PERFORMANCE OF COMBINED CYCLE POWER PLANTS WITH EXTERNAL COMBUSTION

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ABSTRACT

PERFORMANCE OF COMBINED CYCLE POWER PLANTS WITH EXTERNAL COMBUSTİON

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Externally Fired Combined Cycle (EFCC) is a recent technology which utilizes coal as the primary fuel through a gas turbine cycle. EFCC is among the clean coal technologies aiming at improving the coal combustion performance in a cost effective and environmentally friendly manner. Natural gas is used as the supplementary fuel. The ultimate goal of the EFCC concept is coal only option, since natural gas can better be utilized in a modern CCGT plant.

In this study, performance of the EFCC plant is evaluated based on 1st and 2nd law analysis. Different levels of supplementary firing are included in the analyses together with the coal only option. Effect of certain plant parameters on the performance indicators are analyzed. Exergy distibution throughout the plant is obtained It is shown that air heater is the most significant component in terms of irreversibilities.

It is also desired to evaluate the performance of the plant, in economic terms. Thermoeconomic analyses are performed for this purpose, based on the SPECO method and the results of exergy related analysis. Cost balance equations are derived, together with the auxiliary equations formed by using F and P rules, and the resulting set of equations are solved. Cost formation processes are investigated and levelized cost values per unit exergy unit are obtained for each material stream in the plant. It is shown that highest values of exergy unit cost values are observed at the exit stages of compressor and pump.

Results of this study indicate that supplementary firing improves the plant performance, whereas cost of products also increase with supplementary firing. Another observation is that EFCC plant offers performance figures comparable to those for advanced power plant technologies such as IGCC.

Keywords : EFCC, air heater, supplementary firing, first law analysis, second law analysis, thermoeconomics, SPECO method

ÖZ

DIŞTAN YANMALI KOMBİNE ÇEVRİM ENERJİ SANTRALLERİNİN PERFORMANS İNCELEMESİ

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Dıştan Yanmalı Kombine Çevrim santral teknolojisi, bir gaz türbini çevriminde ana yakıt olarak kömür kullanan yeni bir elektrik üretim teknolojisidir. Dıştan Yanmalı Kombine Çevrim teknolojisi kömür yakma performansının çevreyle uyumlu ve maliyet etkin bir şekilde geliştirilmesine yönelik temiz kömür yakma teknolojileri arasındadır. Tamamlayıcı yakıt olarak doğal gaz kullanan bu teknolojide ulaşılmaya çalışılan nihai hedef sadece kömürün yakıt olarak kullanılmasıdır, zira doğal gaz modern bir Kombine Çevrim Güç Santralinde daha etkin bir şekilde kullanılabilmektedir.

Bu çalışmada, Dıştan Yanmalı Kombine Çevrim santralinin performansı birinci ve ikinci kanun çerçevesinde değerlendirilmiştir. Yakıt olarak sadece kömürün kullandığı seçeneğin yanısıra farklı tamamlayıcı yakma seviyeleri de analizlere dahil edilmiştir. Belirli santral parameterelerinin performansa ilişkin göstergeler üzerindeki etkileri incelenmiştir. Tüm çevrim için ekserji dağılımı elde edilmiştir. En fazla tersinmezliğin hava ısıtıcısınca gerçekleştiği görülmüştür.

Bu çalışmada ayrıca, santralin performansının ekonomik yönden de incelenmesi amaçlanmıştır. SPECO yöntemi ve ekserji ile ilgili analizlerin sonuçları kullanılarak termoekonomik analizler gerçekleştirilmiştir. Maliyete ilişkin denklemler ve yardımcı eşitlikler türetilerek oluşturulan denklem seti çözülmüştür. Maliyet oluşum süreçleri incelenmiş ve santraldeki her bir madde akışı için birim ekserji başına bir değere indirgenmiş maliyet değerleri elde edilmiştir. En yüksek birim maliyet değerlerinin kompresör ve pompa çıkışlarında oluştuğu gözlemlenmiştir.

Bu çalışmanın sonuçları, tamamlayıcı yakma işleminin santral performansını iyileştirmekle birlikte, ürün maliyetlerini de artırdığını göstermektedir. Çalışma sonuçları, aynı zamanda, Dıştan Yanmalı Kombine Çevrim santral teknolojisinin Entegre Kömür Gazlaştırma Kombine Çevrim teknolojisi gibi gelişmiş elektrik üretim teknolojileri ile karşılaştırılabilir düzeyde performans sergilediğine işaret etmektedir.

Anahtar Kelimeler : Dıştan Yanmalı Kombine Çevrim, hava ısıtıcısı, tamamlayıcı yakma, birinci kanun analizi, ikinci kanun analizi, termoekonomi, SPECO yöntemi

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LIST OF SYMBOLS

EFCC	Externally Fired Combined Cycle
LHV	Lower heating value (kJ/kmol)
IGCC	Coal Gasification Combined Cycle
PFBC	Pressurized Fluidized Bed Combustion
HRSG	Heat Recovery Steam Generator
PC	Pulverized Coal
CCGT	Combined Cycle Gas Turbine
ST	Steam turbine
GT	Gas turbine
CerHx	Ceramic heat exchanger
REACH	Radiatively enhanced aerodynamically cleaned heat exchanger
η_I	Thermal efficiency of plant
η_R	Rational efficiency
$(W_{el})_{GC}$	Work produced by the gas cycle
$(W_{el})_{SC}$	Work produced by the steam cycle
E_f	Energy of fuel input
TIT	Turbine inlet temperature

r_p	Compressor presure ratio
P_b	Boiler pressure
δ	Supplementary firing level
ΔT_{PP}	Pinch point difference
ΔT_{me}	Temperature difference at the exit of the hot side of HRSG
T_m	Temperature at which the hot gases entering the heat recovery steam generator (K)
T_g	Temperature at which the hot gases exit the adiabatic combustion space (K)
E_i	Rate of exergy inflow
E^Q	Exergy flow associated with heat transfer
E_e	Rate of exergy outflow
E^W	Exergy flow associated with work transfer
$\dot{I}$	Rate of irreversibility
$\dot{E}_{ch}$	Chemical component of flow exergy
$\dot{E}_{ph}$	Physical component of flow exergy
$\tilde{\varepsilon}_{ph}$	Physical component of exergy
$\tilde{\varepsilon}^{\Delta T}$	Thermal part of physical exergy
$\tilde{\varepsilon}^{\Delta P}$	Pressure part of physical exergy
K_p	Equilibrium constant
$\tilde{c}_p^{\ \varepsilon}$	Mean molar isobaric exergy capacity
$\Delta\tilde{s}^0$	Entropy of reaction

$(NCV)^0$	Net calorific value
φ	Ratio of chemical exergy to net calorific value
$\dot{Z}_k^{CI}$	Capital investment cost of kth component
$\dot{Z}_k^{OM}$	Operating and maintanance costs of kth component
$\dot{C}_i$	Cost rate associated with stream *i* (\$/h)
c_i	Average cost per unit exergy (\$/GJ)
N_e	Number of exergy streams exiting the component
$N_{e,F}$	Number of exiting exergy streams associated with the definition of the fuel
$N_{e,P}$	Number of exiting exergy streams included in the product definition.
PEC	Purchased equipment cost

CHAPTER 1

INTRODUCTION

Coal has the largest share in world's proven fossil fuel resources and provides 39 % of world's electricity. Coal will remain as the largest energy source for power generation in the next two decades at least, with abundant and widely dispersed proven reserves throughout the world. [1]

Coal-fired plants represent around 25% of total installed capacity in Turkey. Coal is the second largest energy source after natural gas in electricity generation of Turkey, with around 31% share. [2]

Due to environmental concerns, research activities have been concentrated on innovative coal-fired plant technologies recently. The rate at which clean coal technologies are adopted and the scope at which they are put into place will both be crucial for future coal use in power generation [1]. The long term prospects is expected to be dependent mainly on development of coal combustion technologies that reduce emissions and achieve better performance figures with higher efficiencies.

The conventional mechanism used to generate electricity from coal is direct fired boiler. Utilization of the work potential is limited due to thermodynamic and metallurgical considerations in direct fired power plants. The large temperature difference between the hot coal combustion gases and the working medium is an intrinsic handicap for direct fired boiler systems, which can be overcome by employing a gas turbine cycle.

There are several technologies being developed, to better utilize the work potential of coal combustion in a gas turbine cycle. These are called

Integrated Coal Gasification Combined Cycle (IGCC), Pressurized Fluidized Bed Combustion (PFBC), and Externally Fired Combined Cycle (EFCC).

EFCC is an extension of the well established combined cycle, which is a currently popular technology for power generation using natural gas as the supplementary fuel of choice. EFCC benefits from the advantages of the combined cycle system, while introducing the coal as the main fuel of choice. Desired level of gas turbine inlet temperature can be obtained by employing a specially designed air heater.

Suitable turbine inlet temperature levels for the gas turbine can also be attained by including supplementary firing into the commercial heat exchanging equipment. Natural gas is utilized as the supplementary fuel for that purpose. The ultimate goal for the EFCC plant is the coal only option, where no supplementary firing is needed.

In this study, performance of an Externally Fired Combined Cycle (EFCC) is evaluated. For this purpose, first law and second law analysis for the EFCC is performed. A database is established and employed in energy and exergy related calculations. The performance of the cycle is analyzed based on the variations in selected parameters. A Visual C++ program is developed for that purpose. The effect of different levels of supplementary firing on the EFCC plant performance is also investigated. Exergy distribution throughout the plant is analyzed and rate of irreversibilities are determined on a component basis.

It is also desired in this study to analyze the thermoeconomics of the EFCC plant. A general assessment of plant economics is coupled with the exergy analysis to reach a more meaningful performance assessment for the plant. Performance of the EFCC plant is compared with several other advanced plant technologies.

Chapter 2 contains the information about the main motivation behind this study. Working principle of the EFCC plant is briefly explained and historical development of the EFCC is outlined together with the current research and development activities and needs.

First law analysis is performed in Chapter 3.. First law efficiency and the specific work values are obtained for different values of pressure ratio, boiler pressure, and turbine inlet temperature. Effect of supplementary firing is also investigated.

Second law analysis of the EFCC plant is done in Chapter 4. Rational efficiency values are obtained for different supplementary firing levels. Exergy losses are determined for each component of the cycle, based on different values of the parameters selected.

Chapter 5 deals with the thermoeconomic analysis, which is an essential part to perform a more meaningful performance analysis. Cost formation processes are analyzed and the cost breakdown throughout the cycle is obtained.

Results of the study are given in Chapter 6. Performance of the EFCC plant is compared with the information found in the literature for plants with similar purposes. Different supplementary firing levels are included in the discussions. Concluding remarks and recommendations for the extension of this study are also given in Chapter 6.

CHAPTER 2

EXTERNALLY FIRED COMBINED CYCLE

2.1. Introduction

The predominant mechanism for electricity generation from coal is the direct fired boiler. In direct coal-fired power plants, heat exchange between the hot products of coal combustion and the water takes place in a boiler, and the steam is utilized in a steam turbine to produce work.

The efficiency of direct coal-fired plants are rather low, due to thermodynamic and metallurgic constraints. Modern coal-fired power plants achieve efficiency figures around 38-40% (LHV). Efficiency figures are expected to be not higher than 47% (LHV) for supercritical boilers which operate at pressures around 350 bar and temperatures of 700 ^{0}C. On the other hand, natural gas-fired combined cycle plants achieve 45-52% efficiency (LHV) at relatively moderate parameters. More advanced plants based on natural gas firing can even achieve efficiency figures around 57% (LHV).

This contrast in the efficiency figures is due to the large temperature difference between the coal combustion gases and the working medium, which is more than 1200 K for a typical coal-fired plant. Better utilization of the work potential is possible using a gas turbine cycle that covers a temperature range of 400^{o}C-1400^{o}C.

Since coal is a solid fuel, it is not quite suitable for direct utilization in a gas turbine. An indirect mechanism is therefore needed to overcome this unsuitability and to transfer the chemical exergy of coal to the turbine in an effective way. The technologies utilizing this kind of mechanisms are called "advanced coal conversion technologies".

Integrated Coal Gasification Combined Cycle (IGCC) and the Pressurized Fluidized Bed Combustion (PFBC) are the main advanced coal conversion technologies, designed and being improved to take the advantages of a gas turbine cycle, with coal as the fuel of choice.

Coal is gasified and fed to the turbine upon clean up process in an IGCC plant. A steam boiler is utilized to recover the exhaust heat from the gas turbine. Gas cooling and clean up requires a number of critical components, which, in turn, yields rather high level of complexity and capital costs.

Combustion of coal in a fluidized bed under pressure integrated with a gas turbine is another advanced coal conversion technology called PFBC. The gas turbine is driven by compressed air which is heated in a fluidized bed. PFBC technology is also characterized by its complexity and high capital costs.

Externally Fired Combined Cycle (EFCC) is another technology being developed for effective conversion of coal into electricity by employing a Gas Turbine Cycle.

2.2. EFCC Plant

Externally Fired Combined Cycle (EFCC) is an extension of the well established combined cycle, which is currently popular for power generation applications throughout the world, with natural gas as the fuel of choice.

EFCC plant makes use of a gas turbine as well, but avoids coal combustion gases entering the turbo-machinery. Heat addition in the gas turbine cycle occurs indirectly in a high temperature heat exchanger, which is located in a coal furnace. The exclusion of coal combustion products from the gas turbine avoids the expense of hot gas clean-up and corrosion of turbine blades by coal ash. Since the gas turbine operates on indirectly heated air, it can be totally protected from the impurities in the fuel.

The main advantage of EFCC is that the problems caused by inorganic constituents of coal are switched from the high temperature rotating parts of the gas turbine to the high temperature stationary parts of a heat exchanger.

After expansion in the turbine, air can be partly redirected to the furnace, to supply preheated air for combustion. It is also possible to pass the air to a heat recovery steam generator. Schematic representation of EFCC plant with a specifically designed air heater is given in Figure 2.1.(a)

EFCC offers an advantage in terms of heat utilization, as compared with the direct fired boiler. Air does not require high pressures at high temperatures, in spite of its lower specific heat value than steam. Since the heat exchange between air and the hot combustion gases occur in the gaseous state, the average temperature difference experienced in the heat exchanger is smaller than in the case of steam.

The most critical component of the EFCC system is the high temperature heat exchanger, where air heating takes place. All the other equipment is in use in different plant technologies and of proven technology [3].

It is also possible to modify the existing steam power plants by re-powering. Air heating is allowed to occur in the existing furnace, instead of including a specifically designed high temperature air furnace. This concept may require major reconstruction of the steam boiler. Schematic representation for a EFCC plant in re-powering configuration is given in Figure 2.1.(b).

Another configuration was proposed by Huang [4]. In this system, air is first compressed in the compressor. After being heated to a desirable temperature in the air heater, air expands in the air turbine to produce part of the power output of the combined cycle power plant. Air leaving the air turbine is then used as the combustion air. The hot gas from the air heater is the heat source for the steam power plant. The amount of power output obtainable from the bottoming cycle can be controlled by varying the amount of fuel input to the combustor.

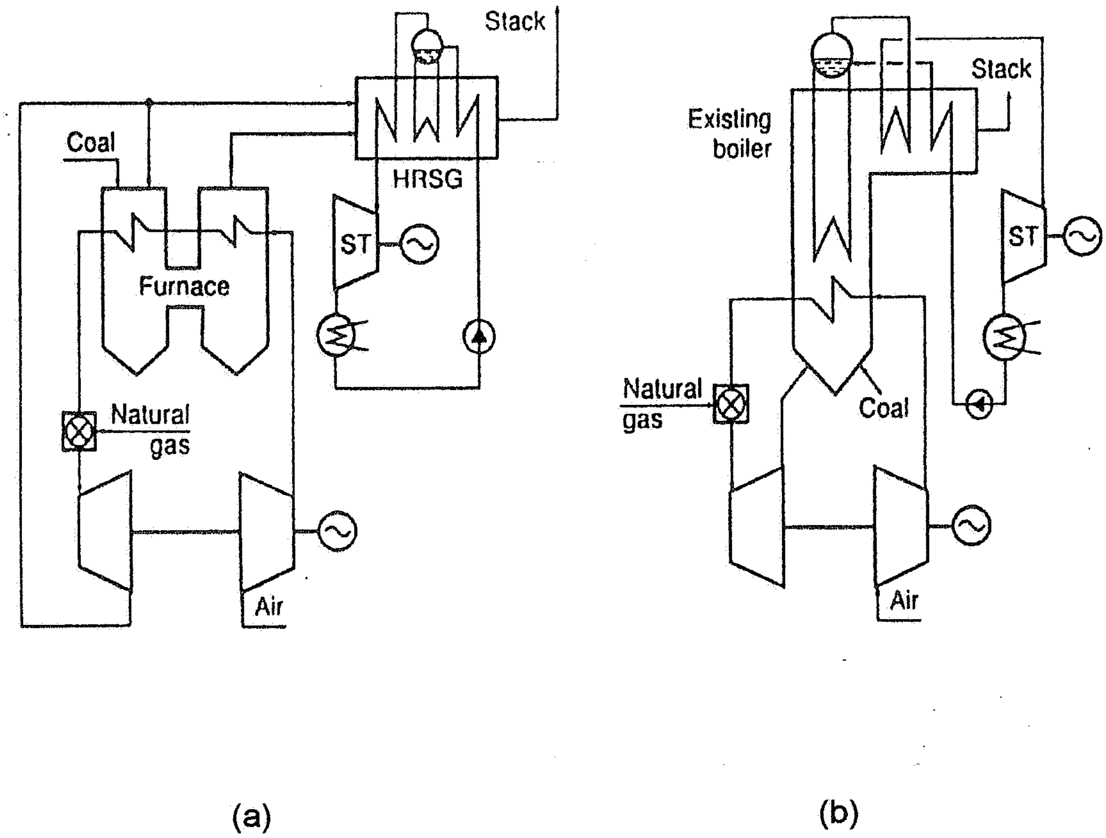


Figure 2.1: Schematic Diagram of an EFCC plant (a) with a specifically designed air heater; b) in re-powering configuration [3]

EFCC plants can achieve 40-49% efficiency values (LHV), depending on the turbine inlet temperature and other parameters [3]. This brings a considerable performance gain, as compared with the modern coal fired plants with direct fired boiler. It was also shown that EFCC plants can operate at efficiency values similar to those of other advanced coal-fired plants such as IGCC.

2.3. Historical Development of the EFCC

An externally-fired gas turbine cycle was proposed first as a possible realization of the Joule-Brayton cycle. However, much interest has been regained in the last decade, and several research activities have been launched in different countries.

In United States this concept has been developed by a group of companies, including Hague International, the developer of the high temperature ceramic heat exchanger. Another group, under the management of the Pittsburg Energy Technology, has been working with industries to develop a similar power cycle called the High Performance Power System. The air heater concept is being designed by the Foster Wheeler and United Technologies teams.

In Europe, works on EFCC is being carried out in Italy by a consortium of Ansaldo, ENEL, and several research institutions, with funding from the European Union. The Netherlands Agency for Energy and Environment (NOVEM) has included EFCC in the New Energy Conversion Technologies program for technical and economic evaluation.

Engineering performance and cost models of the EFCC technology was evaluated in a previous study. A probabilistic modelling capability is incorporated to account for the uncertainties in the performance and cost parameters. EFCC technology is probabilistically compared with the competing advanced coal based power generating technologies, in order to identify the risks and potential pay-offs of the EFCC technology relative to the other technologies.[5]

The main obstacle in the development of a viable EFCC system has been the unavailability of a suitable heat exchanger, which would allow for a sufficiently high gas turbine inlet temperature. Recent developments in ceramic heat exchanger (CerHx) technology for use in the EFCC process have been promising enough to warrant further development of this system [5].

Radiatively Enhanced, Aerodynamically, Cleaned Heat Exchanger (REACH Exchanger) is also among the heat exchanger concepts being developed. The REACH Exchanger is fired by radiative and convective heat transfer from a moderately clean fuel stream and radiative heat transfer from the flame of a much larger un-cleaned fuel stream, which supplies most of the heat [6]. Computation and experiments both indicate that the ceramic heat exchanger can be aerodynamically protected by a tertiary stream with an acceptably low flow rate. It was demonstrated in another study that the use of ceramic oxide coatings for silicon carbide, which is currently used as a structural material for heat exchanger tubes, can improve the corrosion.

