Sudan University of Science and Technology

College of Graduate Studies

Exergy Analysis of Combined Cycle Power Plant (GARRI"2" 180 MW)

تحليل الإكسيرجي للدورة المشتركة في محطة الطاقة (180MW)قري(2)

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DEDICATION

The thanks and the Praises all are always for the **almighty God** who without his Bounty and inspiration nothing will be accomplished.

After that I would like to dedicate this simple work for **my father's soul**, **my mother** and all family members for their encouragement and support throughout the entire process. And special dedication to my uncle **captain: Kamal Abbas** for his continuous encouragement and support.

And last but not least I would like to dedicate this work to my frinds **Mr :(M. rammadan khalil , Osman bashier makkwi**), and all friends and colleagues who helped direct or indirect to complete this project.

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ABSTRACT

In recent years the entire world is a warring about energy resources limitation, specifically Sudan country confronts critical crises in fossil fuel resources and yet more of electricity energy producing from fired thermal power plants. One from these plants Garri"2" combined cycle producing about 180MW. Exergy analysis in the light second law of thermodynamics is powerful tools to investigate from optimization of engineering devices. Exergy analysis has been carried out analytically for Garri"2" to evaluate exergetic efficiency and exergy destruction of each part; exergy balance and entropy generation calculated to achieve it .The results showed that combustion chambers are the main sources of exergy destruction due to high irreversibilities representing 63% from total exergy destruction, followed by gas turbines 13.6%, steam turbines 6.4%, heat recovery steam generators(HRSGs) 6.3%, stacks (exhaust gases) 4.7%, compressors 3.8% and cooling systems 2.3%. The results also showed that thermal and exergetic efficiencies for entire plant are (38%, 49%) respectively.

مستخلص البحث

في السنوات الأخيرة زاد قلق العالم بمحدودية مصادر الطاقة ، جمهورية السودان بصفة خاصة تواجه أزمة حادة في مصادر الوقود الأحفوري ومع ذلك معظم الطاقة الكهربائية تنتج من محطات توليد حراري . واحدة من هذه المحطات محطة قري "2" تعمل بالدورة المشتركة و تنتج حوالي 180 ميقاواط. تحليل الاكسيرجي على ضوء القانون الثاني للديناميكا الحرارية يعتبر أداة فاعلة جدا للتحقق من كفاءة أداء المنظومات الهندسية. تم إجراء تحليل الاكسيرجي لمحطة قري "2" لحساب كفاءة الإكسيرجي وحساب الإكسيرجي المتبدد في كل وحدة من وحدات المحطة على حده؛ تم إستخدام موازنة الاكسيرجي وحساب الانتروبي المتولد لإنجاز الدراسة. أظهرت الدراسة أن غرف الإحتراق هي المصدر الرئيسي في تبديد الإكسيرجي تمثل حوالي 63% من مجموع الاكسيرجي المتبدد في المحطة تليها التوربينات الغازية بنسبة 13.6% ، ثم التوربينات البخارية 6.4%، الغلاية 6.3%، الاكسيرجي المتبدد نتيجة طرد غازات الاحتراق الى الهواء الجوي عند درجة اعلى من درجة حرارة الجو لتجنب تكون احماض على المدخنة 4.7% ، الضواغط 3.8%، أنظمة التبريد 2.3%. كما أظهرت النتائج أن المحطة بظروف التشغيل المدروسة تعمل بكفاءة حرارية38% وكفاءة اكسيرجي 49% .

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NOMENCLATURE

Symbols:

CCPP = combined cycle power plant.

HRSG = heat recovery steam generator.

DV.DA = diverter damper.

HP.SH = high pressure super heater.

HP.EV = high pressure evaporator.

HP.EC = high pressure economizer.

ex = Specific exergy.

c_p = Specific heat.

EX = Exergy rate (MW).

$\mathrm{EX_d}$ = Exergy destruction rate (MW).

EX_{td} = Total exergy destruction.

LHV = Lower heating value (Mj/kg).

ξ = Ration between LHV and chemical exergy of fuel.

T = Absolute temperature (k).

S_{gen} = Entropy generation rate (MW/k).

s^{-} = specific entropy (kj/kmole).

s = specific entropy (kj/kg).

$POW_{net,out}$ = Output power (MW).

Q = Rate oh heat flow (MW).

POW_u = Useful power (MW).

POW_{rev} = Reversible power (MW).

POW_{AC} = Actual compressor power (MW).

POW_C = Compressor power (MW).

$B.POW.R$ = Back power ratio.

n = mole flow rate (kmole/s).

m = mass flow rate (kg/s).

H = Enthalpy rate (MW).

h = specific enthalpy (kj/kg).

h^- = specific enthalpy (kj/kmole).

$\mathrm{EX_d}$ = Exergy destruction rate (MW).

η_{th} = Thermal efficiency.

$\eta_{th,rev}$ = Carnot (maximum available) efficiency.

η_{ex} = exergetic (second law) efficiency.

y = mole fraction.

Subscript and superscript:

f= fuel.

w =water.

p = pressure.

POW= power.

i= in.

e = exit.

0= ambient condition.

el = element.

CHAPTER ONE

INTRODUCTION

1.1 Background

The increased awareness that the world's energy resources are limited has caused many countries to reexamine their energy policies and take drastic measures in eliminating waste [1]. In recent years, the role of combustion engines technology in human life has been highlighted because over 80% of worldwide energy demand has been fulfilled by combustion methods. The augmentations in combustion efficiency and pollutant reduction have become the main concerns of combustion researchers in academic societies and of industrial manufacturers. In the combustion process, a reaction between the fuel and the oxidizer occurs to release heat (thermal energy) and consequently generate electricity. Current researchers focus on increasing combustion performance while reducing the emission of these pollutants. The most important factor driving the increasing focus on combustion performance is energy savings because the anticipated global energy demand is expected to rise by 58% between 2001 and 2025. Figure 1.1 shows the world's energy production by source [2]. From this Figure, we readily observe that the world's three main sources of energy are coal, natural gas and oil; each of which depends upon combustion. In the foreseeable future, these energy sources are expected to continue their domination. Although between 2001 and 2025 the global production of renewable energy is expected to rise by 8%, the expected annual growth of energy demand will rise by 1.9% [3, 4].

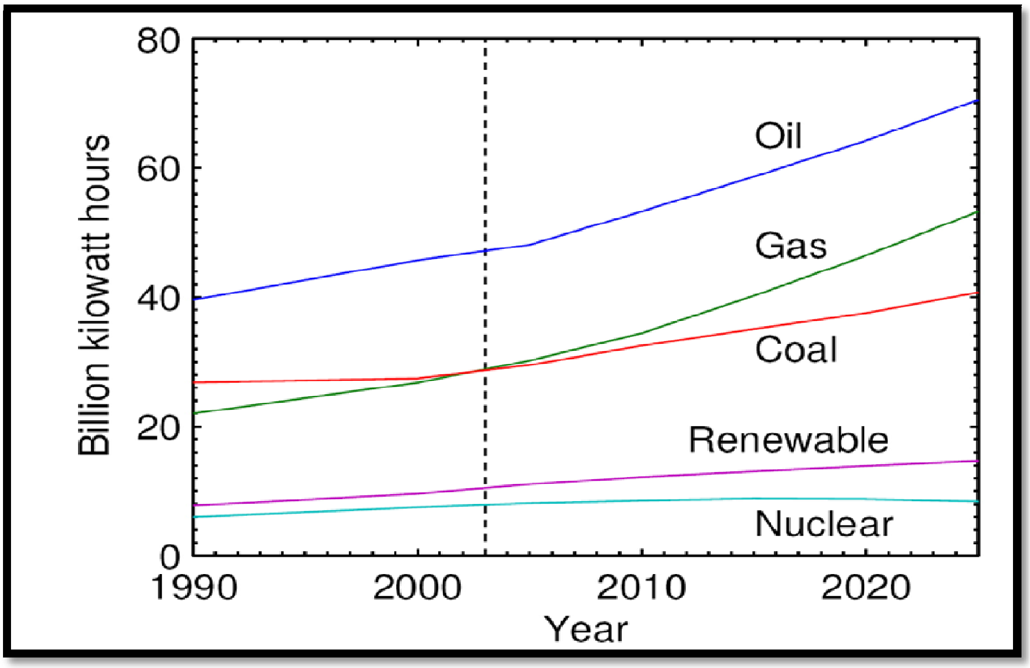


Figure 1.1: World energy productions by sources [2].

Most of the investigations have focused on increasing combustion performance by conserving energy. The first law of thermodynamics deals with the quantity of energy and asserts that energy cannot be created or destroyed. This law merely serves as a necessary tool for the bookkeeping of energy during a process and offers no challenges to the engineer. The second law, however, deals with the quality of energy. More specifically, it is concerned with the degradation of energy during a process, the entropy generation, and the lost opportunities to do work; and it offers plenty of room for improvement [1]. The second law of thermodynamics has proved to be a very powerful tool in the optimization of complex thermodynamic systems. The optimization of power generation systems is one of the most important subjects in the energy engineering field. Due to the high prices of energy and the decreasing fossil fuel resources, the optimum application of energy and the energy consumption management methods are very critical. The combined cycle power plants (CCPPs) have higher thermal efficiency than the separate steam and gas turbine cycle power plants [2]. 80% from electricity energy produced from fired thermal power plant [5] hence non renewable energy (fossil fuel) so that the world aware about the optimum use for this reserve. First law of thermodynamics awarded criteria to energy transfer using energy balance whereas the second law of thermodynamics gives us more insight to evaluate efficient of thermal engineering systems using first law efficiency and second (exergy) efficiency. Exergy is defined as the maximum theoretical useful work that can be obtained as a system interacts with an equilibrium state. The exergy is not generally conserved like energy but is destroyed in the system. Exergy calculation shows the place in the system where losses occur and the magnitude of these losses. Thermal efficiency of engineering systems calculated by the rational between output and input no matter about maximum theoretical efficiency (Carnot efficiency) whilst exergy efficiency calculated by the rational between thermal efficiency and maximum theoretical efficiency to knowing the portion from available energy destroyed due to inefficient processes. The thermal power plant are widely used in Sudan grid net work, CCPP called Garri “2” consider one from these plants producing about 190 MW in design condition this study aimed to estimate exergy destruction due to processes individually and calculate exergy efficiency to know who is process less efficient and made compare with other literature review to satisfy whether plant under study operated with reasonable accuracy or not for either process.

