \dot{m}_{13} \tag{4.34}$$

$$\dot{m}_{10}h_{10} = \dot{m}_{11}h_{11} + \dot{m}_{12}h_{12} + \dot{m}_{13}h_{13} + \dot{W}_{ST_{LP}} \tag{4.35}$$

$$\dot{m}_{10}s_{10} + \dot{S}_{gen,ST_{LP}} = \dot{m}_{11}s_{11} + \dot{m}_{12}s_{12} + \dot{m}_{13}s_{13} \tag{4.36}$$

$$\dot{m}_{10}ex_{10} = \dot{m}_{11}ex_{11} + \dot{m}_{12}ex_{12} + \dot{m}_{13}ex_{13} + \dot{W}_{ST_{LP}} + \dot{E}x_{d,ST_{LP}} \tag{4.37}$$

4.1.1.8 Condenser

The expanding steam in the low-pressure steam turbine is transmitted to the condenser through the pipe 11 and condenses there. Then, it is mixed with the water added to the system from the drainage tank and transmitted to the main water pump via the pipe 19. The heat removal process in the condenser is carried out with sea water. The sea water, which comes to the condenser with the pipe 23, is sent back to the sea with the pipe 24 by increasing its temperature here (Figure 4.11). The mass, energy, entropy and exergy balance equations for the condenser are given as follows, respectively.

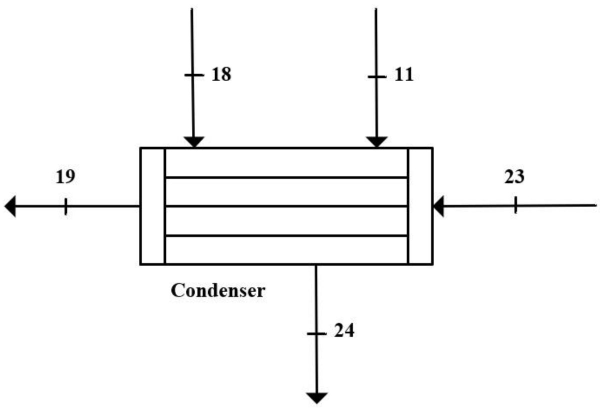

Figure 4. 11 Condenser

$$\begin{aligned} \dot{m}_{11} &= \dot{m}_{18} + \dot{m}_{19}, \\ \dot{m}_{23} &= \dot{m}_{24} \end{aligned} \tag{4.35}$$

$$\dot{m}_{11}h_{11} + \dot{m}_{18}h_{18} + \dot{m}_{23}h_{23} = \dot{m}_{19}h_{19} + \dot{m}_{24}h_{24} + \dot{Q}_{con} \tag{4.36}$$

$$\begin{aligned} &\dot{m}_{11}s_{11} + \dot{m}_{18}s_{18} + \dot{m}_{23}s_{23} + \dot{S}_{gen,con} \\ &\qquad = \dot{m}_{19}s_{19} + \dot{m}_{24}s_{24} + \frac{\dot{Q}_{con}}{T_b} \end{aligned} \tag{4.37}$$

$$\dot{m}_{11}ex_{11} + \dot{m}_{18}ex_{18} + \dot{m}_{23}ex_{23}$$
$$= \dot{m}_{19}ex_{19} + \dot{m}_{24}ex_{24} + \dot{Q}_{con}\left(1 - \frac{T_0}{T_b}\right) \tag{4.38}$$
$$+ \dot{E}x_{d,con}$$

Where, T_b is the immediate boundary temperature for the condenser

4.1.1.9 Feedwater Tank

Bleed steam is taken from the low-pressure steam turbine with pipe 13 to the feed water tank, which has two inlets and one outlet. This steam is used to heat the feedwater tank. Pipe 14 is divided into two different branches and transmits the feedwater to HRSG with a low-pressure pump and a high-pressure pump. Pipe 22, which is connected to the feedwater tank, heats the condensed water coming from the heat exchanger in the HRSG and transmits it to the feedwater tank (Figure 4.12). The mass, energy, entropy and exergy balance equations for the feedwater tank are given as follows, respectively.

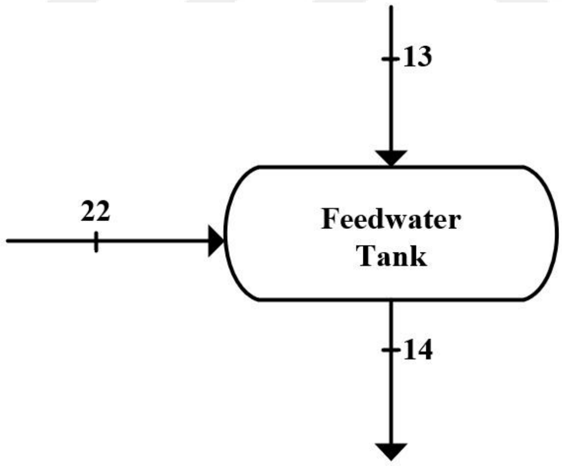

Figure 4. 12 Feedwater tank

$$\dot{m}_{13} + \dot{m}_{22} = \dot{m}_{14} \tag{4.39}$$

$$\dot{m}_{13}h_{13} + \dot{m}_{22}h_{22} = \dot{m}_{14}h_{14} + \dot{Q}_{FW} \tag{4.40}$$

$$\dot{m}_{13}s_{13} + \dot{m}_{22}s_{22} + \dot{S}_{gen,FW} = \dot{m}_{14}s_{14} \tag{4.41}$$

$$\dot{m}_{13}ex_{13} + \dot{m}_{22}ex_{22} = \dot{m}_{14}ex_{14} + \dot{Q}_{FW}\left(1 - \frac{T_0}{T_s}\right) + \dot{E}x_{d,FW} \quad (4.42)$$