Economy and size related considerations of ceramic heat exchangers give the possibility of using metallic heat exchangers with the supplementary firing before the turbine inlet. Due to heat transfer characteristics of ceramic heat exchangers, rather large heat transfer areas are required, which in turn results in greater costs. Nickel based superalloys allows operation at 800 - 825 ^{0}C. Oxygen dispersion alloys can withstand temperatures around 950 ^{0}C. Temperature values up to 1075 ^{0}C were achieved in the experiments reported by Solomon [6]. The level of 1370 ^{0}C was anticipated in the works of Van Roode [7].

CHAPTER 3

ENERGY ANALYSIS OF EFCC POWER PLANT

3.1. Introduction

Determination and maximization of plant efficiency is the major objective of thermodynamic analysis, with the most efficient production of work from a supply of fuel with chemical energy being the major concern. 1st law analysis is the starting point for thermodynamic analysis of power plants.

In this chapter, energy analysis for the EFCC plant is performed. It is rather hard to investigate the overall performance of the plant, based on the changes in many parameters. Therefore, several approaches are considered, to perform a thermodynamic analysis of the power cycles. In this study, Cerri's approach for open circuit / closed cycle plants is considered as a basis, and parameters are selected accordingly. Governing equations are derived using energy balance throughout the cycle. Lower heating value efficiency (LHV) and the specific work values are obtained for different values of pressure ratio, boiler pressure, and turbine inlet temperature, which are the selected parameters. Contribution of steam and gas cycles on the plant performance is analyzed. Effect of supplementary firing is also investigated.

3.2. Plant Configuration

EFCC plant examined in this study consists of a gas turbine cycle and a bottoming steam cycle. Combining these two independent cycles in an appropriate manner yields higher efficiency than that for each of the cycles operating independently. Schematic representation of the EFCC plant analyzed in this study is illustrated in Figure 3.1.

The major components of the gas turbine cycle are a gas turbine, an air heater, a compressor, and a supplementary firing unit. Heat addition to the compressed air takes place in the air heater. Heated air is transferred to the gas turbine, either directly or after further heating in the supplementary firing unit.

At a given turbine inlet temperature, the amount of fuel consumed in the supplementary firing unit is determined solely by the temperature level in the air heater. Methane is burned for the purpose of supplementary firing. Steam sections for all supplementary firing configurations are consisted of a single pressure boiler.

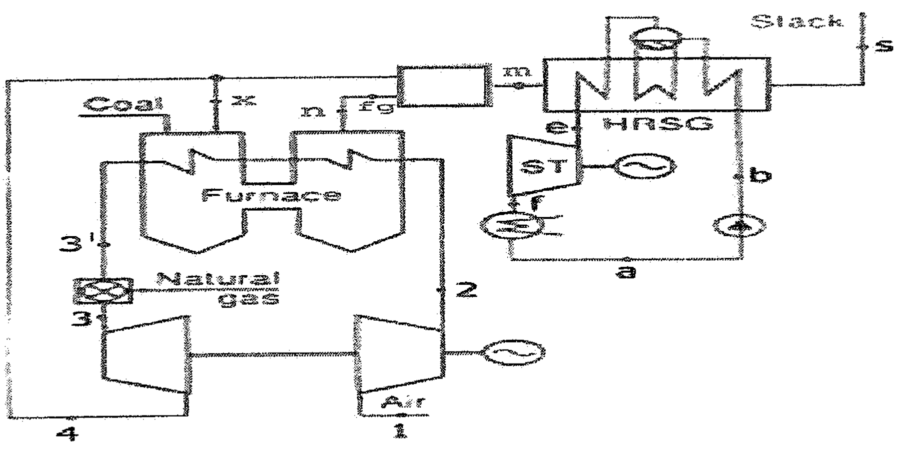


Figure 3.1. Schematic representation of the EFCC Plant

The steam cycle completes the combined cycle plant. This Bottoming steam cycle is mainly composed of a steam turbine, a steam condenser with its cooling system, and a pump.

Energy transfer between the two cycles is realized by means of a heat recovery steam generator. The flue gas temperature at the inlet of the heat recovery steam generator is kept constant at a certain temperature level by means of the gas turbine outlet flow, which is directed partly to the coal furnace.

The flue gases exiting the air heater and the expanded air are mixed before entering the heat recovery steam generator. Combustion products are also directed to the mixing unit if supplementary firing is employed.

Since the EFCC plant involves the coupling of a topping gas turbine cycle with a bottoming steam cycle, the overall system performance is influenced by the design parameters of both cycles. Optimum performance of the combined cycle system is achieved only if the gas turbine and steam turbine cycles are matched properly [8].

3.3. Plant Modeling

The useful products of any combined cycle plant are the power outputs of the steam and gas turbines. The common parameter used to assess the thermodynamic performance of such a plant is its thermal efficiency. Since this definition is based only on energy considerations, the thermal efficiency is also called 1st law efficiency of the plant, and is given by

$$\eta_I = \frac{(W_{el})_{GC} + (W_{el})_{SC}}{E_f} \tag{3.1}$$

where $(W_{el})_{GC}$ is the work produced by the gas cycle, $(W_{el})_{SC}$ is the work produced by the steam cycle, and E_f is the energy of fuel input.

1st law efficiency (LHV) of any thermodynamic cycle depends on many parameters. Some of the parameters have physical limitations due to environmental conditions, while some parameters are bounded by metallurgical constraints. Increasing the performance of a plant requires one to analyze the effect of parameters that can be subject to changes without environmental and metallurgical constraints.

It is rather hard to express the change in plant efficiency with changes in many parameters, some of which are interrelated. Therefore, several approaches have been developed to represent the plant efficiency as a function of a few independent parameters [9]. These approaches aim to reach accuracy and simplicity at the same time, and imply that a number of other parameters are pre-selected and held constant.

In this study, plant efficiency is assumed to be a function of turbine inlet temperature, pressure ratio, boiler pressure, and supplementary firing level.

$$\eta = f(TIT, r_p, P_b, \delta) \tag{3.2}$$

where *TIT* is the turbine inlet temperature, r_p is the pressure ratio, P_b is the boiler pressure, and δ, supplementary firing level, is the ratio of heat supplied in the supplementary firing unit to that supplied in the air heater. Performance assessment for the plant is carried out based on the variations in these four selected parameters. Other parameters affecting the performance of the plant are held constant as implied by the approach selected. The pre-selected parameters for the performance evaluation study of the EFCC plant are tabulated in Table 3.1. Parameters related with the HRSG can be seen in Figure 3.2.

In this study, pressure drop is neglected for all components of the cycle. Adiabatic combustion approximation is followed and heat losses in all components are neglected. Ideal gas model is employed throughout this study.

Table 3.1 Pre-selected parameters for the EFCC Plant

Compressor Inlet Temperature	288 K
Compressor Isentropic Efficiency	0.88
Gas Turbine Isentropic Efficiency	0.856
Pinch Point Difference (ΔT_{PP})	20 K
$T_m - T_c$ (ΔT_{me})	25 K
Steam Turbine Isentropic Efficiency	0.88
Pump Isentropic Efficiency	0.86
Condenser Pressure	5 kPa

Supplementary firing level indicates the extent to which natural gas is utilized within the plant. Since it is not quite simple to reach the turbine inlet temperature values where gas turbine offers significant advantages, combustion of natural gas is allowed to take place in the supplementary firing unit to overcome this shortcoming. In this study, three different supplementary firing levels which correspond to certain exit temperature values for the air heater are investigated.

Heat recovery steam generator is simply a counter flow heat exchanger where the energy transfer between hot gases and the water takes place. Figure 3.2 illustrates the temperature profile at each point in the water (steam) and gas path in the heat recovery steam generator. The location at which the temperature difference experiences its minimum is called pinch point temperature [10]. The minimum temperature difference (pinch point temperature) occurs when the water enters the evaporator as saturated liquid. Pinch point temperature is one of the key factors in the performance of a combined cycle plant [11]. In this study, pinch point difference, ΔT_{PP}, is taken as 20 K, which is a typical value.

Since T_m, temperature at which the hot gases entering the heat recovery steam generator, is determined based on the results of the gas turbine cycle analysis, T_e' can be calculated provided that the temperature difference at the exit of the hot side of the heat recovery steam generator, ΔT_{me}, is taken constant. Hence, the flow rate ratio of two cycles can be evaluated using an energy balance including the evaporator and superheater sections of the heat recovery steam generator. Isentropic efficiency values are selected around the typical values that are observed in operating plants of similar types [12].

Air heater is the key component for the EFCC plant where coal is utilized as an energy source in the gas cycle. Air, after expansion in the gas turbine, enters the adiabatic combustion space with coal, and heat is transferred to the compressed air. Schematic diagram of the air heater is shown in Figure 3.3.

In order to determine the temperature T_g at which the hot gases exit the adiabatic combustion space requires, an iterative procedure based on an energy balance is applied. Amount of coal required per unit air flow is then calculated using an energy balance for the region where heat is transferred to the compressed air.

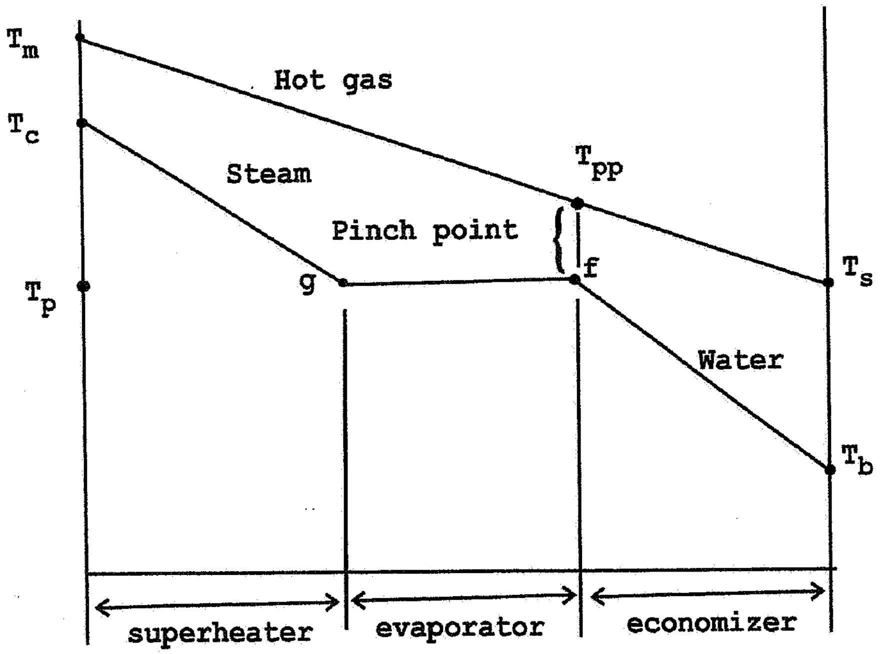


Figure 3.2. Temperature Profile in Heat Recovery Steam Generator

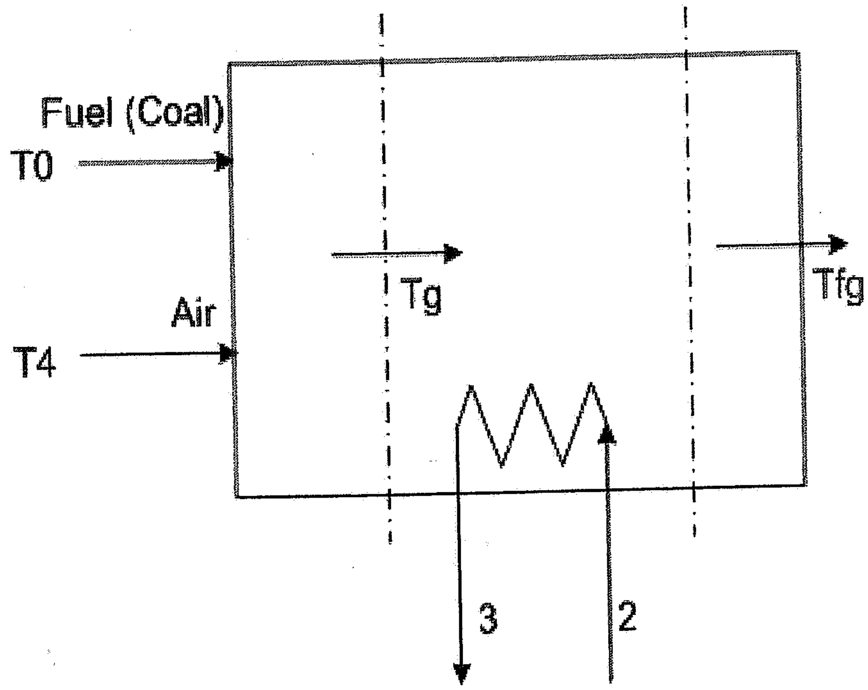


Figure 3.3. Schematic representation of the Air Heater

Table 3.2 Composition of Anthracite Coal [12]

Constituent	Mass Fraction (%)
Carbon	78.2
Hydrogen	2.4
Nitrogen	0.9
Sulphur	1.0
Oxygen	1.5
Water	8.0
Ash	8.0

In this study, anthracite coal and air excess of 30 % over the stoichiometric requirement are used through the calculations. Composition of the anthracite coal is given in Table 3.2.

Environmental effects play an important role in design of modern coal fired power plants. Clean coal technologies are developed with the aims of reducing the emissions in a cost effective way. In this study SO_2 and NO emissions are analyzed to reach an estimate of environmental performance.

It is rather hard to model the NO formation, since it requires the understanding of the complex chemical phenomena involved. NO_x is produced in burners through three mechanisms. The first mechanism is called as thermal NO_x where NO_x is formed in the hottest part of the flame via a high temperature reaction of nitrogen and oxygen existing in the air. Other mechanisms are called prompt NO_x and fuel NO_x, which are of less importance as compared with the thermal NO_x. Thermal NO_x is modeled in this study.

Amount of NO produced is calculated under equilibrium conditions, with ideal gas assumption. The calculations are performed for the temperature at which the hot gases exit the adiabatic combustion space.

Equilibrium constant is given by the relation

$$K_p(T) = \left(\frac{P}{P_0}\right)^b \prod_{i=1}^{m+n} x_i^{v_i} \tag{3.3}$$

where b is the algebraic sum of all the stoichiometric coefficients. Considering the reaction of formation

$$\frac{1}{2}O_2 + \frac{1}{2}N_2 \rightarrow NO \tag{3.4}$$

the mole fractions can be determined using the tabulated values found in the literature [13].

A Visual C++ program is developed, to accomplish the performance assessment for the EFCC plant. Visual C++ is an object oriented programming language that increases software extensibility and reusability, and controls the complexity and cost of software maintenance.

Each component of the EFCC plant is modeled as an individual module. A database is created using MS Access, based on the thermodynamic properties compiled from the literature [12, 14]. Modules in the Visual C++ program are linked properly to constitute the overall performance assessment steps. Performance evaluation steps are outlined in Appendix A.

3.4. Results of Performance Evaluation

1st law efficiency is modeled as a function of three parameters, in addition to the level of supplementary firing, for the whole plant. Analyzing the effect of each parameter independently requires one to fix the other parameters within a reasonable range. A previous study by Korobitzyn with similar purposes suggests typical values where the parameters can be taken constant [3].

In this study, the following values are selected, in order to form a basis for the performance related calculations: TIT=1400 K, r_p=13.7, P_b=14 MPa. These values correspond to the preferred working conditions for the gas turbine engine and the boiler [3]. Calculation procedure for energy analysis is described in Appendix B.

Figure 3.4 illustrates the change in LHV efficiency for the plant with main steam pressure. As main steam pressure increases, LHV efficiency of the cycle also increases. Change in specific work output of the plant with respect to main steam pressure is given in Figure 3.5. As can be seen from the figure, specific work output increases with increasing main steam pressure. Although it is possible to get higher efficiency and work at the same time by increasing the pressure of the main steam, metallurgical constraints impose limitations for performance improvement.

The plots of plant LHV efficiency and specific work for the cycle as a function of turbine inlet temperature are given in Figure 3.6 and Figure 3.7, respectively. Both LHV efficiency and specific work increases as *TIT* is increased. However, turbine inlet temperature is constrained by gas turbine material considerations.

The effect of pressure ratio on the plant performance in terms of LHV efficiency and specific work is shown in Figure 3.8 and Figure 3.9, in respect. Figure 3.8 points out that there exists an optimum pressure ratio for which the LHV efficiency reaches its maximum; with *TIT* and P_b are both kept constant.

On the other hand, specific work output of the EFCC plant decreases with increasing pressure ratio, as shown in Figure 3.9. This is attributed to the fact that, reduction in gas turbine exhaust temperature, which in turn reduces the maximum steam temperature and pressure, thus results in lower steam cycle output.

Change in stack gas temperature with compressor pressure ratio is illustrated in Figure 3.10. Increasing the pressure ratio yields an increasing trend for stack gas temperature, as expected. This trend is experienced in analyzing the effect of main steam pressure as well, as illustrated in Figure 3.11. Cerri argues that there is a minimum stack gas temperature, which is around 430 K, to prevent corrosive condensation in the heat recovery steam generator [9]. Temperature range covering this temperature is included in the analysis.

Figure 3.12 illustrates how contribution of each cycle in power production changes with changing main steam pressure. Analysis shows that gas turbine produces more power than steam turbine, regardless of the main steam pressure. However, as the main steam pressure increases, power output ratio of gas engine to steam turbine decreases.

Effect of compressor pressure ratio on breakdown of power produced by the plant is investigated and illustrated in Figure 3.13. For fixed *TIT* and P_b, contribution of each cycle is the same around r_p=10. Gas turbine cycle produces 20 % more power than the steam cycle for r_p values around 14. On the other hand, power produced by the steam cycle is around 30% higher than that by the gas turbine engine for r_p values close to 6. This analysis also shows that gas turbine engine to steam power plant power ratio is 1.17, for typical working conditions for the EFCC plant analyzed in this study.

An important purpose of this study is to investigate the effect of supplementary firing on the plant performance. Since the ultimate goal of the EFCC concept is to realize the coal-only option, performance assessment of the plant with respect to different supplementary firing levels is of importance in improving the performance.

Figure 3.14 illustrates the degradation in plant performance in terms of LHV efficiency, as supplementary firing level is decreased. Higher supplementary firing level means that more methane is injected into the system. Therefore, mass flow rate in the gas cycle is increased, which, in turn, increases the power produced by the gas turbine engine.

Same considerations are valid for the effect on specific work output for the plant. As can be seen from Figure 3.15, coal only option yields 25 % less power production than that for the 52 % supplementary firing case. Nevertheless, it is not adequate to assess the performance gains of supplementary firing based solely on energy balance. Thermoeconomic analysis coupled with 2nd law investigations leads to more realistic conclusions.

Figure 3.16 illustrates the change in NO emissions with compressor pressure ratio. Selecting pressure ratio in the upper range improves the plants performance regarding NO emissions. Amount of energy required to reach a certain *TIT* decreases, as larger values for r_p are selected. Consequently, the coal requirement is decreased, which implies that less NO is released to the environment.

NO emissions are analyzed for the EFCC plant under consideration. As can be seen from Figure 3.17, NO emissions increase with increasing turbine inlet temperature. This is attributed to the fact that, more coal is required per unit air to reach higher TIT values.

Effect of pressure ratio on plant's SO_2 emissions is shown in Figure 3.18. Same observations and considerations are valid for NO case. Environmental emissions related analyses performed in this study are based on no supplementary firing conditions.

SO_2 emissions are also investigated, since its one of the most determinant factors regarding environment related performance of the plant. SO_2 emissions increase with increasing turbine inlet temperature, as illustrated in Figure 3.19. This is again an expected behavior, since higher amount of coal is required for a given amount of air to reach higher *TIT* values, with r_p and P_b are both held constant.

