GAZİANTEP UNIVERSITY GRADUATE SCHOOL OF NATURAL & APPLIED SCIENCES

EXERGETIC AND THERMOECONOMIC PERFORMANCE ANALYSIS AND OPTIMIZATION OF DIESEL ENGINE POWERED COGENERATION SYSTEMS

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BY
AYŞEGÜL ABUŞOĞLU

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Exergetic and Thermoeconomic Performance Analysis and Optimization of Diesel Engine Powered Cogeneration Systems

Ph. D. Thesis
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Supervisor
Assoc. Prof. Dr. Mehmet KANOĞLU

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ABSTRACT

EXERGETIC AND THERMOECONOMIC PERFORMANCE ANALYSIS AND OPTIMIZATION OF DIESEL ENGINE POWERED COGENERATION SYSTEMS

ABUŞOĞLU, Ayşegül
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In this thesis, we perform exergetic and thermoeconomic performance analysis and optimization of diesel engine powered cogeneration systems. Thermoeconomic analysis is based on exergetic cost accounting method. The procedure and formulation of thermodynamic and thermoeconomic analysis are provided and they are applied to an actual cogeneration system located in Gaziantep, Turkey [Sanko Diesel Engine Powered Cogeneration (DEPC) plant] with an electrical power of 25.32 MW and a steam mass flow rate of 8100 kg/h. A detailed energy and exergy analysis is applied to each component of the system and performance parameters based on both the first law and the second law are defined for diesel cogeneration. The cost functions of each stream in the plant is obtained using specific exergy costing (SPECO) method. Cost relations at the component level are related with certain exergoeconomic variables, and exergoeconomic optimization of the actual DEPC plant is performed. The approach presented in this thesis is an iterative performance improvement procedure in which the integration of the cost and performance data for a given system component permits the calculation of optimum design conditions when the exergy of the component product and the cost of the component fuel based on SPECO approach remain constant.

Some of the main results obtained from this study may be summarized as follows: The fuel utilization efficiency of the plant is 44.6% and power to heat ratio is 143.8.

The exergy input to the plant by fuel is 62,800 kW and 40.4% of this exergy is converted to electrical power while steam production accounts for 0.3% of exergy input. The corresponding exergetic efficiency of this cogeneration plant is 40.6%. It is clear that diesel cogeneration systems involve high electrical output compared to process heat and they should be selected for such applications. The exergetic cost rate and the specific unit exergetic cost of the fuel entering the plant are determined to be 1806 $/h and 2.70 $/GJ, respectively. In optimization studies of the plant, the electricity cost and steam cost are determined for three cases. The electricity cost is 8.90 ¢/kWh in actual base case, 6.70 ¢/kWh in thermoeconomically optimal case, and 23.2 ¢/kWh in thermodynamically optimal case. The steam costs are 5.22 ¢/kWh, 4.50 ¢/kWh, and 10.04 ¢/kWh, respectively, in these three cases. Thermodynamically optimal values do not appear to be cost effective while thermoeconomical optimization results are much more in line with actual base case values indicating the usefullness of combining thermodynamics with economics in design, analysis, and optimization of energy systems. It is also noted that thermoeconomic optimum solution is strongly depended on the cost functions and characteristic coefficient values of the functions defined.

Exhaust emission characteristics of DEPC plant are also assessed, and allocation amounts of emissions of the DEPC system to the power produced and steam generated are calculated. Exhaust emission assessment is done following both energy and exergy based approaches. We determined that using combined power and heat production provide fuel savings of 23.8% in energy approach and 17.3% in exergy approach with respect to separate units of power and heat production.

Key words: Combined Heat and Power (CHP), Cogeneration, Diesel Engine, Energy, Exergy, Exergoeconomic Analysis, Thermoeconomic Analysis, Optimization

ÖZET

DİZEL MOTORLU KOJENERASYON SİSTEMLERİNİN EKSERJETİK VE TERMOEKONOMİK PERFORMANS ANALİZİ VE OPTİMİZASYONU

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Bu çalışmada, termoekonomik analiz ve optimizasyon yaklaşımı, Gaziantep'te bulunan, 25.32 MW elektrik ve 8100 kg/saat buhar çıktısı olan ağır yakıtlı ve dizel motor tahrikli gerçek bir kojenerasyon sistemine, Sanko Dizel Motorlu Kojenerasyon'a uygulanmıştır. Bu amaçla ilk olarak her bir sistem bileşenine detaylı enerji ve ekserji analizi yapılmıştır. Sistemdeki her akımın maliyet fonksiyonları, birim ekserji maliyetlendirme (SPECO) metodu, geleneksel ekonomik metotlarla birleştirilerek elde edilmiş ve hesaplanmıştır. Elde edilen maliyet ilişkileri optimizasyon işleminin başlangıç adımı olarak ekserjiye bağlı tanımlanan uygun ekonomik değişkenlerle ilişkilendirilmiştir. Ayrıca sistemin performans parametreleri de tanımlanmış ve hesaplanmıştır. Son olarak, sistemin termoekonomik optimizasyonu gerçekleştirilmiştir. Bu çalışmada sunulan yaklaşım, herhangi bir sistem bileşeni için verilen maliyet ve performans girdilerinin birleşiminin, sistemdeki bileşenlerin ürün çıktılarının ekserjileri ve yakıt maliyetleri sabit kalmak koşuluyla, optimum dizayn şartlarında hesaplanmasını mümkün kılan iteratif (yinelemeli) bir performans iyileştirme yöntemidir. Kojenerasyon sistemine giren yakıtın ekserjisi 62,800 kW olarak hesaplanmıştır. Sisteme giren toplam yakıt ekserjisinin %40.4' ü elektriğe çevrilmektedir. Sistemin net buhar üretimi ise toplam ekserji girdisinin %0.3' üdür. Dizel motorlu kojenerasyon sisteminin ikinci kanun (ekserji) verimi %40.6 olarak hesaplanmıştır. Gerçek çalışma şartlarında sisteme giren yakıtın ekserjiye bağlı maliyeti ve birim ekserji maliyeti sırasıyla 1806 $/saat ve 2.70 $/GJ bulunmuştur. Sistemdeki dizel motorlar için yakıt girdisi değerleri sırasıyla 1933.3 $/saat ve 2.85 $/GJ olarak hesaplanmıştır.Gerçek çalışma şartlarında elektrik maliyeti 10.31 $/GJ (8.90 ¢/kW-saat) bulunmuştur. Termoekonomik optimizasyon çalışması neticesinde elektrik maliyeti 8.38 $/GJ (6.70 ¢/Kw-saat) olarak hesaplanmıştır, termodinamik optimizasyon sonucunda ise elektrik maliyeti 21.36 $/GJ' e (23.18 ¢/kW-saat) yükselmektedir. Sistemde üretilen buharın gerçek çalışma şartlarında, termodinamik ve termoekonomik optimizasyon çalışmaları sonucunda maliyeti sırasıyla, 5.22 ¢/kW-saat, 10.04 ¢/kW-saat, and 4.50 ¢/kW-saat olarak hesaplanmıştır. Bu çalışmadan elde edilen sonuçlar, termoekonomiye bağlı optimum çözümlemenin, maliyet fonksiyonlarına ve bu fonksiyonların karakteristik katsayı değerlerine kuvvetle bağımlı olduğunu göstermiştir.

Bu çalışmada ayrıca dizel motorlu kojenerasyon sisteminin emisyon karakteristikleri ele alınmış ve emisyonların güç ve buhar üretimine bağlı değerleri, literatürdeki temel metotlar kullanılarak ve geliştirilerek hesaplanmıştır. Egzoz emisyon değerlendirmesi, ekserjiye bağlı ilişkiler geliştirilerek yapılmış ve bulunan sonuçlar enerjiye bağlı metotlarla elde edilen sonuçlarla karşılaştırılmıştır. Analiz sonuçlarına göre, birleşik ısı ve güç kullanımı, ısı ve gücün ayrı üretildiği sistemlere göre enerji ve ekserjiye bazlı analizlerde yakıt tüketiminde sırasıyla %23.84, ve %17.25 oranında bir tasarruf sağlamaktadır.

Anahtar Kelimeler: Birleşik Isı ve Güç Üretimi (CHP), Kojenerasyon, Dizel Motor, Enerji, Ekserji, Eksergoekonomik Analiz, Termoekonomik Analiz, Optimizasyon

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c	cost per unit of exergy (\$/GJ)
$\dot{C}$	cost rate associated with exergy (\$/h)
C	payment (\$)
E	the amount of emission based on energy content (g/kWh)
$\dot{E}$	exergy rate (kW)
$\dot{E}_{\text{heat}}$	rate of exergy transfer by heat (kW)
F	the amount of emission based on shared emission savings in terms of energy (g/kWh)
X_{F}	the amount of emission based on shared emission savings in terms of exergy (g/kWh)
h	enthalpy (kJ/kg)
i	rate of return (%)
$\dot{m}$	mass flow rate (kg/s)
M	the amount of exhaust emission released (g/kWh)
P	present value of the payment (\$)
$\dot{Q}$	rate of heat transfer (kW)
s	entropy (kJ/kg K)
T	temperature (K)
T_0	environment temperature (K)
w	specific work (kJ/kg)
$\dot{W}$	power (kW)
X	the amount of emission based on exergy content (g/kWh)
θ	specific emission amount (g/kWh)
$y_{\text{dest},k}$	component exergy destruction over total exergy input
$y^{*}_{\text{dest},k}$	component exergy destruction over total exergy destruction
$\dot{Z}$	cost rate associated with the sum of capital investment and O&M (\$/h)
$\dot{Z}^{\text{CL}}$	cost rate associated with capital investment (\$/h)

$\dot{Z}^{OM}$	cost rate associated with O&M ($/h)

Abbreviations

BOO	Build Operate and Own
BOT	Build Operate and Transfer
BOTAŞ	Petroleum Pipeline Distribution
CC	Carrying Charges
CRF	Capital Recovery Factor
CO	Carbon monoxide
CO_2	Carbon dioxide
DEPC	Diesel Engine Powered Cogeneration
DeNOx	Denitrification
DeSOx	Desulphurization
DI	Direct Injection
DSİ	State Hydraulic Agency
EES	Engineering Equation Solver
EGR	Exhaust Gas Recirculation
EMRA	Energy Market Regulatory Authority
EML	Electricity Market Law
ESP	Electro Static Precipitators
EU	European Union
EXC	Expenditure Costs
EÜAŞ	Electricity Generation Incorporated Company
FGD	Flue Gas Desulphurization
GAP	Southeast Anatolia Hydropower and Irrigation Project
HFO	Heavy Fuel Oil
HRS	Heat Recovery System
IEA	International Energy Agency
ICE	Internal Combustion Engine

LHV	Lower Heating Value, kJ/ kg
LFO	Light Fuel Oil
LPG	Liquefied Petroleum Gas
MAN	Engine Company and Supplier
MENR	Ministry of Energy and Natural Resources
NO_X	Oxides of Nitrogen
OECD	Organization for Economic Cooperation and Development
OMC	Operating and Maintenance Costs
PEC	Purchased Equipment Cost
SO_2	Sulfur dioxide
TEAŞ	Turkish Electricity Generation Transmission Co.
TEDAŞ	Turkish Electricity Distribution Co.
TEİAŞ	Turkish Electricity Transmission Co.
TETAŞ	Turkish Electricity Trading and Contracting Co.
TEK	Turkish Electricity Administration
TKİ	Turkish Coal Enterprises

Subscripts

eff	effective
L	levelized
k	any component
m	year
dest	destruction
pp	power plant

Greek Letters

η	energy efficiency
η_{turb}	turbine isentropic efficiency

η_{comp}	compressor isentropic efficiency
ε	exergy efficiency
ψ	specific flow exergy (kJ/kg)
τ	total annual operating hours of system at full load (h)
φ	amount of specific emission (g/kWh)

CHAPTER 1

INTRODUCTION

1.1 Background

The utilization of energy and other resources in the industrial world has reached levels never observed before. This leads to a decreasing supply of natural resources and an increasing amount of damage and pollution to the natural environment. At the same time, energy resource conversion networks have become more complicated. Technical improvements are often focused towards less important resource conversions, which do not have significant potential to improve resource use. The scarcity and undesirable side effects of careless utilization of energy resources on economics and ecology require careful analysis and planning for proper energy consumption.

Energy is a vital input for the economic and social development of any country. Rapid population growth and urbanization in Turkey have played an important role for energy consumption. Turkey's energy demand has grown rapidly almost every year and is expected to continue growing. Turkey has different energy sources but these are limited to meet the country's total energy demand. More than half of the net energy consumption of the country is met by imports, and the share of imports of total energy consumption picture continues to increase each year.

An industrial plant may have peak electricity or steam demand during a particular shift each day. In manufacturing sector of Turkey, the industry can use two types of power: the public power source which is cheap but of lower quality (i.e. subject to more voltage fluctuations) and its own in-house power which is more expensive but of a higher quality. The extent of internal generation of power varies among firms and is related to their production technologies. Two most important reasons for producing electricity are the uncertainty about power output and the fluctuations in the voltage of public power which can cause damages to plant, equipment, and intermediate inputs and outputs in the assembly line. To minimize this problem, if not solve it completely, the firms produce their own power to either substitute or supplement the public supply. The endogenously generated power is used to "boost" the power supply obtained from the public sector; to smooth out voltage fluctuations in the public supply; or to supply power when the public power source is interrupted.

Nowadays, cogeneration is considered as one of the most important techniques for achieving a more efficient usage of fuels, natural and financial resources savings and environmental protection. Many countries make efforts to overcome obstacles and to facilitate its spreading. The motivations implemented include the relatively high cost of purchase of surplus power from the power corporations as well as the subsidy of investments. Other measures include communication, energy recording and analyses, research and development support etc. Cogeneration systems can be classified generally according to their prime movers as

- Steam turbine systems
- Gas turbine systems
- Reciprocating internal combustion engine powered systems
- Combined cycle systems
- Standardized cogeneration units (i.e. Packaged Systems)
- Fuel cell powered systems
- Stirling engine powered systems

Cogeneration systems based on reciprocating internal combustion engines such as diesel engines or gas engines can produce higher electricity to steam ratios. The diesel engine is the highest efficiency prime mover commercially available and this makes it particularly well suited to cogeneration applications requiring a relatively high proportion of electric power compared to thermal products.

1.2 Thermoeconomics

Everyday life is immersed in economics. Our society tends to substitute human and natural values by economic ones, i.e. prices. We put prices almost to everything even though we do not know its value. So, what is the alternative? We can approach to know the value of things and living entities if we try to reproduce or replace them. When doing this, we realize how difficult it is. How many resources of all kind of types are needed to get them? How much knowledge is necessary to understand the mechanisms by which resources can be converted into entities? These questions give a sense of ethics of conservation because we should not destroy what we don't know how to construct it.

This is why it is so important to know costs measured as a general lack of resources, independently of the numerical values we use to quantify them. However, as it is well known, the value of cost has its drawbacks. It depends on the limits of the system and on the exergy value in our case and on the purpose (efficiency) of each and every subsystem that inter-relates the productive structure of the system.

We have also another problem: are the costs of things stable with production? In other words, when we say that the cost of electricity is three, does this value change when we demand more electricity? Everybody knows that in small production intervals the average cost of the product remains almost constant. So, what does it mean *small* production intervals? This depends on the purpose of production type. In a macro level, average costs can be used without much error. On the contrary if we are interested in optimizing a power plant or a cogeneration plant as in the case of the present study, we realize that the cost of electricity is very sensitive to the efficiency of the plant components.

The purpose of the study is to optimize an actual cogeneration system costs thermoeconomically by the means of the exergetic cost accounting methodology which have clearly easy to follow procedure and accuracy of results as compared to pure mathematical optimization methods which may end with a complex and unsolvable problem in the case of cogeneration system improvement.

When we try to understand the heat and mass transfer mechanisms in a heat exchanger, we use the techniques provided by the heat and mass transfer sciences.

These sciences use and interpret pressure and temperature as overall values of properties that spatially and temporally change. The understanding of a system interconnection and efficiency is better attained using thermodynamics rather than the sciences of heat and mass transfer. Thermodynamics helps to understand how energy is used and degraded in the different parts of the system. Thermodynamics requires an input-output abstraction of the system. Some information can be obtained from its analyses, some other cannot. There is no contradiction between the different types of analysis. Rather they focus on different aspects of system.

When we focus on exergy accounting, we should have another level of knowledge, now connected to thermodynamics. Thermoeconomics uses thermodynamics but differs in the type of problems it tries to solve as well as the use of new concepts like cost from economics.

Exergy could be considered as the bridge between thermodynamics and cost accounting methodologies. This is because exergy connects with intensive properties like pressure, temperature, energy and so on and on the other side it can be rigorously defined and its cumulative consumption calculated. When we investigate all the properties of exergy costs we are giving a conceptual bridge between classical thermodynamics and economics. On the other hand, we must give a basis for a general resources accounting methodology that can use any other numerical values or even relax the rules of accounting themselves.

Cost has different meanings for different people and practitioners. Cost is most of the times related to money and not so much to physical resources, but in its broader meaning cost is measured in resources, in time, etc. Cost is in many cases mistaken as "price" synonym. We mostly agree on its conceptual difference but we forget it when we apply rules for cost accounting that include external considerations about the finality of the production, or the market's utility of the products we obtain. Thermoeconomics makes it clear and provides a unique way to connect the physical universal measure of loss, i.e. irreversibility with the loss of resources at the overall system level and then to economics. This is perhaps a very strong and motivated reason for the present study and also continuing research on the area of thermoeconomics.

1.3 Scope and Outline of the Study

In this thesis we analyze and optimize diesel engine powered cogeneration systems thermoeconomically by means of the exergetic cost accounting method. The procedure and formulation of such an analysis are provided and they are applied to an actual cogeneration system located in Gaziantep, Turkey [Sanko Diesel Engine Powered Cogeneration (DEPC)]. The outline of the study with respect to chapters is as follows:

In Chapter 2, a comprehensive literature survey on thermoeconomic analysis and optimization of cogeneration systems is presented. The survey is presented under four titles: exergy, thermoeconomics, thermoeconomic optimization, and cogeneration. The survey provides a historical view of thermoeconomics and various methodologies developed over the years. The advantageous and disadvantageous of different methods and their implications are discussed in reference to mentioned literature.

In Chapter 3, an overview of Turkey's energy use is presented. The content of this chapter includes world's energy situation, Turkey's general energy outlook, energy sources in Turkey, electricity market and private power production, and environmental aspects of Turkey's energy use. The role of diesel powered electricity generation and cogeneration applications are discussed as part of the Turkey's overall energy picture. It is shown that diesel engine cogeneration is an effective model for private companies willing to produce their own electricity while meeting part of their heating need.

In Chapter 4, general formulations of thermodynamic analysis including both energy and exergy methods are given.

In Chapter 5, general principles, terminology, and formulation of thermoeconomic analysis, which is also called exergoeconomic analysis are presented. The formulations in Chapters 4 and 5 are in general format and are applicable to various energy conversion systems including diesel powered cogeneration.

General principles and formulations of thermoeconomic optimization are provided in Chapter 6. These methods use a primary optimization performance measure: minimizing the total levelized cost of the system products that includes the cost of external fuel resources, capital investment and operating and maintenance cost.

In Chapter 7, the short overview of cogeneration concept and diesel engine powered cogeneration systems are provided. The detailed technical description of Sanko diesel engine powered cogeneration plant including main and auxiliary system components is presented.

Thermoeconomic analysis and optimization of Sanko DEPC system using actual plant data are performed in Chapter 8. By combining conventional economic analysis with thermoeconomic analyses, exergy cost accounting is given for all subcomponents of the plant and specific exergy costing (SPECO) method is used to obtain the cost formation structure of the plant. Exergetic cost rate balances and corresponding auxiliary equations are formulated for each subsystem of the plant.

In Chapter 9, exhaust emission characteristic of Sanko DEPC plant and operation of the $DeSO_x$ unit in the facility are presented. Exhaust emission assessment is done by using emission allocation and fuel saving analysis methods based on both energy and exergy analyses. Emission results of the cogeneration system are compared to separate applications of power and heat.

In Chapter 10, conclusions drawn from the study are pointed out and certain recommendations are provided.

CHAPTER 2

LITERATURE SURVEY

2.1 Introduction

In this chapter, a comprehensive literature survey on thermoeconomic analysis and optimization of cogeneration systems is presented. The survey is presented under four titles: exergy, thermoeconomics, thermoeconomic optimization, and cogeneration. The survey provides a historical view and various methodology developed over the years. The advantageous and disadvantageous of different methods and their practical implications are discussed in reference to mentioned literature.

2.2 Exergy

The concept of available energy was used by Darrieus in 1930 [1], who defined “thermodynamic efficiency” as being the quotient of the actual work obtained divided by the potential work that could be obtained for materials in steady flow. These ideas were advanced by Keenan in 1932 [2], who gave the name “effectiveness” to the aforementioned efficiency in order to avoid confusion with other efficiencies (e.g. the Carnot efficiency). Keenan described the steady-flow availability equation as promising to be “as revolutionary in its effect on thermodynamic reasoning” as the development of the steady-flow energy equation had been in its time. Unfortunately, Keenan’s insight was not shared by others at that time. Many years later, the importance of availability to the analysis of energy-conversion processes is better recognized elsewhere than in the countries where the ideas first arose. In 1956, Rant coined the term “exergy” for availability [3], which became widely accepted. The literature on the subject grew exponentially in subsequent years.

Baehr has made a useful review of the concept of exergy [4]. In his review, Baehr gave the following concise definition of exergy: “Exergy is that part of energy that can be transformed into any other form of energy”. Szargut appears also to agree on such a wide application of the term: he defines exergy for a system and for a flow process [5]. In addition to Rant, Baehr, and Szargut, significant work was also published by V. Brodyanskii in Russia [6].

After the pioneering works mentioned above, many prominent researchers contributed to the further development of exergy (or second-law) analysis and the dissemination of its application. Work in this area is continuing. Among the works of the rich literature of the exergy concept, the major works are taken below.

Wall [7] presented a number of exergy-based concepts and methods, e.g. efficiency concepts, exergy flow diagrams, exergy utility diagrams, life cycle exergy analysis, and exergy economy optimization. These tools are useful in order to describe, analyze, and optimize energy conversion systems.

Rosen et al. [8] indicated that, the complex array of energy forms involved in cogeneration-based district energy systems make them difficult to assess and compare thermodynamically without exergy analysis depending primarily attributable to the different nature and quality of the three product energy forms: electricity, heat, cool. For these types of systems, exergy analysis provides important insights into the performance and efficiency for an overall system and its separate components.

Dunbar et al. [9] presented the definitions for thermal, strain, and chemical energy/exergy as well as mechanical and thermoeconomical energy/exergy, and the equations of change for these properties were derived. In the resulting equations, terms appeared which explicitly revealed the interconversions between the different forms of energy/exergy, including the breakdown into reversible and irreversible conversions.

A general exergy balance equation which was applicable to any component of thermal systems had been formulated by Oh et al. [10]. The exergy analysis based on this developed equation permitted one to predict the thermal efficiency of the system, the exergy destruction in each component as well as the mass flow rate, the composition, and the temperature of the exhaust gases. They examined the performance of a 1000 kW gas turbine cogeneration system when it was operated at part and full-load conditions through this analysis. The predicted values of the performances for the system had been compared with the actual performance data provided by the gas turbine manufacturer.

The main objective of the study performed by Wall [11] was to show the applicability of exergy for studies of industrial processes. For this, he defined the concept of exergy and applied to industrial processes. It was stated that the application of the exergy concept provides information for long-term planning of resource management.

Rosen [12], according to his belief that understanding of exergy was essential by the public, explained his views about exergy and related subjects. He tried to clarify some confused understandings in energy, exergy, energy crisis in the world, energy conservation, energy and exergy efficiencies, energy and exergy securities etc. The author believed that such understanding is essential for the scientists to better address the energy issues of today and tomorrow.

Dincer [13] discussed the utilization of exergy from several perspectives by redefining exergy, energy and environment policy. The crucial remarks in the paper maybe summarized as follows: Exergy analysis is an effective method that uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the design and analysis of energy systems. Exergy analysis is the best primary tool in addressing the impact of energy resource use on the environment and a suitable technique for furthering the goal of more efficient energy-resource use and hence energy conservation. Exergy analysis is an efficient technique revealing whether or not and by how much it is possible to design more efficient energy systems by reducing the inefficiencies in existing systems. It is an essential indicator to distinguish the quality between energy resources and a beneficial concept in economics. Lastly, exergy analysis method is recognized as a new measure of environmental degradation and hence one of the potential techniques to minimize or eliminate the environmental impact.

Çengel et al. [14] showed that getting bigger was not necessarily better by examining the effect of mixing on exergy destruction. Uniting or combining systems to form larger, more powerful systems is good only if the systems worked in harmony in the new larger unit, and thus there is little or no exergy destruction. That will happen only if the combined system can act like a single homogeneous system. In the paper, the authors concluded that combining systems which are identical or almost identical will result in a larger system with larger exergy content. Combining two systems that are different states will yield a system that is larger in energy content, but smaller in exergy content. To avoid waste of work potential, such systems should be operated separately.

Bisio and Rubatto [15] deduced that the influence of different sources of irreversibilities in work transfer expressions for closed and steady-state open systems systematically. In the various cases, the hypotheses necessary to an analytical formulation and therefore the limits of the steady-state open systems are specified. It is finally concluded that the determination of the entropy production owing to several causes of irreversibilities allows the researchers to analyze the exergy efficiency of various processes.

Nikulshin et al. [16] described an innovative method for the exergy efficiency calculation of a complex energy intensive system with arbitrary structures. The method presented is based on a novel general equation to calculate the total system exergy efficiency, and on an exergy flow graph proposed by the authors. The method is put forward as "invariant" of technical details and structure of the system. This approach allows a user to obtain not only the exergy efficiency of the total system, but also to show the relationship between the exergy efficiency of an individual element and that of the whole system. The method is applied to the thermodynamic exergy analysis of a power plant in the related paper.

Havelsky [17] discussed the problem of energetic efficiency evaluation of cogeneration systems for combined heat, cold and power production. Comparison of energetic efficiency of combined cogeneration systems with contemporary conventional separate production of individual energy flows, on the basis of the mentioned evaluation methods in the paper, shows a large potential for energy saving in systems of combined production of heat and mechanical energy, especially when they are operated at the current need for cold, heat, and power production throughout the year.

Kanoğlu [18] performed an exergy analysis of an actual binary geothermal power plant. The study describes an easy-to-follow procedure for exergy analysis of binary geothermal power plants and how to apply presented procedure to assess the plant performance by pinpointing sites of primary exergy destruction and thus showing the direction for improvements.

Yumrutaş et al. [19] presented a computational model based on the exergy analysis for the investigation of the effects of the evaporating and condensing temperatures on the pressure losses, the exergy losses, the second law efficiency, and the coefficient of performance of a vapor compression refrigeration cycle. It is found that the evaporating and condensing temperatures have strong effects on the exergy losses in the evaporator and condenser, and on the second law efficiency and COP of the cycle but little effects on the other components of the exergy losses. The second law efficiency and the COP increases, and the total exergy loss decreases with decreasing temperature difference between the evaporator and refrigerated space and between the condenser and outside air.

Kanoğlu et al. [20] developed a procedure for the energy and exergy analyses of an open-cycle desiccant cooling systems and it was applied to an experimental unit operating in ventilation mode with natural zeolite as the desiccant. The same procedure and formulations presented in this paper may easily be applied to the units operating in recirculation mode. The analysis through study shows that an exergy analysis can provide useful information with respect to the theoretical upper limit of the system performance, which cannot be obtained from a conventional energy analysis alone. This type analysis allows researchers to identify and quantify the sites with the losses of exergy, and therefore showing the direction for the minimization of exergy losses to approach the reversible coefficient of performance value.

Çengel [21] used thermodynamic concepts and principles in order to analyze merging and breaking-up processes in daily life, using exergy destruction associated with various mixing processes as a guide.

Rosen [22-28] explained his views on the relations between exergy, exergetic efficiency, economics and environment. He pointed out that better coverage of exergy is needed to improve thermodynamics education and to make it more interesting to students, and that a basic level of "exergy literacy" is needed among engineers and scientists-particularly those involved in decision making [29].

Wang et al. [30] applied the Reynolds time average method to the set of equations of exergy transfer for turbulent flows and found that the exergy destruction is mainly due to two factors, i.e. the mean flow dissipations and fluctuating motion dissipations. By using the model, they obtained the numerical solution and analyzed mechanisms of irreversibility for the flow characteristics mentioned. Conclusively, it is shown that for a given fluid, the total exergy destruction per unit length is a function of the geometrical parameters and the boundary conditions as well as the Reynolds number.

Wall [31] demonstrated the usefulness of the exergy concept for analyzing systems which convert energy, material and/or information, e.g., a society or an industrial process. Hence, he discussed physical concepts for resource accounting and suggested a number of basic concepts that could also be valuable in social and economic sciences. He used the methods developed in his thesis to examine the conversions of energy and material resources in Japanese society [32].

A similar study is done for Turkey by İleri and Gürer [33]. In this paper, energy and exergy utilization in Turkey was analyzed. They compared the results obtained for different sectors such as transportation, industrial, residential and commercial applications. Comparison was made by considering magnitudes of energy and exergy inputs and losses, energy and exergy efficiencies and differences in these efficiencies to assess the nature of losses. It was clear that a conscious and planned effort is needed to improve exergy utilization in Turkey, focusing first on the residential and commercial sectors.

Chang and Chuang [34] proposed a two-level idealization concept composed of the reversible operation idealization and the thermodynamic equilibrium operation idealization. By incorporating this concept with the concept of exergy, authors tried to define the intrinsic and extrinsic exergy losses to quantify the extent of deviations from these two types of idealizations. They demonstrated several example cases of different complex levels and the analyses results pinpointed what and where to focus on for improvements.

Wall and Gong [35] outlined the conditions and concepts associated with the use of exergy as an ecologic indicator in the first part of their study. The purpose of that part was to introduce the exergy concept and to give a foundation for the methods that will be introduced together with a number of different ecological indicators and arguments in favor of using exergy as an ecological indicator. The second part of their study [36] was an overview of number of different methods based on concepts presented in the first part and these methods were applied to real systems. A number of different ecological indicators were presented and the concept of sustainable development was clarified.