4.1.1.10 Heat Exchanger

Gland steam separated from the high-pressure steam turbine and the low-pressure steam turbine is transmitted to the heat exchanger by pipe 12 and condensed there. The condensed water line, to which the condenser and the main water pump are connected, transmits the condensed water to the heat exchanger with the pipe 20 and takes the heat from the pipe 12 here. Pipe 18, located at the outlet of the heat exchanger, transmits the condensed water to the condenser, and pipe 21 sends it to the feed water tank (Figure 4.13). The mass, energy, entropy and exergy balance equations for the heat exchanger are given as follows, respectively.

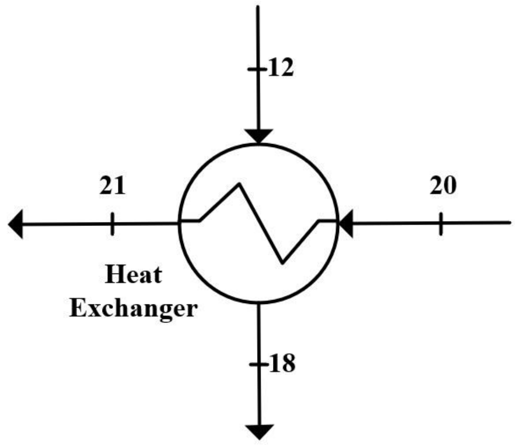

Figure 4. 13 Heat exchanger

$$\dot{m}_{11} = \dot{m}_{18}, \quad \dot{m}_{20} = \dot{m}_{21} \quad (4.43)$$

$$\dot{m}_{12}h_{12} + \dot{m}_{20}h_{20} = \dot{m}_{18}h_{18} + \dot{m}_{21}h_{21} + \dot{Q}_{Hex} \quad (4.44)$$

$$\dot{m}_{12}s_{12} + \dot{m}_{20}s_{20} + \dot{S}_{gen,Hex} = \dot{m}_{18}s_{18} + \dot{m}_{21}s_{21} + \frac{\dot{Q}_{con}}{T_{s,i}} \quad (4.45)$$

$$\dot{m}_{12}ex_{12} + \dot{m}_{20}ex_{20} = \dot{m}_{18}ex_{18} + \dot{m}_{21}ex_{21} + \dot{Q}_{Hex}\left(1 - \frac{T_0}{T_s}\right) + \dot{E}x_{d,Hex} \tag{4.46}$$

4.1.2 Energy and Exergy Analyses Calculation Results

In this section, the energy and exergy analysis of the power plant were carried out by using the actual flow, temperature and pressure data read from the inlet and outlet pipes of each component of the Ambarlı NG-CCPP. While performing these analyses the Engineering Equation Solver (EES) software program was used. These data, taken from the control room of the power plant, were obtained from the instantaneous measurement of the sensors in the pipes on September 12, 2019, at 11:30, under the condition of 25.4°C air temperature (Table 4.1).

The values that cannot be read from the control room of the power plant are completed with the data taken from the literature (Table 4.2) [24].

Table 4. 1 The thermodynamic values of the fluids passing through the pipes

Pipe No	$\dot{m}$ **[kg/s]**	***T*** **[°C]**	***P*** **[kPa]**
1	-	25.40	98.1
2	-	350.00	980
3	-	1050.00	980
4	-	565.00	103.3
5	-	106.10	98.1
6	-	101.20	150
7	-	102.00	7548
8	-	467.40	6008
9	-	210.90	507.5
10	-	190.60	508.4
11	-	37.53	7.5
12	-	97.24	110
13	-	122.10	250
14	-	101.20	150
15	-	101.20	150
16	-	104.50	2500
17	13.93	190.20	504
18	-	32.00	98.4
19	-	37.11	88
20	-	37.20	720
21	-	41.70	700
22	-	97.82	150
23	73.69	24.41	127
24	-	29.22	127

Table 4. 2 Completed thermodynamic values of fluids passing through pipes

Pipe No	**$\dot{m}$ [kg/s]**	**T [°C]**	**P [kPa]**
1	500.00	25.40	98.1
2	500.00	350.00	980
3	511.40	1050.00	980
4	511.40	565.00	103.3
5	511.40	106.10	98.1
6	60.25	101.20	150
7	60.25	102.00	7548
8	60.25	467.40	6008
9	60.25	210.90	507.5
10	74.18	190.60	508.4
11	73.55	37.53	7.5
12	0.14	97.24	110
13	0.49	122.10	250
14	74.18	101.20	150
15	13.93	101.20	150
16	13.93	104.50	2500
17	13.93	190.20	504
18	0.14	32.00	98.4
19	73.69	37.11	88
20	73.69	37.20	720
21	73.69	41.70	700
22	73.69	97.82	150
23	11006	24.41	127
24	11006	29.22	127

The data read by the sensors of each point in the flow diagram of the plant created in Figure 4.1 has been completed as in Table 4.2. In the light of these data, energy and exergy analyses of the power plant were carried out by using the EES. The thermodynamic results obtained for each point in the power plant are given in Table 4.3.