1.2 Problem statement

Current research and development in the field of combustion engines technology was focused to improvements combustion engines performance. There are many methods and approaches to solutions these problems, one of which is Exergy analysis. Analysis in the light of second law of thermodynamics require to evaluate exergy destruction of each part; to determine the parts has major contribute in exergy destruction.

1.3 Objective study

The objective of this study is to evaluate the exergy (second law) analysis of the performance of Garri (2) an existing 180 MW (fuel-fired) electrical combined cycle power plant to identify the potential for improvement.

1.4 Scope

The research scope covers, calculate exergy destruction to main equipment (compressors, combustion chambers, gas turbines, heat recovery steam generators, steam turbine, due to escape of flue gases over ambient and cooling systems).

1.5 Significance of Research

The significance of this study as follows:

1. Facilitate the achievement of better and more efficient combustion engine processes for all concerned industries.

2. This study will assist industrial energy conservation by offering an improved approach to thermal efficiency.

1.6 Thesis outline

The general outline of the thesis is as follows:

- General background, problem statement, the purpose and significance of study (chapter one).
- Literature review consist from general historical background for thermal power plant and its modifications, exergy analysis concepts and formulations specifically for flow processes and finally review of relevant studies (chapter two).
- Exergy analysis of general combined cycle power plants, exergy analysis for Garri”2” combined cycle power plant producing 180 MW (chapter three).
- Results and discussions (chapter four).
- General conclusions drawn from the study and Recommendations including how the study results could be used, possible improvements in the present work, and potential areas of future work (chapter five).

CHAPTER TWO

LITREATURE REVIEW

2.1 Introduction

A power plant is plying very important role in engineering field (also referred to as a generating station, power station, powerhouse, or generating plant) is an industrial facility for the generation of electric power. Most power stations contain one or more generators, a rotating machine that converts mechanical power into electrical power. The relative motion between a magnetic field and a conductor creates an electrical current. The energy source harnessed to turn the generator varies widely. Most power stations in the world burn fossil fuels such as coal, oil, and natural gas to generate electricity. Others use nuclear power, but there is an increasing use of cleaner renewablesourcessuchas solar, wind, wave and hydroelectric[6,7, 8]. A turbo machinery fired thermal power plants are producing most electric energy in the world; this type of plants operate on a deferent cycle and modes as a follow. The Brayton cycle was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed around 1870[1]. Today, it is used for gas turbines only where both the compression and expansion processes take place in rotating machinery. Gas turbines usually operate on an open cycle, Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised. The high pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure. The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power. The exhaust gases leaving the turbine are thrown out (not recirculated), the cycle to be classified as an open cycle. The open gas-turbine cycle described above can be modeled as a closed cycle, by utilizing the air-standard assumptions. Here the compression and expansion processes remain the same, but the combustion process is replaced by a constant-pressure heat-addition process from an external source, and the exhaust process is replaced by a constant pressure heat-rejection process to the ambient air.

The processes taking place in power-generating systems are sufficiently complicated that idealizations are required to develop thermodynamic models. Such modeling is an important initial step in engineering design. They also provide relatively simple settings in which to discuss the functions and benefits of features intended to improve overall performance. The vast majority of electrical generating plants are variations of vapor power plants in which water is the working fluid. The basic components of a simplified fossil-fuel vapor power plant. To facilitate thermodynamic analysis, the overall plant can be broken down into the four major subsystems [9].

The continued quest for higher thermal efficiencies has resulted in rather innovative modifications to conventional power plants. The binary vapor cycle discussed later is one such modification. A more popular modification involves a gas power cycle topping a vapor power cycle, which is called the combined cycle power plant, or just the combined cycle. The combine cycle of greatest interest is the gas-turbine (Brayton) cycle topping a steam turbine (Rankine) cycle, which has a higher thermal efficiency than either of the cycles executed individually Gas-turbine cycles typically operate at considerably higher temperature than steam cycle. The maximum fluid temperature at the turbine inlet is about 620°C for modern steam power plants, but over 1425°C for gas-turbine power plants. It is over 1500°C at the burner exit of turbojet engines. The use of higher temperatures in gas turbines is made possible by recent developments in cooling the turbine blades and coating the blades with high-temperature-resistant materials such as ceramics. Because of the higher average temperature at which heat is supplied, gas-turbine cycles have a greater potential for higher thermal efficiencies. However, the gas-turbine cycles have one inherent disadvantage: The gas leaves the gas turbine at very high temperatures (usually above 500°C), which erases any potential gains in the thermal efficiency. The situation can be improved somewhat by using regeneration, but the improvement is limited. It makes engineering sense to take advantage of the very desirable characteristics of the gas-turbine cycle at high temperatures and to use the high temperature exhaust gases as the energy source for the bottoming cycle such as a steam power cycle. The result is a combined gas–steam cycle. In this cycle, energy is recovered from the exhaust gases by transferring it to the steam in a heat exchanger that serves as the boiler. In general, more than one gas turbine is needed to supply sufficient heat to the steam. Also, the steam cycle may involve regeneration as well as reheating. Energy for the reheating process can be supplied by burning some additional fuel in the oxygen-rich exhaust gases. Recent developments in gas-turbine technology have made the combined gas–steam cycle economically very attractive. The combined cycle increases the efficiency without increasing the initial cost greatly. Consequently, many new power plants operate on combined cycles, and many more existing steam- or gas-turbine plants are being converted to combined-cycle power plants. Thermal efficiencies well over 40 percent are reported as a result of conversion [1].

A 1090-MW Tohoku combined plant that was put in commercial operation in 1985 in Niigata, Japan, is reported to operate at a thermal efficiency of 44 percent. This plant has two 191-MW steam turbines and six 118-MW gas turbines. Hot combustion gases enter the gas turbines at 1154°C, and steam enters the steam turbines at 500°C. Steam is cooled in the condenser by cooling water at an average temperature of 15°C. The compressors have a pressure ratio of 14, and the mass flow rate of air through the compressors is 443 kg/s. A 1350-MW combined-cycle power plant built in Ambarli, Turkey, in 1988 by Siemens of Germany is the first commercially operating thermal plant in the world to attain an efficiency level as high as 52.5 percent at design operating conditions. This plant has six 150-MW gas turbines and three 173-MW steam turbines. Some recent combined cycle power plants have achieved efficiencies above 60 percent [1].

2.2 Exergy analysis

Exergy is composed of two important parts. The first one is the physical exergy and the second one is the chemical exergy. The kinetic and potential parts of exergy are negligible. Exergy is defined as the maximum theoretical useful work that can be obtained as a system interacts with an equilibrium state. The chemical exergy is associated

with the departure of the chemical composition of a system from its chemical equilibrium. The chemical exergy is an important part of exergy in combustion process [10].

2.2.1 Exergy formulation

A general exergy balance equation, applicable to any component of a thermal system may be formulated by utilizing the first and second laws of thermodynamics. The thermo-mechanical (physical) exergy stream may be decomposed into its thermal and mechanical components. The balance gives [10]:

$$ex = ex_{ph} + ex_{ch} \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2-1)}$$

Physical exergy is defined as the follow [11]:

$$\mathrm{e}x_{\mathrm{ph}} = \mathrm{e}x_{\mathrm{T}} + \mathrm{e}x_{\mathrm{p}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2-2)}$$

$$\mathrm{e}x_{\mathrm{T}} = \mathrm{c_p}\left[(\mathrm{T}-\mathrm{T_0}) - \mathrm{T_0}\ln\left({}^{\mathrm{T}}/_{\mathrm{T_0}}\right)\right] \ldots\ldots\ldots\ldots \text{(2-3)}$$

$$\mathrm{e}x_{\mathrm{P}} = \mathrm{RT_0}\ln(\mathrm{p}/\mathrm{p_0}) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2-4)}$$

Subs (3-2), (3-3) in (3-1) give

$$\mathrm{e}x_{\mathrm{ph}} = (\mathrm{h}-\mathrm{h_0}) - \mathrm{T_0}(\mathrm{s}-\mathrm{s_0}) \quad \ldots\ldots\ldots\ldots\ldots \text{(2-5)}$$

If one applies the first and second laws of thermodynamics, one can find the formula for exergy balance as [10, 12].

$$EX_Q + \sum_i m_i ex_i = \sum_e m_e ex_e + EX_{POW} + EX_d \text{ .... (2-6)}$$

$$EX_Q = \left(1 - {}^{T_0}/_{T_i}\right) Q_i \text{ .... (2-7)}$$

$$EX_{POW} = POW \text{ .... (2-8)}$$

The chemical exergy for gas mixtures is defined as follows [12]:

$$ex_{ch}^{mix} = \left[\sum_{i=1}^{n} x_i ex_{ch_i} + RT_0 \sum_{i=1}^{n} x_i \ln x_i\right] \text{ .... (2-9)}$$

For the evaluation of the fuel exergy, the (2-9) formula cannot be used. Thus, the corresponding ratio of simplified exergy is defined as the following [10, 12]:

$$\xi = ex_f / LHV_f \text{ .... (2-10)}$$

The ratio of chemical exergy to LHV_f is usually close to unity. In general fuel with chemical formula $C_x H_y$.

For gaseous fuels [10]:

$$\xi = 1.033 + 0.0169\,{}^{y}\!/_{x} - {}^{0.0689}\!/_{x} \;\ldots\ldots\ldots\ldots\ldots \text{(2-11)}$$

For liquid fuels [13]:

$$\xi = 1.0422 + 0.011925\,{}^{y}\!/_{x} - {}^{0.042}\!/_{x} \quad \ldots\ldots\ldots \text{(2-12)}$$

To find exergy destruction; exergy balance from equation (2-6) can be used and also entropy generation concept applicable to evaluate it if entropy generation can be calculated.

$$EX_d = T_0 S_{gen} \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2-13)}$$

2.2.2 Efficiency laws

Efficiency is the (often measurable) ability to avoid wasting materials, energy, efforts, money, and time in doing something or in producing a desired result. In a more general sense, it is the ability to do things well, successfully, and without waste. In more mathematical or scientific terms, it is a measure of the extent to which input is well used for an intended task or function (output). It often specifically comprises the capability of a specific application of effort to produce a specific outcome with a minimum amount or quantity of waste, expense, or unnecessary effort. Specifically this text present most efficiencies uses in power plant analysis.