Rakopoulos and Giakoumis [37] performed a detailed first and second law analyses on a single-cylinder, naturally aspirated, indirect injection diesel engine to study the energetic and exergetic performance of engine subsystems during various transient operating schedules comprising changes in speed and load.

Szargut [38] defined partial exergy losses appearing in particular parts of thermal systems and formulated balance equations determining these losses. In the paper, sequence method of calculation of partial exergy losses was presented. The method was applied to a cogeneration plant and makes it possible to improve the clarity of obtained results, by the cumulation of some partial losses.

Kumar et al. [39] presented the methodology and some preliminary results in the application of exergy as a second law analysis parameter to a complete diesel engine cycle. For this purpose, a single-cylinder direct injection Diesel engine was simulated. Preliminary results showed that exergy is indeed a powerful tool in quantifying the losses and the irreversibilities in internal combustion engines. Also, a relatively simple analysis such as this can help identify the portions of an engine cycle that are least efficient and which require improvement.

Rakopoulos [40] developed a simulation model of the actual processes occurring during the thermodynamic cycle of a real spark ignition engine. Hence, comprehensive first and second law analyses of a real spark ignition engine cycle were conducted using an advanced single-zone model. Experimental results from a Ricardo E-6 spark ignition engine, located at the author's laboratory were used to test the validity of the first law analysis simulation results. The second law analysis included a detailed evolution of the availability balance, permitting the revelation of the magnitude of work-potential lost during the various cycle processes in a much more realistic way than the first law analysis can. The second law analysis points to several possible ways for improving cycle performance; they include among others, reduction of availability losses in the combustion process, adiabatic combustion and reduction of availability losses in the exhaust process by using the relevant losses there as the availability source for a bottoming cycle (lower temperature power producing cycle).

Caton [41] studied on the destruction of exergy during combustion processes for an adiabatic, constant volume system. The analysis included the computation of entropy, availability, irreversibilities, and the related energy, entropy and availability balances.

Sengupta et al [42] presented the exergy analysis of a coal based thermal power plant using the design data from a 210 MW thermal power plant under operation in India. The exergy efficiency was calculated using the operating data from the plant at different conditions, i.e. at different loads, different condenser pressures, with and without regenerative heaters and with different settings of the turbine governing.

Ballı et al [43] studied the exergetic performance assessment of a combined heat and power (CHP) system installed in Eskişehir. The performance assessments were made in terms of exergy efficiency, the improvement potential, the inlet exergy depletion rate, the fuel depletion rate, the relative exergy consumption, the productivity lack factor and the ratio of exergy consumption of the components to the capital cost of the CHP system.

2.3 Thermoeconomics

The first proposal in the literature to use second law analysis for costing purposes was a paper of Keenan in 1932 [2]. While he did not do exergy costing therein, he refers to it explicitly as the means for appropriately apportioning cost associated with the cogeneration of electric power and steam for distribution. Engineers thought that “obviously” the fuel cost should be allocated to the steam and the power in proportion to their energy content. The result, however, was the cogenerated electricity cost in this manner, was far less expensive than electricity produced in conventional power plants. Keenan pointed out that the value of the steam and the electricity rests in the “availability” not in their energy.

The interest to formulate the interaction between cost and efficiency was first highlighted by Tribus and Evans at UCLA, in the early 1960’s [44]. They were studying desalination processes by exergy analysis, which led them to the idea of exergy costing and its applications to engineering economics, for which they coined the word “*Thermoeconomics*”. The essence of the Evans-Tribus procedure was to trace the flow of money, fuel cost and operation and amortized capital cost through a plant, associating the utility of each stream with its exergy. El-Sayed joined Evans and Tribus in research and they published in 1970 a frequently cited key paper [45], where the mathematical foundation for thermal system optimization was given.

Also in the 1960s Obert and Gaggioli were working in the optimal design of power plant steam piping. They proposed costing the steam exergy at a value to that of power produced, penalizing irreversibilities for electricity which therefore, will not be produced [46]. Gaggioli directed, in the University of Wisconsin, the Ph.D. theses of Reistad (1970) and Wepfer (1979) on “Second Law Costing” methods that include the definition of rules to provide a rational distribution of the cost [47].

Comprehensive effort to apply thermoeconomics to the analysis, optimization and design of thermal systems did not start until the 1980’s. Gaggioli led the Systems Analysis Technical Committee of the Advanced Energy Systems division (AES) of the American Society of Mechanical Engineers (ASME). The first annual international meeting of AES was taken place in Rome in 1997. In 1990's and 2000's, many important works were performed to achieve a greater standardization and formalism. Many articles were published in order to compare, analyze and unify the different methodologies. Majors of these are given below.

Tsatsaronis et al. [48] presented a new approach to exergy costing in exergoeconomics. In the approach presented, single stream costs were used only for transports of heat and power and for the resources supplied externally to the energy system. All calculations were conducted in terms of monetary (cost) flow rates associated with each exergy stream. The single cost values associated with the exergy unit of material streams were not used throughout the paper. In a detailed and systematic accounting process, authors registered every exergy addition – together with the corresponding cost – and every exergy removal from a stream in each process step. This information moved along with the stream from state to state. They illustrated the concept developed by a simplified gas turbine system. In the system, the combustion chamber was replaced by a heat exchanger in which 2.5 MW of exergy was externally supplied at a cost \$3.0/GJ exergy. This resulted in a cost flow rate of \$27/hr associated with the external energy source.

One of the major differences between the approach they developed and the existing methods of exergoeconomic accounting was in the costing procedure when exergy was removed from a stream. This corresponds to the last-in-first-out (LIFO) principle of accounting. Thus, an exergy unit is removed from a stream at the same cost at which it was previously supplied to the same stream. They applied the approach to the system considering the situation of when physical exergy was split into thermal and mechanical exergy and discussed on the appropriateness of LIFO principle for the analysis of exergy costing than any other accounting principle, for instance, FIFO (first-in-first-out). Since the exergy units that were supplied last to the material stream are used first, the cost associated with the exergy units removed from a material stream in a process step provided that this cost can be computed in the previous steps during which the exergy units removed from the stream were then supplied to it.

The analysis of the approach was performed by considering four different cases. In the case A, the presented exergy approach in this paper was used, in case B, conventional approach was performed. In cases C, the new exergy approach presented by authors was applied to the mechanical and thermal exergy separately whereas in case D, the same approach in case B was used for mechanical and thermal exergy of each stream. Among the four cases, case A showed the highest monetary flow rates and the highest cost of exergy destruction in the regenerator. Case C leaded to the lowest cost of mechanical power and exergy destruction in the compressor and the turbine. Case D had the largest number of extreme values among the four cases. According to these results, the reader can make inferences about that the selection of the exergy-costing method has a stronger effect on the results than the decision to distinguish between mechanical and thermal exergy. Among these cases, case C was the best.

The monetary flow rate associated with the thermal, mechanical, and chemical exergy of a material stream at a given state can be calculated by considering the complete previous history of supplying and removing units of the corresponding exergy form to and from the stream being considered. The benefits of using this presented approach significantly outweighed the increased efforts. The approach, combined with some other developments, made exergoeconomics an objective methodology for analyzing and optimizing energy systems.

Tsatsaronis and Moran [49] showed how exergy-related variables can be used to minimize the cost of a thermal system. These variables include the exergetic efficiency, the rates of exergy destruction, and exergy loss, the cost rates associated with exergy destruction, capital investment, and operating and maintenance, a relative cost difference of unit costs and exergoeconomic factor. They applied the iterative exergy-aided cost minimization method to a simple gas turbine cogeneration system. This system develops a net power output of 30 MW and provides 14 kg/s of saturated vapor at 20 bar. The methodology explained in detail in the paper was applied to the system and the effects of the design variables on the costs were studied. The key design variables (i.e. the decision variables) used in this study were the compressor pressure ratio, the isentropic compressor efficiency, the isentropic turbine efficiency, the temperature of the air entering the combustion chamber, and the temperature of the combustion products entering the gas turbine. The analytical and numerical optimization techniques were applied to this specified structure of the cogeneration system. A significant decrease in the product costs could be achieved through changes in the structure of the system. They pointed out that it was not always practical to develop a mathematical model for every promising design configuration of the system since analytical and numerical optimization techniques could not suggest structural changes that have the potential of improving the cost effectiveness.

Tsatsaronis and Ho-Park [50] discussed how to estimate the avoidable and unavoidable exergy destruction and investment costs associated with compressors, turbines, heat exchangers and combustion chambers. In dealing with the inefficiencies associated with a component, one should recognize that the exergies of all material streams exiting a component are considered either at the product side or at the fuel side. Thus the only exergy loss in a component is associated with the transfer of thermal exergy to the environment. The concept of avoidable exergy destruction and avoidable costs was applied to the cogeneration system by many researchers in the past [49,51]. The results presented in this paper were obtained using the same input data and the same assumptions for its exergetic, economic and exergoeconomic analysis as in ref. [51]. Their general procedure, although based on many subjective decisions, facilitates and improves applications of exergoeconomics.

El-Sayed and Gaggioli [52,53] reviewed the development and state of engineering economic applications of the second law of thermodynamics beginning with a historical review, which is important for better comprehension of second law costing, its objectives, its state, and its prospects. First paper [52] was a brief discussion of the relevant cost accounting concepts and gave general descriptions of the different exergy costing methods which have been in existence. There have been basically three methods for costing exergy: 1) charging for exergy destructions and waste losses at a uniform unit cost throughout a device or plant; 2) using algebraic equations – money balances along with auxiliary equations – to get distinct, average unit costs for each exergy stream; 3) obtaining marginal unit costs for each exergy streams, using calculus. The applications can be summarized as follows: 1) determining the unit costs of the different products from plants generating two or more utilities; 2) using unit costs of exergy transfers between units within plant to make informed operating decisions regarding a single unit within the plant; 3) assessing conceptual designs of a plant in order to discover flow sheet or parameter changes which will improve the economy; and in theory; 4) design optimization. Second paper [53], on the other hand, was devoted primarily to calculus methods. The authors pointed out that algebraic cost accounting cannot be achieved rationally without exergy which is evidence that exergy will help to achieve rigorous optimization more efficiently in practice.

Sciubba [54] presented a systematic approach to the evaluation of energy conversion processes and systems, based on an extended representation of their exergy flow diagram. The method called Extended Exergy Accounting (EEA) was applied to the design optimization of a cogenerative power plant in Italy, using realistic cost estimates. In spite of a long tradition of contrary opinion, exergy seems indeed to possess an intrinsic, very strong and direct correlation with economic values: one of the goals of the extended exergy accounting method was to exploit this correlation to develop a formally complete theory of value based indifferently on an exergetic- or on a monetary metric, that is a general valuing or pricing method in which kJ/kg or kJ/kW are consistently equivalent to $/kg and $/kW respectively. EEA is in principle similar to Life Cycle Analysis. Since, time variations of some of the input parameters cannot be neglected; EEA analysis must span the life of the product or plant. The exergetic equivalents of the capital and labor costs were computed based on global data available for Italy in 1994. The results showed that EEA was indeed a practical instrument for performing design optimization tasks and that its result complemented and extended those obtainable via a thermoeconomic analysis.

Kim et al. [55] proposed a combination of exergetic and economic analysis for complex energy systems and derived a general cost-balance equation which could be applied to any component of a thermal system. In proposed method, the exergy of a material stream was decomposed into thermal, mechanical and chemical exergy flows and an entropy production flow. A unit exergy cost was assigned to each disaggregated exergy in the streams at any state and thus it permitted to obtain a set of equations for the unit cost of the various exergies by applying the cost balance equation to each component of the system and to each junction. This exergy-costing method was considered without flow-stream cost calculations and was applied to a gas-turbine cogeneration system based on a 1000 kW gas turbine with a waste-heat boiler in this paper. In the procedure presented, the primary function or primary exergy product from a specified component was used as a measure of the exergies involved in the component, thus the cost balance equation for the component was normalized by dividing by the exergy of primary product. The methodology of exergy costing provided some information that was related to the actual production process within the system. Application results showed that the unit exergy cost increases as the production process continues and that the production cost of electricity increases nearly proportionality with the input cost.

Valero et al. [56] proposed an optimization strategy for complex thermal systems. For this, they presented a procedure of optimization which used the exergetic cost theory and symbolic exergoeconomics. The structural information obtained at the optimum through the exergetic cost theory was quite important because it allowed us to implement the optimization at a local level. With the costs obtained, a correct assessment could be made of the economic savings of fuel associated with an improvement of the efficiency of a component.

Kwon et al. [57] analyzed the effect of annualized cost of a component on the costs of products in 1000 kW gas turbine cogeneration and CGAM (C. Frangopoulos, G. Tsatsaronis, A. Valero, M. von Spakovsky, first initials) system by using two exergy costing methodologies: SPECO (Specific Cost Method) and MOPSA (Modified Productive Structure Analysis) methods. The CGAM problem refers to a cogeneration plant which delivers 30 MW of electricity and 14 kg/s of saturated steam at 20 bar. The relative deviations in the production costs evaluated from SPECO and MOPSA methods lied between 6 and 11%. The unit costs of electricity and steam for the CGAM system were lower than when SPECO method was used. The electricity cost obtained for the CGAM system by using MOPSA method without heat recovery steam generator unit was close to the value by SPECO method. In the SPECO methodology, the cost formation process of electricity was not changed even though heat recovery steam generator was absent. However, the unit cost of electricity from MOPSA method increased significantly since the escaping thermal exergy from the system drastically increased when heat recovery steam generator was absent. According to the results of the study, it was stated that, the unit cost of products was affected mostly by the annuity of the component whose primary production was the same as the system's product so that one can identify the components which affect the unit cost of system's product decisively.

Tsatsaronis and Ho-Park [58] suggested that, in order to evaluate the thermodynamic performance and cost effectiveness of thermal systems and in order to estimate the potential for improvements, it is always useful to know the avoidable part of exergy destruction and the avoidable investment cost associated with a system component. Therefore improvement efforts should focus only on the avoidable parts. This approach is related the virtual effect of avoidable exergy destruction which gives a realistic picture of the potential for improving the thermodynamic effectiveness of the each component in thermal systems. The calculation of avoidable exergy destruction and avoidable investment costs is associated with arbitrary decisions that reflect the maximal and minimal efficiency that can be achieved for the component being considered in today's technological and economic environment. In the author's opinion, this arbitrariness must be accepted, in order for engineers to improve their understanding of the potential for improvements.

Kanoğlu and Çengel [59] compared the economics of geothermal power generation, heating and cooling considering a typical geothermal resource by using conventional economic analysis. Potential revenues were calculated for the five different uses of the same resource including some cogeneration options and a cost comparison was given. Özgener et al. [60] presented an exergoeconomic study of geothermal district heating systems through mass, energy, exergy and cost accounting analyses and a case study for the Salihli geothermal district heating system in Turkey.

Tsatsaronis [61] presented the definitions of some terms used in exergy analysis and exergy costing, and discussed options for the symbols to be used for exergy and some exergoeconomic variables, and presented the nomenclature for the remaining terms. There is a need for some consensus on the symbols to be used in the area. This study will facilitate both the communication among practitioners and the further development of the disciplines of exergy analysis and exergoeconomics.

Zaleta-Aquilar et al. [62] presented a new method of economic evaluation of the turbine steam path component deterioration. This methodology allowed the computation of exergy cost and economic cost of local products and fuels in each plant component especially in steam turbines. The advantage of the model proposed in this paper was that the cost of malfunctions can be interpreted within a productive structure of fuels, products and losses component by component.

Cardona and Piacentino [63] proposed a new approach for the exergoeconomic analysis which overcomes difficulties encountered when applying the traditional exergoeconomic methodologies to energy systems characterized by continuously varying demand. The new approach proposed was applied to a trigeneration plant supplying a medium-size hospital.

Kwak et al. [64] applied an exergy-costing method to a 500 MW combined cycle plant to estimate the unit costs of electricity produced from gas and steam turbines. The exergoeconomic model, which represented the productive structure of the system considered, was used to visualize the cost formation process and the productive interaction between components. Authors called the cost formation method they used in the study as MOPSA (Modified Productive Structure Analysis). If correct information on the initial investments, salvage values and maintenance costs for each component can be supplied, the unit cost of products can be evaluated. The exergy-costing model shows the productive structure, which represents the interaction among components in the energy system.

Hua et al. [65] proposed a binary subsystem model in order to optimize energy systems. By partitioning a total system into two subsystems, they arrived in their analysis at a binary subsystem exergoeconomic model and the corresponding method of optimization for energy systems. The binary subsystem model proposed reflects the general characteristics of energy-conversion systems and shows the essence of sequential, multi-stage utilization and conversion of energy. The binary subsystem method, based on exergoeconomics and a decomposing-coordinating optimization strategy, may simplify the analysis and optimization of complicated energy systems.

Bonnet et al. [66] studied the coupling of an Ericsson engine with a system of natural gas combustion. The authors carried out classic energy, exergy and exergoeconomic analyses. The study is led with a special attempt to describe as accurately as possible what could be the design and the performance of a real engine. It allows balancing energetic performance and heat exchanger sizes, to plot the exergy Grassmann diagram, and evaluating the cost of the thermal and electric energy production. These simple analyses confirm the interest of such systems for micro-cogeneration purposes.

Lozano and Valero [67] presented the theoretical basis and several applications of the theory of exergetic cost, a major approach to the field of thermoeconomics. The theory of exergetic cost linked thermodynamics and economics. It was based on economic concepts such as resources, structure, efficiency, and purpose. It also used the tools of general systems theory. They considered "*P*" as the production of a process (product) and "*F*" as the consumed resources (fuel), both were being assessed in terms of their exergy. Accordingly, the equation $F - P = I \geq 0$ was satisfied, where $I = T_0 S_g$ (Gouy-Stodola theorem) was the quantification, in terms of exergy destruction, of the irreversibility of the process. Its real thermodynamic efficiency was given by $\eta_{\mathrm{b}} = P / F \leq 1$, a non-dimensional number which, because of the fact that it necessarily lied between 0 and 1, represented a universal ratio for assessing the thermodynamic quality of the processes. The inverse of the efficiency function represented the unit exergetic cost of the product, $k_{\mathrm{p}} = F / P = 1 / \eta_{\mathrm{b}} \geq 1$.

The theory separates the physical and productive structures. In the productive structure, a plant was considered as something more than a set of flows and units. According to this consideration, each unit has a particular productive function which contributes to achieving the final aim of production. Following Tsatsaronis [61], it must be clearly indicated which flow or combination of flows constitutes the product of the unit (*P*), which ones are resources or fuel consumed (*F*), and, finally, which flows are losses (*L*), i.e. those flows that leave the unit and the plant and were not subsequently used in the cost formation process of the theory. The authors defined *F-P-L* definition which best represented the productive function of the units. It was necessary to simultaneously examine the energetic transformation that took place in them. For productive units, the *F-P-L* definition has to meet some certain conditions, which can be summarized as follows: 1) all the flows entering or leaving the unit are presented in the *F-P-L* definition only once, 2) all the components, either individual flows or a combination of flows, of *F*, *P* and *L* have a positive or zero exergy, 3) it is possible to specify the exergy balance corresponding to the unit as a function of the flows in terms of *F-P-L=D*, where *D* is the exergy destruction. Using the *F-P-L* definitions, corresponding matrices can be developed and starting from these matrices and using the data from design and operation of the any thermal power plant, it is possible to carry out the exergetic and energetic analysis of the plant. The unit exergetic costs obtained from the procedure can be used for the optimization of the any thermal plant. The use of the second law of thermodynamics through a systematic use of the exergy concept, the fuel-product concept, based on the productive purpose of a component within an energy system, and the mathematical formulization provided by systems theory are the cornerstones of the theory. This theory was recognized with two Edward F. Obert awards.

Cerqueira and Nebra [68] compared four thermoeconomical methodologies: the Thermoeconomic Functional Approach (it was based on the Lagrangian method of mathematical optimization) proposed by Frangopoulos, the Exergoeconomics (it was one of the two possible variants of the exergoeconomics methodology and only the specific cost concept was taken into account in the study) proposed by Tsatsaronis [48], the Theory of Exergetic Cost and the Disaggregating Method of Valero and Lozano [67]. In order to compare, they applied these methodologies to a gas turbine cogeneration problem. They found significant differences in the costs of power and heat by different methodologies. It was shown that if there is consistent definition of the physical units and their products, and equal treatment is given to the cost of external irreversibility, the Thermoeconomic Functional Approach, the Disaggregating Method and the Theory of Exergetic Cost yield equal results.

Sciubba [69] presented and discussed a new exergy-based quantifier recently proposed on the ground of theoretical and thermodynamic considerations as a general paradigm for the assessment of energy conversion systems. The new quantifier, called Extended Exergy, can well treat environmental externalities that are difficult to allocate properly in thermoeconomics. Extended Exergy concept is based on following assumptions: 1) all activities are aggregated production processes that transform flows of a certain number of "inputs" with respect to space, time and properties by means of additional "inputs" consisting of other materials, energy, labor and capital, 2) each activity can be represented by its transfer function, i.e., a relation between output and input flows, 3) the cumulative exergy content of any products is equal to the sum of the raw exergy of the original constituents plus a properly weighted sum of all additional exergetic inputs into the process, 4) non-exergetic externalities also admit an exergetic formulation, in the sense that they have an exergetic equivalent that can be computed from system + environment balances.

Vieira et al. [70] presented the development and automated implementation of a new approach for the design improvement of complex thermal systems based on the integration of an iterative methodology for exergoeconomic improvement with a professional process simulator. The proposed methodology was algebraic and was easily assimilated and applied by practicing engineers of the industrial community. For comparison and evaluation purposes, the proposed method in the paper was applied to the benchmark CGAM cogeneration system [51].

Uhlenbruck and Lucas [71] combined evolution strategy with a particular exergoeconomic method to yield an optimization technique called Exergoeconomically – Aided Evolution Strategy. This method demonstrated the feasibility of applying in the exergoeconomics method in an automated optimization procedure without any engineer's intervention, contrary to the established interactive exergoeconomic optimization [51].

Kwak et al. [72] investigated the cost structure of the CGAM cogeneration system by using a thermoeconomic method called modified productive structure analysis (MOPSA) proposed by Kim et al. [55]. Even though their scheme of exergy costing [55] was similar to that suggested by Lozano and Valero [67], the method can handle any complex power plants more systematically [64]. On the other hand, the specific cost method (SPECO) proposed by Tsatsaronis and Pisa [48] assigns a cost value to the exergy unit of each material and energy streams entering or leaving components so that the method yield many unknowns and consequently requires other assumptions to estimate the production costs.

Çolpan and Yeşin [73] presented a case study of the thermodynamic and economics applied to an existing gas/steam combined cycle cogeneration plant, i.e., Bilkent combined cycle cogeneration plant. SPECO method was used in the analysis. Ünver and Kılıç [74] investigated environmental temperature and load variation influence on exergetic costs of a combined cycle power plant located in Bursa. The plant consisted of two 700 MW powered twin combined cycle blocks i.e. 1410 MW of total power output at 15°C environment temperature.

Paulus and Tsatsaronis [75] explained the differences between the revenues (i.e. a known price of the product) fundamentally different from a calculated specific cost. They presented principles for writing the proper governing equations. The two well proven principles, F and P principles, for calculating costs were applied to the components of the combined cycle power plant. F principle states that the specific cost (cost per unit exergy) associated with the removal of exergy from a fuel stream must be equal to the average specific cost at which the removed exergy was supplied to the same stream in upstream components. In the P principle, each exergy unit is supplied to any stream associated with the product at the same average cost. Specific revenues discussed in this paper are fundamentally different from specific costs. A specific cost represents how much money must be exchanged for a unit exergy. Specific revenue represents how much money one would be willing to pay, or the maximum amount of money that should be exchanged for a unit exergy. As revenues are fundamentally different than costs, the auxiliary equations used to find these revenues are also different. The principles for generating these equations were found to be a "mirror image" of the accepted equations for costs. However, the fuel, and product must be defined for each component in the same manner as when calculating average costs. It was suggested that when a system has fixed fuel input, specific exergy revenues provide more useful information than costs, provided, there is a market for whatever product is produced.

2.4 Thermoeconomic Optimization

Thermoeconomic optimization studies started in 1970's with the papers of El-Sayed and Evans [45] and Szargut [76]. These two were the pioneer works for exergy-based cost optimization studies in the literature. In 1990's, many studies were published with the progressive development of analytical and numerical optimization techniques. In this section major studies of the thermoeconomical optimization in the literature are given with important optimal cost considerations.

Frangopoulos [77] formulated the optimization problem of a combined cycle plant and solved it with two thermodynamic objective functions which were maximization of the system efficiency and the maximization of the net power density and a thermoeconomic one which was minimization of the total cost of owning and operating the system. In the presented problem, the thermoeconomic optimum solution was strongly dependent on the cost functions and coefficients.

Tsatsaronis et al. [78] decided to compare the methodologies of a group of concerned specialists in the field (C. Frangopoulos, G. Tsatsaronis, A. Valero, M. von Spakovsky) by solving a predefined and simple problem of optimization: the CGAM problem, which was named after the first initials of the participating investigators. The CGAM problem they defined referred to a cogeneration plant which delivered 30 MW of electricity and 14 kg/s of saturated steam at 20 bar. The installation consisted of a gas turbine followed by an air preheater that used part of the thermal energy of the gases leaving the turbine, and a heat recovery steam generator in which the required steam was produced. The objective of the CGAM problem was to show how the methodologies were applied, what concepts were used, and what numbers were obtained in a simple and specific problem. In the final analysis, the aim of the CGAM problem was the unification of thermoeconomic methodologies; it was not a competition among methodologies.

Tsatsaronis and Pisa [79] discussed and applied various exergy costing approaches, the exergoeconomic variables, and the procedures used in evaluating and optimizing energy systems on the predefined CGAM problem. The methodology followed in this paper included: the detailed exergy analysis at the plant component level, calculation of the capital costs associated with each plant component, an exergoeconomic analysis using an exergy-based costing method as detailed and objective as possible, and evaluation of the effects of decision variables on selected exergoeconomic variables. This extensive paper provided a powerful and systematic tool for identifying all cost sources and for optimizing the design of complex energy systems.

Frangopoulos [80] applied several methods of analysis and optimization to the same predefined CGAM problem. These methods were direct use of a nonlinear programming algorithm, thermoeconomic functional approach and modular simulation and optimization of the system. All three methods reached the same value of the objective function. The differences in the values of the independent variables were negligible (0.02-0.08%); they were due to numerical approximations and to the fact that the objective function was not very sensitive to the independent variables in the close vicinity of the optimum point. According to the results of the sensitivity analysis of the optimum solution, the change of the optimum values of the independent variables due to an increase of the capital cost by 100% was of the same order of magnitude as the change due to an increase of the fuel price by 100%, but of opposite sign. The direct use of an (usually nonlinear) optimization algorithm is the simplest way, because it requires the least effort in system analysis. As a consequence, it gives no information about the internal economy of the system and physical and economic interrelationships among the components. The direct approach, as given and explained in detail in this problem, may not be applicable for the solution of the complete optimization problem of very complex systems. In the solution of these types of problems, Thermoeconomic Functional Analysis can be used effectively and it is better revealed in more complicated situations when the solution of optimization problem is required. The modular approach which was used in the study as a third optimization method offers significant advantages, when detailed design of components is required and complicated simulation models are used.

Silveira and Tuna [81,82] presented a thermoeconomic analysis of cogeneration plants, applied as a rational technique to produce electric power and saturated steam. The aim of that methodology was the minimum Exergetic Production Cost (EPC), based on the second law of thermodynamics. The advantage of presented method was its lowest computational time because it was a direct algebraic method, easy to handle and change its parameters. The present theoretical method in the first paper was applied to four cogeneration systems of a chemical plant in the second paper. The demand of the plant was 6000 kW of electricity and saturated steam distributed in pressures of 0.25, 0.6, and 1.5 MPa, with flow rates of 0.278, 4.167, and 1.389 kg/s, respectively. The first cogeneration case (case 1) consisted of a back pressure steam turbine with two extractions. Case 2 consisted also of a steam turbine, but a condensation one, with three extractions, supplying all the thermal and electrical needs of the plant. Case 3 consisted of a gas turbine followed by an air pre-heater and heat-recovery steam generator in which the required steam was produced with supplementary firing. Case 4 was equal to case 3 but without supplementary firing. At the end of the numerical analysis, it was determined that, the production costs in exergy basis and in energy basis were the same. This was because of the equality of the purchase and operational costs. The difference appeared in the specific costs of steam and electricity.

Frangopoulos [83] described The Intelligent Functional Approach (IFA) in its general as well as special forms for analysis and optimization of complex systems such as thermal and chemical plants. In this approach, the distribution of functions established interrelations between units or between the system and the environment, which were depicted in the Functional Diagram of the system. The optimization was considered at three levels: synthesis, design and operation. The method can work on any one of these levels in separate or on all three levels simultaneously in order to determine the overall optimum. Frangopoulos [84] applied Intelligent Functional Approach for thermoeconomic optimization of the synthesis, design and operation of a cogeneration system. By rational use of the values of economic indicators (Lagrange multipliers), optimization procedures are much simpler than the general methods could be developed. Thus, time and means are left available, which make it possible to include more variables into the problem and to investigate the effect of more parameters on the optimal solution.

Frangopoulos [85] presented the general formulation and a numerical example for the optimal design or improvement of complex thermal systems by using thermoeconomic functional analysis before proposing the application of the Intelligent Functional Approach for the thermoeconomic optimization problem. He also applied the method to a simplified thermal power plant which was considered to be made up of four units: boiler, turbine-generator, condenser and condensing supply works (e.g. cooling towers, cooling-water-circulating pumps etc.), and boiler feed pump. The author emphasized that the number of units in a system considered was not unique and depended on the available information and desired results. Thus, the designer may select high or low resolution depending on the objectives. He also assumed that the plant produces only electrical power at a specified rate. For the solution of the optimization problem, the method of Lagrange multipliers was chosen, which leads readily to the special cases of decomposition and thermoeconomic isolation.

