

University of Bahrain

College of Engineering



**Department of Mechanical
Engineering**

**ENERGY AND EXERGY ANALYSIS OF A
COMBINED CYCLE POWER PLANT**

**A Thesis Submitted in Partial Fulfillment of the Requirements for the
Master of Science Degree in Mechanical Engineering**

Submitted by

Nedhal Ali Mohammed Ali Ahmed Yusuf

B.SC in Mechanical Engineering, University of Bahrain, 2004

Supervised by

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Ph. Dr. in Mechanical Engineering

University of Bahrain

Kingdom of Bahrain

October 2013

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DECLARATION

I declare that this work is the result of my own investigation and that it has not already been accepted in substance for any degree, nor is being currently submitted for any degree.

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Date : 23/10/2013

The Thesis defence committee considers the Thesis titled:

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Submitted by **Nedhal Ali Mohamed Ahmed Yousif (19990910)** to the Department of Mechanical Engineering satisfactory and acceptable for the Master of Science in Mechanical Engineering.

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DEDICATION

This thesis is dedicated to my family.

AKNOWLEDGMENT

It is a pleasure to thank those who made this thesis possible. To a few of them I offer my word of gratitude.

First, I would like to express my gratitude and sincere thanks to my thesis advisor, Prof. Ahmed Yousif, for his unwavering support and great assistance. I doubt that I would ever be able to convey my appreciation fully, but I owe him my external gratitude.

I would like to thank the entire Department of Mechanical Engineering which provided a friendly and warm environment.

I would also like to thank my whole family for the support and motivations they provided me through my entire life. I must acknowledge my wife without whose love, encouragement and support; I would not have reached this point and completed this thesis.

ABSTRACT (ENGLISH)

A block of 460 MW (two gas turbines and one steam turbine) combined cycle power plant has been modeled and simulated based on energy and exergy analysis approaches. The model equations have been solved by the constructed MATLAB computer program. The model accounts for the actual operating parameters within the block components: compressor, combustion chamber, gas turbine, HRSG and steam turbine. The developed program is utilized to investigate the effect of ambient temperature, compressor pressure ratio, as well part load operation, and heat losses in the combustion chamber on the block performance. To validate the present model, the gas turbine performances at different loads are calculated. The results of the present model are compared with the corresponding ones given by the block's manufacturer. The comparison has shown that the deviations are in the range of 3-5%. Exergy analysis is performed based on the first and second laws of thermodynamics. Its results have shown that the maximum exergy destruction is found to be in the combustion chamber at full and part loads.

The effects of ambient temperature variation on the compressor performance, gas turbine performance and fuel consumption are investigated. The results have shown that the cycle heat rate is found increasing with the net power output of the combined cycle decreases. The steam turbine power output significantly reduces when the ambient temperature is above 25 °C has been observed.

The effects of compressor pressure ratio variation on the compressor performance, gas turbine performance and fuel consumption are investigated. The results have shown

that the fuel consumption is started fallen down when the gas turbine is operated at higher compressor pressure ratio has been observed.

The effect of turbine inlet temperature variation on the gas turbine and fuel consumption are investigated. The results have shown that the fuel consumption has shot up rapidly at higher turbine inlet temperature. Therefore it is preferable to increase the compressor pressure ratio proportionally with turbine inlet temperature.

The effects of the percentage of heat losses within the combustion chamber on the fuel consumption, gas turbine cycle efficiency and turbine work net are investigated. The results have shown that the fuel mass flow rate and work net of the gas turbine are increasing as the heat losses increase. On the other hand the thermal efficiency decreases has been observed.

Exergy analysis is performed over wide range of ambient temperature, compressor pressure ratio, turbine inlet temperature and heat losses within combustion chamber variations. The results have shown that the combustion chamber has the largest exergy destruction in the system in all studied cases.

Method of gas turbine power enhancement which is based on air intake cooling to air compressor is discussed. The effect of spray water fogging system on the compressor performance, gas turbine performance and fuel consumption are investigated. The results have shown that the gas turbine power output increased simultaneously with the fuel consumption. In spite of power output increases, the results have shown that heat rate is dropped eventually when the spray water fogging system is employed.

ABSTRACT (ARABIC)

ملخص البحث

عنوان الأطروحة : تحليل الطاقة و الاضمحلالية للدورة المركبة لمحطة توليد الطاقة

اسم الطالب : نضال علي محمد علي أحمد يوسف

التخصص : الهندسة الميكانيكية السنة : يونيو 2013م

في هذه الرسالة تم عمل نموذج محاكاة، لمجموعة تعمل بالدورة المركبة قدرتها 460 ميغاوات (المجموعة تتكون من توربينين غازيين وتوربين بخاري)، يستند على نهج تحليل الطاقة والاضمحلالية، وقد تم حل معادلات النموذج مع عمل برنامج على الحاسب الآلي باستخدام برنامج مات لاب.

نموذج المحاكاة أخذ في الاعتبار متغيرات التشغيل الحقيقية داخل مكونات المجموعة: الضاغط، غرفة الاحتراق، التوربين البخاري. واستخدم البرنامج لمعرفة مدى تأثير درجة حرارة الهواء الجوي، نسبة الانضغاط، الأحمال الجزئية، الحرارة المفقودة من غرفة الاحتراق على أداء المجموعة.

وللتحقق من صحة النموذج الحالي وبرنامج الحاسب الآلي فقد تم عمل مقارنة بين نتائج النموذج الحالي للتوربين الغازي عند عمله على أحمال مختلفة مع النتائج الفعلية لنفس الأحمال والتي تم الحصول عليها من الشركة المصنعة للمجموعة. وبينت نتائج المقارنة أن الانحراف بين نتائج النموذج والنتائج الفعلية في حدود من 3 إلى 5%.

تم اجراء تحليل الاضمحلال مستنداً على قوانين الديناميكا الحرارية الأولى والثاني، وقد أظهرت النتائج أن أقصى اضمحلال للطاقة حدث في غرفة الاحتراق سواء عند تشغيل المجموعة عند الحمل الكلي أو الأحمال الجزئية.

لقد تم دراسة تأثير التغيير في درجة حرارة الجو المحيط على أداء الضاغط، وأداء التوربين الغازي ومدى استهلاك الوقود. وقد أظهرت النتائج أن معدل الحرارة المضافة للدورة المركبة تزداد بينما القدرة المولدة من الدورة المركبة تناقص مع زيادة درجة حرارة الجو. أما بالنسبة لقدرة التوربين البخاري فقد لوحظ حدوث نقص ملحوظ في القدرة المولدة منها عندما تصبح درجة حرارة الجو المحيط أعلى من 25 درجة مئوية.

تأثير تغيير نسبة الانضغاط للضاغط على أداء كل من الضاغط والتوربين الغازي واستهلاك الوقود تم التحقق منها. وقد أظهرت النتائج أن استهلاك الوقود بدأ في النقصان عندما يعمل الضاغط عند نسبة إنضغاط عالية.

التحقق من تأثير تباين درجة الحرارة عند مدخل التوربين الغازي على أداء التوربين واستهلاك الوقود. أظهرت النتائج أن استهلاك الوقود قد ارتفع سريعاً مع زيادة درجة الحرارة عند مدخل التوربين. لذا فإنه من الأفضل أن تتناسب نسبة الانضغاط مع درجة الحرارة عند مدخل التوربين.

التحقق من الآثار للحرارة المفقودة كنسبة مئوية داخل غرفة الاحتراق على كل من استهلاك الوقود، والكفاءة الحرارية للتوربين تم أجرائها. أظهرت النتائج أن معدل التدفق للوقود وإجمالي الشغل للتوربين الغازي قد زادت مع زيادة النسبة المئوية للحرارة المفقودة. من ناحية أخرى لوحظ انخفاض في الكفاءة الحرارية.

تم إجراء تحليل الاضمحالية خلال مدى واسع من درجة حرارة الجو المحيط، ونسبة الانضغاط في الضاغط، تغيير كل من درجة الحرارة عند مدخل التوربين الغازي والحرارة المفقودة في غرفة الاحتراق. وفي جميع الحالات التي تم دراستها أظهرت النتائج أن غرفة الاحتراق تؤدي إلى أكثر الاضمحلال.

وقد تم مناقشة كيفية تحسين القدرة المولدة من التوربين الغازي والذي تركز على تبريد كمية الهواء عند مدخل ضاغط الهواء للتوربين.

تأثير مياه الرش على شكل رذاذ على أداء كل من الضاغط والتوربين الغازي واستهلاك الوقود تم دراسته. وقد أظهرت النتائج أن القدرة المولدة من التوربين الغازي زادت في وقت واحد مع استهلاك الوقود، وعلى الرغم من الزيادة في القدرة الناتجة فإن النتائج أظهرت أن الحرارة المضافة تناقصت في نهاية المطاف عندما يستخدم رذاذ الماء المقطر.

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NOMENCLATURE

A_f	[m ²]	Surface area of finned tubes
A_T	[m ²]	Surface area of tube without fins
c_p	[kJ/kgK]	Specific heat at constant pressure
$c_{p,m}$	[kJ/kgK]	Specific heat at constant pressure of the mixture
c_v	[kJ/kgK]	Specific heat at constant volume
D_f	[m]	Outer diameter of the fins
D_r	[m]	External diameter of the tube at fin root
$\dot{E}x$	[MW]	Exergy flow rate
$\dot{E}x_d$	[MW]	Exergy destruction rate
h	[kJ/kg]	Specific enthalpy
h_m	[kJ/kg]	Specific enthalpy of the mixture
h_g	[kJ/kg]	Specific enthalpy of saturated vapor
h_v	[kJ/kg]	Specific enthalpy of water vapor
HL	[%]	Heat losses within combustion chamber
k_{air}	[kW/m.K]	Air thermal conductivity
LHV	[kJ/kg]	Lower heating value
L_t	[m]	Length of finned tubes
$\dot{m}$	[kg/s]	Mass flow rate
MW	[kg/kmol]	Molecular weight
N	[kmol]	Number of moles
n_{fan}	[-]	Number of fans
n_t	[-]	Number of finned tubes
$\overline{N_u}$	[-]	Nusselt number
P	[bar]	Pressure
P_r	[-]	Prandtl number
p_t	[m]	Tube pitch in plane perpendicular to air flow
$\dot{Q}$	[MW]	Heat energy transfer
R	[kJ/kgK]	Gas constant
Re	[-]	Reynolds number
r_c	[-]	Pressure ratio
s	[kJ/kgK]	Specific entropy

s_f	[m]	Fins pitch
S_{min}	[m ²]	Minimum air flow area
T	[K]	Temperature
u	[kJ/kg]	Internal energy
V_{max}	[m/s]	Maximum air flow velocity
v_{fan}	[m ³ /s]	Fan volume flow rate
$\dot{W}$	[MW]	Power
w_f	[m]	Fins thickness
y_d	[-]	Exergy destruction ratio
y_d^*	[-]	Exergy destruction rate
y_i	[-]	Fraction of i -component

Special characters

$\bar{\alpha}_{air}$	[kJ/m ²]	Heat transfer coefficient at air side
γ	[-]	Specific heat ratio
ε	[-]	Exergetic efficiency
η	[%]	Efficiency
λ	[%]	Percent of excess air
μ_{air}	[Pa.s]	Air dynamic viscosity
ρ_{air}	[kg/m ³]	Air density
ω	[-]	Absolute humidity
ϕ	[%]	Relative humidity
φ	[-]	Fuel air equivalence ratio

Subscripts

a	Air
c	Compressor
cc	Combustion chamber
dea	Deaerator
exh	Exhaust steam from steam turbine
ext	Extraction steam from steam turbine
HP	High pressure
LP	Low pressure
o	Ambient condition

p	Polytropic
t	Turbine
s	Steam
v	Vapor

ABBREVIATION

<i>HP</i>	[-]	High pressure
<i>IGV</i>	[-]	Inlet guide vane
<i>ISO</i>	[-]	International Organization for Standardization
<i>LP</i>	[-]	Low pressure
<i>TET</i>	[°C]	Turbine exhaust temperature
<i>TIT</i>	[°C]	Turbine inlet temperature
<i>GT</i>	[-]	Gas turbine
<i>CC</i>	[-]	Combustion chamber
<i>HRSG</i>	[-]	Heat recovery steam generator
<i>ACC</i>	[-]	Air cooled condenser
<i>HPFW</i>	[-]	High pressure feedwater
<i>LPFW</i>	[-]	Low pressure feedwater
<i>CEP</i>	[-]	Condensate extraction pump
<i>CRT</i>	[-]	Condensate recovery tank
<i>PP</i>	[-]	Pinch point
<i>VAN</i>	[-]	Turbine variable area nozzle
<i>VIGV</i>	[-]	Compressor Variable inlet guide vans
<i>VSV</i>	[-]	Variable stator vanes

CHAPTER 1

1 INTRODUCTION

1.1 INTRODUCTION

The gas turbine is the most versatile item of turbo machinery today. It can be used in several different modes in critical industries such as power generation, oil and gas, process plants, aviation, as well domestic and smaller related industries.

A gas turbine essentially brings together air, that it compresses in its compressor module, and fuel, to form a mixture which is then ignited. Resulting gases are expanded through a turbine. That turbine's shaft continues to rotate and drive the compressor which is attached on the same shaft, and operation continues. A separate starter unit is called prime mover is used to provide the first rotor motion, until the turbine's rotation is up to design speed and can keep the entire unit running. The work net, turbine work minus compressor work, can be utilized to produce electricity through a generator which is coupled on the same turbine shaft.

The use of combined cycle power plants has been increased recently due to their higher efficiency. Consequently, it has less impact on the environment and reasonable time for construction. The combined cycle power plant utilizes the waste heat that released from the gas turbine exhaust to produce steam through heat recovery steam generator. The improvement in material technology and operating control of gas turbines has led to achieve higher turbine inlet temperature and also higher compression pressure ratio that have influenced the efficiency of combined cycle power plant. In addition to the higher efficiency that reached, high input temperature and low output temperature can be achieved by combining both gas turbine and steam cycles which has led to more utilize of energy and reduce the side effects to the environment. A high thermal efficiency of combined cycle power plants can be reached by putting more attention on the bottoming cycle by optimizing its performance. The heat recovery steam generators

play crucial role in increasing the efficiency of the combined cycle. This can be achieved by choosing the best operating parameters such as temperature profiles, mass flow rate, operating pressures, number of pressure level, surface area, heat exchangers effectiveness, etc.

The needs of continuous save supply of the fossil fuel and the issue of global warming play a vital role in the development and use of the combined cycle power plant. As it has a better performance than a gas or steam turbine power plants. Furthermore studies have been performed in this field nowadays in order to improve the efficiency more and more. One method of measuring the performance of a power plant is based on the conservation of mass and energy principle, which is the first law of thermodynamic. This method, unfortunately, does not cover complete evaluation because it is not indicating how much the energy is utilized within the plant (Yilmazoglu & Amirbdein, 2007). The second method is the exergy analysis which bases on the second law of thermodynamic. It measures the quality of energy as well as its quantity (Nikulshin & Nikulshing, 2001), so it can guide how to utilize within the plant. The worldwide efforts are being directed to develop the gas turbine and combined cycle technology in order to meet the higher operational efficiency of power plants.

The exergy analysis of a combined power plant has been presented by previously by Ahmadi and Dincer (Ahmadi & Dincer, 2011) and Sanjay (Sanjay, 2011). The calculation of the exergy can show the main location of exergy losses, their origins and also, identifies the components or processes, where the highest destruction of exergy appears. This helps in finding away to reduce it and hence the performance will improve. Reddy and Mohamed (Reddy & Mohamed, 2007) analyzed a natural gas fired combined cycle power plant to investigate the effect of gas turbine inlet temperature and pressure ratio on the exergetic efficiency.

The present study is aimed to investigate the performance of a combined cycle power plant operates at full loads and part loads at different operating parameters. In the combined cycle plant the gas turbine is the top cycle and the steam turbine is the bottom cycle. The steam turbine output depends on the gas turbine load. There are two ways of changing the gas turbine geometry for controlling the load either by

compressor variable inlet guides vans (VIGVs) and turbine variable area nozzle (VAN) (Haglind, 2010). The use of VIGVs to control the part load has been presented previously (Baheta & Gilani, 2011) and (Song, et al., 2002). The advantage of VIGVs is to control the exhaust gas temperature of gas turbine so it does not exceed the allowable value. This is achieved by controlling the compressor inlet air flow during the part loads operation. Consequently, it does enhance the heat recovery for the bottom cycle (Song, et al., 2002).

In this work, the effect of ambient temperature, compressor pressure ratio and the turbine inlet temperature on the performance of the gas turbine and the whole combined power plant cycle are studied. In addition to that, the effect of the load on the ratio of mass flow rate of the steam to the mass of the exhaust gases where investigated. The effect of heat losses within the combustion chamber and pressure ratio on the gas turbine and block performance are reported. The Spray water fogging system as method of improving of gas turbine power output has been analysis and its effect on the performance of gas turbine and the whole combined cycle are reported.

Furthermore, the exergy losses at full and part loads in the compressor, combustion chamber, gas turbine, heat recovery steam generation, deaerator, feed water pumps, steam turbine, air cooled condenser, condensate pumps, ejector and feed water heater are analyzed. So, it become possible to identify where the largest quantity of exergy destruction occurs within the whole block.

1.2 THE OBJECTIVES OF THE THESIS

The objectives of this thesis are:

1. Study and analyze the gas turbine and the whole combined cycle power plant component performance based on first and second law of thermodynamics approach.
2. Design a MATLAB program to simulate the performance of the gas turbine and the combined cycle power plant while the operating condition is varying.

3. Investigate the effects of part loads operation of a gas turbine and its effect on the performance of a combined cycle power plant.
4. Investigate the effect of cooling intake air to gas turbine on the performance of the gas turbine as well as the whole combined cycle power plant.
5. Apply the exergy method which uses the conservation of mass and energy principles together with second law of thermodynamics, to identify the amount of losses of each component within the combined cycle power plant.
6. Compare the results with other works presented in literature review.

1.3 MAJOR CONTRIBUTION OF THE THESIS

In this work, the performance of gas turbine and combined cycle power plant at full load and part loads operation were studied. The effect of some of operating parameters such as ambient temperature, turbine inlet temperature, compressor pressure ratio and combustion chamber heat losses variations on the gas turbine and the whole combined cycle power plant performances were reported. Finally, the effect of cooling inlet air to the gas turbine was discussed. Then, the gas turbine and the combined cycle power plant performance enhancements were mentioned.

The simulation program has been constructed by using MATLAB software, the program is being used to analyze the performance of gas turbine and combined cycle power plant. The results have been validated with the available data given by manufacture. The results have shown that the deviations are in the range of 3-5% from the corresponding ones given by manufacture.

The part loads operation of gas turbine and combined cycle power plant are presented thoroughly and the effects of loads on the performance of the plant were reported.

The first law of thermodynamics and second law of thermodynamics were studied thoroughly at each component of the combined cycle power plant in order to evaluate the performance of gas turbine and the whole combined cycle power plant. Exergy analysis approach is performed for each component of the combined cycle power plant at various operating parameters variations such as ambient temperature, compressor

pressure ratio, heat losses within combustion chamber and turbine inlet temperature. The results have shown that the combustion chamber has the highest exergy destructions in all studied cases; the results are agreed with all other works which have been mentioned in the literature review.

The part load results from this thesis have been presented in the 9th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics (HEFAT2012), Malta July 2012. The paper “Modeling combined cycle performance at full and part load, Nedhal Ali and Ahmed Yousif, 9th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, 16-18 July, 2012” was published (Ali & Yousif, 2012).

1.4 ORGANIZATION OF THE THESIS

The thesis consists of five chapters altogether. The chapters are organized as follows:

Chapter 1

This chapter introduces a brief outline of the thesis problems; this chapter also includes the objective of the thesis and the major contribution in this thesis.

Chapter 2

This chapter introduces a brief outline to the literature review research related to the present work.

Chapter 3

This chapter briefly covers the thermodynamics properties of gas mixture, gas turbine modeling, combined cycle power plant modeling. Also the mathematical formulations of each component were discussed and analyzed. Finally, the chapter introduces also the main classical solution methods for the derived equations.

Chapter 4

This chapter includes the simulation results for six operating variables which varied over an appropriate range for each. Those variables are ambient temperature, turbine inlet temperature, compressor pressure ratio, heat losses within combustion chamber, gas turbine with intake air cooling system and gas turbine part loads operation. The performance of combined cycle power plant is studied base on first and second laws of thermodynamics.

Chapter 5

This chapter includes the conclusion and the recommendation for further work.

CHAPTER 2

2 LITERATURE REVIEW

2.1 INTRODUCTION

The following chapter is a review of the theory relating to the modeling of gas turbine as well as combined cycle power plant. Initially an overview of gas turbine modeling will be outlined to give a scope of the use of the proposed design tool within the overall engine design phase. Then, the modeling of combined power plant will be outlined and the effects of different operating parameters on the overall combined cycle power plant will be introduced. The performance of gas turbine and combined cycle part load was reviewed and analyzed based on the first and second laws of thermodynamic were reviewed. In addition to that an introduction of performance improvement of gas turbine such as compressor inlet air cooling will be provided as well as a brief overview of performance prediction will be introduced.

2.2 GAS TURBINES LITERATURE REVIEW

The performance of gas turbine has been analyzed and reported by Badran (Badran, 1999), Sa and Al Zubaidy (Sa & Al Zubaidy, 2011), Rahman and et al. (Rahman, et al., 2011). The studies were based on the first and second law of thermodynamic and it mainly focused on the performance of gas turbine while its design parameters or operating conditions are varying. These parameters are the compressor inlet temperature, compressor and turbine efficiencies, pressure ratio, turbine's inlet-temperature, and combustion efficiency.

The gas turbine performance was discussed by Badran (Badran, 1999), he studies the parameters that effects on the performance of the gas turbine and concentrates on the improvement of parameters concerned with performance of gas turbine. The

compressor inlet temperature, compressor and turbine efficiencies, pressure ratio, turbine's inlet-temperature, and combustion efficiency have been discussed individually and their contributions and effects on the plant performance have been shown also. The same investigations have been carried out by Al Zubaidy (Sa & Al Zubaidy, 2011) and Rahman (Rahman, et al., 2011).

Sa and Al Zubaidy (Sa & Al Zubaidy, 2011) have investigated the effect of ambient temperature variation from the *ISO* condition. An empirical relationship between the gas turbine's ability to generation power when it exposed to ambient condition that differs from the design condition has been developed. It has been observed that the gas turbine loses 1.47MW and 0.1% of its power output and thermal efficiency, respectively when the ambient temperature rises by 1 °C above *ISO* condition. Rahman and et al. (Rahman, et al., 2011) have done the same and besides the influence compression ratio on the gas turbine thermal efficiency for different turbine inlet temperature was discussed.

Rahman and et al. (Rahman, et al., 2011) have investigated the effects of the variation of operating condition on the performance of the gas turbine. Influence of compression ratio on the gas turbine thermal efficiency for different turbine inlet temperature was discussed and it is obviously can be seen that the thermal efficiency increases with compressor ratio at higher turbine inlet temperature. They have found the increase ambient temperature has a negative impact on the thermal efficiency due to the change of air density and compressor work since a lower ambient temperature leads to higher air density and lower compressor work as same was pointed by Badran (Badran, 1999) and Sa and Al Zubaidy (Sa & Al Zubaidy, 2011). The specific fuel consumption increases with increase of air to fuel ratio and ambient temperature has been shown in the results. The Peck thermal efficiency, power and specific fuel consumption occur at higher compression ratio at lower ambient temperature.

2.3 COMBINED CYCLE POWER PLANT LITERATURE REVIEW

Several studies have carried out thermodynamically to investigate the effect of operating condition such as ambient temperature, turbine inlet temperature, and HRSG number of operating pressure on the combined cycle power plant. The studies were based on the first and second law of thermodynamic and it mainly focused on the performance of combined cycle power plant while its parameters or operating conditions are varying. Sanjay (Sanjay, 2011) presents a second thermodynamic law analysis of gas-steam combined cycle. The effect of variation of cycle parameters such as turbine inlet temperature (TIT), compressor pressure ratio, and Heat recovery steam generator (HRSG) configuration on thermodynamic performance are presented and analyzed. The results show that the summation of total exergy destruction of all components is at lower value when the TIT is kept at higher value. The compressor pressure ratio also effects the summation of total exergy destruction of all components, and it is found that at higher value of compressor pressure ratio the exergy destruction is low. The sensitivity of fuel consumption has been studied, and it is observed that at fixed value of compressor pressure ratio when TIT is increased, the fuel consumption increases, but at fixed value of TIT when compressor pressure ratio is increased the fuel consumption decreases. The results show that exergy destruction is lower in reheated HRSG configuration with respect to the same without reheat. In addition to that it is observed that the exergy destruction of the HRSG is lower in case of tri-pressure configuration than the dual or single pressure configurations.

Arrieta and Lora (Arrieta & Lora, 2005) have studied the effect of ambient temperature on the combined cycle power plant. The paper states that the combined cycle power plant is influenced by the ambient conditions which are ambient temperature, atmospheric pressure and the air's relative humidity. They have shown that about two-thirds of the plant's total power output is sensitive to change of ambient conditions from those under *ISO* conditions. The paper is concentrated on the variation of ambient temperature as it has the highest significant impact on the power output from the combined cycle power plant. Finally, they have shown that the power loss in the gas turbine caused by a rise in the ambient temperature is compensated by the

supplementary-firing, for it is possible to generate more steam in the HRSG and consequently, more power output in the steam turbine.

Ahmadi and Dincer (Ahmadi & Dincer, 2011) have conducted a comprehensive dual pressure combined cycle power plant with supplementary firing modeling. The optimum operating parameters of the plant are found by applying a Generic Algorithm (GA) optimization method by defining objective function. The sum of the operating cost related to the fuel consumption and the capital investment for the equipment purchase and maintenance costs in terms of dollar per second is representing the objective function of the study. The design parameters of this study are selected as: compressor pressure ratio, compressor isentropic efficiency, gas turbine isentropic efficiency, gas turbine inlet temperature, and duct burner mass flow rate, high pressure steam, low pressure steam, HP main steam temperature, LP main steam temperature, HP pinch point temperature difference, LP pinch point temperature difference, condenser pressure, steam turbine isentropic efficiency and pump isentropic efficiency. The design parameters are selected in order to achieve the minimum value of the objective function. The sensitivity analysis is performed by taking two important factors which are unit cost of fuel and net output power of the combined power plant. As the fuel cost has the highest significant effect parameter in the objective function, the result investigates the effects of changing in the unit cost of fuel will to objective function and as well as the design parameters.

Polyzakis and et al. (Polyzakis, et al., 2008) have studied the performance of four different gas turbine cycles, namely, simple cycle, intercooled cycle, reheated cycle, and intercooled and reheated cycle with main objective the optimization of combined cycle power plant. Initially, each gas turbine cycle has been investigated independently, and the result shows that the reheated gas turbine is performed very well comparing with other three gas turbine cycles and it is the most attractive one. This is because of its high turbine exhaust gas temperature that resulting high thermal efficiency of the bottoming steam cycle. The next step, once the gas turbine is selected, is to investigate the design point and off design point performance of the combined steam cycle. The impact of the gas turbine on the combined steam cycle has been investigated. The

results show that the three cycles, simple cycle, intercooled cycle, reheated cycle are performing very similar during the off design point (part load operation). Finally, the optimum gas turbine cycle is defined as the reheated cycle. This is due to high exhaust gas temperature and high steam flow rate which is resulting in the steam cycle efficiency as well as the overall efficiency of the combined cycles.

2.4 GAS TURBINE AND COMBINED CYCLE POWER PLANT PART LOAD OPERATION LITERATURE REVIEW

As the ambient conditions are varied from time to time and the power consumptions are varied too, the ability of gas turbine and combined cycle power plant to generate power will vary too. Therefore, most of the time the gas turbine or combined cycle power plant will be operating at partial load or at loads low that its capacity. Hence, lots of efforts have been taken in account to estimate the performance of gas turbine and combined cycle power plant in order to define the optimal parameters that can be operated. Song and el al. (Song, et al., 2002) describes the exergy behavior of gas turbine at part load operating condition. The studied gas turbine is equipped with variable inlet guide van (VIGV) located in the front of the multi-stage compressor. The results show that the VIGV has a negative contribution on the exergy during low load, in spite of, its benefit to the heat recovery steam generators during the part load operating, which avoids the fall of turbine inlet or exhaust temperature. The results show that the exergetic efficiency of compressor and turbine are dropped during the part load operating while the combustor exergetic efficiency varies during part load within less than 2%.

Baheta and Gilani (Baheta & Gilani, 2011) investigate the exergy performance of gas turbine components at various loads. Baheta and Gilani (Baheta & Gilani, 2011) in their work, study a 4.2 MW gas turbine consists of variable stator vanes (VSVs). The VSVs are fully opened when the load is less 50% and started to close at load greater than 50%, so that it would increase the exhaust gas temperature, which is valuable for heat recovery steam generator. The results show that the compressor exergy destruction

is decreasing as the load increases and it reaches the lowest value when the load is equal to 50% and then a sudden increase in exergy destruction is observed at load greater than 50%. The compressor exergy destruction behavior is due to the re-staggering of VSVs. The same is noticed for the turbine but at different values. The combustion chamber exergy destruction is on the contrary to turbine and compressor exergy destruction. The combustion chamber exergy destruction is increasing as the load increases.

Variny and Mierka (Variny & Mierka, 2009) have studied the part load performance of a newly installed 80 MW combined cycle power plant in Slovakia with aim of fine the possibilities for fuel saving. Three approaches have been identified to achieve the goal of fuel saving, these approaches are condensate preheating by activation of the currently idle hot water section; change in steam condensing pressure regulation strategy; and the most important gas turbine inlet air preheating. The first approach is condensate preheating by activation of the currently idle hot water section; this will lead to reduce the amount of extracted steam flow rate from the LP steam turbine that is used to increase the deaerator temperature subsequently, increasing the steam turbine power output. The approach is accomplished by using a heat exchanger with a capacity of 4 MW capable to heat the condensate to 90 °C even in peak load operation. This method can be thereby applied for nearly 5 months in a year, which is still allows it to be economically attractive, saving resulting from this method is approximately 420,000 m^3 annual natural gas saving (5 month operation per year). The second approach is change in steam condensing pressure regulation strategy, the condensing pressure is planned to decrease from 12 kPa to 9 kPa, due to the climatic condition this possible can be achieved only during the six colder months. This corresponds to decrease in the condensing temperature from 49 °C to 44 °C which gives an overview of 180,000 m^3 annual natural gas saving (6 month operation per year). The last approach is gas turbine inlet air preheating. It is important to mention that, the yearly average ambient temperature in the area where the power plant is built is around 11 °C. By preheating the inlet air of gas turbine, the flue gas temperature at the exhaust of gas turbine and the HP steam generation rate in HRSG's were studied. It has found that increasing the temperature of inlet air of gas turbine has a positive effect on the HP steam generation.

However, the bottoming steam cycle efficiency increases during the part load due to the increases of exhaust flue gases temperature. It has been observed that, in part load operation, elevated ambient temperature has a positive effect on the overall electric efficiency of the studied power plant.

Kim (Kim, 2004) has studied the part load performance and design performance of different types of gas turbine power control strategy. The gas turbine operates at constant speed which is mostly used for power generation purpose however the load. Two methods of controlling the part load of the gas turbine were discussed and analyzed. The first is a control with varying fuel flow rate (fuel only control). The second is a control involving the modulation of air flow rate using inlet guide vane (IGV) and variable stator vanes (VSV) of the compressor. The advantage of the second method of controlling the part load of the gas turbine is enhancing the heat recovery performance by maintaining high exhaust gases temperature during the part load operation of gas turbine. Four different gas turbines with different design points have been analyzed. The part load performance behavior of four gas turbines represents low performance, mid-performance, current state-of-the-art high performance, under development and is to be launched in the future has been performed. The results show that the gas turbine with higher design efficiency practices less efficiency degradation regardless of the part load control strategy. Three part load control strategy were adopted in this work which are the maximum air flow control, inlet guide van control up to 30% of air flow reduction with constant turbine exhaust temperature followed by maximum air flow control, and inlet guide van control up to 15% of air flow reduction with constant turbine inlet temperature followed by maximum air flow control. The third method of part load control strategy practices less efficiency degradation in both gas turbine cycle and combined cycle.

2.5 GAS TURBINE WITH INLET AIR COOLING PERFORMANCE LITERATURE REVIEW

Several gas turbine compressor inlet air cooling technologies have been investigated analytically and/or numerically in the literature by many researches e.g. Sanaye and Tahani (Sanaye & Tahani, 2010), Kim and et al. (Kim, et al., 2012), Ehyaei and et al. (Ehyaei, et al., 2011), Mahto and Pal (Mahto & Pal, 2013), Bracco and et al. (Bracco, et al., 2007).

These technologies which have been used to cool the gas turbine compressor inlet air nowadays are thermal energy storage, mechanical cooling, absorption cooling and evaporative cooling. This literature focuses on the evaporative cooling; evaporative cooling is divided into two types either media coolers or high pressure fogging. The effects of evaporative cooling on gas turbine performance has been studied by Sanaye and Tahani (Sanaye & Tahani, 2010), they have concluded that evaporative cooling reduces the compressor inlet air temperature and compressor outlet temperature as well. The reduced compressor outlet temperature will have a positive impact on the turbine inlet cooling mass flow rate, with reduce temperature less mass flow rate will be required. Unfortunately, they found that the evaporative cooling has a negative impact on the bottoming cycle, if it is operating in combined cycle, because the turbine outlet temperature will reduce and it will lead to lower steam generation from the HRSG and finally affecting the steam turbine power output. Kim and et al. (Kim, et al., 2012) are investigating the effect of the amount of the water sprayed on the compressor inlet temperature, they found that the temperature drop is approximately proportional to the amount of sprayed water. Hence, the drop corresponding to the water injection ratio is expected to increase with increasing ambient temperature and decreasing the inlet relative.

Ehyaei and et al. (Ehyaei, et al., 2011) have investigated the effects of inlet fogging system on the first and second law efficiencies for existing power plant which is located Iran. It has seen that the average output power production, first and second low efficiencies are increased through three months of year (June, July and August) by 7%, 5.5% and 6% respectively. The fuel consumption was increased due to the reduction in

the compressor outlet temperature due to cool down inlet air. Also, the gas turbine power production increases due to increase air mass flow rate. To more accurate, increasing air density due to cool down the inlet air.

Mahto and Pal (Mahto & Pal, 2013) have carried out thermodynamics and thermo-economic analysis for five combined cycle configurations with inlet fogging, namely, simple gas turbine combined cycle using single pressure heat recovery steam generator (HRSG), that is, (SGTCC1P), simple gas turbine combined cycle using double pressure HRSG (SGTCC2P), simple gas turbine combined cycle using double pressure HRSG with reheat (SGTCC2PR), simple gas turbine combined cycle using triple pressure HRSG (SGTCC3P) and simple gas turbine combined cycle using triple pressure HRSG with reheat (SGTCC3PR). They have found that the cost/kWh of SGTCC3P configuration has least cost of electricity generation among the five configurations. Further, they found that ratio of coolant mass flow rate to gas flow rate increases with the increase in GTIT in the range of 1200-1800 K. This is due to the fact that with the increase in GTIT, the mean temperature of heat addition increases and requires more coolant to cool the GT blades.

Bracco and et al. (Bracco, et al., 2007) represent the maximum cycle efficiency gain due to the cooling of gas turbine intake air for ambient temperature varying between 10°C to 50°C. The increase in efficiency was found between 0.3% and 1.3%.

In the present work, a MATLAB program has been developed and used to solve the equations which are described the combined cycle power plant. The program is able to analyze the performance of gas turbine as well as the combined cycle power, the performance is analyzed based on the first and second law of thermodynamics. The MATLAB program is able to calculate the performance of combined cycle power plant at different operating condition as well as at different operating parameters. The effect of heat loss thru the combustion chamber on the performance of gas turbine was discussed in this work. The effect of ambient temperature, compressor pressure ratio, turbine inlet temperature, heat loss thru combustion chamber, part load operation and gas turbine with air intake cooling system have been studied thoroughly in wide range of variation.

CHAPTER 3

3 METHODOLOGY

3.1 INTRODUCTION

To predict the gas turbine performance accurately it requires the calculation of thermodynamic properties of different working fluids, namely, air and the products of combustion at different equivalence ratios. There are different methods of calculating the properties of the working fluids; namely:

- Use of property tables.
- By using ready-made polynomials.
- Use the concept of an ideal mixture of gases.

The property tables can be found in the most of the thermodynamic books, and its accuracy depends on the density of data stored in the table. The whole set of tables have to be created especially for combustion products, whenever the equivalence ratios are changed. And, for that reason, makes the use of property table in the analysis of the gas turbine performance to a certain extent hard. The use of ready-made polynomials is the flexible solution to calculate the thermodynamic properties of working fluids. Obviously, if the fuel composition is modified, the polynomial has to change too. Turns (Turns, 2000) defined a set of ready-made polynomials for different type of fuels over compound range (200-1000K). Binnewies and Milke (Binnewies & Milke, 2002) also provided a set of polynomials with the same characteristics. Finally, the thermodynamic concept of an ideal mixture gases is widely used in the gas turbine performance analysis Ahmadi and Dincer (Ahmadi & Dincer, 2011), Sanjay (Sanjay, 2011), Song and et al. (Song, et al., 2002). The ideal mixture of gases method requires two main milestones: a) the characterization of the mixture; and b) the calculation of properties for each component in the mixture. The characterization of air (dry or moist) poses no difficulty as it has a constant composition over a wide range of ambient

conditions (Turns, 2000) and (Binnewies & Milke, 2002). On the other hand, the composition of the combustion products depends on whether a complete or incomplete combustion is assumed. A simple mass balance between reactants and products is done when a complete combustion occurs. The working fluids properties are defined by assuming that it is a mixture of gases. The gases that comprise a mixture are often at high temperature and low pressure relative to the critical-point values of individual gases. In such cases the gas mixture and its components can be treated as ideal gases with negligible error.

In this work, the thermodynamics properties of different working fluid are calculated by using the concept of an ideal mixture of gases with ready-made polynomials equations.

3.2 THEORETICAL METHOD

To determine the properties of a mixture, the concept of thermodynamic ideal mixture of gases is implemented. In order to fulfill this assumption, the characterization of the mixture and the calculation of thermodynamic properties of every component of the mixture have to be done.

Section 3.2.1 gives the theoretical background behind the actual concept of an ideal mixture of gases, while Sections 3.2.2 and 3.2.3 cover the first milestone, the characterization of the mixture, for air (dry and moist) and combustion products, respectively. The decision to use the polynomials in the calculation properties of the mixture components was discussed in detail by Ahmadi and Dincer (Ahmadi & Dincer, 2011) and Sanjay (Sanjay, 2011).

In the present work, MATLAB program was developed and then used to calculate the properties of air and combustion products by knowing the mixture components. Section 3.6 introduced the MATLAB program that used to evaluate the thermodynamic properties.

3.2.1 THERMODYNAMIC PROPERTIES OF AN IDEAL MIXTURE OF GASES

The gas mixture and its components can be treated as ideal gases with negligible error because the gases comprise a mixture are often at high temperature and low pressure relative to the critical point value of individual gases (Cengel & Boles, 2011). In addition, properties like h , u , C_v and C_p of an ideal gas depend on the temperature only and are independent of the pressure or volume of the ideal-gas mixture.

Two different types of properties are existing: extensive and intensive. The extensive properties of a mixture of gases are the total contributions of all components of the mixture, in other words, it depends on the amount of the substance considered (mass or number of mole). For example, the total mass or moles of a mixture is the sum of the masses or moles of the individual components. Conversely, intensive properties of a mixture of gases are estimated by a weighted average of the properties of the components. The weighting factor is either the mass fraction or mole fraction of each component y_i , calculated by dividing the component mass or number of moles over the mixture total mass or moles. For example,

$$h_m = \sum_{i=1}^k y_i h_i \quad (1)$$

or,

$$C_{p,m} = \sum_{i=1}^k y_i C_{p,i} \quad (2)$$

Where h_m and $C_{p,m}$ are the enthalpy and specific heat at constant pressure of the mixture, respectively. Since the mass of a substance can be expressed in terms of moles N and molecular weight M as $m = NM$, the average molecular weight of the mixture M_m can be expressed as

$$M_m = \frac{m_m}{N_m} = \frac{\sum N_i M_i}{N_m} = \sum_{i=1}^k y_i M_i \quad (3)$$

It is clear that to calculate the properties of the mixture, the identification of every component and their concentration is required. The following two sections explain how

to characterize air (dry and moist) and the combustion products for different hydrocarbon based fuels.

3.2.2 CHARACTERIZATION OF DRY AND MOIST AIR

Air consists of a mixture of different elements including water vapor. Dry air is called when the water vapor percentage is nil. The concentrations of other elements in dry are as follow N₂, O₂, CO₂, and Ar. They are remained constant under different conditions. Consequently, the atmospheric air can be brought as the mixture of dry air and water vapor. Table 1 presents the volumetric percentage of dry air elements.