2.2.2.1 Thermal efficiency

The fraction of the heat input that is converted to net work output is a measure of the performance of a heat engine and is called the *thermal efficiency (η_{th})*. For heat engines, the desired output is the net

Work output, and the required input is the amount of heat supplied to the working fluid. Then the thermal efficiency of a heat engine can be expressed as:

$$\eta_{th} = \frac{POW_{net,out}}{Q_{in}} \quad \text{...} \quad (2\text{-}14)$$

Since ($POW_{net,out} = Q_{in} - Q_{out}$) It can also be expressed as:

$$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}} \quad \text{……………..………………..} \quad (2\text{-}15)$$

2.2.2.2 Carnot efficiency

The hypothetical heat engine that operates on the reversible manner cycle is called the Carnot heat engine. The thermal efficiency of any heat engine, reversible or irreversible, is given by equation (2-15). Where Q_{in} heat is rate transferred to the heat engine from a high temperature reservoir at T_H, and Q_{out} is rate heat rejected to a low temperature reservoir at T_L. For reversible heat engines, the heat transfer ratio in the above relation can be replaced by the ratio of the absolute temperatures of the two reservoirs, as given by equation (2-16). Then the efficiency of a Carnot engine, or any reversible heat engine, becomes

$$\eta_{th,rev} = 1 - \frac{T_L}{T_H} \text{ .... (2-16)}$$

2.2.2.3 Exergetic (second law) efficiency

In previous we defined the *thermal efficiency* for devices as a measure of their performance. They are defined on the basis of the first law only, and they are sometimes referred to as the *first law*

efficiency. The first law efficiency, however, makes no reference to the best possible performance, and thus it may be misleading; because it is not refer to maximum efficiency (Carnot efficiency) can be achieved**.** If reversible device, these can be treated by calculate ratio of actual thermal efficiency to the maximum possible (Carnot) efficiency under same condition equation (2-17) (For heat engine).

$$\eta_{ex} = \frac{\eta_{th}}{\eta_{th,rev}} \text{ .... (2-17)}$$

Subs (2-15), (2-16) in (2-17) exergy efficiency can be written as

$$\eta_{ex} = \frac{POW_u}{POW_{rev}} \quad \text{(Power–producing devices)… (2-18)}$$

$$\eta_{ex} = \frac{POW_{rev}}{POW_u} \quad \text{(Power –consuming devices)... (2-19)}$$

Exergy efficiency general formula (2-20) or (2-21):

$$\eta_{ex} = \frac{exergy\ recovered}{exergy\ supplied} \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2-20)}$$

$$\eta_{ex} = 1 - \frac{exergy\ destruction}{exergy\ supplied} \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2-21)}$$

2.3 Exergy analyzed for thermal power plants

There is more researchers studied power plant in the light of second law of thermodynamics whether combined cycle or separately gas turbines and steam turbine cycle. Tremendous number of scientific papers exist in different journals, these text are pointed to some papers; specifically result achieved by study as the follow.

Energy, exergy and exergo-economic analysis of Montazer Ghaem gas turbine power plant which is located near Tehran, capital city of Iran is carried out. The results of this study reveal that the highest exergy destruction occurs in the combustion chamber (CC), where the large temperature difference is the major source of the irreversibility. In addition, the effects of the gas turbine load variations and ambient temperature are investigated to see how system performance changes: the gas turbine is significantly affected by the ambient temperature which leads to a decrease in net power output. The results of the load variation of the gas turbine show that a reduction in gas turbine load results in a decrease in the exergy efficiency of the cycle as well as all the components. As was expected, an increase in ambient temperature has a negative effect on the exergy efficiency of the cycle, so this factor could be countered by using gas turbine air inlet cooling methods. In addition, an exergo-economic analysis is conducted to determine the cost of exergy destruction in each component and to determine the cost of fuel. The results show that combustion chamber has the largest cost of exergy destruction, which is in line with the exergy analysis [14].

Energy and Exergy Analysis of a 348.5 MW Kostolac steam power plant in Serbia country is presented. The results show that energy losses have mainly occurred in the condenser where 421 MW is lost to the environment while only 105.78 MW has been lost from the boiler. Nevertheless, the irreversibility rate of the boiler is higher than the irreversibility rates of the other components. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system (88.2%) followed by the turbines (9.5%), and then the forced draft fan condenser (0.5%). In addition, the calculated thermal efficiency based on the lower heating value of fuel was 39% while the exergy efficiency of the power cycle was 35.77% [15]. Energy and exergy analysis of Al-Hussein steam power plant in Jordan presented. The percentage ratio of the exergy Energy losses mainly occurred in the condenser where 134MW is lost to the environment while only 13 MW was lost from the boiler system. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system (77%) followed by the turbine (13%), and then the forced draft fan condenser (9%). In addition, the calculated thermal efficiency based on the lower heating value of fuel was 26% while the exergy efficiency of the power cycle was 25%. For a moderate change in the reference environment state temperature, no drastic change was noticed in the performance of major components and the main conclusion remained the same; the boiler is the major source of irreversibilities in the power plant. Chemical reaction is the most significant source of exergy destruction in a boiler system which can be reduced by preheating the combustion air and reducing the air–fuel ratio [16]. Exergoeconomic analysis and optimization of a cogeneration system which produces 50 MW of electricity and 15 kg/s of saturated steam at 2.5 bar. He optimized the unit using exergoeconomic principles and evolutionary programming, and showed that the cost of electricity production is 9.9% lower for the optimum case in terms of exergoeconomics compared to a base case [12, 17]. Exergy analysis and optimization of a single-flash geothermal power plant are conducted by developing a mathematical model that is applied to the Dieng geothermal power plant in Indonesia. The exergy of the geothermal fluid that is discharged from the production wells is estimated to be 59.52 MW. This amount of fluid produces 21.71MWof electricity from the power plant overall, with second law efficiency to be 36.48%. There is a considerable amount of waste brine, amounting to 17.98% (10.70 MW) of the total available exergy, which is disposed of in the plants reservoir [18]. Energy and exergy analysis of biomass co-firing based pulverized coal power generation. In that work energy and exergy analyses are carried out for a co-firing based power generation system to investigate the impacts of biomass cofiring on system performance and gaseous emissions of CO2, NOx, and SOx. The power generation system considered is a typical pulverized coal-fired steam cycle system, while four biomass fuels (rice husk, pine sawdust, chicken litter, and refuse derived fuel) and two coals (bituminous coal and lignite) are chosen for the analysis. System performance is evaluated in terms of important performance parameters for different combinations of fuel at different co-firing conditions and for the two cases considered. The results indicate that plant energy and exergy efficiencies decrease with increase of biomass proportion in the fuel mixture [19].

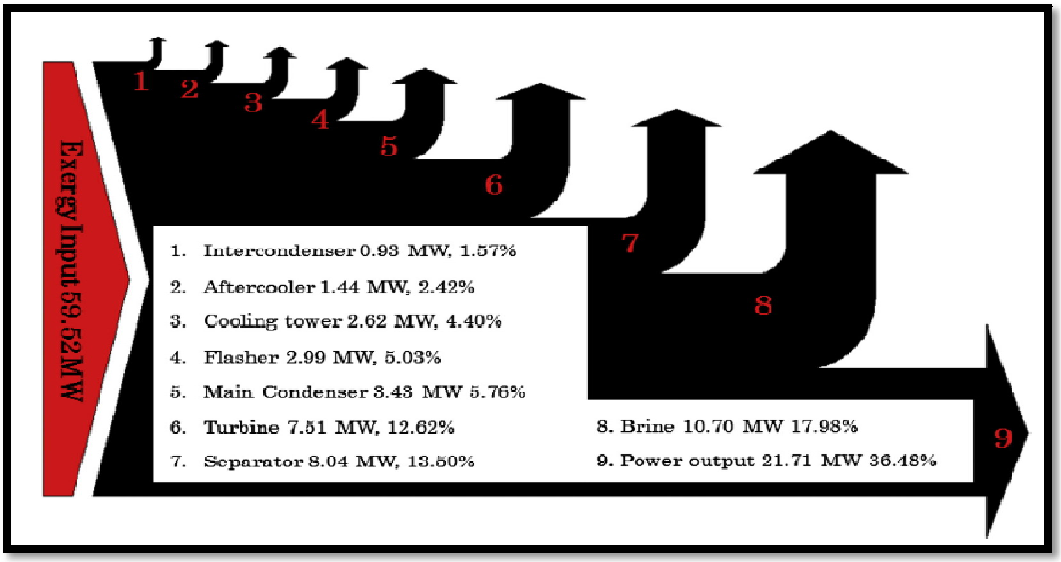


Figure 2.1: Illustrate exergy distributions due to plant [18].

Exergy and exergoeconomic Analyses of combined cycle power plant. Result show that the largest exergy destructions occur in the CCPP combustion chamber, and that increasing the gas turbine inlet temperature decreases the CCPP cost of exergy destruction [12]. Energy and exergy analyses for a combined cycle located in Turkey and suggested modifications to decrease the exergy destruction in CCPPs. Their results showed that combustion chambers, gas turbines and HRSGs are the main sources of irreversibilities, representing over 85% of the overall exergy losses [20] by [12]. Exergy analysis of a 420MW combined cycle power plant. Neka CCPP, is located near the Neka city beside the Caspian Sea(Iran).this power plant has two gas turbines, two compressors, two HRSGs, two deaerators, one steam turbine and one surface condenser with a cooling system that uses seawater as cooling media. The Siemens V94.2 gas turbines of this combined cycle have been installed in 1982. Since these gas turbines are operation for more than 20 years the flue gas parameters are different from new gas turbines..The results show that the combustion chamber, gas turbine, duct burner and heat recovery steam generator (HRSG) are the main sources of irreversibility representing more than 83% of the overall exergy losses. The results show that the greatest exergy loss in the gas turbine occurs in the combustion chamber due to its high irreversibility. As the second major exergy loss is in HRSG, the optimization of HRSG has an important role in reducing the exergy loss of total combined cycle. In this case, LP-SH has the worst heat transfer process. The first law efficiency and the exergy efficiency of CCPP are calculated. Thermal and exergy efficiencies of Neka CCPP are 47 and 45.5% without duct burner, respectively. The results show that if the duct burner is added to HRSG, these efficiencies are reduced to 46 and 44%. Nevertheless, the results show that the CCPP output power increases by 7.38%when the duct burner is used [10].