Manolas et al. [86] applied the genetic algorithms for the operation optimization of a cogeneration system. The capability of genetic algorithms to handle objective functions of any complexity with both discrete and continuous variables, as well as any type of constraint makes them good candidate for the solutions of these types of problems. The procedure they described was applied to a cogeneration system which was the main supplier of an industrial complex with electricity and steam. The system consisted of one gas-turbine generator with a capacity of 99.2 MW, one exhaust-gas boiler of 30.56 kg/s with supplementary firing of gas/oil, four boilers at different technical characteristics and five back-pressure steam-turbine generators with high- and low-pressure stages of the specified characteristics. The objective of the optimization procedure was the maximization of the total electrical power generated for given steam requirements at each pressure levels (i.e. 74.5, 43, 21, and 4.8 bar). The load of each turbine was set empirically, with no optimality criteria. Since the optimization of objective of the genetic algorithm was the maximization of electric power produced and this power had a value, the optimization of the objective should have been the maximization of revenue from the sale of power produced. The calculation of value of power produced, for each combination of values for the independent variables, was performed by the cogeneration system simulation routine developed and this value was multiplied by the specified average price of energy sold, in order to find objective function or fitness. The application results in the paper proved that genetic algorithms can be applied successfully to such types of optimization problems.

Von Spakovsky and Evans [87] discussed a general analytical approach which directly determine the optimum thermodynamic and economic behavior of thermal systems and illustrated a numerical example by using Rankine cycles. The optimum solution vectors were found for the thermoeconomic model of the Rankine reheat cycle with regeneration and the functional diagram of the corresponding cycle.

Because of the difficulties arising from the use of Newton's method in conjunction with numerical differentiation, a certain degree of uncertainty existed in the optimum values found. It was emphasized in the paper that, the thermoeconomic basis model could be created not only for various Rankine cycle configurations but, more generally, for any type of thermal system. Such models, once optimized, can allow the performance and design engineer to know under what set of economic and thermodynamic conditions a particular cycle should operate.

Ionita [88] presented the concept of cost/quality ratio as an optimization criterion for the production of energy, goods and services. All the ideas presented in the paper were based on the general concepts of extended exergy analysis and of thermoeconomics, developed during the last twenty years by several authors in literature.

Von Spakovsky [89] submitted an outline of how to apply a thermoeconomic methodology called Engineering Functional Analysis (EFA) to the optimization and analysis of an energy system. By using EFA, the thermoeconomic structure of an energy system could, as it has been for the CGAM problem, be constructed. Based on EFA, a component function can be defined as having one and only one primary product, which means that it may or may not have a direct correspondence to one of the physical components in the system. As an example, the gas turbine's primary product is the power it produces. Any secondary products, such as, for example, the exhaust gas exiting the turbine, can be handled by means of a branch and branches that have no physical counterpart. The commodities of value which were chosen determine the major loops or interconnections which exist between component functions and, thus, complete the thermoeconomic structure of the system. Having defined this complete structure for the CGAM problem, the next step was to develop its mathematical equivalent. Each of the general mathematical functions was expressed in terms of the decision variables in the paper. These equations were standard definitions for these types of components such as gas turbine, air-preheater, compressor etc. With these equations, the objective function and all corresponding constraints were defined and internal economy of the system was determined by using Lagrange's Method of Undetermined Multipliers.

Vieira et al. [90] proposed an integrated approach for exergoeconomic optimization of complex thermal systems, which coupled a well-known and simple mathematical optimization algorithm to a professional process simulator. The selected mathematical algorithm was the Flexible Polyhedron Method. Integration of the mathematical method with the simulator permitted a two-orders-of magnitude reduction in the number of variables dealt with by the optimization routine while a conventional mathematical optimization approach applied would require the manipulation of more than 800 variables. The application of the proposed integrated optimization approach to a complex cogeneration system (i.e. including more than 800 variables) was carried out in the study. This system consisted of twenty four components. The cogeneration system products were from the gas and steam turbines, the process steam and the process hot water, and the fuel for gas turbines was natural gas. In the cost calculations of paper, the component cost balance equations and the component cost partition equations were organized as in the Theory of the Exergetic Cost [64]. The product, respectively the fuel, of a non-dissipative component consists of the sum of all exergy flow rate values at the outlet which were not associated with material streams. For dissipative components, authors proposed to identify the component fuel and product with entering and exiting flow rates, respectively. Also component cost partitioning follows the two rules stated in ref. [67] one for fuel and one for product. The economic analysis performed in this study was a simplification of the Revenue Requirement Method. To perform the optimization of the cogeneration system, it was assumed that the demand for process steam and hot water were kept constant. As a result of the presented analysis, it was observed that the objective function is almost insensitive to some decision variables. Therefore, to improve the efficiency of the optimization process, the procedure to be applied should be based also on exergoeconomics such that only the decision variables that significantly affect the objective function to be considered.

Bejan [91] presented the fundamentals of the methods of exergy analysis and entropy generation minimization. After reviewing of the some basic concepts, some examples were illustrated beginning from energy storage systems for sensible heat and latent heat through solar energy, and the generation of maximum power in a power plant model with finite heat transfer surface inventory. After analyzing the systems carefully, it was shown that, the physical structure of the system such as geometric configuration and topology, springs out of the process of global thermodynamic optimization subject to global constraints. This generated a principle for the structures not only in engineering but also in physics and biological systems (i.e. constructal theory).

Li et al. [92] analyzed the thermal performance, economics and environment factors of the proposed integrated energy system based on unit performance correlations derived from commercial market. A simultaneous consideration of the thermodynamic, economic and emission criteria regarding both CO_2 and NO_x emissions of a distributed combined heating, cooling and power generation (tri-generation) system in an urban residential area was realized through thermoeconomic optimization based on external cost (e.g. emission taxes) internalization. Since the units of performance functions were non-linear and the components of the system were chosen freely under given load and other physical constraints the optimization was mixed integer and non-linear programming problem and it was difficult to be solved by conventional optimization approach. Thus thermoeconomic optimization was performed by using a genetic algorithm toolbox. As the objective function, system Net Present Value (NPV) was maximized through the optimization process. The sensitivity of the plant optimal configuration and design under different specific economic and environmental legislation conditions was examined by choosing different scenarios, which were defined according to different emission tax rates. The optimization results showed that under the current valid case applied, the plant investment cost and the assumption of avoided grid line construction fee and the electricity capacity expansion fee due to on site energy generation were the key factors to determine optimal plant configuration. With increasing the emission tax rate, the weight of power generation unit capacity and rated electric efficiency became dominant. On the other hand, unit partial load performance which determined the plant economical operation played an important role in the optimal plant configuration determination.

Ceylan and Öztürk [93] developed a genetic algorithm energy demand model for estimating Turkey's future energy demand based on gross national product, population, import and export figures. The genetic algorithm was used to estimate future energy demands under different approaches by appropriately estimating the weighting the parameters with current data.

Frangopoulos and Dimopoulos [94] introduced reliability and availability in to the thermoeconomic model of the system, by doing so, redundancy can be embedded in the optimal solution, and in addition more realistic results can be obtained for the cost and profit, if any. They formulated the optimization problem in two levels as synthesis and design, and operation under time-varying conditions. For the solution of the problem with no failure, the genetic algorithm approach coupled with deterministic one was used. In the case of partial failure, problem was solved by the intelligent functional approach. The methodology, they presented is not restricted to cogeneration systems.

Erdil [95] carried out an exergy optimization for an irreversible combined cogeneration cycle. The cogeneration cycle model was assumed to operate between three heat reservoirs of temperatures T_H, T_L, and T_K. The temperatures of the working fluids exchanging heat with the reservoirs at T_H, T_L, and T_K were T_X, T_Z, and T_U. The exergetic performance of the system rapidly decreased with increasing power to heat ratio, *R*, and than it remained approximately constant. The optimal exergetic performance interval did not vary with *R*. It was also observed that the irreversibility parameter was more effective on the exergy interval than the exergy efficiency interval. In this study, it was shown that the global and optimal exergetic performances gradually decrease as internal irreversibilities increase. He also defined the optimal design regions by considering the exergy output and exergy efficiency of the combined cogeneration system together.

Öztürk et al. [96] performed a thermoeconomic optimization of a steam injected gas turbine system with and without a supplementary combustion chamber. The advantages of combined heat and power (CHP) plants were illustrated by comparing three different types of power stations: a condensing power plant using seawater as coolant, a back pressure combined heat and power plant using water from DH as coolant and an extraction CHP power plant using two alternative coolants: seawater and DH water. The second law effectiveness values for the cases with and without a supplementary combustion chamber were calculated as 37.3 and 41.5%, respectively. Lagrange multipliers were used to obtain the optimum values of the unit exergy costs, and it was used to compromise between economic and thermodynamic analysis based on incorporated exergy-based production costs with economic evaluations. The thermodynamic optimization was carried out by means of the minimization of exergy losses, taking into account only the irreversibility due to the temperature difference between hot and the cold stream.

Mazur [97] pointed out that the main idea of the conventional thermoeconomic analysis is an introduction of exergetic or exergo-environmental costs to normalize monetary and energy units and emphasized that the weak point of this conventional analysis is an implicit assumption regarding the concordance of economic and energetic interests contrary to the real situation. In order to fill the gap which was created by conventional analysis, he tried to introduce the present work which includes an uncertainty in thermoeconomic analysis to find solutions that simultaneously satisfy thermodynamic and economic goals. The combination of Pareto optimality and fuzzy sets concepts can allow the decision makers to conduct a comprehensive study of the energy- transforming systems, considering various combinations of economic goals and thermodynamic constraints. Mazur considered an extended thermoeconomic model of cogeneration plant developed by Toffollo and Lazzaretto [98] with two criteria which need to be minimized: the thermodynamic criterion as a deviation of the exergetic efficiency of the cogeneration plant from an ideal value and the economic criterion as the total cost rate of operation. Lack of concordance for alternative performance criteria raised doubts about the effectiveness of conventional thermoeconomic optimization algorithms. Fuzzy thermoeconomic optimization of a cogeneration plant obviated this difficulty for the decision maker. This study was one of the first attempts to apply the methodology of multicriteria decision making to select the trade-off of working fluids in engineering practice.

Benelmir and Feidt [99] presented the analyses of three existing cogeneration systems which were installed in a public hospital, a process plant, and a paper production plant, by using fundamental techno-economic studies. In the public hospital, the cogeneration system had a production rate of 2.8 MW of electricity and 2.73 MW of heat. In this system, there was a heat engine consuming 720 m^3/h of natural gas. Since the cogeneration system was operated for more than 4704 hours, the amount of heat produced was about 12.8 GWh per year. The electrical, heat and overall efficiencies were about 39%, 38% and 77%, respectively. In the process plant case, the system consisted of a gas turbine, operated with 430 m^3/h natural gas, producing 1.1 MW of electricity, followed by a post-combustion chamber, fed also with natural gas, used as a dryer for a particular chemical product. The electrical, heat and overall efficiencies were about 25.6%, 50% and 75.6%, respectively. In the paper production plant, the system consisted of a gas turbine followed by a post-combustion burner in order to generate steam used for process and to run a steam turbine. The electrical power output of the gas turbine and steam turbine were about 4.5 MW and 1.3 MW respectively. The relative equivalent power of the electricity and process steam were about 10.5 MW and 7 MW, respectively. The electrical efficiencies of the gas turbine and steam turbine were about 30% and 12%, respectively, and the combined gas/steam turbine electrical efficiency was about 37.4%. The overall cogeneration efficiency was about 82.5%. They used the principles of exergy analysis and cost-objective function optimization by focusing on a global approach. This type of analysis is rather approximate but realistic.

Yılmaz [100] conducted a performance analysis based on alternative performance criteria for a reversible Carnot cycle, modified for cogeneration, with external irreversibilities. He defined a model of a reversible Carnot cycle for a cogeneration system. In his model, the external irreversibilities of heat transfer were considered, and other irreversibilities were neglected. It was observed that power to heat ratio $R = 1$ was a critical value for the optimal artificial thermal efficiency.

Von Spakovsky and Evans [101] discussed the detailed thermoeconomic models used in stable economic environment around each function or component in a system. They pointed out that such an environment may allow each component to be optimized in greater detail, isolated from the system, while at the same time maintaining an optimum design for the system as a whole. They also discussed the concept called thermoeconomic isolation in the sense of economically isolating a thermodynamic function from the other functions in a thermal system, i.e., establishing "stable economic environments".

Chejne and Restrepo [102] presented a new approach for thermoeconomical optimization. In this approach, the marginal cost of any stream is linked to the exergy losses and to the exergy itself. Thus, the marginal cost of the flows that leave certain subsystem may be lower than the cost of the flows that enter the subsystem. The authors pointed out that the rules for the exergoeconomic optimization methodology of multiproduct – multicomponent systems developed by Valero and co-workers become insufficient when the system complexity increases. This situation made them to try to modify the rules and other criteria, including the loss of the source's value due to the irreversibility instead of assuming that the unit cost of any stream that enters a subsystem is equal to the cost of the stream that leaves the subsystem. The methodology proposed was applied to CGAM problem [78], and this approach yielded a specific cost for the energy output that was higher than the old and well known approach by Valero and Lozano [67]. On the other hand, for the energy recovery presented in the steam, they obtained a value lower than that presented by Valero and Lozano and also this value was the lowest among the streams.

Valdes et al. [103] tried to find the better optimization strategy to achieve a thermoeconomic optimization of combined cycle gas turbine power plants and used the genetic algorithm optimization tool. For this aim, they proposed two different objective functions: one minimizes the cost of production per unit of output and the other maximizes the annual cash flow and compared the results. In order to cover a wide variety of alternatives, the thermoeconomic optimization model was applied to the following configurations: 1. single pressure combined cycle gas turbine power plant which has been the simplest configuration of all and it was used to fit the genetic algorithm optimization method, 2. dual pressure combined cycle gas turbine power plant with and without reheating, 3. triple pressure combined cycle gas turbine power plant with reheating. These last two configurations are being installed worldwide currently. According to their analysis, the single pressure combined cycle gas turbine power plant was the worst in terms of costs and cash flow and had the lowest efficiency value because energy was poorly used with this configuration. The optimization algorithm was selected an optimum design for the triple pressure combined cycle gas turbine power plant with lower efficiency than the corresponding dual pressure. This meant that a search for a higher efficiency in this configuration would increase the fixed costs too much for the particular conditions selected in the study presented.

Hammond and Akwe [104] used thermodynamic and related exergoeconomic criteria in order to analyze natural gas combined cycle power generation systems with and without carbondioxide removal technologies. Carbon capture was simulated on the basis of CO_2 recovery from the flue gas stream that leaves the heat recovery steam generator via a commercial amine process. Particular attention in this study was paid to carbon sequestration, amine sorbent regeneration heat requirement, natural gas prices, and capacity factors.

Bejan et al. [105] extended constructal theory into the realm of economics and showed that by minimizing the cost in point – to – area or area – to – point transport, it is possible to anticipate the formation and growth of dendritic routes over a growing territory. They also showed that by maximizing the revenue in transactions between a point and area, it is possible to derive not only the dendritic pattern of routes and their interactions, but also the optimal size of the smallest (elemental) interstitial area. The main conclusion of the paper maybe summarized as follows: Since the analytical structure of classical thermodynamics has been used in the past to construct analogies and models in economic theory, this well-organized theory can also be used not only as new leads toward predictive models in economics, but also as a more general theory that unites economic ideas with laws which are well known to govern other physical and biological process.

Rodriguez-Toral et al. [106] presented the development and testing of an equation-oriented mathematical model for the optimization of heat and power systems using a new state-of-the-art sequential quadratic programming. They extended this model by including combined cycle cogeneration plants, the economic optimization of which involved adding equipment investment, cost functions and operating cost models to the stream and unit operation models and gave some details of the modeling extensions along with a number of complex simulations and economic optimization examples of a real cogeneration plant.

2.5 Cogeneration

During the 19th century, when steam engines primarily were reciprocating engines exhausting to atmosphere, the commercial possibilities of steam became evident. In the 1960's, as it sought competitive advantage in the market place, the fuel-energy industry observed that the technology in internal combustion power systems had advanced to the point that electric power could be generated at a building site and that rejected could be utilized for heating and cooling [107]. During 1970's with the dynamics of fuel costs, and in many cases, shortages of gas and liquid gases, the economics of total energy took a turn for the worse. In the 1980's and 1990's, scientists and engineers started to look for more efficient ways to utilize energy from cogeneration facilities [108]. Nowadays, cogeneration is considered as one of the most important techniques for achieving a more efficient usage of fuels, natural and financial resources savings and environmental protection. In the following, major works on cogeneration in the open literature are given.

Yodovard et al. [109] assessed the potential of waste heat thermoelectric power generation for diesel cycle and gas turbine cogeneration in the manufacturing industrial sector in Thailand. In their paper, the potential of waste heat thermoelectric power generators is analyzed using an annual cost method based on stack exhaust from a cogeneration system for different operation hours system life spans, bank interest rates, system prices, maintenance costs, depreciation, internal rates of return, and electricity buy back rates sold to the grid line.

McKay and Rabl [110] presented a paper which was an abbreviated version of a longer report of a feasibility study of the installation of a new central energy supply system in Princeton University, USA. In order to evaluate reasonable alternatives for reducing energy costs, the most promising candidate systems such as a gas turbine cogeneration system and a coal-fired fluidized bed boiler with a high temperature steam cogeneration system were studied. A diesel cogeneration system was also investigated but it turned out to be less attractive in that particular application because much of the cogenerated heat had low temperature. The savings were critically dependent on the economic scenario, in particular the escalation rates for energy prices.

Feng et al. [111] proposed a new thermodynamic performance criterion, the cogeneration efficiency, based on the analysis of various existing criteria. The criterion determines the cost allocation of heat and power in cogeneration systems in addition to assessing the energy utilization effect of systems.

Renedo et al. [112] analyzed different possibilities for providing heating, air-conditioning and hot tap water to a hospital center. For this, several cogeneration systems with diesel engines and gas turbines were considered. It was shown that the size of the facility and the control strategy have a strong influence on the system economy, showing that the most important parameter is the electricity produced.

Hung et al. [113] applied a mathematical model for the purpose of waste heat recovery from an internal combustion engine power plant in seawater desalination and analyzed via Microsoft Excel and Visual Basic. The study showed the feasibility of application of waste heat from combustion engines in the desalination of seawater.

Costa and Balestieri [114] compared the different configurations for the cogeneration system to be proposed for a chemical plant and studied the feasibility of incorporating ammonia / water absorption chillers in comparison to the compression system currently in operation. Economic results taken from this study showed that the reciprocating engine associated to compression refrigerating systems is more attractive when compared to gas turbine and combined type cogeneration systems.

Coelho et al. [115] illustrated the importance of process integration principles and emission concerns in the development of a cogeneration project in a chemical plant located in Portugal. The cogeneration project in this study was based on a diesel cycle engine burning heavy fuel oil, driving an alternator with an exhaust gas heat recovery boiler supplementary fired with either heavy fuel oil or natural gas. A sensitivity analysis was performed for a typical system of such type and size to evaluate the system efficiency against process heat production.

Silveira et al. [116] presented a method and its application for studying the technical and economic feasibility of a cogeneration system utilizing an internal combustion engine. This system produces electrical power and cold air for a university building in Brazil. Cogeneration concepts applied to compact systems as analyzed in this paper, may present some opportunities the Brazilian energy scenario. In third World countries such Brazil, it is not reasonable to assume private capital investment with interest rates less than 15% per year. This restricts the economic feasibility exclusively to the system powered with natural gas and operating in the regime of electric parity.

Bidini et al. [117] analyzed the performance and economic characteristics of an existing combined heat and power plant with an internal combustion engine and district heating for a university building in Italy. Results of data taken were discussed with reference to daily performance of the combined heat and power plant which showed how electric efficiency was only slightly affected by ambient temperature. A comparison of evaluation indexes for cogeneration plants was made with particular attention to the energy index used by Italian legislation as an evaluation parameter to decide whether a combined heat and power plant can have access to financial benefits.

Chicco [118] illustrated and evaluated the possible benefits of adopting different trigeneration alternatives in today's energy system planning. The study was specifically focused on discussing various alternatives for supplying the cooling load. The concepts discussed in the paper, showed that planning trigeneration systems was a challenging problem. For the case studied, the electricity fed machines under the h-partial (which is heating load following with combined heat and power unit – 3202 hours of operation period) regulation strategy resulted to be the most profitable options. Yet, the author pointed out that results obtained from this study could not be generalized to every type of trigeneration system.

Silveira and Gomes [119] presented in their paper some technical information about the most difunding types of the fuel cell demonstration systems in the world and the energetic and economic analysis of a molten carbonate fuel cell in the cogeneration version as applied in the computer center building. The fuel cell powered cogeneration system are preferred in U.S.A, Japan and some countries of European community since they have high efficiency and present low levels of pollutant emissions and may present power capacities from 10 kW to up to 50 MW. The results showed that there is technical and economic feasibility for fuel cell systems investment values between 1000 and 1500 US$/kW for payback values between 3 and 6 years.

Takahashi and Ishizaka [120] constructed a mathematical model employing the concept of information theory to investigate the relationship between cogeneration systems. This model provides an indication of the relative importances of demand indices, and identifies what may become a good measure for assessing the efficiency of the cogeneration system for planning purposes. In their paper, authors built quantitative criteria of relative importance for each energy demand component using the actual energy demand data.

Lucas [121] demonstrated that the saving of primary energy depends sensitively on the parameters of the technology, and in particular, on the match between the power/heat ratio of the cogeneration equipment and the demand. Also, a rational allocation of primary energy and in turn of emissions to the coupled energy forms produced can be derived by a consideration of the exergy losses in the various types of equipment.

Tsay and Lin [122] presented an Evolutionary Programming procedure to minimize the overall cost of the cogeneration system while meeting the requirements of generated steam capacity and electrical power. Developed model considers the connection of the cogeneration system with the utility company in terms of time-of-use rate and various fuel consumptions and also could produce multi-solutions to achieve the real global or nearly global solutions.

Ortiz et al. [123] analyzed a heat-driven metal hydride cogeneration cycle using classical first and second law analyses as well as finite time thermodynamics. The system analyzed was based on an existing cycle and predicted performance was compared to data from two other actual hydride power systems.

Silveira et al. [124] presented a relative fuel cell concept, followed by chemical and technical information on the change of Gibbs' free energy in isothermal fuel oxidation directly into electricity. Fuel cell cogeneration systems performs direct conversion of the chemical energy of the oxidation of hydrogen from fuel with atmospheric oxygen into direct current electricity and waste heat via an electrochemical process relying on the use of different electrolytes. In this sense, they analyzed a direct internal reforming molten carbonate fuel cell associated with an absorption refrigeration system to produce electric power and cold water for a building.

Kato [125] proposed a new conceptual reactor named "low exergy nuclear reactor" and discussed its possibility qualitatively. The necessity of the proposed reactor was discussed based on the present energy technology trend, and cogeneration systems based low exergy nuclear reactor model include absorption heat pump, thermoelectric device, and steam turbine for decentralized energy utilization. Also, thermal efficiency of the proposed reactor was compared with that of conventional light water reactors.

Zheng and Furimsky [126] developed a simulation model for the combined cycle cogeneration plant fuelled by natural gas by using the advanced system for process engineering (ASPEN) software. The results generated were compared with the operational data of the actual commercial plant, and it was showed that the key results generated by this model were in good agreement with the operating data. This made the developed model suitable for simulation of commercial cogeneration plants.

Szklo and Tolmasquim [127] assessed the economic feasibility of a gas-fired cogeneration systems in Brazil and indicated that the use of cogeneration in Brazilian's malls tends to be limited in short term. In their second paper [128] they supplemented its predecessor by analyzing a type of cogeneration rated as strategic by considering possibility of utility in the energy market. The authors analyzed the impact of changes in the regulatory context on the development of cogeneration in Brazil in another study [129].

Fang et al. [130] developed a new system for clean and highly efficient utilization of coal. This new coal-based, multi-product, cogeneration system which combines coal pyrolysis, gasification and combustion to supply town gas, process heat and electricity was analyzed. The system had low emissions and low cost. A demonstration system was constructed in a region of China. Operating results showed that the system had wide fuel adaptability, allowed more than 25% turndown, and had 88% thermal efficiency.

2.6 Conclusions

The extended overview provided in this chapter indicates that there are a large number studies on thermoeconomic analysis and optimization of cogeneration systems in literature. A good fraction of these studies consider CGAM problem when evaluating and comparing different methods. Some studies consider conceptual cogeneration designs with alternative systems and assumed operating conditions. This thesis differs from the previously conducted studies as follows:

a) The presented study is on the exergetic and thermoeconomic performance analysis and optimization of diesel engine powered cogeneration systems. The thesis is original in this scope and content and there is no such study in the open literature, to the best of the author's knowledge and it is the main motivation behind this study.

b) In literature, a small number of studies consider diesel engine powered cogeneration systems and the analysis in these studies are mostly limited to conventional energy analysis and economic considerations.

c) The thesis provides theoretical foundation including procedure and formulation for conventional energy and economic analysis and exergoeconomic analysis and optimization of diesel cogeneration as well as applications on an actual system.

CHAPTER 3

ENERGY USE IN TURKEY

3.1 Introduction

Known energy sources in the earth have been exhausted rapidly over the years. Efficient end effective utilization of energy sources has become important. Turkey's geographical location makes it a natural land bridge connecting Europe to Asia and Middle East. Therefore, it can play a role as an "energy corridor" between the major oil and natural gas producing countries in the Middle East and Caspian Sea and the Western energy markets [131].

Turkey is located between Europe and Asia and its area is 781,000 km^2. Average population density is nearly 80 persons per square kilometer. The annual population growth rate is 1.7, the highest among the IEA (International Energy Agency) countries [132]. As Turkey has improved its economic situation in recent years, the energy demand has increased. There is a need to control atmospheric greenhouse emissions and other pollutants. The increase in the demand rate for energy is 8% per year and it requires installing about 3500 MW of energy generation capacity systems per year. Private and public financial sources must be evaluated to meet this demand [133].

Economic growth of Turkey in recent years has been associated with the privatization of public enterprises. The macro economic performance was boosted by growth in energy sector [134]. In Turkey, electricity is produced by thermal power plants, by consuming coal, lignite, natural gas, fuel oil, and geothermal energy, wind energy, and hydropower plants.

In this chapter, we present an overview of Turkey's energy use. The content of this chapter includes world's energy situation, Turkey's general energy outlook, energy sources in Turkey, electricity market and private power generation, and environmental aspects of Turkey's energy use. The role of diesel powered electricity generation and cogeneration applications are discussed as part of the Turkey's overall energy picture. It is shown that diesel engine cogeneration is an effective model for private companies willing to produce their own electricity while meeting part of their heat need.

3.2 World's Energy Situation

The mix of primary fuels used to generate electricity has changed a great deal over the past three decades on a world basis. Coal has remained the dominant fuel, although electricity generation from nuclear power increased rapidly from the 1970s through the 1980s, and natural gas fired generation have grown rapidly in the 1980s and 1990s. In contrast, due to the OPEC oil embargo of 1973-1974 and the Iranian revolution of 1979, the use of oil for electricity generation has been slowing since mid-1970s [135].

Continued increases in the use of natural gas for electricity generation are expected worldwide. Coal is projected to continue to retain the largest market share of electricity generation, but its importance is expected to be diminished somewhat by the rise in natural gas use. The role of nuclear power in the world's electricity markets is projected to lessen as reactors in industrialized nations reach the end of their life spans and fewer new reactors are expected to replace them. Generation from hydropower and other renewable energy sources are projected to grow by 56% over the next 24 years. The percentage changes of fuel shares of electricity generation for the years 1973 and 2001 are shown in Table 3.1 and Figure 3.1. Coal is largely used for electricity generation as seen from the table and figure.

According to the electricity production figures in the world, the first three leading countries are US with 3864 TWh, China with 1472 TWh and Japan with 1033 TWh. These three countries hold 41.2% of the total production in the world. Electricity productions of the leading countries are given in Table 3.2.

Table 3.1 Fuel shares of world electricity generation in 1973 and 2001 [135]

Year	Hydro %	Coal %	Oil %	Gas %	Nuclear %	Others[a] %	Total % (TWh)
1973	21	38.3	24.7	12.1	3.3	0.6	100 (6117)
2001	16.6	38.7	7.5	18.3	17.1	1.8	100 (15,476)

[a] Geothermal, solar, wind, renewable and waste

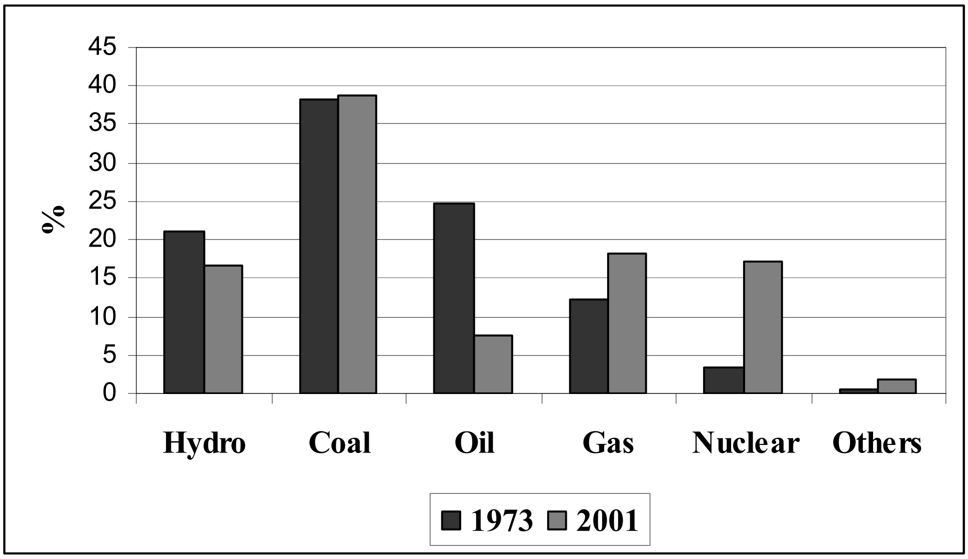

Figure 3.1 Fuel shares of world electricity generation in 1973 and 2001 [135]

Table 3.2 Producers of electricity in the world (2001) [135]

Producers	Production amounts (TWh)	%
US	3864	25
China	1472	9.5
Japan	1033	6.7
Russia	889	5.7
Canada	588	3.8
Germany	580	3.7
India	577	3.7
France	546	3.5
UK	383	2.5
Brazil	328	2.1
Rest of the World	5216	33.8
World Total	**15,476**	**100**

It is known that hydro is the cheapest source of electricity production. Countries, which produce electricity using this method, are given in Table 3.3. Canada is leading country in hydropower production with 333 TWh. China and Brazil follow with 277 and 268 TWh, respectively. The US has the biggest production capacity in the world with 98 GW. Other countries production amounts are more or less parallel to their capacities. Coal has been used widely in electricity generation for decades. US is the biggest generator of all with 1983 TWh. China is second with 1122 TWh. The generation shares of these two countries in the world are as large as 52%. The biggest 10 electricity producers by using coal are shown in Table 3.4.