Table 4. 3 Thermodynamic results of each point in Ambarlı NG-CCPP

Points	$\dot{m}$ [kg/s]	T [°C]	P [kPa]	h [kj/kg]	s [kj/kgK]	$\dot{E}$ [kW]	$\dot{E}x$ [kW]
1	500	25,4	98.1	299	5.706	149500	0
2	500	350	980	598.4	5.742	299200	144645
3	511.4	1050	980	1433	6.642	732836.2	473207
4	511.4	565	103.3	878.2	6.779	449111.5	132316
5	511.4	106.1	98.1	380.3	5.947	194485.4	4772
6	60.25	101.2	150	424.2	1.32	25558.05	86430
7	60.25	102	7548	432.8	1.323	26076.2	86906
8	60.25	467.4	6008	3344	6.776	201476	164210
9	60.25	210.9	507.5	2778	6.883	167374.5	128195
10	74.18	190.6	508.4	2834	7.007	210226.1	159.256
11	73.55	37.53	7.5	2316	7.428	170341.8	110571
12	0.14	97.24	110	2615	7.156	366.1	263.7
13	0.49	122.1	250	2726	6.799	1335.74	1029
14	74.18	101.2	150	424.2	1.32	31467.16	106413
15	13.93	101.2	150	424.2	1.32	5909.106	19983
16	13.93	104.5	2500	427	1.321	5948.11	20018
17	13.93	190.2	504	2834	7.01	39477.62	29887
18	0.14	32	98.4	134.1	0.464	18.774	196
19	73.69	37.11	88	155.5	0.5335	11458.8	103223
20	73.69	37.2	720	156.2	0.5338	11510.38	103270
21	73.69	41.7	700	175.2	0.5946	12910.49	103332
22	73.69	97.82	150	409.9	1.282	30205.53	105501
23	11006	24.41	127	102.4	0.3586	1127014	1541000
24	11006	29.22	127	122.5	0.4257	1348235	1541000

In Table 4.4, exergy destruction values ($\dot{E}x_d$) and entropy production ($\dot{S}_{gen}$) values were determined for each component.

Table 4. 4 Exergy destruction and entropy generation of the system components

System Component	$\dot{\boldsymbol{E}}\boldsymbol{x}_{\boldsymbol{d}}$ **[kW]**	$\dot{\boldsymbol{S}}_{\boldsymbol{gen}}$ **[kW/K]**
Compressor	5379	18.02
Combustion Chamber	89161	352.5
Gas Turbine	20978	70.27
HRSG	13188	37.28
Pump LP	4.159	0.01393
Pump HP	43.41	0.1454
Pump 1	7.96	0.0266
HP Steam Turbine	1921	6.436
LP Steam Turbine	9217	30.87
Condenser	7866	26.35
Heat Exchanger	1055	3.124
Feedwater Tank	49.19	0.1337

The gas turbine, steam turbine powers and the net power value ($\dot{W}_{net,gen}$) obtained from both cycles were calculated (Table 4.5).

Table 4. 5 Net power generations of the system components

System Component	$\dot{W}_{net,gen}$ **[kW]**
Gas Turbine	283.913
HP Steam Turbine	34.093
LP Steam Turbine	38.175
Steam Turbines	72.268
Gas Turbine Cycle	134.188
Steam Turbine Cycle	71.654

The exergetic efficiency of the components (η_{II}) and the exergy efficiency of the combined cycle power plant were calculated by using the thermodynamic results and the balance equations created for each component (Table 4.6).

Table 4. 6 Exergy efficiencies of the CCPP and components

System Component	η_{II} **[%]**
Compressor	96.41
Combustion Chamber	75.62
Gas Turbine	49.85
HRSG	12.34
Pump LP	89.44
Pump HP	91.64
HP Steam Turbine	94.66
LP Steam Turbine	53.99
Condenser	93.14
Feedwater Tank	40.61
Combine Cycle Power Plant	61.22

4.2 THERMOECONOMIC RESULTS

Energy and exergy analyses of EÜAŞ NG-CCPP were carried out in section 4.1. The energy and exergy values and other important thermodynamic data of the fluids passing through the pipes at the inlet and outlet of each component were obtained. In this section, exergy costing will be performed by using the exergy values obtained for each point. The balance equations and auxiliary balances required for the exergy costing will be developed separately for each component and then the cost values of the system will be calculated.

Total capital investment (TCI), operation and maintenance cost (OM) and salvage value (SV) were calculated by using the purchasing costs (PEC) of each component [35, 45]. The present value (PV) was calculated by using Equation 4.47 [73].

$$PV = TCI + OM - SV * PVF \tag{4.47}$$

Before proceeding to the exergoeconomic analysis of the power plant, it is necessary to calculate the important values to be used in the economic analysis. The compound interest rate (i) is accepted as 10%, the life span of the power plant (n) is 20 years, and the fixed value (j) used in the calculation of the salvage value of each component is accepted as 0.12 [75, 78]. It is assumed that the combined cycle power plant continues to operate for 8600 hours per year under full load. The present value factor (PVF) for the power plant is calculated as 0.1174 and the capital recovery factor (CRF) is calculated as 0.1486. Table 4.7 contains the PEC and PV values of each component.