Table 1 Dry air elements

Elements	Volumetric percentage	Molecular weight (kg/kmol)
Nitrogen (N ₂)	78.084%	28.02
Oxygen (O ₂)	20.947%	32.0
Carbon dioxide (CO ₂)	0.035%	44.01
Argon (Ar)	0.934%	39.95

The water vapor enthalpy is a function of temperature only, $h = h(T)$ so the water vapor in the air can be considered an ideal gas without compromising in accuracy. The paper ‘Survey of experimental data and assessment of calculation methods of properties for the air-water mixture’ by X. Ji and et al. (Ji, et al., 2003) studies the thermodynamic properties of the air-water mixture at different temperature and pressure. They have reviewed comprehensive available experimental data and calculation methods for the air-water mixture. It is found that the available experiment data are limited and the determined temperature is within 75 °C. Three types of model (ideal model, ideal mixing model and real model) were carried out in order to assess the calculation of saturated vapor composition and enthalpy for air-water mixture. The conclusion shows that the ideal model can be safely used for low pressures up to 10 bar and temperatures above 280 K, and for high pressures (over 10 bar) only for temperatures over 360 K.

We approximate the enthalpy of water vapor to the enthalpy of saturated vapor for a particular temperature as,

$$\begin{aligned} h_v (T > 280k, low P) \\ h_v (T > 360k, high P) \end{aligned} = h_g(T) \quad (4)$$

The amount of water vapor in air can be specified as an absolute or relative number. The absolute ω value represents the amount of vapor (m_v) per unit of mass of dry air (m_a),

$$\omega = \frac{m_v}{m_a} \quad (5)$$

There is a limit of water vapor the air can hold before any additional moisture starts to condensate. This limit is a function of water saturation pressure (P_g) at a particular temperature. The relative humidity is defined as the ratio of actual vapor mass (m_v) to the maximum amount of vapor air can hold the same temperature,

$$\phi = \frac{m_v}{m_g} \quad (6)$$

Using the ideal gas general equation $PV = mRT$ and combining Eq.5 and Eq.6, the two quantities can be expressed as function of the other,

$$\phi = \frac{\omega P}{(0.622 + \omega)P_g} \quad (7)$$

$$\omega = \frac{0.622 \cdot \phi P_g}{P - \phi P_g} \quad (8)$$

Where P is the total pressure of air, and P_g is the water vapor saturation pressure at a particular temperature. In most practical applications, the amount of dry air remains constant while the water vapors fraction changes.

3.2.3 CHARACTERIZATION OF COMBUSTION PRODUCTS

Combustion is defined as a chemical reaction takes place during which a fuel is oxidized and large quantity of energy is released. This energy is utilized to produce

power. Combustion products thermodynamic properties can be calculated through the characterization of mixture. To obtain the final composition of the combustion products, the mass balance to every particular element in the reaction is to be applied. That is, the total mass of an element in the products has to be equal to the total mass of the same element in the reactants. Two types of combustion process either complete or incomplete.

The complete combustion is defined as process when the reactant burns in oxygen, producing a limited number of products (depends on fuel compositions) such as CO_2 , H_2O , N_2 and SO_2 . The incomplete combustion is occurred when there is not enough oxygen to all fuel to react completely to produce carbon dioxide, water and other components such as C, H_2 , CO, OH or NO_3 and NO_2 . After the bulk of the combustion products are formed, the levels of energy in the mixture decrease making it difficult to maintain the same rate of formation for complex molecules. Therefore, more simple compounds such as CO, OH, NO_x start forming. However, the reversibility nature of these reactions initiates a large number of chemical reactions before reaching a thermodynamic balance. It is important prior to design any combustion process is to define the minimum amount of air for complete oxidation of a fuel which is called stoichiometric or theoretical air.

Any additional air present during the combustion process is called excess air. Theoretically, the excess air will not react during the combustion hence it will be found as part of the products unchanged. The fuel has to bring above its ignition temperature to start the combustion. Moreover, the proportions of the fuel and air must be in proper range in order to establish the combustion and get continue. The chemical equation of the combustion process has to be applied and balanced on the basis of the conservation of mass principle. That is, the total mass of each element on the right-hand side of the reaction equation (the products) must be equal to the total mass of that element on the left-hand side (the reactants) even though the elements exist in different chemical compounds in the reactants and products. Also, the total number of atoms of each element is conserved during a chemical reaction since the total number of atoms is equal to the total mass of the element divided by its atomic mass.

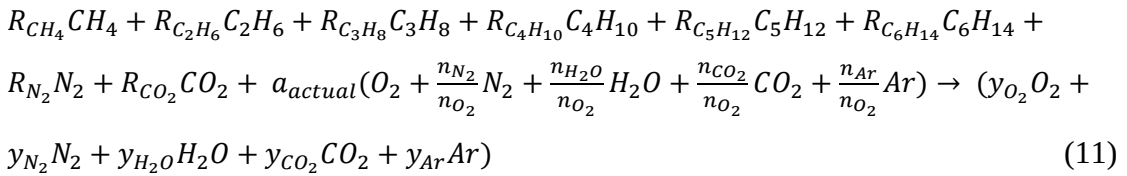
Once the chemical equation is balanced, the fuel-to-air ratio (*FAR*) can be quantified as the ratio of the mass of fuel to the mass of air for a combustion process:

$$FAR = \frac{m_{fuel}}{m_{air}} \quad (9)$$

The ratio of the actual *FAR* to the stoichiometric *FAR* is called the equivalence ratio (Φ). The percentage excess air in a reaction can be expressed in terms of equivalence ratio:

$$excess\ air\ (\%) = \lambda = \frac{100}{\Phi} - 100 \quad (10)$$

In most gas turbine engine applications, the combustion processes are designed to take place in excess air (100% to 250%) to maintain the flame temperature below the mechanical limits of the materials used and to induce a more complete combustion. As a consequence, the composition and properties of the combustion products are closer to those of air than to pure combustion products. For hydrocarbon-based fuels, assuming a complete combustion with no dissociation, there is a general formula to estimate the stoichiometric moles number of air (oxidizer) and the main combustion products.



Above equation can only be used when the chemical composition of the fuel is exact. Any fractional or decimal representation of any fuel is indication of an average of the different types of the same fuel. For example, natural gas chemical composition varies according to geological and geographical characteristics of the reservoir. Therefore, a chromatography analysis of a natural gas sample must be performed before attempting to apply a theoretical model to determine the composition of its combustion products, even if the combustion is considered to be completed.

3.3 THE PRESENTED COMBINED CYCLE POWER PLANT

The system presented in this work is 460 MW combined cycle power plant (two gas turbines and one steam turbine) and details of the system components are shown in Figure 1. The plant is consisted of two identical ALSTOM 13E2 gas turbines (GT) and one KA13E2-2 ALSTOM steam turbine (ST).

The exhaust gas from each gas turbine is diverted to heat recovery steam generator (HRSG). The generated steam from the two HRSG's enters steam turbine (ST). The steam cycle utilizes regenerative feed water heating system. Feed water heating is mainly carried out in Low pressure pre-heater heat exchanger (LP Pre-heater) along with deareator serves as an open feed water heater. The feed water is supplied to HRSGs at two levels of pressure which are high pressure (HP) and low pressure (LP). High pressure feed water pumps (HPFWPs) and low pressure feed water pumps (LPFWPs), respectively are used to feed water.

The HP and LP steam are generated by HRSGs and used to drive the steam turbine. The turbine exhaust is sent to the air cooled condenser (ACC) and the condensate to the condensate recovery tank (CRT). Finally, the condensate is circulated through the condensate extraction pumps (CEPs). Then, the cycle starts over again.

The performance of the given combined cycle power plant as shown in Figure 1 has been modeled and simulated based on energy and exergy analysis approaches.

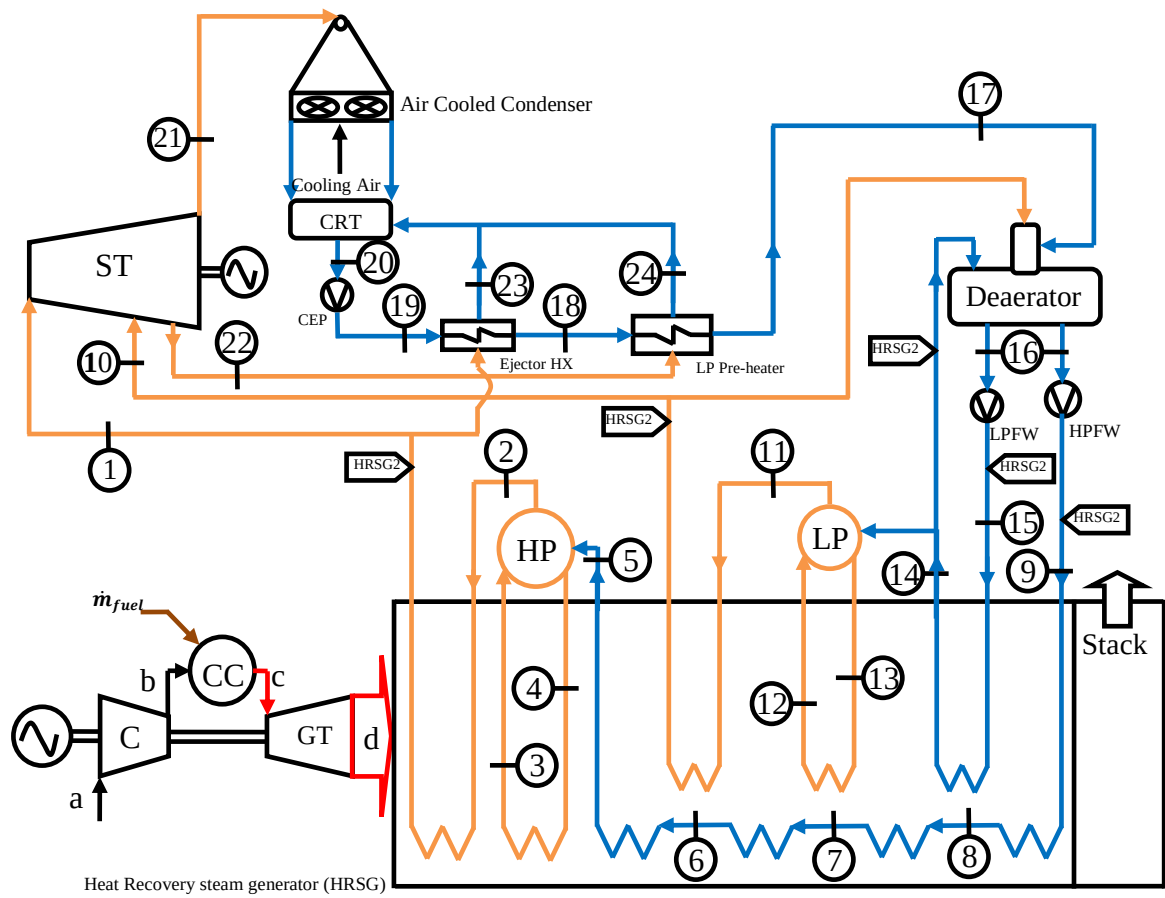


Figure 1 Schematic diagram of the combined power plant

3.4 ENERGY ANALYSIS

The energy concept comes from the first law of thermodynamics which is defined as the conservation of energy. The energy can be identified by two modes which are heat and work. The conservation of energy states that energy interaction at the system boundaries by matter flow, by heat and by work is equal to the change of the energy of the system.

The net change (increase or decrease) in the total energy of the system during a process is equal to the difference between the total energy entering and the total energy leaving the system during that process (Cengel & Boles, 2011).

$$\left(\begin{array}{c} \text{Total energy} \\ \text{entering the system} \end{array} \right) - \left(\begin{array}{c} \text{Total energy} \\ \text{leaving the system} \end{array} \right) = \left(\begin{array}{c} \text{Change in the total} \\ \text{energy of the system} \end{array} \right)$$

or,

$$E_{in} - E_{out} = \Delta E_{system} \quad (12)$$

This relation is often referred to as the energy balance and is applicable to any kind of system undergoing any kind of process by assuming that the flow is steady and uniform.

3.4.1 AIR COMPRESSOR MODEL

The air compressor plays an important role in the gas turbine power plant. It is responsible for providing combustion chamber with all air it needs to accomplish the combustion continuously. It has been found that the compressor is almost taking around 60% of the total power that generates in the gas turbine, so lots of research and efforts were carried out to increase the performance of the air compressor in order to operate it in efficient manner.

Early the air compressor units had pressure ratios of around 5 and required about 10 stages. Over the years the pressure ratios have increased significantly and some turbines have pressure ratios exceeding 30.

This increase in the pressure ratios follows by reduction in the fuel consumption as the air leaving the compressor its temperature raises when the pressure ratios are increased. Therefore, the amount of fuel, that requires achieving the design firing temperature, is decreased.

The atmospheric air is drawn and compressed in axial-flow multi stage compressor providing the high pressure, high volume air to the combustor. The compression process is assumed to be polytropic process taking polytropic efficiency ($\eta_{p,c}$) as 90%. The compressor discharge temperature (T_b) is calculated as a function of inlet air

temperature (T_a), compressor pressure ratio (r_c) and polytropic efficiency, by the Eq.13 taken from (Cohen, et al., 2001):

$$T_b = T_a \cdot (r_c)^{(\gamma-1)/\gamma\eta_{p,c}} \quad (13)$$

Where η_c is the isentropic efficiency and is calculated from (Cohen, et al., 2001):

$$\eta_c = \frac{1-r_c^{(\gamma-1)/\gamma}}{1-r_c^{(\gamma-1)/\gamma\eta_{p,c}}} \quad (14)$$

The energy balance of the air compressor is described as follow:

$$-\dot{W}_c = \dot{m}_{air}(h(T_b) - h(T_a)) \quad (15)$$

The design point performance model of gas turbines is measured when the turbine is operated at the design condition. The gas turbine is designed at *ISO* condition where the ambient temperature, pressure and relative humidity are 15° C, 1.013bar and 60% respectively.

The off-design model of gas turbines uses the parameters from *ISO* full-load model with modifications to the input parameters (ambient temperature, pressure and relative humidity, as well as exhaust pressure modification (backpressure)).

The pressure ratio of the compressor is allowed to vary with the ambient temperature so it is important to know the change of pressure ratio during the off-design performance prior to calculate the gas turbine performance as it has significant value on the gas turbine output and efficiency.

Bracco and et al. (Bracco, et al., 2007) describe the effect of ambient temperature changes on the compressor pressure ratio. This effect can be seen in Eq.16:

$$r_c = r_{c_{ISO}} \cdot \frac{T_{ISO}}{T_o} \quad (16)$$

Where $r_{c_{ISO}}$ is the compressor pressure ratio at the *ISO* conditions and T_o is the ambient temperature. From the equation the pressure ratio, it is noticed that compressor pressure ratio is decreased as the ambient temperature becomes higher than the *ISO* conditions.

3.4.2 COMBUSTION CHAMBER MODEL

Combustion chamber is the place where the fuel is burnt and produced a huge amount of energy; this energy is in the form of heat. The combustion chamber is providing the heat input to the gas turbine. The combustion chamber is a direct-fired air heater in which fuel is burnt almost stoichiometrically or lean combustion with one third or less of the compressed air that delivers from air compressor. Combustion products are the mixed with the remaining air which delivers from air compressor to reach the desire turbine inlet temperature.

There are many types of combustors, but the three major types are tubular, tubo-annular, and annular. Despite the many design differences, all gas turbine combustion chamber have three features: (1) a recirculation zone, (2) a burning zone (with a recirculation zone which extends to the dilution region), and (3) a dilution zone.

The function of the recirculation zone is to evaporate, partly burn, and prepare the fuel for rapid combustion within the remainder of the burning zone. Ideally, at the end of the burning zone, all fuel should be burnt so that the function of the dilution zone is solely to mix the hot gas with the dilution air. The mixture leaving the combustion chamber should have a temperature and velocity distribution acceptable to the guide vanes and turbine (Boyce, 2006).

The mass and energy balance of the combustion chamber for the given turbine inlet temperature (TIT) is expressed by the following equations correspondingly by assuming that the flow is steady and uniform:

$$\dot{m}_{air} + \dot{m}_{fuel} = \dot{m}_{gas} \quad (17)$$

$$\dot{m}_{fuel} LHV \eta_{cc} (1 - HL) + \dot{m}_{air} h_b = \dot{m}_{gas} h_c \quad (18)$$

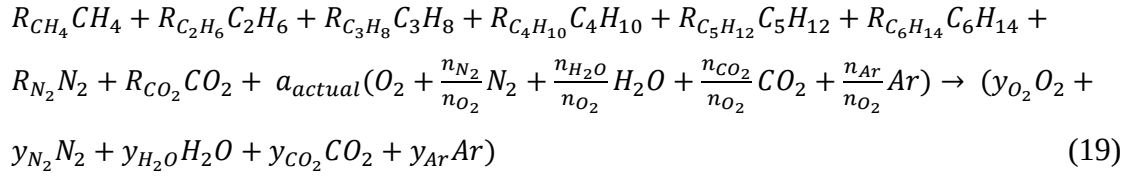
The mass flow rate of fuel ($\dot{m}_{fuel}$) which is required to achieve the desire TIT is obtained from combustion reaction equations, for the used gas fuel which is Khuff gas (Local fuel). Its compositions are given in Table 2. The analysis of the fuel is supplied from Bapco (The local company supplies the fuel to all the power plants in Bahrain).

Table 2 The used fuel composition and its lower heating values

Compositions	LHV (kJ/kg)	Mole %
Methane, CH ₄	50,016	79.49
Ethane, C ₂ H ₆	47,489	1.52
Propane, C ₃ H ₈	46,357	0.36
N-Butane, C ₄ H ₁₀	45,742	0.17
N-Pentane, C ₅ H ₁₂	45,355	0.09
Hexane Plus, C ₆ H ₁₄	45,105	0.25
Nitrogen, N ₂	-	11.74
Carbon Dioxide, CO ₂	-	6.38
Total	34471*	100

*The lower heating value of the fuel is lower than what is reported in the literature for the natural gas because the Khuff gas has lower energy content than a normal natural gas.

The reaction equations are:



Where $N_{air} = (1 + \frac{n_{N_2}}{n_{O_2}} + \frac{n_{H_2O}}{n_{O_2}} + \frac{n_{CO_2}}{n_{O_2}} + \frac{n_{Ar}}{n_{O_2}})$ is the number of air moles.

The equivalence fuel air ratio φ is defined as:

$$\varphi = \frac{1}{1+\lambda} \quad (20)$$

Where λ is percent excess air.

The fuel air ratio is calculated as follow:

$$(FAR)_{actual} = \varphi \cdot (FAR)_{stoic} \quad (21)$$

$$a_{actual} = \frac{MW_{fuel}}{MW_{air}} \cdot \frac{1}{N_{air}} \cdot \frac{1}{(FAR)_{actual}} \quad (22)$$

The different species molar fractions are calculated as following:

$$y_{CO_2} = R_{CH_4} + 2R_{C_2H_6} + 3R_{C_3H_8} + 4R_{C_4H_{10}} + 5R_{C_5H_{12}} + 6R_{C_6H_{14}} + R_{CO_2} + a_{actual} \cdot \frac{n_{CO_2}}{n_{O_2}} \quad (23)$$

$$y_{H_2O} = 2R_{CH_4} + 3R_{C_2H_6} + 4R_{C_3H_8} + 5R_{C_4H_{10}} + 6R_{C_5H_{12}} + 7R_{C_6H_{14}} + a_{actual} \cdot \frac{n_{H_2O}}{n_{O_2}} \quad (24)$$

$$y_{N_2} = R_{N_2} + a_{actual} \cdot \frac{n_{N_2}}{n_{O_2}} \quad (25)$$

$$y_{Ar} = a_{actual} \cdot \frac{n_{Ar}}{n_{O_2}} \quad (26)$$

$$y_{O_2} = 2R_{CO_2} + a_{actual} \cdot \left(2 + 2 \frac{n_{CO_2}}{n_{O_2}} + \frac{n_{H_2O}}{n_{O_2}}\right) - 2y_{CO_2} - y_{H_2O} \quad (27)$$

The a_{stoich} is the stoichiometric quantity of oxidizer, and it is calculated from:

$$a_{stoich} = 2R_{CH_4} + 3.5R_{C_2H_6} + 5R_{C_3H_8} + 6.5R_{C_4H_{10}} + 8R_{C_5H_{12}} + 9.5R_{C_6H_{14}} \quad (28)$$

$$(FAR)_{stoich} = \frac{MW_{fuel}}{MW_{air}} \cdot \frac{1}{N_{air}} \cdot \frac{1}{a_{stoich}} \quad (29)$$

3.4.3 GAS TURBINE MODEL

The gas turbine is a power plant that generates great amount of power for its size and weight however during the last 25 years the gas turbine has found increasing.

The growth of the gas turbine in recent years has been brought about most significantly by three factors (Giampaolo, 2006):

- Metallurgical advance that have made possible the employment of high temperatures in the combustor and turbine components.
- The cumulative background of aerodynamic and thermodynamic knowledge.
- The utilization of computer technology in the design and simulation of turbine airfoils and combustor and turbine blade cooling configurations.

The two factors which most affect high turbine efficiencies are the pressure ratio and turbine inlet temperature. By achieving these factors; ensuring that the gas turbine will

perform high performance of power and efficiency. These turbines which are operating at very high pressure ratio and very high firing temperature are pushing the envelope of technology in the areas of aerodynamics and material science to their limit. The last 30 years has seen a large growth in gas turbine technology. The growth is organized by the growth of material technology, new coatings, and new cooling schemes. This with the conjunction of increase in compressor pressure ratio has increased the gas turbine thermal efficiency from about 15% to over 45%.

Higher performance gas turbines are preferable to operate in combined cycle mode because it releases hot gases at high temperature which can be utilized to generate steam through HRSG and rolling up the steam turbine. The high performance gas turbines which are exceeding efficiencies of over 45% are the basis of the growth of combined cycle power plant; these turbines in a combined cycle mode reach overall plant efficiencies of nearly 60%. The hot gases leave the combustion chamber are expanded in turbine. The expansion process is assumed to be polytropic process taking polytropic efficiency as 95%. The turbine exhaust temperature (TET) has been calculated from inlet turbine temperature (TIT), expansion pressure ratio ($\frac{P_d}{P_c}$) and turbine polytropic efficiency ($\eta_{p,t}$), by Eq.30:

$$TET = TIT \cdot \left(\frac{P_d}{P_c}\right)^{(\gamma-1) \cdot \eta_{p,t} / \gamma} \quad (30)$$

Where is (η_t) the turbine isentropic efficiency and calculated from (Cohen, et al., 2001):

$$\eta_t = \frac{1 - \left(\frac{P_d}{P_c}\right)^{(\gamma-1) \cdot \eta_{p,t} / \gamma}}{1 - \left(\frac{P_d}{P_c}\right)^{(\gamma-1) / \gamma}} \quad (31)$$

The energy balance of the gas turbine is described as follow:

$$\dot{W}_t = \dot{m}_{gas}(h(TIT) - h(TET)) \quad (32)$$

The work net from the gas turbine is equal to:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_c \quad (33)$$

3.4.4 HEAT RECOVERY STEAM GENERATOR (HRSG) MODEL

Heat recovery steam generator, also known as waste heat recovery boilers, it plays an essential part of chemical plants, refineries, power plants and process systems because it is utilized the waste heat from the process to generate steam which can be used to drive steam turbine or used in a process (Cogeneration). The HRSGs consist of three major components: the economizer, evaporator, and superheater.

The HRSGs are categorized into vertical and horizontal types. In vertical type HRSGs, exhaust gases flow vertically over the horizontal tubes but in horizontal type HRSGs, exhaust gases flow horizontally over the vertical types. The main criterion to classify the HRSG types is the way the water flow through the heat exchanger of the boiler takes place, this water flow across the boiler is called the circulation. There exist basically four types: the natural circulation type, the forced circulation type, the assisted circulation type, and the once through type. The natural circulation arises from the density difference between water and steam. The forced circulation arises from the work done by water circulation pump. The assisted circulation takes place from the work done by water circulation pump that is only service during the start-up of the HRSG. The once through circulation takes place from the weight force of the water entering the boiler at a very high point and in this type there is no recirculation taking place in the evaporator. All mentioned types of circulation normally have drum except the once through type. The HRSGs performance has a large impact on the overall performance of the combined cycle power plant; the HRSGs have several factors that influence it performance. These factors were reported by Ong'iro and et al (Ong'iro, et al., 1977):

- The gas and water flow rates, temperatures, pressure and gas compositions which vary with fuel type.
- Water and air temperature change due to diurnal and seasonal temperature changes.
- Fouling of the heat-transfer surface which vary with time and lead to significant changes in heat-transfer rate and pressure loss.
- The geometry of the HRSG.

- Fluctuating conditions at the gas turbine exhaust, e.g. gas velocity and temperature.

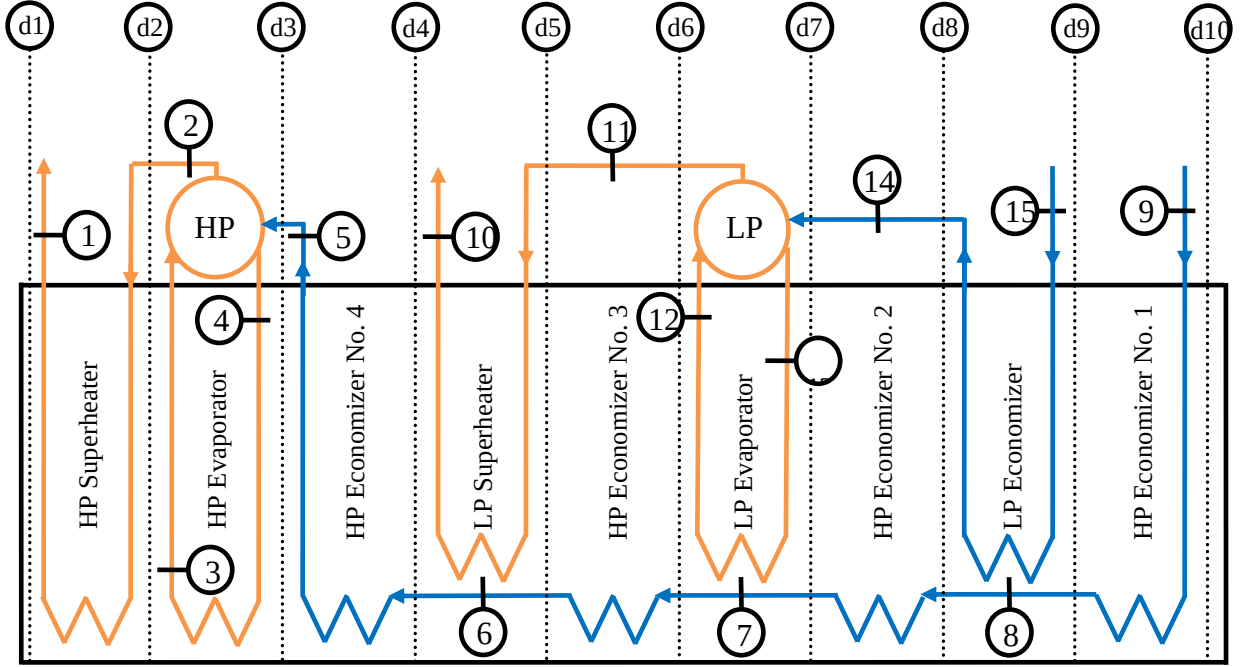


Figure 2 Details of heat recovery steam generator

The HRSG is one of the most important components of the combine cycle power plant that significantly affect the efficiency of the whole plant, hence more efforts have been taken nowadays to optimize the operating parameters such as pinch point (temperature difference between the gas and steam) and steam pressure in the HRSG in order to achieve maximum efficiency and extract the maximum allowable energy from the exhaust gases to enhance the efficiency of the combined cycle power plant. A dual pressure HRSG shown in Figure 2 is analyzed by applying the mass and energy balance equation for water and gas at each part of HRSG.

The temperature of gas entering the HP evaporator can be calculated from the pinch point temperature of HP evaporator (PP_{HP}) and HP drum temperature (T_4), therefore the temperature of gas entering the HP evaporator (T_{d3}) can be written as follow:

$$T_{d3} = T_4 + PP_{HP} \quad (34)$$

The same can be applied for the gas entering the LP evaporator; the temperature can be calculated as follow:

$$T_{d7} = T_{14} + PP_{LP} \quad (35)$$

Where T_{14} and PP_{LP} are LP drum and LP evaporator pinch point temperatures respectively. The energy balance equation was applied at each section of the HRSG as given below:

High-pressure superheater:

$$\dot{m}_{s,HP}(h_1 - h_2) = \eta_{HRSG} \dot{m}_{gas}(h_{d1} - h_{d2}) \quad (36)$$

High-pressure evaporator:

$$\dot{m}_{s,HP}(h_3 - h_4) = \eta_{HRSG} \dot{m}_{gas}(h_{d2} - h_{d3}) \quad (37)$$

High-pressure economizer No. 4:

$$\dot{m}_{s,HP}(h_5 - h_6) = \eta_{HRSG} \dot{m}_{gas}(h_{d3} - h_{d4}) \quad (38)$$

Low-pressure superheater:

$$\dot{m}_{s,LP}(h_{10} - h_{11}) = \eta_{HRSG} \dot{m}_{gas}(h_{d4} - h_{d5}) \quad (39)$$

High-pressure economizer No. 3:

$$\dot{m}_{s,HP}(h_6 - h_7) = \eta_{HRSG} \dot{m}_{gas}(h_{d5} - h_{d6}) \quad (40)$$

Low-pressure evaporator:

$$\dot{m}_{s,LP}(h_{12} - h_{13}) = \eta_{HRSG} \dot{m}_{gas}(h_{d6} - h_{d7}) \quad (41)$$

High-pressure economizer No. 2:

$$\dot{m}_{s,HP}(h_7 - h_8) = \eta_{HRSG} \dot{m}_{gas}(h_{d7} - h_{d8}) \quad (42)$$

Low-pressure economizer:

$$(\dot{m}_{s,LP} + \dot{m}_{dea,LP})(h_{14} - h_{15}) = \eta_{HRSG} \dot{m}_{gas}(h_{d8} - h_{d9}) \quad (43)$$

High-pressure economizer No. 1:

$$\dot{m}_{s,HP}(h_8 - h_9) = \eta_{HRSG} \dot{m}_{gas}(h_{d9} - h_{d10}) \quad (44)$$

3.4.5 STEAM TURBINE MODEL

The steam turbine is defined as a form of heat engine in which the power is occurred as resultant of transferring the steam into kinetic energy by means of expansion through nozzles and this kinetic energy of the resulting jet is in turn converted into force doing work on rings of blades mounted on rotating part. The steam turbine operates on the Rankine Cycle.

The steam turbine work is calculated from the energy balance equation as described in the following:

$$(2\dot{m}_{s,HP} - \dot{m}_{ejector}) \cdot h_1 + (2\dot{m}_{s,LP} - \dot{m}_{dea,st}) \cdot h_{10} - \dot{m}_{ext} \cdot h_{22} - \dot{m}_{exh,ST} \cdot h_{21} = W_{ST} \quad (45)$$

3.4.5.1 IMPLEMENTATION OF PART-LOAD OPERATION:

Sliding pressure part-load controls typically used in combined cycle steam turbine operate on master-slave scheme where live steam pressures (p), temperature (T) and mass flow ($\dot{m}$) are linked by the following constant flow function (K) as described in Eq.46 (Haglund, 2010):

$$K = \frac{\dot{m}\sqrt{T}}{\sqrt{p}} \quad (46)$$

It is usual that, whenever possible, live steam temperature be held constant at its rated value in order to achieve maximum efficiency while mass flow and pressure vary proportionally. However, at low loads, typically lower than 50%, pressure is maintained at a minimum value by means of a control valve which used to modify mass flow independently of pressure.

Depending on the part-load pattern of gas turbine exhaust temperature, it might also be necessary to decrease the steam temperature below certain load. This can be achieved either by de-superheating the live steam stream or just by increasing the mass flow rate of feedwater to the steam generator as shown in the work of de Escalona and et al. (de Escalona, et al., 2012).

3.4.6 DEAERATOR MODEL

Deaerator is equipment which is used for the removal of oxygen and other dissolved gases from the feedwater to the steam generating boiler. Deaerators can be considered as open feedwater heaters; that serve the dual purpose of heating the feedwater to improve the overall plant efficiency and deaerating the feedwater to remove gases that could cause corrosion of equipment and piping system. Deaerators also provide the storage of high-quality feedwater for the boiler feed pump.

Heat transfer in deaerators is by direct contact between the feedwater and the turbine extraction steam, and various design techniques are used such as bubbling, tray, spray, or various combinations of these.

Satisfactory boiler operation depends on the heating and conditioning of the feedwater. The impurities in water affect boiler performance; these impurities can be either suspended or dissolved matter and consist of atmospheric gases (O_2 , N_2 , and CO_2).

Oxygen (O_2) is also present in feedwater, and this gas can be removed by mechanical and chemical processes. The mechanical means of removing oxygen is by use of a deaerator where the feedwater interacts with steam and oxygen is released to the atmosphere at the deaerator vent. The rest of the oxygen, which is capable of causing severe corrosion, is scavenged chemically using chemical compounds such as hydrazine.

The energy balance of the deaerator is described as follow:

$$2\dot{m}_{dea,LP} \cdot h_{14} + \dot{m}_{dea,st} \cdot h_{10} + (2 \cdot (\dot{m}_{s,HP} + \dot{m}_{s,LP}) + \dot{m}_{dea,LP}) \cdot h_{17} = 2 \cdot \{\dot{m}_{s,HP} + \dot{m}_{s,LP} + \dot{m}_{dea,LP}\} \cdot h_{16} \quad (47)$$

Where $h_{17} = h_f(T_{dea})$

3.4.7 LP PRE-HEATER MODEL

LP pre-heater is a type of closed feedwater heater and it is basically U-tube heat exchanger. The feedwater enters the lower portion of the heater through the inlet nozzle and proceeds to the feedwater outlet nozzle through the horizontal U-tubes in basically a two-pass system. Steam from extraction stages in the steam turbine enters at the top of the heater and flows in a counter direction, also in a two-pass flow arrangement. Then flows over the tubes where it condenses and leaves the feedwater heater through the drain outlet nozzle.

Feedwater heaters have two primary functions in power plants: (1) to provide the means for increasing the feedwater temperature, which improves the overall plant efficiency and (2) to minimize the thermal effects in the boiler.

The number and type of feedwater heaters used depend on the steam cycle, the steam operating pressure, turbine capacity, and the plant economics. The lower operating costs can offset the additional capital cost expenditure. In general, the number is between 5 to 8 heaters, usually the smaller plants have fewer units.

The energy balance of the LP pre-heater is described as follow:

$$(2 \cdot (\dot{m}_{s,HP} + \dot{m}_{s,LP}) + \dot{m}_{dea,LP}) \cdot (h_{17} - h_{18}) = \dot{m}_{ext} \cdot (h_{22} - h_{24}) \quad (48)$$

3.4.8 EJECTOR HEAT EXCHANGER MODEL

The ejector can be classified according to the type of pump that uses the venture effect of a converging-diverging nozzle. In this process the nozzle converts the pressure energy of a motive steam to velocity energy which creates a low pressure zone that draws in and entrains a suction fluid which to be evacuated. The ejector has two inlets: one to admit the motive steam and the other to admit the gas/vapor mixture which to be evacuated usually from the condenser.

The motive steam at high pressure is extracted from the boiler superheater output and then the steam pressure reduces by the ejector motive to the steam design pressure. The motive steam under high pressure enters convergent-divergent nozzle, the pressure is being converted to high jet-velocity at the throat of the convergent-divergent nozzle which creates a low pressure at that point. The gas/vapor mixture which to be evacuated enters at that point, to the throat of the convergent-divergent nozzle, where the pressure is low. The low pressure draws the suction fluid into the convergent-divergent nozzle where it mixes with the motive steam.

The mixture of motive steam and gas/vapor mixture leaves the ejector and enters the heat exchanger where the remaining steam is condensed and exchange the heat with the feedwater. There will be a negligible increase in the feedwater temperature after leaving the ejector heat exchanger however the steam can be recovered instead of venting to atmosphere.

The energy balance of the ejector heat exchanger is described as follow:

$$(2.(\dot{m}_{s,HP} + \dot{m}_{s,LP}) + \dot{m}_{dea,LP}).(h_{18} - h_{19}) = \dot{m}_{ejector}.(h_1 - h_{23}) \quad (49)$$

3.4.9 AIR COOLED CONDENSER MODEL

Dry-cooling or direct air cooled condenser has come in picture due to the decreasing availability and rising of cost of cooling water. The air cooled condensers are employed to reject heat to the environment in modern power plant integrating steam turbines.

Unfortunately, the effectiveness of these cooling system decreases with increasing in the ambient temperature which resulting in a corresponding reduction in steam turbine efficiency and output during hot ambient temperature. In Bahrain, the ambient temperature can go to 48 °C.

In the present work, the condenser is an air cooled condenser. It is basically a cross flow heat exchanger consists of four “A” roof type streets of finned tubes elements. The axial flow fans locate underneath the roof; force the air through the fins. The numbers of operating fans are depended on the steam back pressure, ambient temperature as well as the steam mass flow rate.

The steam temperature is assumed to be equal the tube surface temperature i.e. the effect of tube thickness is negligible. Therefore, the condensate steam temperature can be found as follow (Hewitt, et al., 1994):

$$T_{ACC,steam} = T_{air,in} + \Delta T_{LM} \left(\frac{A_{ACC} \cdot \bar{\alpha}_{air}}{\dot{m}_{air,ACC} \cdot c_{p,air}} \right) / \left\{ 1 - \exp \left(\frac{-A_{ACC} \cdot \bar{\alpha}_{air}}{\dot{m}_{air,ACC} \cdot c_{p,air}} \right) \right\} \quad (50)$$

The pressure of air cooled condenser is the saturated pressure at $T_{ACC,steam}$.

Where, the heat transfer coefficient at air side ($\bar{\alpha}_{air}$) can be calculated as follow (Hewitt, et al., 1994):

$$\bar{\alpha}_{air} = \frac{\overline{N_u} \cdot k_{air}}{D_r} \quad (51)$$

Where D_r is the external diameter of the tube at fin root.

The recommended correlation Nusselt number ($\overline{N_u}$) for in-line, high-fin tube arrays taken from (Hewitt, et al., 1994) and is given by Eq.55:

$$\overline{N_u} = 0.269 Re^{0.625} \left(\frac{A_f}{A_T} \right)^{-0.375} P_r^{0.333} \quad (52)$$

Where P_r is the air Prandtl number as a function of air temperature T and is given by (Turns, 2000):

$$P_r(T) = 1.2 \times 10^{-6} \cdot T^2 - 9.2 \times 10^{-4} \cdot T + 0.875 \quad (53)$$

The surface area of the finned tubes (n_t) of length (L_t) is given by Eq.54:

$$A_f = \frac{\pi n_t L_t}{(s_f + w_f)} \left\{ \frac{1}{2} (D_f^2 - D_r^2) + D_f w_f + D_r s_f \right\} \quad (54)$$

Where s_f , w_f , and D_f are fins pitch, fins thickness, and outer diameter of fins, respectively.

The total tube surface area without fins is:

$$A_T = \pi n_t L_t D_r \quad (55)$$

The Air thermal conductivity (k_{air}) is a function of air temperature T and is given by (Turns, 2000):

$$k_{air}(T) = 3.13908 \times 10^{-9} \cdot T^3 - 3.40606 \times 10^{-5} \cdot T^2 + 9.78225 \times 10^{-2} \cdot T - 0.10903 \quad (56)$$

The Reynolds number (Re) is calculated as follow:

$$Re = \frac{V_{max} D_r \rho_{air}}{\mu_{air}} \quad (57)$$

Where μ_{air} is air dynamic viscosity at air temperature T and it is given by (Turns, 2000):

$$\mu_{air}(T) = \mu_o \frac{(T_o + C_o)}{(T + C_o)} \cdot \left(\frac{T_o}{T} \right)^{1.5} \quad (58)$$

Where T_o , C_o and μ_o are constant and equal to 291.15 K, 120 K and $18.27 \times 10^{-6} Pa.s$, respectively.

The maximum air flow velocity (V_{max}) taken from (Hewitt, et al., 1994) and is given by Eq.59:

$$V_{max} = \frac{\dot{m}_{air,ACC}}{\rho_{air} S_{min}} \quad (59)$$

The minimum air flow area (S_{min}) for most cross-flow heat exchanger, is perpendicular to the flow direction taken from (Hewitt, et al., 1994) and is given by Eq.60:

$$S_{min} = n_t L_t (p_t - D_r - \frac{2w_f h_f}{(w_f + s_f)}) \quad (60)$$

Where p_t is tubes pitch in plane perpendicular to flow.

The total axial flow fans mass flow rate is calculated as follows:

$$\dot{m}_{air,ACC} = n_{fan} \cdot v_{fan} \cdot \rho_{air} \quad (61)$$

Where n_{fan} , v_{fan} , and ρ_{air} are number of fans, fan volume flow rate, and air density, respectively.

The energy balance of the ACC is described as follow:

$$\dot{m}_{air,ACC} (T_{air,ACC_{out}} - T_{air,ACC_{in}}) = \dot{m}_{exh,ST} \cdot (h_{21} - h_{20}) \quad (62)$$

3.5 EXERGY ANALYSIS

Exergy is defined as the maximum theoretical useful work obtained as the system interacts with the environment state without violating any thermodynamic laws (Cengel & Boles, 2011).

In the present work, the dead state condition (15 °C, 101.3 kPa and 60% relative humidity. The chemical composition and mole fraction of one mole of dry is N_2 , O_2 , CO_2 and Ar : 78.084%, 20.947%, 0.035% and 0.934% respectively) is chosen for as thermo-mechanical and chemical equilibrium condition for the exergy analysis.