Table 3.3 Producers of hydropower in the world (2001) [135]

Producers	**Production amounts (TWh)**	**%**
Canada	333	12.6
China	277	10.5
Brazil	268	10.1
US	223	8.4
Russia	176	6.7
Norway	124	4.7
Japan	94	3.6
Sweden	79	3.0
France	79	3.0
India	74	2.8
Rest of the world	919	34.6
World total	**2646**	**100**

US is the leading country in electricity producers using oil. As seen in Table 3.5, Saudi Arabia, Iraq and Iran have a reasonable share with 13% in total of the world. Electricity generation by using gas has increased due to its lower cost compared to coal. Russia and US have the 36% share of the total production in the world by this source as seen in Table 3.6.

Table 3.4 Electricity producers by using coal in the world (2001) [135]

Producers	**Production amounts (TWh)**
US	1983
China	1122
India	452
Germany	301
Japan	239
South Africa	199
Australia	170
Russia	169
Poland	137
UK	134
The rest of the world	1086
World total	**5992**

Nuclear electricity satisfies both economic and environmental protection goals. The controlled fission of small amounts of uranium fuel creates large volumes of electricity without combusting carbon-based fuel sources. This avoids the release of both residual carbon gases and other combustion by product emissions such as nitrogen oxides. Improved production techniques have actually reduced the amounts of used fuel created while increasing electricity output. The world biggest 10 countries that produce electricity by nuclear power plants and corresponding installed capacities are seen in Table 3.7.

Table 3.5 Electricity producers by using oil in the world (2001) [135]

Producers	**Production amounts (TWh)**
US	134
Japan	117
Mexico	93
Saudi Arabia	87
Italy	75
China	47
Iraq	34
Russia	30
Chinese Taipei	30
Iran	28
Rest of the world	493
World total	**1168**

Table 3.6 Electricity producers by using gas in the world (2001) [135]

Producers	Production amounts (TWh)
US	646
Russia	377
Japan	257
UK	143
Italy	104
Iran	97
Thailand	72
Germany	57
Egypt	56
Malaysia	56
Rest of the world	963
World total	**2828**

Table 3.7 Electricity producers by nuclear power plants in the world and corresponding installed capacities (2001) [135]

Producers	Production amounts (TWh)	Installed Capacity (GW)
US	808	95
France	421	63
Japan	320	44
Germany	171	21
Russia	138	21
Korea	112	14
UK	90	13
Canada	77	12
Ukraine	76	11
Sweden	72	9
Rest of the world	369	53
World total	2654	356

3.3 Energy Outlook of Turkey

Turkey is an energy-importing country and more than about 70% of energy consumption in the country is met by imports. The share of imports grows each year. While the primary energy consumption in 1970 was 18.84 millions petroleum equivalent energy (PEE), it reached 78.41 millions PEE with an increase rate of 316% in 2003. While 76.9% of the consumed energy in 1970 was met by the domestic energy sources, this percentage decreased to 28% in 2003 [136, 137].

Turkey is dependent on foreign countries especially in terms of oil, natural gas and hard coal. In 1990, the percentages of import were 87.3% of oil, 95.7% of natural gas, and 66.5% of hard coal. The figures in 2003 were 92.3%, 97.4%, and 88.6%, respectively. While the primary energy production growth rate was 0.81% for the period 1985-2003, the consumption growth rate was 4.23% [138, 139]. The electricity production of Turkey was 23,200 GWh in 1983; it reached 140,500 GWh in 2003 demonstrating a six times increase. In spite of such an increase, the electricity production per capita for the year 2003, which was 1581 kWh/person, could not reach the world production per capita, which was 2333 kWh/person [140]. The distribution of electricity production by energy sources for the year 2003 was as follows: hard coal 2%, lignite 17%, hydraulic 25%, oil 7%, and natural gas 45%. The percentage of domestic sources was about 44%.

Turkey spent 9.5 billion dollars overall for energy imports in 2005. This amount constitutes 20% of the overall exports in 2005 and 14% of the overall imports. This means that Turkey spends one-fifth of the income it obtains from overall exports for energy imports. Oil is the most imported energy source with a percentage of 50% (4.7 billion dollars) of the overall energy source imports and it is followed by natural gas with a percentage of 40% (3.8 billion dollars) and hard coal with a percentage of 10% (0.8 billion dollars) [140].

3.4 Energy Sources in Turkey

3.4.1 Coal

Both hard coal and lignite deposits exist in Turkey. Hard coal is mostly located in the northwestern part of the country, in the Zonguldak province, which has more than 700 million metric tons of workable reserves, about 80% of which can be coked. Lignite deposits are wide spread and plentiful; reserves are estimated at more than 8 billion metric tons (7^{th} largest in the world), most of which is economically mineable, though only about 7% has a heat content of more than 3000 kcal/kg (12,500 kJ/kg).

Coal (lignite and hard coal together) has an important role in electricity production of Turkey in terms of installed capacity and electricity production.

Turkey realized the importance of energy production based on domestic sources after the oil crisis of 1973. Annual lignite production increased from 10 million tones to 65 million tones in a bout 10-15 years. However, with the importance given to the prevention of environmental pollution and tendency to use natural gas, the electricity production using coal decreased to 20% [140].

Coal is a non-renewable energy source, and therefore, we will inevitably run out of it. Coal is a fuel that burns with a thick black smoke that is released into the atmosphere. The smoke causes problems ranging form emphysema to discoloration of paint to even acid rain. As looked towards to the future of energy, it is clear that energy obtained from coal is not environment friendly source for many reasons. The sulfur dioxide and nitric acid can bind with water in the atmosphere and create sulfuric and nitric acid, or acid rain. The large amounts of carbondioxide released from the burning of coal cause greenhouse effect which cause global warming [141, 142]. A summary of coal fired thermal power plants in Turkey is shown in Table 3.8.

Table 3.8 Coal fired thermal power plants in Turkey [143]

Generating Facility	**Owner**	**Location (Province)**	**Fuel**	**Capacity (MW)**
Afşin-Elbistan A	TEAŞ	Malatya	Coal	1.36
Soma	TEAŞ	Manisa	Coal	1034
Yatağan	TEAŞ	Muğla	Coal	630
Kemerköy	TEAŞ	Muğla	Coal	630
Seyitömer	TEAŞ	Kütahya	Coal	600
Çayırhan	Park Termik Inc.	Ankara	Coal	450
Tunçbilek	TEAŞ	Kütahya	Coal	429
Yeniköy	TEAŞ	Muğla	Coal	420
Kangal	TEAŞ	Sivas	Coal	300
Çatalağzı	TEAŞ	Zonguldak	Coal coke	300
Orhaneli	TEAŞ	Bursa	Coal	210
Total				**5004.36**

3.4.2 Petroleum

Turkey's oil reserves are relatively small. Oil is produced mainly in the southeast, with a small amount coming from the northwest of the country. Since its peak in 1991, domestic oil production has been declining owing to the depletion of resources. While Turkey produced 3.7 million tones of oil in 1999, production was 2.5 million tones in 2003 and is expected to be decline by almost half by 2010 [142].

The use of oil has been increasing gradually for the last several decades. Especially keeping with the pace of globalization and the rapid industrial development of the world, this result is inevitable in Turkey. Approximately, 42% of Turkey's total energy needs have been fulfilled by oil. By the greater contribution of natural gas, this rate has started to decrease. Roughly, 90% of Turkey's oil supplies are imported. This importation comes from the Middle East, Saudi Arabia, Iran, Iraq, Syria and Russia.

TPAO (Turkish Petroleum Corporation) is the largest oil producer and accounts for 68.4% of the total oil production, followed by 25% for Turk Perenco N.V (see Table 3.9). In 2003, 29 companies, 19 foreign and ten domestic, carried out exploration activities in Turkey. Twelve companies, two domestic and ten foreign, produced oil, either individually or as a part of joint ventures. In 2002, 8.3% of the total electrical energy generation came from oil fired plants in Turkey.

Table 3.9 Crude oil production of Turkey in 2003 [143]

Company	**Production (thousand tones)**
TPAO	1624
Perenco N.V.	594
Madison Oil Turkey Inc.	13
Petrom (Dorchester)	118
Others	26
Total	**2375**

3.4.3 Natural Gas

Turkey's known natural gas producing fields are mostly located in southeast Anatolia region. The natural gas reserve in Turkey's known regions, as reservoir total and as producible gas total, is 12 bcm (billion cubic meters), but this domestic gas production corresponds to 3% of the total gas demand making the country almost fully dependent on gas imports [140]. BOTAŞ (Petroleum Pipeline Corporation) is the sole natural gas importer. It has eight long-term sales and purchase contracts with six different supply sources. In Table 3.10, these natural gas import contracts are given according to 2003 data.

In Turkey, energy policies have had an important effect on the availability of natural gas and its development as a fuel for electricity generation. The driving force behind the growth in Turkey is the increased consumption of natural gas for electric power generation. The first auto-producer natural gas fired plant was installed in 1992. Following this application, many other cogeneration facilities using natural gas were established and use of natural gas for power production has risen dramatically [144]. According to the 2004 data, five municipalities, six industrial zones, 200 industrial plants, two fertilizer production facilities, and seven power plants utilize natural gas as an energy source [143]. In Table 3.11, natural gas and fuel oil fired thermal power plants are given.

The demand of natural gas in Turkey is expected to increase. The industrial usage of natural gas differs from year to year within sectors. The hosing sector's rate of 19% in 2000 decreased to 17% in 2005 and is expected to reincrease and reach to 20% in 2010 [145]. In the fertilizer sector, the situation does not vary much. However, the industry sector, which was 13% in 2000, increased to 31% in 2005, and is expected to increase 32% in 2010. In the electric sector, there is a decrease. The 66% of usage rate in 2000 decreased to 50% in 2005 and is expected to decrease to 46% in 2010 [140].

3.4.4 Hydro

Turkey's geography is highly conducive to hydroelectric power generation. The gross hydropotential, which is a function of topography and hydrogeology, is estimated to be 432,986 GWh/year [146]. Turkey has about 1% of the total world hydroelectric potential. There are many rivers in Turkey and five separate watersheds. In 2002, Turkey had 125 hydroelectric power plants in operation. Most of them are owned and operated by DSİ (State Hydraulic Agency). Independent companies who own hydroelectric projects in Turkey include Birecik Inc., which owns a 672 MW power plant on the Euphrates River, and CEAŞ which presently has more than 1000 MW generating capacity.

Table 3.10 Turkey's natural gas import contracts [142]

Existing agreements	Volume (bcm/yr)	Signature date	Length (years)	Operation date	Volumes delivered in 2003 (bcm)
Russia (West)	6	1986	25	1987	11.4 (total)
Algeria (LNG)	4	1988	20	1994	3.8
Nigeria (LNG)	1.2	1995	22	1999	1.1
Iran	10	1996	25	2001	3.5
Russia (Black Sea)	16	1997	23	2003	1.2
Russia (West)	8	1998	23	1998	See above
Turkmenistan	16	1999	30	-	0
Azerbaijan	6.6	2001	15	-	0

Table 3.11 Natural gas and fuel oil fired thermal power plants in Turkey [143]

Generating Facility	Owner	Location (Province)	Fuel	Capacity (MW)
İskenderun Works	İsdemir	Hatay	Fuel oil	220
Aliağa Refinery	Tüpraş	İzmir	Fuel oil	207
Aliağa Refinery	TEAŞ	İzmir	Fuel oil	180
Total				**607**
Bursa	TEAŞ	Bursa	Natural gas	1.4
Ambarlı	TEAŞ	İstanbul	Natural gas	1349
Hamitabat	TEAŞ	Tekirdağ	Natural gas	1.2
Uni-Mar	Int. Power, Enron, Midlands	Tekirdağ	Natural gas	504
Trakya Elektrik	Int. Power	Tekirdağ	Natural gas	498
Gebze	Çolakoğlu	Kocaeli	Natural gas	247
Esenyurt Doğa	Edison Energy	İstanbul	Natural gas	180
Bursa	Bis Energy	Bursa	Natural gas	174
Total				**2954.6**

Turkey has significant hydroelectric power resources such as the Southeast Anatolia Hydropower and Irrigation Project, which is also known as GAP Project. It has more than 104 total plants, installed capacity over 10.2 GW, and it is developing a great deal more, especially as part of the $32 billion project. GAP is such a significant project that when completed, it will be considered to be one of the biggest water development projects ever undertaken. It will include 21 dams, 19 hydro plants, around 7.5 GW of power generating capacity, and a network of tunnels and irrigation canals. The main Turkish hydro dams are: Atatürk, 2400 MW; Karakaya, 1800 MW; Ilısu, 1200 MW; Cizre, 240 MW; Silvan, 240 MW; Hakkari, 208 MW; Alpaslan II, 200 MW; Batman, 198 MW; Konaktepe, 180 MW; and Karkamış, 180 MW [141].

3.4.5 Geothermal

Turkey has significant potential for geothermal power production, possessing one-eighth of the world's total geothermal potential. Much of this potential is of relatively low enthalpy. So, it is not suitable for electricity production but it is still useful for direct heating applications. At the end of 1999, Turkey's total installed capacity for direct heating was 820 MWh, of which about 390 MWh provided heating for 51,600 residences, about 100 MWh provided heating for about 45 hectares of greenhouses, and about 330 MWh was used to provide heated water for about 200 spas. By 2010, as many as 500,000 residences could be heated by geothermal power, which would represent the use of about 3500 MWh [140].

Turkey presently has one operating geothermal power plant, a 20 MW facility in the Denizli-Kızıldere geothermal field in the south-western of Turkey, Denizli province. The facility includes nine production wells, also has an integrated liquid carbondioxide and dry ice production factory that can produce a combined total of 40,000 m^3 annually of the two products. Another 20 MW power production unit is being planned for this facility. Recently, new geothermal power plants are put in operation and new ones are being built.

3.4.6 Wind

Turkey has a considerable potential for electricity generation from wind. A study carried out in 2002 concluded that Turkey has a theoretical wind energy potential of nearly 90,000 MW and an economical wind energy potential of about

10,000 MW. The most promising region is in northwestern region including the area around Marmara sea [147].

Turkey is now encouraging the construction of wind power plants by private power developers. The first wind power facility in Turkey, the Ares winds farm, was commissioned in 1998, and is located near the city of İzmir in western Turkey. That facility has 12 wind turbines for a total capacity of 7.2 MW, and is owned by Güçbirliği Holding Inc. Bozcaada wind farm, also near İzmir, went into operation in 2000. It has 17 turbines for a total capacity of 10.2 MW, and it is owned by Demirer Holding Inc. Turkey has a goal of deriving 2% of its electricity from wind power. In 2000, the Turkish Government had offered a tender for up to 390 MW of electricity from wind power. About 25 potential sites for wind-power projects had been identified and were undergoing evaluation, but the tender was cancelled as part of the IMF induced economic policy changes [148].

3.4.7 Nuclear Power

Since the discovery of atomic energy in the 1940s, it has been the cause of thousands of deaths, but has also been the subject of some of the most amazing discoveries in the 20th century. The amount of energy that could be released from a small amount of fuel seemed to be the answer to all of our energy problems, but today, the dangers in this kind of energy generation remains as concern.

The Turkish Government has announced plans to build 10 nuclear reactors in 2020. The first reactor is planned in Akkuyu Bay on southeast Mediterranean coast [149]. In July 2000, Turkey cancelled its plans for building a 1400 MW nuclear power plant at Akkuyu Bay. Prior to the cancellation, three international consortia were bidding for the $2.5 billion contract. The cancellation was directly caused by Turkey's economic situation, and the Turkish Treasury Department's refusal to grant sovereign guarantee for the project. However, there had been much opposition to the project for a variety of reasons, including a significant seismic risk in the area where the power plant was to be sited [150, 151]. Just recently, a legislation prepared by the government is passed that allows building nuclear power plants in Turkey and provides the framework for the rules of it.

3.5 Electricity Market and Private Power Generation in Turkey

3.5.1 Historical Background

In 1960s, the Turkish government started the “development plans era”. The Ministry of Energy and Natural Resources (MENR) was established in 1963, and was responsible for Turkey’s energy policy. This was followed in 1970 by the creation of Turkish Electricity Administration (TEK), which would have a monopoly in the Turkish electricity sector at almost all stages apart from distribution, which were left to the local administrations [152].

In the early 1980s, the Turkish electricity industry was dominated by a state-owned vertically integrated company, TEK. Starting from the 1980s, the government sought to attract private participation into the industry in order to ease the investment burden on the general budget. In 1982, the monopoly of public sector on generation was abolished and the private sector was allowed to build power plants and sell their electricity to TEK. In 1984, TEK was restructured and gained the status of state-owned enterprise.

Various private sector participation models short of privatization were put into practice. The first law setting up a framework for private participation in electric industry was enacted in 1984 (Law no. 3096). This law forms the legal basis for private participation through Build Operate and Transfer (BOT) contracts for new generation facilities, Transfer of Operating Rights (TOOR) contracts for existing generation and distribution assets, and the autoproducer system for companies to produce their own electricity. Under BOT concession, a private company would build and operate a plant for up to 99 years (subsequently reduced 49 years) and then transfer it to the state at no cost. Under a TOOR, the private enterprise would operate (and rehabilitate where necessary) an existing government-owned facility through a lease-type arrangement [153].

In 1993, TEK was incorporated into privatization plan and split into two separate state-owned enterprises, namely Turkish Electricity Generation Transmission Co. (TEAŞ) and Turkish Electricity Distribution Co. (TEDAŞ). However, the constitutional court of Turkey issued a series of rulings in 1994 and

1995 making the privatization almost impossible to implement in electricity industry. To overcome the deadlock; in August 1999, the parliament passed a constitutional amendment permitting the privatization of public utility services and allowing international arbitration for resolving disputes. However, during this interval, Turkey not only lost five invaluable years in terms of reform process that could never get back but also, and more importantly, tried to enhance the attractiveness of BOT projects by providing “take or pay” guarantees by the Undersecretariat of Treasury for adding new generation capacity to meet anticipated demand. An additional law, namely the Build Operate and Own (BOO) Law (No.4283), for private sector participation in the construction and operation of new power plants was also enacted in 1997 again with guarantees provided by the Treasury [154, 155]. Current structure of the contracts concluded based on these laws acts as a major barrier to the development of competition in the electricity sector.

3.5.2 Reforms in Turkish Electricity Market

By the end of 1990s, it became clear that quasi-privatization with Treasury guarantees was not going to be feasible given the rapidly deteriorating fiscal situation. Therefore, Turkey turned to a radically different framework for the design of energy market.

On March 2001, Electricity Market Law (EML, No. 4628) came into force and aimed at establishing a financially strong, stable, transparent, and competitive electricity market. In line with new law, TEAŞ was restructured to form three new state-owned public enterprises, namely Turkish Electricity Transmission Co. (TEİAŞ), Electricity Generation Co. (EÜAŞ) and Turkish Electricity Trading and Contracting Co. (TETAŞ). The new law also created an autonomous regulatory body, namely Electricity Market Regulatory Authority (EMRA). EMRA has administrative and financial autonomy; it receives no financing from state budget. It collects its revenues from electricity and gas licensing fees and from a surcharge on electricity TPA (regulated third part access for transmission and distribution of electricity related with electricity market law) tariff (maximum 1%) [152].

Along the lines of developments in electricity sector, some other reforms were also introduced in other segments of the energy industry. On May 2001, Natural Gas Market Law (NGML, No. 4646) also came into force and aimed at achieving similar objectives in natural gas market. It also renamed the regulatory body as EMRA. As a final step, Petroleum Market Law (PML, No. 5015) and Liquefied Petroleum Gas Market Law (LPGML, No. 5307) came into force on December 2003 and March 2005, respectively, and EMRA was granted the responsibility to regulate these markets as well [153, 154].

3.5.3 Current Assessment of the Turkish Electricity Market

In 2006, the total electricity power generation was 175,893 GWh with an annual increase of 8.6% while the total electricity power consumption was 174,231 GWh with an annual average increase of 8.3%. This year, Turkey exported the highest amount of electricity power recorded up to date. A total of 2236 GWh electricity power were exported to Iraq, Naxcivan, Georgia and Syria. The electricity power import was 573 GWh from Georgia and Turkmenistan.

According to Table 3.12, 48.1% of Turkey’s total electricity power generation in 2006 was produced by the generation facilities within EÜAŞ and subsidiaries. The share of power stations within the scope of BO model, BOT model and TOOR models are 24.3%, 8.4% and 2.3% respectively whereas autoproducers and autoproducer groups has a share of 9.4%. The contributions of power stations belonging to the private sector and holding generation license and mobile power plants have the portions of 7.2% and 0.3% respectively. The total generation of the companies holding generation license except public sector was 12,664 GWh in 2006.

EMRA granted 103 licenses for electricity production and 25 for autoproducers and autoproducer groups corresponding to the total powers of 4466.2 MW and 172.1 MW respectively and four wholesale licenses within 2006. Of the 132 licenses, 128 belong to the private sector, while the rest relate to the plants in operation and under construction within the body of TEİAŞ. The distribution of power plants installed by the private sector is as follows: 86 licenses with the total power of 2745.4 MW for hydraulic plants, 33 with the total power of 1747.8 MW for fuel oil, naphtha, biogas fired plants and geothermal power plants, 8 corresponding to 139.3 MW for wind and 1 corresponding to 5.8 MW for landfill gas fired electricity generation power plants. EMRA granted licenses to 9 generation facilities with an installed capacity of 608.5 MW under the scope of BOT model in 2006.

Table 3.12 The contribution of Turkey's total electricity production in 2006 according to the generation types by EMRA [156]

Electricity Generation Types	**Installed Capacity (GW)**
EÜAŞ	84,604.5
BO	42,742
BOT	14,775
TOOR	4045.5
Autoproducers and Autoproducer Groups	16,534
Privileged Companies	12,664
Mobile Power Plants	527.7
Total	**175,893**

There are currently 509 projects corresponding to an installed capacity of 18,511 MW for licensing from EMRA. The distribution of these projects according to the power capacity is as follows: 8267.5 MW for hydraulic power plants, 7274.5 MW for wind power plants, 15.9 MW for other renewable resources and 2952.8 MW for diesel engine powered plants, biogas power plants and landfill gas fired power plants.

3.5.4 Diesel Engine Powered Cogeneration and Mobile Power Plants

Most developing countries are characterized by deficiencies of various degrees in the substructure services produced and delivered by public sector. Deficiencies may be observed in power sector, telecommunication, transport services, water and waste disposal. In rapidly developing countries such Turkey, deficiencies occur because the rapid growth in the demand for those substructure congests the capacity of the public sector to deliver services of a uniform quality. The other deficiencies may be due to a combination of rapid growth in urban areas and a lack of equipment, and adequately trained personnel.

Firms can eliminate those deficiencies in a number of ways. In the case of electric power deficiency in manufacturing sector, the firms blend two types of power: the public power source which is cheap but of lower quality (i.e. subject to more voltage fluctuations) and its own in-house power which is more expensive but of a higher quality. The extent of internal generation of power varies among firms and is related to their production technologies. Two most important reasons for producing electricity are the uncertainty about power outputs and the fluctuations in the voltage of public power which can cause damages to plant, equipment, and intermediate inputs and outputs in the assembly line. To minimize this problem, blender firms produce their own power to either substitute or supplement the public supply. The endogenously generated power is used to "boost" the power supply obtained from the public sector smoothing out voltage fluctuations in the public supply or it is used to supply plant when the public power source is interrupted [157].

Power generation using reciprocating engines was not as common three decades ago as it is today. The main application for engine derived power was in small backup plants for hospitals, airports, hotels, and industry that needed to ensure a reliable power supply at all times.

In order to meet the intermediate needs for power capacity EÜAŞ introduced "mobile power plants" concept in 1998. A total of 75 MW diesel-fired power plants were built in 1999. In 2006, the total installed capacity of mobile power plants and diesel engine powered cogeneration plants reached 749.7 MW and 179.9 MW respectively [156]. The diesel engine powered mobile power plants and cogeneration facilities in Turkey with corresponding installed capacities are given in Table 3.13 and Table 3.14, respectively.

Heavy fuel oil fired diesel power plant maybe a good solution for Turkey. These power plants run on cheap and almost domestic fuel, RF4 (i.e. Heavy fuel oil no: 6), which is the low grade product of all seven Turkish refineries. Such power plants have proven to be reliable and economical means of power generation worldwide and Turkey is no exception.

Table 3.13 Diesel engine powered mobile power plants in Turkey (2006) [156]

Mobile power plants	Location	Fuel	Installed Capacity (MW)
Batman	Batman	Fuel oil	117.9
Esenboğa	Ankara	Fuel oil	53.8
Hakkari	Hakkari	Fuel oil	24.8
Isparta	Isparta	Fuel oil	27.9
Kırıkkale	Kırıkkale	Fuel oil	153.9
Kızıltepe	Mardin	Fuel oil	34.1
PS3 – A 2 & PS3 – 2	Şırnak	Fuel oil	24.4
Samsun 1	Samsun	Fuel oil	131.3
Samsun 2	Samsun	Fuel oil	131.3
Siirt	Siirt	Fuel oil	25.6
Van	Van	Fuel oil	24.7
Total			**749.7**

Table 3.14 Diesel engine powered cogeneration facilities in Turkey (2006) [156]

Diesel engine powered cogeneration plants	Location	Fuel	Installed Capacity (MW)
Karkey (Silopi 1)	Şırnak	Fuel oil	45.5
Karkey (Silopi 2)	Şırnak	Fuel oil	30.5
Karkey (Silopi 3)	Şırnak	Fuel oil	34.1
Karkey (Silopi 4)	Şırnak	Fuel oil	33.2
PS3 – A1 (İdil Energy)	Şırnak	Fuel oil	11.4
Sanko Energy	Gaziantep	Fuel oil	25.2
Total			**179.9**

3.6 Pollution and Environmental Concerns

Turkey has been undergoing major economic changes in the 1990s and 2000s, marked by rapid overall economic growth and structural changes (privatization of state enterprises, price liberalization, integration in the European and global economy). However, the share of the governmental sector in the Turkish economy remains high. Turkey's population has reached 80 million and remains one of the fastest growing from 1990 to 1999 in the OECD (Organization for Economic Cooperation and Development) [158]. Major migrations from rural areas to urban, industrial and tourist areas continue.

Air pollution is becoming a great environmental concern in Turkey. Air pollution from energy utilization in the country is due to the combustion of coal, lignite, petroleum, natural gas, wood and agricultural and animal wastes. On the other hand, owing mainly to the rapid growth of primary energy consumption and the increasing use of domestic lignite, SO_2 emissions, in particular, have increased rapidly in recent years in Turkey. The major source of SO_2 emissions is the power sector, contributing more than 50% of the total emissions [146].

3.6.1 CO_2 Emissions

Turkey's total carbon dioxide (CO_2) emissions amounted to 193 million tones (Mt) in 2002. Emissions grew by 4% compared to 2001 levels and by just over 50% compared to 1990 levels. Oil has historically been the most important source of emissions, followed by coal and natural gas. Oil represented 42% of total emissions in 2002, while coal represented 40% and natural gas 18%. The contribution of each fuel has however changed significantly owing to the increasingly important role of natural gas in the country's fuel mix starting from the mid-1980s. In Figure 3.2, CO_2 emissions by fuel types between 1973 and 2002 are given.

According to the recent projections, total primary energy supply (TPES) will almost double between 2002 and 2020, with coal accounting for an increasingly important share, rising from 26% in 2002 to 36% in 2020, principally replacing oil, which is expected to drop from 40% to 27%. Such trends will lead to a significant rise in CO_2 emissions, which are projected to reach nearly 600 Mt in 2020, over three times of 2002 levels [142].

Per capita CO_2 emissions were at 2.8 tones in 2002, much lower than the OECD average of 11 tones. Between 1990 and 2002, per capita emissions in Turkey grew by 21% while on average they grew by only 4% at the OECD level and dropped by 3% in the IEA Europe region. However, owing to the important growth in emissions that took place over the 1990s, by 2002 CO_2 emissions per unit of GDP were only marginally lower than the OECD average [143].

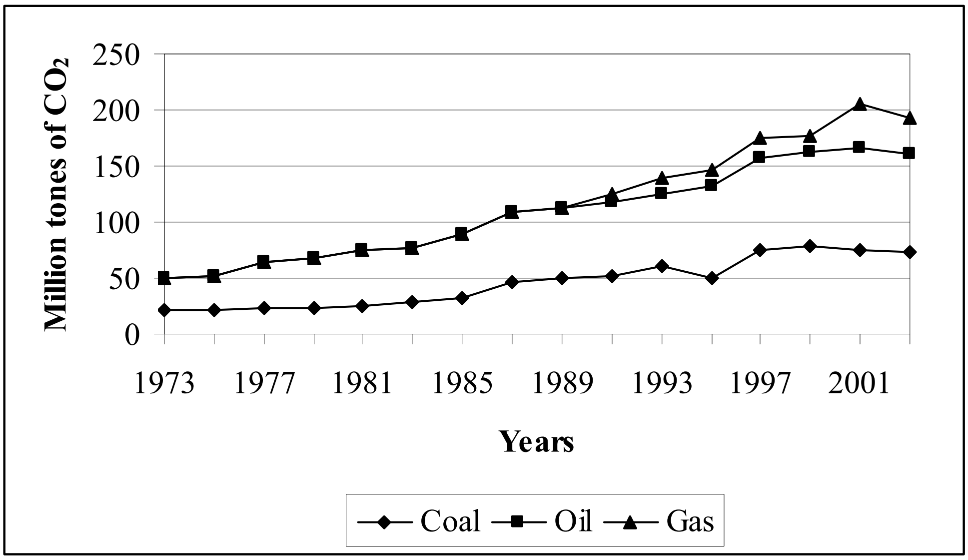

Figure 3.2 CO_2 emissions by fuel types between 1973 and 2002 [142]

3.6.2 NO_x and SO_x Emissions

The main air pollutants related to the production and use of energy is sulfur oxides (SO_x) – in particular sulfur dioxide (SO_2), - nitrogen oxides (NO_x) and suspended particulates. These emissions come mostly from the combustion of solid and liquid fuels. The use of high-sulfur lignite in particular is an important source of air pollution.