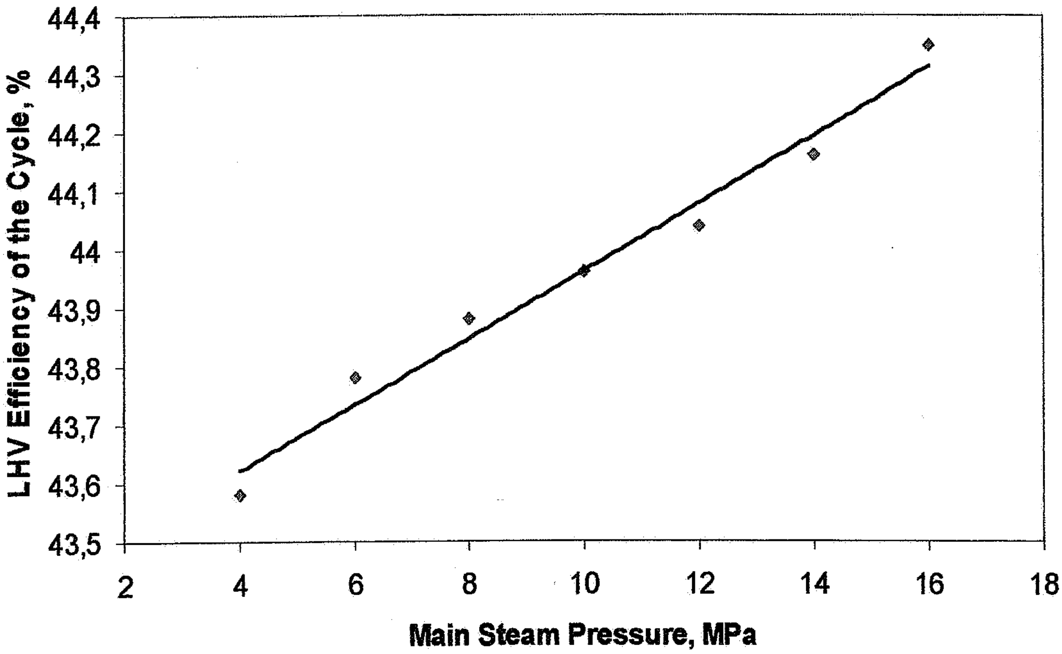


Figure 3.4 LHV efficiency of EFCC plant versus main steam pressure at TIT = 1438K and r_p = 13.7

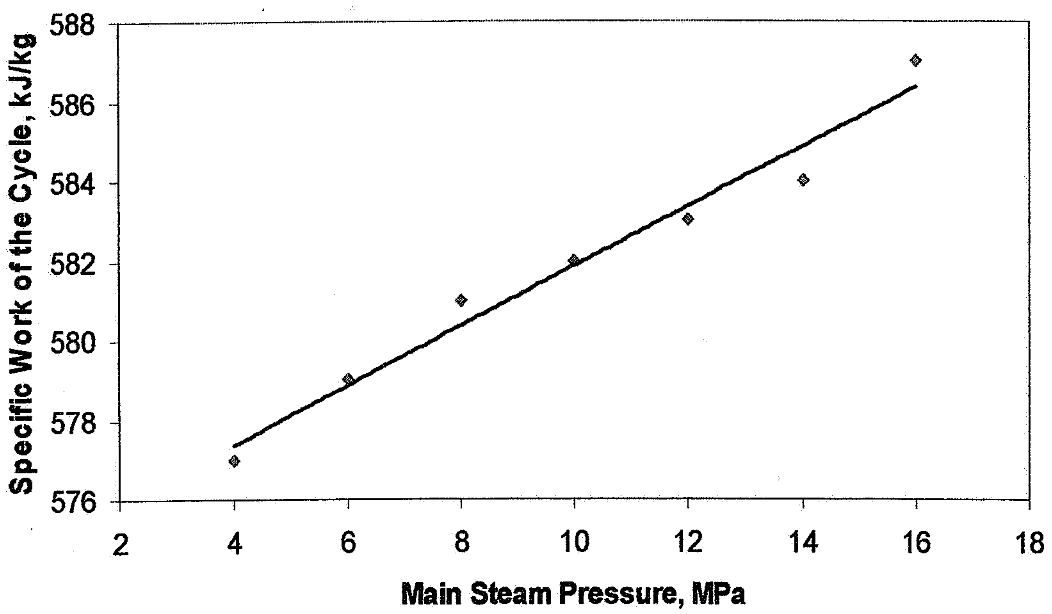


Figure 3.5 Specific work of EFCC plant versus main steam pressure at TIT = 1438K and r_p = 13.7

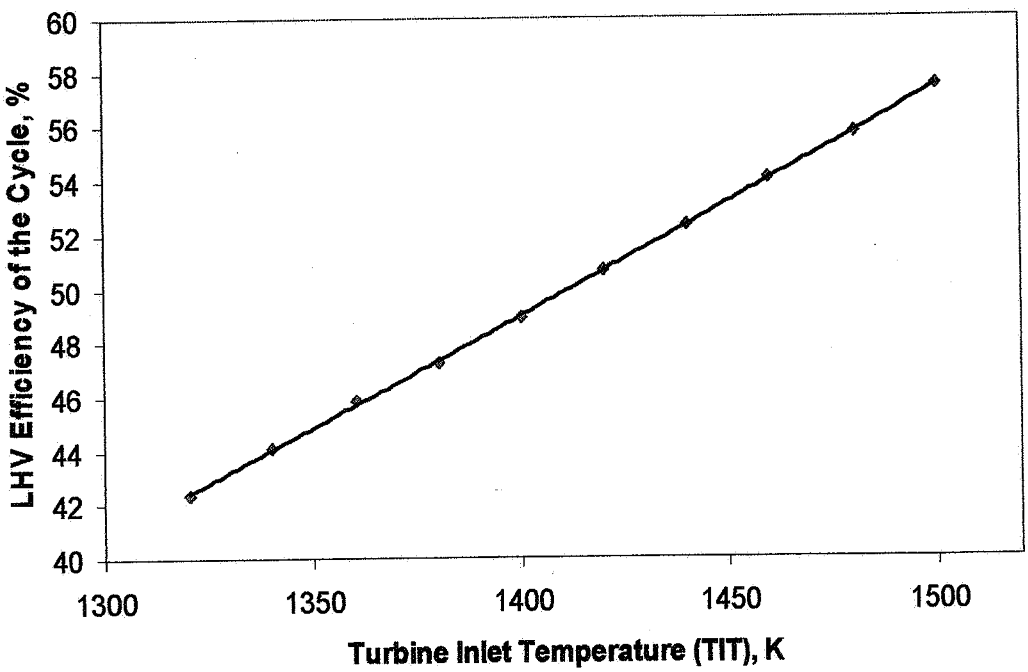


Figure 3.6 LHV efficiency of EFCC plant versus turbine inlet temperature at P_b = 14 MPa and r_p = 13.7

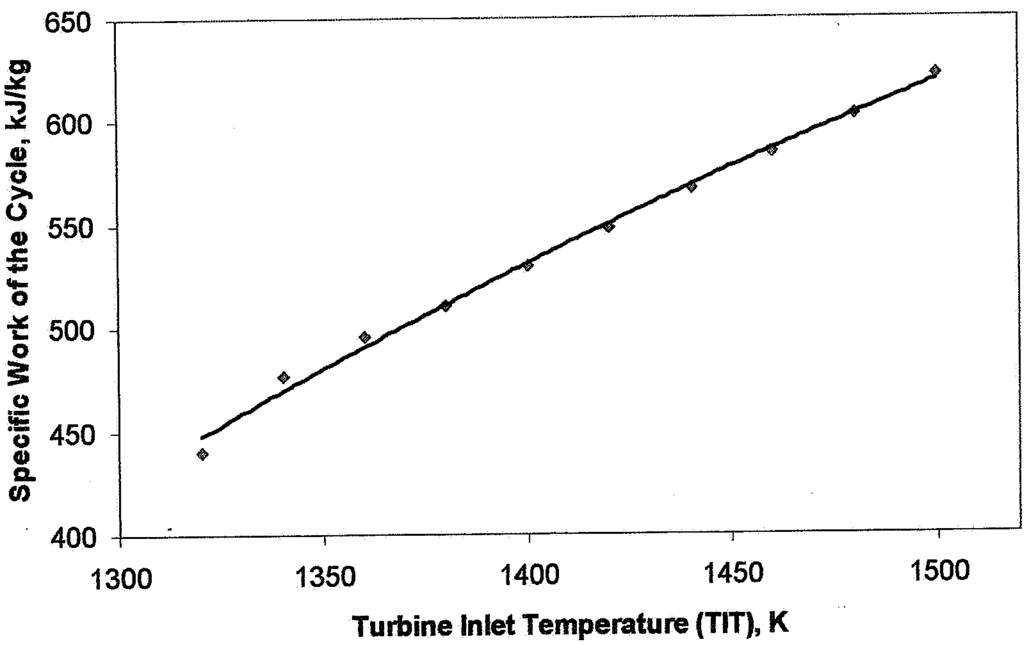


Figure 3.7 Specific work of EFCC plant versus turbine inlet temperature at P_b = 14 MPa and r_p = 13.7

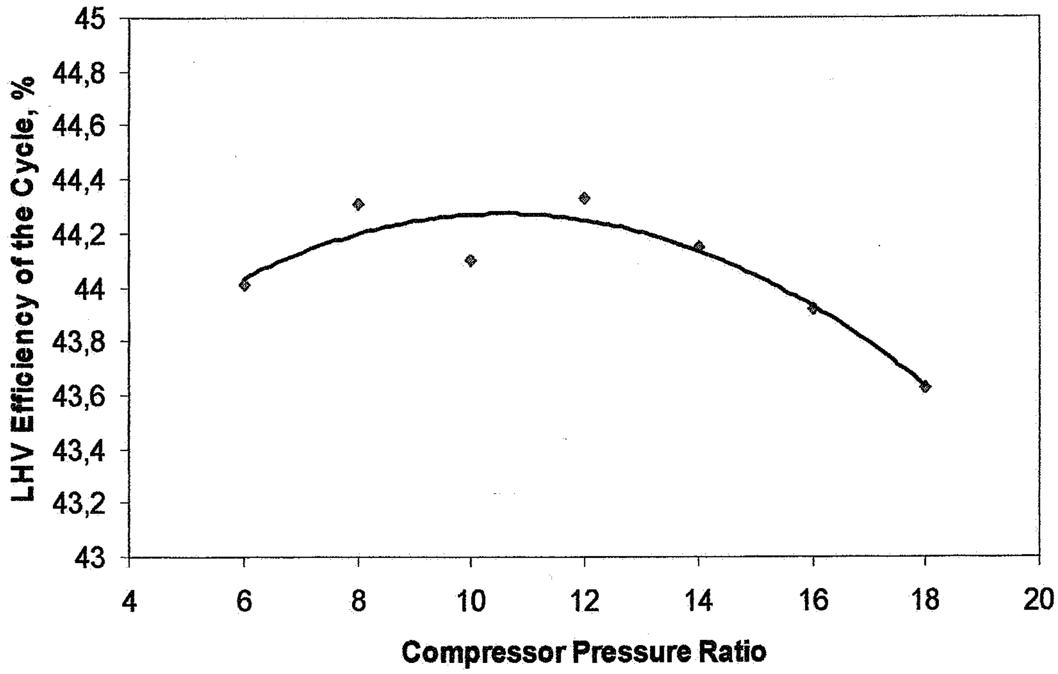


Figure 3.8 LHV efficiency of EFCC plant versus compressor pressure ratio at P_b = 14 MPa and TIT = 1438 K

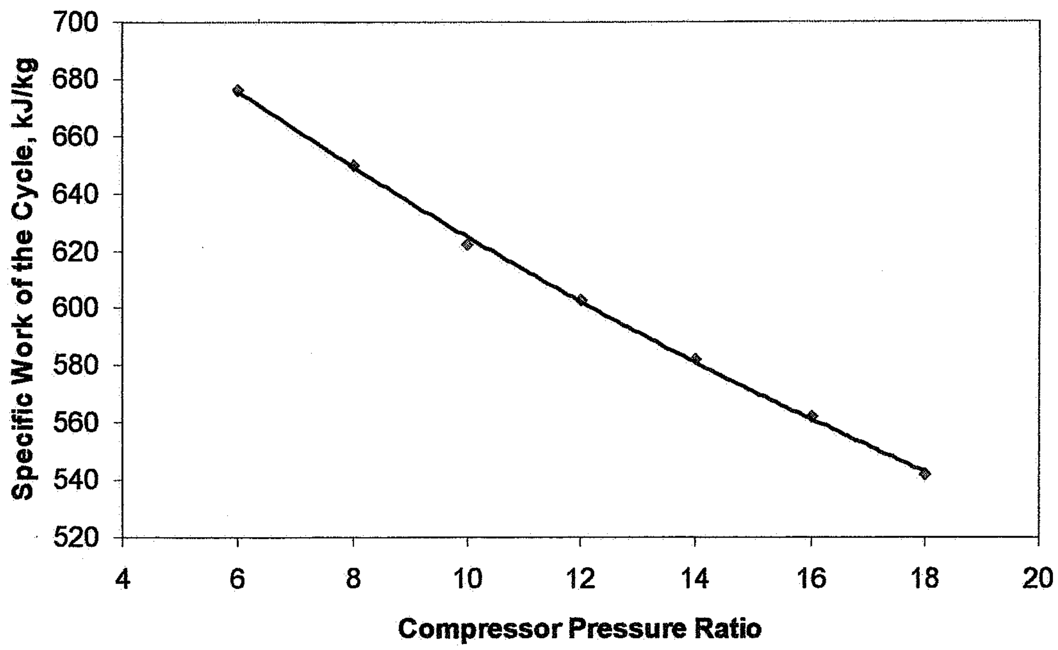


Figure 3.9 Specific work of EFCC plant versus compressor pressure ratio at P_b = 14 MPa and TIT = 1438K

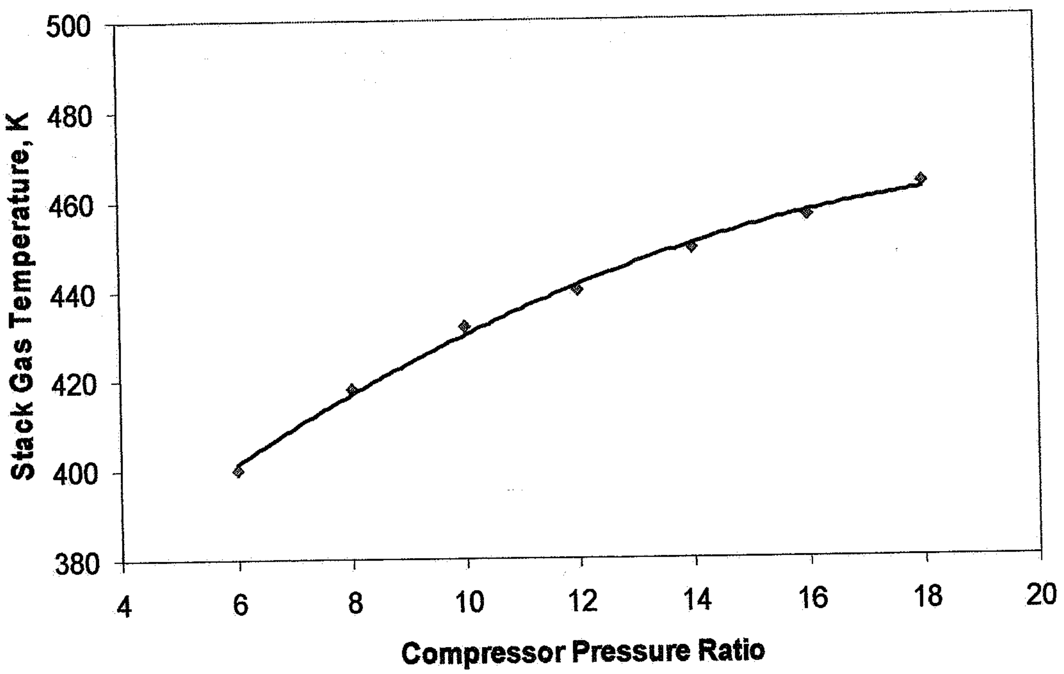


Figure 3.10 Stack gas temperature of EFCC plant versus turbine inlet temperature at P_b = 14 MPa and r_p = 13.7

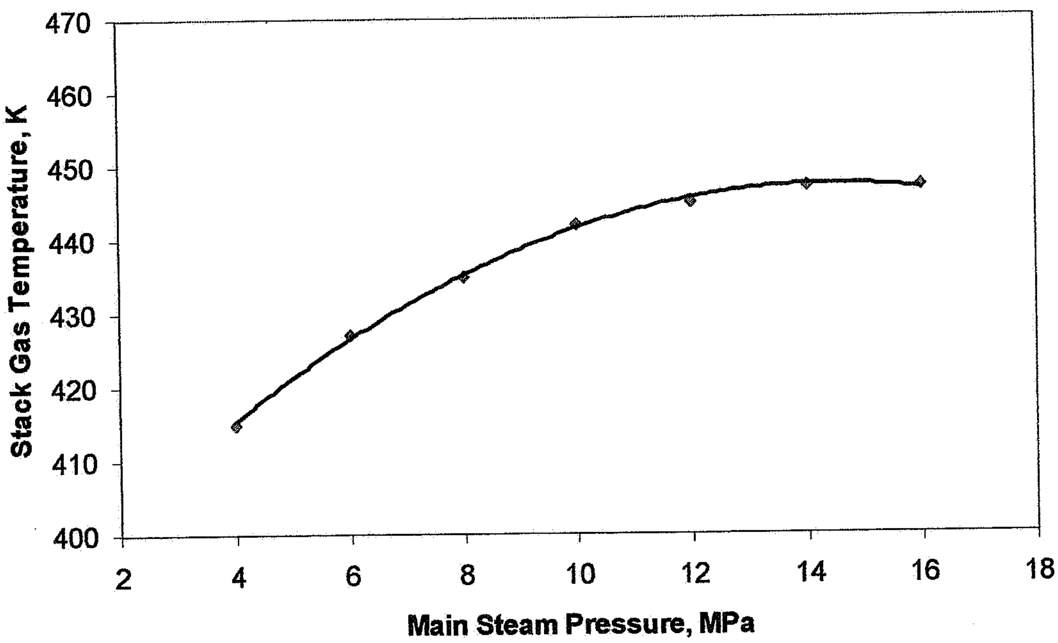


Figure 3.11 Stack gas temperature of EFCC plant versus compressor pressure ratio at Pb = 14 MPa and TIT = 1438K

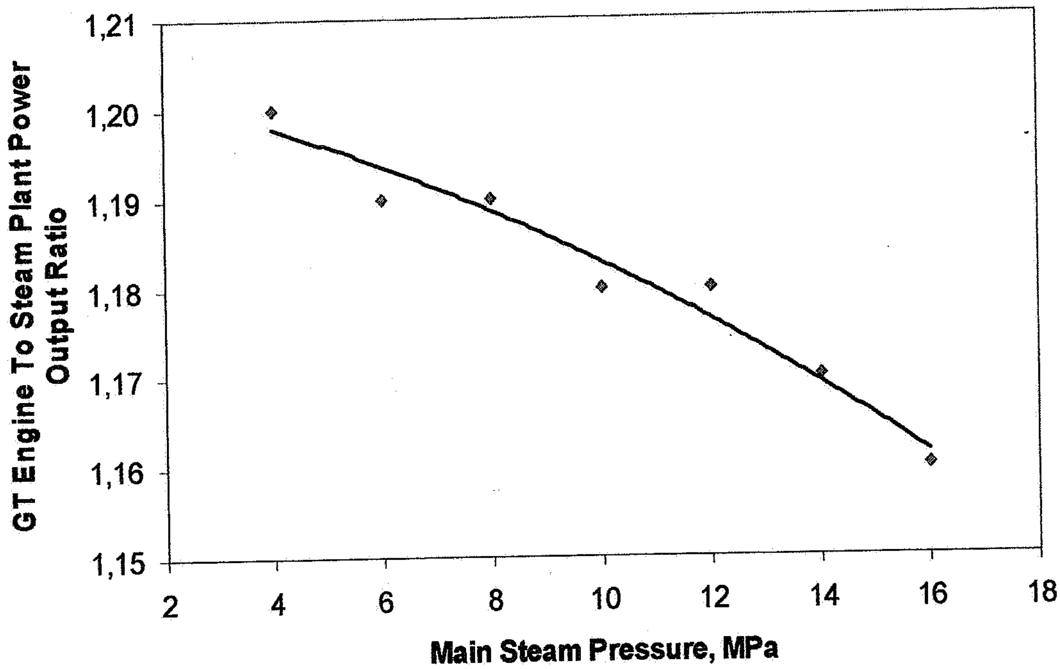


Figure 3.12 Specific work ratio of gas turbine to steam turbine versus main steam pressure at *TIT* = 1438 K and r_p = 13.7