3.5.1 MECHANICAL AND CHEMICAL EXERGY

For a control volume the specific exergy has been calculated by the following equation (Moran & Shapiro, 2008) is given by Eq.63:

$$\dot{Ex}_Q + \sum_i \dot{m}_i \cdot ex_i = \sum_e \dot{m}_e \cdot ex_e + \dot{Ex}_W + \dot{Ex}_d \quad (63)$$

Where the subscripts i and e are the specific exergy of control volume inlet and exit, and $\dot{E}x_d$ is the exergy destruction. Where $\dot{E}x_Q$ and $\dot{E}x_W$ are exergy of heat transfer and work respectively which cross the boundaries.

$$\dot{E}x_Q = \sum_j (1 - \frac{T_o}{T_j}) \cdot \dot{Q}_j \quad (64)$$

$$\dot{E}x_W = \dot{W} \quad (65)$$

The total exergy rate ($\dot{E}x$) is divided into four main components which define as follow:

$$\dot{E}x = \dot{E}x_{ph} + \dot{E}x_{ch} + \dot{E}x_{potential} + \dot{E}x_{kinetic} \quad (66)$$

$$\dot{E}x = \dot{m} \cdot ex \quad (67)$$

In the present work, the potential ($\dot{E}x_{potential}$) and kinetic ($\dot{E}x_{kinetic}$) exergy are assumed to be negligible as the elevation and speed changes influences are negligible. The other two components $\dot{E}x_{ph}$ and $\dot{E}x_{ch}$ are physical and chemical exergy respectively.

$$\dot{E}x_{ph} = \dot{m} \cdot [(h - h_o) - T_o(s - s_o)] \quad (68)$$

The mixture of chemical exergy is defined as follows:

$$\dot{E}x_{mix}^{ch} = \dot{m} \cdot (\sum_{i=1}^n x_i \cdot ex_i^{ch} + RT_o \sum_{i=1}^n x_i \cdot \ln x_i) \quad (69)$$

Where x_i is mole fraction of a species.

3.5.2 FUEL EXERGY

The fuel exergy is a combination of physical ($\dot{E}x_{ph}$) and chemical ($\dot{E}x_{mix}^{ch}$) exergy, and they are taken from the work of Adam and et al. (Adam, et al., 2011):

$$\dot{E}x_{ph} = \dot{m}_{fuel} \left[c_p^{-h} (T - T_o) - T_o \left\{ c_p^{-s} \ln\left(\frac{T}{T_o}\right) - R \ln\left(\frac{P}{P_o}\right) \right\} \right] \quad (70)$$

Where,

$$c_p^{-h} = \frac{1}{T-T_o} \int_{T_o}^T c_p dT \text{ and } c_p^{-s} = \frac{1}{\ln(\frac{T}{T_o})} \int_{T_o}^T c_p \frac{dT}{T}$$

c_p^{-h} and c_p^{-s} are mean molar isobaric exergy capacities for evaluating enthalpy and entropy changes.

$$\dot{E}x_{mix}^{ch} = \dot{m}_{fuel} [\sum_{i=1}^n x_i \cdot ex_i^{ch} + RT_o \sum_{i=1}^n x_i \cdot \ln x_i] \quad (71)$$

$$\dot{E}x_{fuel} = \dot{E}x_{ph} + \dot{E}x_{mix}^{ch} \quad (72)$$

3.5.3 EXERGY DESTRUCTION

The exergy destruction of the k^{th} component in the system can be written as the difference between fuel ($\dot{E}x_{f,k}$) and product ($\dot{E}x_{p,k}$) exergy of the k th component in the system as proposed by Abusoglu and Kanoglu (Abusoglu & Kanoglu, 2009):

$$\dot{E}x_{d,k} = \dot{E}x_{f,k} - \dot{E}x_{p,k} \quad (73)$$

In addition to exergy destruction $\dot{E}x_d$, the exergetic efficiency and two exergy destruction ratios are presented in this work (Abusoglu & Kanoglu, 2009) and (Aljundi, 2009).

The exergetic efficiency (ε_k) of the k^{th} component is the ratio of the product exergy ($\dot{E}x_{p,k}$) to the fuel exergy ($\dot{E}x_{f,k}$):

$$\varepsilon_k = \frac{\dot{E}x_{p,k}}{\dot{E}x_{f,k}} \quad (74)$$

The exergy destruction ratio $y_{d,k}$ is the ratio compares the exergy destruction in the k^{th} component with total fuel exergy supplied $\dot{E}x_{fuel}$ to overall the system:

$$y_{d,k} = \frac{\dot{E}x_{d,k}}{\dot{E}x_{fuel}} \quad (75)$$

On the other hand, the exergy destruction rate of the k^{th} component can be compared to the total exergy destruction ($\dot{E}x_{d,Total}$) (Sum of all k components in the system):

$$y^*_{d,k} = \frac{\dot{E}x_{d,k}}{\dot{E}x_{d,Total}} \quad (76)$$

3.6 MATLAB PROGRAM

A MATLAB program code has been developed in order to simulate the performance of the combined cycle power plant given in Figure 1. The code structure is based on the methodology equations (1 to 76) which are described in the previous sections, and it is programmed in MATLAB. The program consists of eight functions and they are linked together to build out the main program. These functions are as follow: gas properties, air compressor, combustion chamber, gas turbine, heat recovery steam generator, air cooled condenser, steam turbine and finally exergy analysis. The gas properties function uses the weather condition as the main input data for the main program and it is a key for evaluating the performance of the gas turbine model. The thermodynamic properties (e.g. enthalpy, specific heat and entropy) of the inlet air are estimated from the gas properties function, this function works based on the input data of the weather condition. The flowcharts as given in Figure 47 and Figure 48 in Appendix A; show how the gas properties of wet air and combustion gases are estimated.

In the present work, the specific heat of gas at constant pressure is defined as a function of temperature, given by the polynomial adopted from (Turns, 2000).

$$c_p(T) = R_u (a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4) \quad (77)$$

The constants of the different species are given in

Table 3. Therefore, the enthalpy and the entropy of gas turbine working fluid can be calculated using the above function in the following equations:

$$h(T) = \int c_p(T) dT \quad (78)$$

$$s(T, P) = \int c_p(T) \cdot \frac{dT}{T} - R_u \ln \frac{P}{P_a} \quad (79)$$

Where P_a is ambient pressure and R_u is universal gas constant which is equal to 8.314 kJ/kmol.K.

Table 3 The coefficients of Eq.77 of the different species

Species	a_1 [kJ/kg]	$a_2 \cdot 10^2$ [kJ/kg.k]	$a_3 \cdot 10^6$ [kJ/kg.k ²]	$a_4 \cdot 10^9$ [kJ/kg.k ³]	$a_5 \cdot 10^{14}$ [kJ/kg.k ⁴]
Temperature 250 to 1000K					
N ₂	3.299	0.141	-3.963	5.642	-244.485
O ₂	3.213	0.113	-0.576	1.314	-87.686
CO ₂	2.276	0.992	-10.409	6.867	-211.728
Ar	2.500	0	0	0	0
H ₂ O	3.387	0.347	-6.355	6.969	-250.659
Temperature above 1000K					
N ₂	2.927	0.149	-0.568	0.101	-0.675
O ₂	3.698	0.061	-0.126	0.018	-0.114
CO ₂	4.454	0.314	-1.278	0.239	-1.669
Ar	2.500	0	0	0	0
H ₂ O	2.672	0.306	-0.873	0.120	-0.639

The second function is air compressor model which is described in the flowchart as given in Figure 49 in Appendix B. This function is used the thermodynamic properties of the inlet air, which are generated from gas properties function, to calculate the air compressor work in and compressor discharge temperature. The discharge temperature again is being used as input data for the gas properties function in order to estimate the compressor discharge thermodynamic properties.

The third function is combustion chamber model and it is described in the flowchart as given in Figure 50 in Appendix B. This function is the core of the program and it has three main input data which are air compressor exit properties, fuel gas properties and the turbine inlet temperature. The fuel air ratio and fuel consumption are calculated in this function and satisfying the combustion chamber energy balance equation. The

combustion chamber exit again is being used as input data for the gas properties function in order to estimate its thermodynamic properties.

The fourth function is gas turbine model and it is described in the flowchart as given in Figure 51 in Appendix B. This function is calculated turbine work output and exhaust temperature from combustion chamber exit thermodynamic properties and compressor pressure ratio. Turbine exhaust temperature again is being used as input data for the gas properties function in order to estimate its thermodynamic properties.

The fifth function is heat recovery steam generator model and it is described in the flowchart as given in Figure 52 in Appendix B. The main input data for this function is the turbine exhaust temperature and its thermodynamic properties, from these inputs the steam mass flow rate and steam temperature are computed.

The sixth function is air cooled condenser model and it is described in the flowchart as given in Figure 53 in Appendix B. The steam turbine back pressure is defined through this function. The main input parameters for this function are the steam mass flow rate and the weather condition. The condensation steam temperature is calculated as a function of steam mass flow rate and ambient temperature, this temperature is the saturated temperature of the steam turbine exhaust end and from it the steam turbine back pressure is being calculated.

The seventh function is steam turbine model and it is described in the flowchart as given in Figure 54 in Appendix B. The steam turbine model function uses the steam properties exit from HRSG and the condensate temperature from the air cooled condenser to compute the steam turbine work output.

Finally, the eighth function is the exergy analysis which is used all the output computed data from the earlier functions to compute the exergy destruction, exergetic efficiency, exergy destruction ratio and rate of all combined cycle power plant components.

CHAPTER 4

4 RESULTS AND DISCUSSION

4.1 INTRODUCTION

In this chapter, the MATLAB program is run to produce the results of the given system, shown Figure 1. The input data at the design point used in the present work to run the thermodynamic simulation program are given in Table 4. The data cover, the air compressor, combustion chamber, the gas turbine, heat recovery steam generator and its components, steam turbine, and air cooled condenser. The program is used to reproduce the power at the full and the part load for the gas turbine (Top cycle) because the performance of the combined cycle depends mainly on the performance of the gas turbine.

The performance of the combined cycle power plant is investigated using the above equations (1 to 79). The MATLAB program is used to solve the equations and a function called X-Steam, as given in appendix A, is being used to get the steam properties at different operating condition for all steam cycle. In addition to that, a MATLAB function is constructed and used to get the air and hot gases properties at different operating condition for the gas turbine cycle and the heat recovery steam generator.

The obtained results from the present model for the gas turbine power are given in Table 5 along with those given by the manufacture. The deviations between the calculated power of the gas turbine and the actual ones are in the range of 3-5%, as shown in Table 5. These results indicate that the present model is capable to produce the behavior of the gas turbine as well as the combined cycle at different operating condition. It is expected that the obtained results from the gas turbine model to be greater than the actual ones because the present model has not considered all the losses within the gas turbine cycle.

Table 4 Main data for the thermodynamic simulation of the cycle at the design points

Unit	Parameter	Value
Compressor	i. Pressure loss of entry pressure	0.5%
	ii. Polytropic efficiency	90%
	iii. Mechanical efficiency	99%
	iv. Pressure ratio	14.2
	v. Compressor volume flow rate	429 m ³ /s
Combustion chamber	i. Combustor efficiency	98%
	ii. Pressure loss of entry pressure	0.2%
	iii. Heat losses	2%
Gas turbine	i. Polytropic efficiency	95%
	ii. Exhaust pressure	1.113 bar
	iii. Turbine inlet temperature	1100 °C
HRSG	i. HRSG efficiency	99.5%
	ii. Temperature difference between flue gases and HP steam	20 °C
	iii. Temperature difference between flue gases and LP steam	11 °C
	iv. Degree of superheat of LP live steam	100 °C
	v. HP evaporator pinch point temperature difference	10 °C
	vi. LP evaporator pinch point temperature difference	18 °C
	vii. HP economizer approach temperature difference	6 °C
	viii. LP economizer approach temperature difference	6 °C
	ix. HP feed water pressure to HRSG	105 bar
	x. LP feed water pressure to HRSG	18 bar
	xi. HP drum pressure	70 bar
	xii. LP drum pressure	4.2 bar
	xiii. Extracted water from LP economizer to deaerator	30 kg/s
Deaerator	Deaerator temperature	125 °C
Pumps	Isentropic efficiency	85%
LP pre-heater	i. Condensate water outlet temperature	85 °C
	ii. Condensate pressure	11.5 bar
Ejector	i. Steam entry pressure	15 bar
	ii. Steam mass flow rate	0.1 kg/s
Steam turbine	i. Minimum steam quality at exhaust	0.875
	ii. Isentropic efficiency	95%
	iii. Extraction steam pressure	0.85 bar
Air cooled condenser	i. Number of fans	16
	ii. Fan air volume flow rate	757 m ³ /s
	iii. Total heat transfer surface area	619000 m ²
	iv. External diameter of tube	35 mm
	v. Tubes length	11.25 m
	vi. Fins outer diameter	73 mm
	vii. Fins thickness	0.3 mm
	viii. Fins pitch	2.8 mm
	ix. Distance between tubes in plane perpendicular to flow	54 mm
	x. Number of tubes	7840
	xi. Log mean temperature difference	15.35 °C

Table 5 The comparison between the model results of the gas turbine with the corresponded ones given by the manufacture at the design point of gas turbine

%Load	Given by Manufacture (MW)	Calculated Load (MW)	% Deviations
0	0	0	0
25	40	42.0347	5.087
50	81	84.0738	3.795
75	120	126.0861	5.072
80	131	136.1730	3.949
85	137	142.8976	4.305
90	145	151.3033	4.347
100	162	168.1148	3.775

Six simulations are conducted for the give combined cycle configuration seen in Figure 1. Five operating parameters are fixed while one operating variable is varied over an appropriate range. The investigated parameters in the present work are: ambient temperature, turbine inlet temperature, compressor pressure ratio, heat losses through combustion chamber, gas turbine with intake air cooling system and gas turbine part load operation. The effect of these variables on the performance are investigated and reported in the coming sections.

4.2 EFFECTS OF AMBIENT TEMPERATURE

In the first simulation the ambient temperature varies from 15°C to 45°C. This hot climate is particularly relevant to the Bahrain weather where the ambient temperature varies from 15°C to 45°C during the winters and summers. This simulation is intended to show the effect ambient temperature has on the performance of combined cycle system. Table 6 presents the effects of two operation ambient temperature on the combined cycle power plant components.

Combined cycle power plant would be required to operate over a wide range of temperature depending on its location. In Bahrain the ambient temperature could be

varied from 15°C in the winter season to 45°C in the peak of summer. The effect of ambient temperature on the performance of the combined cycle power plant is measurable. This section will be evaluating the performance of the combined cycle power plant components based on energy and exergy analysis method.

Table 6 The performance of combined cycle power plant at two different ambient condition

Parameters	At ISO Condition ^(a)	At site Condition ^(b)
Compressor pressure ratio (r_c)	14.20	12.98
Compressor discharge temperature (°C)	396.75	432.09
Compressor efficiency (%)	85.80	86.01
Air mass flow rate (kg/s)	523.52	468.73
Turbine inlet temperature (°C)	1100	1100
Fuel mass flow rate (kg/s)	14.09	12.38
Turbine discharge temperature (°C)	511.89	533.9
Hot gases mass flow rate (kg/s)	537.61	481.11
Turbine efficiency (%)	96.27	96.22
Gas turbine thermal efficiency (%)	35.21	32.37
Gas turbine net power output (MW)	167.59	135.33
Gas turbine heat rate (kJ/kWhr)	10,225	11,122
Back work ratio (%)	55.50	59.43
HP steam mass flow rate (kg/s) – Two boilers	119.56	118.80
HP steam temperature (°C)	491.89	513.90
HP steam pressure (bar)	70	70
LP steam mass flow rate (kg/s) – Two boilers	33.77	23.88
LP steam temperature (°C)	245.38	245.38
LP steam pressure (bar)	4.2	4.2
HRSG flue gases outlet temperature (°C)	148.92	159.60
Steam turbine power output (MW)	131.89	127.31
Steam turbine back pressure (mbar)	140	324
Overall cycle thermal efficiency (%)	48.08	46.64
Combined cycle power output (MW)	463.55	393.76
Combined cycle heat rate (kJ/kWhr)	7,393	7,646

^(a) Ambient condition: Temperature is 15 °C, Pressure is 1.013bra and Relative humidity is 60%.

^(b) Ambient condition: Temperature is 42 °C, Pressure is 1.013bra and Relative humidity is 80%.

In this section the effect of ambient temperature variation has been analyzed by using two approaches:

- Energy analysis.
- Exergy analysis.

4.2.1 ENERGY ANALYSIS

The performance of the air compressor at different ambient temperature is shown on Figure 3. The air compressor efficiency, discharge temperature, pressure ratio, mass flow rate are presented in Figure 3 relative to *ISO* condition values which are given in Table 6. The results show that, the air compressor efficiency tends to increase as the ambient temperature increases. The relation between compressor efficiency and ambient temperature is linear as shown in Figure 3 even though the increase in efficiency is very negligible. It is around 0.254% over the range of the given ambient temperature.

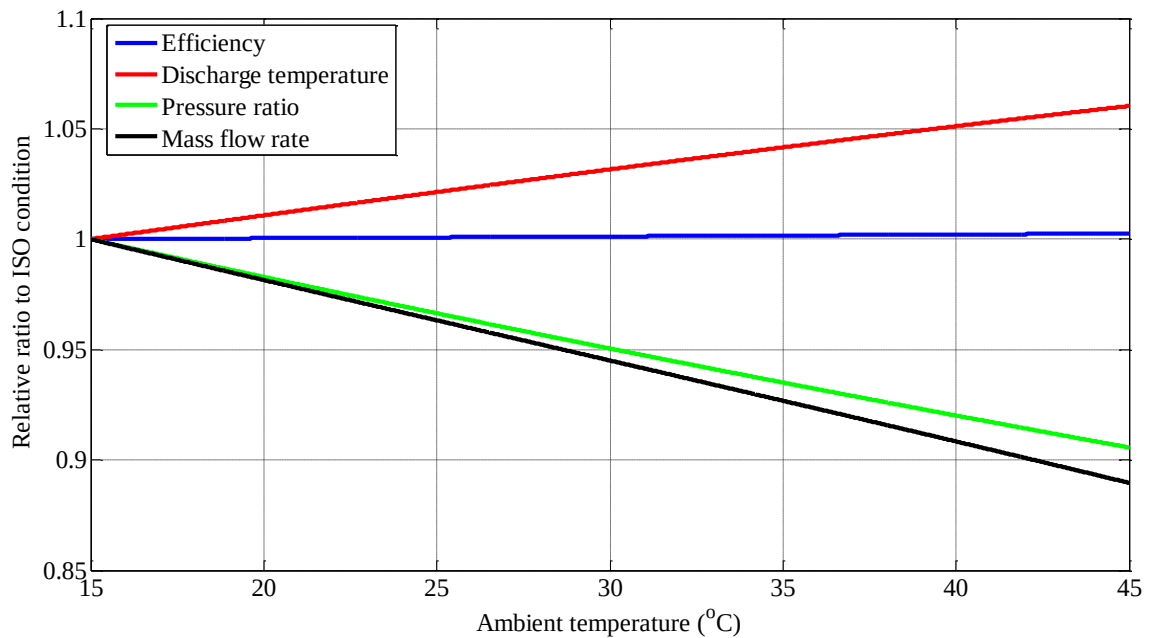


Figure 3 The effect of ambient temperature on compressor performance

The ambient temperature has a significant effect on the compressor pressure ratio as shown in Figure 3. The pressure ratio starts to fall down as the ambient temperature increases. The relation between the compressor pressure ratio and ambient temperature was presented in the work of Bracco and et al. (Bracco, et al., 2007) and is given by Eq.16. The pressure ratio decreases by 1.356 bar from the design value. The reason behind that is the air density drops at high ambient temperature and causing the pressure ratio to drop too.

Figure 3 represents also the effects of ambient temperature on the air compressor discharge temperature; it is obvious that the air compressor discharge temperature increases as the ambient temperature increases. The difference between the air compressor discharge temperature at 15 °C and 45 °C is found to be 40.4 °C consequently this increase in the air compressor discharge temperature will case an increase in the compressor work as shown in Figure 4. In addition to that the fuel consumption will be reduced as the air leaving the compressor becomes hotter which is required less amount of energy to achieve the desired turbine inlet temperature.

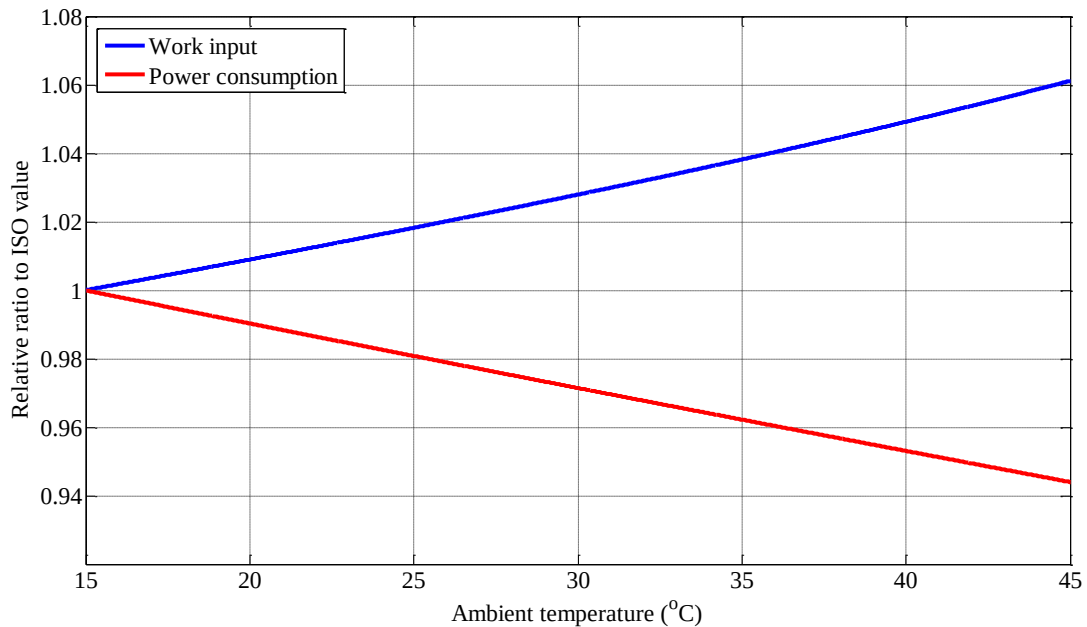


Figure 4 Compressor work input and power consumption with different ambient temperature

As it is noticed in Figure 4 the compressors work input is increasing when the ambient temperature increases but the compressors power consumption behaves in contradictory manner. Figure 3 shows the compressor deliver air mass flow rate is fallen as the ambient temperature increases and this justifies reason why the compressors power consumption is decreasing as the ambient temperature increases.

The compressor air mass flow rate is decreasing due to the air density tends to decrease when the temperature increases. Finally, it is understandable that the ambient temperature has a negative impact on the compressor performance when it increases relative to *ISO* operating conditions.

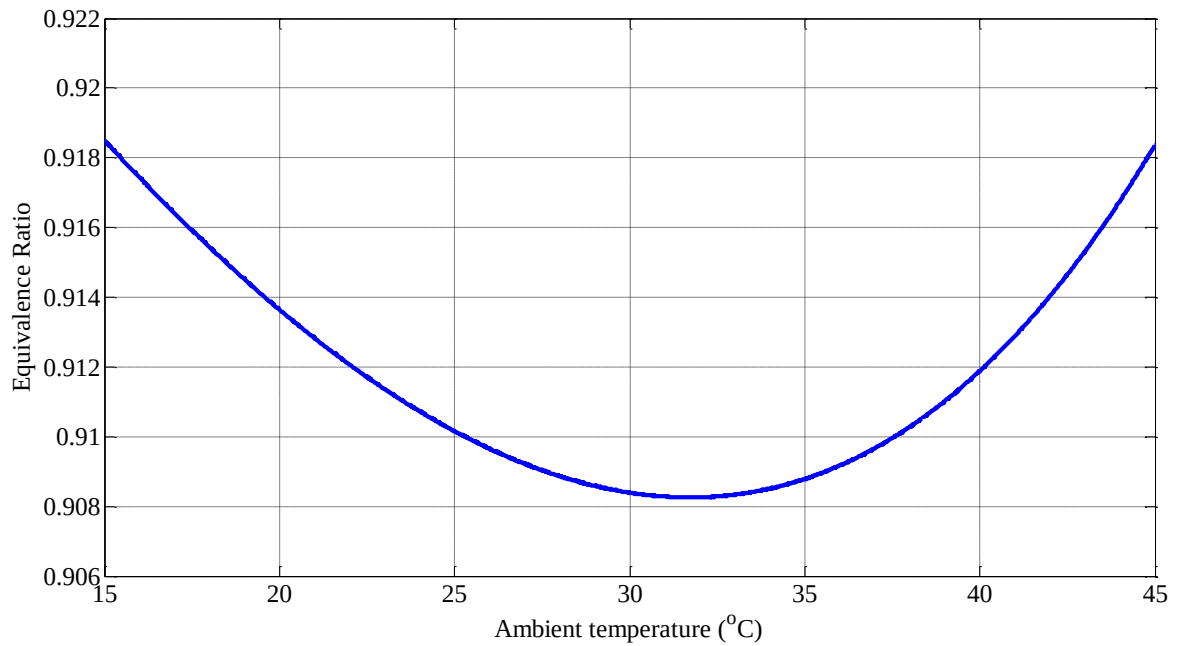


Figure 5 The relation between equivalence ratio and ambient temperature

The relation between equivalence ratio and ambient temperature is presented on Figure 5. The results show that, the equivalence ratio is changed in the shape of parabolic compared with ambient temperature variation. It is obvious that the combustion process remains lean since the equivalence ratio remains below the unity as shown in Figure 5. The equivalence ratio drops as the ambient temperature increases and reaches it lower value when the ambient temperature is equal to 32 °C.

The equivalence ratio decreases from 0.9185 to its minimum value of 0.9083 when the ambient temperature is varied from 15 °C to 32 °C. The reason for this reduction is caused by that the rate of change of fuel mass flow rate is higher than the rate of change of air mass flow rate due to the ambient temperature increases. While for ambient temperature varies from 32 °C to 45 °C the rate of change of air mass flow rate is higher than the rate of change of fuel mass flow rate as it shown in both Figure 3 and Figure 6.

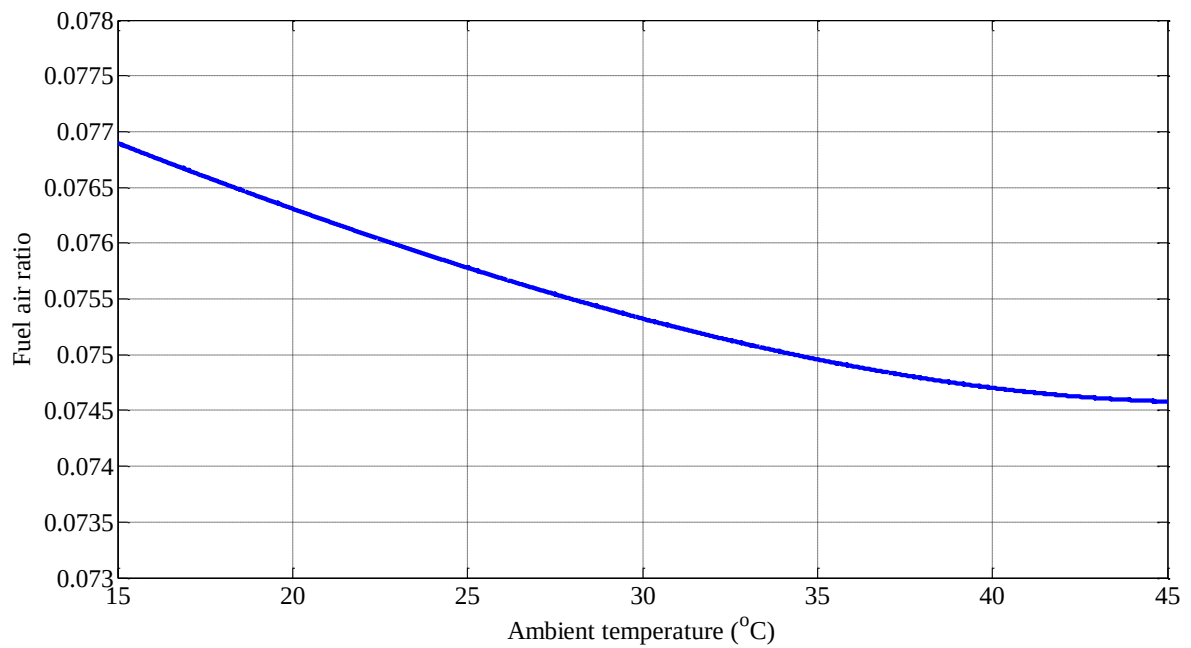


Figure 6 The effect of ambient temperature on fuel air ratio

The fuel air ratio is decreasing as the ambient temperature increases as shown in Figure 6. The reason of this reduction in the fuel air ratio is that, as the ambient temperature increases the air compressor discharge temperature, as shown in Figure 3, is increased subsequently the air enters the combustion chamber becomes hotter as a result of this. Finally, as a result for that, the amount of energy requires achieving the desired turbine temperature is reduced, in other word; the fuel mass flow rate will be decreased (Sanjay, 2011). The results show that, the fuel air ratio reduces nonlinearly with ambient temperature variation as shown in Figure 6.

The relative fuel consumption to the *ISO* condition compared with ambient temperature is shown in Figure 7. The relative fuel consumption is decreasing proportionally as the ambient temperature increases. The reason for that is as air inlet temperature to compressor increases, fuel relative consumption decrease to compensate for the increases temperature of the air exiting the compressor as shown in Figure 3.

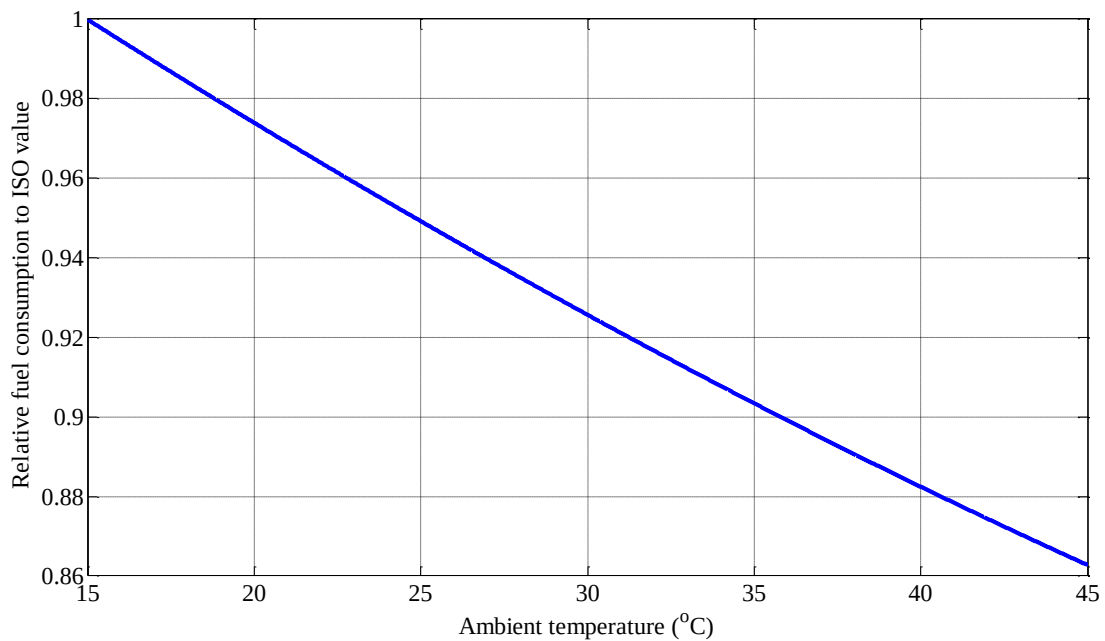


Figure 7 The effect of ambient temperature on relative fuel consumption

The results show that, the relative fuel consumption is reduced from unity to 0.8625 which equal to 13.75% drops when the ambient temperature is varied from 15°C to 45°C. This reduction is like an indication that the gas turbine power output will be decreased as the input energy to the system decreases.

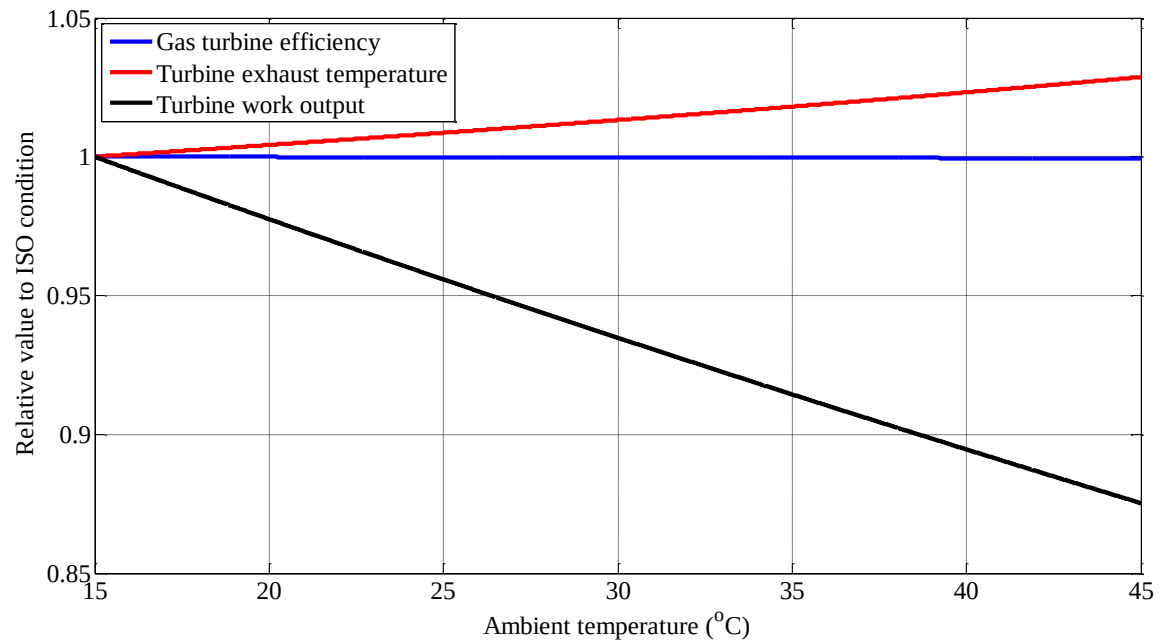


Figure 8 The effect of ambient temperature on gas turbine performance

The gas turbine performance such as efficiency, exhaust temperature and work output are presented in Figure 8 compared with ambient temperature. The results show that, the gas turbine efficiency is tended to remain constant when the ambient temperature is varied from 15 °C to 45 °C even though there is a small reduction in the efficiency which is around 0.061% but it can be considered as negligible changes over the range of ambient temperature variation.

There is a significant change in turbine exhaust temperature and turbine work output can be seen in Figure 8. The turbine exhaust temperature is a function of pressure ratio and turbine inlet temperature as given in Eq.30. The only variable that can affect the turbine exhaust temperature is the pressure ratio since it is varying over the range of ambient temperature variation, the exhaust temperature inversely proportional to the pressure ratio as given in Eq.30. Therefore, turbine exhaust temperature increases as the pressure ratio is decreased, as shown in Figure 3, when the ambient temperature is varied from 15 °C to 45 °C.

The gas turbine work output decreases as the turbine exhaust temperature increases since the work output is a function between the inlet temperatures and exhaust

temperature as given in Eq.31. The results show that, the turbine work output decreases by 12.46% as the ambient temperature increases 15 °C to 45 °C, as shown in Figure 8.

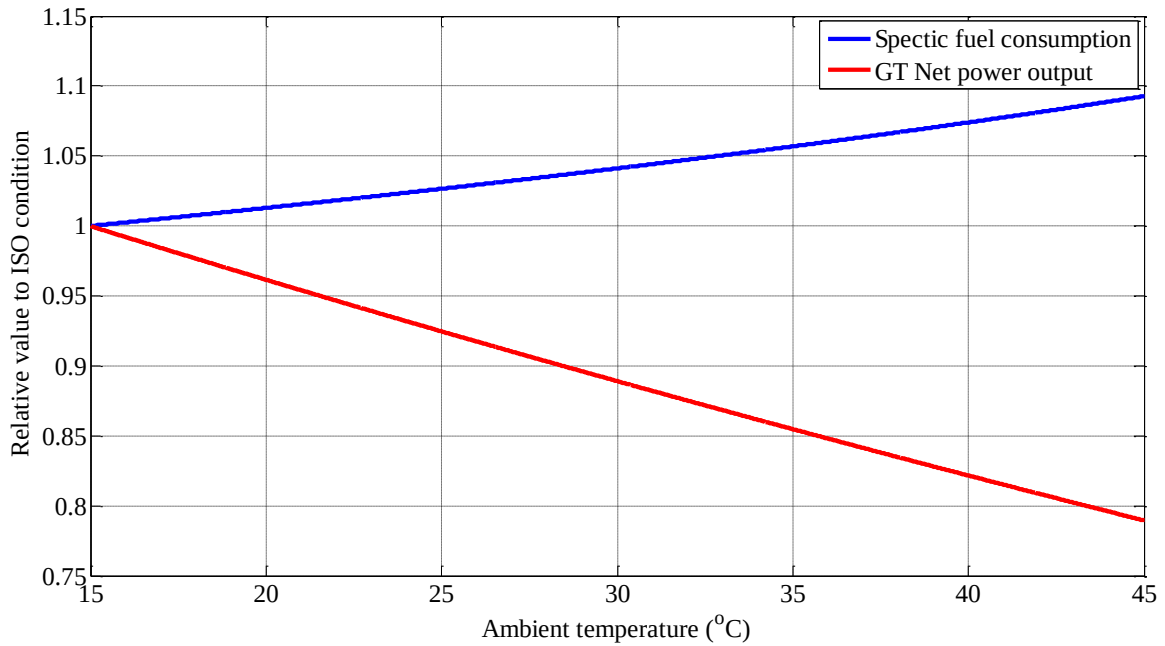


Figure 9 Specific fuel consumption and GT net power output vs. ambient temperature

The gas turbine net power output decreases as shown in Figure 9 when the ambient temperature is varied from 15 °C to 45 °C (Rahman, et al., 2011). The net gas turbine power output reduces linearly by 21.04% which is equivalent to 35.26MW. The reason for that is the air density is decreasing as the ambient temperature increases.

Figure 10 presents the back work ratio, the ratio of compressor work to the turbine work. The back work ratio increases from 55.5% to 59.9% when the ambient temperature is varied from 15 °C to 45 °C. this means air compressor requires more power to deliver the same volumetric rate of air to the combustion chamber at higher temperature.

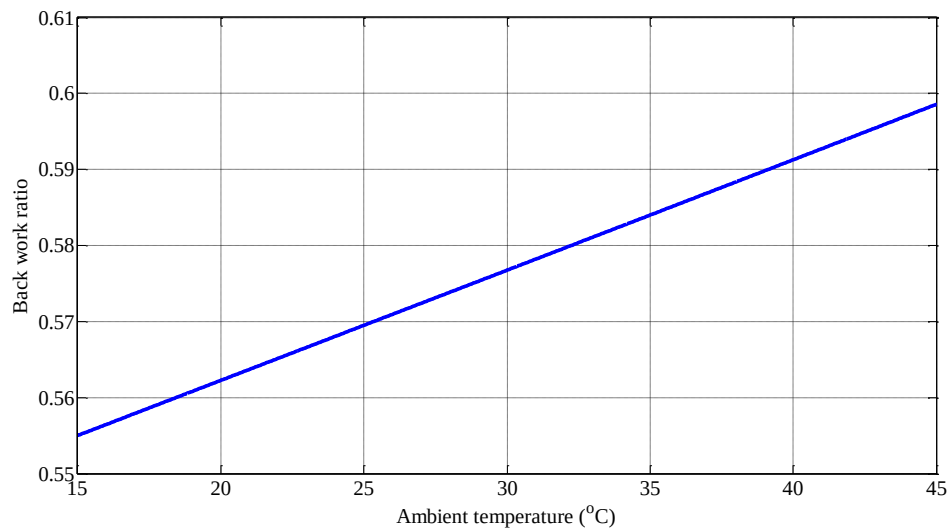


Figure 10 The relation between back work ratio and ambient temperature

The gas turbine heat rate is affected by ambient temperature as shown in Figure 11 due to the change of air density and compressor work since a lower ambient temperature leads to a higher air density and a lower compressor work that in turn gives a higher gas turbine power output (Rahman, et al., 2011).