As a consequence of efforts to move away from high-sulfur lignite to either imported coal or natural gas, air pollution concentration levels have reduced significantly in most large cities since the early 1990s [143]. In 2001, Turkey emitted a total of 2.08 Mt of SO_2, equivalent to 30.4 kg per capita. This is slightly below the OECD average, which at the end of the 1990s was 32.9 kg per capita [142]. In terms of emissions per unit of GDP, Turkey emitted 5.5 kg per 1000 US$ in 2001, among the highest levels in OECD countries where the average was approximately 1.5 kg per 1000 US$. Electricity generation and industry are by far the largest contributors to SO_2 emissions in the country, representing respectively 65% and 21% of total emissions in 2001 [158].

Emissions of NO_x totaled approximately 0.90 Mt in 2003, slightly below 2000 levels of 0.92 Mt. NO_x emissions have nevertheless been rising over the past decades. According to the OECD, over the 1990s only, NO_x emissions grew by 48%. On a per capita level, emissions were of 12.8 kg in 2003, substantially below the OECD average of approximately 40 kg at the end of the 1990s. On the other hand, emissions per unit of GDP were at 2.1 kg per 1000 US$ in 2003, above the OECD average, which at the end of the 1990s was around 1.9 kg per 1000 US$. Transportation and predominantly road-based transport, is the largest source of NO_x emissions, representing 36% of total emissions. Electricity generation and industry represent over 20% each [142]. In Table 3.15, total emission estimations with five years intervals in Turkey are given.

Table 3.15 Total emission estimates with five year intervals in Turkey (mg/yr) [158]

Years	PM	SO_x	NO_x	VOC	CO	CH_4
1990	4,976,456	2,620,105	612,345	427,864	1,443,276	153,441
1995	6,012,112	3,123,344	696,678	413,976	1,584,554	138,334
2000	6,964,224	3,486,623	834,776	443,568	1,786,645	142,873
2005	7,789,677	4,134,543	956,744	465,765	1,986,865	149,673
2010	8,986,687	4,875,789	1,214,762	504,443	2,243,543	154,534
2015	9,345,256	5,668,922	1,764,322	539,543	2,544,567	159,789
2020	10,122,342	6,234,544	2,344,176	591,344	2,943,876	162,356

VOC: volatile organic compounds, PM: particulate matter, CH_4: methane

3.6.3 Air Quality Regulation

The main principles of the Turkish environmental policy have been identified as management of natural resources enabling continuous economic development through protection of human health and natural balance and leaving natural, physical and social environment to the future generations which they deserve. On the other hand, the fundamental objective of the energy sector is to supply reliable, inexpensive and high-quality energy to all consumer sectors wherever and whenever required at appropriate price to sustain economic and social development in an environmentally sound way [131].

Air quality standards for four pollutants, namely SO_2, nitrogen dioxide (NO_2), particulate matter (PM), and ozone (O_3) are set under the 1986 Air Quality Protection regulation. As shown in Table 3.16, these standards are much less stringent than those set by the WHO (World Health Organization). The monitoring of ambient air pollution has improved over recent years but remains a problem, particularly with regards to NO_2 and O_3. Until recently, the 1986 regulation was also responsible for setting air pollution standards for combustion plans. It was amended in October 2004 by the new Industrial Air Pollution Control Regulation.

Table 3.16 Turkish and WHO air quality standards [142]

	Turkish Standards		WHO Standards	
	LTS ($\mu g/m^3$)	STS ($\mu g/m^3$)	LTS ($\mu g/m^3$)	STS ($\mu g/m^3$)
SO_2	150	400[a]	50	125
NO_2	100	300	-	150
PM	150	300	50	120
O_3	240[b]		100-200	-

$\mu g/m^3$: micrograms per cubic meter
LTS: long term standards (maximum annual average)
STS: short term standards (maximum daily average)
PM: particulate matter with particles less than or equal to 10 micrometers (µm) in diameter
- not applicable
[a] Turkey's ambient air quality standard for SO_2 on an hourly basis is 900 $\mu g/m^3$
[b] This represents the maximum value allowable in any one-hour period

The regulation sets standards for the emissions of NO_x, SO_2, CO (carbon monoxide) and PM. NO_x and SO_2 standards have not changed compared to 1986 standards, while PM and CO standards have been lowered for both solid and liquid fuel-fired plants. In the case of PM, standards have been lowered from 150 $\mu g/m^3$ to 100 $\mu g/m^3$ for solid fuel-fired power plants. For CO, standards have been lowered from 250 $\mu g/m^3$ to 200 $\mu g/m^3$ in the case of solid fuel-fired plants and from 175 $\mu g/m^3$ to 150 $\mu g/m^3$ in the case of liquid fuel-fired plants.

Given the high sulfur content of domestic lignite, new lignite-fired power plants have been equipped with flue gas desulphurization (FGD) technology in order to comply with the regulation. To reduce emissions from pre-1986 lignite-fired power plants, these plants are progressively being retrofitted with FGD technology.

At present, six out of eleven lignite power plants have been retrofitted. No schedule has been defined for the five remaining plants. Both new and old power plants have been fitted electrostatic precipitators (ESP). However, owing to technical problems, not all ESP are working at maximum efficiency.

The Industrial Air Pollution Control Regulation sets limits and penalties for non-compliance with emission standards for power plants and gives the Ministry of Environment and Forestry responsibility for plant authorization and enforcement. Under the new regulation, the plant operators are responsible for continuous monitoring of stack emissions. Plant operators are also responsible for contracting with an independent authorized laboratory to provide compliance monitoring and plant vicinity air quality assessments. This is a notable from the 1986 regulation where the Ministry of Health and the Ministry of Environment and Forestry shared monitoring responsibilities. The dual role of the government as owner and operator of most power plants on the one hand and as the air quality enforcement authority on the other has historically made the enforcement of air quality standards difficult. It is unclear whether giving plant operators emissions monitoring responsibilities, as well as the responsibility for contracting for compliance monitoring and air quality assessment, will provide sufficient independence to improve this situation. In addition, the enforcement capacity of the Ministry of Environment and Forestry is limited owing to resource constraints.

The emission standards for power plants in Turkey are given in Table 3.16 with the revision in 2004. These standards remain significantly less stringent than those currently in force at the EU (European Union) level as defined by the revised Large Combustion Plants (LCP) directive [142]. For example, for new solid fuel-fired power plants (authorized after November 2003) with a thermal input greater than 300 MW, the NO_x emissions limit is set at 200 mg/Nm^3 at the EU level, while the NO_x emissions limit is 800 mg/Nm^3 in Turkey.

The “approximation” process with EU legislation has important implications for the energy sector, particularly as regards the LCP directive and the Integrated Pollution Prevention and Control Directive [142]. A number of studies on how to comply with the EU LCP directive are underway. First estimates show that achieving the standards defined under the LCP directive would entail investments of over 1 billion US$. This would include investments in the retrofitting of installed FGD and ESP equipment and the adoption of advanced and environment-friendly coal technologies. The 2004 Industrial Air Pollution Control Regulation is an important step towards aligning air quality standards with EU regulations, but more efforts will be needed.

Table 3.17 Emission Standards for power plants in Turkey [142]

1986 REGULATION (mg/Nm^3)									
	PM		**CO**	**NO_x**		**SO_2**			
	OP	NP		OP	NP	<300 MW_{th}		>300 MW_{th}	
						ROH>20,000	NP	ROH<50,000 and OP	ROH>50,000 and NP
SFFP	250	150	250	1000	800	3200	2000	3200	1000
LFFP	110	110	175	1000	800	3200	1700	1700	800
GFP	10	10	100	500	500	60	60	60	60
2004 REGULATION (mg/Nm^3)									
SFFP	100		200	800		2000 (<100 MW_{th}) 1300 (100 -300 MW_{th})		1000 ≥300 MW_{th}	
LFFP	110-170 ≥15 MW_{th}		150	800		1700 (1% S[a]) 2400 (1.5% S) (<100 MW_{th}) 1700 (100-200 MW_{th})		800 ≥300 MW_{th}	
GFP	10		100	500		60		60	

SFFP: Solid fuel-fired plants, LFFP: Liquid fuel-fired plants, GFP: Gas-fired plants,
OP & NP: Old Plants and New Plants, refer to power plants built before and after Air Quality Protection regulation came into force in 1986
ROH: Remaining operating hours
[a] Sulfur content (mass percentage) of the fuel used

The industry and residential sectors are also responsible for significant air pollution, mainly as a result of lignite consumption. In order to reduce emissions from these sectors, the state owned Turkish Coal Enterprises (TKİ) has developed significant lignite washing capacity. By the end of 2003, total washing capacity was approximately 10 Mt, equivalent to current coal demand from both sectors. In addition, the use of high-sulfur coal in residential heating is prohibited. Lastly, the substitution of natural gas as distribution networks are expanded in urban areas should further contribute to reduce air pollution.

In the transport sector "the Gasoline and Diesel Oil Quality Regulation" was enacted in June 2004. It provides the necessary arrangements for harmonization of the gasoline and diesel oil standards with the most recent EU standards. While the directive specifies a transitional period between 2005 and 2009 for full compliance with the new standards, in the Turkish regulation the transitional period has been started in 2007.

In the first half of 2004, unleaded gasoline represented 73% of total gasoline sold while the share was only 63% in 2003. This reflects the increasing proportion of the car fleet being fitted with catalytic converters. It is forecasted that by 2012, the entire car fleet will be fitted with such converters. Since 2000, all imported and domestically produced new automobiles are equipped with catalytic converters and EU standards are in place. The government also promotes the use of unleaded gasoline through a preferential pricing policy. Finally, the government is in the process of upgrading İzmit and İzmir refineries through the construction of hydro cracking and izomerization units. The rehabilitation project is planned to be concluded at the end of 2007. Kırıkkale refinery already complies with current and post-2005 EU regulations on petroleum quality fro both leaded and unleaded gasoline [142].

In regards to diesel, new desulphurization units are under construction and are planned to be operational in three major refineries so as to comply with the EU standards by 2007, as envisaged in the 2004 regulation. In parallel, the Ministry of Industry and Trade has issued regulations in 2003 in order to transpose into national law the EU directives related to vehicle standards for emissions of gaseous pollutants.

3.7 Conclusions

Still being a developing country, Turkey's economy is growing at a high rate and this is accompanied by an annual average 8% increase in energy use. Turkey imports most of its energy and there is a relatively high diversification of energy sources including coal, oil, natural gas, hydro, wind, and geothermal. As the country is moving forward with the establishment of nuclear power plants this diversification will be nearly complete. Diesel engine powered cogeneration is significant as being an effective method for private industries to produce their own electricity while meeting part of their heating need.

CHAPTER 4

THERMODYNAMIC ANALYSIS

4.1 Introduction

Energy is the most fundamental term in thermodynamics and energy engineering. Energy analysis is often one of the most significant parts of engineering analysis. Energy can be stored within a system in various macroscopic forms, it can be transformed from one form to another, and it can be transferred between systems. The total amount of energy is conserved in all transformations and transfers. Energy balances are widely used in the design and analysis of energy conversion systems. Although energy balances can determine energy supply requirements in the form of material streams, heat, and shaft work, they do not provide sufficient information on how efficiently energy is used.

The only inefficiencies detected by the energy analysis of a system are the energy transfers out of the system that are not further used in the overall installation. Hence, the heat transfer to the environment is often used as a measure of the so-called energy loss. This approach is misleading for two reasons: (1) the heat rejection to the environment is sometimes unavoidable (e.g., even in the reversible Carnot cycle there is heat rejection to the environment as a consequence of the second law of thermodynamics), and (2) thermodynamic inefficiencies mainly occur within a system (e.g., reducing the pressure of a fluid in an adiabatic throttle is a dissipative process without heat transfer to the environment).

Energy balance focuses on the quantity of energy and fails to account for the quality of energy. The true thermodynamic value (quality) of an energy resource is expressed by its potential to cause a change, that is, "to do something useful", such as heat a room, compress a gas, or promote an endothermic chemical reaction.

Kinetic, potential, mechanical, and electric energy can be fully converted in an ideal process to any other form of energy, whereas the quality of thermal and chemical energy depends on parameters (temperature, pressure, and chemical composition) of the energy carrier and of the environment. Electricity clearly has a greater quality than low-pressure steam or cooling water stream in a power plant. In thermodynamics, the quality of a given quantity of energy is characterized by its exergy [51].

Exergy is the theoretical maximum of useful work (shaft work or electrical work) obtainable from a thermal system as this is brought into thermodynamic equilibrium with the reference environment while heat transfer occurs with this environment only. Alternatively, exergy is the theoretical minimum of work (shaft work or electrical work) required to form a quantity of matter from substances present in the environment and to bring the matter to a specified state. Hence, exergy is a measure of the departure of the state of the system from the state of the reference environment. The processes in all real energy conversion systems are irreversible and a part of the exergy supplied to the total system is destroyed. Only in a reversible process does the exergy remain constant [160].

The second law of thermodynamics complements and enhances an energy balance by enabling calculation of both the true thermodynamic value of an energy carrier, and the real thermodynamic inefficiencies in processes or systems. The concept of exergy is extremely useful for this purpose. The real inefficiencies of a system are exergy destruction, occurring within the system boundaries, and exergy losses, which are exergy transfers out of the system that are not further used in the overall installation. Some of the common causes for exergy destruction include chemical reaction, heat transfer across a finite temperature difference, fluid friction, flow throttling, and mixing of dissimilar fluids.

In this chapter we present general formulations of thermodynamic analysis including energy and exergy methods. The formulations are applicable to thermal systems including diesel powered cogeneration.

4.2 Energy Analysis

Energy conservation is expressed by energy balances and together with corresponding mass balances they are widely used in the modeling and analysis of energy conversion systems.

4.2.1 Mass Balance

The conservation of mass principle can be expressed as the net mass transfer to or from a system during a process is equal to the net change (increase or decrease) in the total mass of the system during that process [160]. In the rate form it is expressed as

$$\sum \dot{m}_{\mathrm{i}} - \sum \dot{m}_{\mathrm{e}} = \frac{dm_{\mathrm{system}}}{dt} \tag{4.1}$$

where *i* and *e* refer to inlet and exit states of the any control volume, respectively. During a steady flow process, the total amount of mass contained within a control volume does not change with time (m_{CV} = constant). Then the conservation of mass principle requires that the total amount of mass entering a control volume equal the total amount of mass leaving it. For a general steady-flow system with multiple inlets and exits, the conservation of mass principle can be expressed in the rate form as

$$\sum \dot{m}_{\mathrm{i}} = \sum \dot{m}_{\mathrm{e}} \tag{4.2}$$

4.2.2 Energy Balance

Based on experimental observations, the first law of thermodynamics states that energy can be neither created nor destroyed; it can only change forms. Therefore, every bit of energy should be accounted for during a process [160-162]. The conservation of energy principle may be expressed as follows: 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 leaving the system during that process. Energy balance for any system undergoing any kind of process can be expressed more compactly in the rate form as [160]

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \tag{4.3}$$

During a steady-flow process, the total energy content of a control volume remains constant (E_{CV} = constant), and thus the change in the total energy is zero. Therefore, the amount of energy entering a control volume in all forms (by heat, work, and mass) must be equal to the amount of energy leaving it. Then the rate form of the general energy balance reduces for a steady-flow process to

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \tag{4.4}$$

Noting that energy can be transferred by heat, work, and mass only, the energy balance above for a general steady-flow system can also be written more explicitly as

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} + \sum \dot{m}_{\text{i}} \left(h_{\text{i}} + \frac{V_{\text{i}}^2}{2} + gz_{\text{i}} \right) = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \sum \dot{m}_{\text{e}} \left(h_{\text{e}} + \frac{V_{\text{e}}^2}{2} + gz_{\text{e}} \right) \tag{4.5}$$

where h_{i}, h_{e}, V_{i}, V_{e}, z_{i}, z_{e} represent enthalpy, velocity, and elevation of mass entering and leaving the control volume, respectively.

4.3 Exergy Analysis

For the evaluation and improvement of thermal systems, it is essential to understand the sources of thermodynamic inefficiencies and the interactions among system components. All real energy conversion processes are irreversible due to dissipative effects such as chemical reaction, heat transfer through a finite temperature difference, mixing of matter at different compositions or states, unrestrained expansion, and friction. Exergy balances assist in calculating the exergy destruction within system components. Thus, the thermodynamic inefficiencies and the processes that cause them are identified. Only a part of the thermodynamic inefficiencies can be avoided by using the best currently available technology. Improvement efforts should be centered on avoidable inefficiencies. Dimensionless variables can be used for performance evaluations. Appropriately defined exergetic efficiency unambiguously characterizes the performance of a system from the thermodynamic viewpoint.

4.3.1 Reference Environment and Exergy Components

The environment, which appears in the definition of exergy, is a large equilibrium system in which the state variables (T_0, p_0) and the chemical potential of the chemical components contained in it remain constant when in a thermodynamic process heat and materials are exchanged between another system and the environment. This environment is called exergy-reference environment or thermodynamic environment. The temperature T_0 and pressure p_0 of the environment are often taken as standard-state values, such as 298.15 K and 1.013 bar. However, these properties may be specified differently depending on the application. For example, T_0 and p_0 may be taken as the actual or average ambient temperature and pressure, respectively, for the time and location at which the system under consideration operates or is designed to operate. For example, if the system uses air, T_0 would be specified as the average air temperature. If both air and water from the natural surroundings are used, T_0 would usually be specified as the lower of the temperatures for air and water when the installation operates above the ambient temperature [160,162,163].

Although the intensive properties of the environment are assumed to remain constant, the extensive properties can change as a result of interactions with other systems. It is important that no chemical reactions can take place between the environmental chemical components. The exergy of the environment is equal to zero. The environment is part of the surroundings of any thermal system.

In the absence of nuclear, magnetic, electrical, and surface tension effects, the total exergy of a system (E_{sys}) can be divided into four components: Physical exergy, $E_{\mathrm{sys}}^{\mathrm{PH}}$, kinetic exergy E^{KN}, potential exergy, E^{PT}, and chemical exergy, E^{CH}. Then the total exergy of a system is given by

$$E_{\mathrm{sys}} = E_{\mathrm{sys}}^{\mathrm{PH}} + E^{\mathrm{KN}} + E^{\mathrm{PT}} + E^{\mathrm{CH}} \tag{4.6}$$

The subscript *sys* distinguishes the total exergy and physical exergy of a system from other exergy quantities, including transfers associated with streams of matter. The total specific exergy on a mass basis e_{sys} is

$$e_{\text{sys}} = e_{\text{sys}}^{\text{PH}} + e^{\text{KN}} + e^{\text{PT}} + e^{\text{CH}} \tag{4.7}$$

The physical exergy associated with a thermodynamic system is given by

$$E_{\text{sys}}^{\text{PH}} = (U - U_0) + p_0(V - V_0) - T_0(S - S_0) \tag{4.8}$$

where U, V and S represent the internal energy, volume and entropy of the system, respectively. The subscript 0 denotes the state of the same system at the temperature T_0 and pressure p_0 of the environment. The rate of physical exergy $\dot{E}^{\text{PH}}$ associated with a material stream is

$$\dot{E}^{\text{PH}} = (H - H_0) - T_0(S - S_0) \tag{4.9}$$

where H and S denote the enthalpy and entropy, respectively. The subscript 0 denotes property values at the temperature T_0 and pressure p_0 of the environment. The physical exergy of a system consists of thermal exergy $\dot{E}^{\text{T}}$ (due to system temperature) and mechanical exergy $\dot{E}^{\text{M}}$ (due to system pressure):

$$\dot{E}^{\text{PH}} = \dot{E}^{\text{T}} + \dot{E}^{\text{M}} \tag{4.10}$$

An unambiguous calculation of the specific thermal and specific mechanical exergy is possible only for ideal gases and incompressible liquids:

$$e^{\text{T}} = \int_{T_0, p_0}^{T_0, p} c\left(1 - \frac{T}{T_0}\right)\mathrm{d}T \tag{4.11}$$

$$e^{\text{M}} = \int_{T_0, p_0}^{T_0, p} v\,\mathrm{d}p \tag{4.12}$$

where v denotes specific volume. For any fluid, the specific thermal exergy of a stream at temperature T and pressure p is expressed as

$$e^{\mathrm{T}} = e^{\mathrm{PH}}(T, p) - e^{\mathrm{PH}}(T_0, p) \tag{4.13}$$

The mechanical exergy is determined from

$$E^{\mathrm{M}} = E^{\mathrm{PH}} - E^{\mathrm{T}} \tag{4.14}$$

Kinetic and potential exergies are equal to kinetic and potential energies, respectively.

$$E^{\mathrm{KN}} = \frac{1}{2} m\vec{v}^{2} \tag{4.15}$$

$$E^{\mathrm{PT}} = mgz \tag{4.16}$$

Here $\vec{v}$ and z denote velocity and elevation relative to coordinates in the environment $(\vec{v}_0 = 0, z_0 = 0)$. Equations 4.15 and 4.16 can be used in conjunction with both systems and material streams. The exergy associated with shaft work, flow of electricity, kinetic energy, or potential energy is equal to the energy amount of each of these quantities.

Chemical exergy is the theoretical maximum useful work obtainable as the system at temperature T and pressure p is brought into chemical equilibrium with the reference environment while heat transfer occurs only with this environment. Thus, for calculating the chemical exergy, not only the temperature T_0 and pressure p_0 but also the chemical composition of the environment $x_{\mathrm{i}}^{\mathrm{e}}$ have to be specified. By definition, the exergy of the reference environment is equal to zero and there is no possibility of developing work from interactions between parts of the environment.

The standard molar chemical exergy $e_{\mathrm{sub}}^{\mathrm{CH}}$ of any substance consisting of its elements can be determined using the change in the specific Gibbs function $\Delta\overline{g}$ for the formation of this substance from the reaction of chemical elements present in the environment:

$$e_{\mathrm{sub}}^{\mathrm{CH}}(T_0, p_0) = \bar{g}_{\mathrm{sub}}(T_0, p_0) - \sum_{\mathrm{i}=1}^{\mathrm{M}} v_{\mathrm{i}} \left[e_{\mathrm{i}}^{\mathrm{CH}}(T_0, p_0) - \bar{g}_{\mathrm{i}}(T_0, p_0) \right] \tag{4.17}$$

where $\bar{g}_{\mathrm{i}}$, v_{i} and $e_{\mathrm{i}}^{\mathrm{CH}}$ denote, for the *i*-th chemical element, the Gibbs function at T_0 and p_0, the stoichiometric coefficient in the reaction, and the standard chemical exergy, respectively. The chemical exergy of a gas *i,* having the mole fraction $x_{\mathrm{i}}^{\mathrm{e}}$ in the environmental gas phase is [50,51]

$$e_{\mathrm{i}}^{\mathrm{ch}} = -\bar{R} T_0 \ln x_{\mathrm{i}}^{\mathrm{e}} \tag{4.18}$$

The chemical exergy of an ideal mixture of N ideal gases is given by

$$e_{\mathrm{M,ig}}^{\mathrm{ch}} = \sum_{i=1}^{N} x_{\mathrm{i}} e_{\mathrm{i}}^{\mathrm{ch}} + \bar{R} T_0 \sum_{i=1}^{N} x_{\mathrm{i}} \ln x_{\mathrm{i}} \tag{4.19}$$

where T_0 is the environmental temperature, $e_{\mathrm{i}}^{\mathrm{ch}}$ is the standart molar chemical exergy of the i-th substance and x_{i} is the mole fraction of the k-th substance in the system at T_0. For the chemical exergy calculations of liquids, the chemical exergy can be obtained if the activity coefficients γ_{k} are known such as

$$e_{\mathrm{M,l}}^{\mathrm{ch}} = \sum_{i=1}^{N} x_{\mathrm{i}} e_{\mathrm{i}}^{\mathrm{ch}} + \bar{R} T_0 \sum_{i=1}^{N} x_{\mathrm{i}} \ln(\gamma_{\mathrm{k}} x_{\mathrm{i}}) \tag{4.20}$$

The standard chemical exergy of a substance not present in the environment can be calculated by considering a reversible reaction of the substance with other substances for which the standard chemical exergies are known. For energy conversion processes, calculation of the exergy of fossil fuels is particularly important. The chemical exergy of a fossil fuel $e_{\mathrm{f}}^{\mathrm{ch}}$ on a molar basis can be derived from exergy, energy, and entropy balances for the reversible reaction:

$$e_f^{ch} = -\left(\Delta \bar{h}_R - T_0 \Delta \bar{s}_R\right) + \Delta e^{ch} = -\Delta \bar{g}_R + \Delta e^{ch} \tag{4.21}$$

with $$\Delta\bar{h}_R = \sum_i v_i \bar{h}_i = -\bar{h}_f + \sum_k v_k \bar{h}_k = -HHV$$

$$\Delta\bar{s}_R = \sum_i v_i \bar{s}_i = -\bar{s}_f + \sum_k v_k \bar{s}_k$$

$$\Delta\bar{g}_R = \Delta\bar{h}_R - T_0 \Delta\bar{s}_R$$

$$\Delta e^{ch} = \sum_k v_k e_k^{ch}$$

where i and k denote O_2, CO_2, H_2O, SO_2 and N_2. $\Delta\bar{h}_R$, $\Delta\bar{s}_R$, and $\Delta\bar{g}_R$ denote the molar enthalpy, entropy and Gibbs function, respectively of the reversible combustion reaction of the fuel with oxygen. *HHV* is the molar higher heating value of the fuel and v_k is the stoichiometric coefficient of the k-th substance in this reaction. For some fuels such as coal and oil, the enthalpy and entropy values of the fuel must be estimated using available approaches before the chemical exergy can be calculated. The higher heating value is the primary contributor to the chemical exergy of a fossil fuel. The molar chemical exergy of a fossil fuel may be estimated with the aid of its molar higher heating value as

$$\frac{e_f^{ch}}{HHV} \approx \begin{cases} 0.95 - 0.985 & \text{for gaseous fuels except } H_2 \text{ and } CH_4 \\ 0.98 - 1.0 & \text{for liquid fuels} \\ 1.0 - 1.04 & \text{for solid fuels} \end{cases}$$

For hydrogen and methane this ratio is 0.83 and 0.94, respectively.

4.3.2 Exergy Balance, Exergy Destruction, and Exergy Loss

All thermodynamic processes are governed by the laws of conservation of mass and energy. These conservation laws state that mass and energy can neither be created nor destroyed in a process. Exergy, however, is not conserved but is destroyed by irreversible processes within a system. Consequently, an exergy balance must contain a destruction term, which vanishes only in a reversible process. Furthermore, exergy is lost, in general, when a material or energy stream is rejected to the environment.

The exergy destruction represents the exergy destroyed $\dot{E}_{\mathrm{D}}$ due to irreversibilities (entropy generation) within a system. The irreversibilities are caused by chemical reaction, heat transfer through a finite temperature difference, mixing of matter, and unrestrained expansion and friction. The exergy destruction is calculated with the aid of either (a) an exergy balance formulated for the system being considered, or (b) the entropy generation, $\dot{S}_{\mathrm{gen}}$, within the system (calculated from an entropy balance) and the relationship [31,51]

$$\dot{E}_{\mathrm{D}} = T_0 \dot{S}_{\mathrm{gen}} \tag{4.22}$$

The former way is recommended when a comprehensive exergetic evaluation is conducted. The exergy destruction in the overall system is equal to the sum of the exergy destruction in all system components:

$$\dot{E}_{\mathrm{D,total}} = \sum_{\mathrm{k}=1}^{\mathrm{n_k}} \dot{E}_{\mathrm{D,k}} \tag{4.23}$$

The rate of exergy destruction in the *k*th component of a system is given by

$$\dot{E}_{\mathrm{D,k}} = \dot{E}_{\mathrm{F,k}} - \dot{E}_{\mathrm{P,k}} - \dot{E}_{\mathrm{L,k}} \tag{4.24}$$

where, $\dot{\mathrm{E}}_{\mathrm{F,k}}$ and $\dot{\mathrm{E}}_{\mathrm{P,k}}$ are the so-called exergetic fuel and exergetic product, respectively, and $\dot{E}_{\mathrm{L,k}}$ represents the exergy rate loss in the *k*th component, which is usually zero when the component boundaries are at T_0. For an overall system, $\dot{E}_{\mathrm{L,total}}$ includes the exergy flow rates of all non-useful streams rejected by this system to the surroundings.

The total exergy destruction value is also obtained from the exergy balance written for the overall system

$$\dot{E}_{\mathrm{D,total}} = \dot{E}_{\mathrm{F,total}} - \dot{E}_{\mathrm{P,total}} - \dot{E}_{\mathrm{L,total}} \tag{4.25}$$

A useful splitting of the total exergy destruction within a component is between avoidable and unavoidable exergy destruction. Unavoidable $\dot{E}_{\mathrm{D,k}}^{\mathrm{UN}}$ is that part of exergy destruction within one component that cannot be eliminated even if the best available technology in the near future. The avoidable exergy destruction rate $\dot{E}_{\mathrm{D,k}}^{\mathrm{AV}}$ is the difference between the total and the unavoidable exergy destruction rate [50].

$$\dot{E}_{\mathrm{D}} = \dot{E}_{\mathrm{D,k}}^{\mathrm{AV}} + \dot{E}_{\mathrm{D,k}}^{\mathrm{UN}} \tag{4.26}$$

It is apparent that all efforts to improve the thermodynamic efficiency of a component or system should focus on avoidable exergy destruction.