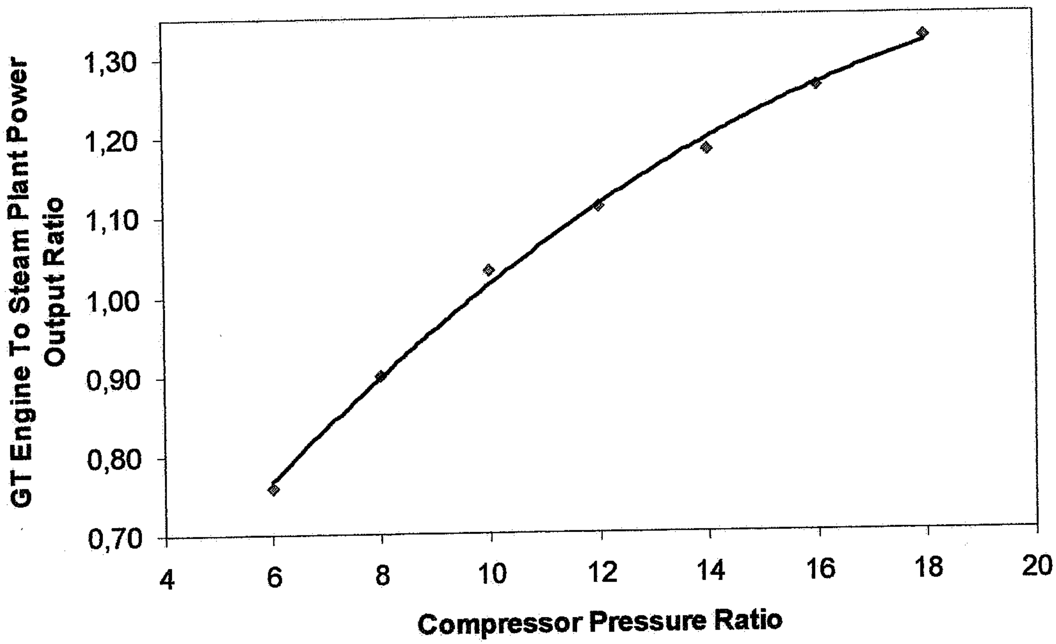


Figure 3.13 Specific work ratio of gas turbine to steam turbine versus compressor pressure ratio at *TIT* = 1438 K and *Pb* = 14 MPa

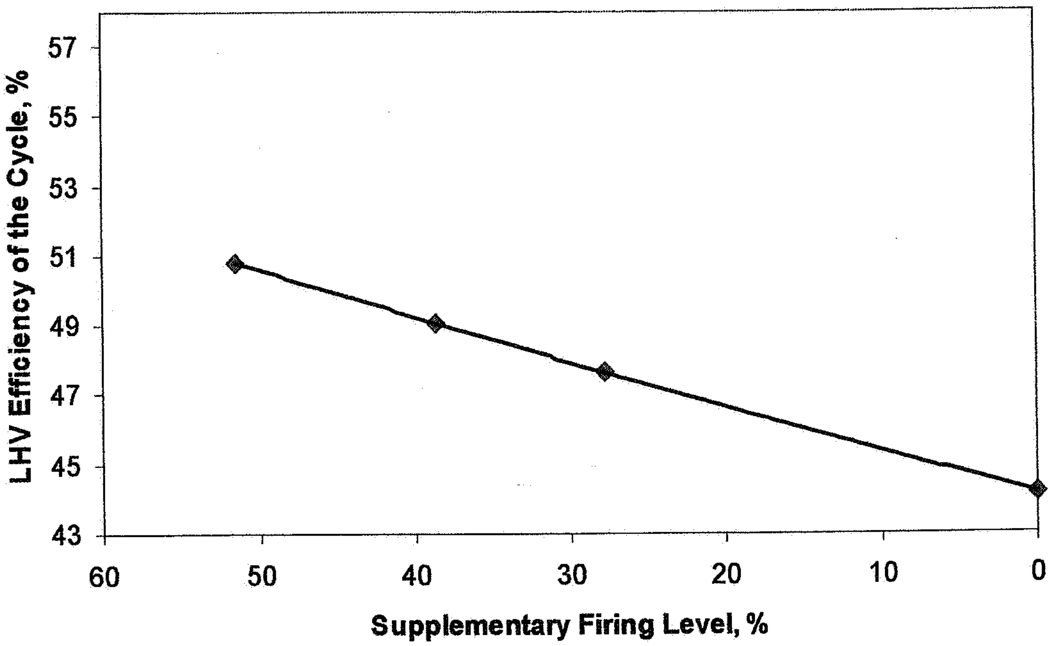


Figure 3.14 LHV Efficiency of EFCC Plant for different levels of supplementary firing.

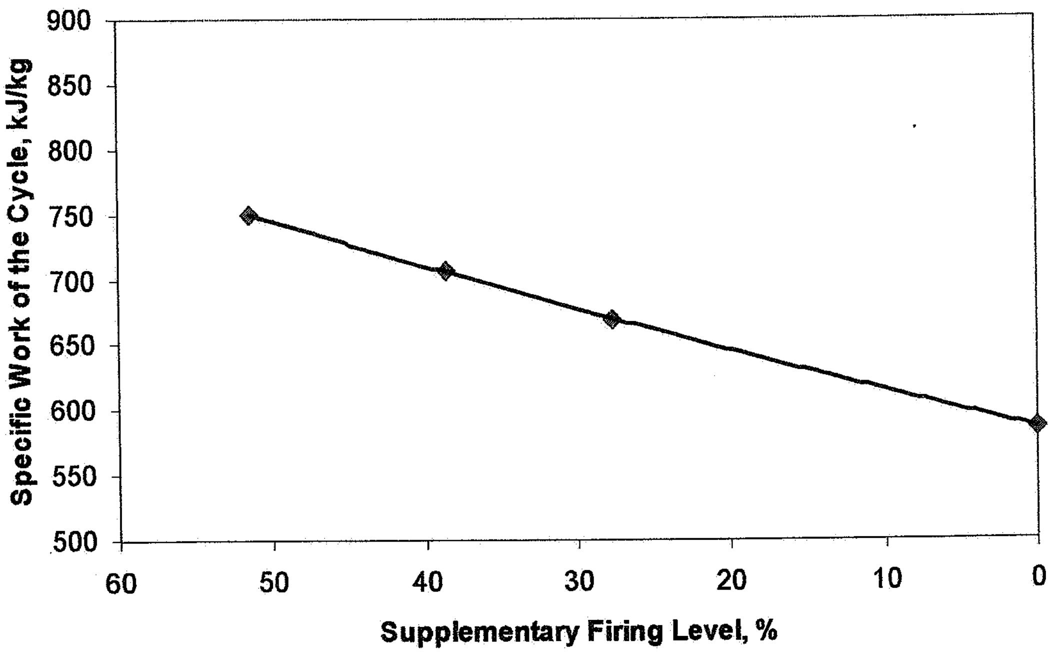


Figure 3.15 Specific work of EFCC Plant for different levels of supplementary firing.

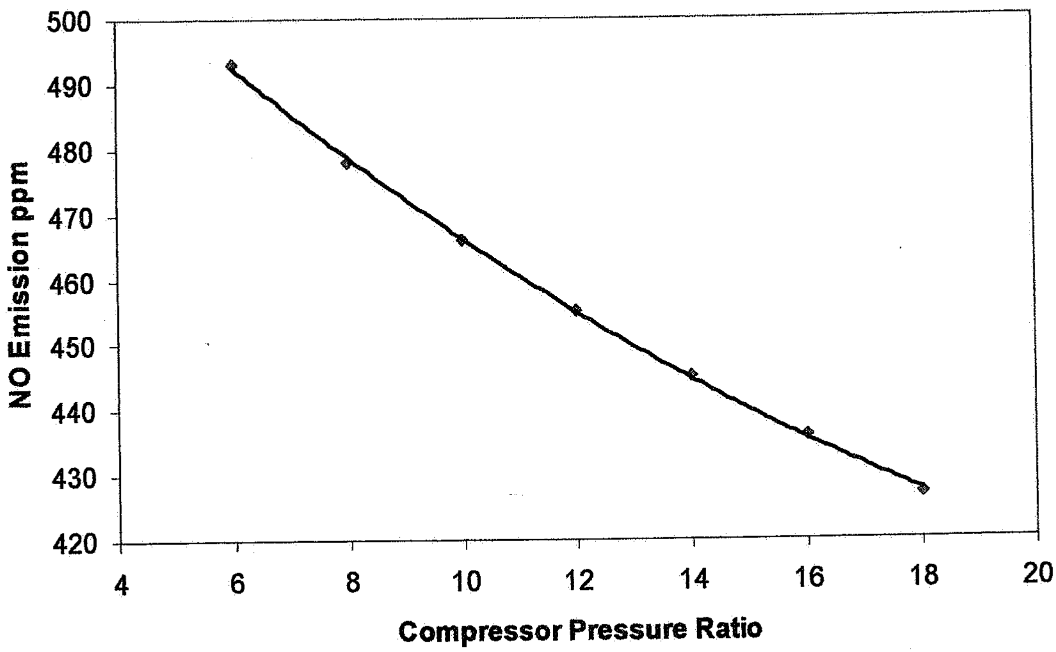


Figure 3.16 NO emission versus compressor pressure ratio at P_b = 14 MPa and *TIT* = 1438K

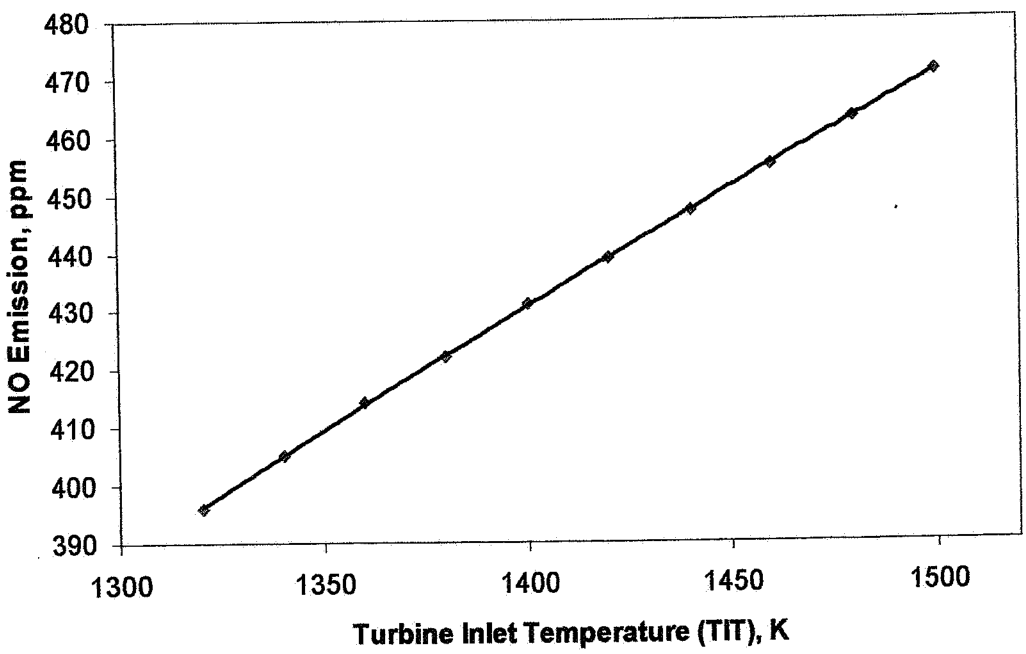


Figure 3.17 NO emission versus turbine inlet temperature at P_b = 14 MPa and r_p = 13.7

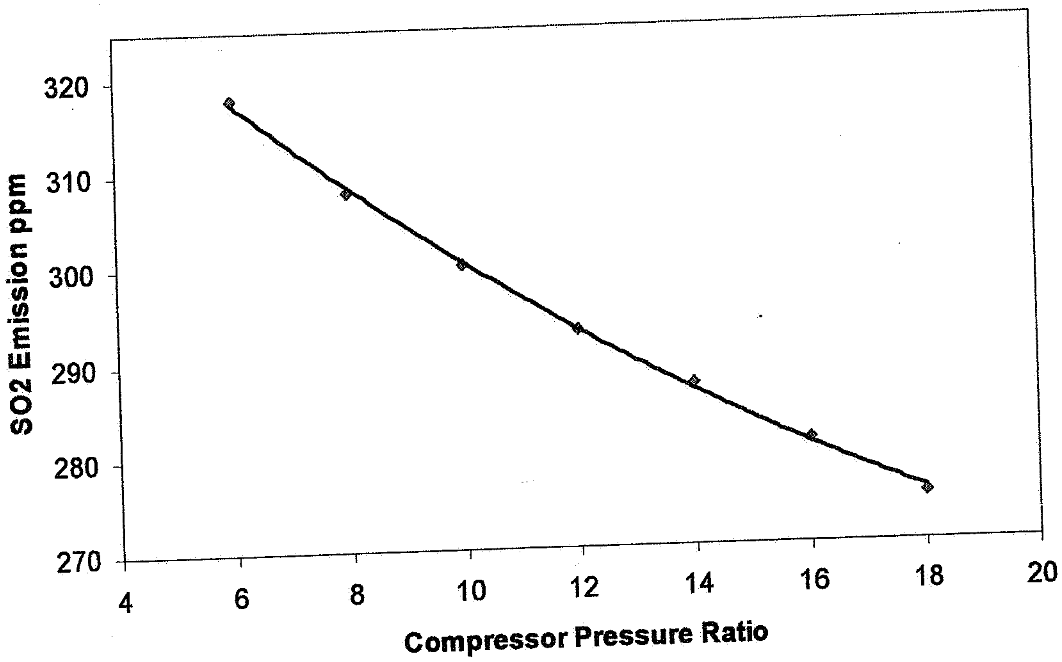


Figure 3.18 SO_2 emission versus compressor pressure ratio at P_b = 14 MPa and *TIT* = 1438 K

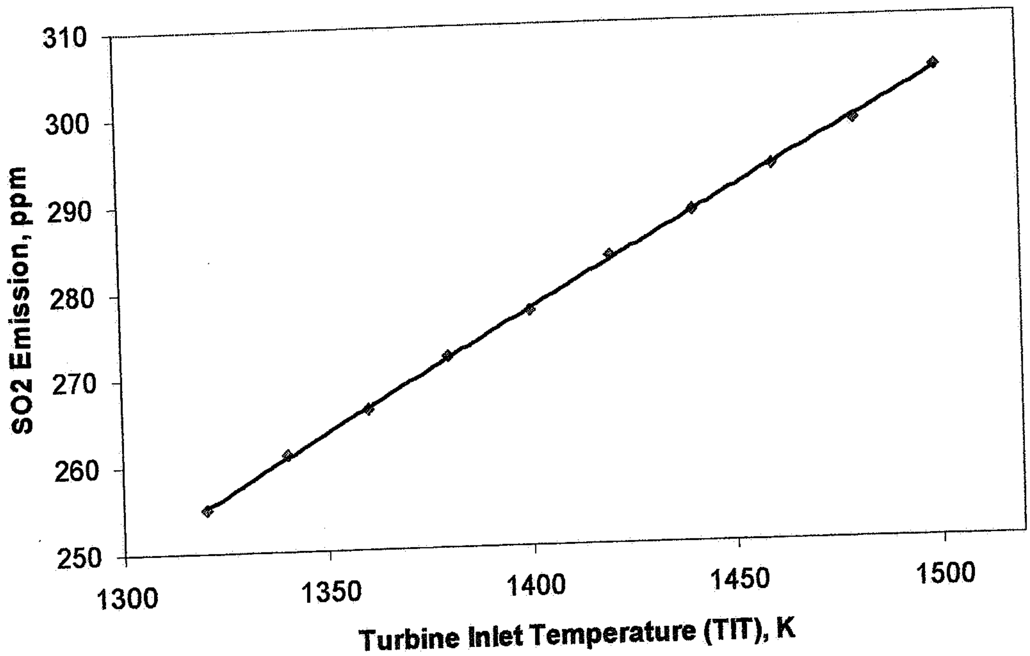


Figure 3.19 SO_2 emission versus turbine inlet temperature at P_b = 14 MPa and r_p = 13.7

CHAPTER 4

EXERGY ANALYSIS OF EFCC POWER PLANT

4.1. Introduction

Exergy analysis has become an increasingly important tool for the design and analysis of thermal systems, with regard to the importance of developing thermal systems which utilize energy resources in an effective manner. The method of exergy analysis is extending the concept of performance assessment based on thermodynamic analysis for a power plant, since it enables the location, cause and true magnitude of waste and loss to be determined. It also provides the basis for thermoeconomic analysis.

Recent developments in exergy analysis allow definition of new performance criteria, which offers more realistic treatment than the traditional ones. Rational efficiency is a performance criterion that can be formulated for the whole plant or a plant component. Rational efficiency η_R is given by the ratio of exergy transfer rate associated with the component output to the exergy transferred rate associated with the corresponding exergy input [12].

$$\eta_R = \frac{\sum \Delta E_{out}}{\sum \Delta E_{in}} \tag{4.1}$$

where $\Sigma\Delta E_{out}$ and $\Sigma\Delta E_{in}$ are sums of all exergy transfers making up the output and input, respectively.

In this chapter, exergy analysis for the EFCC plant is performed. For this purpose, the same approach in the preceding chapter is considered. Governing equations regarding 2nd law analysis are derived and solved using the computer model developed. Appendix C outlines the calculation steps for second law analysis performed in this chapter. Rational efficiency values are obtained for changes in selected parameters. Exergy destruction throughout the EFCC plant is investigated. Performance of each component is determined based on exergy analysis. Different levels of supplementary firing are included in the analyses.

4.2. Governing Equations

The general form of exergy balance can be given by

$$\dot{E}_i + \dot{E}^Q = \dot{E}_e + \dot{E}^W + \dot{I} \tag{4.2}$$

where $\dot{E}_i$ is the rate of exergy inflow, $\dot{E}_e$ is the rate of exergy outflow, $\dot{E}^Q$ is the exergy flow associated with heat transfer, $\dot{E}^W$ is the exergy flow associated with work transfer, and $\dot{I}$ is rate of irreversibility.

Any flow exergy is composed of physical and chemical parts.

$$\dot{E} = \dot{E}_{ch} + \dot{E}_{ph} \tag{4.3}$$

A general statement for physical component of exergy is given by

$$\tilde{\varepsilon}_{ph} = (\tilde{h} - \tilde{h}_0) - T_0(\tilde{s} - \tilde{s}_0) \tag{4.4}$$

For an ideal gas, the physical exergy can conveniently be expressed in terms of the thermal component $\tilde{\varepsilon}^{\Delta T}$ and the pressure component $\tilde{\varepsilon}^{\Delta P}$ as

$$\tilde{\varepsilon}_{ph} = \tilde{c}_p{}^{\varepsilon}(T - T_0) + \tilde{R}T_0 \ln\frac{P}{P_0} \tag{4.5}$$

where $\tilde{c}_p{}^{\varepsilon}$ is defined as mean molar isobaric exergy capacity and is expressed by

$$\tilde{c}_p{}^{\varepsilon} = \frac{1}{T - T_0}\left[\int_{T_0}^{T} \tilde{c}_p dT - T_0 \int_{T_0}^{T} \frac{\tilde{c}_p dT}{T}\right] \tag{4.6}$$

The tabulated $\tilde{c}_p{}^{\varepsilon}$ values for different temperature values are used in the exergy related calculations in this study [12].

For a gas mixture, exergy is represented by the following general equation

$$\dot{E} = \dot{n}\left[\sum_i x_i \tilde{\varepsilon}_{oi} + \tilde{R}T_0 \sum_i x_i \ln x_i + (T - T_0)\sum_i x_i \tilde{c}_{p,i}{}^{\varepsilon} + \tilde{R}T_0 \ln(P/P_0)\right] \tag{4.7}$$

where x_i is the molar fraction of each constituent of the mixture, $\tilde{\varepsilon}_{oi}$ is the corresponding molar exergy, for which the values are tabulated in the literature [12].

Estimation of fuel exergy values is a crucial step in any exergy related analysis, since the fuel is the only source of exergy input to the plant. Provided the necessary data are available, the chemical exergy of a combustible substance can be obtained by the following relationship

$$\tilde{\varepsilon}^0 = -\Delta\tilde{h}^0 + T^0 \Delta\tilde{s}^0 + \tilde{R}T^0\left(x_{O_2} \ln\frac{P_{O_2}{}^{00}}{P^0} - \sum_k x_k \ln\frac{P_k{}^{00}}{P^0}\right) \tag{4.8}$$

where the subscript k refers to the components of the combustion products. This expression can directly be applied to the calculation of chemical exergy of gaseous fuels for which the chemical composition can be determined and thermochemical data for the components can be obtained. However, solid and liquid fuels are constituted by numerous chemical compounds of unknown nature. It is rather difficult to determine the entropy of reaction, $\Delta\tilde{s}^0$, for those fuels with a reasonable degree of accuracy. Several approaches are found in the literature to address this problem. Based on the assumption suggested by Szargut and Styrylska that the ratio of chemical exergy ε^o to the net calorific value $(NCV)^0$ for solid and liquid industrial fuels is the same as for pure chemical substances having the same ratios of constituent chemicals, and using the following expression for this ratio

$$\varphi = \frac{\varepsilon^0}{(NCV)^0} \tag{4.9}$$

φ values can be obtained for numerous pure organic substances, containing C, H, O, N and S. These values can be used to express the dependence of φ on the atomic ratios H/C, O/C, N/C, and S/C [12]. The applicability of such expressions can be extended to cover industrial fuels.