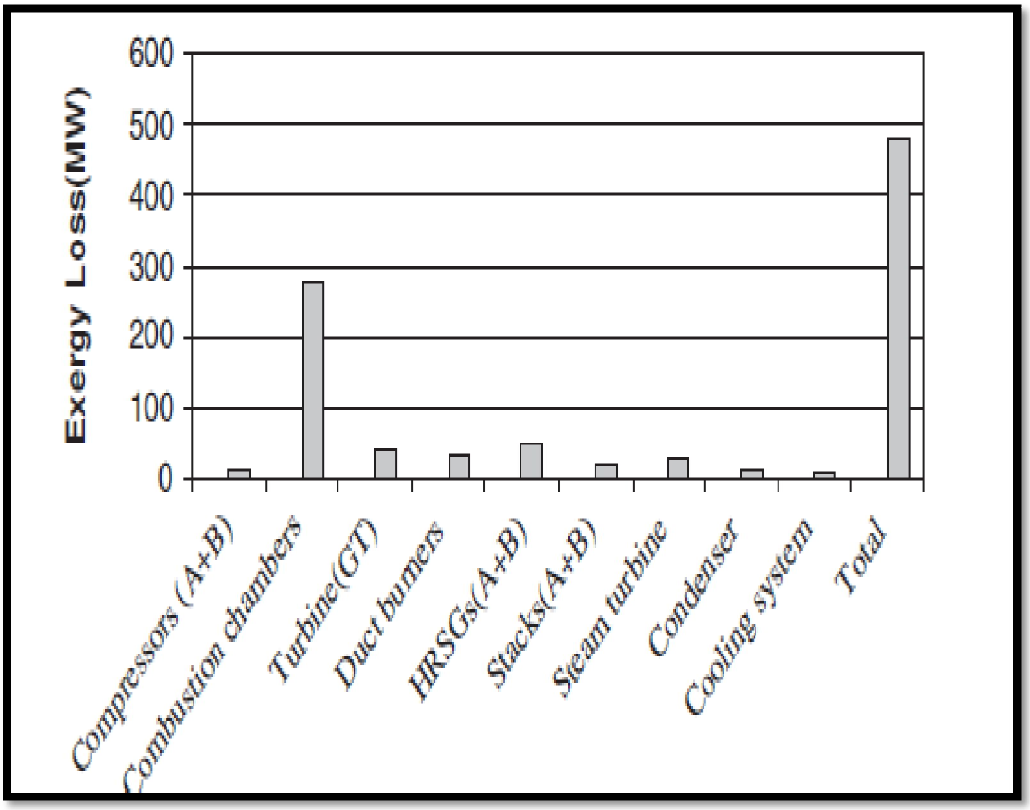


Figure2.6: contribute of each part on exergy destruction for Neka CCPP [10].

CHAPTER THREE

EXERGY ANALYSIS OF GARRI "2" POWER PLANT

3.1Exergy analysis of combined cycle power plant

In this text exergy analysis for general combined cycle power plant will carried out and could be used for either combined cycle power plans with respect any difference in design.

3.1.1 Exergy analysis of gas cycle component

In this section, the exergy analysis will be done for the gas turbine components.

3.1.1.1 Exergy analysis of compressor

This section decomposed of two parts, first exergy destruction due to compressor and second the power required rotating a compressor.

(I) Exergy destruction due to compressor

To calculate it uses equation (2-13), entropy generation in that equation with exact solution, should be calculated by equation (3-1) [1].

$$s_2 - s_1 = s_2^{0p} - s_1^{0p} - Rln\left({}^{p_2}/_{p_1}\right)\ldots\ldots\ldots\ldots \text{(3-1)}$$

Since adiabatic compression the deference in entropy means entropy generation as the follow:

$$s_{gen} = s_2 - s_1 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3-2)}$$

$$S_{gen} = m_{air} * s_{gen} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3-3)}$$

(Π) Compressor power and back power ratio

Useful compressor power is the deference between exergy in and out to compressor it can be achieved by equation (3-5). And actual compressor work required (exergy supply) achieved by useful work addition to exergy destruction on compressor, equation (3-4), and back work ratio represent the fraction of the turbine power used to drive the compressor calculated by equation (3-5).

$$\mathrm{POW_{AC}} = \mathrm{POW_{C}} + \mathrm{EX_{d}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3-4)}$$

$$\mathrm{B.POW.R} = \frac{\mathrm{POW_{AC}}}{\mathrm{POW_{net}}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3-5)}$$

3.1.1.2 Exergy analysis of combustion chamber

The objective of this section to calculate exergy destruction on combustion process and this can be achieved by exergy balance from equation (2-6). Since adiabatic combustion and no work in or out due to process; term(EX_Q) and term (EX_{POW}) can be neglected. To find exergy entered combustion chamber term $\sum_i m_i ex_i$ using equation (2-1); because exergy in combustion chamber is composed of two main parts (physical and chemical), Physical exergy exist in compressed air and chemical exergy in fuel (with neglect kinetic and potential exergy). Note that physical exergy of fuel neglected as well as chemical exergy due to mixtures. Physical exergy of air calculated directly from equation (2-5) by using properties of air from [21], and chemical exergy from equation (2-10), (2-12). To find exergy exiting combustion chamber term ($\sum_e m_e ex_e$) in equation (2-6) by equation (2-1) the working fluid here is flue gases hence to find enthalpy and entropy using equation (3-6)-(3-9). And for calculate rate of heat added to combustion chamber using equation (3-11).

$$H_g = n_f * \sum_{el=1}^{n} n_{el} \, \bar{h}_{el} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3-6)}$$

$$S_g = n_f * \sum_{el=1}^{n} n_{el} \;\, \bar{s}_{el} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3-7)}$$

$$s_{el}^{-}(T,p) = s_{el}^{0p-}(T,p_0) - R_u ln\left(\frac{y_{el}p_t}{p_0}\right) \text{ …….… (3-8)}$$

$$y_{el} = {}^{n_{el}}/_{n_t} \text{ ……………………..……………... (3-9)}$$

$$n_f = {}^{m_f}/_{M_f} \quad \text{ ……...………………………... (3-10)}$$

$$Q_{in} = H_o - H_{in} \text{ ……………..…………………..… (3-11)}$$

Equations (3-6)-(3-9) required chemical equation for combustion process to knowing number of moles and mole fraction for each element.

3.1.1.3 Exergy analysis of turbine

These section aimed to calculate exergy destruction of turbine unite due to expansion process this can be achieved by equations (2-5), (2-6). With respect adiabatic expansion process term(EX_Q) in equation (2-6) are executed. And term (EX_{POW}) in that equation represent useful power on turbine unit this achieved by output power plus actual compressor work. Note that output power given at outer electric generator and compressor work calculated at compressor unit; so that mechanical transport efficiency neglected in calculation.

3.1.2 Exergy analysis of steam cycle component

In this section exergy analysis for steam turbine cycle should be done specifically to calculate exergy destruction.

3.1.2.1 Exergy analysis of HRSG

Assume fully adiabatic and isobaric flow due to all stages in HRSG, Equation (2-13) applicable to calculate exergy destruction. Entropy generation at that equation is calculated by entropy balance equation (3-12) [1, 22]. And To calculate exergy of escape flue gases due to chimney equation (2-5) applicable.

$$S_{in} - S_{out} + S_{gen} = \Delta S_{sys} \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(3-12)}$$

Steady flow rate; entropy change term (ΔS_{sys}) will be zero, hence equation (3-12) becomes:

$$S_{gen} = S_{out} - S_{in} \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(3-13)}$$

The four component of HRSG deals as a heat exchanger and should be analyzed individually to knowing distribution of exergy destruction on HRSG. Entropy out (term (S_{out})) represent entropy out on flue gases and steam water, calculated by equation (3-14). entropy in as well as entropy out, by equation (3-15).

$$S_{out} = m_{gas}s_{gas,e} + m_w s_{w,e} \quad \text{……………..…… (3-14)}$$

$$S_{in} = m_{gas}s_{gas,i} + m_w s_{w,i} \quad \text{…………………..…….. (3-15)}$$

Substitute (3-14), (3-15) in (3-13) and rearranged

$$S_{gen} = m_{gas}\left(s_{gas,e} - s_{gas,i}\right) - m_w\left(s_{w,e} - s_{w,i}\right) \quad (3\text{-}16)$$

Term $\left(s_{gas,e} - s_{gas,i}\right)$ calculated by equation (3-17)

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{p_2}{p_1} \quad \text{…………………….. (3-17)}$$

Note: $\left(s_{w,e}, s_{w,i}\right)$ taken directly from steam table at specified state for each part in heat exchanger [21].

3.1.2.2 Exergy analysis of turbine

Exergy balance in equation (2-6) applicable, assume fully adiabatic expansion flow hence term $((\mathrm{EX_Q}))$ will be zero, neglect mechanical transport and generator losses. Term $(\mathrm{EX_P})$ represents output power .and terms ($\sum_i m_i ex_i, \sum_e m_e ex_e$) can be calculated by equation (2-5).

3.1.2.3 Exergy analysis for condenser

Exergy balance in equation (2-6) applicable, assume fully adiabatic heat exchanging hence term $((\mathrm{EX_Q}))$ will be zero and there is no work; $(\mathrm{EX_{POW}})$ equals zero .and terms ($\sum_i m_i ex_i, \sum_e m_e ex_e$) can be calculated by equation (2-5).