An exergy transfer across the boundary of a control volume system can be associated with either a material stream or an energy transfer by work or heat. By taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the general form of the exergy balance for a control volume involving multiple inlet and outlet streams of matter and energy can be expressed as

$$\frac{dE_{\mathrm{CV}}}{dt} = \sum\left(1-\frac{T_0}{T_{\mathrm{k}}}\right)\dot{Q}_{\mathrm{k}} - \left(\dot{W} - P_0\frac{dV_{\mathrm{CV}}}{dt}\right) + \sum\dot{E}_{\mathrm{i}} - \sum\dot{E}_{\mathrm{e}} - \dot{E}_{\mathrm{D}} \tag{4.27}$$

where $\dot{E}_{\mathrm{i}}$ and $\dot{E}_{\mathrm{e}}$ are the total exergy transfer rates at the inlet and outlet, respectively for the total, physical, chemical, kinetic, and potential exergy associated with mass transfers. The term $\dot{Q}_{\mathrm{k}}$ represents the rate of heat transfer at the location on the boundary where the temperature is T_{k}. The associated rate of exergy transfer $\dot{E}_{\mathrm{q,k}}$ is given by

$$\dot{E}_{\mathrm{q,k}} = \left(1-\frac{T_0}{T_{\mathrm{k}}}\right)\dot{Q}_{\mathrm{k}} \tag{4.28}$$

For $T_{\mathrm{k}}>T_0$, the exergy rate $\dot{E}_{\mathrm{q,k}}$ associated with heat transfer is always smaller than the heat transfer rate $\dot{Q}_{\mathrm{k}}$. In applications below the temperature of the environment,

$T_k < T_0$, and $\dot{E}_{q,k}$ and $\dot{Q}_k$ have opposite signs: When energy is supplied to the system, exergy is removed from it and vice versa. For steady-flow systems, $\frac{dE_{CV}}{dt} = 0$, and Equation 4.27 becomes

$$0 = \sum \left(1 - \frac{T_0}{T_k}\right)\dot{Q}_k - \left(\dot{W} - P_0 \frac{dV_{CV}}{dt}\right) + \sum \dot{E}_i - \sum \dot{E}_e - \dot{E}_D \tag{4.29}$$

4.3.3 Exergetic Efficiency

Dimensionless criteria are used for performance evaluations. Appropriately defined exergetic efficiency unambiguously characterizes the performance of a system or system component from the thermodynamic view point. The exergetic efficiency should also be used to compare the performance of similar components operating under similar conditions. For the comparison of dissimilar components the exergy destruction ratio may be used.

The exergetic efficiency of the *k*th component ε_k is defined as the ratio between product and fuel. The exergy rates of product $\dot{E}_{P,k}$ and the fuel $\dot{E}_{F,k}$ are defined by considering the desired result produced by the component, and the exergetic resources expended to generate this result, respectively:

$$\varepsilon_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} = 1 - \frac{\dot{E}_{D,k} + \dot{E}_{L,k}}{\dot{E}_{F,k}} \tag{4.30}$$

The definition of exergetic efficiency must be meaningful from both the thermodynamic and the economic view points. General guidelines for defining exergetic efficiencies can be found in the literature [7,9,16,31,35,36,51,160,163]. A distinction between (a) physical and chemical exergy, or (b) thermal, mechanical and chemical exergy, or (c) thermal mechanical, reactive and non-reactive exergy may allow the definitions of more rational exergetic efficiencies for some components.

For the comparison of dissimilar components operating in the same system, modified exergetic efficiency can be defined based on the avoidable and unavoidable exergy destruction concept:

$$\varepsilon_{\mathrm{k}}^{*} = \frac{\dot{E}_{\mathrm{P,k}}}{\dot{E}_{\mathrm{F,k}} - \dot{E}_{\mathrm{D,k}}^{\mathrm{UN}}} = 1 - \frac{\dot{E}_{\mathrm{D,k}}^{\mathrm{AV}} + \dot{E}_{\mathrm{L,k}}}{\dot{E}_{\mathrm{F,k}} - \dot{E}_{\mathrm{D,k}}^{\mathrm{UN}}} \tag{4.31}$$

4.3.4 Exergy Destruction Ratio and Exergy Loss Ratio

In addition to the exergy destruction $\dot{E}_{\mathrm{D,k}}$ and the exergetic efficiencies ε_{k}, the exergy destruction ratio $y_{\mathrm{D,k}}$ is used in the thermodynamic evaluation of a component. This ratio compares the exergy destruction in the *k*th component with the total fuel exergy supplied $\dot{E}_{\mathrm{F,total}}$ to the overall system:

$$y_{\mathrm{D,k}} = \frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{F,total}}} \tag{4.32}$$

Alternatively, the exergy destruction rate of the *k*th component can be compared to the total exergy destruction rate $\dot{E}_{\mathrm{D,total}}$:

$$y_{\mathrm{D,k}}^{*} = \frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{D,total}}} \tag{4.33}$$

The exergy loss ratio is defined similarly to Equation 4.32, by comparing the exergy loss to the total fuel exergy supplied to the overall system

$$y_{\mathrm{L,total}} = \frac{\dot{E}_{\mathrm{L,total}}}{\dot{E}_{\mathrm{F,total}}} \tag{4.34}$$

The difference between the exergy destruction ratio and the exergetic efficiency is that in the former the exergy destruction within a component is related to the fuel exergy supplied to the overall system, whereas the latter refers the same exergy destruction to the fuel exergy supplied to the component. The exergy destruction ratio expresses the percentage of the decrease of the exergetic efficiency for the overall system caused by the exergy destruction in the *k*th system component:

$$\varepsilon_{\mathrm{total}} = \frac{\dot{E}_{\mathrm{P,total}}}{\dot{E}_{\mathrm{F,total}}} = 1 - \frac{\dot{E}_{\mathrm{D,total}} + \dot{E}_{\mathrm{L,total}}}{\dot{E}_{\mathrm{F,total}}} = 1 - y_{\mathrm{D,total}} - y_{\mathrm{L,total}} \tag{4.35}$$

Since in almost every case no exergy loss is defined at the component level, the exergy loss ratio is defined only for the overall system.

4.4 Performance Assessment Parameters in Cogeneration Systems

Human beings need to develop a criterion for every decision making process. To judge the feasibility or usefulness of a cogeneration system, we use some parameters, which for every system accept a numerical value. The magnitude of this numerical value is the key for assessing the capabilities of the system under consideration. This is the fundamental approach in every engineering decision making process.

There are several performance assessment parameters of cogeneration systems in literature. Huang [164] describes ten of these parameters: fuel-utilization efficiency, efficiency of power generation, fuel chargeable to power, power-to-heat ratio, energy saving index, fuel energy saving ratio, fuel saving rate, second law efficiency, economic efficiency and PURPA efficiency. Among these parameters, fuel utilization efficiency is the most widely used parameter. However, power to heat ratio and second law efficiency (exergetic efficiency) are stated to be the most useful parameters by Huang.

The concept of Fuel utilization efficiency is based on the assumption that the unit energy carried out by the process heat is equally valuable as the unit energy carried out by the produced work or electricity. In other words, any unit amount of energy transferred for useful application by the system is taken into account with equal value. Fuel utilization efficiency is defined to be the ratio of the energy output rate of the cycle, which is used either as a process heat or electricity, to the energy input rate of the fuel employed by the cogeneration system:

$$\mathrm{FUE} = \frac{\left(\dot{W}_{\mathrm{el}}\right)_{\mathrm{plant}} + \Delta H_{\mathrm{process}}}{\dot{m}_{\mathrm{fuel}} \cdot LHV_{\mathrm{fuel}}} \tag{4.36}$$

where $\left(\dot{W}_{\text{el}}\right)_{\text{plant}}$ is the power produced by cogeneration plant, $\Delta H_{\text{process}}$ is the process heat produced by the same plant and the term in the denominator is the total fuel energy given to the cogeneration plant.

Between the two outputs of a cogeneration system, the value of electricity is higher than that of the process heat. This observation leads us to a simple conclusion: The larger electricity we produce for the same amount of process heat, the better the performance of the cogeneration system is. Hence, the ratio of electricity to the process heat of the cycle provides valuable information for the comparison of different cogeneration system designs. Mathematically power to heat ratio is expressed as

$$\text{PHR} = \frac{\left(\dot{W}_{\text{el}}\right)_{\text{plant}}}{\Delta H_{\text{process}}} \tag{4.37}$$

Fuel utilization efficiency, which is also called the first law efficiency, has the following weakness: Energy cannot always be exported from a system in the form of work. However, the very foundation of the science of thermodynamics is based on the observation that energy and work are not entirely interchangeable. To remove the weakness of the first law efficiency, a new efficiency needs to be defined for the cogeneration system. This new definition, not only employs the first law of thermodynamics, but also the second law. To define the second law efficiency, the various outputs of the cogeneration system, namely the process heat and the electricity, are measured in terms of their capabilities to produce work. The amount of work that the input fuel can ultimately produce is also evaluated. The ratio of the output to the input, measured in terms of exergy rate gives the second law efficiency (i.e., exergetic efficiency) as

$$\varepsilon_{\text{PLANT}} = \frac{\left(\dot{W}_{\text{el}}\right)_{\text{plant}} + \Delta\dot{E}_{\text{process}}}{\dot{E}_{\text{fuel}}} \tag{4.38}$$

Where $\Delta\dot{E}_{\text{process}}$ exergy of the process heat is produced and $\dot{E}_{\text{fuel}}$ is the total fuel exergy given to the cogeneration plant.

4.5 Conclusions

In this chapter, we provided general formulations for mass, energy, and exergy analyses of energy systems as well as performance assessment parameters of cogeneration. These formulations will be used in thermodynamic analysis of diesel powered cogeneration systems. The detailed formulations in component level will be provided in later chapters after the cogeneration system operation is explained.

CHAPTER 5

THERMOECONOMIC ANALYSIS

5.1 Introduction

The increasing demand for natural resources by current energy conversion technologies and the concern for the impact on the environment due to emission, waste disposal and signs of global warming have brought about the creation of new disciplines that help to understand how to improve the design and operation of energy systems and prevent residues from damaging the environment.

Thermoeconomics (i.e. exergoeconomics) is, in its widest possible sense, the science of natural resources saving that connects physics and economics by means of the second law of thermodynamics. It is the branch of engineering that combines exergy analysis and economic principles to provide system designer or operator with information not available through conventional energy analysis and economic evaluations but crucial to the design and operation of a cost-effective system [51].

The production process of a complex energy system can be analyzed in terms of its economic profitability and efficiency with respect to resource consumption. An economic analysis can calculate the cost of fuel, investment, operation and maintenance for the total plant or even individual components but provide no means on how to allocate costs among them and its products. On the other hand, thermodynamic analysis allows us to calculate the efficiencies of the individual processes of the plant. It locates and quantifies the irreversibilities but it cannot evaluate their significance in terms of the overall production process. Thermoeconomic analysis combines economic and thermodynamic analysis by applying the concept of cost, originally an economic property, to exergy. Most analysts agree that exergy is an adequate thermodynamic property to which we allocate cost because it accounts for the quality of energy [47,53]. The exergy balance accounts for the degradation of the exergy. The input exergy into a process will always be greater than the exergy output:

$$\textit{Exergy Input} - \textit{Exergy Output} = \textit{Irreversibilities} > 0$$

This expression shows that there are irreversibilities during a process. There is an implicit classification of the flows crossing the boundary of the system: the flows that are the production objective, the resources required to carry out the production and those that are residual. This information is not implicit in the second law and is the most important conceptual leap separating and at the same time uniting physics with economics. The following equation

$$\textit{Resources (F)} - \textit{Products (P)} = \textit{Residues (R)} + \textit{Irreversibilities (I)} > 0$$

is of outmost importance because it places "*purpose*" in the heart of thermodynamics. The concept of efficiency defined as

$$\textit{Efficiency} = \textit{Product} / \textit{Resource}$$

is older than thermodynamics and measures the quality of a process. The desire to produce a certain product is external to the system, and must be defined beforehand. Once this has been done, the design of the system and its functional structure will fit the aim of using available resources (capital, raw material, man power). Every definition of efficiency demands a comparison of the product obtained with the resources needed to obtain it. Its inverse value is

$$\textit{Unit Consumption} = \textit{Resource} / \textit{Product}$$

This expression is also a definition of the unit average cost when resources refer to the overall plant instead of individual processes. This concept is the key of thermoeconomics. A logical chain of concepts can be established (see Figure 5.1) which allows connecting physics with economics.

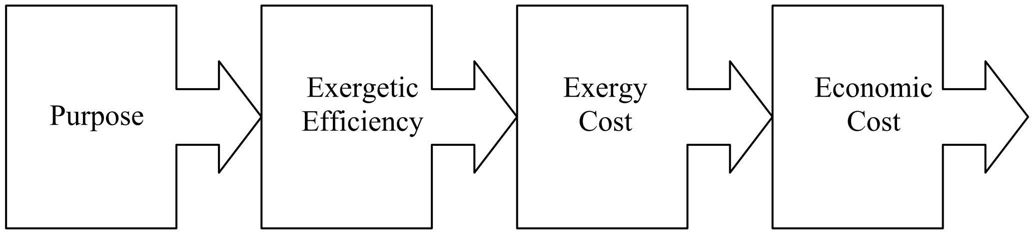

Figure 5.1 Logical chain of thermoeconomic concepts

Thus, thermoeconomics assesses the cost of consumed resources, money and system irreversibilities in terms of the overall production process. They help to point out how resources may be used more effectively in order to save them. Money costs express the economic effect of inefficiencies and are used to improve the cost effectiveness of production processes. Assessing the cost of the flow streams and processes in a plant helps to understand the process of cost formation, from the input resources to the final products.

These analyses can solve problems related to complex energy systems that could not be solved by using conventional energy analyses. Among other applications thermoeconomics are used for:

- Rational prices assessment of plant products based on physical criteria.
- Optimization of specific process unit variables to minimize the final product cost, i.e. global and local optimization.
- Detection of inefficiencies and calculation of their economic effects in operating plants, i.e. plant operation thermoeconomic diagnosis.
- Evaluation of various design alternatives, operation decisions and profitability maximization.
- Energy audits.

5.2 Economic Analysis

The successful completion of a thermal design project requires estimation of the major costs involved in the project [e.g. total capital investment, fuel costs, operating and maintenance (O & M) expenses, and cost of the final products] considering various assumptions and predictions referring to the economic, technological, and legal environments, and using techniques from engineering economics [51].

One of the most important factors affecting the selection of a design option for a thermal system is the cost of the final products. The cost of an item is the amount of money paid to acquire or produce it. The market price of an item is, in general, affected not only by the production cost of the item and the desired profit but also by other factors such as demand, supply, competition, regulation and subsidies.

The total cost of an item consists of fixed costs and variable costs. The term "fixed costs" identifies those costs that do not depend strongly on the production rate. Costs for depreciation, taxes on facilities, insurance, maintenance, and rent belong to this category. "Variable costs" are those costs that vary more or less directly with the volume of output. These include the costs for materials, labor, fuel, and electric power [166].

Good cost estimation is a key factor in successfully completing a design project. Cost estimates should be made during all stages of design to provide a basis for decision making at each stage. Each company has its own preferred approach for conducting an economic analysis and calculating the cost of main products (i.e. unit price of electricity and steam).

5.2.1 Time Value of Money

Decisions about capital expenditures generally require consideration of the earning power of money. A dollar in hand today is worth more than a dollar received one year from now because the dollar in hand now can be invested for the year. Thus, as the cost evaluation of a project requires comparisons of money transactions at various points in time, we need methods that will enable us to account for the value of money over time.

Future Value: If "P" dollars (present value) are deposited in an account earning "i" percent interest per time period and the interest is compounded at the end of each of "n" time periods, the account will grow to a future value, "F"

$$F = P(1+i)^{\mathrm{n}} \tag{5.1}$$

Interest is the compensation paid for the use of borrowed money. The interest rate is usually stated as a percentage; in equations, however, it is expressed as a decimal (e.g., 0.07 instead of 7%). Instead of the term interest rate, we will use the terms rate of return for an investment made and annual cost of money for borrowed capital [167].

Compounding Frequency: In engineering economy, the unit of time is usually taken as the year. If compounding occurs "*p*" times per year ($p \geq 1$) for a total number of "*n*" years ($n \geq 1$), and "*i*" is the annual rate of return, Equation 5.1 becomes

$$F = P\left(1+\frac{i}{p}\right)^{\mathrm{np}} \tag{5.2}$$

Here the product "np" is the number of periods and "*i/p*" is the rate of return per period. In this case, the annual rate of return "*i*" is known as the nominal rate of return. The effective rate of return is the annual rate of return that would yield the same results if compounding were done once a year instead of "*p*" times per year. The effective rate of return, which is higher than the nominal rate of return, is obtained by eliminating *F/P* from Equations 5.1 and 5.2 as

$$i_{\mathrm{eff}} = \left(1+\frac{i}{p}\right)^{\mathrm{p}} - 1 \tag{5.3}$$

If continuous compounding of money ($p \to \infty$) is used, the future value is calculated from

$$F = Pe^{\mathrm{in}} \tag{5.4}$$

It is apparent that in the case of continuous compounding the effective rate of return becomes

$$i_{\mathrm{eff}} = e^{\mathrm{i}} - 1 \tag{5.5}$$

In Equations 5.4 and 5.5, "i" is the nominal annual rate of return and "n" is the total number of years. If the time is less than one year, the simple interest formula can be used to calculate the future value:

$$F = P(1 + ni_{\mathrm{eff}}) \tag{5.6}$$

where "n" is now a fraction of a year and "i_{eff}" is the annual effective rate of return. Equations 5.2 and 5.4 can be expressed in the same form as Equation 5.1:

$$F = P(1 + i_{\mathrm{eff}})^{\mathrm{n}} \tag{5.7}$$

The term $(1 + i_{\mathrm{eff}})^{\mathrm{n}}$, referred to as the single – payment compound amount factor (SPCAF).

Unless otherwise indicated, the terms interest, rate of return, and annual cost of money refer to their effective values. Also, to simplify calculations, when the cost of money is calculated for one or more years plus a fractional part of a year, Equation 5.7 is applied with a non-integer exponent [166].

Present Value: When evaluating projects, we often need to know the present value of funds that we will spend or receive at some definite periods in the future. The present value (or present worth) of a future amount is the amount that if deposited at a given rate of return and compounded would yield the actual amount received at a future date. From Equation 5.7 we see that a given future amount F has a present value P:

$$P = F \frac{1}{(1 + i_{\mathrm{eff}})^{\mathrm{n}}} \tag{5.8}$$

The term $1/(1 + i_{\mathrm{eff}})^{\mathrm{n}}$, called the single – payment present – worth factor or the single – payment discount factor (SPDF). Since the difference between the future value and the present value is often called discount, in this case the term i_{eff} is called the effective discount rate.

Annuities: An annuity is a series of equal amount money transactions occurring at equal time intervals (periods). Usually, the time period corresponds to one year. Money transactions of this type can be used, for instance to pay off a debt or accumulate a desired amount of capital. Annuities are used in this study to calculate the levelized costs of the final product, fuel, and so forth. An annuity term is the time from the beginning of the first time interval to the end of the last time interval.

If A dollars are deposited at the end of each period in an account earning i_{eff} percent per period (effective rate of return per period), the future sum F (amount of the annuity or future value of the annuity) acquired at the end of the n$^{\mathrm{th}}$ period is

$$F = A\frac{\left(1+i_{\mathrm{eff}}\right)^{\mathrm{n}}-1}{i_{\mathrm{eff}}} \tag{5.9}$$

The term $\left[\left(1+i_{\mathrm{eff}}\right)^{\mathrm{n}}-1\right]/i_{\mathrm{eff}}$ is called the uniform – series compound – amount factor (USCAF), and the reciprocal term of it is called the uniform – series sinking fund factor (USSFF). By combining Equations 5.8 and 5.9, we obtain

$$\frac{P}{A} = \frac{\left(1+i_{eff}\right)^{\mathrm{n}}-1}{i_{eff}\left(1+i_{eff}\right)^{\mathrm{n}}} \tag{5.10}$$

The expression on the right side of this equation is called the uniform – series present – worth factor (USPWF). The reciprocal of this factor is the capital recovery factor (CRF):

$$\mathrm{CRF} = \frac{A}{P} = \frac{i_{\mathrm{eff}}\left(1+i_{\mathrm{eff}}\right)^{n}}{\left(1+i_{\mathrm{eff}}\right)^{n}-1} \tag{5.11}$$

The CRF is used to determine the equal amounts A of a series of n money transactions, the present value of which is P.

Capitalized Cost: An asset (e.g., a piece of equipment) of fixed – capital cost C_{FC} will have a finite economic life of n years. The economic life (or book life) of an asset is the best estimate of the length of time that the asset can be used. The salvage value of an asset is the estimated economic worth of the asset at the end of its economic life.

Engineers often want to determine the total cost of an asset under conditions permitting perpetual replacement of the asset without considering inflation. The so-called capitalized cost C_K is defined in engineering economics as the first cost of the asset plus the present value of the indefinite annuity that corresponds to the perpetual replacement of the asset every *n* year. Assuming that the renewal cost of the asset remains constant (no inflation) at $C_{FC} - S$, and that both the useful life of the asset and the rate of return remain constant, the present value of the indefinite annuity is calculated from Equation 5.8 as [51]

$$(C_K - C_{FC}) = (C_K - S)/(1 + i_{eff})^n \tag{5.12}$$

That is, the capitalized cost C_K is in excess of the fixed – capital cost C_{FC} by an amount which, when compounded at an effective rate of return i_{eff} for *n* years, will have a future value of C_K minus the salvage value *S* of the asset. Solving the last equation for C_K, we obtain the capitalized cost as

$$C_K = \left[C_{FC} - \frac{S}{(1 + i_{eff})^n} \right] \left[\frac{(1 + i_{eff})^n}{(1 + i_{eff})^n - 1} \right] \tag{5.13}$$

The second factor in square brackets on the right side of the equation is called the capitalized – cost factor (CCF). The capitalized – cost factor is equal to the capital – recovery factor of an ordinary annuity (Equation 5.11) divided by the effective rate of return.

The use of the term capitalized cost is more meaningful in accounting than in engineering economics where the term merely characterizes a special case of present – value calculation referring to an infinite project life. However, because the term capitalized cost is encountered very often in the literature of both engineering economics and accounting, it is important to be familiar with the different meanings that may be attached to it [51,167].

5.2.2 Inflation, Escalation, and Levelization

Inflation: General price inflation is the rise in price levels associated with an increase in available currency and credit without a proportional increase in available goods and services of equal quality [51]. The consumer price index, which is tabulated by the government, is composite prices index that measures general inflation.

When inflation occurs, costs change every year. Cost changes in past years are considered using appropriate cost indices. For future years a varying annual inflation rate can be used, but such a rate always represents a prediction. For simplicity we assume a constant average annual inflation rate (r_i) for future years.

Escalation: The real escalation rate of expenditure is the annual rate of expenditure change caused by factors such as resource depletion, increased demand, and technological advances [168]. The first two factors lead to a positive real escalation rate whereas the third factor results in a negative rate. The real escalation rate (r_r) is independent and exclusive of inflation.

The nominal (or apparent) escalation rate (r_n) is the total annual rate of change in cost and includes the effects of both real escalation rate and inflation:

$$(1+r_\mathrm{n}) = (1+r_\mathrm{r})(1+r_\mathrm{i}) \tag{5.14}$$

To simplify calculations, we assume that all costs except fuel costs and the values of by-products change annually with the constant average inflation rate r_i; that is, we take $r_\mathrm{r} = 0$. Since fuel costs are expected over a long period of future years to increase on the average faster than the predicted inflation rate, a positive real escalation rate for fuel costs may be appropriate for the economic analysis of thermal systems.

Levelization: Cost escalation applied to an expenditure (e.g., fuel costs or O&M costs) over n-year period results in a non-uniform cost schedule in which the expenditure at any year is equal to the previous year expenditure multiplied by $(1+r_\mathrm{n})$, where r_n is the constant rate of change, the nominal escalation rate. The constant – escalation levelization factor (CELF) is used to express the relationship between the value of expenditure at the beginning of the first year (P_0) and an equivalent annuity (A), which is now called a levelized value. The levelization factor depends on both the effective annual cost – of – money rate, or discount rate i_{eff} and the nominal escalation rate r_{n}:

$$\frac{A}{P_0} = \mathrm{CELF} = \frac{k\left(1-k^{\mathrm{n}}\right)}{1-k}\mathrm{CRF} \tag{5.15}$$

where

$$k = \frac{1+r_{\mathrm{n}}}{1+i_{\mathrm{eff}}} \tag{5.16}$$

and the variables CRF and r_{n} are determined from Equations 5.11 and 5.14, respectively. Equation 5.15 assumes that all transactions are made at the end of their respective years and (P_0) is the cost at the beginning of the first year.

The concept of levelization is general and is defined as the use of time – value – of – money arithmetic to convert a series of varying quantities to a financially equivalent constant quantity (annuity) over a specified time interval. We will apply the concept of levelization to calculate the levelized fuel and O&M costs, the levelized total revenue requirements and the levelized total cost of the main product of a thermal system [51].

In the economic analysis of the thermal systems, the annual values of carrying charges, fuel costs, raw water costs, and operating and maintenance (O&M) expenses supplied to the overall system are the necessary input data. However these cost components may vary significantly within the economic life. Therefore, levelized annual values for all cost components should be used in the economic analysis and evaluations of the overall system. The levelized cost is given by [49]

$$A = \mathrm{CRF}\sum_{\mathrm{m}=1}^{\mathrm{n}} P_{\mathrm{m}} = \frac{i_{\mathrm{eff}}\left(1+i_{\mathrm{eff}}\right)^{\mathrm{n}}}{\left(1+i_{\mathrm{eff}}\right)^{\mathrm{n}}-1}\sum_{\mathrm{m}=1}^{\mathrm{n}} P_{\mathrm{m}} \tag{5.17}$$

where

$$P_{\mathrm{m}} = C_{\mathrm{m}} \frac{1}{(1 + i_{\mathrm{eff}})^{\mathrm{m}}} \tag{5.18}$$

The cost rate associated with the capital and O&M expenses for the *k*th component of a thermal system is

$$\dot{Z}_{\mathrm{k}} = \frac{\mathrm{CC_L}}{\tau} \frac{\mathrm{PEC_k}}{\sum_{\mathrm{k}} \mathrm{PEC_k}} + \frac{\mathrm{OMC_L}}{\tau} \frac{\mathrm{PEC_k}}{\sum_{\mathrm{k}} \mathrm{PEC_k}} \tag{5.19}$$

The first term in the nominator of the right hand side of the equation gives $\dot{Z}_{\mathrm{k}}^{\mathrm{CI}}$, and the second term gives $\dot{Z}_{\mathrm{k}}^{\mathrm{OM}}$. The levelized cost rate of the expenditure (fuel, raw water) supplied to the overall system is

$$\dot{\mathrm{C}}_{\mathrm{EX}} = \frac{\mathrm{EXC_L}}{\tau} \tag{5.20}$$

5.2.3 Time Assumptions

In an economic analysis, all available cost numbers (e.g., land costs, total plant facilities investment, other outlays, O&M costs, fuel costs, and by-product values) must be escalated to the date they are expended. In the evaluation of economic analysis, the following assumptions are made:

- Land costs incur at the beginning of the first year of the design and construction period.
- The total capital investment is allocated to the individual years of the design and construction period. The expenditures for each year are incurred in the middle of the year.
- The startup costs are expended in the middle of the last year of design and construction.
- The working capital and the costs of licensing, research, and development are escalated to the end of the last year of design and construction.

- The allowance for funds used during construction (AFUDC) is paid annually during the design and construction period; the sum of AFUDC is calculated at the end of last year of this period.
- The costs of fuel, operation, and maintenance are incurred in the middle of each year of the system economic life.
- The revenues from the sale of products are received in the middle of each year of the system economic life.

For economic analysis of a thermal system, system engineer or plant operator must register the date of reference for each cost number and specify (a) the beginning and length of the design and construction period, (b) the anticipated economic life, and (c) the life for tax purposes. The beginning of commercial operation (beginning of economic-life period) is assumed to coincide with the end of the design and construction period [169].

5.2.4 Depreciation

Depreciation reflects the fact that the value of an asset tends to decrease with age (or use) due to physical deterioration, technological advances, and other factors that ultimately will lead to the retirement of the asset. In addition, depreciation is a mechanism for repaying the original amount obtained from debt holders if the debt is to be retired. Finally, depreciation is an important accounting concept serving to reduce taxes during plant operation. In that respect, depreciation is not strictly related to the physical or economic lifetime of an asset. The asset life used for tax purposes (as determined by statute) could be shorter than the asset's anticipated economic life.

There are many methods for depreciating the value of an asset. Some of these methods – straight line, sum of the years digits, and declining balance methods – give no consideration to interest costs, whereas others (sinking fund and present worth methods) take into account the interest on investment. Table 5.1 summarizes the mathematical relationships that can be used to calculate the depreciation allocation at the end of a year of the property life, and the cumulative depreciation allocation at the end of a year. The difference between the original cost of a property and the cumulative depreciation at the end of a year is defined as the book value at the end of that year.

5.2.5 Financing and Required Returns on Capital

The money to cover the total capital requirement of an investment can come through the following sources:

- Borrowing capital, for instance by selling bonds (debt financing)
- The sale of common and preferred stock (equity financing)
- Existing fund of the company (self – financing)
- A combination of these

Table 5.1 Summary of selected tax depreciation methods [51]

Method	Depreciation Allocation at the End of Year z (DP_z)	Cumulative Depreciation Allocation at the End of Year z $CDP_z = \sum_{i=1}^{z} DP_i$
Straight line	$\frac{C_0 - S}{n}$	$(C_0 - S)\frac{z}{n}$
Sum of the years digits	$(C_0 - S)\left[\frac{2(n+1-z)}{n(n+1)}\right]$	$(C_0 - S)\left[\frac{z(2n+1-z)}{n(n+1)}\right]$
Double declining balance $(n \geq 3)$	$C_0\left(\frac{2}{n}\right)\left(\frac{n-2}{n}\right)^{z-1}$	$C_0\left[1-\left(\frac{n-2}{n}\right)^{z}\right]$
125 % declining balance	$C_0\left(\frac{1.25}{n}\right)\left(\frac{n-1.25}{n}\right)^{z-1}$	$C_0\left[1-\left(\frac{n-1.25}{n}\right)^{z}\right]$
Sinking fund	$(C_0 - S)\left[\frac{i(1+i)^{z-1}}{(1+i)^{n}-1}\right]$	$(C_0 - S)\left[\frac{(1+i)^{z}-1}{(1+i)^{n}-1}\right]$

C_0 = total depreciable investment (TDI) at the beginning of the economic life period (dollars)
S = salvage value of the property at the end of the (tax or economic) life considered in the depreciation (dollars)
n = tax life or economic life considered in the depreciation calculations (years)
i = interest rate (decimal ratio)
z = attained age of the property (years)

The average cost of money in a project depends on the fractions of the total capital requirement financed through debt, preferred stock, and common stock and on the required return on each type of financing.