For dry organic substances contained in solid fossil fuels consisting of C, H, O and N with the mass ratio o/c being less than 0.667, the following expression can be used in terms of mass ratios

$$\varphi_{dry} = 1.0437 + 0.1882\frac{h}{c} + 0.0610\frac{o}{c} + 0.0404\frac{n}{c} \tag{4.10}$$

This expression is valid, and is utilized in this study for chemical exergy calculations of anthracite coal, for which the mass ratio o/c is around 0.015. In order to take into account the effect of moisture and the sulphur content in the fuel, the following modification is applied throughout the calculations

$$\varepsilon^0 = \left[\left((NCV)^0 + \omega h_{fg}\right)\right]\varphi_{dry} + \left[\left(\varepsilon_s^0 - (NCV)_s^0\right)\right]s \tag{4.11}$$

4.3. Results of Performance Evaluation

In this section, results of the 2nd law analysis for the EFCC plant are presented. Exergy analysis is performed based on the integration of 2nd law related equations into the model developed in the preceding section. Exergy balance is employed for each component of the EFCC plant, and the corresponding rates of irreversibilities are determined based on the governing equations briefly outlined in the preceding section. Dependence of rational efficiency on selected plant parameters is investigated. Exergy lost in plant components for which the rate of irreversibility poses significance is analyzed.

Figure 4.1 illustrates the change in rational efficiency for the plant with main steam pressure. Increasing the main steam pressure increases the plant performance in terms of rational efficiency. The plot of rational efficiency as a function of compressor pressure ratio is given in Figure 4.2. As it can be seen from the figure, with *TIT* and P_b are both kept constant, there exists an optimum pressure ratio for which the rational efficiency achieves a maximum. Effect of turbine inlet temperature on rational efficiency is investigated, while P_b and r_p are held constant. Plant rational efficiency increases linearly with increasing *TIT* as shown in Figure 4.3.

Effect of supplementary firing on rational efficiency for the whole cycle is illustrated in Figure 4.4. Plant performance decreases, as the level of supplementary firing is being decreased. Coal only option results in around 25 % less rational efficiency than that for the 52 % supplementary firing case, as can be seen from the figure. This observation is in accordance with the results of the analysis performed in Chapter 3, with both investigations pointing out the fact that supplementary firing improves the plant performance. Nonetheless, a detailed performance assessment for coal only option can only be achieved upon the finalization of thermoeconomic analysis.

Exergy lost throughout the cycle working in coal only conditions is determined per unit fuel input. Figure 4.5 illustrates the contribution of each component in exergy destruction, together with the useful work output per unit fuel exergy input. As it can be seen from the figure, air heater is the most critical component, where more than half of the fuel exergy is lost. Approximately 15 % of fuel exergy is destroyed in the other plant components, while 33 % is converted to useful work.

Breakdown of exergy lost within the plant with different supplementary firing levels are illustrated in Figure 4.6 and Figure 4.7. For the case of 27 % supplementary firing level, which corresponds to a turbine inlet temperature of 1253 K, 38 % of fuel exergy can be converted to useful work, while the rest is lost throughout the cycle, with air heater being the major contributor to the lost exergy. It is also shown in Figure 4.6 that amount of exergy lost within the supplementary firing unit corresponds to 7% of total fuel exergy input, while exergy lost within the other components is around 15%.

Figure 4.7 illustrates the distribution of the fuel exergy input throughout the plant working in 52 % supplementary firing conditions. As compared with the 27 % supplementary firing level, exergy lost within the supplementary firing unit is two times higher, while the plant rational efficiency is further improved to reach around 43%. Air heater is again the most significant component in terms of irreversibilities, where approximately 29% of fuel exergy input is diminished. Both in coal only option and in different supplementary firing level conditions, mixer and heat recovery steam generator are the leading ones among the other plant components regarding the exergy losses, while around 3% of fuel exergy is lost in each of these components.

Exergy lost in the compressor per unit fuel exergy input as a function of compressor pressure ratio is shown in Figure 4.8,. It can be observed that, the amount of exergy losses in compressor increases as the pressure ratio is increased. It can also be seen that exergy lost within the compressor is around 1% of fuel exergy input to the plant.

Air heater is the most significant component of the EFCC plant analyzed in this study, in terms of irreversibilities. Figure 4.9. figures out the variation in exergy lost in the air heater per unit fuel exergy input, with compressor pressure ratio. It is observed that exergy lost decreases as the pressure ratio is increased. Nevertheless, more than 50% of the fuel exergy input is lost in the air heater, regardless of the selected pressure ratio.

Figure 4.10 illustrates the variation in the lost exergy within the gas turbine with pressure ratio. Increasing the pressure ratio results in increase in the exergy lost for gas turbine. Increasing the pressure ratio twice increases the amount of exergy lost in the gas turbine by around 60 %.

Effect of pressure ratio on the lost exergy within the heat recovery steam generator is investigated and shown in Figure 4.11. Around 3% of fuel exergy input is diminished in the heat recovery steam generator. It is observed that amount of exergy lost is smaller within the low pressure ratio range and is getting larger as the pressure ratio is increased.

Variation in the amount of exergy lost within heat recovery steam generator with main steam pressure is analyzed. As can be seen from Figure 4.12, more exergy is lost within the heat recovery steam generator within the low steam pressure range, and increasing the main steam pressure has a decreasing effect on the exergy lost term.

Figure 4.13 figures out the change in the exergy lost per unit fuel exergy input within the steam turbine as a function of main steam pressure. Increasing the main steam pressure increases the amount of exergy lost. It can also be seen that around 2% of the fuel exergy input is lost within the steam turbine.

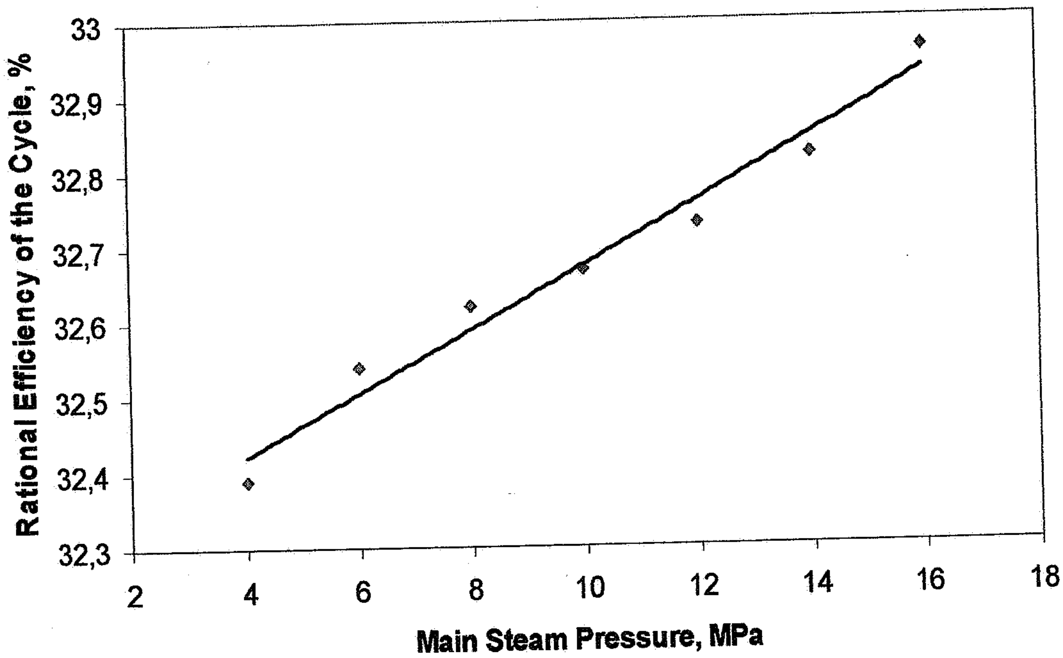


Figure 4.1 Rational efficiency of EFCC plant versus main steam pressure at $TIT =$ 1438K and $r_p = 13.7$

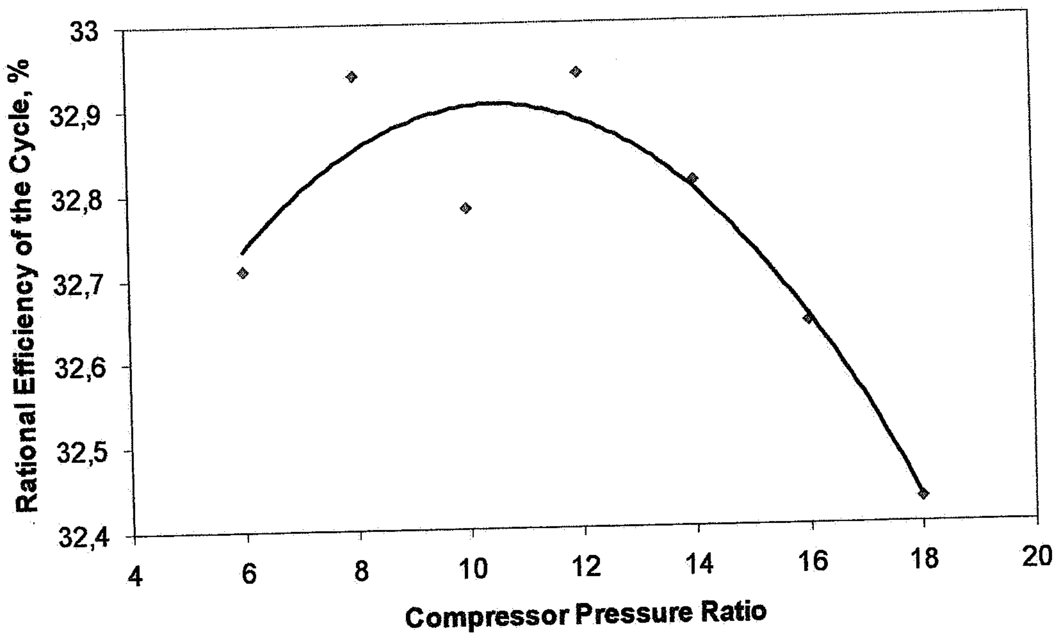


Figure 4.2 Rational efficiency of EFCC plant versus compressor pressure ratio at P_b = 14 MPa and $TIT =$ 1438 K

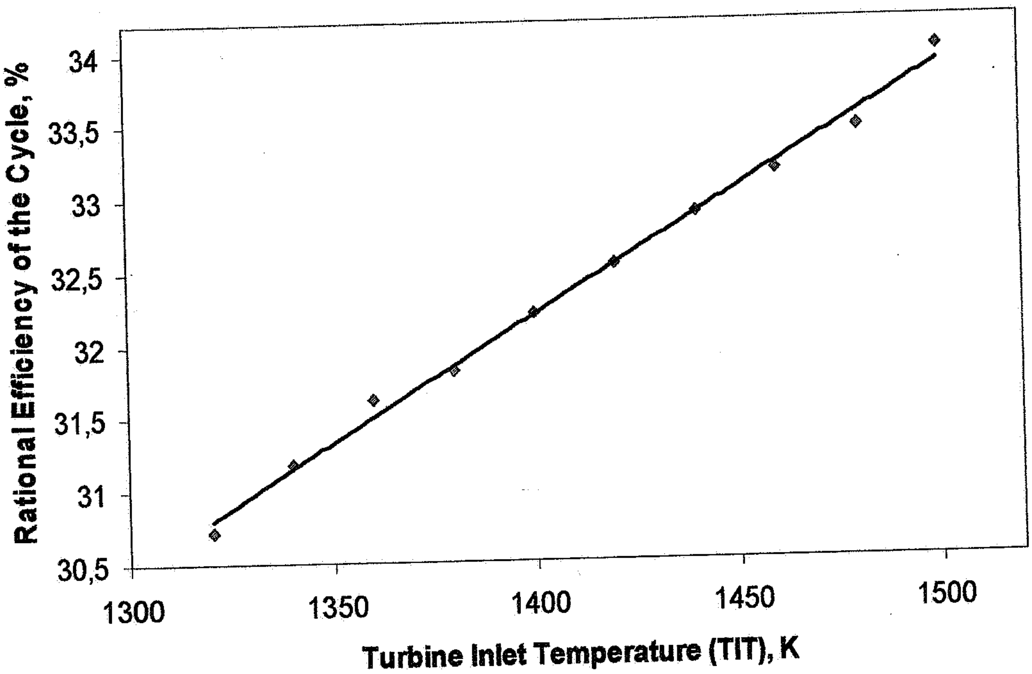


Figure 4.3 Rational efficiency of EFCC plant versus turbine inlet temperature at P_b = 14 MPa and r_p = 13.7

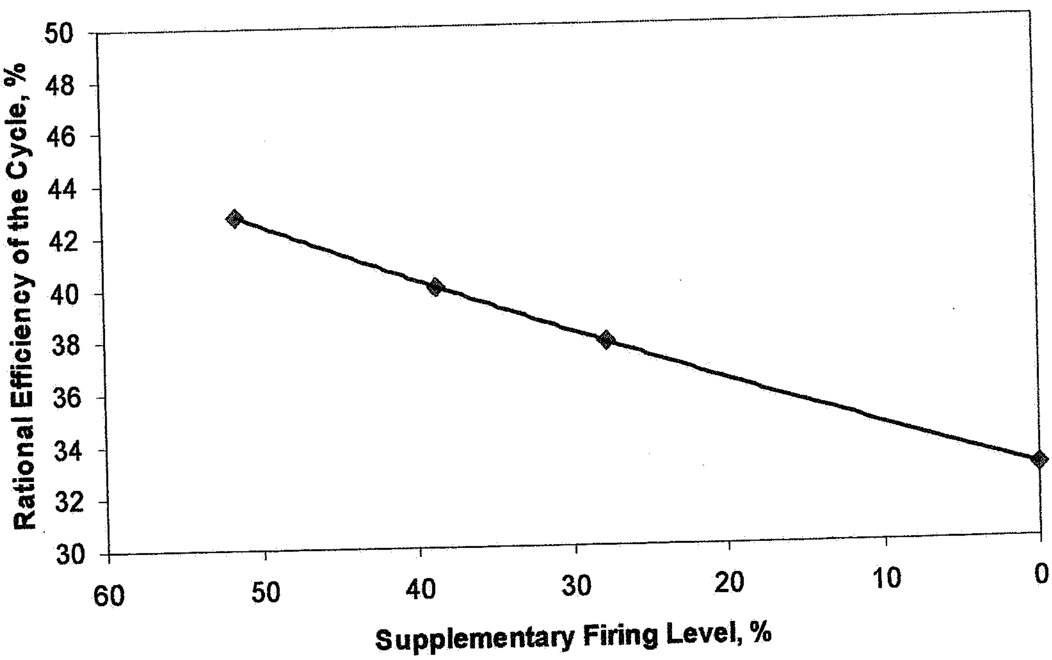


Figure 4.4 Rational efficiency of EFCC plant for different levels of supplementary firing.