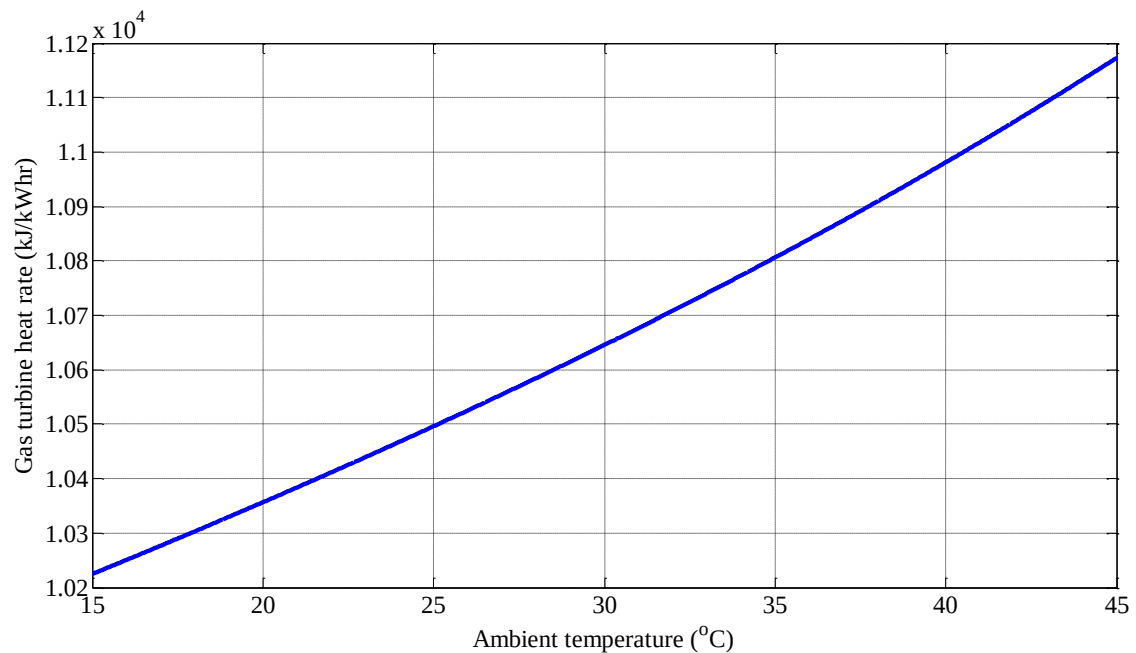


Figure 11 Gas turbine heat rate vs. ambient temperature

The results show that, the heat rate increases from 10,225 kJ/kWhr to 11,172 kJ/kWhr this means the gas turbine overall efficiency decreases as the ambient temperature is increased as the amount of energy to produce one kWhr is increased as shown in Figure 11.

The ambient temperature has a significant influence on the gas turbine within the range of values that was analyzed, the gas turbine performance starts to get worse, resulting droops in turbine power output and overall thermal efficiency. In contrary of that the turbine exhaust temperature is increasing so it is preferably to utilize the high exhaust gases temperature released from gas turbine to generate steam by installing a heat recovery steam generator then utilize it to run a steam turbine.

The effect of ambient temperature on the relative steam generation from heat recovery steam generator is shown in Figure 12. It is obvious that the relative steam generation starts to decrease as the ambient temperature increase. The reason for that is the flue hot gas mass flow rate from gas turbine exhaust reduces, as shown in Figure 3, as the ambient temperature increases.

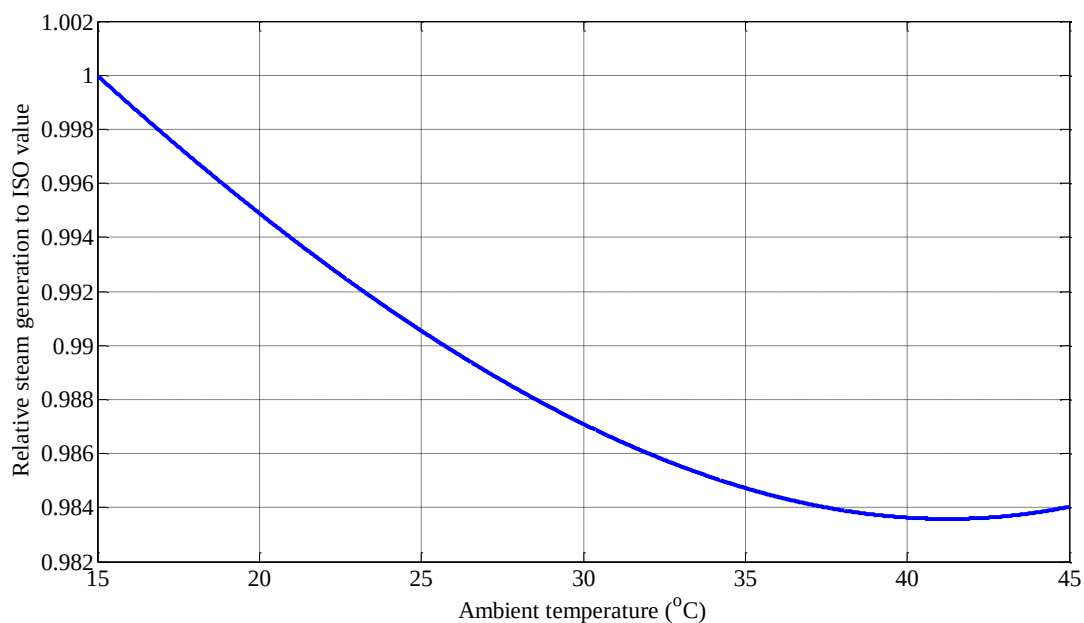


Figure 12 The effect of ambient temperature on relative steam generation from HRSG

The relative steam generation decreases to its minimum value when the ambient temperature is 41.3°C and then a slight increase can be observed from Figure 12. This phenomenon is occurred despite the fact that the flue gases mass flow rate reduces is due to at high ambient temperature the turbine exhaust temperature is increasing as shown in Figure 8, this increases in exhaust temperature will lead to increase the steam mass flow rate as the steam pressure is maintained constant over the range of the given ambient temperature.

The results show that, the steam generation from the heat recovery steam generator decreases by 1.64% at ambient temperature is 41.3°C then a minor increase about 0.04% is observed when ambient temperature reaches 45°C as shown in Figure 12.

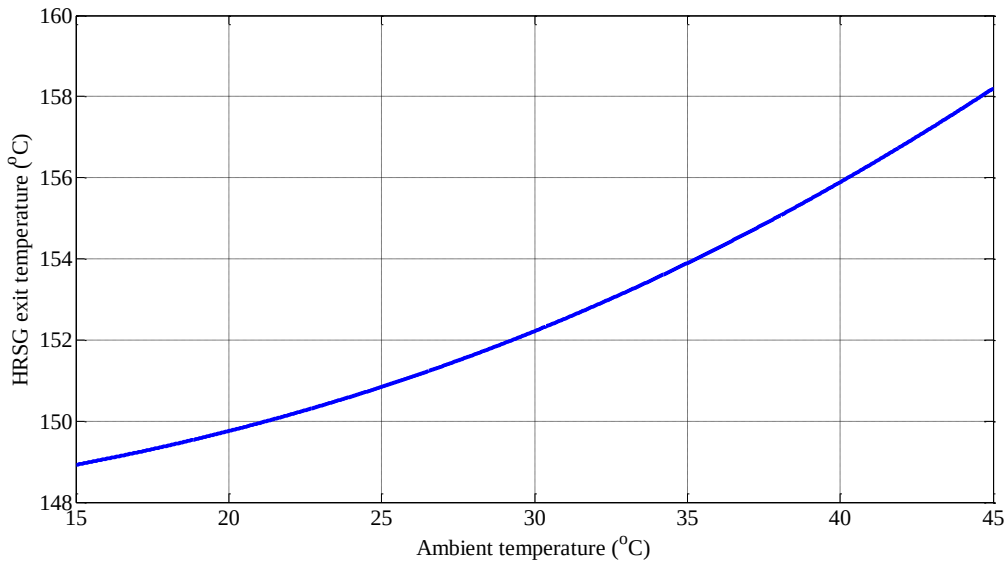


Figure 13 The effect of ambient temperature on HRSG flue gases exit temperature

The flue gases exit temperature from the heat recovery steam temperature is a very essential parameter on the heat recovery steam generator; the flue exit temperature has an effect on the performance of the heat recovery steam generator. Hence the heat extracts from hot gases is increased as the exit temperature decreases. In our model it has assumed that the exit temperature is a function of the dew point temperature of the flue gases compositions. In order to avoid any acid corrosion formation within the boiler tubes, the temperature distribution inside the heat recovery steam generator is higher than the acid dew point temperature of the flow hot gases. The acid dew point

temperature of a flue gas is defined as the temperature at a given pressure at which any gaseous acid in the flue gas will start condense into liquid acid.

The dew point temperature of the flue hot gases is considerable if the fuel gas contains any aggressive composition such as sulfur which can result acid corrosive formation.

The results show that, the HRSG exit temperature is increasing by 9.27°C over the range of the ambient temperature due to the water substance increases from 6.88% to 11.14% of the total flue gases compositions.

The relative steam turbine power output is decreasing over the range of ambient temperature variation as shown in Figure 14. It has noticed that the steam turbine power output drops with different slop at two zones. The first zone is when the ambient temperature is varied from 15°C to 25.3°C , the steam turbine power output drops with rate of $0.123\%/^{\circ}\text{C}$ which equivalent to 1.27%. The second zone is when the ambient temperature is varied from 25.3°C to 45°C , the steam turbine power output drops with rate of $0.158\%/^{\circ}\text{C}$ which equivalent to 4.39%.

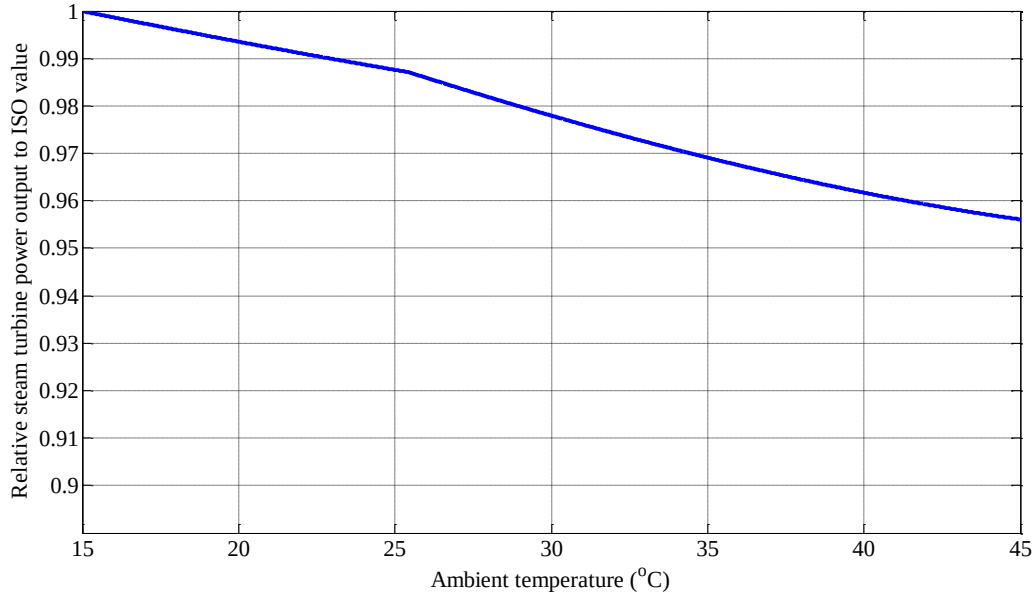


Figure 14 Relative steam turbine power output to ISO value vs. ambient temperature

The steam turbine power output drops in two zones at different slops for two reasons. First reason is the steam mass flow rate is decreasing as the ambient temperature is

increased as shown in Figure 12 and the second reason is the back pressure starts to drop when the ambient temperature is above 25.3 °C as shown in Figure 14. Therefore, the steam turbine drops at lower rate when the ambient temperature is below 25.3 °C because the steam turbine back pressure remains constant. As soon as the back pressure starts to vary when the ambient temperature is above 25.3 °C, the steam turbine power output drops at higher rate.

The steam turbine back pressure plays an essential role of steam turbine power output. It has a direct impact on the power out if there is any variation from the design value. The effect of back pressure can be seen in Figure 14. Figure 15 shows the effect of ambient temperature on the steam turbine back pressure, as the block is equipped with air cooled condenser type, the steam turbine back pressure is very sensitive to the ambient temperature as shown in Figure 15.

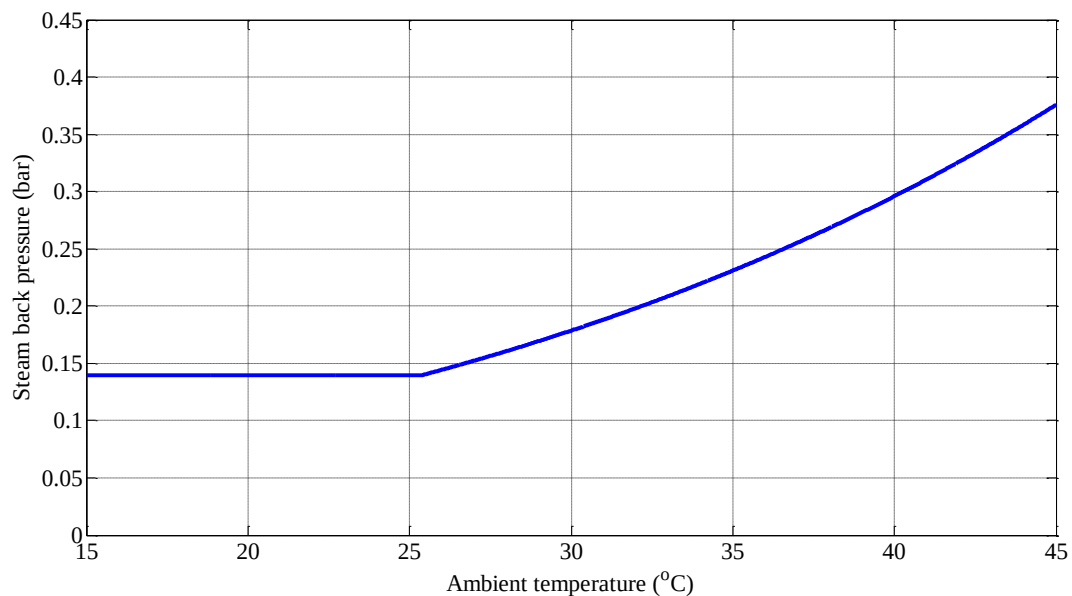


Figure 15 The effect of ambient temperature on steam turbine back pressure

The results show that, steam turbine back pressure is remained constant from 15 °C to 25.3 °C and as the ambient temperature increases more the back pressure is increased too. The reason for that is when the ambient temperature varies from 15 °C to 25.3 °C, the number of air fans in service increase from the minimum fans required to condense the exhaust steam within the air cooled condenser to the maximum number of fans

available in the air cooled condenser. For temperature above 25.3 °C, the heat transfer coefficient of air is reduced as the ambient temperature increases which will follow an increase in steam turbine back pressure as a resultant. This increase in the back pressure will be followed by reduction in the steam turbine power output. The reduction in steam turbine power output is 1.23% when the ambient varies from 15 °C to 25.3 °C, and it is 3.14% when the ambient varies from 25.3 °C to 45 °C.

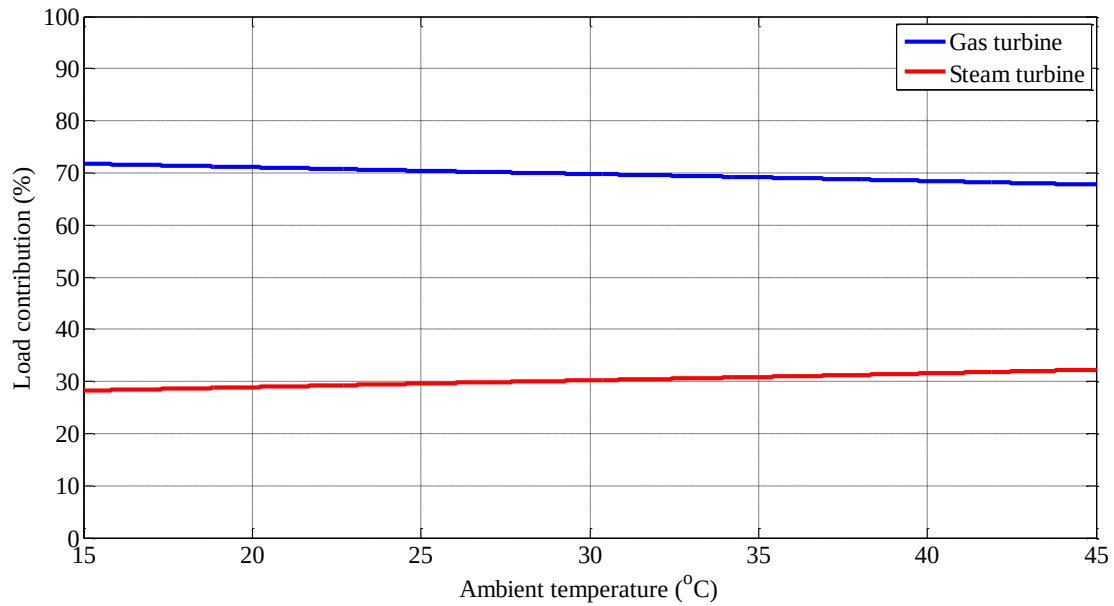


Figure 16 Gas and steam turbines load contribution

Both gas and steam turbines contribution loads on the combined cycle power plant total loads are presented in Figure 16. The results show that, gas turbines contribution load is decreased as the ambient temperature increases while the steam turbine load is increased as the ambient temperature increases.

The drop in gas turbines contribution load is about 5.65% while the steam turbine load is increased by 14.31% as the ambient temperature varies from 15 °C to 45 °C. Figure 16 shows that the gas turbine load is affected negatively when the ambient temperature increases, but the steam turbine load during combined cycle operation tries to recover the lost of gas turbine loads. This reduction in gas turbine load is due to as the ambient temperature increases the air density decreases which will result a reduction in the air mass flow rate enters the gas turbine block and finally causes a reduction in the gas

turbine power output as discussed earlier. More effort to be studied to increase the steam turbine load during the variation of ambient temperature in order to maintain the total combined cycle power output almost constant regardless any vary in the ambient temperature.

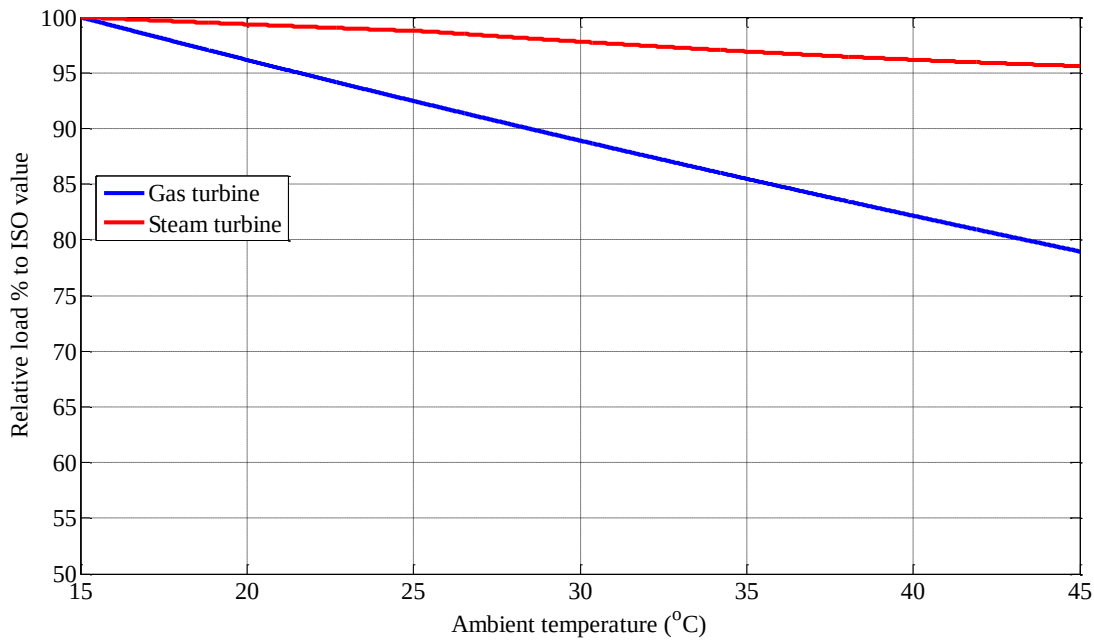


Figure 17 Gas and steam turbines relative loads to ISO condition

Figure 17 shows the effect of ambient temperature on gas turbine and steam turbine output load. Figure 17 represent the power output relatives to the *ISO* power output of each gas and steam turbine when ambient temperature varies from 15 °C to 45 °C. It is clear that the gas turbine output power is more sensitive to the ambient temperature than the steam turbine output.

This reduction in gas turbine's power output is due the air density as discussed earlier where the steam turbine power output reduction is due to the increases in the back pressure of steam turbine; the back pressure of steam turbine is very sensitive to the ambient temperature as shown in Figure 15 because the media of heat transfer in the condenser is air and this reduction could be avoided if water cooled condenser is being used in this system.

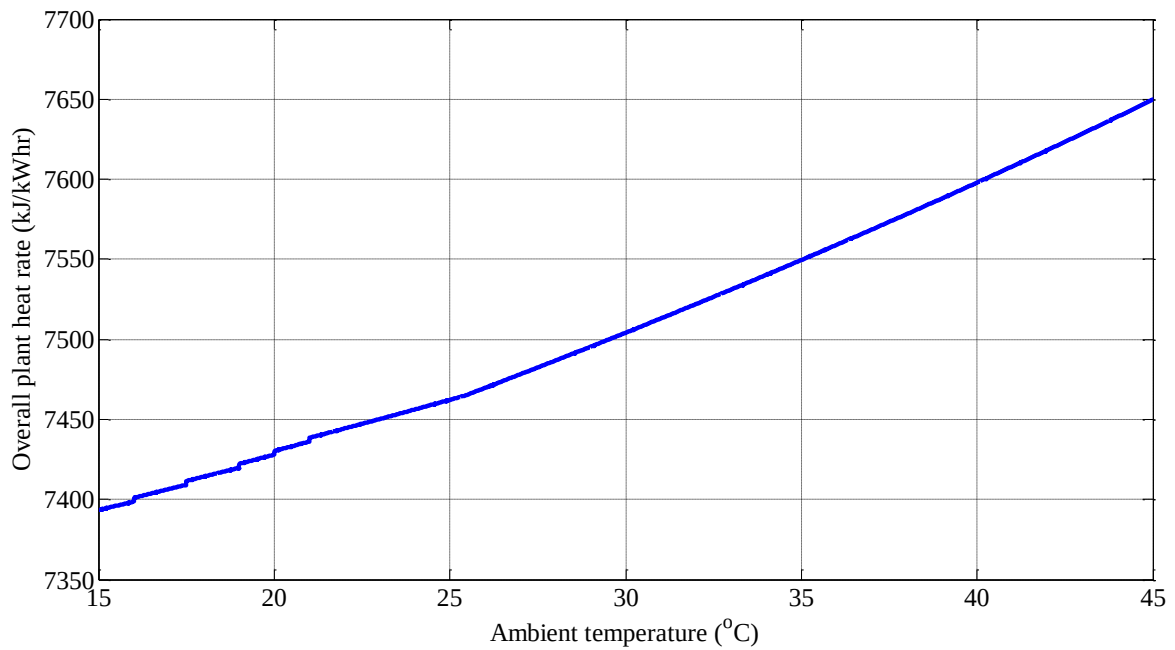


Figure 18 Overall plant heat rate vs. ambient temperature

Figure 18 represents overall plant heat rate changes over the range of investigated ambient temperature. It is obviously shown that the heat rate is increasing as the ambient temperature increases. The results show that, the overall plant heat rate is increased by 2.99%.

On the other hand, the results show that the specific fuel consumption in kg/s is dropped from 28.18 (Two gas turbines) to 24.31 (Two gas turbines) when the ambient temperature varies from 15 to 45 °C while the total Block power output is dropped from 467.5 MW to 391.3 MW. The reason for this drop is due the air compressor discharge temperature increases at high ambient temperature which is approximately about 40.4 °C above the design temperature at the *ISO* condition, this increase will lead to decrease the fuel consumption.

4.2.2 EXERGY ANALYSIS

The exergy analysis is used to identify the most inefficient and efficient components of the given system shown in Figure 1. It locates the largest source of exergy destruction

and losses in the system. Table 7 gives the exergy destruction for all of the components at *ISO* condition. Four different methods are being used to display destructions and losses, as discussed in Eq.73 to Eq.76:

- The exergy destruction, $\dot{E}x_d$.
- The exergetic efficiency, ε_{Exergy} .
- The exergy destruction ratio, y_d .
- The exergy destruction rate, y_d^* .

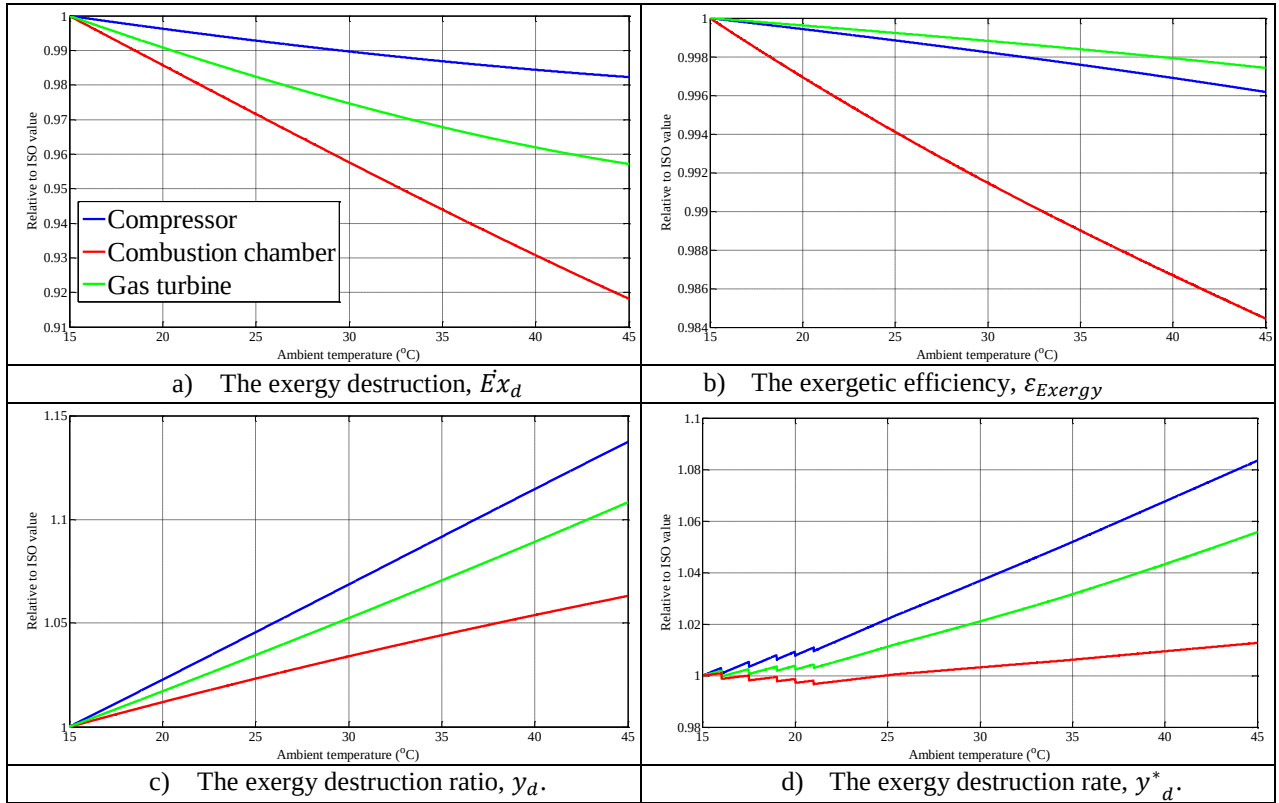


Figure 19 Compressor, combustion chamber and GT exergy destruction analysis vs. ambient temperature

The results show that, the total exergy destruction of the given system is around 530.90 MW. It is easily identified that the combustion chamber is the largest source of destruction which is around 37.30% of the total entire system. The next largest source of destruction is the steam turbine, accounting for 5.68% and followed by the air

compressor at 3.39%. The total of these three components account for 87.06% of the all destruction while other component such as heat recovery steam generator, air cooled condenser and gas turbine are at 2.83%, 2.71% and 2.01% respectively. The HRSG has a loss because the final exhaust gas is rejected to the environment and any energy contained therein is a loss.

The deaerator, LP Pre-heater, HP feed water pumps, ejector, condensate extraction pumps and LP feed water pumps exergy destruction rates are 0.26%, 0.22%, 0.03%, 0.02%, 0.006% and 0.004% respectively.

Table 7 Exergy destruction for all block elements at ISO condition

Components	$\dot{E}_d$ [MW]	ε_{Exergy} [%]	y_d^* [%]	y_d [%]
Compressor	18.00	91.39	3.53	3.39
Combustors	198.02	73.62	38.87	37.30
GT	10.69	97.24	2.10	2.01
HRSGs	15.05	87.24	2.95	2.83
Deaerator	1.35	91.78	0.13	0.26
HPFWP	0.17	88.95	0.02	0.03
LPFWP	0.02	87.40	0.002	0.004
ST	30.18	81.38	2.96	5.68
ACC	14.38	49.42	1.41	2.71
CEP	0.03	84.82	0.003	0.006
Ejector	0.08	95.33	0.008	0.02
LP Pre-Heater	1.16	80.92	0.11	0.22

The exergetic efficiency of all components is given in Table 7, the highest efficiency is identified as 95.33% and the lowest efficiency is 49.42%. Most of components exergetic efficiency is varying from 73% to 95%; except the air cooled condenser which is around 49.42% therefore there is big room for improvement here. Another way of viewing destruction, given in Table 7, is the exergy destruction ratio. This parameter does not total to 100% because not all of the fuel exergy is destroyed, a large portion of it is converted to useful work and heat output. Table 7 represents the exergy destruction, exergetic efficiency, exergy destruction rate, and exergy destruction ratio at

full load. The results show that, the higher exergy destruction rate is in combustion chamber which is 38.87% while the ACC exergy destruction rate is 1.41%. According to the first law of thermodynamics, energy losses associated with ACC is significant because it represent about 29.8% of the total energy input to the cycle and the results of the present work is agreed with the work of (Aljundi, 2009). These results reflect the importance of more work to be carried out to investigate the reasons behind the big destruction in the combustion chamber in order to reduce. The exergy destruction, exergetic efficiency, exergy destruction ratio and exergy destruction rate for compressor, combustion chamber and gas turbine relative to the *ISO* condition are presented in Figure 19.

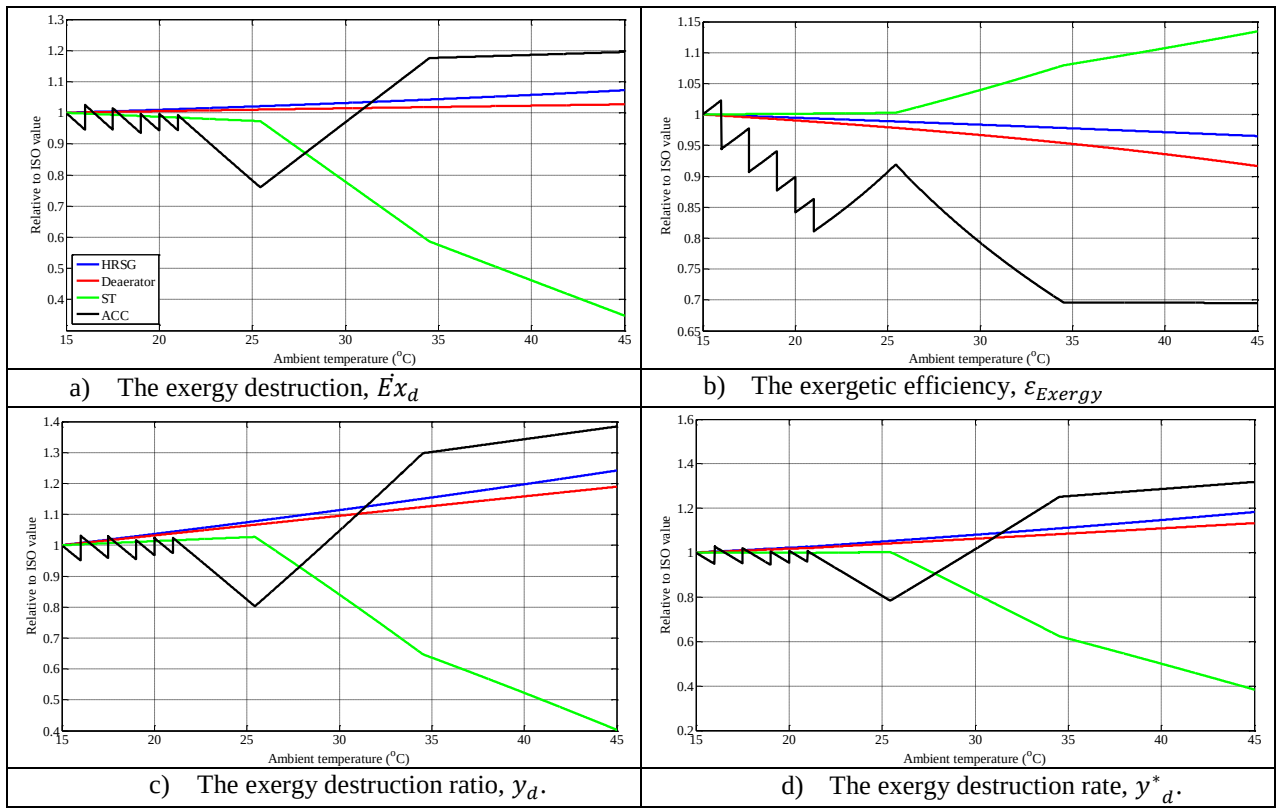


Figure 20 HRSG, deaerator, ST and ACC exergy destruction analysis vs. ambient temperature

The results show that, both exergy destruction and exergetic efficiency of compressor, combustion chamber and gas turbine are decreasing linearly as the ambient temperature increases while the exergy destruction ratio and exergy destruction rate of compressor, combustion chamber and gas turbine are increasing proportionally with ambient

temperature increases. It has noticed also that the exergy of combustion chamber is very sensitive to the ambient temperature as shown in Figure 19. The reason for that is the fuel mass ratio is decreasing as the ambient temperature increases as shown in Figure 7 consequently the exergy destruction and exergetic efficiency of combustion chamber are fallen down faster than compressor and gas turbine.

The exergy destruction, exergetic efficiency, exergy destruction ratio and exergy destruction rate for heat recovery steam generator, deaerator, steam turbine and air cooled condenser relative to the *ISO* condition are presented in Figure 20. The results show that, the exergy destruction, exergy destruction ratio and exergy destruction rate of heat recovery steam generator and deaerator are increasing linearly as the ambient temperature increases while for steam turbine is decreasing. The air cooled condenser (ACC) exergy destruction increases rapidly when the ambient temperature exceeds 25 °C and due to that the exergetic efficiency shrinks. It has noticed from Figure 20 that the steam turbine exergetic efficiency is growth as the ambient temperature increases. This supports our observation that at higher ambient temperature the steam cycle is not affected by the ambient temperature variation if the back pressure of steam turbine is maintained constant.

4.3 EFFECTS OF TURBINE INLET TEMPERATURE

The second simulation varies the turbine inlet temperature from 900 °C to 1400 °C. This simulation is intended to show the effect turbine inlet temperature has on the performance of combined cycle system. The turbine inlet temperature is increased by adding more fuel into the combustion chamber while the air mass flow rate remains constant, basically increasing the fuel air ratio.

In this section the effect of turbine inlet temperature variation has been analyzed by using two approaches:

- Energy analysis.
- Exergy analysis.

4.3.1 ENERGY ANALYSIS

As it is expected that will not be any changes in the compressor performance when the turbine inlet temperature varies. The reason for that is the compressor operation parameters will be held constant and the aims of this study is to study the performance of the combustion chamber, gas turbine as well as the combined cycle efficiency and performance.

The effects of turbine inlet temperature are studied in this section by varying the temperature from 900 °C to 1400 °C.

The relation between the turbine inlet temperature and relative fuel consumption (relative to design value, 1100 °C) is presented in Figure 21. It has been analyzed the effect of turbine inlet temperature on the fuel consumption when the turbine inlet temperature varies from 900 °C to 1400 °C. The results show that, the change of relative fuel consumption with respect to turbine inlet temperature is linearly proportional, and it is increasing as the turbine inlet temperature increases.

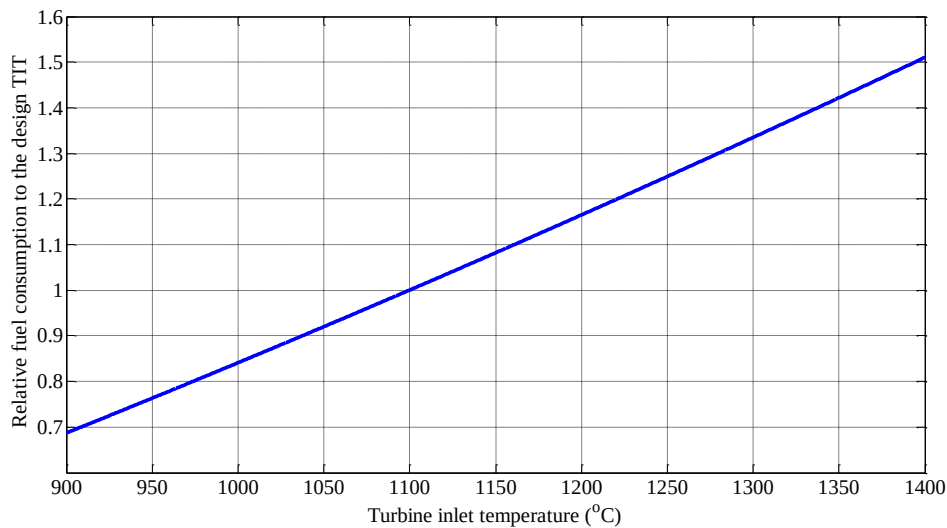


Figure 21 Relation between relative fuel consumption and TIT

The fuel consumption is increasing from 0.6865 to 1.5099 relative to fuel consumption at turbine inlet temperature equal to 1100 °C (design value); fuel consumption at 1100 °C is equal to 14.0892 kg/s at ISO condition, which means that the fuel consumption is

increased by 219.94% when the turbine inlet temperature varies from 900 °C to 1100 °C. Unfortunately, this increase in the fuel consumption has a side effect on the combustion process which is converted the combustion process from lean combustion to rich combustion. The results show that, the rich flame can be established if the turbine inlet temperature is above 1150 °C as it shown in Figure 22.

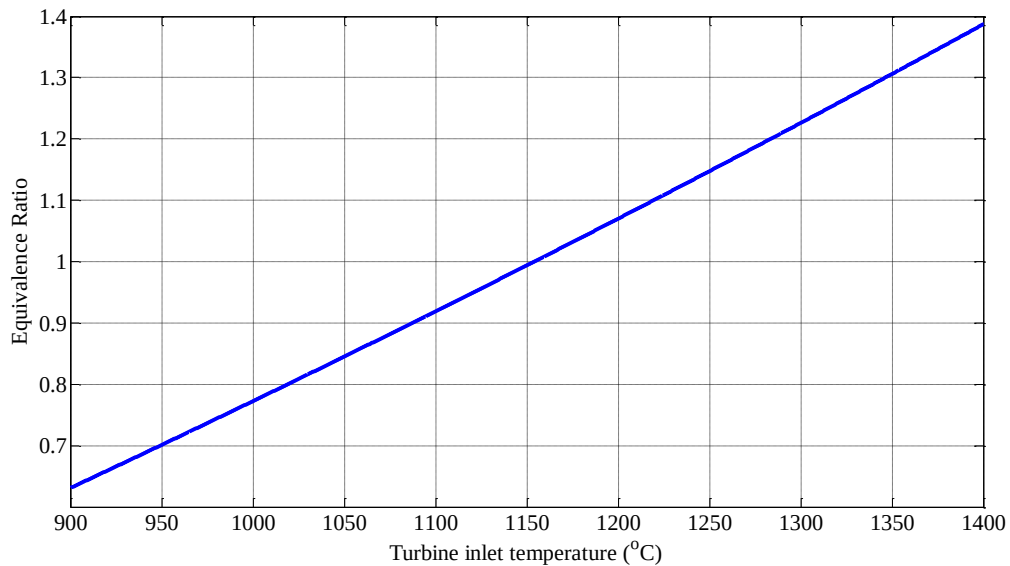


Figure 22 The effects of varying TIT on the equivalence ratio

As the combustion becomes rich, the unburned fuel might be presented at the exit of the combustion process which means that the fuel is not fully utilized to produce the require amount and quality of the hot gases.

Figure 22 presents the relation between the air equivalence ratios with TIT. The results show that, the equivalence ratio increases as the TIT rises. Figure 22 shows also an important point which is as the TIT is more than 1150 °C the combustion condition is converted from lean to rich combustion accordingly higher thermal NO_x emission will be produced as it is function of Combustion temperature. Moreover to NO_x emission, as the combustion becomes richer that means an excess of fuel will be presented in the exhaust gases. There will be unburned hydrocarbons in the exhaust gases due to incomplete combustion. Another produce which is formed due to incomplete combustion is CO because of no enough Oxygen to create CO_2 .

In order to meet the higher TIT along with lean combustion the best practical way is to increase the compression pressure ratio of the compressor eventually increase the compressor discharge air temperature which enters the combustion chamber as discussed earlier. As the air temperature enters the combustion chamber increases the amount of fuel consumption will decay. Finally, equivalence ratio will drop and the lean combustion will be achieved.

On other hand is the advantage of higher operating TIT is to achieve higher gas turbine efficiency and higher work net for the same compression ratio.