3.1.3 Summarize

Exergy efficiency of each part and whole plant in previous section calculated by equation (2-17) and (2-21); to check the methodology and calculation, both equations must be identical, and exergy destruction of whole combined cycle power plant should be achieved by summation of exergy destruction of each part equation (3-18). Table 3.1 gives summarize for exergy analysis for combined cycle power plant specifically single pressure type figure (3.1).

$$EX_{td} = \sum_c EX_d \quad \text{……………………………………………….} \quad (3\text{-}18)$$

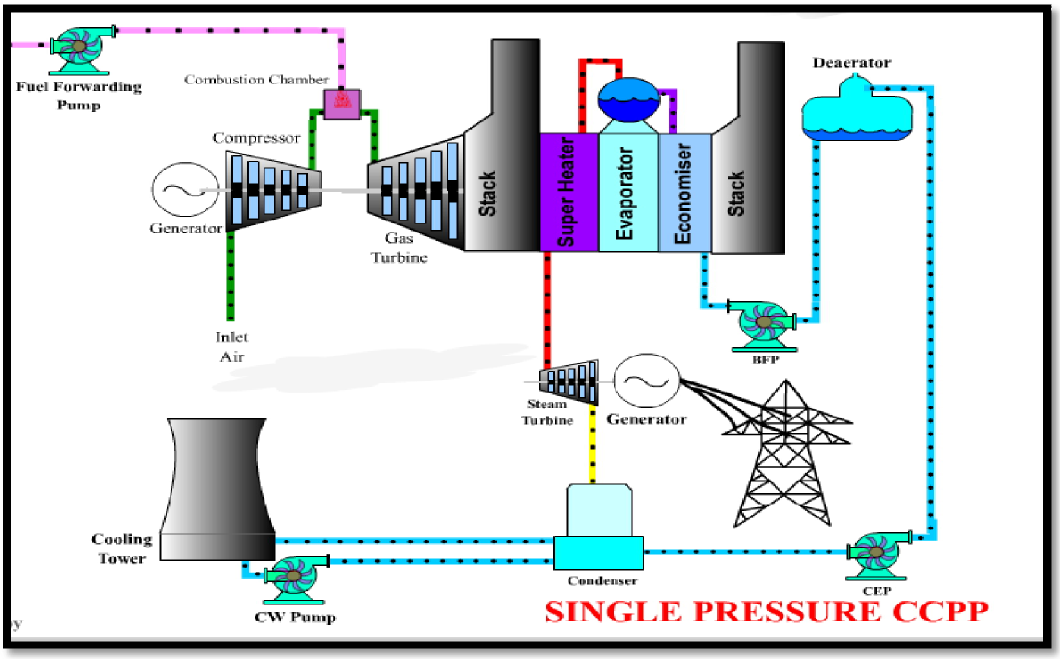


Figure 3.1: Illustrate single pressure CCPP.

Table 3.1: Expressions for exergy destruction rate and exergy efficiency for combined cycle power plant components

Plant component	Exergy destruction rate(MW)	Exergy efficiency
compressor	$EX_d = EX_{in} - EX_{out} + POW_{AC}$	$1 - \frac{EX_d}{POW_{AC}}$
Combustion chamber	$EX_d = EX_{in} - EX_{out}$	$\frac{EX_{out}}{EX_{in}}$
Gas turbine	$EX_d = EX_{in} - EX_{out} - POW_{AC} - POW_{net}$	$\frac{POW_{AC} + POW_{net}}{EX_{in} - EX_{out}}$
HRSG	$S_{gen} = m_{gas}(s_{gaso} - s_{gasi}) - m_w(s_{wo} - s_{wi})$	$1 - \frac{EX_d}{EX_{in}}$
HP.SH	$T_0 * S_{gen}$	
HP. EC	$T_0 * S_{gen}$	
HP.EV	$T_0 * S_{gen}$	
LP.EV	$T_0 * S_{gen}$	
stack	EX_{out}	Non
Steam turbine	$EX_d = EX_{in} - EX_{out} - POW_{net}$	$\frac{POW_{net}}{EX_{in} - EX_{out}}$
Condenser	EX_{in}	Non

3.2 Applied exergy analysis on Garri"2" power plant

First all details about Garri"2" power plant required to applied exergy analyzed in previous section **(3.1)**; this part explains main information and all data at specified operation condition to Garri"2" power plant as the follow.

3.2.1 Plant location

The power plant had been built in the north of the Khartoum refinery. The Khartoum refinery is about 70 km north from Khartoum and the distance between the refinery and the Nile River is about 12.5 km. about 5.5 km west of the refinery, the power plant site altitude is about 400 m, there is a highway passing from Khartoum city to northern Sudan, with which the access road of the refinery had been connected. The national highway and access road both is asphalt pavement. There is a railway lining about 3 km west of the Khartoum refinery [23].

3.2.2 Plant specification

The combined cycle power plant consists of two 206B combined cycle, two gas turbine. Two HRSGs and one steam turbine are used per block, as shown in figure 3.2. Each gas turbine type is PG 6001B; its output is approximately 40MW for ISO condition, the gas turbine is normally operated with light diesel oil. The gas turbine generator, which is driven at 3000 rpm, is with air-cooler. Each gas turbine exhausts gas lead to its associated HRSG. There is a diverter damper between the gas turbine exhaust and the HRSG. It allows the gas turbine to operate either in open cycle mode or in combined cycle mode. The exhaust gas flow and temperature characteristics at the gas turbine exhaust will be changed with their load. The HRSGs are of the single-pressure type. The main stream lines of each HRSG are led to the steam turbine. Steam turbine is condensing type with extraction steam [23].

3.2.3 Processes flow of plant

Ambient air is filtered and led to the compressor of the gas turbine, where it is compressed and fed to the combustors. In the combustors the compressed air is heated up to the turbine inlet temperature. Fuel combusted in combustion chamber and then expanding in turbine, after expansion the flue is led to the HRSG. Steam is generated in the HRSG by heat transfer from the flue gases to the feed water. The HRSG is a single pressure boiler. From the two parallel HRSGs the superheated HP steam is fed to the steam turbine. The expanded steam is condensed in a water cooled condenser, all this processes illustrated in figure 3.2[23]. In order to obtain optimum utilization of the steam, the pressure at the exhaust is optimized to the condenser cooling system. Air and non condensable gases entering the water/steam cycle are collected at the coldest part of the condenser and evacuated. During normal operation the vacuum is maintained with Liquid-Ring vacuum pump. The condensate and make up water accumulating in the condenser hot -well is delivered by condensate pumps to the dearator water tank each condensate pump is provided with separate suction lines from the hot-well ensuring a short and direct connection. One pump is in operation, the second serve as standby and is switched on automatically in case of failure of the running pump. The level in hot well is kept constant by control of the make-up feed [23].

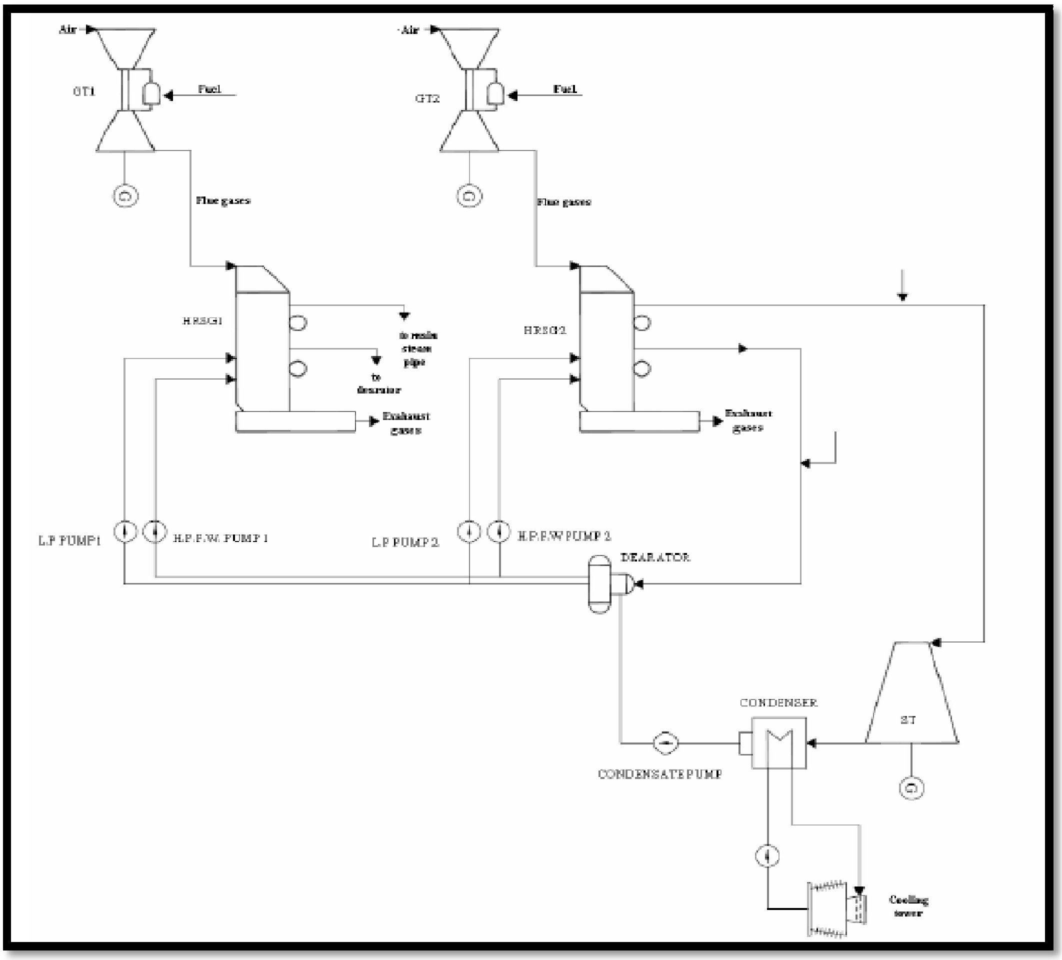


Figure 3.2: Illustrate Garri”2” one block configuration.

3.2.4 Operation data

The main fuel uses in Garri”2” combined power plant is the light diesel oil (L.D.O) produced in Khartoum refinery. Therefore, the exergy analysis is performed for this type of fuel. The fuel composition is approximate as $C_{12.3}H_{22.14}$. The air that is used in the current analysis can be approximated as 21% oxygen and 79% nitrogen. Exergy analysis of plant will perform for one block and generalized for whole plant because both blocks are symmetrical. Stoichiometric air fuel ratio (AFR) for specified type of fuel 14.5 (kg(air)/(kg(fuel)) and approximate air fuel ratio 55(kg(air)/(kg(fuel)) hence very lean mixture, the highest temperature of gases about 1320K so that dissociation of carbon dioxide and steam water cannot be done because dissociation is happened at very high temperature. The temperature and pressure for process given in table 3.3, Hence combustion formulas should be written and tabulated in tables 3.5, 3.6. **Note**: all properties required could be takes from [21].