The average rate of the cost of money (discount rate) calculated in this way is the before – tax rate. The after – tax discount rate (i_{at}) reflects the effect of the deductibility of debt return on the government income tax calculation for the company, and is calculated from

$$i_{\mathrm{at}} = i - f_{\mathrm{d}} i_{\mathrm{d}} t \tag{5.21}$$

where *i* is the before – tax discount rate, f_{d} and i_{d} represent the fraction of the total capital requirement financed through debt, and the corresponding rate of return, respectively, and *t* is the total income tax rate [51].

5.2.6 Fuel, Operating, and Maintenance Costs

Fuel costs are usually part of the operating and maintenance costs. However, because of the importance of fuel costs in cogeneration systems fuel costs are considered separately from the O&M costs. The O&M costs can be divided into fixed and variable costs. The fixed O&M costs are composed of costs for operating, labor, maintenance labor, maintenance materials, overhead, administration and support, distribution and marketing, research and development, and so forth. The variable operating costs depend on the average annual system capacity factor, which determines the equivalent average number of hours of system operation per year at full load.

The fuel costs and the variable operating costs can be easily calculated from the flow diagrams. Once we know the flow of a raw material stream or of a utility, we simply multiply the flow by its unit cost and by the average total time of operation per year to obtain the contribution of the flow being considered to the total annual costs.

5.2.7 Taxes and Insurance

Income taxes are calculated by multiplying the income tax rate by the taxable income, which is the difference between total revenue and all tax-deductible expenditures. Income tax rates have varied significantly in recent years [51].

Tax deductible expenditures include fuel costs, O&M charges, return on debt, and investment cost recovery (depreciation calculated for tax purposes). In any year of the economic life of a system, the difference between the income taxes actually paid and the income taxes that would have been paid if a straight-line depreciation had been used is called the deferred income tax.

Depending on the location, the annual property taxes are usually between 1% and 4% of the plant facilities investment [167]. The annual insurance costs are typically between 0.5% and 1.5% of the plant facilities investment. Design engineers can contribute to a reduction in insurance costs by understanding the different types of insurance available, the legal responsibilities of a company with regard to accidents and emergencies, and other factors that must be considered in obtaining adequate insurance.

5.3 Thermoeconomic Analysis

Cost accounting in a company is concerned primarily with (a) determining the actual cost of products or services, (b) providing a rational basis for pricing goods and services, (c) providing a means for allocating and controlling expenditures, and (d) providing information on which operating decisions may be based and evaluated [51]. This frequently calls for the use of cost balances. In a conventional economic analysis, a cost balance is usually formulated the overall system operating at steady state

$$\dot{C}_{\mathrm{P,TOT}} = \dot{C}_{\mathrm{F,TOT}} + \dot{Z}_{\mathrm{TOT}}^{\mathrm{CI}} + \dot{Z}_{\mathrm{TOT}}^{\mathrm{OM}} \tag{5.22}$$

The cost balance expresses that the cost rate associated with the product of the system $\dot{C}_{\mathrm{P}}$ equals the total rate of expenditures made to generate the product, namely the fuel cost rate $\dot{C}_{\mathrm{F}}$ and the cost rates associated with capital investment $\dot{Z}^{\mathrm{CI}}$ and operating and maintenance $\dot{Z}^{\mathrm{OM}}$. When referring to a single stream associated with a fuel or product, the expression fuel stream or product stream is used. The rates $\dot{Z}^{\mathrm{CI}}$ and $\dot{Z}^{\mathrm{OM}}$ are calculated by dividing the annual contributions of capital investment and the annual operating and maintenance (O&M) costs, respectively, by the number of time units (usually hours or seconds) of system operation per year. The sum of these two variables is denoted by $\dot{Z}$

$$\dot{Z} = \dot{Z}^{\mathrm{CI}} + \dot{Z}^{\mathrm{OM}} \tag{5.23}$$

5.3.1 Exergy Costing

Cost may be defined as the amount of resources needed to obtain a functional product. On one hand, resources take a general meaning. On the other hand, cost is associated with the purpose of production. It is associated neither with price nor with the resources that could be saved if the production process were less efficient or more conventional one [165]. Cost is an emergent property. It cannot be measured as a physical magnitude of a flow stream as temperature or pressure; it depends on the system structure and appears as an outcome of the system analysis. Therefore, it needs precise rules for calculating it from physical data. Cost is a property that cannot be found in the product itself [51,67].

In thermoeconomics, the words history, degradation, exergy, quality, cost, resource, consumption, purpose and causality are related to one another. In the cost formation process, it is essential to analytically search for the locations and physical mechanisms that make up a specific productive flow [170]. The resources are used to provide physico-chemical qualities to the intermediate products until a finished product is obtained. The main problem to be solved using exergy is how to measure and homogenize the accounting of these qualities.

Since exergy measures the true thermodynamic value of the effects associated with heat, work and mass interactions through systems, it is meaningful to use exergy as a basis for assigning costs in thermal systems. Indeed, thermoeconomics rests on the notion that exergy is the only rational basis for assigning costs to the interactions that a thermal system experiences with its surroundings and to the sources of inefficiencies within it. This approach is referred as “exergy costing”.

In exergy costing a cost is associated with each exergy stream. Thus for entering and exiting streams of matter with associated rates of exergy transfer, power and the exergy transfer rate associated with heat transfer may be written, respectively as

$$\dot{C}_{\mathrm{i}} = c_{\mathrm{i}}\dot{E}_{\mathrm{i}} = c_{\mathrm{i}}(\dot{m}_{\mathrm{i}}e_{\mathrm{i}}) \tag{5.24}$$

$$\dot{C}_{\mathrm{e}} = c_{\mathrm{e}}\dot{E}_{\mathrm{e}} = c_{\mathrm{e}}(\dot{m}_{\mathrm{e}}e_{\mathrm{e}}) \tag{5.25}$$

$$\dot{C}_{\mathrm{w}} = c_{\mathrm{w}}\dot{W} \tag{5.26}$$

$$\dot{C}_{\mathrm{q}} = c_{\mathrm{q}}\dot{E}_{\mathrm{q}} \tag{5.27}$$

where c_{i}, c_{e}, c_{w}, and c_{q} denote average costs per unit of exergy of material stream at inlet and exit, power and heat respectively and $\dot{C}_{\mathrm{i}}$, $\dot{C}_{\mathrm{e}}$, $\dot{C}_{\mathrm{w}}$ and $\dot{C}_{\mathrm{q}}$ are the corresponding cost rates, $\dot{E}_{\mathrm{i}}$ and $\dot{E}_{\mathrm{e}}$ are exergy transfers for entering and exiting streams of matter, $\dot{W}$ is power, and $\dot{E}_{\mathrm{q}}$ is the exergy transfer rate associated with heat transfer.

Accordingly, for a component receiving heat transfer and generating power, we may write [10,51,57]

$$\sum_{\mathrm{e}}\left(c_{\mathrm{e}}\dot{E}_{\mathrm{e}}\right)_{\mathrm{k}} + c_{\mathrm{w,k}}\dot{W}_{\mathrm{k}} = c_{\mathrm{q,k}}\dot{E}_{\mathrm{q,k}} + \sum_{\mathrm{i}}\left(c_{\mathrm{i}}\dot{E}_{\mathrm{i}}\right)_{\mathrm{k}} + \dot{Z}_{\mathrm{k}} \tag{5.28}$$

This equation simply states that the total cost of the exiting exergy streams equals the total expenditure to obtain them: the cost of the entering exergy streams plus the capital and other costs. Note that when a component receives power (as in a compressor or a pump) the second term of the left hand side would move with its positive sign to the right side of this expression. Cost balances are generally written so that all terms are positive.

The exergy rates exiting and entering the k^{th} component are calculated using exergy relations in Chapter 4. The term $\dot{Z}_k$ may be obtained by first calculating the capital investment and operating and maintenance (O&M) costs associated with the k^{th} component and then computing the levelized values of these costs per unit of time (year, hour, or second) of system operation. Based on these costs the general equation for the cost rate $(\dot{Z}_i)$ in \$/s associated with capital investment and the maintenance costs for the k^{th} component is

$$\dot{Z}_k = \frac{Z_k (CRF)\varphi}{(N \times 3600)} \tag{5.29}$$

where Z_k is the purchase cost of the k^{th} component (\$), *CRF* is the annual capital recovery factor; *N* is the number of hours of plant operation per year, and φ is the maintenance factor.

When two or more products, by-products and residues are produced simultaneously, how costs can be allocated? Indeed, the main problem of allocating costs has been to find a function that adequately characterizes every one of the internal flows in a system and distributes cost proportionally. This function needs to be universal, sensitive and additive. That is, it needs to have an objective value for every possible material manifestations and it needs to vary when these manifestations do so and each internal flow property needs to be represented additively. There is a wide international consensus that the best function, at least for energy systems, is exergy, which can contain in its own analytical structure of the flow history [165,171].

5.3.2 Aggregation Level for Applying Exergy Costing

For calculating approximate average costs, we can stop our analysis by disaggregating our system at not very detailed level since the level at which the cost balances are formulated affects the results of a thermoeconomic analysis. Cumulative exergy consumption analysis does not go into process details but focuses on the overall exergy consumption.

Accordingly, in thermal design, it is recommended that the lowest possible aggregation level be used [51,163,170,171]. This level is usually represented by the individual components (compressors, turbines, heat exchangers etc.). Even in cases where the available information is insufficient for applying exergy costing at the component level, it is generally preferable to make appropriate assumptions that enable exergy costing to be applied at the component level than to consider only groups of components [51].

5.4 Thermoeconomic Variables for Component Evaluation

The following quantities, known as thermoeconomic variables, play a central role in the thermoeconomic evaluation and optimization of thermal systems:

- the average unit cost of fuel, $c_{\mathrm{F,k}}$ (i.e. $c_{\mathrm{F,k}} = \dfrac{\dot{C}_{\mathrm{F,k}}}{\dot{E}_{\mathrm{F,k}}}$)
- the average unit cost of product, $c_{\mathrm{P,k}}$ (i.e. $c_{\mathrm{P,k}} = \dfrac{\dot{C}_{\mathrm{P,k}}}{\dot{E}_{\mathrm{P,k}}}$)
- the cost rate of exergy destruction, $\dot{C}_{\mathrm{D,k}}$
- the relative cost difference, r_{k}
- the exergoeconomic factor, f_{k}

In this chapter, three of these variables are discussed: $\dot{C}_{\mathrm{D,k}}$, r_{k}, and f_{k} while all five thermoeconomic variables are applied to the thermoeconomic analysis and evaluation of the diesel engine powered cogeneration system (see Chapter 9).

5.4.1 Cost of Exergy Destruction

In the cost balance formulas (i.e. Equations 5.22 and 5.28), there is no cost term directly associated with exergy destruction. Accordingly, the cost associated with the exergy destruction in a component or process is a hidden cost, but very important one, that can be revealed only through thermoeconomic analysis. Using the specific exergetic costs associated with fuel, product, and exergy loss for the *k*th component, the cost rate balance can be written as

$$c_{P,k}\dot{E}_{P,k} = c_{F,k}\dot{E}_{F,k} - \dot{C}_{L,k} + \dot{Z}_k \tag{5.30}$$

Using Equation 4.20 from Chapter 4, in order to eliminate $\dot{E}_{F,k}$, we obtain

$$c_{P,k}\dot{E}_{P,k} = c_{F,k}\dot{E}_{P,k} + \left(c_{F,k}\dot{E}_{L,k} - \dot{C}_{L,k}\right) + \dot{Z}_k + c_{F,k}\dot{E}_{D,k} \tag{5.31}$$

or to eliminate $\dot{E}_{P,k}$, we obtain

$$c_{P,k}\dot{E}_{P,k} = c_{F,k}\dot{E}_{F,k} + \left(c_{P,k}\dot{E}_{L,k} - \dot{C}_{L,k}\right) + \dot{Z}_k + c_{P,k}\dot{E}_{D,k} \tag{5.32}$$

In both Equations 5.31 and 5.32, the last term on the right hand side involves the rate of exergy destruction. Assuming that the product, $\dot{E}_{P,k}$ is fixed and that the unit cost of fuel, $c_{F,k}$ of the *k*th component is independent of the exergy destruction, the cost of exergy destruction can be expressed as

$$\dot{C}_{D,k} = c_{F,k}\dot{E}_{D,k} \tag{5.33}$$

As the fuel rate $\dot{E}_{F,k}$ must account for the fixed product rate $\dot{E}_{P,k}$, and the rate of exergy destruction rate $\dot{E}_{D,k}$, we may interpret $\dot{C}_{D,k}$ in Equation 5.33 as the cost rate of the additional fuel that must be supplied to the *k*th component.

Alternatively, assuming that the fuel $\dot{E}_{F,k}$ is fixed and that the unit cost of product $c_{P,k}$ of the *k*th component is independent of exergy destruction, we can define the cost of exergy destruction by the last term of Equation 5.32 as

$$\dot{C}_{D,k} = c_{P,k}\dot{E}_{D,k} \tag{5.34}$$

When exergy of fuel $\dot{E}_{F,k}$ is fixed, the exergy destruction $\dot{E}_{D,k}$ reduces to the product of the *k*th component $\dot{E}_{P,k}$, and therefore Equation 5.34 can be interpreted as the monetary loss associated with the loss of product.

5.4.2 Relative Cost Difference

The relative cost difference r_k for the *k*th component is defined as

$$r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} \tag{5.35}$$

The variable expresses the relative increase in the average cost per exergy unit between fuel and product of the component. The relative cost difference is a useful variable for evaluating and optimizing a system component. In an iterative cost optimization of a system, if the cost of fuel of a major component changes from one iteration to the next, the objective of the cost optimization of the component should be to minimize the relative cost difference instead of minimizing the cost per exergy unit of the product with this component.

If Equation 5.35 is rewritten for revealing the real cost sources associated with the *k*th component, using Equations 5.23 and 5.31 and taking $\dot{C}_{L,k} = 0$, we obtain

$$r_k = \frac{c_{F,k}\left(\dot{E}_{D,k} + \dot{E}_{L,k}\right) + \left(\dot{Z}_k^{CI} + \dot{Z}_k^{OM}\right)}{c_{F,k}\dot{E}_{P,k}} \tag{5.36}$$

Using the exergetic efficiency of the *k*th component, and using Equation 4.26 from chapter 4, Equation 5.36 may be written as

$$r_k = \frac{1-\varepsilon_k}{\varepsilon_k} + \frac{\dot{Z}_k^{CI} + \dot{Z}_k^{OM}}{c_{F,k}\dot{E}_{P,k}} \tag{5.37}$$

5.4.3 Exergoeconomic Factor

As Equations 5.36 and 5.37 indicate, the cost sources in a component may be grouped into two categories. The first consists of non-exergy related costs (capital investment, and operating and maintenance expenses), while the second category consists of exergy destruction and exergy loss. In evaluating the performance of a component, we want to know the relative significance of each category. This is provided by the exergoeconomic factor, f_{k} defined for the *k*th component as

$$f_{\mathrm{k}} = \frac{\dot{Z}_{\mathrm{k}}}{\dot{Z}_{\mathrm{k}} + c_{\mathrm{F,k}}\left(\dot{E}_{\mathrm{D,k}} + \dot{E}_{\mathrm{L,k}}\right)} \tag{5.38}$$

The total cost rate causing the increase in the unit cost from fuel to product is given by the denominator in Equation 5.38. Accordingly, the exergoeconomic factor expresses as a ratio the contribution of the non-exergy related cost to total cost increase. A low value of the exergoeconomic factor calculated for a major component suggests that cost savings in the entire system might be achieved by improving the component efficiency (reducing the exergy destruction) even if the capital investment for this component will increase. On the other hand, a high value of this factor suggests a decrease in the investment costs of this component at the expense of its exergetic efficiency.

5.5 The Specific Exergy Costing (SPECO) Method

The costs associated with each material and energy stream in a system are calculated with the aid of (a) cost balances written for each system component, and (b) auxiliary costing equations. Assuming that the costs of the exergy streams entering a component known, a cost balance is not sufficient to determine the costs of the exiting exergy streams when the number of exiting streams is larger than one. In this case, auxiliary costing equations must be formulated for the component being considered, the number of these equations being equal to the number of exiting streams minus one [57,167,171].

Different approaches for formulating efficiencies and auxiliary costing equations have been suggested in the literature. These approaches can be divided into two groups: (1) The exergoeconomic accounting methods [47-51,54-58,67,72] aim at the costing of product streams, the evaluation of components and systems, and the iterative optimization of energy systems; (2) The Lagrangian-based approaches [45,52,77,80,83-87] aim in optimizing the overall system and the calculation of marginal costs. In literature only total exergy values were used and the auxiliary costing equations were formulated explicitly by using assumptions derived from experience, postulates, or the purpose of the system being analyzed.

A different approach, based on the LIFO (Last In First Out) accounting principle, was presented in refs. [78,79]. In this approach, fuels, products, and costs are defined systematically registering exergy and cost additions and removals from each material and energy stream. In this way, "local average costs" are obtained since the cost per exergy unit of the exergy used in a component is evaluated at the cost at which the removed exergy units were supplied by upstream components. An automatic criterion to generate the auxiliary costing equations based on this principle can be achieved by using computer implementation and an algebraic formulation [98]. In this study, the name SPECO, specific exergy costing method, was given to this approach because of the need of using specific exergies and costs for registering all additions and removals of exergy and cost.

The basic principles of the SPECO approach were then directly applied to exergy streams instead of material and energy streams [171]. It was demonstrated that these principles are sufficient for systematically defining fuel and product of the components and for formulating the auxiliary costing equations used to calculate either average costs (AVCO approach) or local average costs (LIFO approach).

Lagrangian-based approaches, on the other side, employ mathematical techniques to arrive at costs. It can be easily demonstrated that the same cost balances and auxiliary equations used in accounting methods can be obtained through partial derivatives in the Lagrangian-based approaches.

The SPECO method consists of the following three steps:

Step 1- identification of exergy streams: Initially, a decision must be made with respect to whether the analysis of the components should be conducted using total exergy or separate forms of the total exergy of a material stream (e.g. thermal, mechanical, and chemical exergies). Considering separate exergy forms improves the accuracy of the results. However, this improvement is often marginal and not necessary for extracting the main conclusions from the exergoeconomic evaluation.

Step 2- definition of fuel and product: The product is defined to be equal to the sum of all the exergy values to be considered at the outlet (including the exergy of energy streams generated in the component) plus all the exergy increases between inlet and outlet (i.e. the exergy additions to the respective material streams) that are in accord with the purpose of the component. Similarly, the fuel is defined to be equal to all the exergy values to be considered at the inlet (including the exergy streams supplied to the component) plus all the exergy decreases between inlet and outlet (i.e. the exergy removals form the respective material streams) minus all the exergy increases (between inlet and outlet) that are not in accord with the purpose of the component.

Step 3- cost equations: Exergoeconomics rests on the notion that exergy is the only rational basis for assigning costs to the interactions a thermal system experiences with its surroundings and to the sources of inefficiencies within it [51]. All the equations given in section 5.3 are used throughout the analysis at this step.

5.5.1 The $\dot{\mathrm{F}}$ and $\dot{\mathrm{P}}$ Principles

The $\dot{\mathrm{F}}$ (fuel) principle refers to 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 $\dot{\mathrm{F}}$ principle states that the total cost associated with this removal of exergy must be equal to the cost at which the removed exergy has supplied to the same stream in the upstream components.

The $\dot{\mathrm{P}}$ (product) principle refers to the supply of exergy to an exergy stream within the component being considered. The $\dot{\mathrm{P}}$ principle states that each exergy unit is supplied to any stream associated with the products at the same average cost c_p. This cost is calculated from the cost balance and the equations obtained by $\dot{\mathrm{F}}$ principle. Aggregation level influences accuracy of the results, so it should be set at a lower level [51].

5.6 Conclusions

In this chapter, we provided general principles, terminology, and formulation of thermoeconomic analysis, which is also called exergoeconomic analysis. The procedure and formulation are applicable to all energy systems including diesel powered cogeneration systems. Detailed formulations considering the operation of the entire system and components will be provided in Chapter 8.

CHAPTER 6

THERMOECONOMIC OPTIMIZATION

6.1 Introduction

Thermal system design involves answering questions such as: What processes or equipment items should be selected and how should they be arranged? What is the preferred size of a component? What are the best temperature, pressure, flow rate, and chemical composition of each stream in the system? To answer these questions, engineers need to formulate an appropriate optimization problem.

The first step in the definition of optimization problem is to define clearly the boundaries of the system to be optimized. All the subsystems that significantly affect the performance of the system under study should be included in the optimization problem. The selection of criteria on the basis of which the system design will be evaluated and optimized is the key element in formulating an optimization problem. Optimization criteria may be *economic* (total capital investment, total annual levelized costs, annual levelized net profit), *technological* (thermodynamic efficiency, production time, production rate, fuel consumption) and *environmental* (rates of emitted pollutants). An optimized design is characterized by a minimum or maximum value, as appropriate for each selected criterion [51,89].

Another essential element in formulating the optimization problem is the selection of the *design variables* that adequately characterize the possible design options. In selecting these variables, it is necessary to include all the important variables that affect the efficiency and the cost effectiveness of the system. Each component and the system as a whole are defined by a set of quantities. Some of them are fixed by external conditions (e.g. environmental pressure and temperature, fuel price) and are called parameters. The remaining are variables, i.e. their value may change during the optimization procedure. Those variables, the values of which do not depend on another variables or parameters, are called *independent* or *design* variables. The rest can be determined by the solution of a set of *equality* constraints and they are called *dependent* variables.

The mathematical model for an optimization problem consists of:

- An objective function to be minimized
- A set of equality constraints
- A set of inequality constraints

Thermoeconomic optimization methods use a primary optimization performance measure: minimize the total levelized cost of the system products that includes the cost of external fuel resources, capital investment and maintenance cost. Also multicriteria optimization and environmental constraints can be considered.

6.2 Thermoeconomic Optimization Approaches

The balance between thermodynamic measures and capital expenditures is an economic feature, which applies to the thermal system as a whole and to each component individually. The costs of resources usually vary to the opposite direction of the cost of equipment with respect to the design variables. An improvement on the structure or the efficiency of the equipment implies a reduction of the resources consumption but an increase of the capital investment.

The *equality constraints* are provided by appropriate thermodynamic and cost models as well as appropriate boundary conditions. These models must include the flow rate and energy balances for each component, relations associated with the engineering design, such as the local efficiencies of the components. The model adopted by thermoeconomic optimization relates the input (fuels) of each component with its outputs and design variables

The model can also contain *inequality constraints* that specify the allowable operating ranges, the maximum and minimum performance requirements, and bounds on the availability of resources. When optimum is reached with only equality constraints, we obtain the shadow costs, one for each independent variable. If an inequality constraint is active in the optimum, the cost becomes an opportunity cost for the constrained variable.

There are cost optimization procedures which make no use of the exergy concept. So cost-effectiveness of every change carried out on a plant component must be assessed in terms of the overall system parameters, e.g. its effect on the consumption of fuel resources. This makes optimization very complex and computer time consuming. With thermoeconomic optimization these difficulties may be overcome. For example, with proper thermoeconomic analysis and under certain conditions, the decomposition is applicable, which facilitates the solution of the problem, because it allows the optimization problem of the whole system to be decomposed into a set of optimization problems of subsystems or components, which are of smaller dimension (i.e. they have fewer independent variables) and can be solved more easily. There was basically, at the beginning, two different thermoeconomic approaches: the *structural* method that use the local unit cost of the irreversibilities and the *autonomous* method introduced by Evans and El-Sayed [45] in 1970 that is the starting point of other state-of-the-art techniques.

There are several approaches to Thermoeconomic optimization that were presented in a set of articles in 1993, as a result of the project CGAM [48-51,58,78,79]. *The Exergoeconomic Optimization Approach*, proposed by Tsatsaronis [78] uses an iterative design improvement procedure that does not aim at calculating the global optimum of a predetermined objective function, as the conventional optimization methods do, but tries to find a "good" solution for the overall system design. The basic idea lies in a commonly accepted concept from the cost view point: at constant capacity for a well designed component, group of components, or total system, a higher investment cost should correspond to a more efficient component and vice versa.

The Functional Analysis proposed by Frangopoulos [77,80,83-85,94] and *the Engineering Functional Analysis* proposed by von Spakovsky [87,89,101] used the method of the Lagrange multipliers and decomposition procedures. Valero and coworkers, present a similar approach, but propose to use the unit average exergy costs instead of the Lagrange multipliers.

El- Sayed proposed also, in order to avoid problems with the isolation of the decision variables to divide the decision variables into local variables and global variables, in general the number of global variables is much smaller than the local variables, iterate to find the local optimum of each component respect its local variables and the global optimum respect to the global variables.

The decomposition strategy is based on the *Principle of Thermoeconomic Isolation* (TI) introduced by R. Evans in 1980 [47]: A component of a thermal system is thermo-economically isolated form the rest of the system if its production and the unit cost of the resources are known quantities and independent from the rest of the component variables. It is an ideal condition which cannot be fully achieved for most of real systems, but the more the TI conditions are fulfilled the fewer iterations are required to achieve the optimal solution for the whole system. Therefore the thermoeconomic model of the system is subdivided or decomposed into subgroups. Each subgroup is optimized in turn, according to a sequential process, iterating around the system until the system's internal economy converges, within prescribed tolerances, to a single set of values. Decomposition may only approach the global optimum since the degree of thermoeconomic isolation of the independent variables, the choice of the subgroups and their functions, and the nature of the dependent variables greatly affects how close the approach will be. Nonetheless, the advantages of this strategy facilitates the optimal design of individual units in highly interdependent complex systems, and let the designers to concentrate their efforts on designing the variables of single components, while resting assured that these efforts improve the overall system.

6.3 Cost Optimal Exergetic Efficiency for An Isolated System Component

Several mathematical approaches may be applied to optimize the design of a single system component in isolation from the remaining system components. Some of these approaches can be found in literature [76,77,80,84-90,94]. In this study, the thermoeconomic approach that illustrates clearly the connections between thermodynamics and economics are used in the analysis and evaluations of diesel engine powered cogeneration system. With this approach, the cost optimal exergetic efficiency can be obtained for a component isolated from the remaining system components.

The thermoeconomic optimization approach is based on following assumptions which are expressed analytically:

Assumption A1: The exergy flow rate of the product $\dot{E}_{\mathrm{P,k}}$, and the unit cost of the fuel $c_{\mathrm{F,k}}$ remain constant for the *k*th component to be optimized:

$$\dot{E}_{\mathrm{P,k}} = \text{constant} \tag{6.1}$$

$$c_{\mathrm{F,k}} = \text{constant} \tag{6.2}$$

These equations, which represent constraints of the optimization problem, define mathematically what is meant by isolation in Chapter 8.

Assumption A2: For every system component, we expect the investment costs to increase with increasing capacity and increasing exergetic efficiency of the component. Here, we assume that for the *k*th component the total capital investment $\mathrm{TCI_k}$ can be represented at least approximately by the following relation [79]:

$$\mathrm{TCI_k} = B_{\mathrm{k}} \left(\frac{\varepsilon_{\mathrm{k}}}{1-\varepsilon_{\mathrm{k}}} \right)^{\mathrm{n_k}} \dot{E}_{\mathrm{P,k}}^{\mathrm{m_k}} \tag{6.3}$$

where the term $\left(\frac{\varepsilon_{\mathrm{k}}}{1-\varepsilon_{\mathrm{k}}} \right)^{\mathrm{n_k}}$ expresses the effect of efficiency (i.e. thermodynamic performance), while the term $\dot{E}_{\mathrm{P,k}}^{\mathrm{m_k}}$ expresses the effect of capacity (i.e. component size) on the value of $\mathrm{TCI_k}$. The parameter B_{k} is given as constant in the cost equations of the *k*th component, $\mathrm{n_k}$ and $\mathrm{m_k}$ are expressed as efficiency and capacity exponents in cost equations respectively. Within a certain range of design options, these three terms are constant [51].

Assumption A3: Usually a part of the operating and maintenance (O&M) costs depends on the total investment costs and another part on the actual production rate. The annual levelized operating and maintenance costs attributed to the *k*th component are represented as [76]

$$Z_k^{OM} = \gamma_k(TCI_k) + \omega_k \tau \dot{E}_{P,k} + R_k \tag{6.4}$$

In Equation 6.4, γ_k is a coefficient that accounts for the part of the fixed O&M costs depending on the total capital investment associated with the *k*th component. In large conventional electric power plants, an average value for the coefficient γ_k of 0.015×CELF may be assumed for all plant components where CELF is the constant-escalation levelization factor. For relatively small thermal systems, the coefficient γ_k can be taken as high as 0.10 [79]. ω_k is a constant that accounts for the variable O&M costs associated with the *k*th component and denotes the O&M cost per unit of product exergy, τ is the average annual time of plant operation at the nominal load; and R_k includes all the remaining O&M costs that are independent of the total capital investment and the exergy of the product.