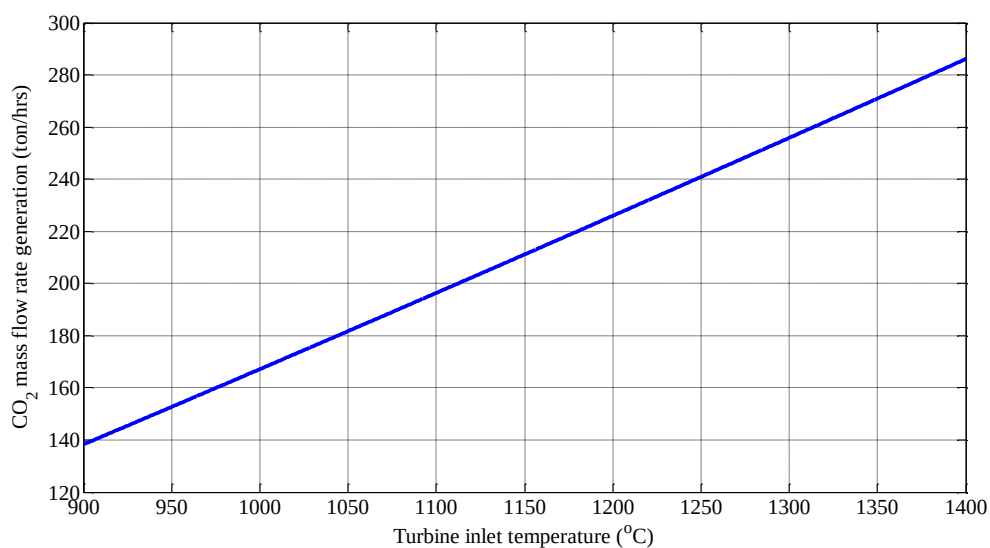


Figure 23 Relation between CO₂ mass flow rate generation and TIT

This increase in the fuel consumption it has an advantage point which is increasing the work net output from the gas turbine but unfortunately it has a negative effect on the environment by increasing rapidly the CO₂ gas that generated from the combustion chamber as the combustion process becomes rich combustion as shown in Figure 22. The CO₂ gas generated from the combustion chamber is shown in Figure 23. The results show that, CO₂ gas generated from gas turbine is increased by 105.68% by varying the turbine inlet temperature from 900 °C to 1400 °C.

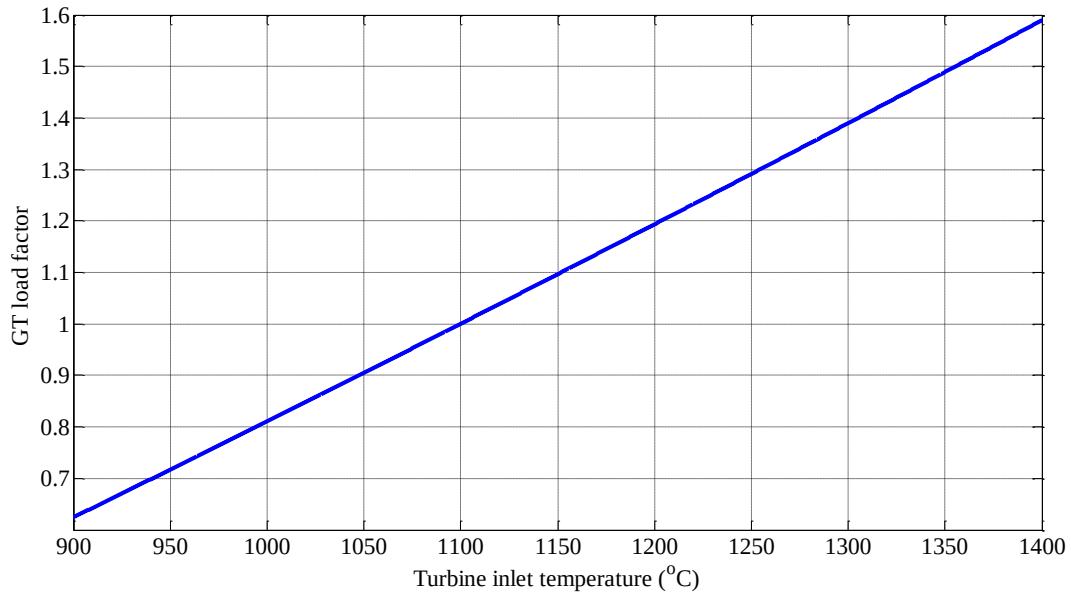


Figure 24 Relation gas turbine load factor and turbine inlet temperature

Figure 24 shows the effect of varying the turbine inlet temperature on the gas turbine power output, as it is expected the gas turbine power output is increasing as the turbine inlet temperature increases providing that the compressor pressure ratio is held constant. Figure 24 shows that the gas turbine power output relative to the gas turbine power output at the design turbine inlet temperature (1100°C in our case). The results show that, the gas turbine power output is increased from 0.624 of design value (167.59 MW in our case) to 1.589 of design value with rate of 0.323 MW/ °C. which means the gas turbine power output is boosted by more than 161.7 MW over the range of turbine inlet temperature investigated.

The turbine inlet temperature has a significant effect on the power output of gas turbine; the power gain is around 32.3MW for every 100°C increased the turbine inlet temperature.

In spite of the gas turbine power enhancement, the fuel gas consumption is getting increased rapidly as shown in Figure 23. Hence, the only method is being used now a day to reduce the fuel consumption is to increase the pressure ratio so that the air inlet temperature to combustion chamber is increased and result the gas consumption reduces.

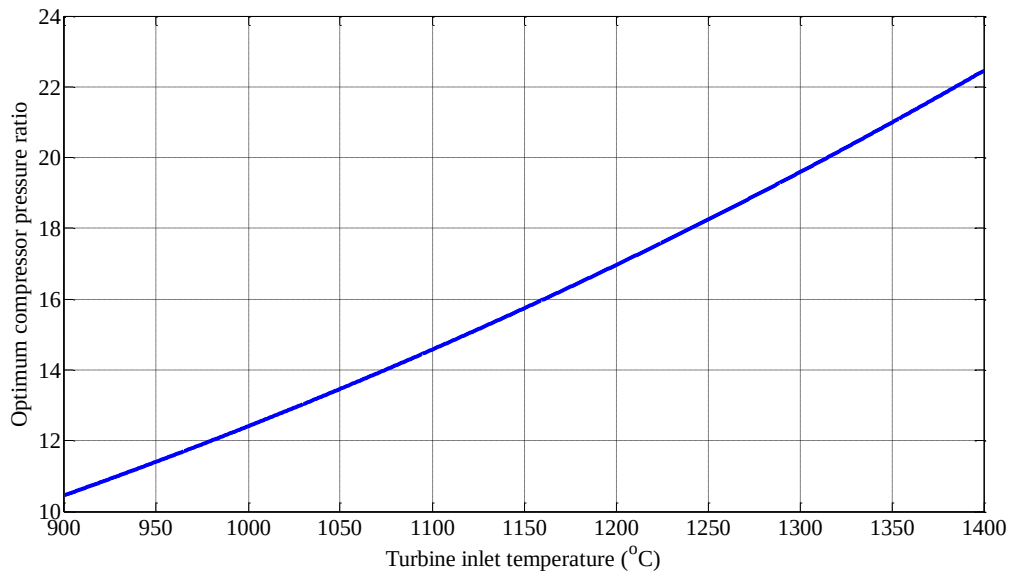


Figure 25 Optimum compressor pressure ratio at different TIT

The compressor pressure ratio has a significant effect on the gas turbine efficiency and power out. Therefore, the optimum pressure ratio has been derived to satisfy the maximum power out from the gas turbine and without any deteriorate the gas turbine thermal efficiency at certain operating conditions. The optimum pressure ratio is presented in Figure 25, it is obvious that the optimum pressure ratio is increased as the turbine inlet temperature increases. Therefore the turbine inlet temperature improvement depends on two factors which are the material technology and the compressor pressure ratio. The material technology allows us to increase the turbine inlet temperature to certain limit that withstands the thermal stress causes by the hot gases flow. The second factor is the compressor pressure ratio as shown in Figure 25 is increased as the turbine inlet temperature increases, the pressure ratio makes the turbine operates in effective manner by optimizing the fuel consumption that used to operate the gas turbine. Finally, the increase of turbine inlet temperature is preferable when the pressure ratio is high enough in order to maximize the power output without any scarification in thermal efficiency.

The results show that, the steam turbine power output increases too due to the turbine inlet temperature increases, the steam turbine power output is boosted by more than 169 MW over the range of turbine inlet temperature investigated.

The combined cycle overall efficiency will be improved at higher turbine inlet temperature due to the increases of the exhaust turbine temperature which leads to increase the steam flow generation through the heat recovery steam generator and finally of course increase the steam turbine power output. The combined cycle overall thermal efficiency has shot up from 42.92% to 52.99% over the range of turbine inlet temperature variation.

4.3.2 EXERGY ANALYSIS

The effect of varying turbine inlet temperature on exergy is investigated in this section. From the energy viewpoint, it has shown that higher operating temperature always has positive benefits on both gas turbine and combined cycle power plant.

As the compressor is not subjected to any changed while investigating the effect of turbine inlet temperature changes, therefore the exergetic efficiency would not be differed from the early results which have been seen in the previous section. Hence the fuel consumption is increased as turbine inlet temperature increases therefore the combustion chamber exergetic efficiency is predictable to change.

Figure 26 displays how the exergy destruction, exergetic efficiency, exergy destruction ratio and exergy destruction rate for combustion chamber, gas turbine, heat recovery steam generator and steam turbine are affected by varying the turbine inlet temperature.

The combustion chamber exergy destruction is proportionally increased with the turbine inlet temperature, as the major source of exergy destruction in the combustion chamber is the chemical exergy. Accordingly, the fuel increases when the turbine inlet temperature is increased which will cause more chemical exergy destruction within the combustion chamber.

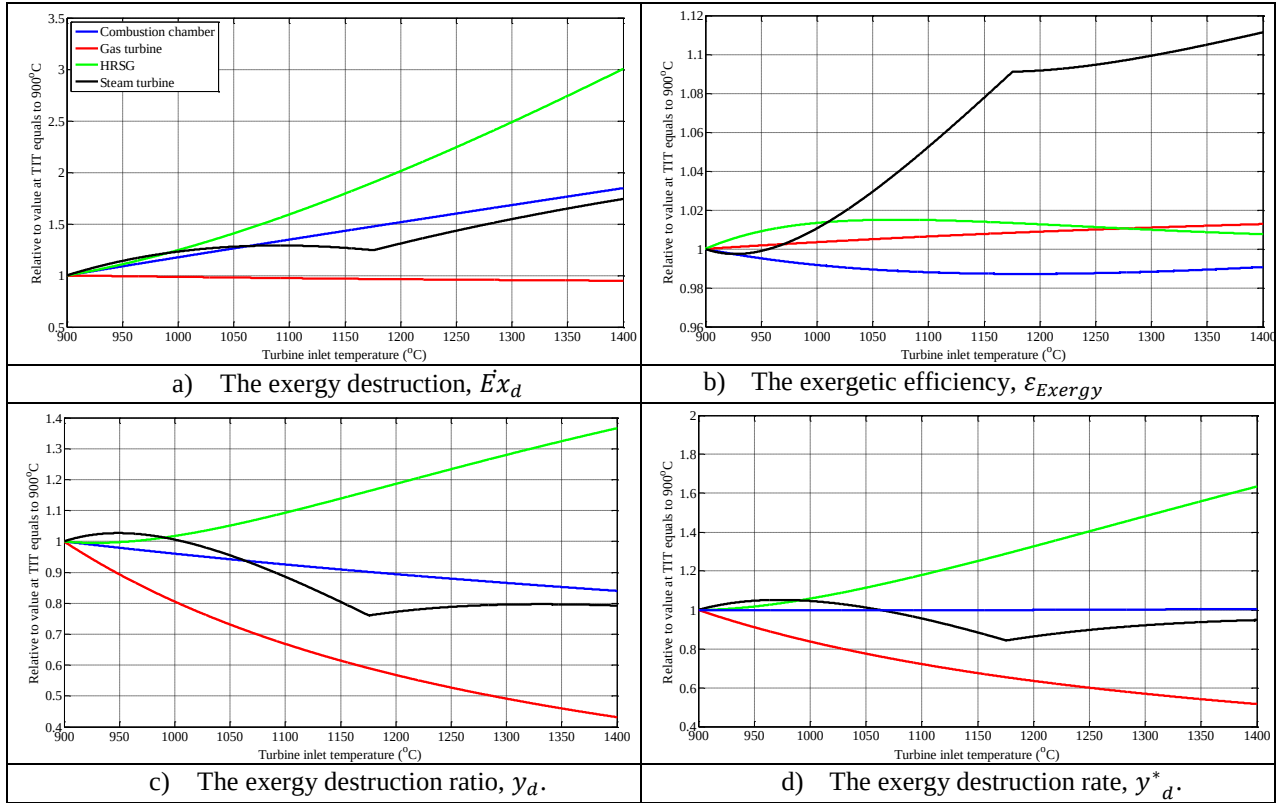


Figure 26 Combustion chamber, gas turbine, heat recovery steam generator and steam turbine exergy destruction analysis vs. turbine inlet temperature

The results shows that, the HRSG and steam turbine exergy destructions are increasing as the turbine inlet temperature increases while the gas turbine exergy destruction remains almost constant. The steam turbine exergetic efficiency is increasing by more than 11.5% over the range of TIT investigated while the gas turbine and HRSG are increasing by less than 2%. The reason for that the live steam temperature increases rapidly as the TIT increases.

The results show that, the exergy destruction ratio and exergy destruction rate of gas turbine are decreasing by 55% and 50% respectively when the TIT increases. Contradictory, the HRSG behaves as the exergy destruction ratio and exergy destruction rate are increasing by 35% and 63%.

4.4 EFFECTS OF COMPRESSOR PRESSURE RATIO

The third simulation varies the compressor pressure ratio from 9 to 22. This simulation is intended to show the effect of compressor pressure ratio has on the performance of combined cycle system. As the compressor pressure ratio increases, the air leaving the compressor becomes hotter, therefore less fuel mass flow rate is required will be predicted to reach the desired turbine inlet temperature.

In this section the effect of compressor pressure ratio variation has been analyzed by using two approaches:

- Energy analysis.
- Exergy analysis.

4.4.1 ENERGY ANALYSIS

The compressor performance at different compressor pressure ratio is presented in Figure 27. The results show that, as the pressure ratio increases, the air leaving the compressor becomes hotter; therefore less fuel is required to reach the desired turbine inlet temperature, for a fixed gas flow to the gas turbine.

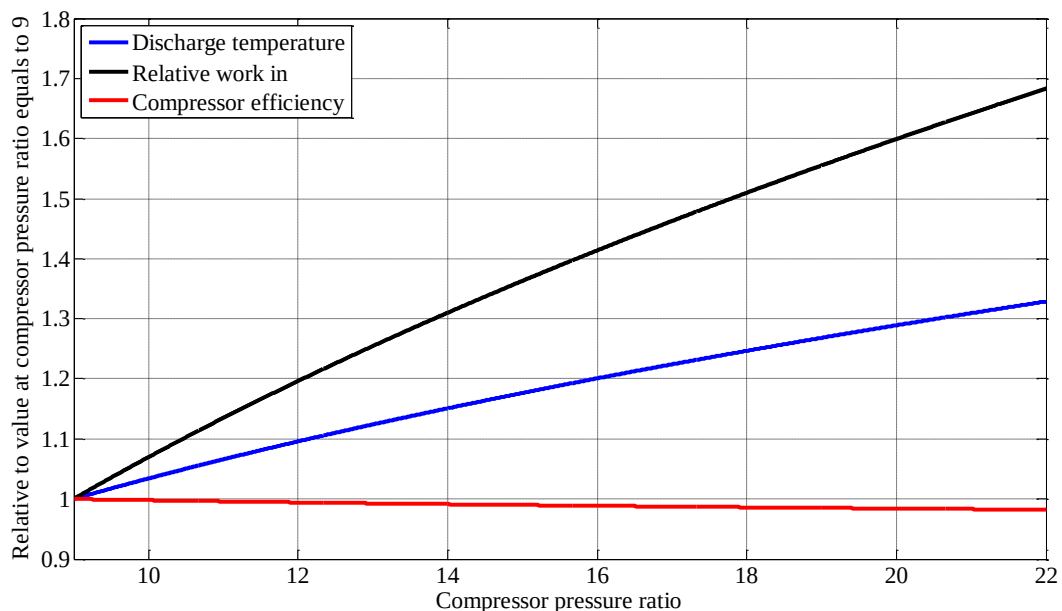


Figure 27 Compressor performances at different pressure ratio

The work required in the air compressor is rapidly increased with pressure ratio; it has shown that the work in required increases by more than 68% over the range of pressure ratio investigated. The results show that, the compressor efficiency is not much affected by the variation of compressor pressure ratio, as shown in Figure 27, but in general it drops slightly.

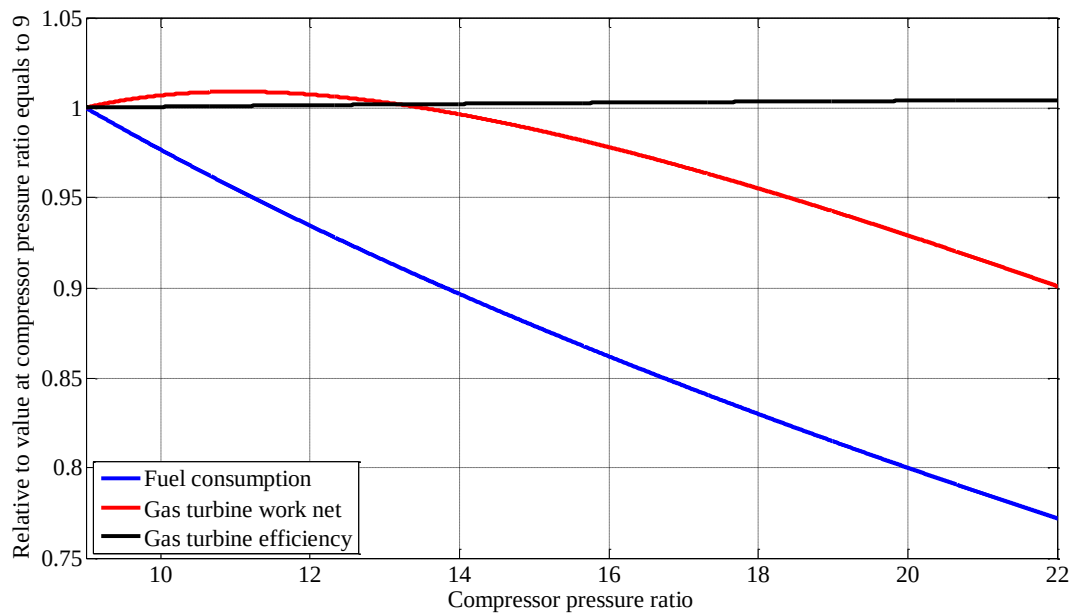


Figure 28 Relation between compressor pressure ratios with fuel consumption, gas turbine efficiency and gas turbine work net

Figure 28 shows the relation between compressor pressure ratios with fuel consumption, gas turbine efficiency and gas turbine work net output. The results show that, as the compressor pressure ratio is increasing the fuel consumption decreases. The study shows that the fuel consumption is decreased by 22.78% when the compressor pressure ratio is varying from 9 to 22.

Since compressor discharge temperature is a function of the compressor pressure ratios and it is increased proportionally with compressor pressure ratios as it is shown in Figure 27. Therefore, the temperature of air enters the combustion chamber is increased as the pressure ratio increases; consequently the amount of fuel to achieve the required turbine inlet temperature is reduced as the turbine inlet temperature is maintained constant. Sanjay (Sanjay, 2011) in his work has discussed the effect of pressure ratio on

fuel consumption and the same was observed. The effect of compressor pressure ratio variation on fuel air equivalence ratio is presented in Figure 29. The results show that, the change in fuel air equivalence ratio is radically affected by the pressure ratio. The fuel air equivalence ratio is dropping from 1.028 to 0.7942 which means 22.74%. This means, the combustion process becomes leaner when the pressure ratio increasing, so this is a clear indication that increasing turbine inlet temperature is preferred with higher compressor pressure ratio.

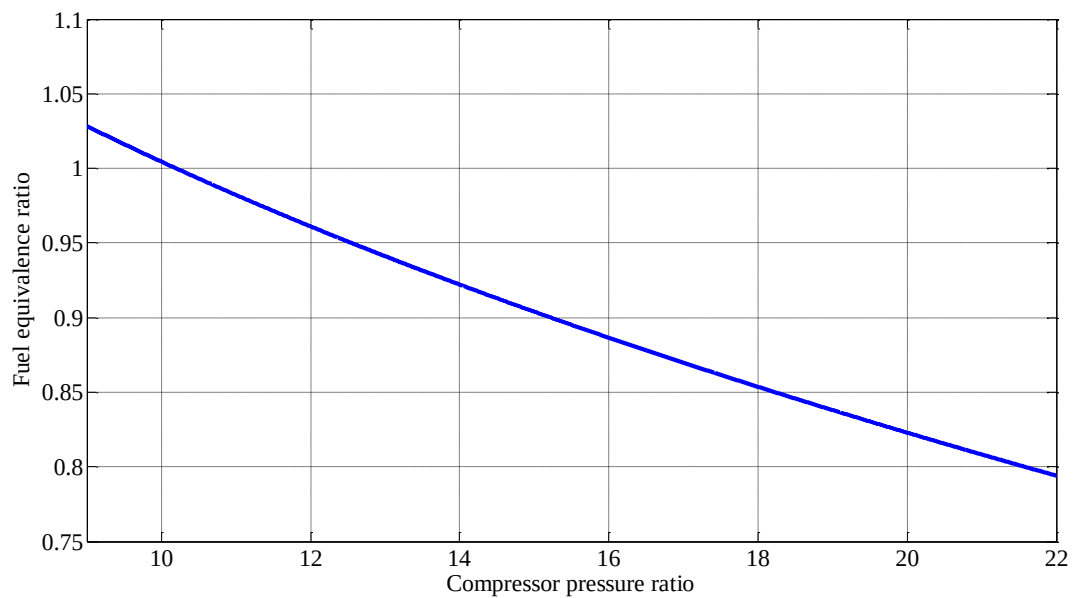


Figure 29 Effect of compressor pressure ratios on fuel equivalence ratio

The higher compressor pressure ratio causes a larger drop in entropy when expansion occurs in the gas turbine, producing more work but exhausting the gas at lower temperature. This lower gas temperature causes less steam production in the heat recovery steam generator, therefore lowering the outputs of steam cycle.

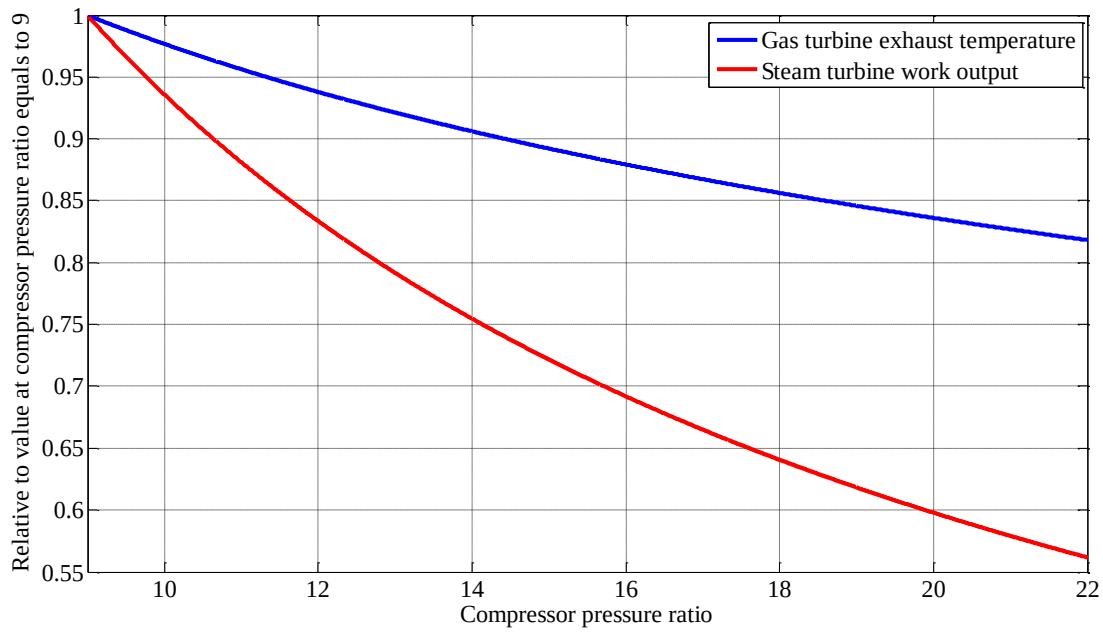


Figure 30 Relative value of GT exhaust temperature and ST output at different pressure ratio

The results show that, the gas turbine exhaust temperature drops by 18.17% when the pressure ratio is varying from 9 to 22. The effect of this drop is noticed on the steam turbine work output as it is dropped by more than 43.81%.

4.4.2 EXERGY ANALYSIS

The effect of varying compressor pressure ratio on exergy is investigated in this section. The exergy destruction in air compressor is noticed to be more at higher compressor pressure ratio due to increased compression work in as shown in Figure 31.

In combustion chamber, exergy destruction 27.6% is lower at higher pressure ratio. Gas turbine exhibits higher level of exergy destruction 67.3% at higher value of pressure ratio due to higher expansion ratio resulting in more entropy generation.

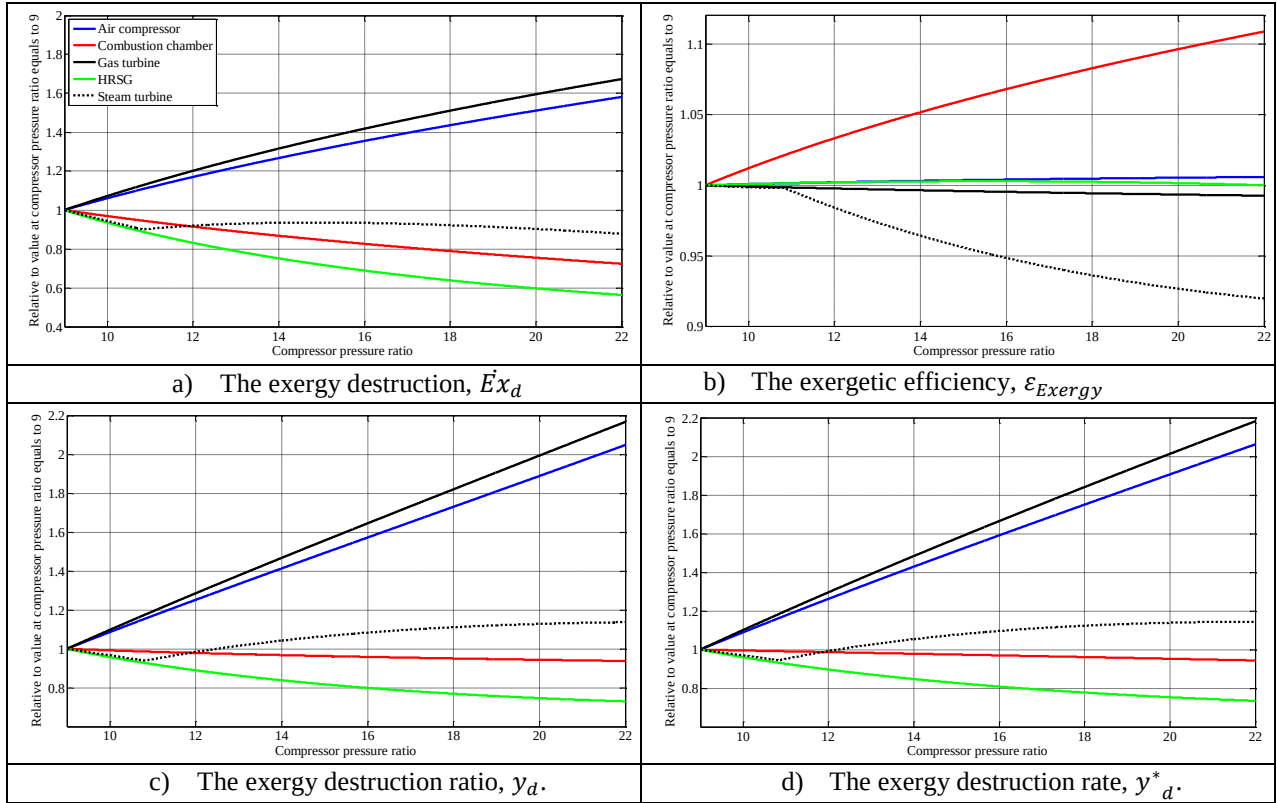


Figure 31 Air compressor, combustion chamber, gas turbine, heat recovery steam generator and steam turbine exergy destruction analysis vs. compressor pressure ratio

In HRSG, exergy destruction is higher at lower pressure ratio, this is attributed to the fact that at lower pressure ratio lower expansion ratio provides more heat energy available to HRSG.

The steam turbine exergetic efficiency is dropped by more than 7% at higher value of compressor pressure ratio. The reason for this is the live steam temperature is dropped due to reduction gas turbine temperature. In combustion chamber the exergetic efficiency is increased by more than 11% due to reduction in the fuel consumption.

The results show that, the air compressor and gas turbine exergy destruction ratio and rate are increasing by more than 100% due to higher expansion ratio resulting in more entropy generation. In the HRSG, the exergy destruction ratio and rate are decreasing by more than 22% as shown in Figure 31.

4.5 EFFECTS OF HEAT LOSS WITHIN COMBUSTION CHAMBER

In forth simulation the heat loss thru combustion chamber is varied from 0 to 5%. This simulation is intended to show the effect heat loss within combustion chamber has on the performance of combined cycle system.

In this section the effect of heat loss within combustion chamber variation has been analyzed by using two approaches:

- Energy analysis.
- Exergy analysis.

4.5.1 ENERGY ANALYSIS

Figure 32 shows the effects of percentage of heat losses within the combustion chamber on the fuel consumption, gas turbine cycle efficiency and turbine work net compare to the case of nil heat losses. The heat losses is accounted by introducing the term $(1 - \%HL)$ in the left hand side of heat addition from the fuel Eq. 21. The fuel consumption increases linearly with increases of the percentage of heat losses. This is due to more fuel will be required to achieve the desired turbine inlet temperature. The extra fuel is added to compensate for the heat losses from the combustion chamber. In the present study, it has been assumed that the percentage of heat loss is varying from 0% to 5%. When the heat losses are 5% the fuel consumption increases by 5.87%, as shown in Figure 32.

As the fuel consumption is increasing, with the increasing of the percentage of heat losses, the gas turbine work net is increased too. Figure 32 shows that the relation between the percentage of heat loss and the gas turbine work net. The gas turbine work net increases slightly by 0.49% as the heat losses is increased by 5%. This small increase in gas turbine work net is very negligible. It is 0.833 MW while the gas turbine full power is 167.27 MW. This is due to the increase of fuel mass flow rate enters to the

gas turbine combustion chamber. This increase is to compensate the heat losses and consequently fix the inlet gas temperature to the turbine.

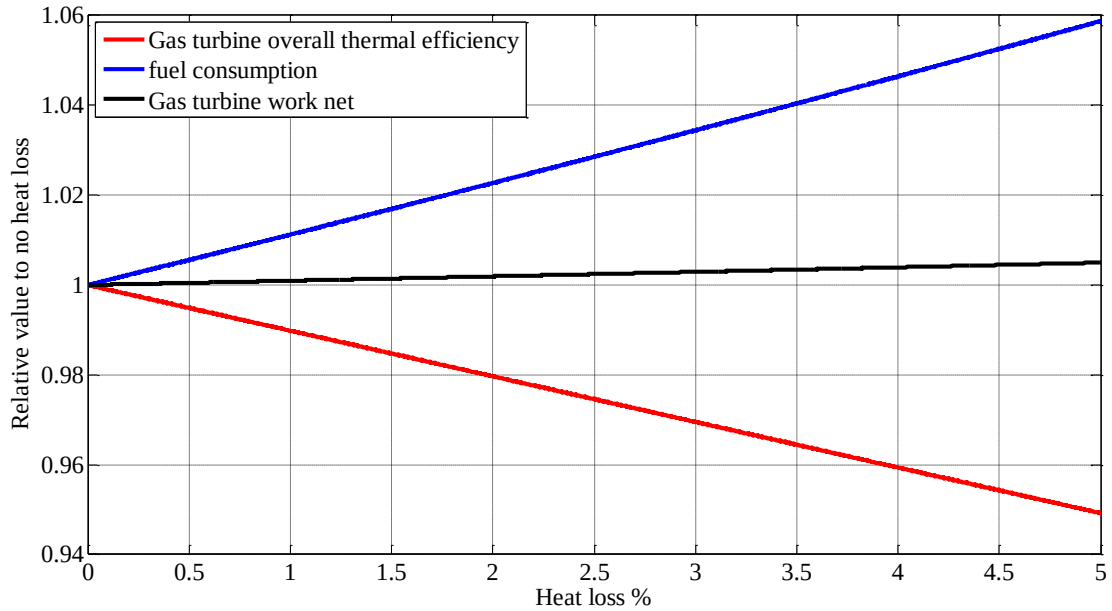


Figure 32 The effect of heat losses within combustion chamber on GT performance

In addition to that, Figure 32 shows that there is a drop in the cycle thermal efficiency with the increase in the heat losses. The gas turbine overall thermal efficiency drops by 5.076% when the heat loss is 5%. This observation reflects in the importance of reducing the heat losses in the combustion chamber by improving the insulation of the combustion chamber.

4.5.2 EXERGY ANALYSIS

The combustion chamber exergy destruction is expected to increase as the heat loss increases due to fuel consumption increases as shown Figure 32. This can be observed in Figure 33, this increase is due the chemical reactions which are the most significant source of exergy destruction. The exergy destruction in the combustion chamber is mainly affected by the fuel mass flow rate. The results show that the exergy destruction

of the combustion chamber is increased by 20.4% when the heat loss within the combustion chamber is varied from 0 to 5% as shown in Figure 33.

In addition to that the CO₂ emission increases from 192.3 to 202.7 ton/hr, therefore the heat loss within the combustion chamber has significant effect on both the exergy destruction and the environment as the CO₂ generation is tended to increase.

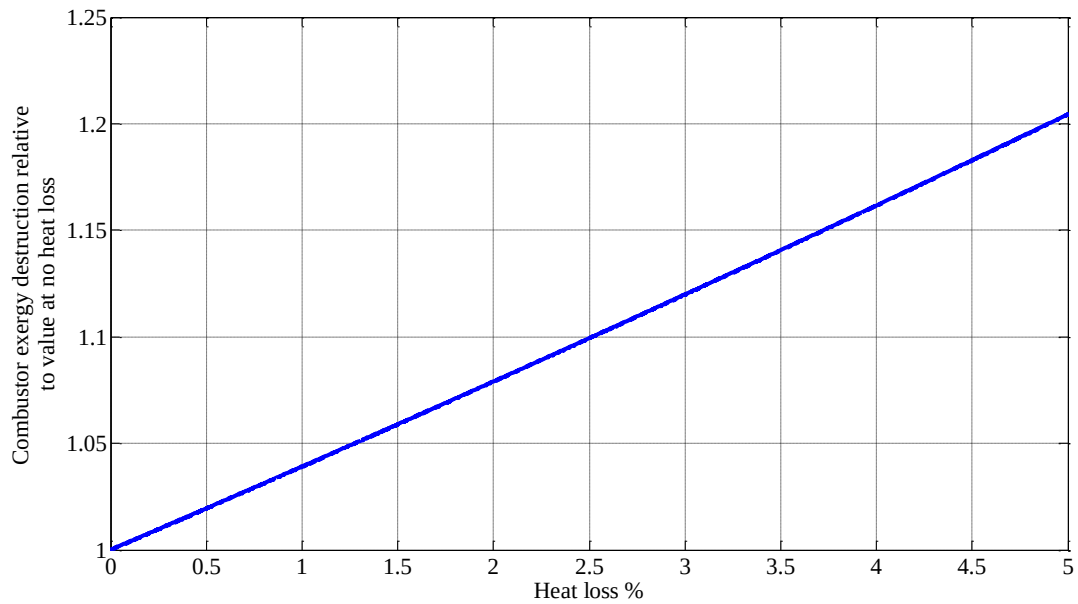


Figure 33 The effect of heat losses within combustion chamber on combustion chamber exergy destruction

The combustion chamber exergetic efficiency is slightly affected by the heat loss variation, it is noticed that the efficiency increases from 73.36 to 74.02% which less than 0.9% changes.

The results show that, all other components within the combined cycle power plant their exergy performance are remained constant.

4.6 EFFECTS OF GAS TURBINE WITH INTAKE AIR COOLING SYSTEM

The inlet fogging system is a method of cooling compressor inlet air by spraying a tiny mist at the compressor inlet. Determining the required amount of spray water is very important to achieve the highest results without affecting the compressor. The over spray water can be caused a serious damage in the compressor blades specially the first stages. The size of the water droplet is determined by the size of the spray nozzles and the operating pressure of the spray water, big size of droplet can be caused erosion in the compressor blades and in worse scenario it might cause a vibration in the machine.

The inlet fogging system consists of ten high pressure water pumps that design to boost up the spray water from 5 bar to above 200 bar, then the water is fed to the air intake filter house through high pressure pipe lines and gets squirted through number of nozzles which are distributed in such way that the water gets mixed properly with air. Usually, demineralized water is used as spray water because it has had it is mineral ions removed to prevent the spray nozzles blocked. The basic function of the control system is to correctly inject the appropriate amount of fog to attain the desired ending temperature near the inlet to the compressor (typically approaching the wet bulb temperature).

The required amount of spray water can be defined as function of ambient condition such as ambient temperature (T_o) and relative humidity (RH_o) as shown in Eq.80:

$$\dot{m}_{fog} = f(T_o, RH_o) \quad (80)$$

The inlet fogging cooling is a constant wet-bulb process that takes advantage of the latent heat of vaporization to cool the air stream and it is more efficient at drier and hot climates.

From the ambient condition the wet-bulb temperature can be calculated as an explicit function of ambient temperature and relative humidity, given by the mathematical relation taken from (Stull, 2011) as shown in Eq. 81:

$$T_{wb} = T_o \tan^{-1}[0.151977(RH_o\% + 8.313659)^{0.5}] + \tan^{-1}(T_o + RH_o\%) - \tan^{-1}(RH_o\% - 1.676331) + 0.00391838(RH_o\%)^{1.5} \tan^{-1}(0.023101 RH_o\%) - 4.686035 \quad (81)$$

Eq. 81 is valid for a pressure of 101.325 kPa and for relative humidity between 5% and 99% and for air temperature between -20 °C and 50 °C, except for situations having both low humidity and cold temperature.

There are three types of inlet fogging system which are described as follow:

1. Saturated fogging when the air is reaching saturation at the wet bulb temperature before entering the compressor.
2. Over spray if there are water droplets remaining after the air flow reaching saturation at the wet bulb temperature, the remaining droplets will enter the compressor.
3. Inter-stage fogging when the water is sprayed after the air enters the compressor.

In summary, inlet fogging system has the following effects (Cataldi, et al., 2006):

1. Water droplets evaporate by absorbing heat from the air flow and cause increase in air density.
2. Increase of the gas-phase mass flow rate along the compressor passage because of the evaporation of the water droplets.
3. Increase in C_p due to change in gas-phase composition.
4. Change in volumetric flow rate of compressor with increase in the front stages and decrease in the back stages, causing alteration of the velocity triangles and consequently backward shifting of the pressure build-up along the compressor.
5. Aerodynamic load transfer toward the back stages.
6. Increased airfoil trailing edge flow angle deviations on wet blades, which result in weaker stage characteristics for the first blade rows.
7. Changes in compressor characteristics according to surge limit.

In fifth simulation the gas turbine with intake air cooling system is implemented. This simulation is intended to show the effect cooling intake air has on the performance of combined cycle system. The ambient temperature varies from 15 °C to 45 °C while the

ambient relative humidity is remained constant at 60%; the results were compared with ambient temperature effect results which were discussed in Section 4.2.

In this section the effect of gas turbine with intake air cooling system has been analyzed by using two approaches:

- Energy analysis.
- Exergy analysis.

4.6.1 ENERGY ANALYSIS

The inlet fogging has been widely noticed as a method of cooling inlet air of gas turbine, which take the advantage of the evaporative heat of water to cool down the intake air. This method is more applicable where the gas turbine is operated at hot ambient condition because it drops the inlet air temperature to the compressor; theoretically it reaches the wet-bulb temperature by spraying a tiny mist of water.

In practical the fogging system is consisted of number of high pressure water pumps that use to supply the water through a very small nozzles to the compressor inlet. The number of pumps that should come in operation is depending on the ambient condition such as the ambient temperature and relative humidity. The amount of spray water required to achieve the wet bulb temperature is noticed varying as the ambient temperature varies as shown in Figure 34.

The amount of required spray water to achieve the wet bulb temperature at the entrance of air compressor varies with ambient temperature; it is obviously shown that the amount of spray water increases proportionally with ambient temperature in case the ambient relative humidity remains constant at 60%.