Table 3.2: Main data for Garri “2”

Parameter	Quantity
Output power at (ISO) condition for one gas turbine (MW)	39.174
Output power at operation condition for one gas turbine (MW)	29.7
Atmospheric pressure in (bar)	0.966
Atmospheric temperature in (C^0)	35
Lower heating value of fuel (LHV_f) in Mj/kg	43.2
$A.F.R_{ST}$	14.5
$A.F.R_{AC}$	55
Fuel consumption in (kg/s)	2.55
Output power from one steam turbine at operation condition in (MW)	30.3
Total output power from one block in (MW)	89.7
Total output power from Garri “2” in (MW)	179.4

Table 3.3: Operation data of gas turbine cycle

Equipment	Position	T (k)	p(bar)
Compressor	in	308	0.966
	out	630	10.2
Combustion chamber	in	630	10.2
	out	1320	10.2
Gas turbine	in	1320	10.2
	out	850	1.27

Table 3.4: Properties of air thrown due to compressor

Equipment	Position	h (Mj/kg)	s_0 (Mj/kg.k)
Compressor	In	0.308	0.0017
	Out	0.639	0.0025

Table 3.5: Reactants properties

Element	Numbers of kmoles	Mole fraction%
$C_{12.3}H_{22.14}$	1	4.1
O_2	12.3	50.47
N_2	11.07	45.4

Table 3.6: Combustion product of one kmole of fuel

Element	Numbers of kmoles	Mole fraction%
CO_2	12.3	3.74
H_2O	11.07	3.36
O_2	50.13	15.24
N_2	255.56	77.66

Table 3.7: Properties of flue gases exiting combustion chamber and entering turbine (T=1320K)

Element	h^{-} *(Mj/kmole)*	s_0^{-}*(Mj/kmole.k)*
CO_2	60.666	284.722
H_2O	49.707	244.564
O_2	42.753	253.325
N_2	40.853	237.353

Table 3.8: Properties of flue gases escaping to atmosphere due to chimney (T=431K)

Element	h^{-} *(Mj/kmole)*	s_0^{-} *(Mj/kmole.k)*
CO_2	14.670	0.228
H_2O	14.423	0.201
O_2	12.65	0.216
N_2	12.547	0.203

Table 3.9: Properties of flue gases at ambient condition T (308k)

Element	$\bar{h}$ (*Mj/kmole)*	$\bar{s_0}$ *(Mj/kmole.k)*
CO_2	9.732	0.215
H_2O	10.235	0.199
O_2	8.971	0.206
N_2	8.956	0.192

Table 3.10: Operation data in HRSG

Equipment	position	flue gases		steam	
		T (K)	p(bar)	T (K)	p(bar)
Diverter Damper	In	850	1.27	Non	
	Out	830	1.2		
HP.SH	In	830	1.2	559	70.5
	Out	761	1.18	738	70
HP.EV	In	761	1.18	559	70.5
	Out	568	1.1	559	70.5
HP.EC	In	568	1.1	337	82
	Out	463	1	552	70.5
LP.EV	In	463	1	142	3.9
	out	431	0.967	142	3.9

Table 3.11: Operation data for steam turbine

<table>
<tr><td colspan="2">Equipment</td><td>position</td><td>T(K)</td><td>P(bar)</td><td>m(kg/s)</td></tr>
<tr><td colspan="2" rowspan="2">Turbine</td><td>In</td><td>738</td><td>67</td><td>32</td></tr>
<tr><td>Out</td><td>321</td><td></td><td>32</td></tr>
<tr><td rowspan="4">condenser</td><td rowspan="2">Cooled water</td><td>In</td><td>321</td><td>0.115</td><td>32</td></tr>
<tr><td>out</td><td>321</td><td>0.115</td><td>32</td></tr>
<tr><td rowspan="2">Cooling water</td><td>in</td><td>299</td><td>2.4</td><td>2270</td></tr>
<tr><td>out</td><td>305</td><td>1.97</td><td>2270</td></tr>
</table>

CHAPTER FOUR

RESULTS AND DISCUSSIONS

This chapter is present all results achieved for Garri"2" as the follow. The exergy losses and efficiency for the components of the Garri"2" gas turbine cycle present by Figures (4.1) and (4.2) . The total exergy losses of the gas turbine plant are also shown. The results show that the greatest exergy loss in the gas turbine cycle takes place at the combustion chamber; because of chemical reaction and the large temperature difference between the burners and working fluid. That means its exergetic efficiency is less than other components. The turbine has a second major loss. Also, the results reveal that the compressor of the gas turbine has the largest exergy efficiency compared with the other gas turbine components. Figure (4.1) shows the exergy losses for all steam turbines component and appears that the greatest exergy losses in steam turbine cycle takes place at turbine unite due to irreversibilities and mechanical losses associated with transport power to electrical generator, and second exergy losses at HRSG; this required to calculate the exergy loss for each part of HRSG. For each element of HRSG, there are two inputs and two outputs. Inputs are hot flue gas and cold water and outputs are cold flue gas and hot water. The result is shown in Figure (4.4) that DE.DA has a largest exergy losses followed by HP.SH, HP.EV, HP.EC and LP.EV respectively. However, it does not mean that the heat transfer process in these parts is inefficient as well. In the other words, if one wishes to find the most irreversible heat transfer processes in the HRSG, one should also consider the amount of heat transfer that occurs in each part. Figure (4.3) also illustrate that considerable amount of exergy destroyed due to escape hot exhaust gases to ambient at $(158C^0)$ to avoid acid formation at the stack, and dissipate exergy at cooling systems to condense saturated steam that comes out from turbine.

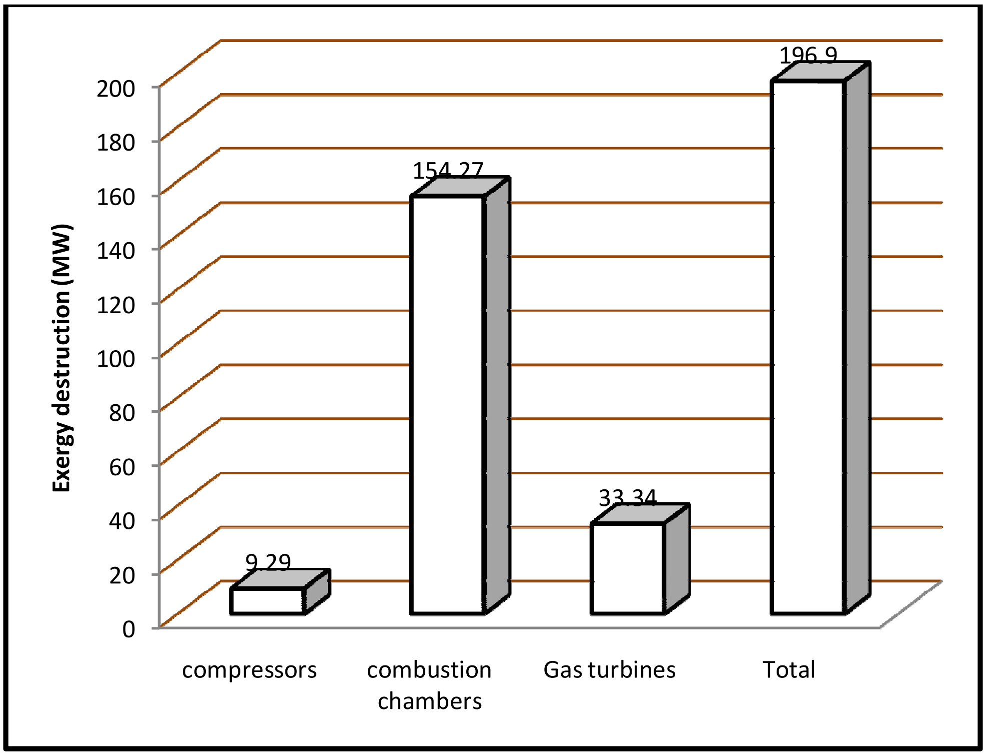


Figure4.1: Exergy destruction rate of whole gas turbines and its components'