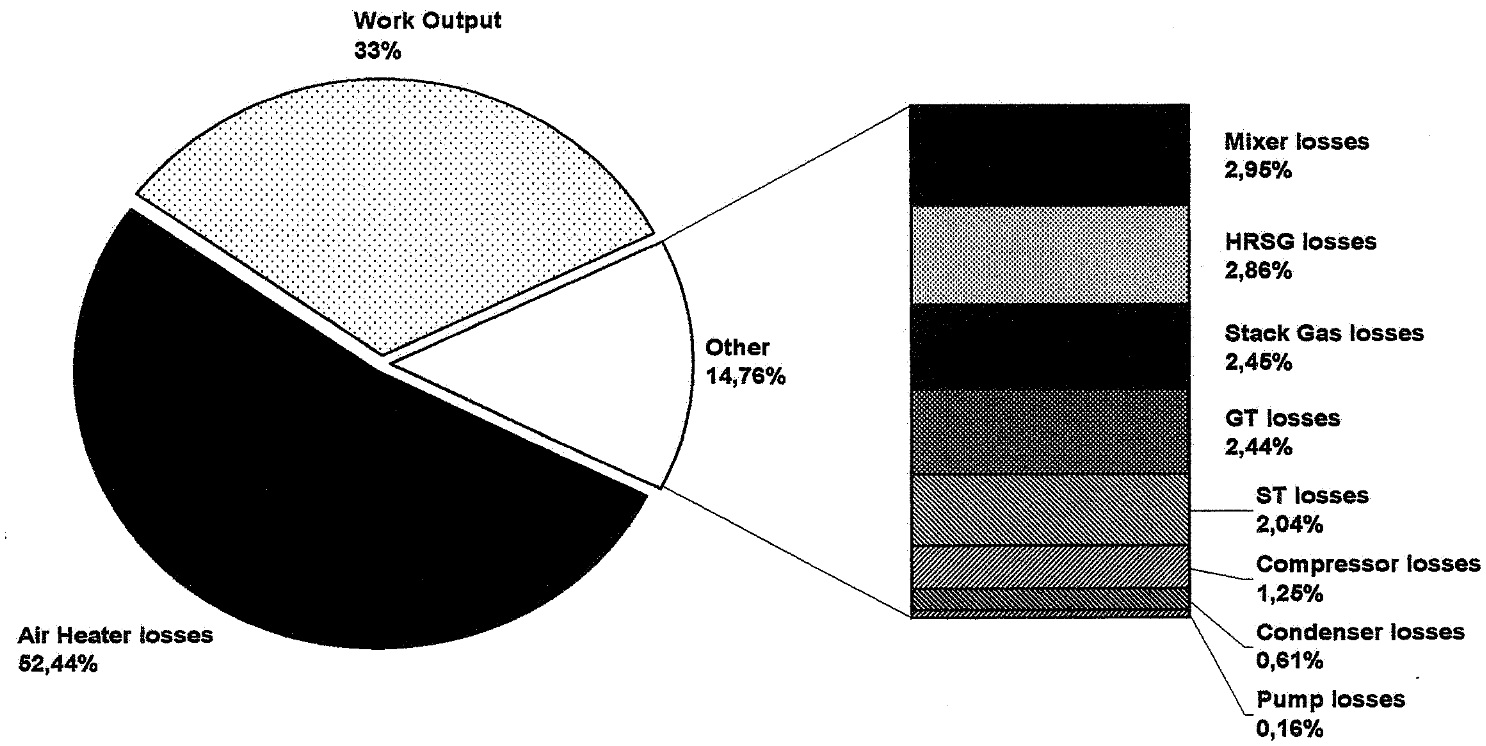


Figure 4.5 Exergy distribution for coal only option

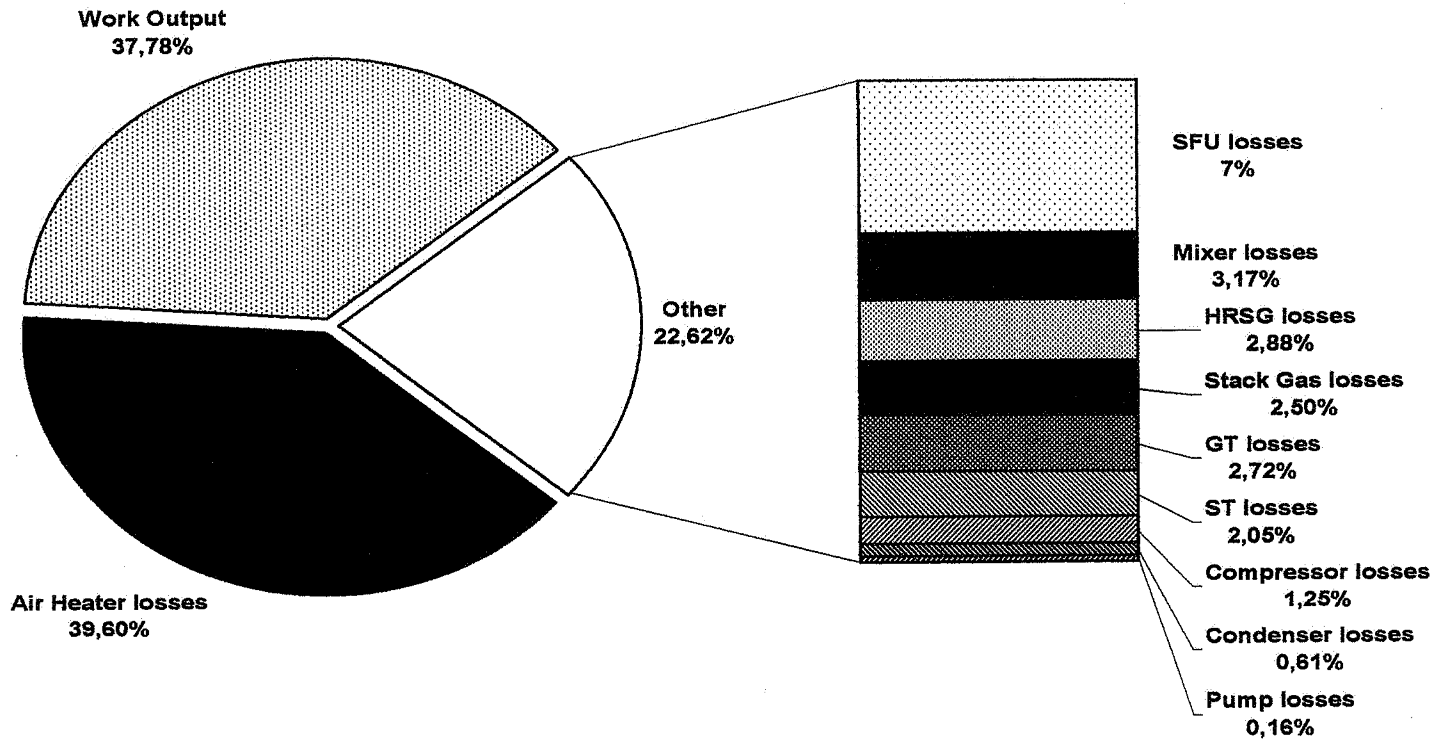


Figure 4.6 Exergy distribution for 27% supplementary firing level conditions

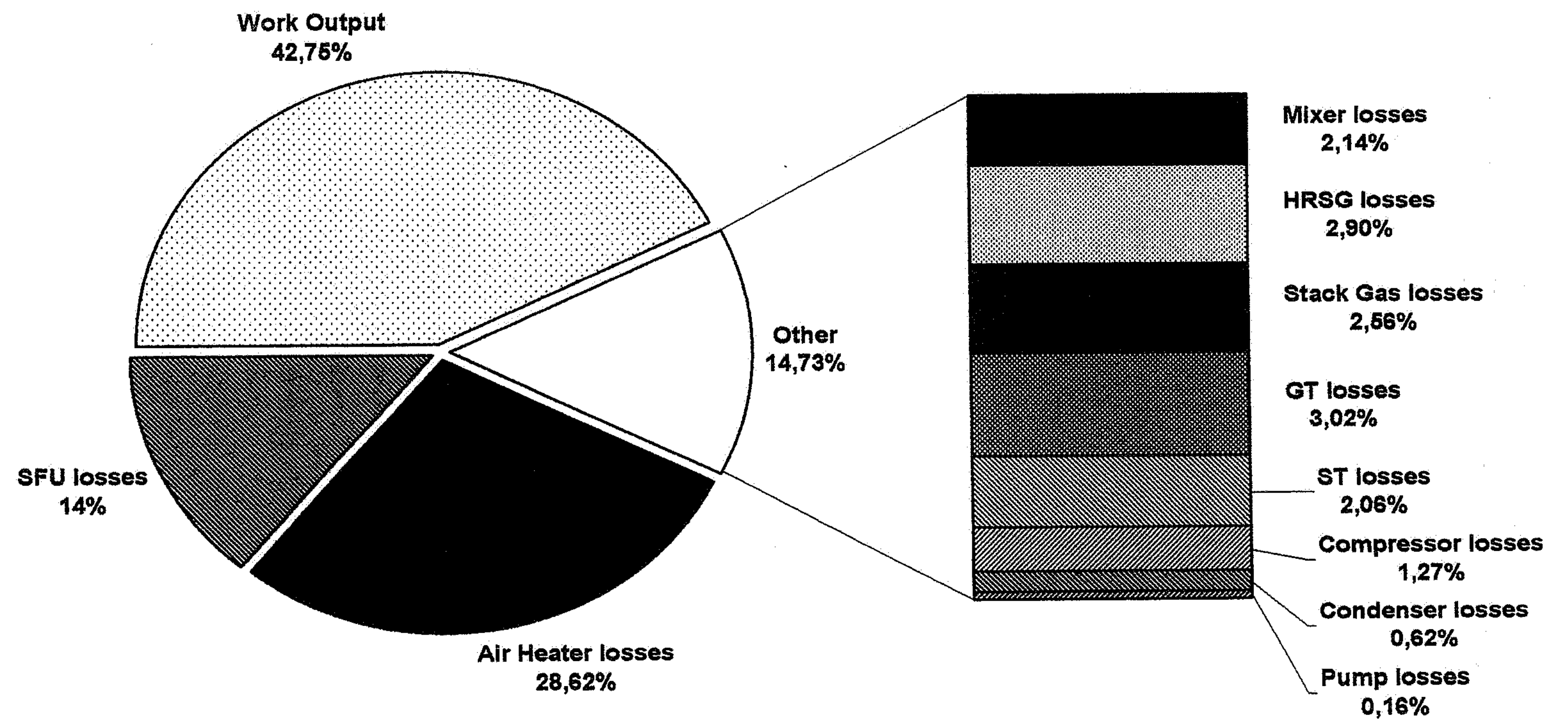


Figure 4.7 Exergy distribution for 52% supplementary firing level conditions

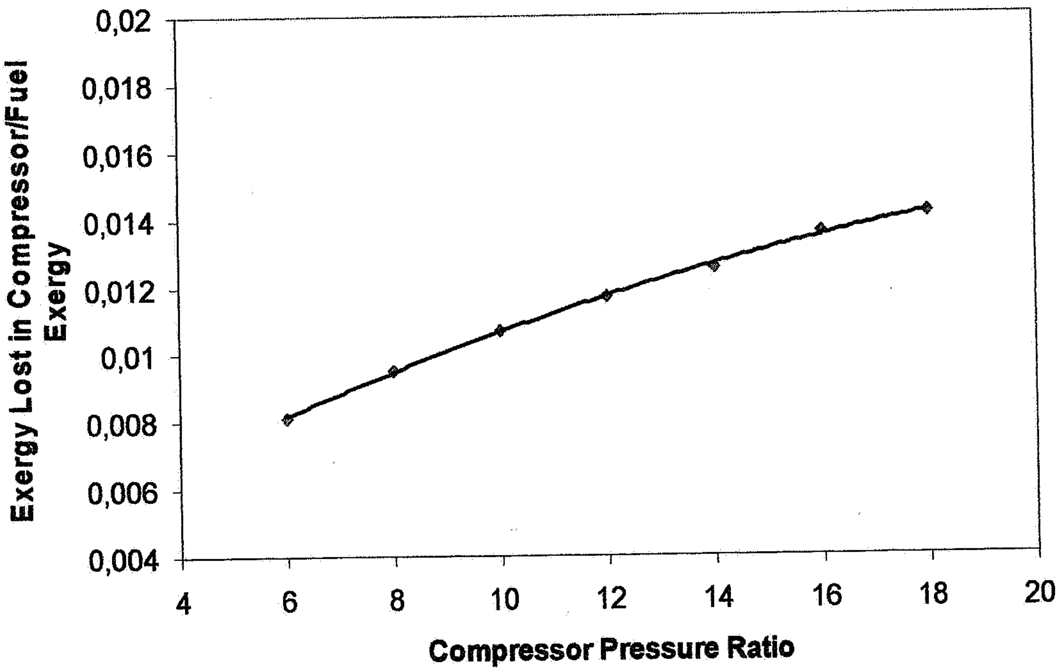


Figure 4.8 Exergy lost in the compressor per fuel exergy input versus compressor pressure ratio at P_b = 14 MPa and *TIT* = 1438 K

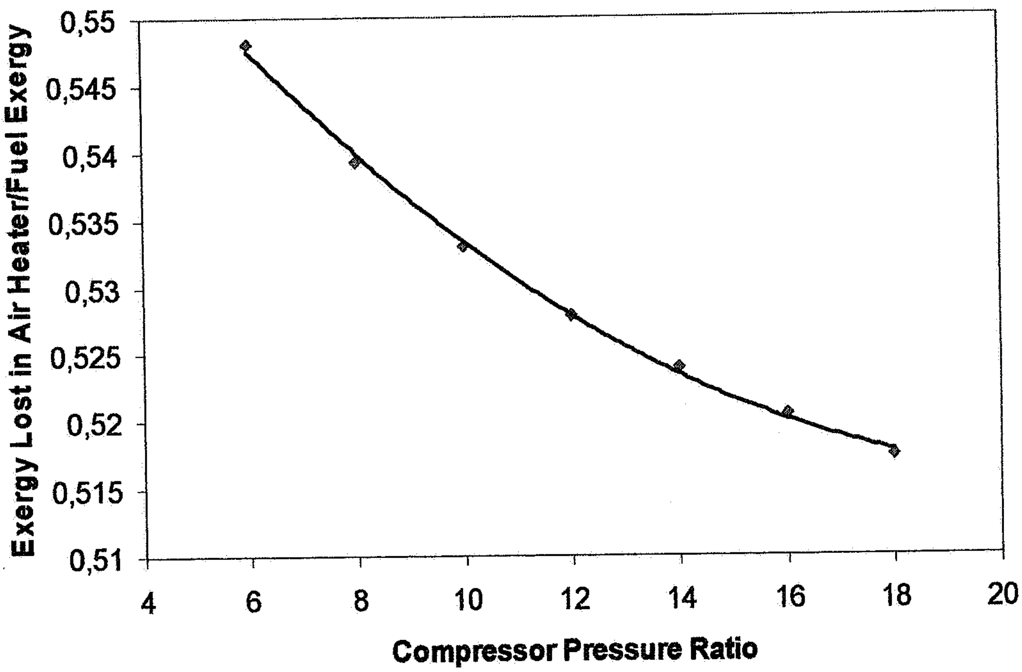


Figure 4.9 Exergy lost in the air heater per fuel exergy input versus compressor pressure ratio at P_b = 14 MPa and *TIT* = 1438 K

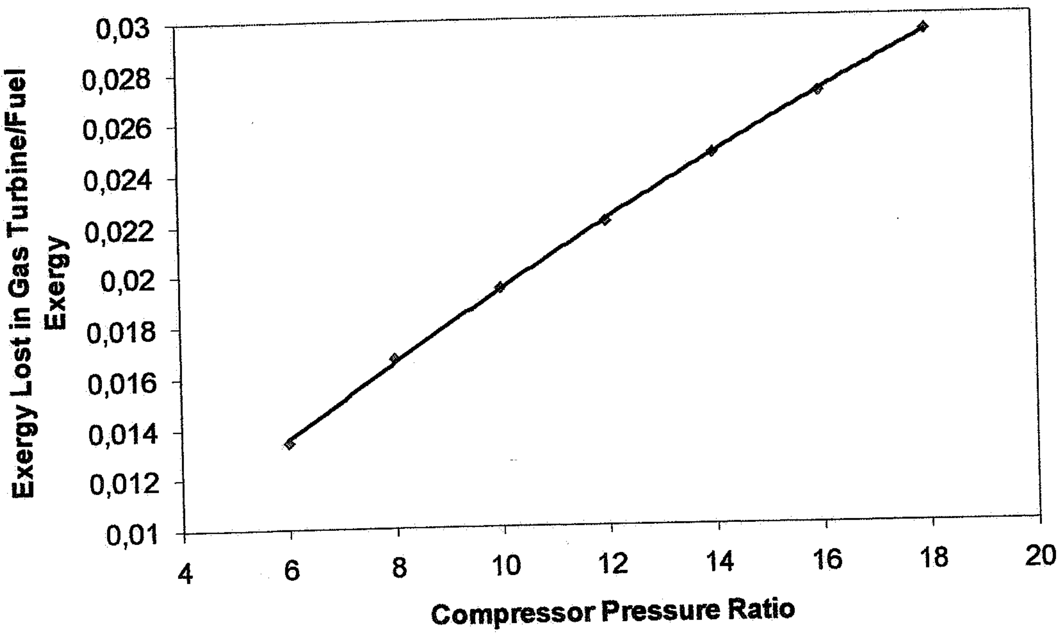


Figure 4.10 Exergy lost in the gas turbine per fuel exergy input versus compressor pressure ratio at P_b = 14 MPa and TIT = 1438 K

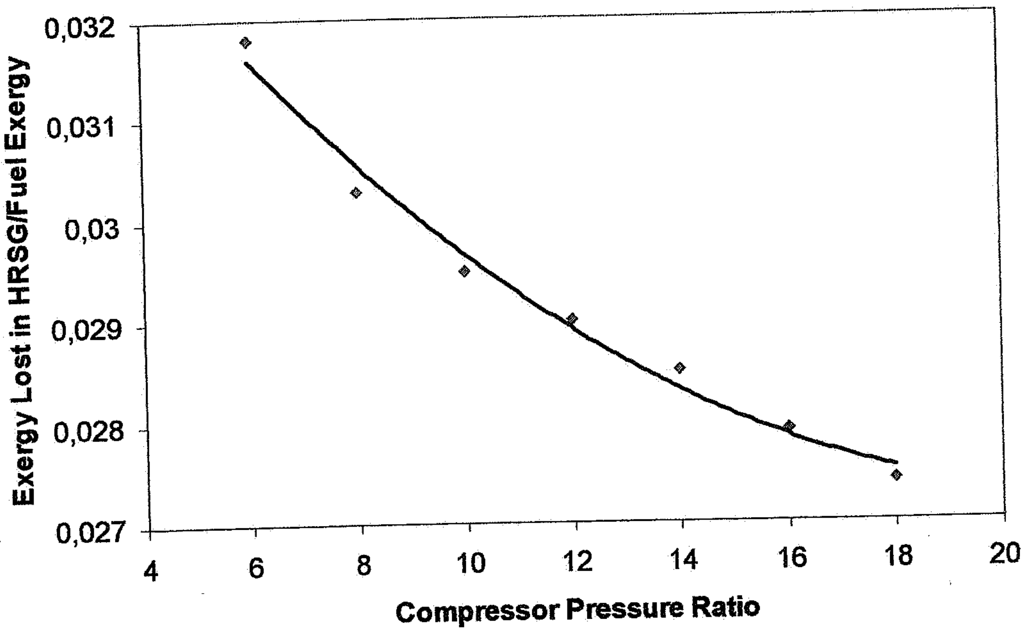


Figure 4.11 Exergy lost in the heat recovery steam generator per fuel exergy input versus compressor pressure ratio at P_b = 14 MPa and TIT = 1438 K

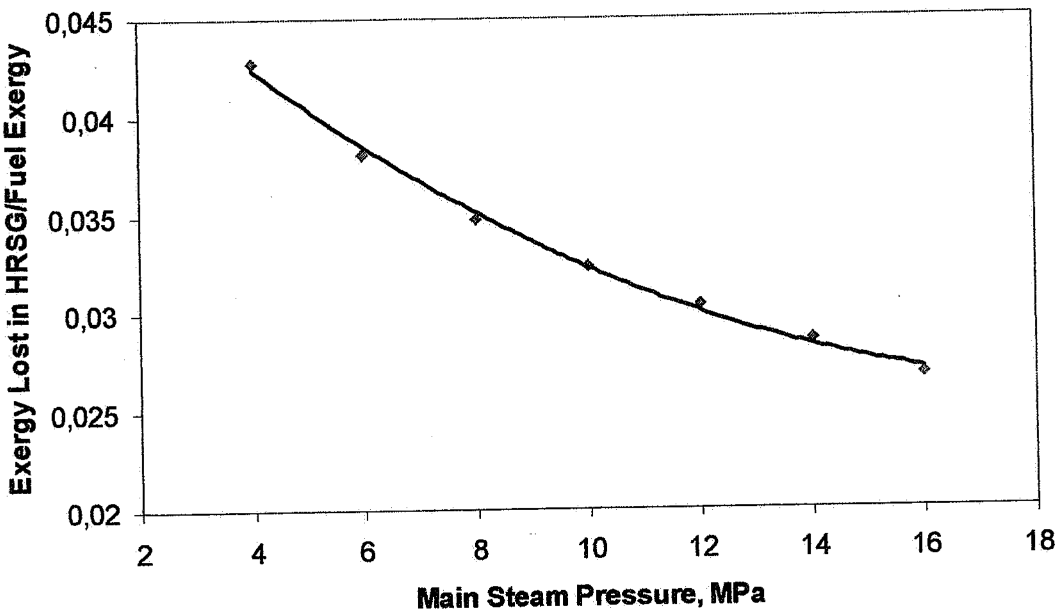


Figure 4.12 Exergy lost in the heat recovery steam generator per fuel exergy input versus main steam pressure at $TIT = 1438$K and $r_p = 13.7$

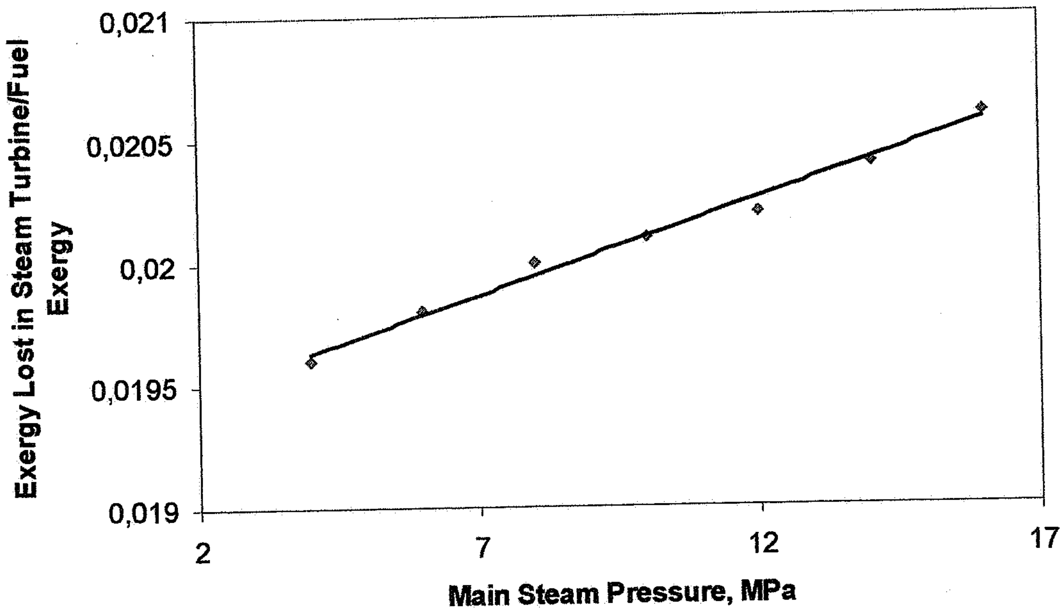


Figure 4.13 Exergy lost in the steam turbine per fuel exergy input versus main steam pressure at $TIT = 1438$K and $r_p = 13.7$

CHAPTER 5

THERMOECONOMIC ANALYSIS FOR EFCC POWER PLANT

5.1. Introduction

Detailed thermodynamic evaluations of EFCC plant is performed in previous chapters. Exergy destructions and exergy losses are determined as a measure of thermodynamic inefficiencies within the system. Nevertheless, knowledge on costs of these inefficiencies is a useful guide for improving the performance of the system and reducing the costs of final products as well.

In this chapter, thermoeconomic analysis are performed for the EFCC plant to reach a more realistic conclusion about the plant performance. In this respect, cost formation processes are analyzed and flow of costs throughout the EFCC plant is determined.

5.2. Fundamentals of Thermoeconomics

Thermoeconomics is the branch of thermal sciences that combines a thermodynamic analysis with economic principles. The main purpose of thermoeconomics is to mathematically combine in a single model the second law of thermodynamic analysis with the economic factors. It provides the understanding of an energy-conversion system with information which is not available through conventional thermodynamic analysis and economic evaluation. Such information is crucial to the design and operation of a cost-effective system.