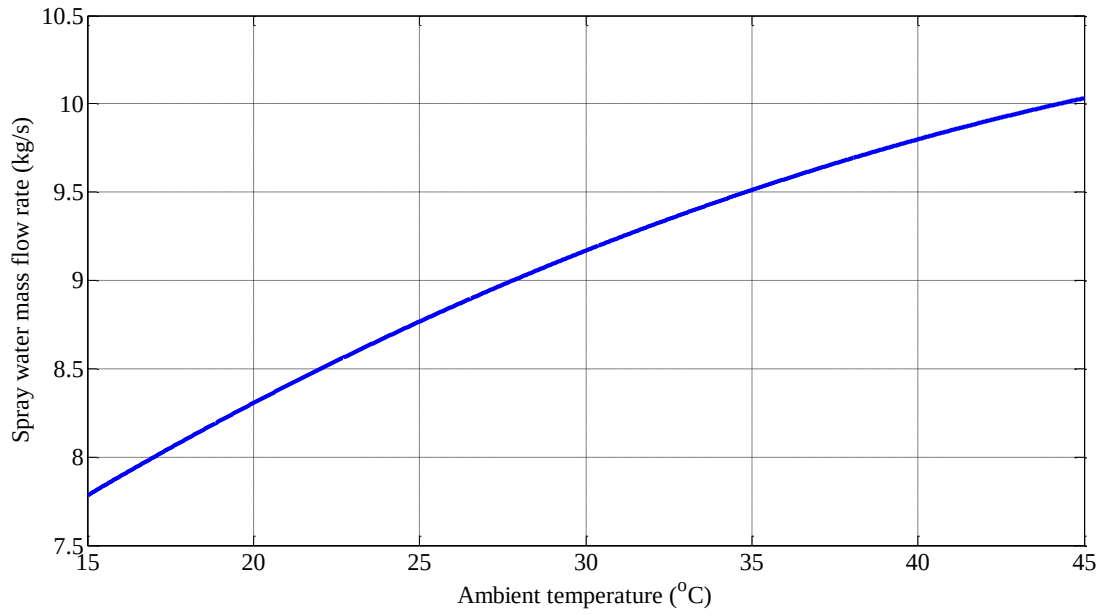


Figure 34 The amount of required spray water vs. ambient temperature

Figure 34 shows the relation between the amounts of spray water mass flow rate in kg/s and ambient temperature variation, the spray water required to achieve the wet bulb temperature at compressor entrance is found to be 7.78 kg/s at 15 °C ambient temperature and increased to 10.03 kg/s at 45 °C. The air enters the air compressor is dramatically decreased due to the spray water as shown in Figure 35, the air inlet temperature drops from 15 °C to 10.52 °C and from 45 °C to 37.47 °C. Hence, the spray water has a significant effect on the inlet air as it is capable to drop the intake air temperature by minimum 5 °C as shown in Figure 35.

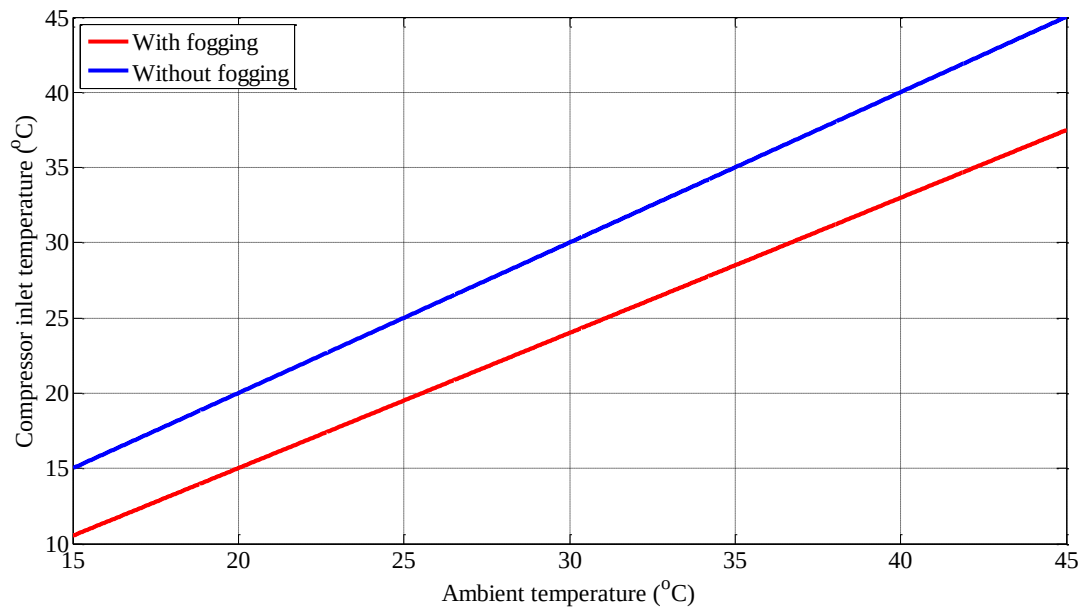


Figure 35 The effect of spray water system on the compressor inlet temperature

In addition to that the compressor works in is dropped by 1.51% at low ambient temperature and by 2.21% at high ambient temperature due to compressor inlet temperature drops. This drops in compressor work in can be increased the work net output of gas turbine cycle as the compressor work in is taken around 60% of the total gas turbine work output.

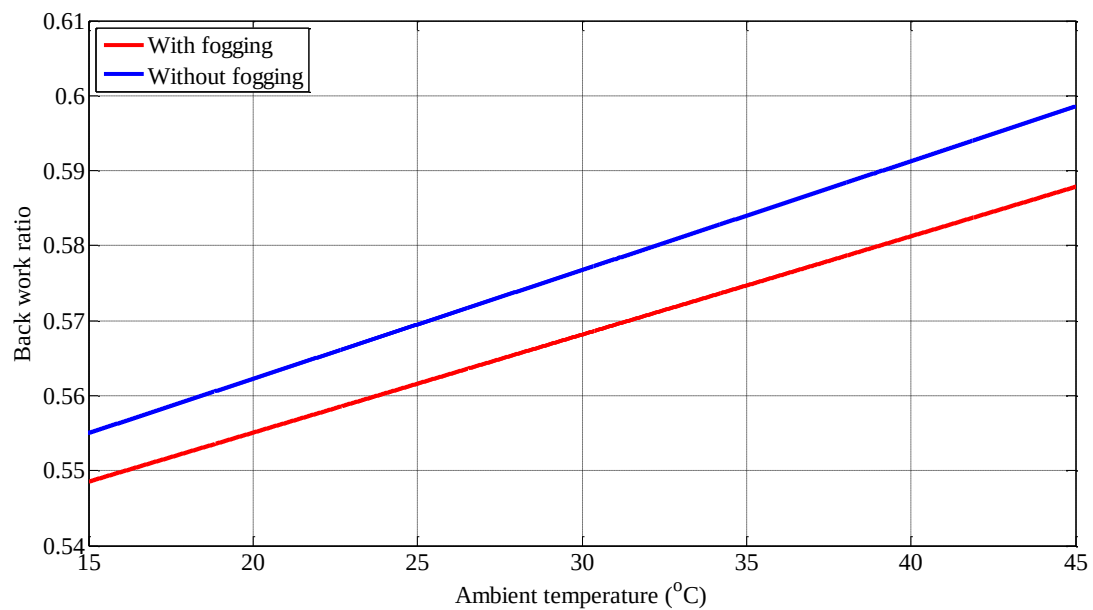


Figure 36 The effect of spray water fogging system on back work ratio

The effect of spray water fogging system on the gas turbine back work ratio is presented in Figure 36. The results show that, the back work ratio is increasing when the ambient temperature increases. The reason for that the air compressor work in requires to compress the air at higher temperature is more than at lower temperature due to air density. As shown in Figure 35, air enters the compressor is fallen down by 5 to 7°C when the spray water fogging system is in service. This will cause a drop in the compressor work in.

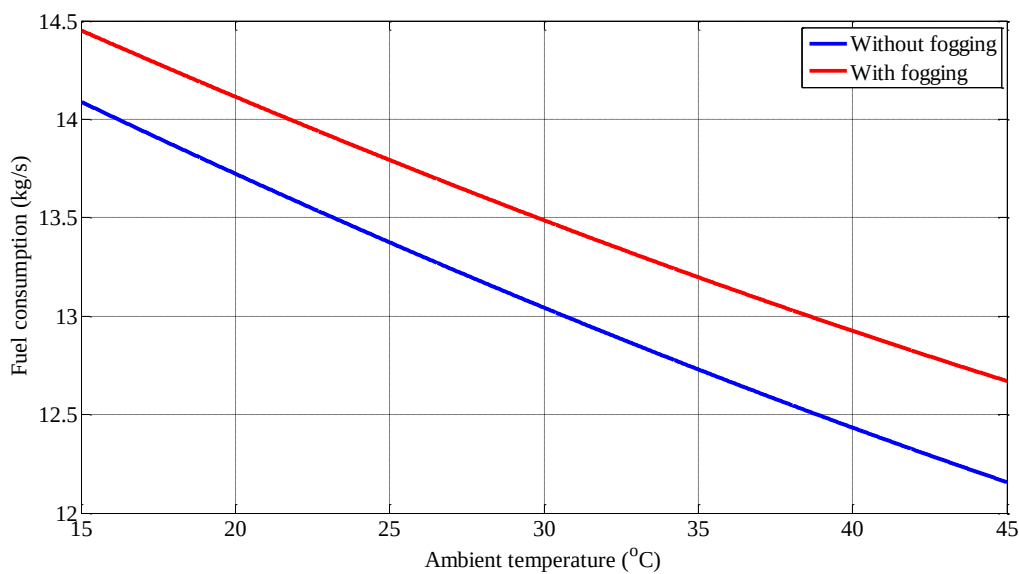


Figure 37 The effect of spray water fogging system on fuel consumption

In addition to the drop of compressor work in, the work output of gas turbine will increase as the temperature of air enters the compressor decreases due to the increase of fuel consumption as shown in Figure 37. This increase in the fuel consumption will lead to increase the work output of the gas turbine. Therefore the back work ratio will be reduced when the spray water fogging system is in operation.

The effect of spray water fogging system on the gas turbine power output is presented in Figure 38. Figure 38 compares between the gas turbine power output in case of spray fogging system is in service or not, it is clearly shown that the power output increases when the spray water fogging system is in service for the same ambient condition.

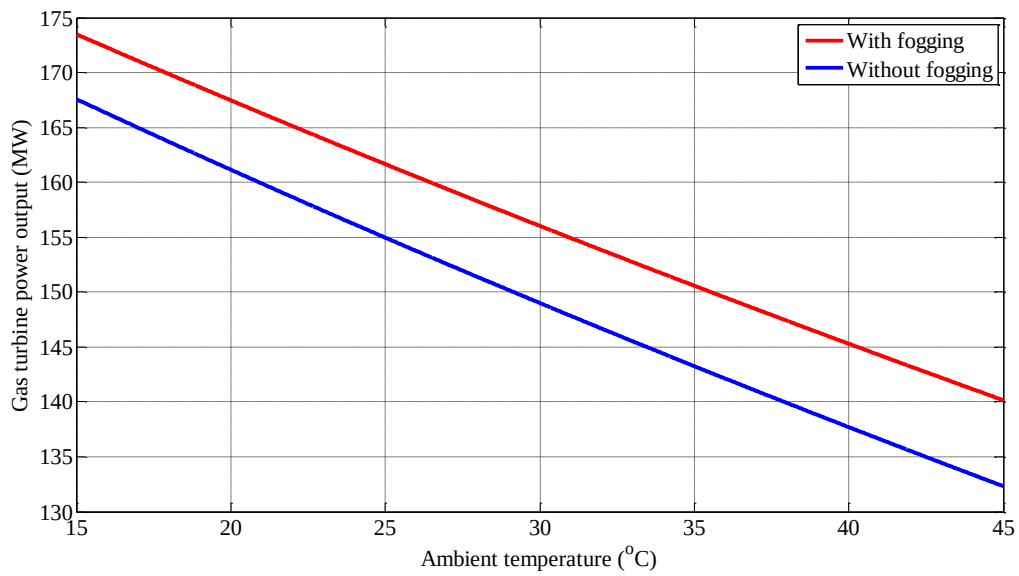


Figure 38 The effect of spray water fogging system on gas turbine power output

The turbine exhaust temperature was noticed affected by the spray water fogging system as shown in Figure 39, the turbine exhaust temperature is decreased by 2 to 3 °C. This reduction the exhaust temperature follows by very small reduction the steam generation. This reduction in steam generation is not worth mentioning as it is less than 0.5 to 1%.

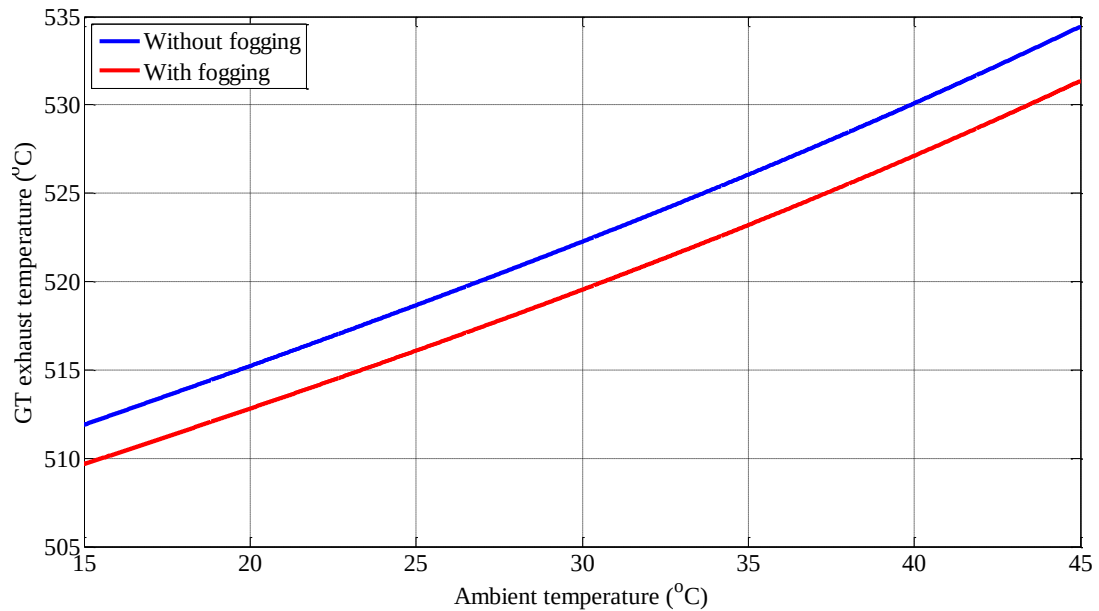


Figure 39 The effect of spray water fogging system on gas turbine exhaust temperature

The effect of spray water fogging system on steam turbine is very minor as the live steam temperature enters the steam turbine however remain constant and the flow steam remain constant too.

4.6.2 EXERGY ANALYSIS

Table 8 presents the exergy analysis of all components of the combined cycle power plant, as shown in Figure 1, at 42 °C and 60% relative humidity ambient condition without spray water fogging system in operation. It is obvious that the combustion chamber has the highest exergy destruction comparing to other components due to the chemical reaction are the most significant source of exergy destruction in the combustion chamber.

Table 8 Exergy destruction for all block elements at 42 °C and 60% relative humidity

Components	$\dot{E}_d$ [MW]	ε_{Exergy} [%]	y_d^* [%]	y_d [%]
Compressor	17.70	91.08	3.97	3.64
Combustors	183.30	72.58	41.11	37.70
GT	10.27	97.02	2.30	2.11
HRSGs	16.01	84.53	3.59	3.29
Deaerator	1.39	85.22	0.16	0.29
HPFWP	0.18	87.91	0.02	0.04
LPFWP	0.02	86.22	0.003	0.005
ST	12.57	90.96	1.41	2.58
ACC	17.10	34.36	1.92	3.52
CEP	0.02	86.46	0.003	0.005
Ejector	0.07	92.44	0.008	0.015
LP Pre-Heater	0.31	84.60	0.04	0.06

Table 9 presents the exergy analysis of all components of the combined power plant as shown in Figure 1 at 42 °C and 60% relative humidity ambient condition with spray water fogging system in operation. The effects of spray water system can be noticed by comparing between Table 8 and Table 9.

The exergy destruction of air compressor, combustion chamber and gas turbine are slightly higher when the spray water fogging system is in operation. The reason for air compressor exergy destruction increases is the air mass flow rate is increasing when the fogging system in operation due to the addition water adds to the system. The combustion chamber exergy destruction is increases due to fuel consumption increases when the fogging in operation as discussed earlier. The reason for gas turbine exergy destruction increase is that the hot gases mass flow rate and work output are increased. The HRSG and deaerator exergy destruction are decreased when the spray water fogging system in operation. The reason for that the steam temperature decreases due to gas turbine exhaust temperature is reduced when fogging system in service.

Table 9 Exergy destruction for all block elements at 42 °C with spray water fogging system in operation

Components	$\dot{E}_d$ [MW]	ε_{Exergy} [%]	y_d^* [%]	y_d [%]
Compressor	17.81	91.14	3.84	3.54
Combustors	187.86	72.81	40.50	37.37
GT	10.44	97.06	2.25	2.08
HRSGs	15.90	85.24	3.43	3.16
Deaerator	1.38	87.40	0.15	0.27
HPFWP	0.18	88.19	0.02	0.04
LPFWP	0.02	86.53	0.002	0.005
ST	13.22	90.65	1.42	2.63
ACC	23.46	26.08	2.52	4.67
CEP	0.02	86.67	0.003	0.005
Ejector	0.07	94.45	0.008	0.014
LP Pre-Heater	0.34	87.01	0.04	0.07

4.7 EFFECTS OF GAS TURBINE PART LOAD OPERATION

In sixth simulation the gas turbine part load operation is studied. This simulation is intended to show the gas turbine part load operation has on the performance of combined cycle system.

In this section the effect of gas turbine part load operation has been analyzed by using two approaches:

- Energy analysis.
- Exergy analysis.

4.7.1 ENERGY ANALYSIS

Figure 40 shows the operating part-loads performance obtained from the gas turbine model. The turbine operating parameters are controlled in such a way that turbine performance deterioration is minimized during the part-load operation. The VIGV position will be around 54% opening during the start up of the gas turbine and it will stay at its position during turbine loading up until turbine exhaust temperature reaches the limit point (550°C in the studied gas turbine). At that time the VIGV will be operating and opening more to maintain the turbine exhaust temperature within the allowable temperature limit.

Once the turbine inlet temperature reaches the maximum allowable temperature (1100°C in the studied gas turbine) the exhaust temperature will decay as the VIGV is varying to open more and admitting more air mass flow rate in the cycle.

In the present study a minimum gas turbine load of 40% is considered. This is the minimum load which a gas turbine can run in a case of combined cycle, as per the manufacture requirement.

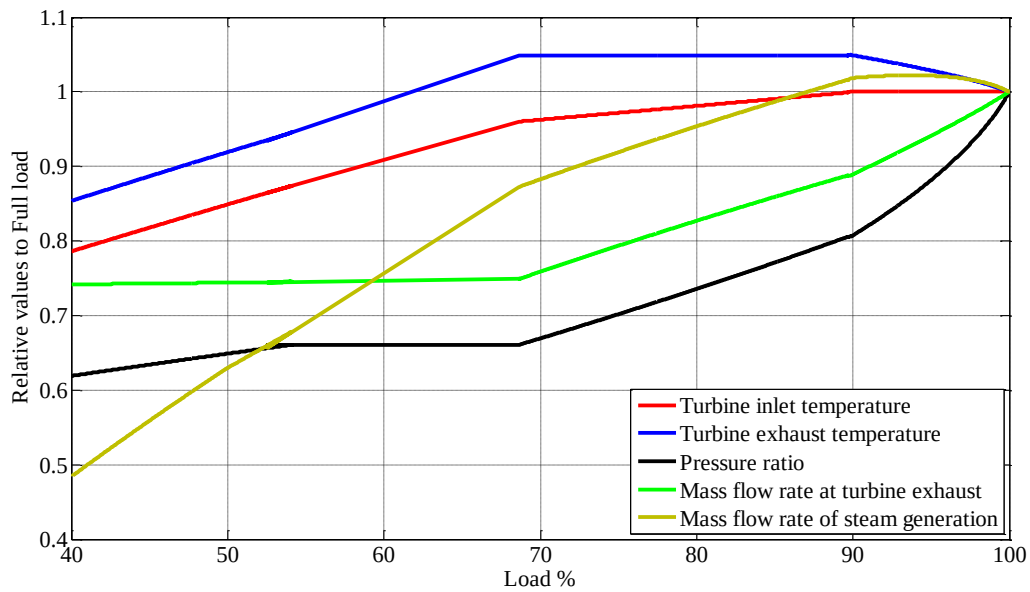


Figure 40 Part load performance of the gas turbine

The philosophy of a gas turbine is to control turbine inlet temperature and the exhaust temperature within the allowable limits of part load operation. Of course this will enhance the exhaust heat recovery for the bottom cycle as the gas turbine exhaust temperature is maintained at maximum value. This has been achieved by controlling the compressor inlet air mass flow rate by using VIGV variation when loads higher than 69% (Song, et al., 2002). The gas turbine operational characteristics can be classified into three zones as shown Figure 40. First zone occurs when the load is range of 0 to 69%. The turbine runs at conditions below the design value, and at this zone the VIGV will be opened at 53.85% and allows only 75% of the full load compressor inlet air mass flow rate. Second zone occurs when the load is range of 69% to 90%, and at this zone the VIGV open will vary in order to control the exhaust gas temperature so it does go beyond the maximum allowable temperature which is 550°C . To achieve that it admits more mass flow rate of air and fuel. Third zone occurs when the load is above 90% to full load, and at this zone the turbine inlet temperature and other parameter are equal the design values.

Figure 41 shows the relation between the ratio of the steam mass flow rate and exhaust gases mass flow rate. As it is shown the steam ratio starts to increases linearly up to 74% load. This is due to: (a) the VIGV will be remained at 53.85% open, (b) the

increase of fuel mass flow rate, and (c) a constant air mass flow rate. This will lead to increase the turbine exhaust gases temperature. Therefore, the steam mass flow rate will increase as shown in Figure 41.

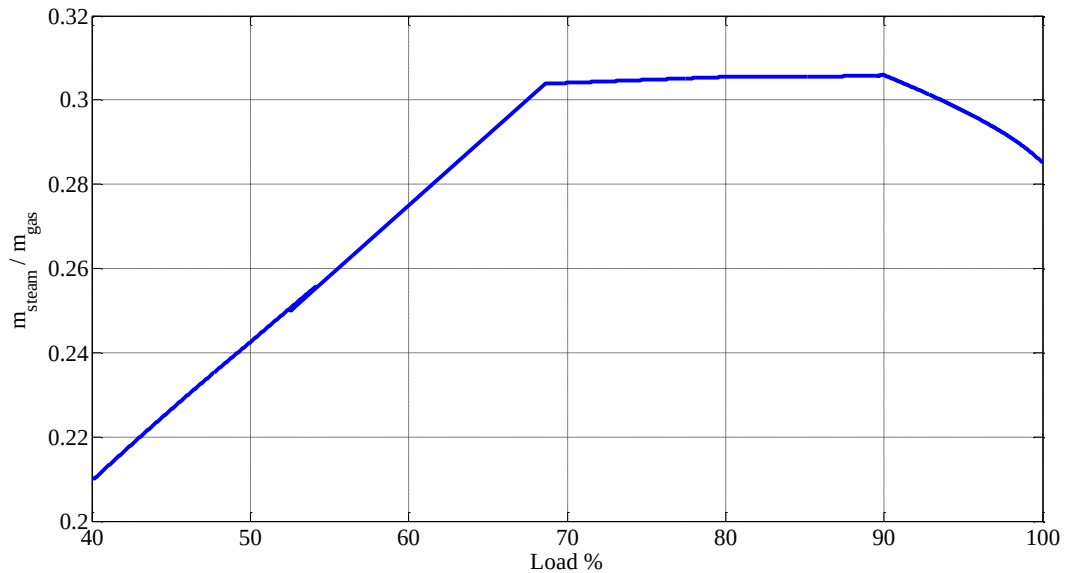


Figure 41 Ratio between steam generations and exhaust gas mass flow rate of gas turbine

For loads range from 74% to 94% the exhaust gas turbine temperature reaches the maximum allowable temperature as shown in Figure 40. In this range, the ratio between steam mass flow rate and exhaust gases mass flow rate reaches the maximum value, as shown in Figure 41.

For loads above 94%, where the gas turbine controller tends to fix the inlet gas turbine temperature at the design value, the exhaust temperature will decay as a result of admitting more air mass flow rate to the combustion chamber by opening the VIGV more until becomes fully open. Therefore, the steam to gas mass ratio, in this zone, decreases, as shown in Figure 41.

Figure 42 shows effect of the load on the ratio of steam turbine power to the total block power. The ratio is increasing as the load increases and it is reached to maximum at 74% of total cycle load. At this load the ratio of steam turbine power output is equal to 0.324 of the total block power output. By referring to Figure 40 it is cleared that at this point the turbine exhaust temperature is equal the maximum allowable temperature

which is 550°C and at the same time the exhaust gas mass flow rate is the minimum value. Consequently, the generated steam will reach its maximum temperature. In the other hand at loads greater than 74% the ratio of steam turbine power to the block one decreases as the load is increasing and this pattern agrees with the results of Figure 40.

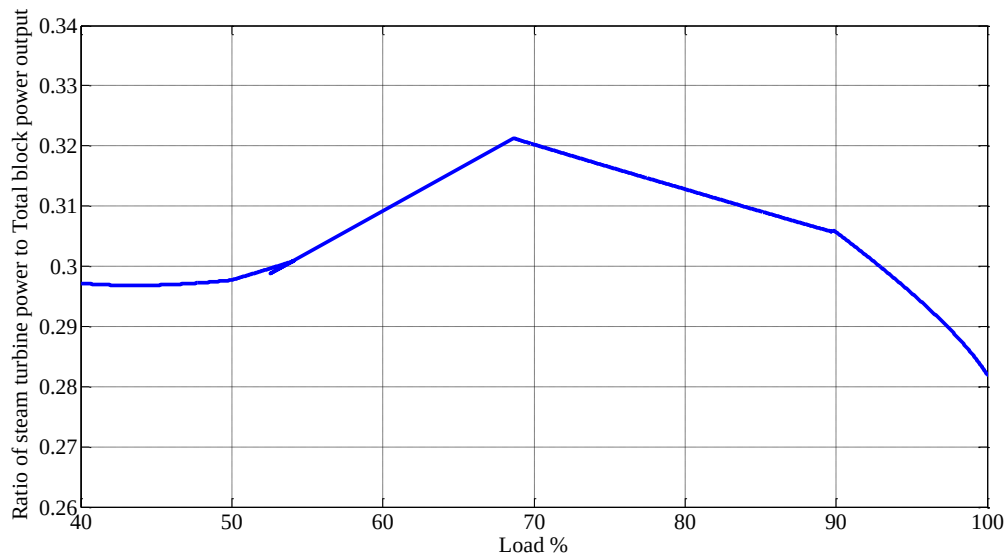


Figure 42 The effect of the load on the ratio of ST power to the total power

The heat rate of the block is equal 7397 kJ/kWh at full load and it is dropped slightly by 0.49% when the total cycle load is 94% . Then, it has increased linearly by 2.5% when it reaches 74% load. When the load is less than 74% a sudden increased is observed as shown in Figure 43.

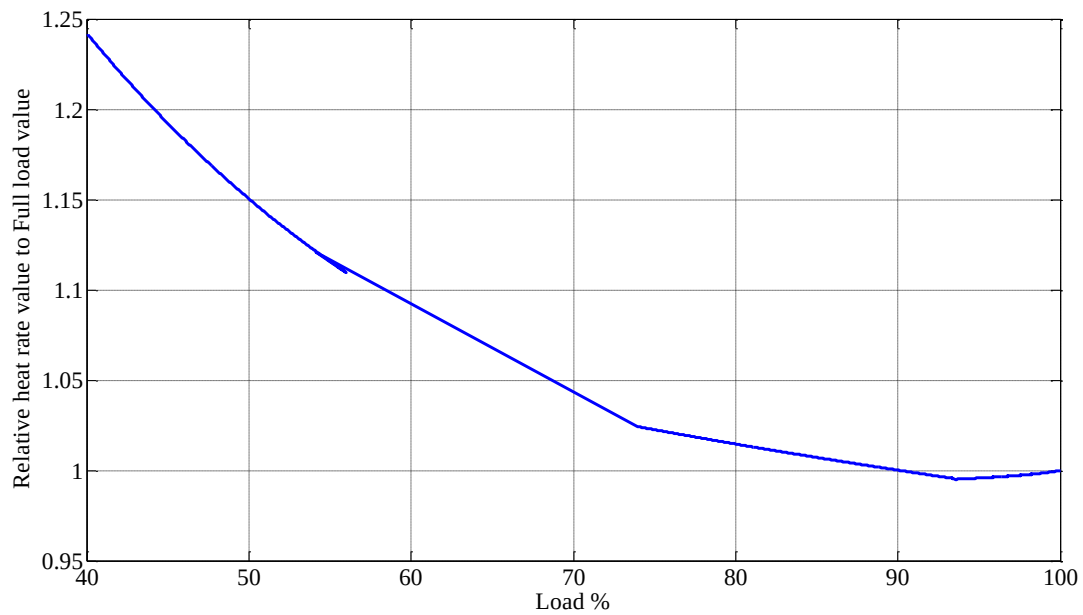


Figure 43 The effect of the load on the heat rate (Relative values)

Figure 43 indicates that for the range of load 74% to 100% is the optimum range which the cycle operates without significant increase in the heat rate relative to the full load value. The minimum heat rate occurs at the full load of the block. It is important to run the block at the minimum heat rate because this will accompanied with less fuel consumption. This is lead to less emission and less green house gases per kWh.

4.7.2 EXERGY ANALYSIS

Figure 44 shows the variation of relative exergy destruction rate in the compressor, combustion chamber and gas turbine with the load. The general trend, the relative ratio increases as the load increase. For the load of 69% of the full load the exergy destruction ratio relative to the full load one of air compressor, combustion chamber and gas turbine is 60.9%, 79.6% and 59.3%, respectively.

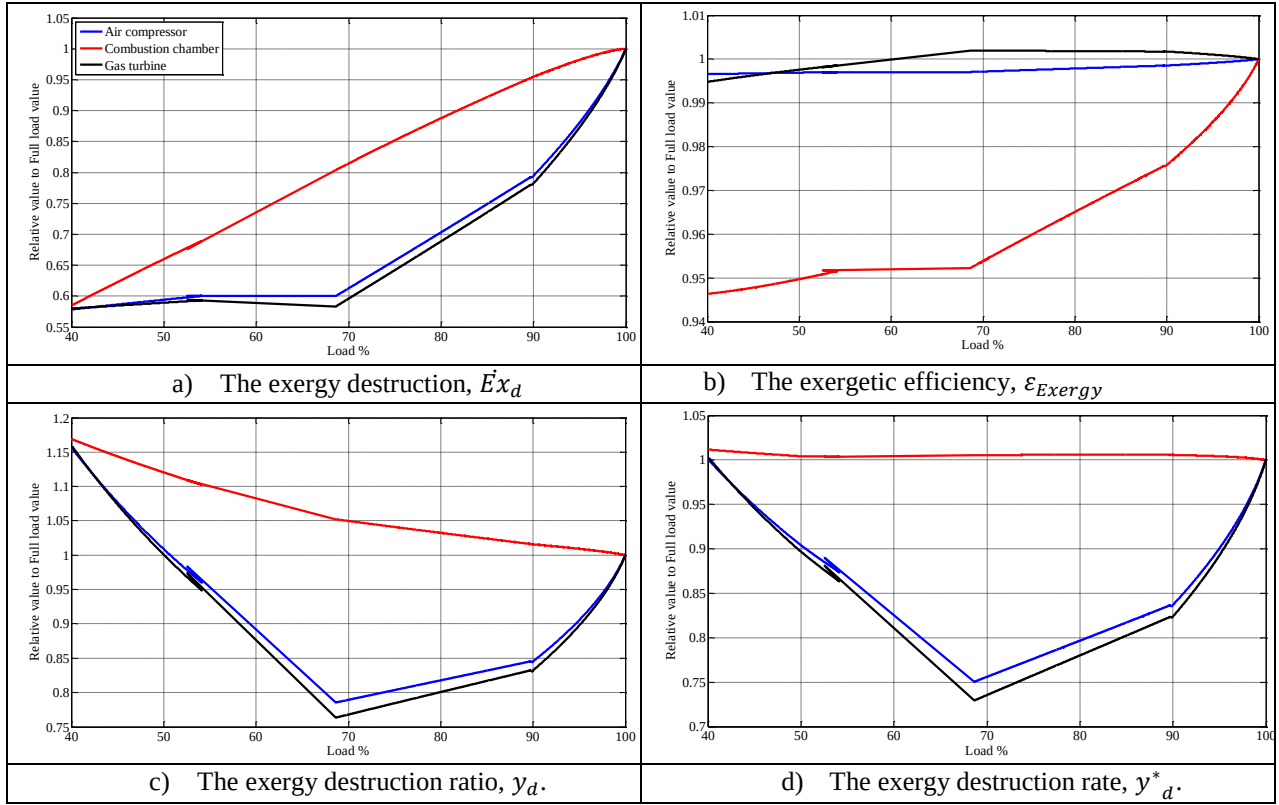


Figure 44 The effect of the loads on exergy destruction analysis in compressor, combustion chamber and gas turbine

This behavior is due to the VIGV which is controlling the air mass flow rate entering the compressor. At the load of 90% the exergy destruction of air compressor, combustion chamber and gas turbine is 80.4%, 94.4% and 79.3% relative to the full load exergy destruction, respectively. The maximum destructions occur in the combustion chamber.

The compressor and turbine exergy destruction are proportional to the air mass flow rate. Figure 40 indicates that the mass flow rate is decreased while load decreasing. While the combustion chamber exergy destruction is not affecting by the air mass flow rate it depends the chemical reaction which is most significant source of exergy destruction than the thermo-mechanical exergy which are entering and leaving the combustion chamber.

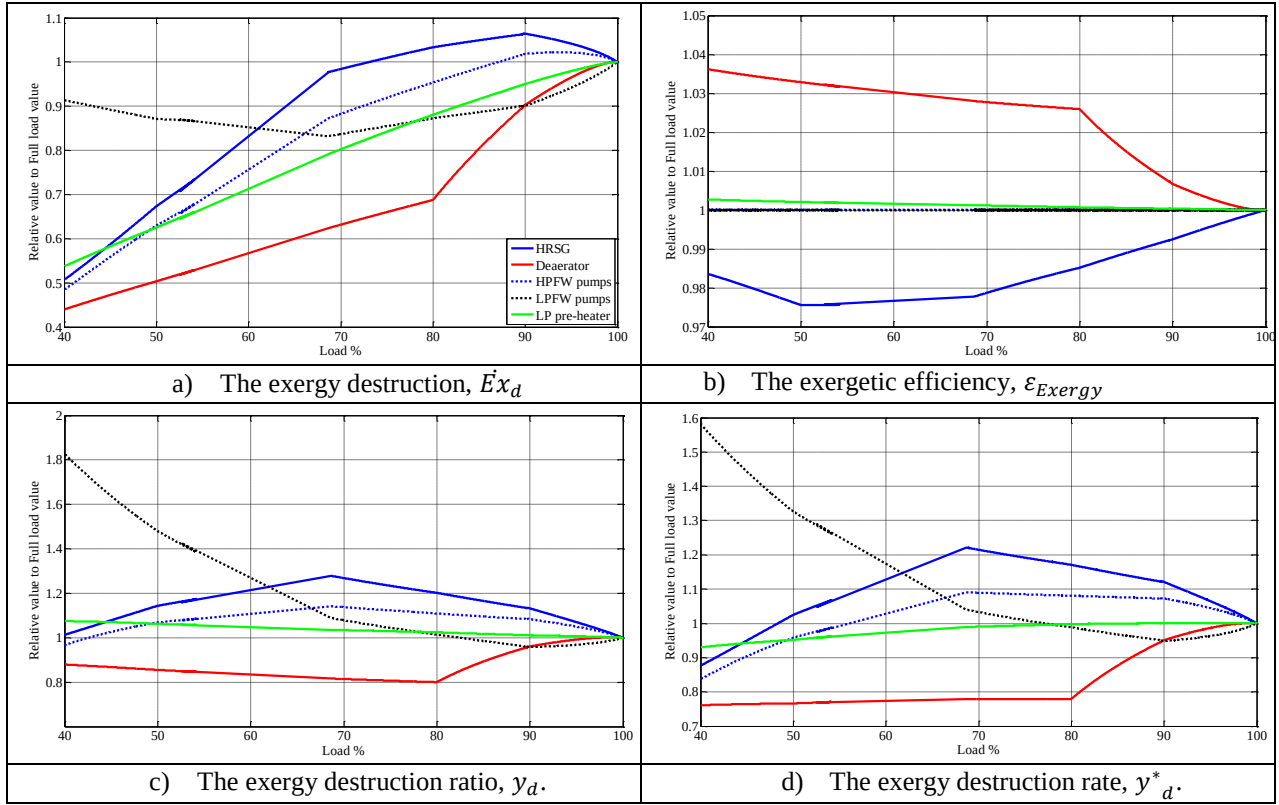


Figure 45 The effect of the loads on exergy destruction analysis in HRSG

The same pattern was accomplished in the work of Baheta and Gilani (Baheta & Gilani, 2011). It shows that variation of exergy destruction rate in the compressor and gas turbine are started to fall down from 20% load up to 50% load where the VSVs (Variable stator vanes) is fully opened. As the load reaches 50% the VSVs is started to close which is caused to reduce the air mass flow rate entering the compressor. Figure 45 shows the effect of load on the exergy destruction rate, relative to the full load value, for the HRSG, deaerator, LP pre-heater, HPFWs and LPFWs. From the figure it is clear that for all the exergy destruction is decreasing when the load decreases except for the HRSG is increasing at the range of 84.25% to 100% load. This last observation is due to the hot gases temperature which enters the HRSG is increased as the load increases as shown in Figure 40. Then, all components are decreasing as the load is decreased. Figure 46 shows the relative exergy destruction rate to full load for the steam turbine, ACC, CEP and ejector. The ejector exergy destruction rate is increasing as the load is decreased and it reaches its maximum value at load of 69%. The reason for that is the steam admitting to the ejector has a maximum temperature.

This is causing the exergy destruction to increase as the load decreases. The steam turbine exergy destruction rate is falling down to 60% relative to full at 70% load and then a minor reduction is occurred when the load is varying from 40% to 70%. ACC exergy destruction rate is increasing as the load increases due the reduction in the steam mass flow rate passes the ACC comparing to the cooling air that uses to condensate the exhaust steam. CEP exergy destruction rate is decreasing with load. The reason is the condensate mass flow rate plays the essential role of the exergy destruction and it is decreasing linearly with the load.

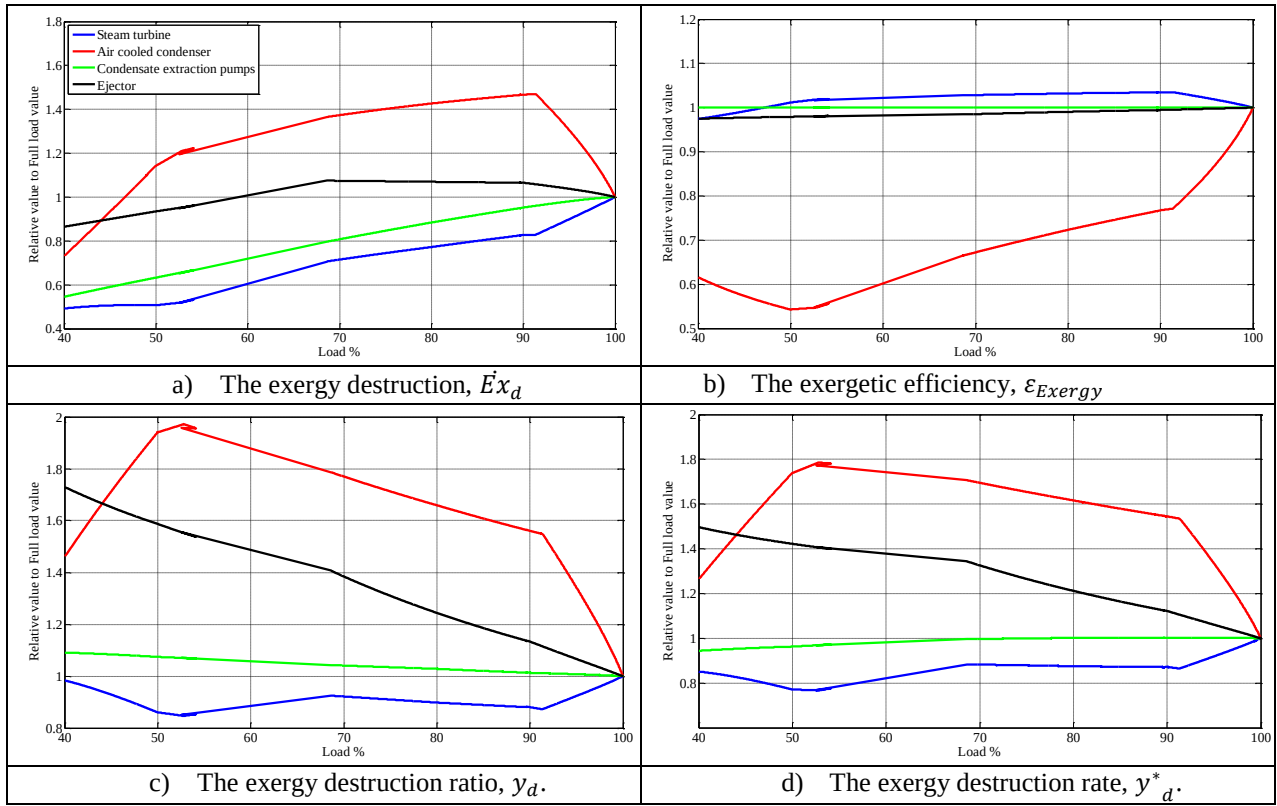


Figure 46 The effect of the loads on exergy destruction analysis in ST

CHAPTER 5

5 CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

In the present study, the effects of varying the ambient temperature, turbine inlet temperature, compressor pressure ratio, heat losses within combustion chamber and loads on the performance and exergy destruction of a combined cycle block consists of two gas turbine and one steam turbine have been demonstrated. In addition to that, the effect of spray water fogging system as power augmentation method on the performance and exergy destruction of a combined cycle power plant has been investigated too. To validate the present model, the gas turbine power at different loads is calculated. The model results are compared with the corresponding ones given by the block's manufacturer. The comparison shows the deviations to be in the range of 3-5%.