Table 4. 7 PEC and PV values of each component.

COMPONENT	PEC ($)	PV ($)
Compressor	16728701	108417475
Combustion Chamber	22373305	144999731
Gas Turbine	39389543	255280712
HRSG	3171366	20553385
Pump 1	61024	395492
HP Pump	299841	1943247
LP Pump	48054	311433
Feedwater Tank	2961497	19193240
Heat Exchanger	461401	2990308
Condenser	130652	846749
HP Steam Turbine	8933403	57896720
LP Steam Turbine	9669334	62666240

The unit cost of natural gas was calculated as 1.571 TL/m^3 by dividing the cost of the total consumed fuel and the total amount of fuel consumed annually. Both values were obtained from the power plant record. The exchange rate used in the calculations, according to the data of the Central Bank of the Republic of Turkey (TCMB), has been accepted as 5.77 TL/$, taking into account the measurement date at the power plant [78]. Using all these values, the exergetic unit cost (c_{NG}) of the fuel was calculated as 7.183 $/GJ.

The electricity cost value that is used in the exergoeconomic analyses of the compressor, gas turbine, low-pressure and high-pressure steam turbines and a total of three pumps in the power plant should be calculated. The electricity price has been determined as 0.448 TL/kWh by the Republic of Turkey Energy Market Regulatory Authority (EPDK) [74, 79]. The exergetic unit cost of electricity (c_w) is calculated as $21.58/GJ, together with the exchange rate obtained from the TCMB data.

The total cost ($\dot{Z}_k$) of each component was calculated by using the calculated and accepted values above, together with the operation and maintenance cost ($\dot{Z}_k^{OM}$) and capital investment cost ($\dot{Z}_k^{CI}$) values for each component. Table 4.8 shows the total cost of each component of the power plant.

Table 4. 8 Total cost of each component ($\dot{Z}_k$)

COMPONENT	$\dot{Z}_k$ ($/h)
Compressor	1873.90
Combustion Chamber	2506.19
Gas Turbine	4412.31
HRSG	355.25
Pump 1	6.84
HP Pump	33.59
LP Pump	5.38
Feedwater Tank	331.74
Heat Exchanger	51.68
Condenser	14.64
HP Steam Turbine	1000.69
LP Steam Turbine	1083.13

The assumptions in exergoeconomic analysis are listed as follows;

- Atmospheric air sucked from the inlet of the compressor, which is one of the components of the Brayton cycle, does not require additional cost. For this reason, it is accepted as $c_1 = 0$.
- The sea water used for the condenser does not require additional costs. For this reason, it is accepted as $c_{23} = c_{24} = 0$.
- Since the water leaving the heat exchanger and coming to the condenser does not require additional costs, it is accepted as $c_{18} = 0$ [35].
- The additional cost for the water at the inlet and outlet of low pressure and high-pressure pumps is considered as $c_{16} = c_{17} = 0$ and $c_6 = c_7 = 0$ [35]. For this reason, LP Pump and HP Pump are not considered in the exergoeconomic analysis.

4.2.1 Exergoeconomic Balance Equations of Components

- **Compressor**

The main cost balance equation, expanded cost balance equation and auxiliary balances for the compressor in Figure 4.1 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_1 + \dot{C}_{w,c} + \dot{Z}_c = \dot{C}_2 \tag{4.48}$$

Expanded cost balance equation:

$$c_1\dot{E}x_1 + c_w\dot{W}_c + \dot{Z}_c = c_2\dot{E}x_2 \tag{4.49}$$

Auxiliary balances:

$$c_1 = 0 \tag{4.50}$$

- **Combustion Chamber**

The main cost balance equation and expanded cost balance equation for the combustion chamber in Figure 4.2 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_2 + \dot{C}_{NG} + \dot{Z}_{cc} = \dot{C}_3 \tag{4.51}$$

Expanded cost balance equation:

$$c_2\dot{E}x_2 + c_{NG}\dot{Q}_{cc} + \dot{Z}_{cc} = c_3\dot{E}x_3 \tag{4.52}$$

- **Gas Turbine**

The main cost balance equation and expanded cost balance equation for the gas turbine in Figure 4.3 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_3 + \dot{C}_{W,GT} + \dot{Z}_{GT} = \dot{C}_4 \tag{4.53}$$

Expanded cost balance equation:

$$c_3\dot{E}x_3 + c_w\dot{W}_{GT} + \dot{Z}_{GT} = c_4\dot{E}x_4 \tag{4.54}$$

- **HRSG**

The main cost balance equation, expanded cost balance equation and auxiliary balances for the HRSG in Figure 4.4 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_4 + \dot{C}_7 + \dot{C}_{16} + \dot{C}_{21} + \dot{Z}_{HRSG} = \dot{C}_5 + \dot{C}_8 + \dot{C}_{17} + \dot{C}_{22} \tag{4.55}$$

Expanded cost balance equation:

$$\begin{aligned} c_4\dot{E}x_4 + c_7\dot{E}x_7 + c_{16}\dot{E}x_{16} + c_{21}\dot{E}x_{21} + \dot{Z}_{HRSG} \\ = c_5\dot{E}x_5 + c_8\dot{E}x_8 + c_{17}\dot{E}x_{17} + c_{22}\dot{E}x_{22} \end{aligned} \tag{4.56}$$