Combination of second law of thermodynamics with economics using exergy analysis has become a very powerful tool for the systematic study and optimization of energy systems in recent years. Thermoeconomics rests on the notion that the exergy is the only rational basis for assigning costs to the interactions a thermal system experiences with its surroundings to the sources of inefficiencies within the system [15]. Since the thermodynamic considerations of thermoeconomics are based on the exergy concept, the terms exergoeconomics and thermoeconomics are used interchangeably.

There are different approaches developed for formulating efficiencies and auxiliary costing equations. Each of these approaches is characterized by some degree of subjectivity. In this study, SPECO (Specific Exergy Costing Method) is considered as a basis to implement thermoeconomic analysis.

The basic principle of SPECO method is the calculation of costs by systematically registering exergy and cost additions and removals from each material stream.

A cost balance applied to the kth component shows that the sum of cost rates of all exiting exergy streams equals the sum of all cost rates associated with all entering streams plus charges due to capital investment ($\dot{Z}_k^{CI}$) and operating and maintenance costs ($\dot{Z}_k^{OM}$). The sum of the last two terms is denoted by $\dot{Z}_k$

$$\dot{Z}_k = \dot{Z}_k^{CI} + \dot{Z}_k^{OM} \tag{5.1}$$

Accordingly, for a component operating at steady state with a number of entering and exiting material streams as well as both heat and work interactions with surroundings the relevant cost balance can be expressed as number of this equations being equal to the number of exiting streams minus one.

If there are N_e exergy streams exiting the component being considered, we have N_e unknowns and only one equation, the cost balance. Therefore, one needs to formulate $N_e - 1$ auxiliary equations. This task is accomplished with the aid of the F and P rules [15].

The F rule is based on the consideration that the removal of exergy from an exergy stream within the component being considered when for this stream the exergy difference between inlet and outlet is considered in the definition of the fuel. The F rule states that the total cost associated with this removel of exergy must be equal to the cost at which the removed exergy was supplied to the same stream in upstream components. One auxiliary equation per each removal of exergy can be obtained in this way so that the number of auxiliary equations provided by the F rule is always equal to the number $N_{e,F}$ of exiting exergy streams which are associated with the definition of the fuel for the component.

The P rule refers to the supply of exergy to an exergy stream within the component being considered. The P rule states that each exergy unit is supplied to any stream associated with the product at the same average cost. The cost is calculated using the cost balance and the equations obtained by applying the F rule. Since each stream to which exergy is supplied corresponds to an exiting stream, the number of auxiliary equations provided by the P rule is always equal to $N_{e,P} - 1$ where $N_{e,P}$ is the number of exiting exergy streams which are included in the product definition.

Since the total number of exiting streams N_e is equal to the sum of $N_{e,F} + N_{e,P}$, the F and P rules together provide the required $N_e - 1$ auxiliary equations.

5.3. Cost Equations for EFCC Plant

In this section, cost equations formulated for the EFCC plant are presented. Cost balance equations are derived for each component based on the SPECO method. F and P rules are taken into account to formulate the relevant auxiliary equations.

Schematic representation for the EFCC plant is given in Figure 5.1. All exergy streams indicated in the figure are taken into account to perform an overall thermoeconomic analysis for the plant.

There are several assumptions behind the formulation. This is mainly due to the lack of reliable data for working conditions of some components, especially the air heater. The annual carrying charges and operating and maintanance costs are apportioned among the system components according to the contribution of each component to the sum of purchased-equipment costs. The relevant data found in the literature is used for carrying charges and operating and maintenance costs [10]. A zero unit cost is assumed for air entering the air compressor. All data are for operation at steady state. The results of the preceding chapter are utilized for exergy streams.

Cost equations are given below for each component, together with the assumptions and the auxiliary equations.

Compressor :

$$\dot{C}_1 = 0 \qquad \text{(Assumption)} \qquad (5.8)$$

$$\dot{C}_1 + \dot{C}_{11} + \dot{Z}_{ac} = \dot{C}_2 \qquad \text{(Cost Balance)} \qquad (5.9)$$

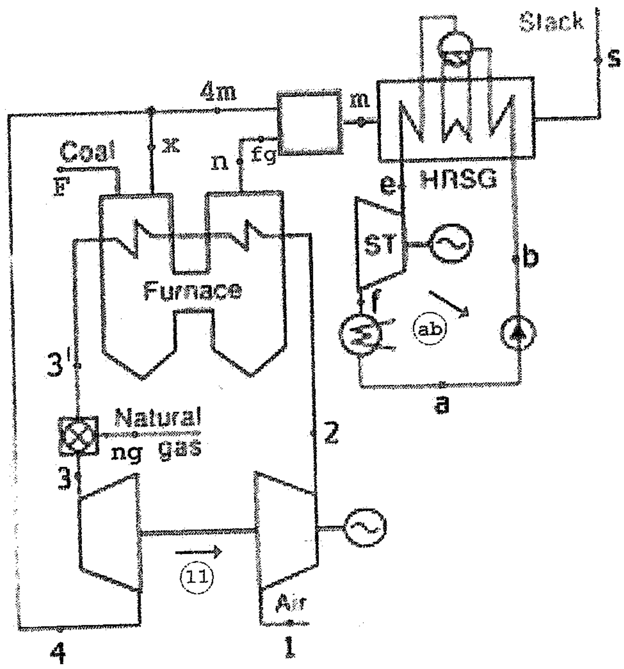


Figure 5.1. Schematic representation of the EFCC plant with all exergy streams

Gas turbine :

$$\dot{C}_3 + \dot{Z}_{gt} = \dot{C}_4 + \dot{C}_{11} + \dot{C}_{34} \qquad \text{(Cost Balance)} \qquad (5.10)$$

$$\frac{\dot{C}_3}{\dot{E}_3} = \frac{\dot{C}_4}{\dot{E}_4} \qquad \text{(Auxiliary Equation – F Rule)} \qquad (5.11)$$

$$\frac{\dot{C}_{11}}{\dot{W}_{ac}} = \frac{\dot{C}_{34}}{\dot{W}_{34}} \qquad \text{(Auxiliary Equation – F Rule) for Compressor and Gas turbine} \qquad (5.12)$$

Branch point x :

$$\dot{C}_4 = \dot{C}_{4m} + \dot{C}_x \qquad \text{(Cost Balance)} \qquad (5.13)$$

$$\frac{\dot{C}_{4m}}{\dot{E}_{4m}} = \frac{\dot{C}_r}{\dot{E}_r} \qquad \text{(Auxiliary Equation – F Rule)} \qquad (5.14)$$

Air heater :

$$\dot{C}_x + \dot{C}_F + \dot{Z}_{cc} = \frac{\dot{E}_g}{\dot{E}_{fg}} \dot{C}_{fg} \qquad \text{(Cost Balance for Combustor)} \qquad (5.15)$$

$$\left(\frac{\dot{E}_g}{\dot{E}_{fg}} - 1 \right) \dot{C}_{fg} + \dot{C}_2 + \dot{Z}_{ah} = \dot{C}_3 \qquad \text{(Cost Balance for Heat exchanger)} \qquad (5.16)$$

Heat Recovery Steam Generator :

$$\dot{C}_m + \dot{Z}_{sg} + \dot{C}_b = \dot{C}_e + \dot{C}_s \qquad \text{(Cost Balance)} \qquad (5.17)$$

$$\frac{\dot{C}_m}{\dot{E}_m} = \frac{\dot{C}_s}{\dot{E}_s} \qquad \text{(Auxiliary Equation – F Rule)} \qquad (5.18)$$

Mixing Unit :

$$\dot{C}_{4m} + \dot{C}_{fg} + \dot{Z}_m = \dot{C}_m \qquad \text{(Cost Balance)} \qquad (5.19)$$

Pump :

$$\dot{C}_a + \dot{Z}_{pump} + \dot{C}_{ab} = C_b$$ (Cost Balance) (5.20)

Steam turbine :

$$\dot{C}_e + \dot{Z}_{st} = \dot{C}_f + \dot{C}_{ef} + \dot{C}_{ab}$$ (Cost Balance) (5.21)

$$\frac{\dot{C}_e}{\dot{E}_e} = \frac{\dot{C}_f}{\dot{E}_f}$$ (Auxiliary Equation – F Rule) (5.22)

$$\frac{\dot{C}_{ab}}{\dot{E}_{ab}} = \frac{\dot{C}_{ef}}{\dot{E}_{ef}}$$ (Auxiliary Equation – F Rule) for Pump and Steam turbine (5.23)

Condenser :

$$\dot{C}_f + \dot{Z}_{cond} = \dot{C}_a + \dot{C}_{qout}$$ (Cost Balance) (5.24)

$$\frac{\dot{C}_f}{\dot{E}_f} = \frac{\dot{C}_a}{\dot{E}_a}$$ (Auxiliary Equation – F Rule) (5.25)

Supplementary firing unit :

$$\dot{C}_{3'} + \dot{C}_{q_{ng}} + \dot{Z}_{sp} = \dot{C}_3$$ (Cost Balance) (5.26)

The following relationships, which represent the PEC terms as a function of system parametes, are utilized for calculating the purchased-equipment cost (PEC) terms [10]. For some of the components, PEC terms are not described in the literature as a function of sytem parameters. Instead, scaling exponent approach can be employed in the absence of relevant cost information. Scaling exponent approach aims at estimating the cost of a component based on the information for the same equipment with different capacity or size [10].

Compressor :

$$PEC_c = \left(\frac{71.10\dot{m}_{air}}{0.9 - \eta_{sc}} \right) r_p \ln r_p \tag{5.27}$$

Combustion chamber :

$$PEC_{cc} = \left(\frac{46.08\dot{m}_{air}}{0.995 - \frac{P_3}{P_{3'}}} \right) \left[1 + \exp\left(0.018T_3 - 26.4 \right) \right] \tag{5.28}$$

Gas turbine :

$$PEC_{gt} = \left(\frac{479.34\dot{m}_g}{0.92 - \eta_{st}} \right) \ln r_p \left[1 + \exp\left(0.036T_3 - 54.4 \right) \right] \tag{5.29}$$

Heat recovery steam generator :

$$PEC_{hrsg} = 6570 \left[\left(\frac{\dot{Q}_{ec}}{\Delta T_{lm,ec}} \right)^{0.8} + \left(\frac{\dot{Q}_{ev}}{\Delta T_{lm,ev}} \right)^{0.8} + \left(\frac{\dot{Q}_{sup}}{\Delta T_{lm,sup}} \right)^{0.8} \right] + 21276\dot{m}_{st} + 1184.4\dot{m}_g \tag{5.30}$$

Studies on air heater have been gaining interest in recent years. Therefore, detailed cost figures are not available in the literature. In this study, air heater is considered to be composed of two parts. The heat exchanger part is modelled as a standard heat exchanger. The other part, where combustion takes place, is formulated based on a standard combustor model.

5.4. Results of Performance Evaluation

The solution steps for the thermoeconomic analysis are outlined in Appendix D. The linear equation set derived for the EFCC plant is solved to obtain the average cost values per unit exergy for each material stream. Cost formation processes are investigated in a detailed manner and cost distribution is assessed for the whole plant.

Table 5.1 illustrates the results of the thermoeconomic analysis for coal only option. Highest exergy unit cost values are achieved at streams 2 and b, which correspond to the exit states of compressor and pump, respectively. This is attributed to the fact that all exergy available at the exit is supplied by mechanical power, which indeed is the most expensive fuel within the system.

Similar results are obtained for the EFCC plant with supplementary firing conditions. Table 5.2 and Table 5.3 outline the cost distribution for the 27 % and 52 % supplementary firing levels, in respect. As can be seen from the tables, in supplementary firing conditions, costs for work streams obtained from gas and steam turbines are higher, than that for coal only option. Another observation is that, product costs increase as the level of supplementary firing is increased. This is due to the increase in utilization of natural gas in the supplementary firing unit. Unit cost of natural gas is more than two times higher than that for coal to produce the same amount of energy [16].

Table 5.1. Cost Distribution for coal only option

Stream	Exergy E (MW)	Levelized Cost rate C ($/h)	Levelized cost per exergy unit c ($/GJ)
x	1.21	94.04	21.65
4m	11.46	893.56	21.65
4	13.09	987.59	20.96
3	50.30	3795.23	20.96
11	18.80	1039.29	15.35
34	34.98	1933.50	15.35
2	6.18	1164.52	52.31
fg	7.99	288.60	10.03
m	21.25	1208.99	15.80
s	2.24	127.25	15.80
e	16.41	1191.50	20.16
b	0.01	2.42	66.15
f	1.86	40.27	6.01
Qout	1.05	100.95	26.69
a	0.005	0.35	20.16
ab	0.03	2.05	22.80
ef	13.99	1148.88	22.80
1	0	0	0
3x		-	-

Table 5.2. Cost Distribution for 27 % supplementary firing level

Stream	Exergy E (MW)	Levelized Cost rate C ($/h)	Levelized cost per exergy unit c ($/GJ)
x	1.07	99.06	25.73
4m	9.95	921.29	25.73
4	13.14	1020.34	21.57
3	49.64	3854.90	21.57
11	16.42	978.41	16.55
34	34.34	2046.60	16.55
2	5.40	1122.83	62.40
fg	5.50	282.50	14.26
m	18.60	1235.34	18.45
s	1.98	131.77	18.45
e	14.34	1232.31	23.87
b	0.009	2.50	78.38
f	0.49	42.03	23.87
Qout	1.62	41.69	7.13
a	0.004	0.36	23.87
ab	0.02	2.11	26.99
ef	12.23	1188.20	26.99
1	0	0	0
3x	26.17	3624.69	38.47

Table 5.3. Cost Distribution for 52 % supplementary firing level

Stream	Exergy E (MW)	Levelized Cost rate C ($/h)	Levelized cost per exergy unit c ($/GJ)
x	0.97	95.29	27.38
4m	9.42	928.29	27.38
4	13.20	1023.58	21.54
3	49.22	3816.84	21.54
11	14.67	912.04	17.27
34	25.69	1905.28	20.60
2	4.82	1083.72	62.40
fg	3.56	272.06	21.22
m	16.62	1238.42	20.72
s	1.80	133.86	20.70
e	12.79	1259.44	27.36
b	0.008	2.57	90.21
f	0.44	42.98	27.36
Qout	1.45	42.65	8.18
a	0.004	0.37	27.36
ab	0.02	2.16	30.94
ef	10.90	1214.33	30.94
1	0	0	0
3x	17.13	2742	55.52

In this study, effect of pressure ratio on gas turbine unit exergy cost is also investigated. Figure 5.1 illustrates the unit exergy cost of work obtained from gas turbine as a function of the pressure ratio. Although PEC for the gas turbine increases with increasing pressure ratio, unit exergy cost for the gas turbine is decreased. This is because of the fact that, increasing pressure ratio increases the temperature at the air heater inlet, which in turn implies that less fuel consumption is required to achieve the specified turbine inlet temperature. Unit exergy cost for the gas turbine is also related with the performance of other components, especially the compressor.

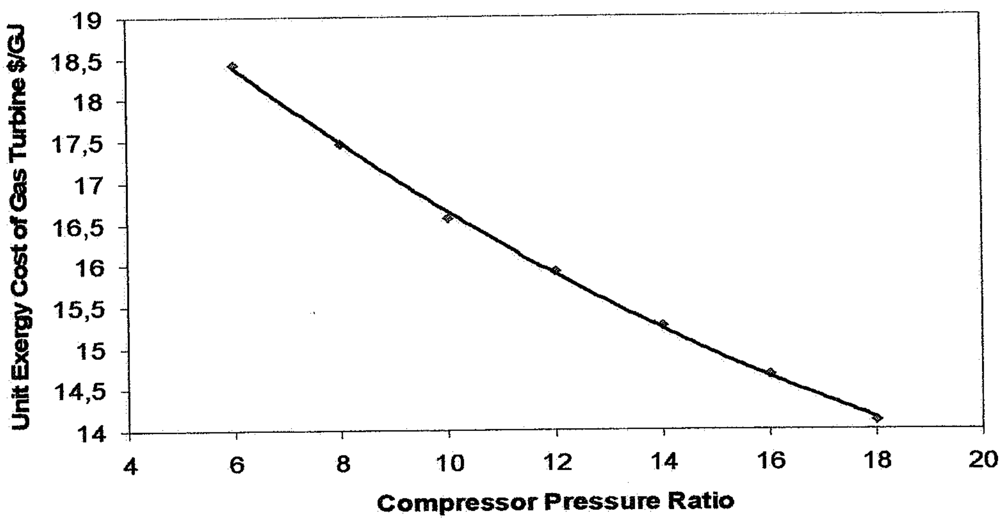


Figure 5.2 Unit exergy cost of work obtained from GT versus compressor ratio

CHAPTER 6

RESULTS AND CONCLUSIONS

In this study, performance of the EFCC plant is investigated. Performance assessment is done based on energy and exergy analysis. Thermoeconomic analysis coupled with exergy related analysis is also performed to evaluate the overall performance of the system. Effect of supplementary firing is also analyzed.

Energy analysis for the cycle is performed, as the first step for the performance evaluation. Turbine inlet temperature, compressor pressure ratio and boiler pressure are selected as the parameters for which effect on LHV efficiency and specific work output of the cycle is analyzed.

It is demonstrated that specific work output and the LHV efficiency of the EFCC plant increases with increasing turbine inlet temperature. The same observation is valid for the effect of boiler pressure. On the other hand, effect of the pressure ratio shows a different behavior. There exists an optimum pressure ratio for which the LHV efficiency reaches its maximum, with *TIT and* P_b are both kept constant, while specific work output of the EFCC plant decreases with increasing pressure ratio, as figured out in Figure 6.1.

Exergy analysis is performed and rational efficiency values are obtained for changes in selected parameters. It is shown that increasing the main steam pressure increases the plant performance in terms of rational efficiency. It is further observed that there exists an optimum pressure ratio for which the rational efficiency achieves a maximum, while plant rational efficiency increases linearly with increasing *TIT*.

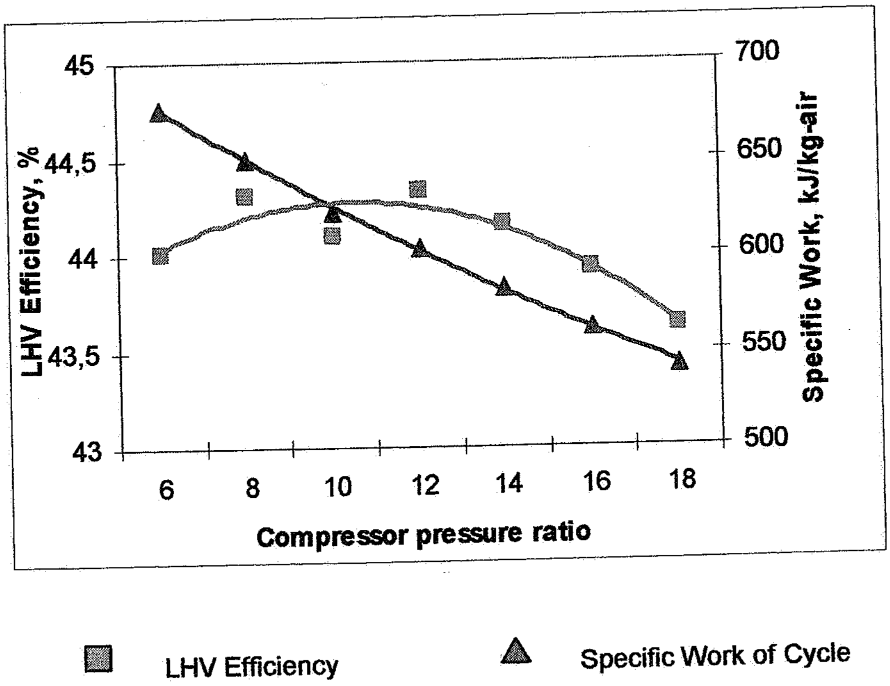
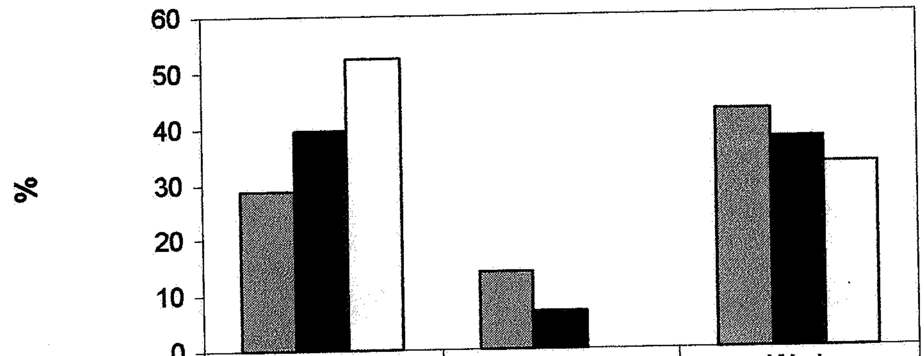


Figure 6.1. Effect of compressor pressure ratio on performance of the EFCC plant, based on energy analysis.