The results of ambient temperature variation have shown that it has a significant influence on the performance of the air compressor as well as the gas turbine. The air compressor work in is increased by 6% and follow by reduction in the total air mass flow rate delivers by the compressor. The gas turbine power output is reduced by more than 20% as the ambient temperature varies from 15 °C to 45 °C. It has been observed that the gas turbine heat rate is increased by 9.26%. The steam turbine back pressure is significantly affected by the ambient temperature when it rises above 25 °C. the steam turbine power output is reduced by 4.5% due to increase in back pressure. The overall heat rate of the combined cycle has been increased by 2.99% over the range of investigated ambient temperature. It has observed that the gas turbines contribution loads is dropped as the ambient temperature increases while the steam turbine contribution load vice versa.

It has been observed from the result of turbine inlet temperature variation that, the fuel consumption is increased by more than 219% as the turbine inlet temperature increases. In addition to that the combustion process is converted from lean to rich combustion. It

has been observed that gas turbine power output is increased by more than 160 MW with a rate of 32.3 MW for every 100 °C increase in the turbine inlet temperature. Further, the steam turbine power output is increased by 169MW. On the other hand, the combined cycle overall thermal efficiency has shot up from 42.92% to 52.99% over the range of turbine inlet temperature variation.

The results of the compressor pressure ratio variation have shown that the air compressor work in is rapidly increased by more than 68% over the range of pressure ratio investigated. It has been observed that the fuel mass flow rate in increased as the pressure ratio increases. The maximum increase in the fuel consumption is 22.78% corresponding to 22 compressor pressure ratio. On the other hand, the fuel air equivalence ratio is decreased by 22.74% and causing the combustion process to become leaner. The steam turbine power output in decreasing by more than 43.81% has been observed due to gas turbine exhaust temperature reduction, the reduction is found to be 18.17%.

The results of the heat losses in the combustion chamber have shown that it has a significant influence on the fuel mass flow rate. The fuel mass flow rate is increased as the heat losses within the combustion chamber increases. The maximum increase in the fuel consumption is 5.87% corresponding to a 5% heat losses. On the other hand, the work net of the gas turbine is slightly increased. Further, it causes a 5.07% reduction the thermal efficiency of the gas turbine.

The results of gas turbine with intake air cooling method have shown that the air intake temperature is dropped by 5 °C at low ambient and 7 °C at high ambient temperature which resulting an increase in the gas turbine power output by 5 MW at low ambient and 8 MW at high ambient temperature. On the other hand, the fuel mass flow rate has been observed increased by 2% at low ambient and 12% at high ambient temperature. It has been observed that the air compressor work in is dropped by 1.51% at low ambient and 2.21% at high ambient temperature.

The results of gas turbine load variation have shown that the ratio of the steam turbine power to the block power reaches its maximum value of 0.324 at a load of 74%. This is corresponding to the maximum exhaust gases temperature of 550° C. It has been

observed that the heat rate of the plant is reached the lowest value when the block load is 94% of the full load. This is due to during part load the gas exhaust temperature increases slightly and reached the maximum allowable temperature i.e. improving the heat recovery steam generator.

The results of the exergy destruction have shown that the highest destruction to be in the combustion chamber in all the variable parameters that have been studied. This is because the chemical reactions are the most significant source of exergy destruction. The exergy destruction in the combustion chamber is mainly affected by the fuel mass flow. So, it can be reduced by preheating the inlet air and increasing the compressor pressure ratio i.e. increasing the temperature of the air entering the combustion chamber.

5.2 RECOMMENDATIONS

The original proposed thesis objectives are satisfied in the present work. However, there are number of opportunities for further works have been identified.

First, it is recommended to perform thermoeconomic analysis for the give combined cycle power plant shown in Figure 1. The thermoeconomic or exergoeconomics is helped to provided the best operative cost effective parameters of the combined cycle power plant.

It is recommended that the combined cycle power plant operating parameters be optimized by defining an objective function which can be directly effective the cost of operating the given combined power plant.

Further work can be performed on heat recovery steam generator in order to identify the best operating parameter of the HRSG that can achieve maximum energy extraction from the flue hot gases. It is also that a triple pressure HRSG can be employed to decrease temperature difference between hot gases and water streams.

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APPENDIX A: GAS MIXTURE AND STEAM PROPERTIES

This appendix presents the procedure used to calculate the gas mixture properties. The appendix is divided into three sections: wet air properties, combustion mixture properties and water/steam mixture properties.

A.1 WET AIR PROPERTIES

The wet air properties such as enthalpy, specific heat and entropy at different ambient condition (temperature and relative humidity) are calculated as shown in Figure 47:

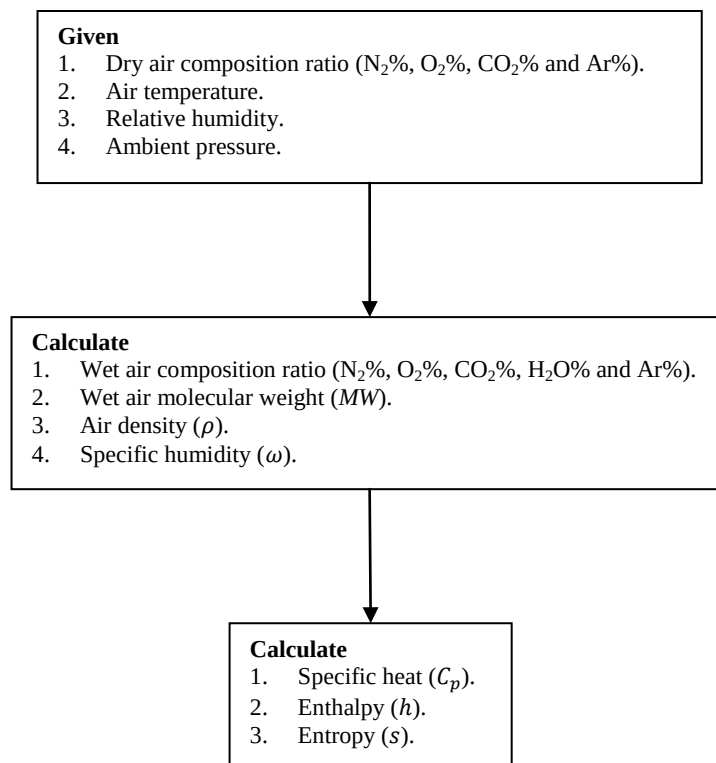


Figure 47 Flowchart describes wet air properties calculation

A.2 COMBUSTION GASES PROPERTIES

The combustion gases properties are calculated at different temperature and pressure from the combustion chamber reaction equation as shown in Figure 48:

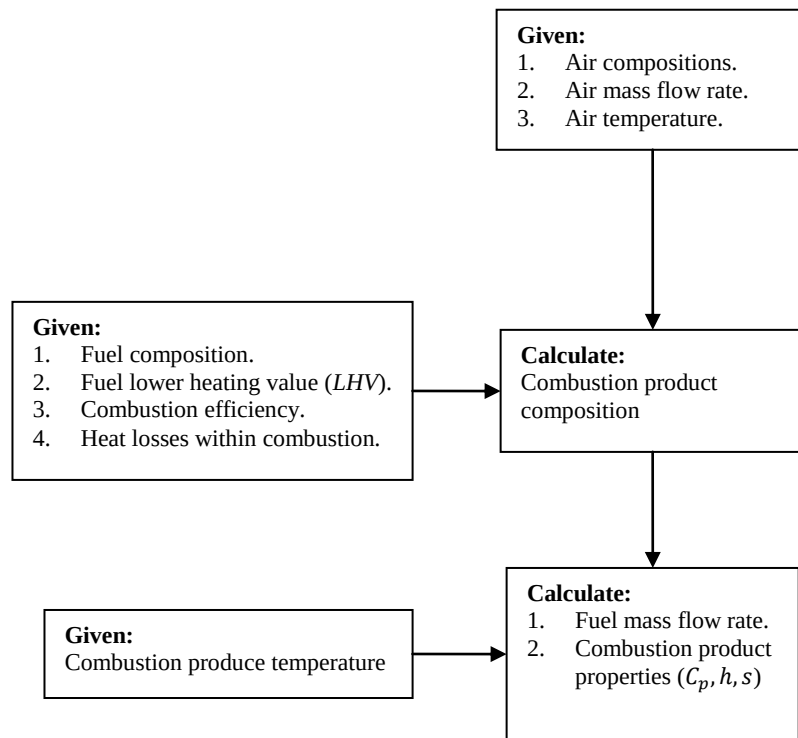


Figure 48 Flowchart describes combustion gases properties calculation

A.3 WATER/STEAM PROPERTIES

The water and steam properties at different operating condition are obtained from a MATLAB function called X-Steam.

The X-Steam function for MATLAB is an implementation of IAPWS IF97 standard formulation and it is programmed as MATLAB .m file. It provides accurate thermo hydraulic data for water, steam and mixtures of water and steam in the region:

- $0^{\circ}\text{C} < \text{temperature} < 2000^{\circ}\text{C}$ for $0.00611 \text{ bar} < \text{pressure} < 100 \text{ bar}$.
- $0^{\circ}\text{C} < \text{temperature} < 1000^{\circ}\text{C}$ for $0.00611 \text{ bar} < \text{pressure} < 1000 \text{ bar}$.

For accuracy and further information on IAPWS IF97 formulation, see homepage of the international association for properties of water and steam (www.iapws.org).

APPENDIX B: MODELS

This appendix presents the procedure used to calculate the air compressor, combustion chamber, gas turbine, heat recovery steam generation, air cooled condenser and steam turbine. In addition to that the optimum compressor pressure ratio estimation is described. The appendix is divided into seven sections:

B.1 AIR COMPRESSOR MODEL

The air compressor model is designed based on the given equations in section 3.4.1:

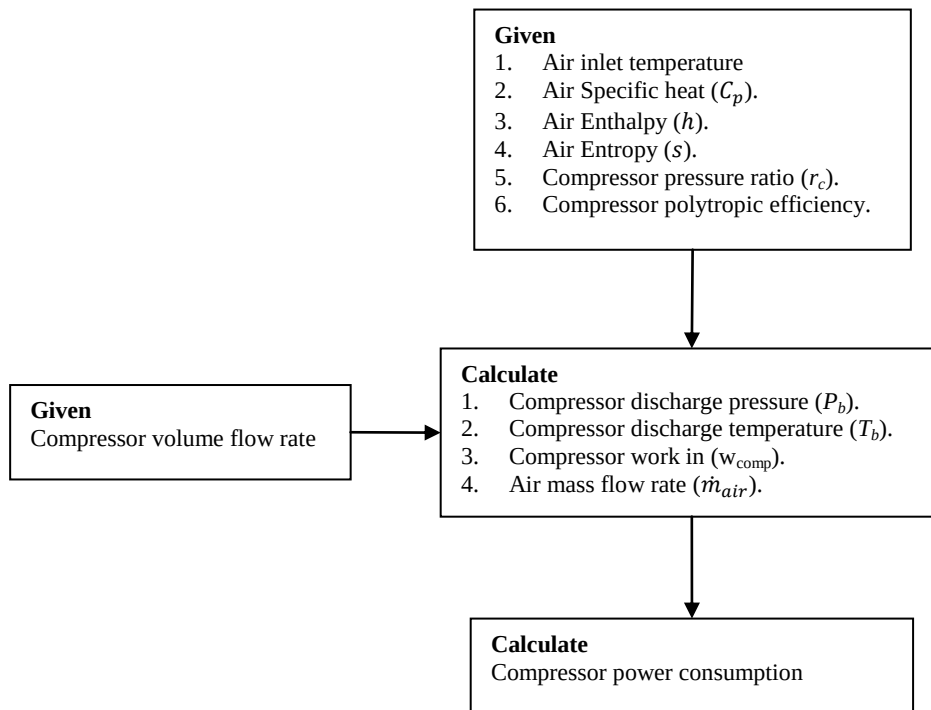


Figure 49 Flowchart describes air compressor model

B.2 COMBUSTION CHAMBER MODEL

The combustion chamber model is designed based on the given equations in section 3.4.1:

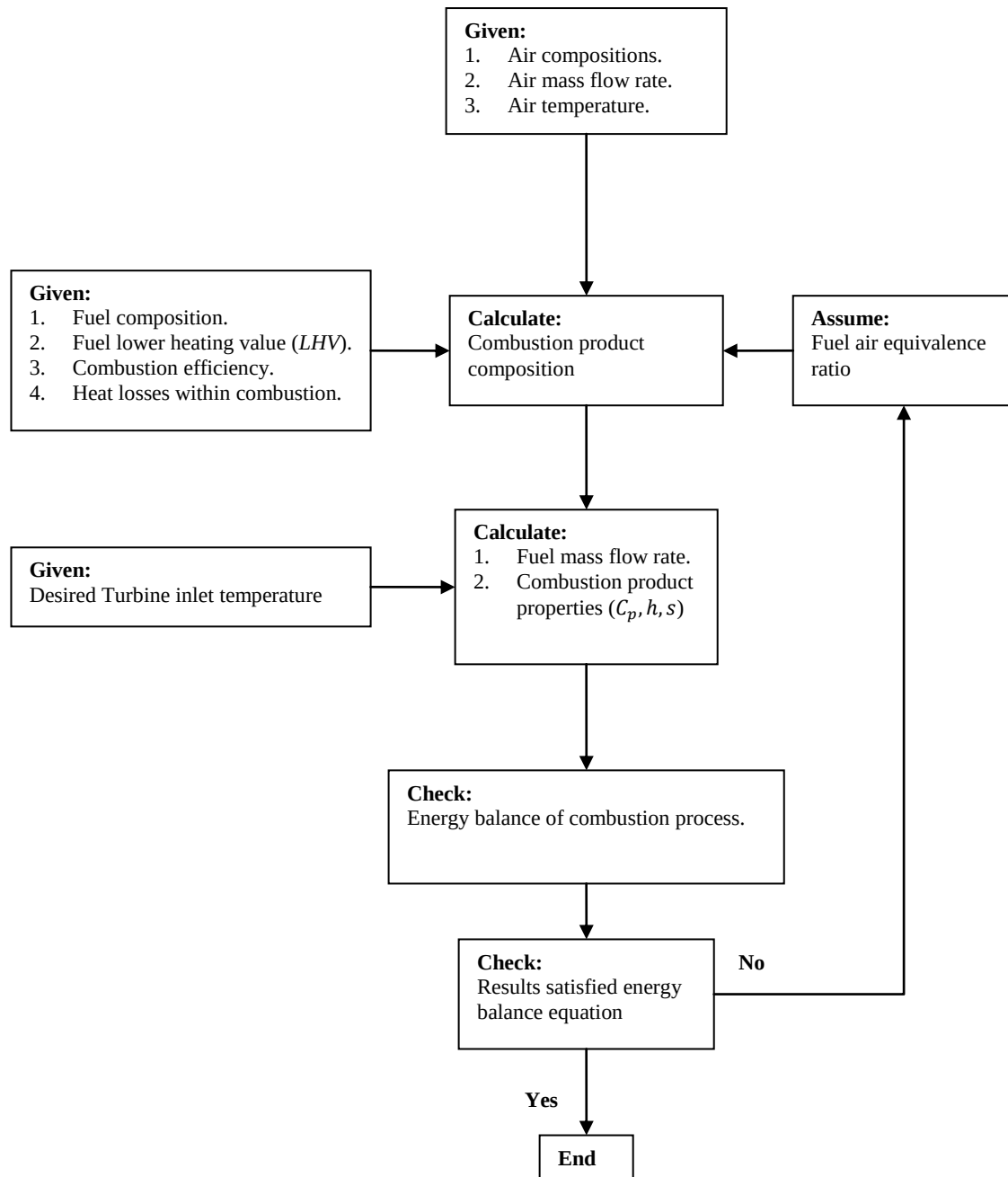


Figure 50 Flowchart describes combustion chamber model

B.3 GAS TURBINE MODEL

The gas turbine model is designed based on the given equations in section 3.4.3:

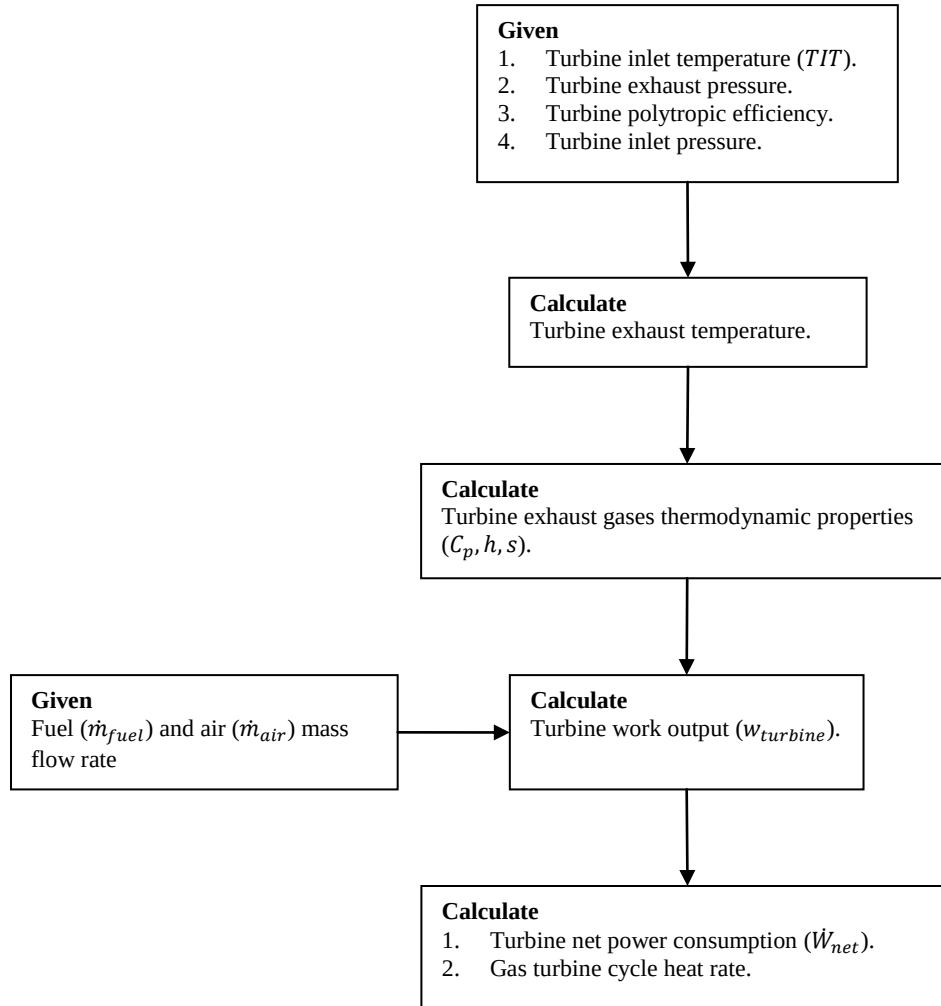


Figure 51 Flowchart describes gas turbine model

B.4 HEAT RECOVERY STEAM GENERATOR MODEL

The heat recovery steam generator model is designed based on the given equations in section 3.4.3:

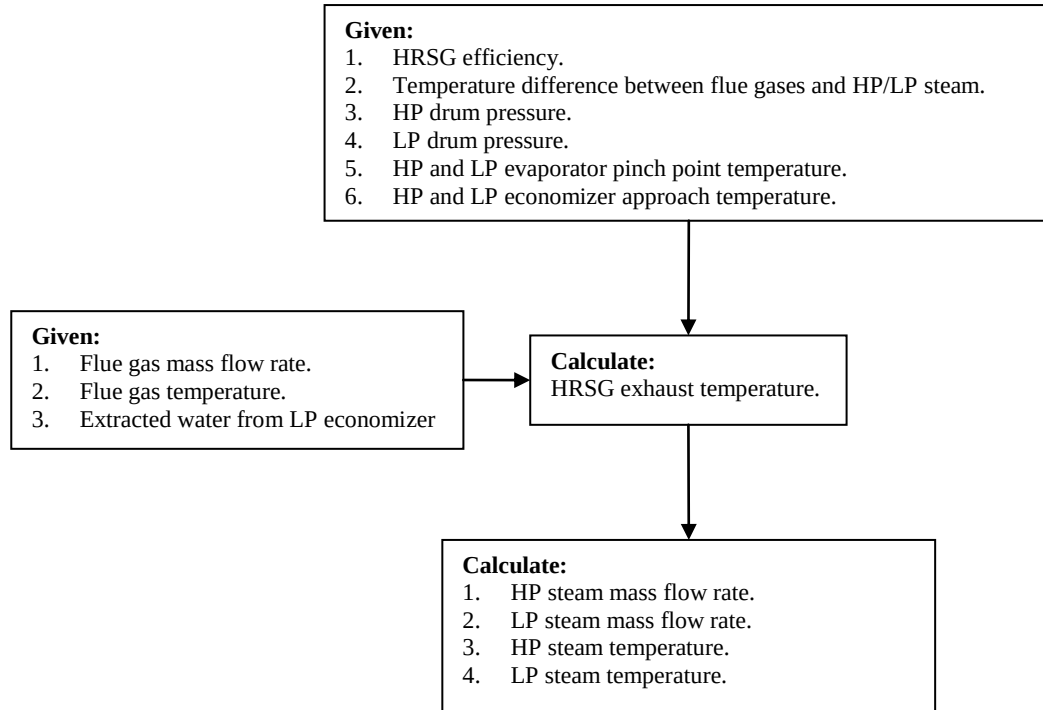


Figure 52 Flowchart describes heat recovery steam generator model

B.5 AIR COOLED CONDENSER MODEL

The air cooled condenser model is designed based on the given equations in section 3.4.3:

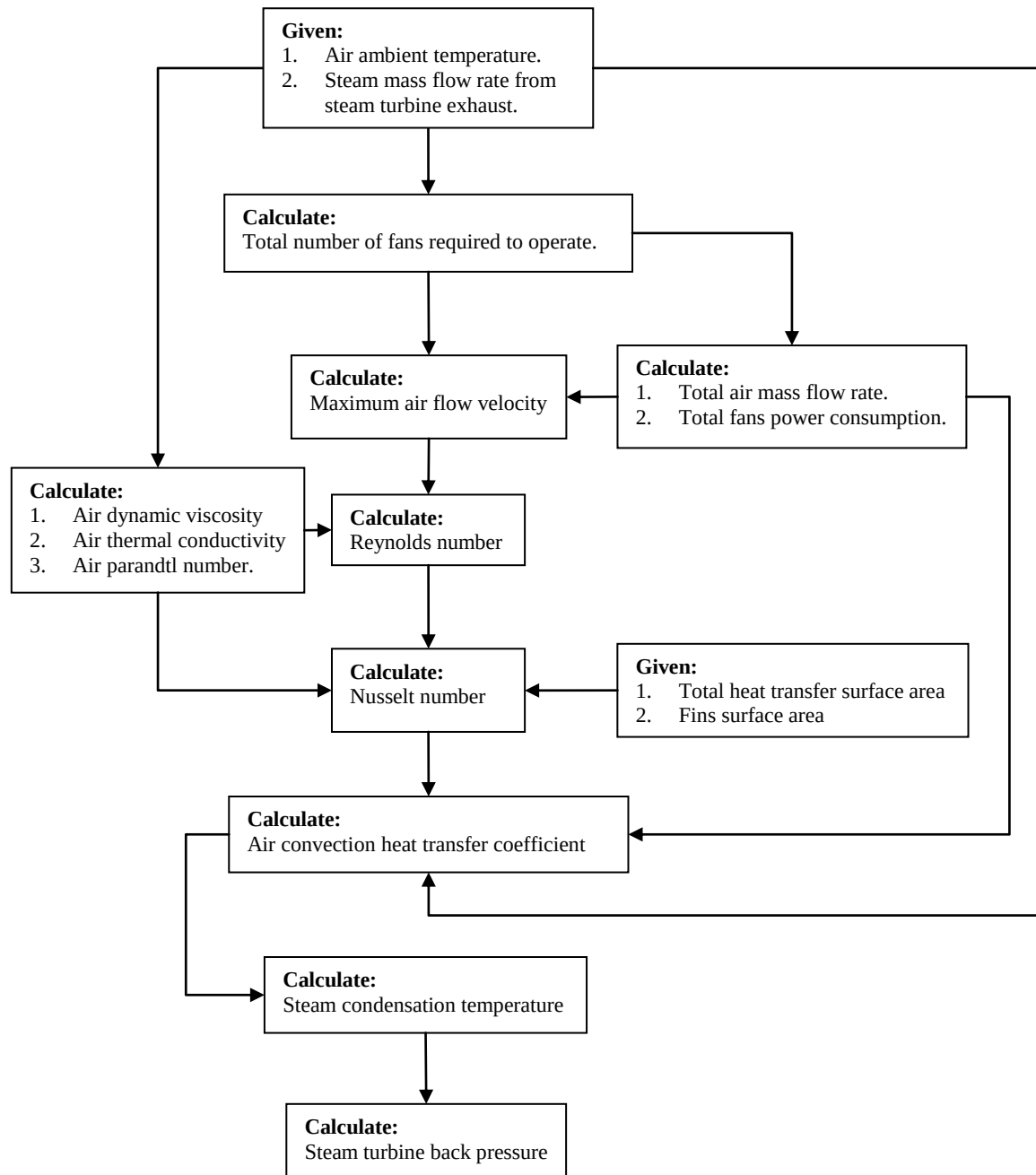


Figure 53 Flowchart describes air cooled condenser model

B.6 STEAM TURBINE MODEL

The steam turbine model is designed based on the given equations in section 3.4.3:

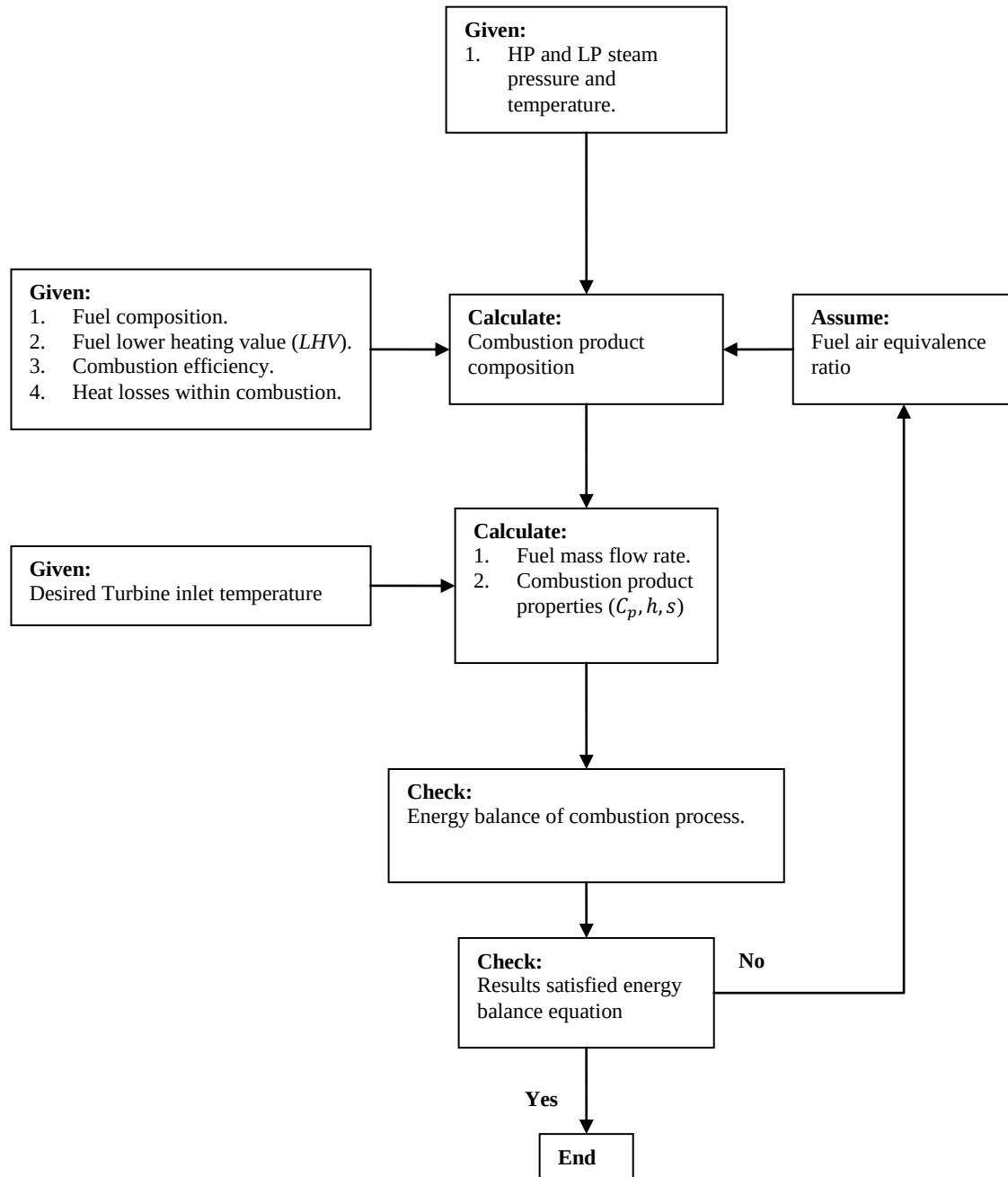


Figure 54 Flowchart describes steam turbine model

B.7 OPTIMUM COMPRESSOR PRESSURE RATIO ESTIMATION

The optimum compressor pressure ratio is calculated by assuming that the maximum work net output can be achieved from the gas turbine. The gas turbine work net output is the difference between compressor work in and turbine work output as given in Eq. 82:

$$\dot{W}_{net} = \dot{W}_{turb} - \dot{W}_{comp} \quad (82)$$

Where the air compressor and gas turbine works can be written as a function of their inlet and outlet temperatures as given in Eq. 83 and Eq. 84:

$$\dot{W}_{comp} = \dot{m}_{air} c_{p,air} (T_b - T_a) \quad (83)$$

$$\dot{W}_{turb} = (\dot{m}_{air} + \dot{m}_{fuel}) c_{p,gas} (T_c - T_d) \quad (84)$$

Where T_a and T_b are compressor inlet and outlet temperatures respectively. T_c and T_d are turbine inlet and outlet temperatures respectively. The compressor and turbine outlet temperature can be calculated as a function of the inlet temperature as given in Eq. 85 and Eq. 86:

$$T_b = T_a \cdot (r_c)^{(\gamma_a - 1)/\gamma_a \eta_{p,c}} \quad (85)$$

$$T_d = T_c \cdot \left(\frac{1}{r_c}\right)^{(\gamma_g - 1) \cdot \eta_{p,t}/\gamma_g} \quad (86)$$

By combining all equation in the gas turbine work net equation and assume that the $\dot{m}_{fuel}$ is very small compare to $\dot{m}_{air}$. Therefore can be ignored, then Eq. 82 can be written as given in Eq. 87:

$$\frac{\dot{W}_{net}}{\dot{m}_{air} c_{p,air}} = \frac{c_{p,gas}}{c_{p,air}} T_c \left(1 - \left(\frac{1}{r_c}\right)^{(\gamma_g - 1) \cdot \eta_{p,t}/\gamma_g}\right) - T_a ((r_c)^{(\gamma_a - 1)/\gamma_a \eta_{p,c}} - 1) \quad (87)$$

In order to evaluate the value of optimum compressor pressure ratio ($r_{c,opt}$), Eq. 87 is differentiated with respect to r_c and equalized to zero. Then $r_{c,opt}$ is equal to:

$$r_{c,opt} = \left[\frac{c_{p,gas}}{c_{p,air}} \frac{T_c}{T_a} \left(\frac{(\gamma_g - 1) \cdot \eta_{p,t}/\gamma_g}{(\gamma_a - 1)/\gamma_a \eta_{p,c}}\right) \right]^{1/((\gamma_a - 1)/\gamma_a \eta_{p,c} + (\gamma_g - 1) \cdot \eta_{p,t}/\gamma_g)} \quad (88)$$

APPENDIX C: MATLAB PROGRAM CODE

This appendix presents the MATLAB program code which has been used to calculate the performance of the give combined cycle power plant as shown in Figure 1:

```
%=====
%
%                               INPUT                               %
%=====
%
%               STANDARD REFERENCE CONDITION (Dead State)               %
%
clc
clear all

Po=1.013;           % Pressure (bar)
To=15:0.003:45;    % Temperature (C)
RHo=0.6;           % Relative Humidity

%=====
%
%               FUEL GAS PARAMETERS                               %
%
P_fuel=30;          % Fuel gas pressure bar
T_fuel=60+273.15;   % Fuel gas temperature

%
%               FUEL COMPOSITIONS                               %
%
R_CH4 = 0.7949;      %input('Fuel Compositions - Methane (CH4)
mole%: ')/100;

R_C2H6 = 0.0152;     %input('Fuel Compositions - Ethane (C2H6)
mole%: ')/100;

R_C3H8 = 0.0036;     %input('Fuel Compositions - Propane (C3H8)
mole%: ')/100;

R_C4H10 = 0.0017;    %input('Fuel Compositions - Butane (C4H10)
mole%: ')/100;

R_C5H12 = 0.0009;    %input('Fuel Compositions - Pentane (C5H12)
mole%: ')/100;

R_C6H14 = 0.0025;    %input('Fuel Compositions - Hexane (C6H14)
mole%: ')/100;

R_N2 = 0.1174;       %input('Fuel Compositions - Nitrogen (N2)
mole%: ')/100;
```

```

R_CO2 = 0.0638;      %input('Fuel Compositions - Carbon Dioxide (CO2)
mole%:')/100;

%=====
%
%               AIR COMPRESSOR PARAMETERS               %
%
dP_comp=0.500/100;  %input('Inlet Compressor Pressure drop %:');
rc_ISO=14.2;        %input('Compressor Pressure Ratio:');
eta_polyt= 0.90;    % Compressor polytropic efficiency
eta_m=0.99;         % Compressor Mechanical efficiency
Vcomp=429;          % Compressor volume flow rate m3/s
%=====
%
%               COMBUSTOR PARAMETERS                     %
%
eta_cc=0.98;        %input('Efficiency of Combustion Chamber %')/100;
dP_cc=0.2/100;      %input('Combustion Chamber Pressure drop %:');
HL=0.02;            %input('Heat Losses in Combustion Chamber%')/100;
X_mix=0.35;         % Primary air ratio to Combustion Chamber
%=====
%
%               GAS TURBINE PARAMETERS                   %
%
TIT=1100+273.15;    %input ('Turbine inlet
temperature(DegC):')+273.15; % Turbine inlet temperature in K
eta_polyt=0.95;     % Turbine polytropic efficiency
P_back=0.1;         %input('Back Pressure at Turbine (bar):')/1000;
%=====
%
%               HRSG PARAMETERS                         %
%
N_boiler=2;         % Number of Boilers
eta_HRSG=0.995;     % HRSG Efficiency
dTsg_HP=20;         % The temperature difference between the gas
turbine exhaust gases and HP steam [C]
dTsg_LP=11;         % The temperature difference between the gas
turbine exhaust gases and LP steam [C]
LP_DegreeSH=100;    % Degree of superheat of LP live steam [C]
PP_HP=10;           % HP evaporator pinch point temperature difference
[C]

```

```

PP_LP=18;           % LP evaporator pinch point temperature difference
[C]

AT_HP=6;           % HP economizer approach temperature difference
[C]

AT_LP=6;           % LP economizer approach temperature difference
[C]

Pl_HP=105;         % HP Water pressure to HRSG in bar

Pl_LP=18;          % LP Water pressure to HRSG in bar

P_LP=4.2;          % LP Drum pressure in bar

P_hp=70;           % HP Drum pressure in bar

T_dea=125;         % Deaerator Temperature [C]

mdot_LPtoDea=30;   % Extracted water from LP Economizer to Deaerator
[kg/s]

%=====

%                AIR COOLED CONDENSER PARAMETERS                %

V_fan=757;         % Fan Air volume flow rate (m3/s)

dP_static=0.109;   % Total static pressure loss (kPa)

eta_Fan=0.606;     % Fan efficiency

eta_gearbox=0.97;  % Gearbox efficiency

eta_motor=0.956;   % Motor efficiency

wf=0.3e-3;         % Fins thickness

hf=19e-3;          % Fins height

sf=2.8e-3;         % Fins Pitch

Df=73e-3;          % Outer diameter of Fins

Pt=54e-3;          % Pitch of tubes in plane perpendicular to flow

Dr=35e-3;          % External diameter of the tube at fin root

Lt=11.25;          % Length of tubes

nt=7840;           % Total number of tubes

A_ACC=619000;      % Total heat transfer surface area (m2)

deltaT_lm=15.35;   % Log mean temperature difference

%=====

%                STEAM TURBINE PARAMETERS                %

eta_ST=0.95;       % Steam turbine Isentropic efficiency

```

```

%=====
%
%                                AMBIENT CONDITIONS                                %
Ta=To+273.15;          %input('Ambient Air Temperature (DegC):')+273.15 ;
% Inlet air temperature to air compressor in K

Pamb=Po;               %input('Ambient Pressure (bar):');
RH=RHo;                %input('Relative Humidity %:']/100;
%=====
%
%                                AIR COMPOSITION WITH MOISTURE                                %
R=8.314;               %Universal Gas Constant (kJ/kmol.K)

% Dry Air Compositions Mole Fraction %
n_N2d= 0.78084;
n_O2d= 0.20947;
n_CO2d= 0.00035;
n_Ard= 0.00934;
% Air Composition Molecular Weight
M_N2= 28.02;
M_O2= 32;
M_CO2= 44.01;
M_Ar= 39.95;
M_H2O= 18.016;

n_airdry= 1;           % No of moles of dry air
M_airdry=n_N2d*M_N2 + n_O2d*M_O2 + n_CO2d*M_CO2 + n_Ard*M_Ar;    % Dry
Air Molecular Weight

n=length(To);
for i=1:n
Pg(i)=Xsteam('psat_T',To(i));
Pv(i)=Pg(i)*RH;
N_H2O(i)=-n_airdry/(1-Pamb/Pv(i));
N_air(i)= N_H2O(i) + n_airdry;    % No of moles of wet air
% Wet Air Compositions Mole Fraction %
n_N2(i)= n_N2d*n_airdry/N_air(i);
n_O2(i)= n_O2d*n_airdry/N_air(i);

```

```

n_CO2(i)= n_CO2d*n_airdry/N_air(i);

n_H2O(i) = N_H2O(i)/N_air(i);

n_Ar(i)= n_Ard*n_airdry/N_air(i);

M_air(i)= n_N2(i)*M_N2 + n_O2(i)*M_O2 + n_CO2(i)*M_CO2 + n_Ar(i)*M_Ar
+ n_H2O(i)*M_H2O; % Air Molecular Weight

%Air density calculation

rho_air(i)=(Pamb*100)*M_air(i)/(R*Ta(i));

Omega(i)=0.622*Pv(i)/(Pamb-Pv(i)); % Specific humidity (kg
water/kg dry air)

% Dead State properties for gases

cpa_o(i)=cpa(Ta(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i))
;

h_o(i) =h(Ta(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i));

s_o(i)=s(Ta(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i),Po);

% Dead State properties for water

cpw_o(i)=XSteam('cp_pt',Po,To(i));

hw_o(i)=XSteam('h_pt',Po,To(i));

sw_o(i)=XSteam('s_pt',Po,To(i));

end

%=====
%
% FUEL COMPOSITIONS
%

% Molecular Weight

M_CH4 = 16.042;

M_C2H6 = 30.068;

M_C3H8 = 44.094;

M_C4H10 = 58.12;

M_C5H12 = 72.146;

M_C6H14 = 86.172;

M_N2 = 28.02;

M_CO2 = 44.01;

M_fuel=M_CH4*R_CH4+M_C2H6*R_C2H6+M_C3H8*R_C3H8+M_C4H10*R_C4H10+M_C5H12
*R_C5H12+M_C6H14*R_C6H14+M_N2*R_N2+M_CO2*R_CO2;

%=====