Auxiliary balances:

$$\begin{aligned} c_4 &= c_5 \\ c_6 &= 0 \\ c_7 &= 0 \\ c_{16} &= 0 \\ c_{17} &= 0 \end{aligned} \tag{4.57}$$

- **Main Water Pump (Pump 1)**

The main cost balance equation and expanded cost balance equation for the main water pump in Figure 4.6 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_{19} + \dot{C}_{w,P_1} + \dot{Z}_{P_1} = \dot{C}_{20} \tag{4.58}$$

Expanded cost balance equation:

$$c_{19}\dot{E}x_{19} + c_w\dot{W}_{P_1} + \dot{Z}_{P_1} = c_{20}\dot{E}x_{20} \tag{4.59}$$

- **High Pressure Steam Turbine**

The main cost balance equation, expanded cost balance equation and auxiliary balances for the high-pressure steam turbine in Figure 4.9 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_8 + \dot{Z}_{HPST} = \dot{C}_9 + \dot{C}_{w,HPST} \tag{4.60}$$

Expanded cost balance equation:

$$c_8\dot{E}x_8 + \dot{Z}_{HPST} = c_2\dot{E}x_2 + c_w\dot{W}_{HPST} \tag{4.61}$$

Auxiliary balances:

$$c_8 = c_9 \tag{4.62}$$

- **Low Pressure Steam Turbine**

The main cost balance equation, expanded cost balance equation and auxiliary balances for the low-pressure steam turbine in Figure 4.10 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_{10} + \dot{Z}_{LPST} = \dot{C}_{11} + \dot{C}_{12} + \dot{C}_{13} + \dot{C}_{w,LPST} \tag{4.63}$$

Expanded cost balance equation:

$$c_{10}\dot{E}x_{10} + \dot{Z}_{LPST} = c_{11}\dot{E}x_{11} + c_{12}\dot{E}x_{12} + c_{13}\dot{E}x_{13} + c_w\dot{W}_{LPST} \tag{4.64}$$

Auxiliary balances:

$$c_{10} = c_{11} = c_{12} = c_{13} \tag{4.65}$$

- **Condenser**

The main cost balance equation, expanded cost balance equation and auxiliary balances for the condenser in Figure 4.11 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_{11} + \dot{C}_{18} + \dot{C}_{23} + \dot{Z}_{con} = \dot{C}_{19} + \dot{C}_{24} \tag{4.66}$$

Expanded cost balance equation:

$$c_{11}\dot{E}x_{11} + c_{18}\dot{E}x_{18} + c_{18}\dot{E}x_{18} + \dot{Z}_{con} = c_{19}\dot{E}x_{19} + c_{24}\dot{E}x_{24} \tag{4.67}$$

Auxiliary balances:

$$c_{23} = c_{24}$$
$$c_{18} = 0 \tag{4.68}$$
$$c_{23} = 0$$

- **Feedwater Tank**

The main cost balance equation and expanded cost balance equation for the feedwater tank in Figure 4.12 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_{13} + \dot{C}_{22} + \dot{Z}_{FW} = \dot{C}_{14} \tag{4.58}$$

Expanded cost balance equation:

$$c_{13}\dot{E}x_{13} + c_{22}\dot{E}x_{22} + \dot{Z}_{FW} = c_{14}\dot{E}x_{14} \tag{4.59}$$

- **Heat Exchanger**

The main cost balance equation, expanded cost balance equation and auxiliary balances for the heat exchanger in Figure 4.13 are given as follows, respectively.

Main cost balance equation:

$$\dot{C}_{12} + \dot{C}_{20} + \dot{Z}_{Hex} = \dot{C}_{21} \tag{4.63}$$

Expanded cost balance equation:

$$c_{12}\dot{E}x_{12} + c_{20}\dot{E}x_{20} + \dot{Z}_{Hex} = c_{21}\dot{E}x_{21} \tag{4.64}$$

Auxiliary balances:

$$c_{18} = 0 \tag{4.65}$$

4.2.2 Exergoeconomic Analysis Calculation Results

The main cost balance equation, expanded cost balance equation and auxiliary balances obtained for the input and output currents of each component are solved by using the EES software program. Exergy values $\dot{E}x$ (GJ/h), unit exergy cost values c (\$/GJ) and exergy cost values $\dot{C}$ (\$/h) were calculated for each point. These values are given in Table 4.9.

Table 4. 9 Exergy, specific exergy cost and exergetic cost values of each pipe

Pipe No	$\dot{Ex}$ [GJ/h]	c [$/GJ]	$\dot{C}$ [$/h]
1	0	0	0
2	519.6	25.99	13506
3	1574	17.3	27225
4	476.3	20.11	9581
5	17.18	20.11	345,5
6	311.15	0	0
7	312.86	0	0
8	591.2	12.71	7514
9	461.5	12.71	5866
10	573.3	11.03	6326
11	398.1	11.03	4392
12	0.949	11.03	10,47
13	3.704	11.03	40,87
14	383.1	11.87	4546
15	71.94	0	0
16	72.07	0	0
17	107.6	19.31	2077
18	0.706	0	0
19	371.6	11.86	4407
20	371.8	11.88	4418
21	372	12.04	4480
22	379.8	10.99	4174
23	55476	0	0
24	55476	0	0

The relative cost difference ($\boldsymbol{r_k}$), which gives the relationship between the average product and the average fuel cost, and the exergoeconomic factor ($\boldsymbol{f_k}$) values were calculated for each component. Relative cost difference and exergoeconomic factor values for each component are given in Table 4.10.