Exergy destruction throughout the EFCC plant is investigated in this study. Performance of each component is determined based on exergy analysis. Exergy lost throughout the cycle is determined per unit fuel input.

Air heater is the most significant component in terms of exergy losses. For a plant working in coal only option, around half of the total exergy input is diminished within the air heater, while 33% is converted to useful work. Exergy breakdown throughout the cycle is also investigated for different levels of supplementary firing. Results of the analysis indicate that supplementary firing improves the plant performance in terms of rational efficiency. For the EFCC plant working in 27% supplementary firing level, rational efficiency reaches around 37%, whereas nearly 43% rational efficiency is achieved in 52% supplementary firing conditions. Figure 6.2 illustrates the exergy loss characteristics for the EFCC plant investigated in this study, together with the work obtained for unit fuel exergy input.



	Air Heater losses	SFU losses	Work
SF-52%	28,62	14	42,75
SF-27 %	39,6	7	37,78
CO	52,44	0	33

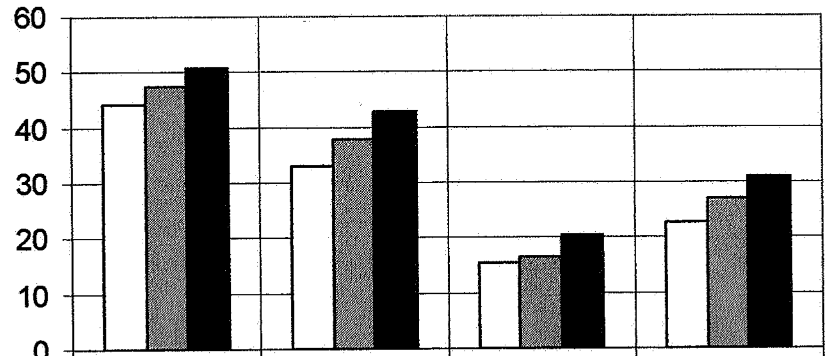
Figure 6.2 Comparison of 2nd law related performance of EFCC plant for different levels of supplementary firing.

Thermoeconomic analyses are also performed for the EFCC plant to reach a more realistic conclusion about the plant performance. Cost formation processes are analyzed and flow of costs throughout the EFCC plant is determined.

SPECO method is employed for the calculation of costs by systematically registering the exergy and cost additions and removals for each material stream. Cost balance equations are derived for each component. F and P rules are implemented for auxiliary equations. Levelized cost rates and levelized cost per exergy unit are determined for each material stream, based on the results of exergy analysis.

It is shown that, highest exergy unit cost values are achieved at the exit stages of compressor and pump. This is attributed to the fact that all exergy available at the exit is supplied by mechanical power, which indeed represents the most expensive fuel within the system. Costs for work streams obtained from gas and steam turbines are higher in supplementary firing conditions than that for coal only option. This is due to addition of natural gas into the cycle, as a more expensive fuel as compared with the coal.

Overall performance assessment task is accomplished with the results of thermoeconomic analysis. Key performance indicators obtained by this study are given in Figure 6.3. As can be seen form the figure, although plant performance is improved by supplementary firing in terms of 1st and 2nd law efficiency figures, the same observation is not valid regarding thermoeconomic performance.



	LHV efficiency(%)	rational efficiency (%)	unit exergy cost for GT ($/GJ)	unit exergy cost for ST ($/GJ)
□ Coal Only	44,16	32,82	15,35	22,8
▨ SF - level 27 %	47,63	37,78	16,55	26,99
■ SF - level 52 %	50,79	42,75	20,6	30,94

Figure 6.3 Overall performance of the EFCC plant for different levels of supplementary firing.

Another observation of this study is that gas turbine to steam turbine power ratio is getting larger as the level of supplementary firing is increased. Gas turbine produces around 17% more power than steam turbine for coal only option. As the natural gas is utilized within the system in supplementary firing conditions, gas turbine to steam turbine power ratio increases, and reaches around 1.77 for 52% supplementary firing condition. The results of this study outlines a distinct feature of the EFCC plant in this respect, as compared with CCGT, IGCC, and the PFBC plants.

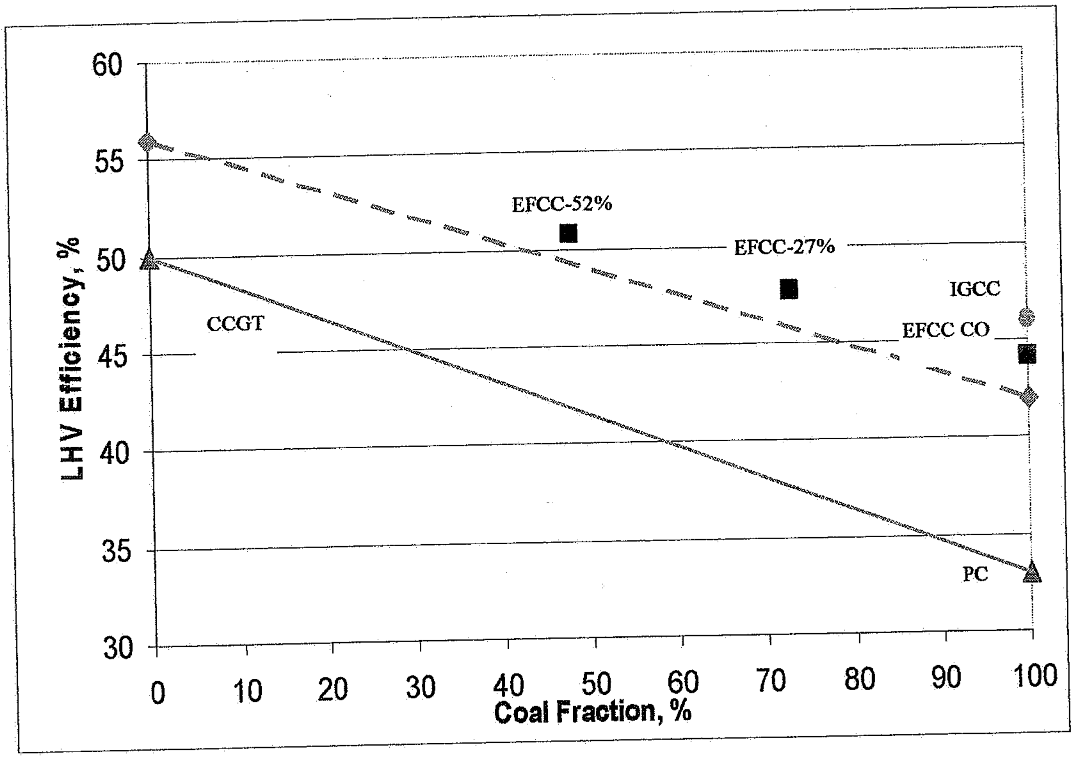


Figure 6.4 LHV efficiency for different plant technologies as a function of coal fraction in the total fuel input.

Results of this study outline the high performance characteristics of the EFCC plant. Figure 6.4 represents the LHV efficiency figures obtained in this study together with the results found in the literature for several other advanced plant technologies. As illustrated in the figure, EFCC offers LHV efficiency figures higher than those for conventional coal fired plants such as PC, and similar to those of advanced coal fired plants such as IGCC. EFCC has an advantage above the mean efficiency line between the PC and CCGT. On the other hand, when compared with the advanced plants, depicted by dashed line on the chart, the advantage becomes smaller as the coal fraction in the fuel is decreased. This observation indicates that the effect of supplementary firing is rather limited in the long term, since natural gas can better be utilized in CCGT plant.

EFCC plant is based on the currently available technology, except for the air heater, which is the most critical component. EFCC plant offers performance figures similar to other clean coal technologies such as IGCC. Supplementary firing modifies the performance of the plant in terms of 1^{st} and 2^{nd} law efficiencies, but cost of the products also increases.

As the uncertainty associated with cost data is larger than that for thermodynamic data, economic calculations do not have the same accuracy as thermodynamic calculations. Therefore a more detailed thermodynamic analysis can be performed to reach a more realistic conclusion about the overall performance.

For future work, a comprehensive study can be performed to decrease the exergy losses in the major plant components such as the air heater. This, in turn, would improve the plant rational efficiency. Different levels of pressure can be investigated for the heat recovery steam generator.

Clean coal technologies aim at improving the environmental characteristics in a cost effective manner. In this study, a rather simple model is employed to analyze the NO emissions for the EFCC plant. More detailed approaches can be preferred to determine the environmental effects, especially the NO emissions. Furthermore, a thermoeconomic study taking into account the emission control related components can also be a useful extension of this study.

In this study anthracite coal is considered as the primary fuel for the EFCC plant under consideration. Performance evaluation can be repeated based on different coal types, such as lignite. Re-powering configuration which is the modification of the steam boiler in an existing steam power plant can also be analyzed and compared with the results of this study, especially in economic terms.

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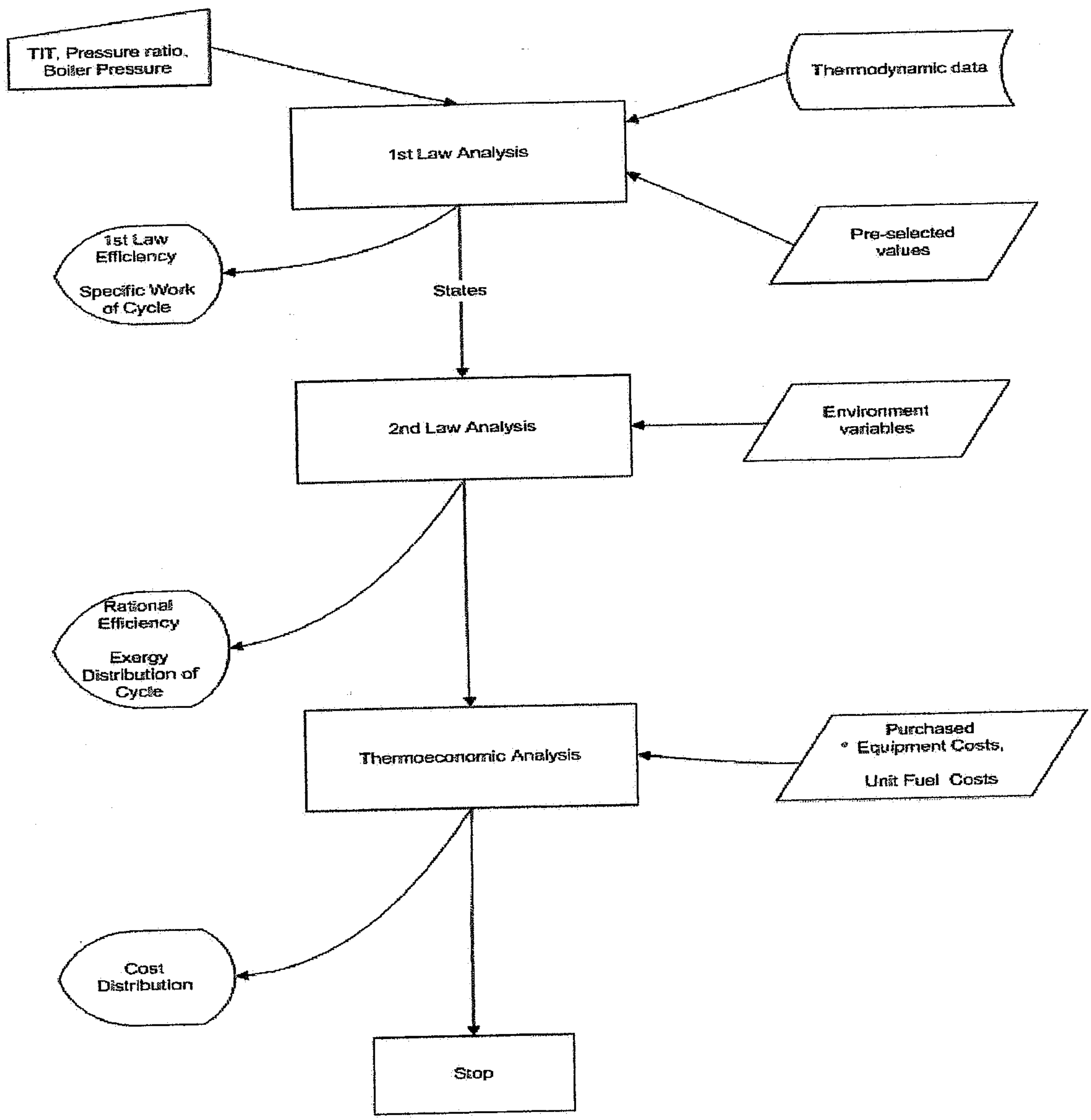
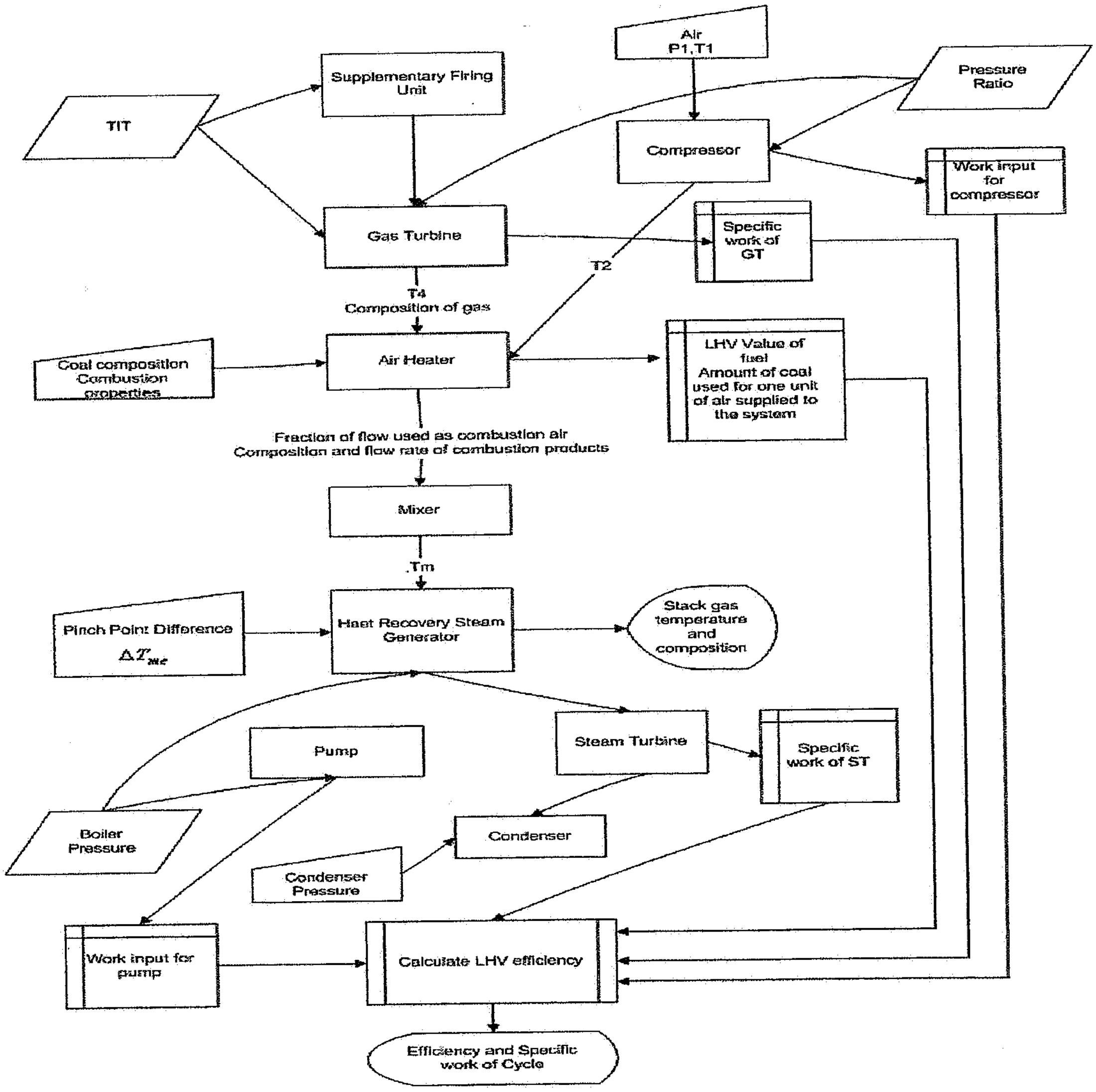
APPENDIX A

PERFORMANCE ASSESSMENT STEPS



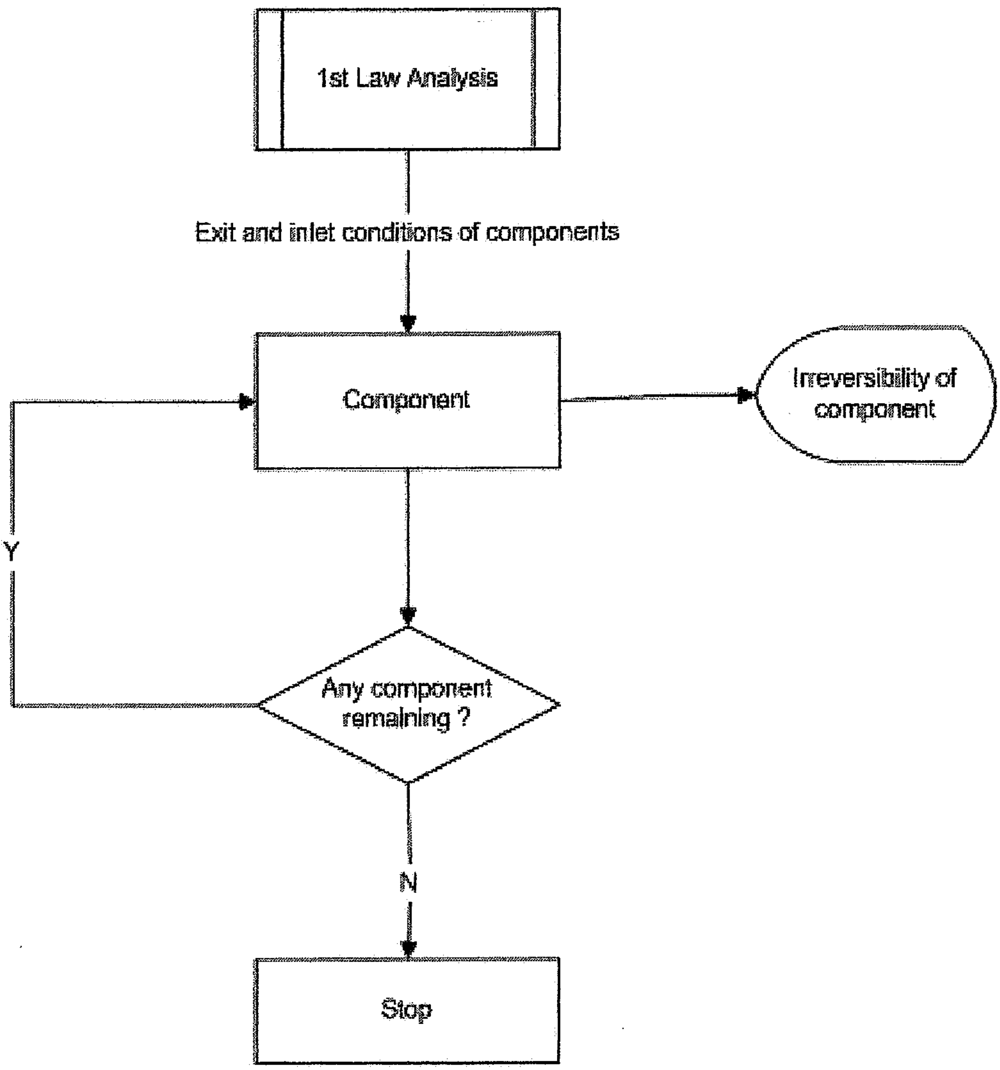
APPENDIX B

FIRST LAW ANALYSIS



APPENDIX C

SECOND LAW ANALYSIS



APPENDIX D

THERMOECONOMIC ANALYSIS