```

```

% Standard molar Chemical exergy

x_CH4 = 831650;           %Standard molar chemical exergy, kJ/kmol
x_C2H6 = 1495840;        %Standard molar chemical exergy, kJ/kmol
x_C3H8 = 2154000;        %Standard molar chemical exergy, kJ/kmol
x_C4H10 = 2805800;       %Standard molar chemical exergy, kJ/kmol
x_C5H12 = 3463300;       %Standard molar chemical exergy, kJ/kmol
x_C6H14 = 4113300;       %Standard molar chemical exergy, kJ/kmol

for i=1:n

x_N2(i)=-R*Ta(i)*log(n_N2(i));   %Standard molar chemical exergy,
kJ/kmol

x_O2(i)=-R*Ta(i)*log(n_O2(i));   %Standard molar chemical exergy,
kJ/kmol

x_CO2(i)=-R*Ta(i)*log(n_CO2(i)); %Standard molar chemical exergy,
kJ/kmol

x_Ar(i)=-R*Ta(i)*log(n_Ar(i));   %Standard molar chemical exergy,
kJ/kmol

x_H2O(i)=-R*Ta(i)*log(n_H2O(i)); %Standard molar chemical exergy,
kJ/kmol

end

%=====
%                               COMPRESSOR MODEL                               %
%=====

Pk1=(1-dP_comp)*Pamb;

for i=1:n

cpa_a(i)=cpa(Ta(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i))
; % Specific heat of air at point a

h_a(i)=h(Ta(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i));

s_a(i)=s(Ta(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i),Pk1)
;

xph_a(i)=h_a(i)-h_o(i)-Ta(i)*(s_a(i)-s_o(i));

xch_a(i)=(n_N2(i)*x_N2(i)+n_O2(i)*x_O2(i)+n_CO2(i)*x_CO2(i)+n_H2O(i)*x
_H2O(i)+n_Ar(i)*x_Ar(i)+R*Ta(i)*(n_N2(i)*log(n_N2(i))+n_O2(i)*log(n_O2
(i))+n_CO2(i)*log(n_CO2(i))+n_H2O(i)*log(n_H2O(i))+n_Ar(i)*log(n_Ar(i)
)))/M_air(i);

xtot_a(i)=xph_a(i)+xch_a(i);

```

```

R_air(i)=R/M_air(i); % kJ/kg.K

gamma_a(i)=cpa_a(i)/(cpa_a(i)-R_air(i)); % Air specific heat ratio

rc(i)=rc_ISO*(288.15/Ta(i)); % Compressor Pressure ratio

Pk2(i)=rc(i)*Pk1; % Compressor Discharge Pressure

% CALCULATION COMPRESSOR ISENTROPIC EFFICIENCY %

n_comp(i)=(gamma_a(i)-1)/gamma_a(i);

eta_comp(i)=(1-rc(i)^n_comp(i))/(1-rc(i)^(n_comp(i)/eta_polyc)); %
Isentropic efficiency compressor

Tb(i)=Ta(i)*(rc(i)^(n_comp(i)/eta_polyc)); % Compressor outlet
temperature in K

cpa_b(i)=cpa(Tb(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i))
; % Specific heat of air at b

h_b(i) = h(Tb(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i));

s_b(i)=s(Tb(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i),Pk2(
i));

xph_b(i) = h_b(i)-h_o(i) -Ta(i)*(s_b(i)-s_o(i));

xch_b(i)=(n_N2(i)*x_N2(i)+n_O2(i)*x_O2(i)+n_CO2(i)*x_CO2(i)+n_H2O(i)*x
_H2O(i)+n_Ar(i)*x_Ar(i)+R*Ta(i)*(n_N2(i)*log(n_N2(i))+n_O2(i)*log(n_O2
(i))+n_CO2(i)*log(n_CO2(i))+n_H2O(i)*log(n_H2O(i))+n_Ar(i)*log(n_Ar(i)
)))/M_air(i);

xtot_b(i)=xph_b(i)+xch_b(i);

% CALCULATION COMPRESSOR WORK IN (kJ/kg) %

wcomp(i)=(h_b(i)-h_a(i))/eta_m; % Compressor work in kJ/kg

Pc(i)=(1-dP_cc)*Pk2(i);

end

%=====
% COMBUSTION CHAMBER MODEL %
%=====

% Lower Heating Value of Fuel Gas (kJ/kg)

LHV_CH4 = 50016; % Lower Heating Value of Methane (CH4)-(kJ/kg)
LHV_C2H6 = 47489; % Lower Heating Value of Ethane (C2H6)-(kJ/kg)
LHV_C3H8 = 46357; % Lower Heating Value of Propane (C3H8)-(kJ/kg)
LHV_C4H10 = 45742; % Lower Heating Value of Butane (C4H10)-(kJ/kg)
LHV_C5H12 = 45355; % Lower Heating Value of Pentane (C5H12)-(kJ/kg)

```

```

LHV_C6H14 = 45105; % Lower Heating Value of Hexane (C6H14)-(kJ/kg)

LHV=(R_CH4*LHV_CH4*M_CH4+R_C2H6*LHV_C2H6*M_C2H6+R_C3H8*LHV_C3H8*M_C3H8
+R_C4H10*LHV_C4H10*M_C4H10+R_C5H12*LHV_C5H12*M_C5H12+R_C6H14*LHV_C6H14
*M_C6H14)/M_fuel;

for i=1:n

xph_fuel(i)=xph(T_fuel,P_fuel,R_CH4,R_C2H6,R_C3H8,R_C4H10,R_C5H12,R_C6
H14,R_N2,R_CO2,Ta(i),Po);

xch_fuel(i)=(R_CH4*x_CH4+R_C2H6*x_C2H6+R_C3H8*x_C3H8+R_C4H10*x_C4H10+R
_C5H12*x_C5H12+R_C6H14*x_C6H14+R_N2*x_N2(i)+R_CO2*x_CO2(i)+R*Ta(i)*(R
_CH4*log(R_CH4)+R_C2H6*log(R_C2H6)+R_C3H8*log(R_C3H8)+R_C4H10*log(R_C4H
10)+R_C5H12*log(R_C5H12)+R_C6H14*log(R_C6H14)+R_N2*log(R_N2)+R_CO2*log
(R_CO2)))/M_fuel;

xtot_fuel(i)=xph_fuel(i)+xch_fuel(i);

end

%=====
%                               COMPLETE COMBUSTION                               %
%                               FUEL AIR RATION (STOICHIOMETRIC)                       %

ea(2)=0;

for i=1:n

%   AY=b

A1=-n_CO2(i)/n_O2(i);
A2=-n_H2O(i)/n_O2(i);
A3=-(2+2*n_CO2(i)/n_O2(i)+n_H2O(i)/n_O2(i));
A4=-n_N2(i)/n_O2(i);
A5=-n_Ar(i)/n_O2(i);

A=[A1 1 0 0 0;A2 0 1 0 0;A3 2 1 0 0;A4 0 0 1 0;A5 0 0 0 1];

b1=R_CH4+2*R_C2H6+3*R_C3H8+4*R_C4H10+5*R_C5H12+6*R_C6H14+R_CO2;
b2=2*R_CH4+3*R_C2H6+4*R_C3H8+5*R_C4H10+6*R_C5H12+7*R_C6H14;
b3=2*R_CO2;

b4=R_N2;

b=[b1 b2 b3 b4 0]';

Y=inv(A)*b;

y_gth(i)=Y(2,:)+Y(3,:)+Y(4,:)+Y(5,:); % Number of mole flue gases
Complete combustion

```

```

a_th(i)=Y(1,:);

yth_CO2(i)=Y(2,:); % Number of moles

yth_H2O(i)=Y(3,:); % Number of moles

yth_N2(i)=Y(4,:); % Number of moles

yth_Ar(i)=Y(5,:); % Number of moles

M_gth(i)= (yth_N2(i)*M_N2+yth_CO2(i)*M_CO2 + yth_Ar(i)*M_Ar +
yth_H2O(i)*M_H2O)/y_gth(i);

NN_air(i)=(n_N2(i)+n_O2(i)+n_CO2(i)+n_H2O(i)+n_Ar(i))/n_O2(i);

FA_theoritical(i)= (M_fuel/M_air(i))*(1/NN_air(i))*(1/a_th(i));

Solu=-1;

EA(i)=0.95*ea(i+1);

while Solu<=0

EA(i)=EA(i)+0.00001;

phi(i) = 1/(EA(i)+1); % Equivalence Ratio

FA_actual(i) = phi(i)*FA_theoritical(i);

a_act(i)= (M_fuel/M_air(i))*(1/NN_air(i))*(1/FA_actual(i));

ng_CO2(i)=(R_CH4+2*R_C2H6+3*R_C3H8+4*R_C4H10+5*R_C5H12+6*R_C6H14+R_CO2
+a_act(i)*n_CO2(i)/n_O2(i));

ng_H2O(i)=(2*R_CH4+3*R_C2H6+4*R_C3H8+5*R_C4H10+6*R_C5H12+7*R_C6H14+
a_act(i)*n_H2O(i)/n_O2(i));

ng_N2(i)=(R_N2+a_act(i)*n_N2(i)/n_O2(i));

ng_Ar(i)=(a_act(i)*n_Ar(i)/n_O2(i));

ng_O2(i)=R_CO2+a_act(i)+a_act(i)*n_CO2(i)/n_O2(i)+0.5*a_act(i)*
n_H2O(i)/n_O2(i)-ng_CO2(i)-0.5*ng_H2O(i);

N_gas(i)= ng_CO2(i)+ ng_H2O(i)+ ng_N2(i)+ ng_Ar(i)+ ng_O2(i);

% Mixing with Hot air with ratio Hot gases/Hot air = 0.35/0.65

y_CO2(i)=X_mix*ng_CO2(i)/N_gas(i)+(1-X_mix)*n_CO2(i);

y_H2O(i)=X_mix*ng_H2O(i)/N_gas(i)+(1-X_mix)*n_H2O(i);

y_N2(i)=X_mix*ng_N2(i)/N_gas(i)+(1-X_mix)*n_N2(i);

y_Ar(i)=X_mix*ng_Ar(i)/N_gas(i)+(1-X_mix)*n_Ar(i);

y_O2(i)=X_mix*ng_O2(i)/N_gas(i)+(1-X_mix)*n_O2(i);

M_gas(i)= y_N2(i)*M_N2 + y_O2(i)*M_O2 + y_CO2(i)*M_CO2 + y_Ar(i)*M_Ar
+ y_H2O(i)*M_H2O;

```

```

cpg_c(i)=cpg(TIT,y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i));
h_c(i)=h(TIT,y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i));
Solu=(h_c(i)-h_b(i))/(eta_cc*LHV*(1-HL)-h_c(i))-X_mix*FA_actual(i);
end

EA(i)=EA(i); % Excess Air Ratio
ea(i+2)=EA(i);

s_c(i)=s(TIT,y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i),Pc(i))
;

xph_c(i)=h_c(i)-h_o(i)-Ta(i)*(s_c(i)-s_o(i));

xch_c(i)=(y_N2(i)*x_N2(i)+y_O2(i)*x_O2(i)+y_CO2(i)*x_CO2(i)+y_H2O(i)*x
_H2O(i)+y_Ar(i)*x_Ar(i)+R*Ta(i)*(y_N2(i)*log(y_N2(i))+y_O2(i)*
log(y_O2(i))+y_CO2(i)*log(y_CO2(i))+y_H2O(i)*log(y_H2O(i))+y_Ar(i)*
log(y_Ar(i))))/M_gas(i);

xtot_c(i)=xph_c(i)+xch_c(i);

%=====
% TURBINE MODEL %
%=====

Pd = P_back + Pamb;

R_gas(i)=R/M_gas(i);

gamma_g(i)= cpg_c(i)/(cpg_c(i)-R_gas(i));

n_turb(i)=(gamma_g(i)-1)/gamma_g(i);

%=====

eta_t(i) = (1-(Pd/Pc(i))^(eta_polyt*n_turb(i)))/(1-
(Pd/Pc(i))^n_turb(i)); % Isentropic efficiency of Turbine

Td(i)=TIT*((Pd/Pc(i))^(eta_polyt*n_turb(i)));

cpg_d(i)=cpg(Td(i),y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i))
;

h_d(i)=h(Td(i),y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i));

s_d(i)=s(Td(i),y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i),Pd);

xph_d(i)=h_d(i)-h_o(i)-Ta(i)*(s_d(i)-s_o(i));

xch_d(i)=(y_N2(i)*x_N2(i)+y_O2(i)*x_O2(i)+y_CO2(i)*x_CO2(i)+y_H2O(i)*x
_H2O(i)+y_Ar(i)*x_Ar(i)+R*Ta(i)*(y_N2(i)*log(y_N2(i))+y_O2(i)*
log(y_O2(i))+y_CO2(i)*log(y_CO2(i))+y_H2O(i)*log(y_H2O(i))+y_Ar(i)*
log(y_Ar(i))))/M_gas(i);

xtot_d(i)=xph_d(i)+xch_d(i);

```

```

wturb(i) =(1+X_mix*FA_actual(i))*(h_c(i)-h_d(i));    % Turbine work in
kJ/kg

%=====

%Air Mass flow rate Calculation

mdot_dryair(i)=(Pamb-Pv(i))*100*Vcomp)*M_airdry/(R*Ta(i));
mdot_air(i)=(1+Omega(i))*mdot_dryair(i);
mdot_fuel(i)=mdot_air(i)*FA_actual(i)*X_mix;
mdot_gas(i)=mdot_fuel(i)+mdot_air(i);

%=====

Wturb(i) = mdot_air(i)*wturb(i);
Wcomp(i)=mdot_air(i)*wcomp(i);
r_bw(i)=wcomp(i)/wturb(i); % Back work ratio
Wnet_GT(i)=(Wturb(i)-Wcomp(i))/1000; % in MW
wnet(i)=Wnet_GT(i)/mdot_air(i)*1000; % in kJ/kg
eta_GTcycle(i)=Wnet_GT(i)/(eta_cc*mdot_fuel(i)*LHV)*1000;
HR_GTcycle(i)=3600/eta_GTcycle(i);
end

%=====
%                               Exergy Analysis of Gas Turbine Cycle                               %
%=====

for i=1:n

% Compressor Analysis

E_a(i)=mdot_air(i)*xtot_a(i);
E_b(i)=mdot_air(i)*xtot_b(i);
Ed_Comp(i)=(Wcomp(i)+E_a(i)-E_b(i))/1000; % Air Compressor Exergy
Destruction in MW
eta_compII(i)=(E_b(i)-E_a(i))/Wcomp(i);

%=====

% Combustion Analysis

Q_loss(i) = mdot_fuel(i)*eta_cc*LHV*HL;
E_c(i)=mdot_gas(i)*xtot_c(i);
E_fuel(i)=mdot_fuel(i)*xtot_fuel(i);

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```

Tj(i)=(Tb(i)+Ta(i))/2;

Ed_CombCh(i)=(1-Ta(i)/Tj(i))*Q_loss(i)+E_b(i)+E_fuel(i)-
E_c(i))/1000;% Combustion Chamber Exergy Destruction in MW

eta_CombChII(i)=(E_c(i)+Q_loss(i))/(E_b(i)+E_fuel(i));

%=====

% Gas Turbine Analysis

E_d(i)=mdot_gas(i)*xtot_d(i);

Ed_GasTurb(i)=(E_c(i)-E_d(i)-Wturb(i))/1000;% Gas Turbine Exergy
Destruction in MW

eta_GasTurbII(i)=Wturb(i)/(E_c(i)-E_d(i));

end

%=====

%                               Feed Water Pumps                               %
%=====

P_dea=XSteam('Psat_T',T_dea); % Deaerator pressure in bar
v_dea=XSteam('vL_T',T_dea); % Specific volume of Feed water
h_dea=XSteam('hL_T',T_dea);

w_HPFWP=v_dea*(Pl_HP-P_dea)*100; % HP Feed water pump work done in
kJ/kg

eta_HPFWP=0.85;

w_LPFWP=v_dea*(Pl_LP-P_dea)*100; % LP Feed water pump work done in
kJ/kg

eta_LPFWP=0.85;

%=====

%                               Heat Recovery Steam Generator                               %
%=====

for i=1:n

T1(i)=Td(i)-273.15-dTsg_HP;

h1(i)=XSteam('h_pT',P_hp,T1(i));

s1(i)=XSteam('s_pT',P_hp,T1(i));

x_1(i)=h1(i)-hw_o(i)-Ta(i)*(s1(i)-sw_o(i));

T4(i)=XSteam('Tsats_p',P_hp);

h4(i)=XSteam('hL_p',P_hp);

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```

T3(i)=T4(i);
h3(i)=XSteam('hV_p',P_hp);
T2(i)=T3(i);
h2(i)=h3(i);
T5(i)=T4(i)-AT_HP;
h5(i)=XSteam('h_pT',Pl_HP,T5(i));
T_d3(i)=(T4(i)+273.15)+PP_HP;
h_d3(i)=h(T_d3(i),y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i));
SG_HPratio(i)=(eta_HRSG*(h_d(i)-h_d3(i)))/(h1(i)-h5(i));
mdot_HP(i)=SG_HPratio(i)*N_boiler*mdot_gas(i);
T13(i)=XSteam('Tsats_p',P_LP); % in degC
h13(i)=XSteam('hL_p',P_LP);
T14(i)=T13(i)-AT_LP;
h14(i)=XSteam('h_pT',Pl_LP,T14(i));
s14(i)=XSteam('s_pT',Pl_LP,T14(i));
x_14(i)=h14(i)-hw_o(i)-Ta(i)*(s14(i)-sw_o(i));
T_d7(i)=T13(i)+PP_LP;
h_d7(i)=h(T_d7(i),y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i));
T12(i)=T13(i);
h12(i)=XSteam('hV_p',P_LP);
T11(i)=T12(i);
h11(i)=h12(i);
T10(i)=T11(i)+LP_DegreeSH;
h10(i)=XSteam('h_pT',P_LP,T10(i));
s10(i)=XSteam('s_pT',P_LP,T10(i));
x_10(i)=h10(i)-hw_o(i)-Ta(i)*(s10(i)-sw_o(i));
T_d4(i)=T10(i)+dTsg_LP+273.15;
h_d4(i)=h(T_d4(i),y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i));
%Dew point temperature of gases
P_H2O(i)=y_H2O(i)*Pamb;
T_dp(i)=XSteam('Tsats_p',P_H2O(i));

```

```

T_d10(i)=(T_dp(i)+110)+273.15;

h_d10(i)=h(T_d10(i),y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i)
);

s_d10(i)=s(T_d10(i),y_N2(i),y_O2(i),y_CO2(i),y_H2O(i),y_Ar(i),M_gas(i)
,Pd);

xph_d10(i)=h_d10(i)-h_o(i)-Ta(i)*(s_d10(i)-s_o(i));

xch_d10(i)=(y_N2(i)*x_N2(i)+y_O2(i)*x_O2(i)+y_CO2(i)*x_CO2(i)+y_H2O(i)
*x_H2O(i)+y_Ar(i)*x_Ar(i)+R*Ta(i)*(y_N2(i)*log(y_N2(i))+y_O2(i)*
log(y_O2(i))+y_CO2(i)*log(y_CO2(i))+y_H2O(i)*log(y_H2O(i))+y_Ar(i)*
log(y_Ar(i))))/M_gas(i);

xtot_d10(i)=xph_d10(i)+xch_d10(i);

h9(i)=h_dea+w_HPFWP/eta_HPFWP;

T9(i)=XSteam('T_ph',Pl_HP,h9(i));

s9(i)=XSteam('s_pT',Pl_HP,T9(i));

x_9(i)=h9(i)-hw_o(i)-Ta(i)*(s9(i)-sw_o(i));

h15(i)=h_dea+w_LPFWP/eta_LPFWP;

T15(i)=XSteam('T_ph',Pl_LP,h15(i));

s15(i)=XSteam('s_pT',Pl_LP,T15(i));

x_15(i)=h15(i)-hw_o(i)-Ta(i)*(s15(i)-sw_o(i));

SG_LPratio(i)=(eta_HRSG*(h_d(i)-h_d10(i))+SG_HPratio(i)*(h9(i)-
h1(i))+mdot_LPtoDea*(h15(i)-h14(i))/mdot_gas(i))/(h10(i)-h15(i));

mdot_LP(i)=SG_LPratio(i)*N_boiler*mdot_gas(i);

end

%=====
%
% Deaerator
%
%=====

for i=1:n

mdot_ejector=0.1; % 360 kg/hrs steam flow

T_cond=85;

P_cond=11.5;

h_cond=XSteam('h_PT',P_cond,T_cond);

s_cond=XSteam('s_PT',P_cond,T_cond);

x_cond(i)=h_cond-hw_o(i)-Ta(i)*(s_cond-sw_o(i));

```

```

h_pegging(i)=h10(i);
T_pegging(i)=XSteam('T_ph',P_dea,h_pegging(i));
s_pegging(i)=XSteam('s_PT',P_dea,T_pegging(i));
x_pegging(i)=h_pegging(i)-hw_o(i)-Ta(i)*(s_pegging(i)-sw_o(i));
h_LPtoDea(i)=h14(i);
T_LPtoDea(i)=T14(i);
s_LPtoDea(i)=XSteam('s_PT',Pl_LP,T_LPtoDea(i));
x_LPtoDea(i)=h_LPtoDea(i)-hw_o(i)-Ta(i)*(s_LPtoDea(i)-sw_o(i));
s_dea=Xsteam('sl_T',T_dea);
x_dea(i)=h_dea-hw_o(i)-Ta(i)*(s_dea-sw_o(i));
mdot_pegging(i)=(mdot_HP(i)+mdot_LP(i)+N_boiler*mdot_LPtoDea)*h_dea-
(mdot_HP(i)+mdot_LP(i))*h_cond-N_boiler*mdot_LPtoDea*h_LPtoDea)/
(h_pegging-h_cond);
end

%=====
%                               Feed Water Pumps                               %
%=====

for i=1:n
    HPFWP(i)=mdot_HP(i)*w_HPFWP/eta_HPFWP/1000; % HP Feed water pump power
consumption in MW
    LPFWP(i)=(mdot_LP(i)+N_boiler*mdot_LPtoDea)*w_LPFWP/eta_LPFWP/1000; %
LP Feed water pump power consumption in MW
end

%=====
%                               AIR COOLED CONDENSER                               %
%=====

% Back pressure calculation
for i=1:n
    Tair_in(i)=To(i); % Inlet Air temperature in degC
    if Tair_in(i)>21
        N_fan(i)=16; % Total numbers of fans is operation
    elseif Tair_in(i)>20
        N_fan(i)=15; % Total numbers of fans is operation
    end
end

```

```

elseif Tair_in(i)>19
N_fan(i)=14; % Total numbers of fans is operation
elseif Tair_in(i)>17.5
N_fan(i)=13; % Total numbers of fans is operation
elseif Tair_in(i)>16
N_fan(i)=12; % Total numbers of fans is operation
elseif Tair_in(i)>14
N_fan(i)=11; % Total numbers of fans is operation
elseif Tair_in(i)>12
N_fan(i)=10; % Total numbers of fans is operation
elseif Tair_in(i)>10
N_fan(i)=9; % Total numbers of fans is operation
elseif Tair_in(i)>7
N_fan(i)=8; % Total numbers of fans is operation
elseif Tair_in(i)>3.5
N_fan(i)=7; % Total numbers of fans is operation
else Tair_in(i)>2
N_fan(i)=6; % Total numbers of fans is operation
end

W_Fan=V_fan*dP_static/(eta_Fan*eta_gearbox*eta_motor); % Fan power
required kW

P_Fans(i)=W_Fan*N_fan(i)/1000; % Total Fans Power required MW

mfan_air(i)=N_fan(i)*V_fan*rho_air(i); % Total fans air mass flow rate
(kg/s)

Stmin=nt*Lt*(Pt-Dr-(2*wf*hf)/(wf+sf)); %Minimum flow area

Vmax(i)=mfan_air(i)/(rho_air(i)*Stmin); % Maximum flow velocity

% Air dynamic Viscosity (mu)

mu_o=18.27e-6; %(Pa.sec)

cmu=120; % Constant in K

Tmu=291.15; % Constant in K

mu(i)=(mu_o*(Tmu+cmu)/(Ta(i)+cmu))*((Tmu/Ta(i))^1.5);

Re(i)=Vmax(i)*Dr*rho_air(i)/mu(i); % Reynolds Number

```

```

AF=nt*Lt*pi/(sf+wf)*(0.5*(Df^2-Dr^2)+Df*wf+Dr*sf); % Surface area of
fins

AT=nt*Lt*pi*Dr; % Total surface area

% Air Thermal Conductivity (Lambda)

Lambda(i)=(3.13908e-9*Ta(i)^3-3.40606e-5*Ta(i)^2+9.78225e-2*Ta(i)-
1.0903e-1)*1e-3;

% Air Parandtl Number (Pr)

Pr(i)=1.2e-6*Ta(i)^2-9.2e-4*Ta(i)+8.75e-1;

% Nusselt Number (Nu)

Nu(i)=0.269*Re(i)^0.625*((AF/AT)^-0.375)*Pr(i)^0.333;

h_air(i)=Nu(i)*Lambda(i)/Dr; % Heat transfer coefficient air side
(W/m2K)

T_back(i)=Tair_in(i)+deltaT_lm*(A_ACC*h_air(i)/(mfan_air(i)*cpa_a(i)*1
000))/(1-exp(-A_ACC*h_air(i)/(mfan_air(i)*cpa_a(i)*1000)));

if T_back(i)>XSteam('Tsats_p',0.140)

P_4t(i)=XSteam('Psat_T',T_back(i));

T_4t(i)=XSteam('Tsats_P',P_4t(i));

v_4t(i)=XSteam('vl_P',P_4t(i));

T_back(i)=T_back(i);

else

P_4t(i)=0.140;

T_4t(i)=XSteam('Tsats_P',P_4t(i));

v_4t(i)=XSteam('vl_P',P_4t(i));

T_back(i)=XSteam('Tsats_p',P_4t(i));

end

w_CEP(i)=v_4t(i)*(P_cond-P_4t(i))*100;

eta_CEP=0.85;

h_CEPs(i)=XSteam('hl_T',T_4t(i));

s_CEPs(i)=XSteam('sl_T',T_4t(i));

x_CEPs(i)=h_CEPs(i)-hw_o(i)-Ta(i)*(s_CEPs(i)-sw_o(i));

h_CEPd(i)=h_CEPs(i)+w_CEP(i)/eta_CEP;

s_CEPd(i)=XSteam('s_ph',P_cond,h_CEPd(i));

x_CEPd(i)=h_CEPd(i)-hw_o(i)-Ta(i)*(s_CEPd(i)-sw_o(i));

```

```

mdot_cond(i)=mdot_HP(i)+mdot_LP(i)-mdot_pegging(i);

CEP(i)=mdot_cond(i)*w_CEP(i)/eta_CEP/1000;% Condensate Extraction pump
power consumption in MW

end

%=====
%                               Ejector                               %
%=====

for i=1:n
hejector_in(i)=h1(i);
P_ejector=15;%Ejector operating pressure (bar)
Tejector_in(i)=XSteam('T_Ph',P_ejector,hejector_in(i));
sejector_in(i)=XSteam('s_PT',P_ejector,Tejector_in(i));
xejector_in(i)=hejector_in(i)-hw_o(i)-Ta(i)*(sejector_in(i)-sw_o(i));
hejector_out(i)=XSteam('hl_P',P_ejector);
sejector_out(i)=XSteam('sl_P',P_ejector);
xejector_out(i)=hejector_out(i)-hw_o(i)-Ta(i)*(sejector_out(i)-
sw_o(i));
hcond_in(i)=h_CEPd(i);
hcond_out(i)=mdot_ejector*(hejector_in(i)-
hejector_out(i))/mdot_cond(i)+hcond_in(i);
Tcond_out(i)=XSteam('T_Ph',P_cond,hcond_out(i));
scond_out(i)=XSteam('s_Ph',P_cond,hcond_out(i));
xcond_out(i)=hcond_out(i)-hw_o(i)-Ta(i)*(scond_out(i)-sw_o(i));
end

%=====
%                               LP Pre-Heater                               %
%=====

for i=1:n
% Extraction steam mass flow rate
P_Ext=0.85;% Extracted steam pressure in bar
hExt_in=XSteam('hv_p',P_Ext);
sExt_in=XSteam('sv_p',P_Ext);

```

```

xExt_in(i)=hExt_in-hw_o(i)-Ta(i)*(sExt_in-sw_o(i));
hExt_out=XSteam('hl_P',P_Ext);
sExt_out=XSteam('sl_p',P_Ext);
xExt_out(i)=hExt_out-hw_o(i)-Ta(i)*(sExt_out-sw_o(i));
hLPheater_in(i)=hcond_out(i);
hLPheater_out=h_cond;
mdot_Ext(i)=mdot_cond(i)*(hLPheater_out-hLPheater_in(i))/(hExt_in-
hExt_out);
end

%=====
%                               Steam Turbine                               %
%=====

for i=1:n
h_1t(i)=h1(i);
h_2t(i)=h10(i);
h_3t=hExt_in;
h_4ts(i)=XSteam('h_ps',P_4t(i),s1(i));
h_4t(i)=h_1t(i)-eta_ST*(h_1t(i)-h_4ts(i));
x_4t(i)=(h_4t(i)-XSteam('hl_P',P_4t(i)))/(XSteam('hv_P',P_4t(i))-
XSteam('hl_P',P_4t(i))); % Exhaust steam quality
if x_4t(i)>=0.875
x_4t(i)=x_4t(i);
else
x_4t(i)=0.875;
end

s_4t(i)=XSteam('sl_p',P_4t(i))+x_4t(i)*(XSteam('sv_p',P_4t(i))-
XSteam('sl_p',P_4t(i)));
x4t(i)=h_4t(i)-hw_o(i)-Ta(i)*(s_4t(i)-sw_o(i));
W_ST(i)=(mdot_HP(i)-mdot_ejector)*(h_1t(i)-h_4t(i))+...
(mdot_LP(i)-mdot_pegging(i))*(h_2t(i)-h_4t(i))-
mdot_Ext(i)*h_3t)/1000;%in MW
W_Auxiliary(i)=CEP(i)+HPFWP(i)+LPFWP(i)+P_Fans(i);
eta_STcycle(i)=(W_ST(i)-W_Auxiliary(i))/(2*mdot_gas(i)*h_d(i)/1000);

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eta_cycle(i)=(2*Wnet_GT(i)+W_ST(i))/(2*LHV*mdot_fuel(i)/1000);

HR(i)=(2*mdot_fuel(i)*3600*LHV/1000*eta_cc)/(2*Wnet_GT(i)+W_ST(i)-
W_Auxiliary(i));

P_Totalcycle(i)=2*Wnet_GT(i)+W_ST(i)-W_Auxiliary(i);

end

%=====
%           Calculate the outlet air temperature from ACC           %
%=====

for i =1:n

h_rejected(i)=x_4t(i)*(XSteam('hV_P',P_4t(i))-XSteam('hL_P',P_4t(i)));

H_rejected(i)=h_rejected(i)*(mdot_HP(i)-mdot_ejector+mdot_LP(i)-
mdot_pegging(i)-mdot_Ext(i));

hair_in(i)=h_a(i);

xair_in(i)=xtot_a(i);

hair_out(i)=H_rejected(i)/mfan_air(i)+hair_in(i);

ACC=Axy(Ta(i),n_N2(i),n_O2(i),n_CO2(i),n_H2O(i),n_Ar(i),M_air(i));

T_AIROUT=roots([ACC(5)/5, ACC(4)/4, ACC(3)/3, ACC(2)/2, ACC(1), -
hair_out(i)]);

Tair_out(i)=T_AIROUT(5);

sair_out(i) = s(Tair_out(i), n_N2(i), n_O2(i), n_CO2(i), n_H2O(i),
n_Ar(i), M_air(i), Pamb);

xphair_out(i) = hair_out(i)-h_o(i) -Ta(i)*(sair_out(i)-s_o(i));

xchair_out(i)=(n_N2(i)*x_N2(i)+n_O2(i)*x_O2(i)+n_CO2(i)*x_CO2(i)+n_H2O
(i)*x_H2O(i)+n_Ar(i)*x_Ar(i)+...

R*Ta(i)*(n_N2(i)*log(n_N2(i))+n_O2(i)*log(n_O2(i))+n_CO2(i)*log(n_CO2(
i))+n_H2O(i)*log(n_H2O(i))+n_Ar(i)*log(n_Ar(i)))/M_air(i);

xair_out(i)=xphair_out(i)+xchair_out(i);

end

%=====
%           CO2 Generation           %
%=====

for i=1:n

mdot_CO2(i)=2*3600*y_CO2(i)*mdot_gas(i)*M_CO2/M_gas(i); % kg/hr

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m_CO2gen(i)=mdot_CO2(i)/(1000*(2*Wnet_GT(i)+W_ST(i))); % kg/kWhr

end

%=====
%
%               Exergy Analysis of Water Steam Cycle               %
%=====

for i=1:n

% Heat Recovery Steam Generator (HRSG) Analysis

Q_HRSGloss(i) = (1-eta_HRSG)*mdot_gas(i)*(h_d(i)-h_d10(i));
Tj_HRSG(i)=(Td(i)+T_d10(i))/2;
E_1(i)=mdot_HP(i)*x_1(i)/N_boiler;
E_9(i)=mdot_HP(i)*x_9(i)/N_boiler;
E_10(i)=mdot_LP(i)*x_10(i)/N_boiler;
E_14(i)=mdot_LPtoDea*x_14(i);
E_15(i)=(mdot_LPtoDea+mdot_LP(i)/N_boiler)*x_15(i);
E_d10(i)=mdot_gas(i)*xtot_d10(i);

Ed_HRSG(i)=((1-Ta(i)/Tj_HRSG(i))*Q_HRSGloss(i)+E_9(i)+E_15(i)+E_d(i)-
E_1(i)-E_10(i)-E_14(i)-E_d10(i))/1000; % HRSG Exergy Destruction in MW

eta_HRSGII(i)=((1-Ta(i)/Tj_HRSG(i))*Q_HRSGloss(i)+(E_1(i)-
E_9(i))+(E_10(i)+E_14(i)-E_15(i)))/(E_d(i)-E_d10(i));

% Deaerator Analysis

E_cond(i)=mdot_cond(i)*x_cond(i);
E_pegging(i)=mdot_pegging(i)*x_pegging(i);
E_LPtoDea(i)=N_boiler*mdot_LPtoDea*x_LPtoDea(i);
E_dea(i)=(mdot_HP(i)+mdot_LP(i)+mdot_LPtoDea*N_boiler)*x_dea(i);
Ed_Deaerator(i)=(E_cond(i)+E_pegging(i)+E_LPtoDea(i)-E_dea(i))/1000; %
Deaerator Exergy Destruction in MW

eta_DeaeratorII(i)=E_dea(i)/(E_cond(i)+E_pegging(i)+E_LPtoDea(i));

% Feed Water Pumps

E_HPin(i)=mdot_HP(i)*x_dea(i);
Ed_HPFWP(i)=HPFWP(i)+(E_HPin(i)-E_9(i)*N_boiler)/1000; % HP Feed Water
Pumps Exergy Destruction in MW

eta_HPFWPII(i)=(E_9(i)*N_boiler-E_HPin(i))/(HPFWP(i)*1000);
E_LPin(i)=(mdot_LP(i)+mdot_LPtoDea*N_boiler)*x_dea(i);

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Ed_LPFWP(i)=LPFWP(i)+(E_LPin(i)-E_15(i)*N_boiler)/1000; % LP Feed
Water Pumps Exergy Destruction in MW

eta_LPFWP_II(i)=(E_15(i)*N_boiler-E_LPin(i))/(LPFWP(i)*1000);

% Steam Turbine

E_1t(i)=(mdot_HP(i)-mdot_ejector)*x_1(i);

E_2t(i)=(mdot_LP(i)-mdot_pegging(i))*x_10(i);

E_3t(i)=mdot_Ext(i)*xExt_in(i);

E_4t(i)=(mdot_HP(i)-mdot_ejector+mdot_LP(i)-mdot_pegging(i)-
mdot_Ext(i))*x4t(i);

Ed_SteamTurb(i)=(E_1t(i)+E_2t(i)-E_3t(i)-E_4t(i))/1000-W_ST(i); %
Steam Turbine Exergy Destruction in MW

eta_SteamTurb_II(i)=(W_ST(i)*1000)/(E_1t(i)+E_2t(i)-E_3t(i)-E_4t(i));

% Air Cooled Condenser

Eair_in=0;

Eair_out(i)=mfan_air(i)*xair_out(i);

Esteam_ACC(i)=E_4t(i);

Econd_ACC(i)=(mdot_HP(i)-mdot_ejector+mdot_LP(i)-mdot_pegging(i)-
mdot_Ext(i))*x_CEPs(i);

Ed_ACC(i)=P_Fans(i)+(Eair_in+Esteam_ACC(i)-Eair_out(i)-
Econd_ACC(i))/1000; % Air Cooled Condenser Exergy Destruction in MW

eta_ACC_II(i)=(Eair_out(i)+Econd_ACC(i))/(P_Fans(i)*1000+Eair_in+Esteam
_ACC(i));

% Condensate Extraction Pumps

ECEP_in(i)=mdot_cond(i)*x_CEPs(i);

ECEP_out(i)=mdot_cond(i)*x_CEPd(i);

Ed_CEP(i)=CEP(i)+(ECEP_in(i)-ECEP_out(i))/1000; % Condensate
Extraction Pumps Exergy Destruction in MW

eta_CEP_II(i)=(ECEP_out(i)-ECEP_in(i))/(CEP(i)*1000);

% Ejector

Eejector_in(i)=mdot_ejector*x_ejector_in(i);

Eejector_out(i)=mdot_ejector*x_ejector_out(i);

Econd_out(i)=mdot_cond(i)*xcond_out(i);

Ed_Ejector(i)=(Eejector_in(i)+ECEP_out(i)-
(Eejector_out(i)+Econd_out(i)))/1000; % Ejector Exergy Destruction in
MW

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eta_EjectorII(i)=(Eejector_out(i)+Econd_out(i))/(Eejector_in(i)+
ECEP_out(i));

% LP Pre-Heater

E_LPHin(i)=mdot_Ext(i)*xExt_in(i);

E_LPHout(i)=mdot_Ext(i)*xExt_out(i);

Ed_LPH(i)=(E_LPHin(i)+Econd_out(i))-(E_LPHout(i)+E_cond(i))/1000; %
LP Pre-Heater Exergy Destruction in MW

eta_LPHII(i)=(E_LPHout(i)+E_cond(i))/(E_LPHin(i)+Econd_out(i));

end

%=====
%
% EXERGY ANALYSIS
%
%=====

for i=1:n

Ed_TOT(i)=2*(Ed_Comp(i)+ Ed_CombCh(i)+ Ed_GasTurb(i)+ Ed_HRSG(i))+
Ed_Deaerator(i)+ Ed_HPFWP(i)+ Ed_LPFWP(i)+
Ed_SteamTurb(i)+Ed_ACC(i)+Ed_CEP(i)+ Ed_Ejector(i)+ Ed_LPH(i);

Ey_TOT(i)= Ed_TOT(i)/(2*E_fuel(i)/1000);

% Compressor Analysis

Ey_Comp(i)= Ed_Comp(i)/(E_fuel(i)/1000);

Ep_Comp(i)=Ed_Comp(i)/Ed_TOT(i);

% Combustion Analysis

Ey_CombCh(i)=Ed_CombCh(i)/(E_fuel(i)/1000);

Ep_CombCh(i)=Ed_CombCh(i)/Ed_TOT(i);

% Gas Turbine Analysis

Ey_GasTurb(i)=Ed_GasTurb(i)/(E_fuel(i)/1000);

Ep_GasTurb(i)=Ed_GasTurb(i)/Ed_TOT(i);

% Heat Recovery Steam Generator (HRSG) Analysis

Ey_HRSG(i)=Ed_HRSG(i)/(E_fuel(i)/1000);

Ep_HRSG(i)=Ed_HRSG(i)/Ed_TOT(i);

% Deaerator Analysis

Ey_Deaerator(i)=Ed_Deaerator(i)/(2*E_fuel(i)/1000);

Ep_Deaerator(i)=Ed_Deaerator(i)/Ed_TOT(i);

% Feed Water Pumps

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Ey_HPFWP(i)=Ed_HPFWP(i)/(2*E_fuel(i)/1000);
Ep_HPFWP(i)=Ed_HPFWP(i)/Ed_TOT(i);
Ey_LPFWP(i)=Ed_LPFWP(i)/(2*E_fuel(i)/1000);
Ep_LPFWP(i)=Ed_LPFWP(i)/Ed_TOT(i);
% Steam Turbine
Ey_SteamTurb(i)=Ed_SteamTurb(i)/(2*E_fuel(i)/1000);
Ep_SteamTurb(i)=Ed_SteamTurb(i)/Ed_TOT(i);
% Air Cooled Condenser
Ey_ACC(i)=Ed_ACC(i)/(2*E_fuel(i)/1000);
Ep_ACC(i)=Ed_ACC(i)/Ed_TOT(i);
% Condensate Extraction Pumps
Ey_CEP(i)=Ed_CEP(i)/(2*E_fuel(i)/1000);
Ep_CEP(i)=Ed_CEP(i)/Ed_TOT(i);
% Ejector
Ey_Ejector(i)=Ed_Ejector(i)/(2*E_fuel(i)/1000);
Ep_Ejector(i)=Ed_Ejector(i)/Ed_TOT(i);
% LP Pre-Heater
Ey_LPH(i)=Ed_LPH(i)/(2*E_fuel(i)/1000);
Ep_LPH(i)=Ed_LPH(i)/Ed_TOT(i);
end

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