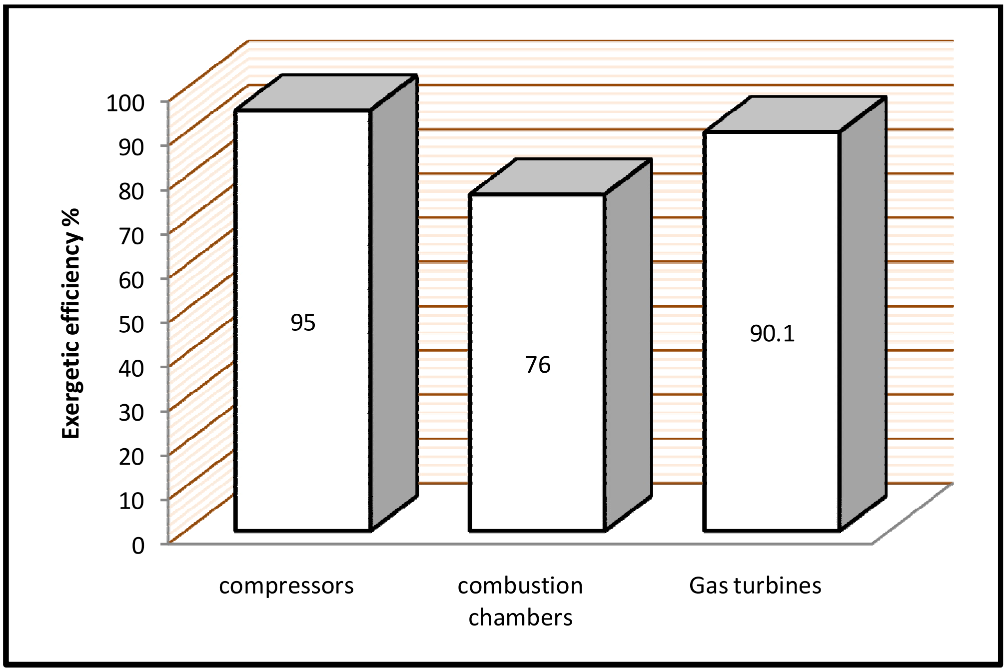


Figure 4.2: Exergetic efficiency of the gas turbine components

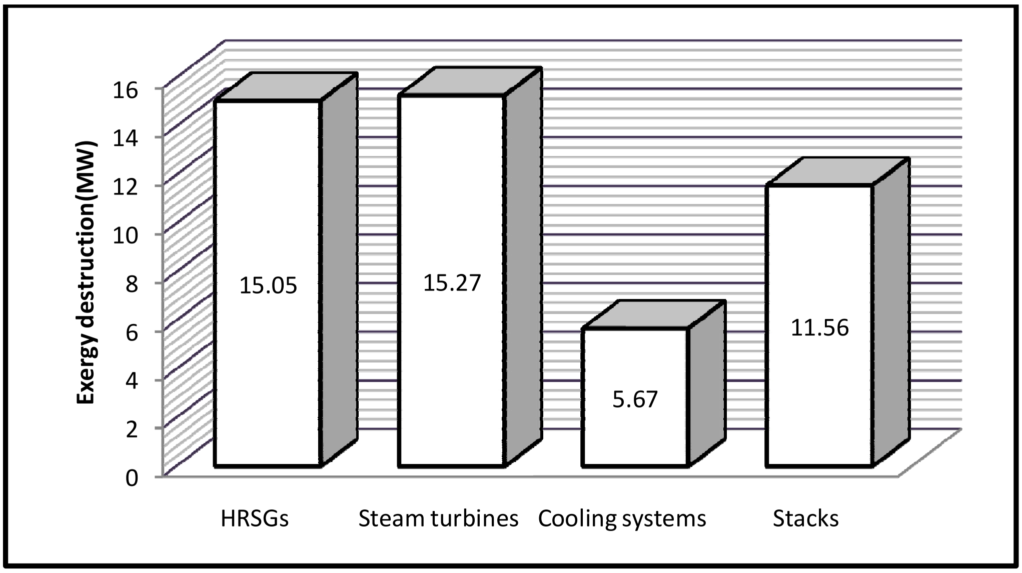


Figure 4.3: Exergy destruction rate of whole steam turbines components

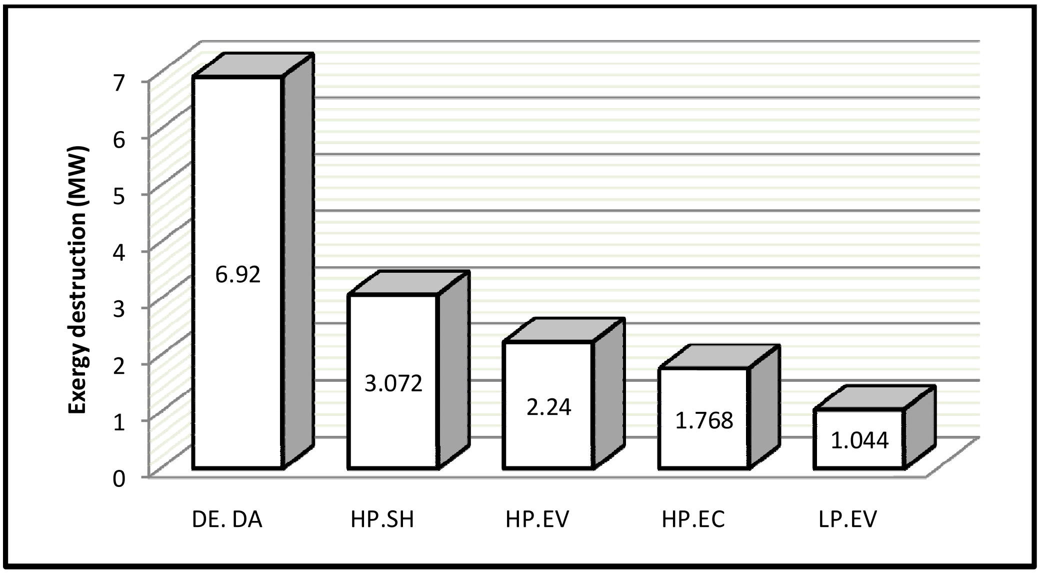


Figure 4.4: Exergy destruction rate due to HRSGs unites for whole plant

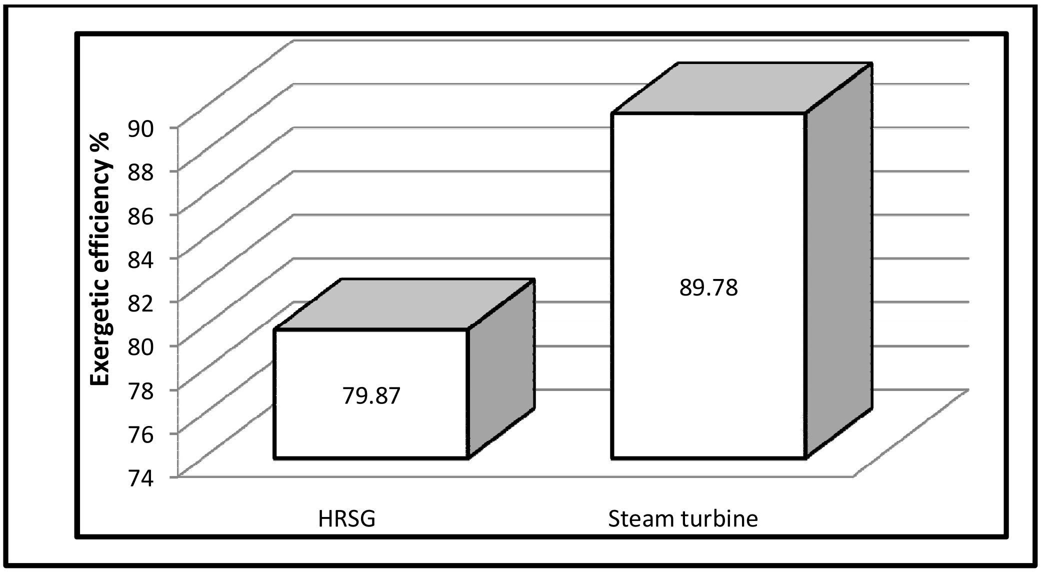


Figure 4.5: Exergetic efficiencies for main components of steam turbine cycle

Due to above analysis and figures we are knew efficient of each part and exergy destruction compared with others component. This text will present significance of equipment due to contribute each part on total exergy destruction for whole plant. Figure (4.6) illustrates exergy destruction for component, no matter about represent specified part on total exergy destruction. Figure (4.7) gives us more insight in analysis; because it present contributes ratios for whole plant. Note that from figure (4.2) exergy efficiency for gas turbine and combustion chamber (76%, 90%) respectively this not consider huge different and may be by misleading; whilst figure (4.7) show that 63% from total exergy destruction on combustion chambers, versus 13.6% for gas turbines. Figure (4.9), (4.8) show destruction ratio for Garri"2" combined cycle power plant versus similarly Neka combined cycle power plant (double pressure type) installed in Iran country in 1982. The result show that Gas turbines have an identical efficiency whilst compressors and combustion chambers for Garri"2" more efficient than Neka one. Figure also shows that the exergy destruction ratio for stacks and steam turbine close to identical but HRSGs and cooling systems are not. Exergetic efficiency for entire plant calculated by two ways (equations (2.17), (2.21)) and reveals (48%), (50%) respectively and could be takes (49%) as an average exergetic efficiency for Garri"2" versus exergetic efficiency (45.5%) for Neka. The result also shows that the thermal efficiency for Garri "2" is 38%. Table (4.1) illustrate summarize of exergy analysis for Garri"2" CCPP.

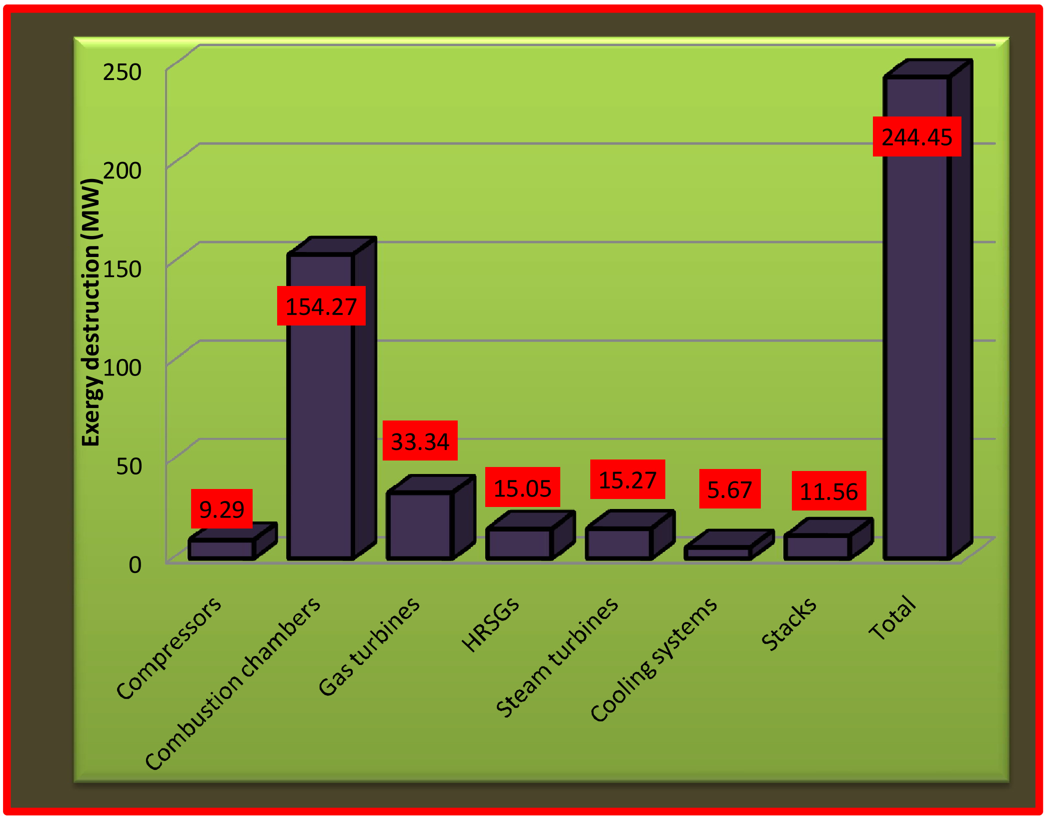


Figure 4.6: Exergy destruction rate for whole Garri"2" CCPP and its components'