Table 4. 10 Relative cost difference and exergoeconomic factor of each component

COMPONENT	r [%]	f [%]
Compressor	20.44	98.98
Combustion Chamber	46.65	88.65
Gas Turbine	34.24	98.32
HRSG	9285	88.21
Pump 1	205.2	98.58
Feedwater Tank	7.99	99.95
Heat Exchanger	3478	93.15
Condenser	7.48	34.07
HP Steam Turbine	69.79	99.31
LP Steam Turbine	95.57	97.03

4.3 LIFE CYCLE ASSESSMENT RESULTS

Energy, exergy and exergoeconomic analyses of the power plant were carried out in sections 4.1 and 4.2, respectively. The exergy values were calculated for each pipe in the power plant. The exergy cost was also calculated for each point by using the determined exergy values together with the unit exergy cost in the exergoeconomic analysis. In this section, environmental impact ratios will be calculated by using the exergy values obtained for each point. The balance equations and auxiliary balances required within the scope of environmental impact values will be developed separately for each component and the environmental impact ratios of the system will be calculated.

Before proceeding to the exergoenvironmental analysis of the plant, a life cycle assessment should be performed. So, it is necessary to determine the environmental impact scores of the components. The lifespan of the power plant to be used in the calculations has been accepted as 20 years, and the annual operating hours of the power plant under full load have been assumed as 8600 hours.

The type of raw material of each component of the power plant and the proportion of these materials are required to calculate the total Eco-indicator 99 scores of the components. The Eco-indicator 99 score of each component resulting from the material is calculated by using determined the Eco-indicator 99 score of each material. The data about the raw material type and the ratios of each component constituting the EÜAŞ NG-CCPP are given in Table 4.11 [79, 80].

Table 4. 11 The ratio of raw materials [79, 80] and Eco-indicator 99 point of each component [81]

COMPONENT	Raw material ratio [%]	Eco-indicator 99 point [mPts/kg]
Compressor	Steel: 33 Steel low alloy: 45 Cast iron: 22	Steel: 86 Steel low alloy: 110 Cast iron: 240
Combustion Chamber	Steel: 33 Steel high alloy: 77	Steel: 86 Steel high alloy: 910
Gas Turbine HP Steam Turbine LP Steam Turbine	Steel: 25 Steel high alloy: 75	Steel: 86 Steel high alloy: 910
HRSG Feedwater Tank	Steel: 100	Steel: 86
Heat Exchanger Condenser	Steel: 26 Steel high alloy: 74	Steel: 86 Steel high alloy: 910
Main Water Pump HP Pump LP Pump	Steel: 35 Cast Iron: 65	Steel: 86 Cast iron: 240

The total scores of the components within the scope of the life cycle assessment are obtained by summing the environmental impacts of the components resulting from their operation and maintenance, their manufacturing and their disposal as salvage by using the equation in Equation 3.103.

The total material Eco-indicator 99 scores and the weights of the components are given in Table 4.12. The data required to calculate the weights of the compressor, combustion chamber and gas turbine components that constitute the Brayton cycle is not sufficient. For this reason, information about the weights of these components was taken from the literature [81].

Table 4. 12 Total material points and weight of each component

COMPONENT	**Total Material Point [mPts/kg]**	**Weight [kg]**
Compressor	130.68	80000
Combustion Chamber	729.08	108200
Gas Turbine	704	181400
HRSG	86	804422
Pump 1	186.1	275
HP Pump	186.1	342
LP Pump	186.1	417
Feedwater Tank	86	50745
Heat Exchanger	704	279142
Condenser	704	4379
HP Steam Turbine	704	64423
LP Steam Turbine	704	69967

The environmental impact scores resulting from the operation and maintenance of the components were accepted as 25% of the environmental impact scores from the manufacturing, and the environmental impact scores resulting from the disposal as salvage were accepted as 12% of the environmental impact scores from the manufacturing [73]. The environmental impact scores resulting from the production of the components, the environmental impact scores from the operation and maintenance, the environmental impact scores from the disposal of the components as salvage and the total environmental impact scores are calculated and shown in Table 4.13. Also, some life cycle assessment results from literature and this study's results are given in Table 4.14.

Table 4. 13 Total Eco-indicator 99 scores and Eco-indicator 99 scores resulting from operation and maintenance, manufacturing and disposal as a salvage of each component.

COMPONENT	$\dot{Y}_k^{CO}$ **[mPts/h]**	$\dot{Y}_k^{OM}$ **[mPts/h]**	$\dot{Y}_k^{DI}$ **[mPts/h]**	$\dot{Y}_k$ **[mPts/h]**
Compressor	60.78	15.19	7.29	83.27
Combustion Chamber	458.64	114.66	55.03	628.34
Gas Turbine	742.47	185.61	89.09	1017.19
HRSG	402.21	100.55	48.26	551.03
Pump 1	0.3	0.07	0.04	0.41
HP Pump	0.37	0.09	0.04	0.51
LP Pump	0.45	0.11	0.05	0.61
Feedwater Tank	25.37	6.34	3.04	34.76
Heat Exchanger	1142.53	285.63	137.10	1565.27
Condenser	17.92	4.48	2.15	24.55
HP Steam Turbine	263.68	65.92	31.64	361.25
LP Steam Turbine	286.37	71.59	34.36	392.34
Total	3401.11	850.28	408.13	4659.53

Table 4. 14 Some different LCA results from literature and this study's LCA results.