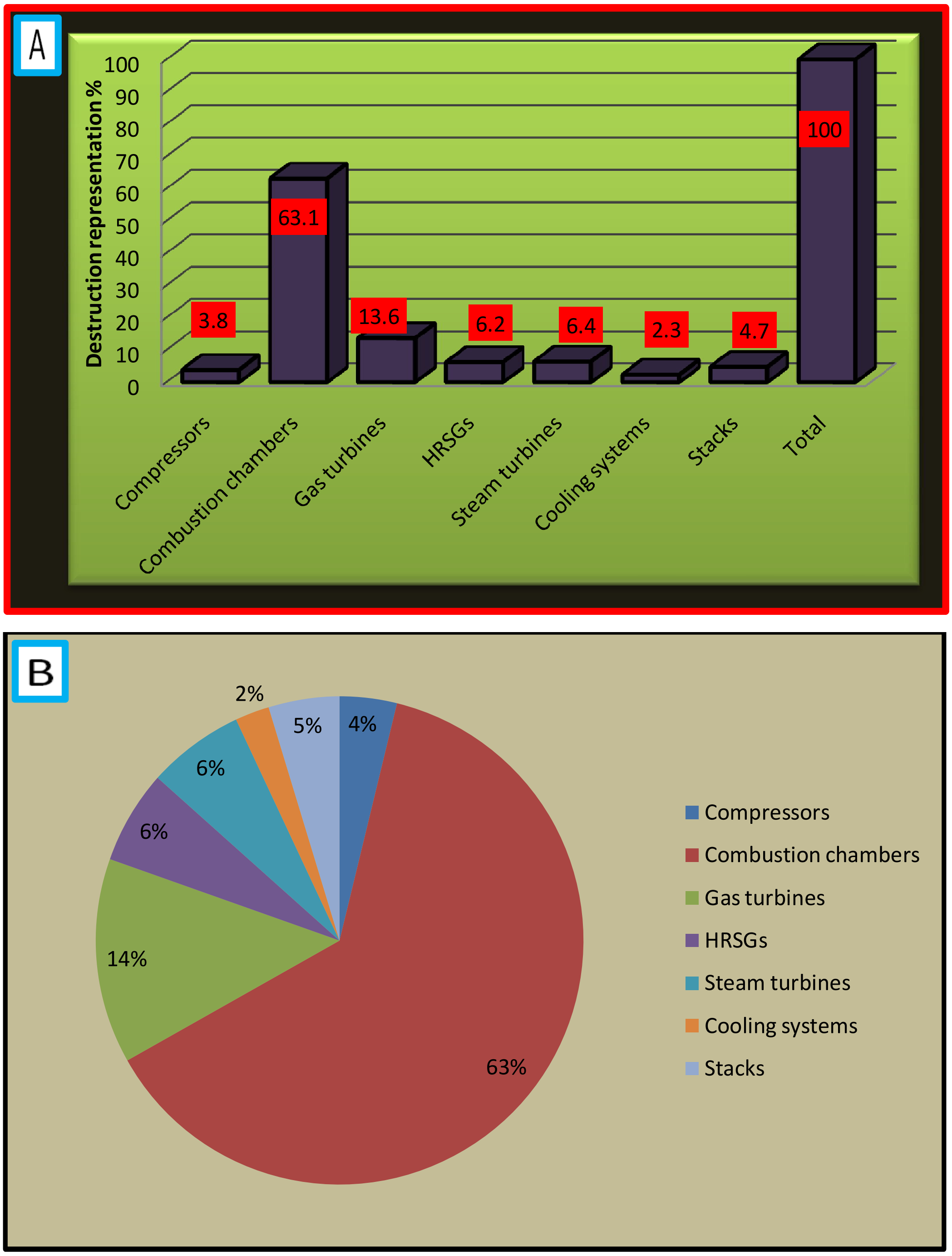


Figure 4.7 A &B: Contribute each part on exergy destruction

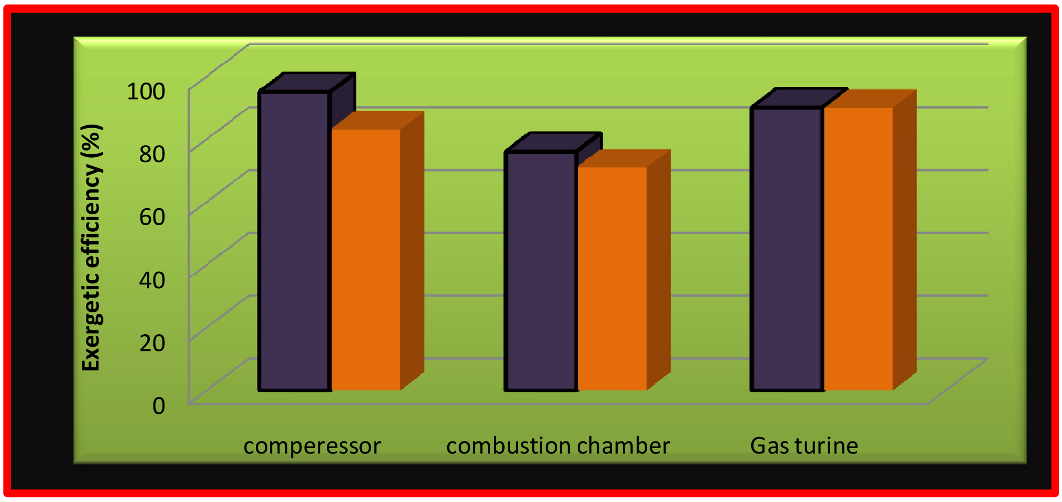
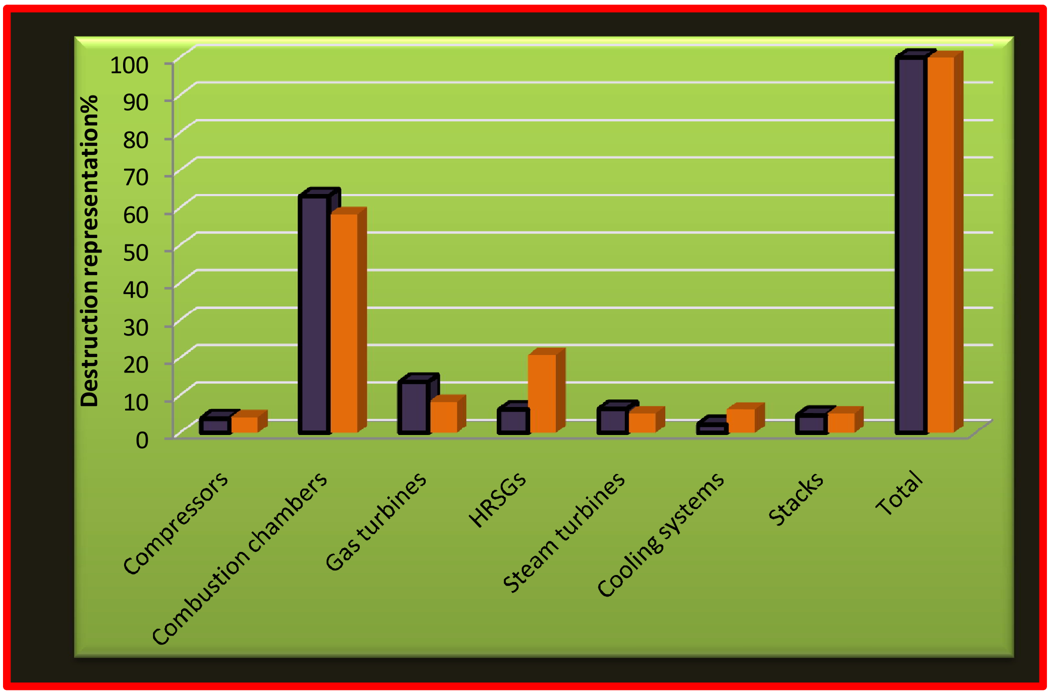


Figure 4.8: Exergetic efficiencies of Garri"2" and Neka gas turbine components Garri"2", Neka [10]



Figur4.9: Contribute each part on exergy destruction for both Garri"2" and Neka [10] CCPPs

Table 4.1: Summarize of exergy analysis for Garri”2” CCPP.

Equipment	Exergy destruction (MW)	Exergetic efficiency (%)	Representation destruction (%)
compressors	9.29	95	3.8
Combustion chambers	154.27	76	63.1
Gas turbines	33.34	90	13.6
HRSGs	15.1	89.8	6.2
Steam turbines	11.58	79.9	6.4
stacks	15.27	Non	2.3
Cooling systems	5.67	Non	4.7
Total plant	244.52	49	100

CHAPTER FIVE

CONCLUSION

&

RECOMMENDATIONS

5.1 Conclusion

Fired thermal power plant is one of the major power generations in Sudan country. This thesis study the exergy destruction (irreversibility) of each part for Garri"2" combined cycle power plant produced 180 MW – through the exergy analysis by using the performance data of this plant. The results show that the exergy efficiency of the combustion chamber is much lower than the efficiency of other combined cycle components due to its high irreversibility; represents 63% from total destruction for whole Garri"2". The second major exergy losses are in the gas and steam turbines respectively. Optimization of these equipments has an important role in reducing the exergy losses of total combined cycle. This plant achieved thermal and exergetic efficiencies (38%, 49%) respectively; that means 51% from entered exergy destroyed due to irrevesibilities and escape exergy to ambient and cooling tower.

5.2 Recommendations

from these results and discussions appear that the optimization of combustion chamber have an important role in reducing exergy destruction; this divided of two ways first one design of combustion chamber; this needed international company, and second one to manipulates on air fuel ratio since operation condition appear air fuel ratio about 4 times from Stoichiometric one with respect metallurgy issues. Effect of different fuel must be study in the light of exergoecnomic analysis. Effect of air pre-cooler on exergy destruction

Required, since it’s installed on Garri power plants by Sudanese engineers and achieved increase in output power. Exergy analyzed for Neka combined cycle power plant [10] achieved that when using supplementary firing on HRSG the output power increased by (7.38%) whilst exergy efficiency decreased by (1.5%). Since capacity of steam turbine about 40MW and produced about 31 MW now, Possibility of uses supplementary firing for Garri combined cycle power plants required; because Sudan country in present confront high shortage in total electrical energy produced.

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