<table>
<tr><th rowspan="2">COMPONENT</th><th colspan="6">Total Eco-indicator 99 scores $\dot{Y}_k$ [mPts/h]</th></tr>
<tr><th>Ref. 1 [80]</th><th>Ref. 2 [82]</th><th>Ref. 3 [83]</th><th>Ref. 4 [84]</th><th>Ref. 5 [85]</th><th>This Study</th></tr>
<tr><td>Compressor</td><td>65.1</td><td>236</td><td>378</td><td>-</td><td>7.2</td><td>83.27</td></tr>
<tr><td>Combustion Chamber</td><td>337.6</td><td>381</td><td>219.6</td><td>-</td><td>-</td><td>628.34</td></tr>
<tr><td>Gas Turbine</td><td>624.7</td><td>1126</td><td>219.6</td><td>-</td><td>50.4</td><td>1017.2</td></tr>
<tr><td>HRSG</td><td>1934.8</td><td>2708</td><td>1519.2</td><td>-</td><td></td><td>551</td></tr>
<tr><td>HPST</td><td rowspan="2">727.9</td><td>276</td><td rowspan="2">576</td><td>1058.4</td><td rowspan="2">2.7</td><td>361.3</td></tr>
<tr><td>LPST</td><td>493</td><td>1224</td><td>392.3</td></tr>
<tr><td>Pump</td><td>0.7</td><td>-</td><td>-</td><td>-</td><td>3.6</td><td>1.56</td></tr>
<tr><td>Condenser</td><td>3.7</td><td>-</td><td>190.8</td><td>-</td><td>-</td><td>24.6</td></tr>
<tr><td>Deaerator</td><td>12.1</td><td>-</td><td>-</td><td>32.4</td><td>-</td><td>37.76</td></tr>
<tr><td>Boiler</td><td>-</td><td>-</td><td>-</td><td>36194.4</td><td>-</td><td>-</td></tr>
<tr><td>Heat Exchanger</td><td>-</td><td>-</td><td>-</td><td>-</td><td>57.6</td><td>1565.3</td></tr>
<tr><td>Gasifier</td><td>-</td><td>-</td><td>-</td><td>-</td><td>147.6</td><td>-</td></tr>
<tr><td>Heater</td><td>-</td><td>-</td><td>-</td><td>28.8</td><td>-</td><td>-</td></tr>
</table>

CHAPTER 5

DISCUSSION AND CONCLUSION

In this study, energetic, exergetic and thermoeconomic analyses and life cycle assessment of EÜAŞ Ambarlı Natural Gas Fired Combined Cycle Power Plant were performed. Exergoeconomic analysis was taken into account within the thermoeconomic analysis of the power plant, and exergoenvironmental analysis was taken into account within the life cycle assessment. In this chapter, the obtained results are evaluated.

The mass, energy, entropy and exergy balance equations are developed separately for each component of the power plant, and the energy and exergy values of the streams in the pipes at the inlet and outlet of the components and other important thermodynamic values were calculated. While performing the exergy analysis, potential and kinetic exergies were neglected, and physical and chemical exergies were considered. The exergy destruction and the exergy efficiency of each component were determined with these obtained values.

After the energy and exergy analysis, the power plant was examined in terms of exergoeconomics. In the exergoeconomic analysis, firstly, the economic analysis of the power plant was performed. The present value of the power plant was determined by calculating the purchase equipment cost, total capital investment, operation and maintenance costs and salvage value of all components. The total costs of all components were calculated by using these values. The capital recovery factor value was obtained by assuming specific values of the compound interest rate, the annual operating hour of the plant and the total lifespan of the plant. After the economic analysis, exergy values that were obtained for each point in the power plant in the energy and exergy analysis were assigned to the values obtained as a result of the economic analysis, and then exergoeconomic analysis was performed. As a result of exergoeconomic analysis, unit exergy cost values were calculated for each point. By using these calculated values and the exergy values, exergy cost values were obtained for each point. Finally, the relative cost difference and exergoeconomic factor values of the components were calculated as a percentage.

In the exergoenvironmental analysis, similar to the exergoeconomic analysis, the LCA results of the components in the plant were calculated firstly, and these values were assigned to the exergy streams, and then the exergoenvironmental analysis of the plant was performed. Within the scope of LCA, firstly, the weights of each component were calculated. Then, the Eco-indicator 99 points, which are from the material, of the components were calculated by using the Eco-indicator 99 points of each material. Then, the environmental impacts of the components in operation and maintenance, manufacturing and disposal as scrap were calculated as Eco-indicator 99 points. With all these calculated values for LCA, the hourly environmental impact of each component was calculated. After the life cycle assessment, the environmental impact values were calculated for each point by using the exergy values and specific environmental impact values calculated for each point in the power plant. Finally, the environmental effects of the components were obtained and the exergoenvironmental analysis was completed.

As a result of the energy and exergy analysis of the power plant, the component with the highest exergy destruction was determined as the combustion chamber with a value of 89161 kW. The values of exergy destruction for other components, from high to low, were as follows: 20978 kW for gas turbine, 13188 kW for HRSG, 9217 kW for low-pressure steam turbine, 7866 kW for the condenser, 5379 kW for compressor, 1921 kW for high-pressure steam turbine, 1055 kW for heat exchanger, 49.19 kW for feed water tank, 43.41 kW for the high-pressure pump, 7.96 kW for the main water pump and 4.16 kW for low-pressure pump exergy destruction values were found. The exergy efficiencies of the components were calculated from the highest efficiency to the lowest efficiency as follows: 96.41% for the compressor, 94.66% for high-pressure steam turbine, 93.14% for condenser, 91.64% for the high-pressure pump, 89.44% for low-pressure pump, 75.62% for combustion chamber, 53.99% for low-pressure steam turbine, 49,85% for gas turbine, 40.61% for feedwater tank and 12.34% for HRSG.

According to the results of the exergoeconomic analysis of the plant, the highest unit exergy cost value was calculated as 25.99$/GJ in pipe number 2 located at the outlet of the compressor. Except for atmospheric air and water, which unit exergy cost of it is zero, the pipe with the lowest unit exergy cost value is calculated as pipe number 22 coming from the HRSG and going to the feedwater tank at a value of 10.99 $/GJ.

Considering the exergy cost values calculated by using the exergy values in the pipes together with the unit exergy values, the highest exergy cost value was calculated at the value of 27225 $/h in pipe number 3 located at the combustion chamber exit. The pipe with the lowest exergy cost, on the other hand, was calculated at a value of 10.47 $/h in pipe number 12 connected to the heat exchanger from the low-pressure steam turbine outlet. When the relative cost difference between the average unit product cost and the average unit fuel cost is examined, the highest difference occurred in HRSG with 9285%, and the lowest difference occurred in the condenser at a rate of 7.48%. Considering the exergoeconomic factor values comparing investment costs and exergy destruction values, the component with the highest value was the feed water tank with 99.95%, while the component with the lowest exergoeconomic factor value was the condenser with 34.07%.

As a result of the energy and exergy analysis carried out, the highest exergy destruction was determined in the combustion chamber. The most important reason for the exergy destruction is the chemical reactions taking place inside the component in the combustion chamber. The combustion reaction, which occurs as a result of chemical reactions, increases the exergy destruction of the component, in other words, increases its irreversibility. Within the scope of suggestions to reduce exergy destruction in the combustion chamber, increasing the insulation to increase the combustion efficiency and ensuring that the heat transfer remains within the component can be given.

The component with the highest exergy destruction after the combustion chamber was the gas turbine. The temperature and pressure difference at the gas turbine inlet and outlet are the main causes of exergy destruction. Reducing the temperature difference in the turbine will reduce the exergy destruction. Regular maintenance of turbine blades and replacing blades with higher heat resistance with old ones can be said as the main options. The third highest exergy destruction occurred in the heat recovery steam generator. The most important reason for this is the temperature difference in HRSG. Among the applications to reduce the exergy destruction of HRSG can be said the regular maintenance of the pipe bundles inside.

In order to reduce the exergy destruction values of low-pressure and high-pressure steam turbines, which are among the other components of the power plant, some suggestions can be given such as; increasing the steam quality, performing regular maintenance of turbine blades, using new blades with high thermal insulation, and replacing new technological steam turbines with old ones. Increasing the necessary maintenance of the components, especially at the inlets and outlets of the components and regular cleaning and replacing the components with low exergy efficiency with new ones can be given as some suggestions for the others such as the condenser, feed water tank and pumps.

The unit exergy costs were calculated for each pipe in the power plant according to the SPECO method chosen for exergoeconomic analysis. Exergy costs were calculated by using the unit exergy cost and exergy values for again each pipe. The highest unit exergy cost was obtained in pipe number 2, and the highest exergy flow occurred in pipe number 3. Reducing the unit exergy cost in pipe number 2 cannot be possible. Because, the unit exergy cost of the atmospheric air sucked from pipe number 1 at the compressor inlet is always zero, and therefore the inlet of the compressor cannot be intervened. Instead, the application can be performed to reduce the exergy cost in pipe number 3. Decreasing the exergy value of pipe number 3 located at the exit of the combustion chamber can be said as a suggestion.

The environmental impacts resulting from the manufacturing, operation and maintenance and disposal of each component as salvage were calculated and total environmental impact values were determined for the life cycle assessment of the power plant. According to the results, the component with the highest environmental impact value was determined as the heat exchanger, while the main water pump has the lowest value. These values are respectively as follows, 1142.53 mPts/kg, 285.63 mPts/kg and 137.10 mPts/kg. The component with the lowest environmental impact score due to its manufacturing, operation and maintenance and disposal as scrap was the main water pump. The environmental impact scores for the main water pump are 0.3 mPts/kg, 0.07 mPts/kg, 0.04 mPts/kg and 0.41 mPts/kg, respectively. Considering the total environmental impact scores, the highest value was obtained for the heat exchanger with 1565.27 mPts/kg, while the lowest value was seen in the main water pump with 0.41 mPts/kg.

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