Assumption A4: The economic analysis of the system being considered is simplified by neglecting the effects of financing, inflation, taxes, insurance, and construction time and by considering the startup costs, working capital, and the costs of licensing, research, and development together with the total capital investment. The annual carrying charge associated with the *k*th component is then obtained by multiplying the total capital investment for this component TCI_k by the capital recovery factor, β:

$$Z_k^{CI} = \beta(TCI_k) \tag{6.5}$$

Assumptions A1 through A4 (Equations 6.1 through 6.5) form the "cost model". The total annual levelized costs excluding fuel costs associated with the *k*th component are obtained by combining Equations 6.4 and 6.5

$$Z_k = Z_k^{CI} + Z_k^{OM} = (\beta + \gamma_k)(TCI_k) + \omega_k \tau \dot{E}_{P,k} \tag{6.6}$$

The corresponding cost rate $\dot{Z}_k$ is obtained by dividing Equation 6.6 by τ,

$$\dot{Z}_k = \frac{\beta + \gamma_k}{\tau}(TCI_k) + \omega_k \dot{E}_{P,k} + \frac{R_k}{\tau} \tag{6.7}$$

From Equation 6.3,

$$\dot{Z}_k = \frac{(\beta + \gamma_k)B_k}{\tau}\left(\frac{\varepsilon_k}{1-\varepsilon_k}\right)^{n_k} \dot{E}_{P,k}^{m_k} + \omega_k \dot{E}_{P,k} + \frac{R_k}{\tau} \tag{6.8}$$

The objective function to be minimized expresses the cost per exergy unit of the product for the *k*th component. Accordingly, using Equations 5.31 from Chapter 5 and taking $\dot{C}_{L,k} = 0$, we can write

$$\text{Minimize}\, c_{P,k} = \frac{c_{F,k}\dot{E}_{F,k} + \dot{Z}_k}{\dot{E}_{P,k}} \tag{6.9}$$

Using Equation 4.26 from Chapter 4, and Equation 6.8, this objective function may be expressed as

$$\textit{Minimize}\, c_{P,k} = \frac{c_{F,k}}{\dot{E}_{P,k}} + \frac{(\beta + \gamma_k)B_k}{\tau\, \dot{E}_{P,k}^{1-m_k}}\left(\frac{\varepsilon_k}{1-\varepsilon_k}\right)^{n_k} + \omega_k + \frac{R_k}{\tau\, \dot{E}_{P,k}} \tag{6.10}$$

The values of parameters $\beta, \gamma_k, B_k, \tau, \omega_k$, and R_k remain constant during optimization process, and so $c_{P,k}$ varies only with ε_k [51,76,79]. Thus the optimization problem reduces to the minimization of Equation 6.10 subject to the constraints expressed by Equations 6.1 and 6.2. The minimum cost per exergy unit of product is obtained by differentiating Equation 6.10 and setting the derivative to zero:

$$\frac{dc_{P,k}}{d\varepsilon_k} = 0 \tag{6.11}$$

The resulting cost-optimal exergetic efficiency [76] is

$$\varepsilon_k^{OPT} = \frac{1}{(1 + F_k)} \tag{6.12}$$

where

$$F_k = \left(\frac{(\beta + \gamma_k)B_k n_k}{\tau c_{F,k}\, \dot{E}_{P,k}^{1-m_k}}\right)^{1/(n_k+1)} \tag{6.13}$$

Equations 6.12 and 6.13 show that the cost-optimal exergetic efficiency increases with increasing cost per exergy unit of fuel, $c_{F,k}$, increasing annual number of hours of system operation τ, decreasing capital recovery factor β, decreasing fixed O&M cost factor γ_k, and decreasing cost exponent n_k. Equation 6.12 may be rewritten as

$$F_k = \frac{1-\varepsilon_k^{OPT}}{\varepsilon_k^{OPT}} \tag{6.14}$$

or using Equations 4.20 and 4.26 from Chapter 4,

$$F_k = \left(\frac{\dot{E}_{D,k} + \dot{E}_{L,k}}{\dot{E}_{P,k}}\right)^{OPT} \tag{6.15}$$

Since the exergy rate of the product is assumed constant during optimization, the cost-optimal value of the sum $\left(\dot{E}_{D,k} + \dot{E}_{L,k}\right)$ is given by

$$\left(\dot{E}_{D,k} + \dot{E}_{L,k}\right)^{OPT} = \dot{E}_{P,k} F_k = \dot{E}_{P,k}\left(\frac{1-\varepsilon_k^{OPT}}{\varepsilon_k^{OPT}}\right) \tag{6.16}$$

At this point, a simplification of assumption A3 allows some additional results to be obtained: In Equations 6.4 (and in Equations 6.6, 6.7, 6.8 and 6.10) we may neglect the last two terms on the right side referring to a certain portions of the O&M costs since these costs are often small compared with the remaining terms on the same side of the respective equation [76,79]. With this simplification and using Equation 4.26, Equation 6.10 can be expressed in terms of $\left(\dot{E}_{D,k} + \dot{E}_{L,k}\right)$ as

$$\text{Minimize } c_{P,k} = c_{F,k}\left(1 + \frac{\dot{E}_{D,k} + \dot{E}_{L,k}}{\dot{E}_{P,k}}\right) + \frac{(\beta + \gamma_k) B_k}{\tau\, \dot{E}_{P,k}^{1-m_k}}\left(\frac{\dot{E}_{P,k}}{\dot{E}_{D,k} + \dot{E}_{L,k}}\right)^{n_k} \tag{6.17}$$

By differentiating Equation 6.17 with respect to $\left(\dot{E}_{D,k} + \dot{E}_{L,k}\right)$ and setting the derivative to zero, we obtain after some manipulation the following relation between the cost-optimal values of the cost rates expressed by $c_{F,k}\left(\dot{E}_{D,k} + \dot{E}_{L,k}\right)$ and $\dot{Z}_k$:

$$n_{\mathrm{k}} = \frac{c_{\mathrm{F,k}}\left(\dot{E}_{\mathrm{D,k}} + \dot{E}_{\mathrm{L,k}}\right)^{\mathrm{OPT}}}{\dot{Z}_{\mathrm{k}}^{\mathrm{OPT}}} \tag{6.18}$$

Thus, under assumptions A1, A2, A4 and the simplified assumption A3, when the *k*th component is optimized in isolation, the cost exponent n_k in Equations 6.3, 6.8, 6.10 and 6.15 express the ratio between the cost optimal rate associated with exergy destruction and exergy loss and the cost-optimal rate associated with capital investment.

Using Equations 6.16 and 6.18, we obtain the following expressions for the cost-optimal values of the non-fuel related cost rate $\dot{Z}_{\mathrm{k}}$, the relative cost difference r_{k}, Equation 5.37, the exergoeconomic factor f_{k}, and Equation 5.38, we obtain

$$\dot{Z}_{\mathrm{k}}^{\mathrm{OPT}} = c_{\mathrm{F,k}}\dot{E}_{\mathrm{P,k}}\frac{F_{\mathrm{k}}}{\mathrm{n_k}} \tag{6.19}$$

$$r_{\mathrm{k}}^{\mathrm{OPT}} = \frac{\mathrm{n_k}+1}{\mathrm{n_k}}F_{\mathrm{k}} \tag{6.20}$$

$$f_{\mathrm{k}}^{\mathrm{OPT}} = \frac{1}{1+\mathrm{n_k}} \tag{6.21}$$

The use this optimization approach we must be able to express the total capital investment of a system component as a function of exergetic efficiency and the capacity through a relation similar to Equation 6.3. In Chapter 8, these definitions are given for all components of the diesel engine powered cogeneration system for which meaningful exergetic efficiencies are defined.

6.4 Thermoeconomic Optimization Methodology of Existing Complex Systems

The usual approach to the optimization of complex thermal systems is to iteratively optimize subsystems and/or ignore the influence of some structural changes and decision variables. An alternative to this approach is an iterative thermoeconomic optimization technique that consists of the following steps:

1. In the first step, the detailed schematics and inputs of the existing system must be evaluated. The use of actual data, vendor's quotations (even contractor's actual operating manuals) can reduce, therefore, the total number of iterations required.
2. A detailed thermoeconomic analysis and evaluations are conducted for the actual system with the data taken in the previous step. In this step, we can easily obtain the decision variables that affect both the exergetic efficiency and the investment costs.
3. If the system has one or two components for which the sum of the cost rates $\left(\dot{Z}_{\mathrm{k}} + \dot{C}_{\mathrm{D,k}}\right)$ is significantly higher than the same sum for the remaining components, the improvements of these components can be modified to approach their corresponding cost-optimal exergetic efficiency, given by Equation 6.12. This is meaningful only for components where each of the terms $\dot{Z}_{\mathrm{k}}$ and $\dot{C}_{\mathrm{D,k}}$ has a significant contribution to the costs associated with the respective component.
4. For the remaining components, particularly the ones having a relatively high value of the sum $\left(\dot{Z}_{\mathrm{k}} + \dot{C}_{\mathrm{D,k}}\right)$, the relative deviations of the actual values from the cost-optimal values for the exergetic efficiency and relative cost difference are calculated:

$$\Delta\varepsilon_{\mathrm{k}} = \frac{\varepsilon_{\mathrm{k}} - \varepsilon_{\mathrm{k}}^{\mathrm{OPT}}}{\varepsilon_{\mathrm{k}}^{\mathrm{OPT}}} \times 100 \tag{6.22}$$

$$\Delta r_{\mathrm{k}} = \frac{r_{\mathrm{k}} - r_{\mathrm{k}}^{\mathrm{OPT}}}{r_{\mathrm{k}}^{\mathrm{OPT}}} \times 100 \tag{6.23}$$

5. Finally a parametric study may be conducted to investigate the effect on the optimization results of some parameters and/or assumptions made in the optimization procedure.

6.5 Conclusions

In this chapter, we provided general principles and formulation of thermoeconomic optimization. The detailed formulation for diesel powered cogeneration is provided and applied to an existing system in Chapter 8.

CHAPTER 7

SANKO DIESEL ENGINE POWERED COGENERATION

7.1 Introduction

The term “cogeneration” characterizes energy conversion processes, in which energy is generated for a dual purpose, usually to produce both electricity and heat. The heat output is conventionally in the form of steam or hot water. Such dual purpose processes can substantially improve the efficiency of energy utilization for electrical power generation.

Reciprocating engines such as diesel and gasoline engines have been used for power generation almost 70 years. In these internal combustion engines, fuel combustion occurs within the engine cylinders, and the chemical energy created by the combustion is converted directly to mechanical work transmitted through a driven shaft to the generator. Because reciprocating engines use internal combustion, they often function at higher temperatures than do steam turbines and should theoretically be even more efficient in producing power from a given fuel-energy input [172].

In this chapter, we present a short overview of cogeneration concept and diesel engine powered cogeneration systems first, and then the detailed description of Sanko Diesel Engine Powered Cogeneration (DEPC) plant including main and spare parts of this actual cogeneration plant equipments: diesel engine, turbocharger, waste heat boiler unit, heat exchangers with different goals within the system, process flows of fuel, water, steam, lubrication oil, and exhaust gas and also exhaust gas treatment unit.

7.2 Cogeneration Overview

Cogeneration is the simultaneous generation of heat and power, both of which are used. It encompasses a range of technologies, but always includes an electricity generator and a heat recovery system. Cogeneration is also known as “combined heat and power (CHP)” and a “total energy”.

In conventional electricity generation, further losses around 5-10% are associated with the transmission of electricity from relatively remote power stations via the electricity grid. These losses are the greatest when electricity is delivered to the smallest consumers. Through the utilization of heat, the efficiency of cogeneration plant is normally used locally, and then transmission and distribution losses will be negligible. Cogeneration therefore offers energy savings ranging between 15-40% when compared against the supply of electricity and heat from conventional power stations and boilers [173].

Because transporting electricity over long distances is easier and cheaper than transporting heat, cogeneration installations are usually sited as near as possible to the place where the heat is consumed and, ideally, are built to a size to meet the heat demand. This is the central and most fundamental principle of cogeneration. When less electricity is generated than needed, it is necessary to buy extra. However, when the scheme is sized according to the power demand, normally more electricity than needed is generated. The surplus electricity can be sold to the grid or supplied to another customer via the distribution system [174].

Cogeneration uses a single process to generate both electricity and usable heat or cooling. The proportions of power and heat needed (power/heat ratio) vary from site to site, so the type of plant must be selected carefully and an appropriate operating regime must be established to match demands as closely as possible. The plant may therefore be set up to supply part or all of the cite electricity and heat loads, or an excess of either may be exported if a suitable customer is available. Cogeneration plant consists of four basic elements: a prime mover (engine), an electricity generator, a heat recovery system and a control system. Depending on site requirements, the prime mover may be a steam turbine, reciprocating engine or gas turbine. In the future new technology options will include micro-turbines, Stirling engines and fuel cells. The prime mover drives the electricity generator and usable heat is recovered.

Cogeneration plants are available to provide outputs from 1 kW to 500 MW. For larger scale applications (greater than 1 MW), there is no standard cogeneration kit: equipment is specified to maximize cost-effectiveness for each individual site. For small-scale cogeneration applications equipment is normally available in packaged units, helping to simplify installations [172].

7.2.1 Cogeneration in Industrial Plants

Industrial cogeneration schemes are typically located on sites that have a high demand for process heat and electricity all year. Suitable examples are found in the refining, paper, chemicals, oil, greenhouses and textile sectors. The bulk of cogeneration capacity on industrial sites comes from schemes of over 1 MW, and these tend to be designed on an individual basis to meet the specific requirements of each application. A much larger number of industrial sites have smaller systems, using technologies similar to cogeneration systems used in buildings and commerce.

Industrial cogeneration installations can operate for 8000 hours/year or more. Therefore, in industrialized countries, the heat potential in industry is large enough to enable cogeneration to provide a significant proportion of –or in some cases all of- the base load demand for electricity.

Industrial cogeneration contributes a relatively by large share to Turkey power generation. Combined heat and power generation has developed rapidly owing to the past favorable legal framework ad the existence of heat demand in the energy-intensive industries. Abolishing the favorable legal framework recently and increasing oil and gas prices have caused financial difficulties for the CHP operators. The Turkish government is drafting new energy efficiency legislation part of which will be aligning the CHP policies to the recent EU – CHP Directive [142].

In any possible future policies and measures the government may consider for the promotion of cogeneration, cost-effectiveness should be a driving force. Inefficient cogeneration or CHP that does not bring real fuel savings and emissions reductions as compared to state-of-the-art separate electricity generation and heat production should not be promoted.

7.2.2 Diesel Engine Powered Cogeneration (DEPC)

The reciprocating engines used in cogeneration applications are internal combustion engines operating on the same familiar principles as their gasoline and diesel engine automotive counterparts. Although conceptually the system differs very little from that of gas turbines, there are important differences. Reciprocating engines give a higher electrical efficiency, but it is more difficult to use the thermal energy they produce, since it is generally at lower temperatures and is dispersed between exhaust gases and engine cooling systems [173].

The compression ignition (i.e. diesel) engines for large scale cogeneration are predominantly four stroke direct injection machines fitted with turbochargers and intercoolers. Diesel engines can be fuelled with diesel oil, heavy fuel oil and natural gas. The natural gas is in a dual fuel mode, as a small quantity of diesel oil (about 5% of the total heat input) has to be injected with the natural gas to ensure ignition; as the engine can also run on diesel oil only it is suited to interruptible gas supplies. Shaft efficiencies are 35 to 45% and output range is up to 65 MW. Compression-ignition engines run at speeds of between 500 and 1500 rev/min. In general, engines up to about 0.5 MW (and sometimes up to 2 MW) are derivatives of the original automotive diesels, operating on diesel oil and running at the upper end of their speed range. Engines from 0.5 MW to 20 MW evolved from marine diesels and are dual-fuel or residual fuel oil machines running at medium to low speed.

Modern engines use delayed ignition timing and increased compression ratios to limit NO_x formation whilst maintaining high levels of power output and efficiency. This requires sophisticated fuel injection and engine management system. Although reciprocating engines can be designated to achieve TA-luft (Technical Instructions on Air Quality Control) requirements through primary reduction methods (i.e. limiting NO_x formation with the engine) larger compression ignition engines are often fuelled by heavy fuel oil. $DeNO_x$ treatment of the exhaust gases is then required to reduce emissions to acceptable levels. The scale of these installations can make the cost of this after-treatment acceptable within the plant's overall capital and operating cost.

7.3 Description of Sanko DEPC

Sanko Diesel Engine Powered Cogeneration (DEPC) plant was installed by Sanko Textile Industry and Trade Incorporated Company and M.A.N Power joint venture, in the Gaziantep Third Organized Industrial Region; in 2002. The project contract was signed between SANKO and EÜAŞ on 2001. The construction period took twelve months and on January 2002, commercial electricity generation was started at the plant. The overview of the SANKO DEPC plant is given in Figure 7.1. The total installed electricity generation and steam capacities of the plant is 25.32 MW and 8.1 tons/hr respectively.

Figure 7.1 The plant overview of Sanko DEPC

The electricity is generated by three, diesel engine actuated; generator sets (see Figure 7.2). The each engine is four-stroke compression ignition engine with 18 cylinders in a V configuration. Heavy fuel oil is being used as fuel for engines. The engine – generator sets were imported from M.A.N. Engine Company and A.B.B. Generator Company. These companies are well known largest international companies at their sectors. The permissible annual electricity production of plant is 217 GWh and the annual fuel consumption is nearly 45,000 tons at designed operating conditions. Plant operation life is to be twenty five years. The total investment of plant is nearly about 35 million US Dollars.

Figure 7.2 The diesel engine actuated generator sets in SANKO DEPC

The plant consists of three main section made of precast – insulated type. These are power house, heat recovery-flue gas treatment units, fuel oil loading and storage area. Plant is located on a 4000 m^2 area where 2030 m^2 of it is closed area. The system that are mainly installed in plant are engine-generator sets, turbocharger systems, fuel forwarding module, Heavy fuel oil (HFO) and light fuel oil (LFO) systems, lubrication oil system, compressed air system, cooling system, waste heat boiler units, fire hydrant system, PLC monitoring control system.

Two heavy fuel oil tanks with capacity of 1000 m^3 and one light fuel oil tank with capacity of 250 m^3 are used for storage of fuels (see Figure 7.3). The daily fuel consumption of plant at full load is about 130 m^3. The DEPC plant has also $DeNO_x$ treatment system in order to get rid of harmful engine emissions, formed as a result of burning heavy fuel oil. The water requirement of plant is supplied a deep water well and treated for usage.

Figure 7.3 The views of the HFO and LFO day tanks in Sanko DEPC

7.4 Engine – Generator Sets

There are three engine-generator sets inside of the power house of SANKO DEPC as shown in Figure 7.2. All sets are installed parallel and separately from each other. Each sets generates electricity independently and consists of an engine, a generator, two turbocharger units, a waste heat recovery steam boiler, an air vessel, an oil filter, a fuel booster, a charge air filter, a lubrication oil cooler, and air-water radiator unit. All of the sets are controlled from the control room by PLC systems.

The sets are designed 8000 hours per year. The sets run at 514-600 rpm which is classified as middle speed. The maximum output of each engine is 8.44 MW.

7.4.1 Engine Data (M.A.N 18 V 32/40 Diesel Engines)

The engines used in power house as actuator are named as M.A.N 18 V 32/40 Diesel engine and showed from top, front and side views in Figures 7.4, 7.5 and 7.6 respectively. Engines were manufactured by M.A.N Company at the factories in UK and Germany and imported to Turkey. The “18 V 32/40” represents number of cylinder, configuration form and, bore and stroke dimensions. The engine is four stroke cycled and turbocharged. Fuel injection system in the engine is direct injection type. HFO and LFO can be used both as fuel in this type of engine by the help of fuel booster. The compressed air, as starter, is used to start engine to run. The ambient air is compressed to 40 bars and collected inside of the air vessels. This compressed air is used as start air and as control air for pneumatic systems at a value of 10 bars.

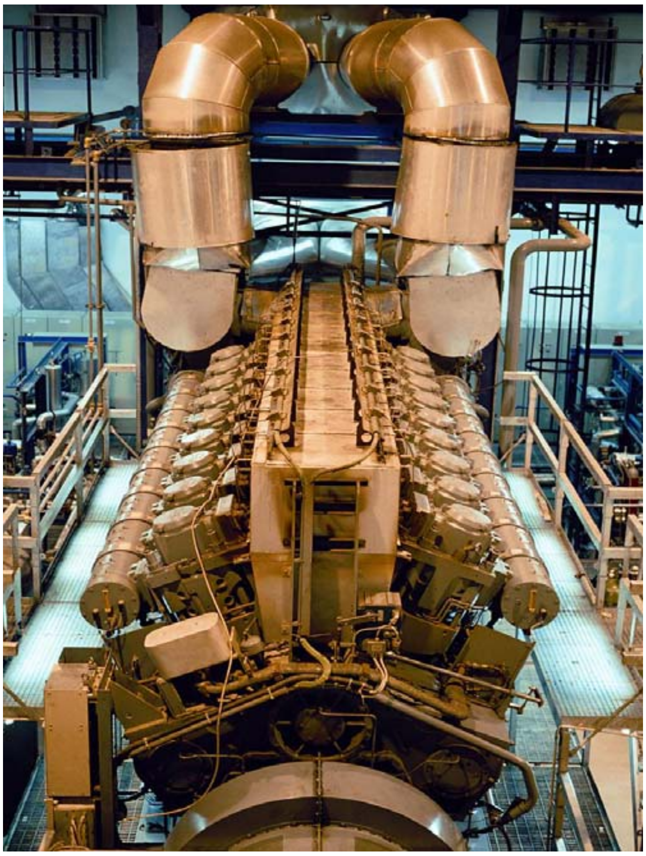

Figure 7.4 The top view of M.A.N Engine 18 V 32/40 in SANKO DEPC Plant

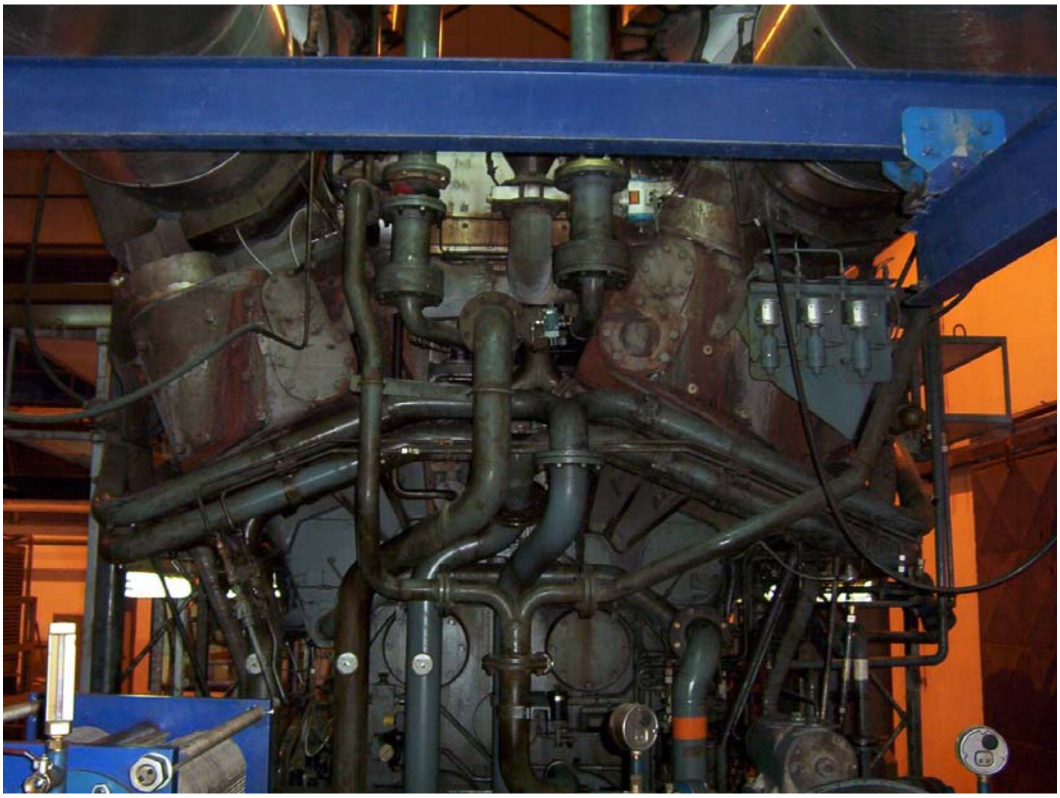

Figure 7.5 The front view of M.A.N Engine 18 V 32/40 in SANKO DEPC Plant

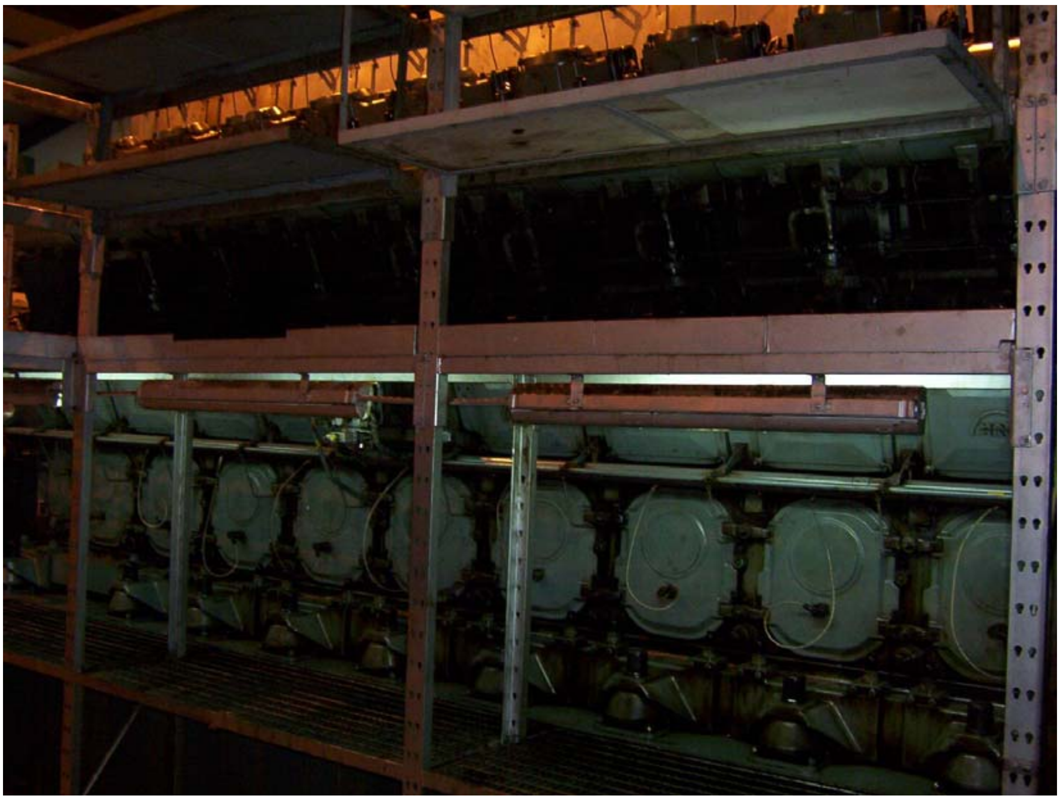

Figure 7.6 The side view of M.A.N Engine 18 V 32/40 in SANKO DEPC Plant

Since engine is turbocharged, there is an inter-cooler system for cooling of charged air before it is sent into the engine cylinders. For engine cooling, an air-water radiator system is chosen. The coolant liquid is water and cycled in a closed loop. The heat transfer at radiators is performed by usage of air fans. The basic data of the engine (i.e. M.A.N 18 V 32/40) is listed in Table 7.1.

Ignition sequence of M.A.N 18 V 32/40 Diesel engine is given in the following:
18 Cylinder engine, ignition sequence of bank *A*
Clockwise : A1-B1-A3-B3-A5-B5-A7-B7-A9-B9-A8-B8-A6-B6-A4-B4-A2-B2
Counter-Clock: A1-B2-A2-B4-A4-B6-A6-B8-A8-B9-A9-B7-A7-B5-A5-B3-A3-B1

18 Cylinder engine, ignition sequence of bank *B*
Clockwise : A1-B1-A6-B6-A3-B3-A2-B2-A8-B8-A7-B7-A4-B4-A9-B9-A5-B5
Counter-Clock: A1-B5-A5-B9-A9-B4-A4-B7-A7-B8-A8-B2-A2-B3-A3-B6-A6-B1

7.4.2 Generator (Alternator) Data

Following is the list of data describing Leroy-Somer generator (see Figure 7.7)

Type / Serial Number	: LSA 60 105-8P / 167375
Output	: 10,550 kVA
Voltage	: 11 kV
Current	: 129 A
Power Factor	: 0.80
Frequency	: 50 Hz
Weight	: 78 tones
Speed	: 750 rpm
Direction of Rotation	: CCW
Cooling Method	: Symmetric Fan Cooling
Actuator	: M.A.N B&W 18 V 32/40
Efficiency	: 93.8

Table 7.1 Basic engine characteristic data of M.A.N 18 V 32/40 [175]

Cylinder Bore		320 mm
Piston Stroke		400 mm
Crank Offset		200 mm
Number of Cylinder in Each Bank		18 V-Shape
Cylinder Distance		1000 mm
Speed		514-600 rpm
Mean Piston Speed		10 m/s
Brake Mean Effective Pressure		24.9 bar
Maximum Cylinder Pressure		200 bar
Specific Fuel Consumption		181 g/kWh
Lower Heating Value (LHV) of fuel-oil		42,700 kJ/kg
Charged Air Flowrate / One Engine		18.4 kg/s
Lube Oil Consumption		20.0 kg/s
Specific Lube Oil Consumption		0.5 g/kWh
Swept Volume/Cylinder		0.03217 m^3/cylinder
Power Output/Cylinder		469 bkW
Torque		114.6 Nm
Exhaust Mass Flowrate/One Engine		17.0 kg/s
Compression ratio		12.37
Stroke/Injection Type		4/Direct Injection
Valve Timing of M.A.N 18 V 32/40 Engine		
Inlet Valve	opens / closes	52 ° before TDC / 38 ° after BDC
Exhaust Valve	opens / closes	63 ° before BDC / 44 ° after TDC
Overlap		96 °
Starting Valve	opens / closes	2-3 ° after TDC / 116 ° after TDC

Figure 7.7 The Leroy-Somer generator in SANKO DEPC

7.4.3 Transformer Data

Following is the list of data describing Alstom transformators (see Figure 7.8):

Number of transformers	: 3 pieces
Type / Standart	: ALSTOM TCU 5036 / TS 267
Rated Power Output	: 11,000 kVA
Rated frequency	: 50 Hz
Rated Voltage	: 31.5 / 11 kVa
Off Load Losses	: 11 kW
On Load Losses	: 62 kW
Short circuit impedance	: 8.5%

Distribution Transformer 1

Type / Standart	: ALSTOM DCU 4136 / TS 267
Rated Power Output	: 1,250 kVA
Rated frequency	: 50 Hz
Rated Voltage	: 31.5 / 0.4 kVa
Off Load Losses	: 2.2 kW
On Load Losses	: 17 kW
Short circuit impedance	: 6%

Distribution Transformer 1

Type / Standart	: ALSTOM DCU 4336 / TS 267
Rated Power Output	: 2,250 kVA
Rated frequency	: 50 Hz
Rated Voltage	: 31.5 / 0.4 kVa
Off Load Losses	: 3.2 kW
On Load Losses	: 24 kW
Short circuit impedance	: 6%

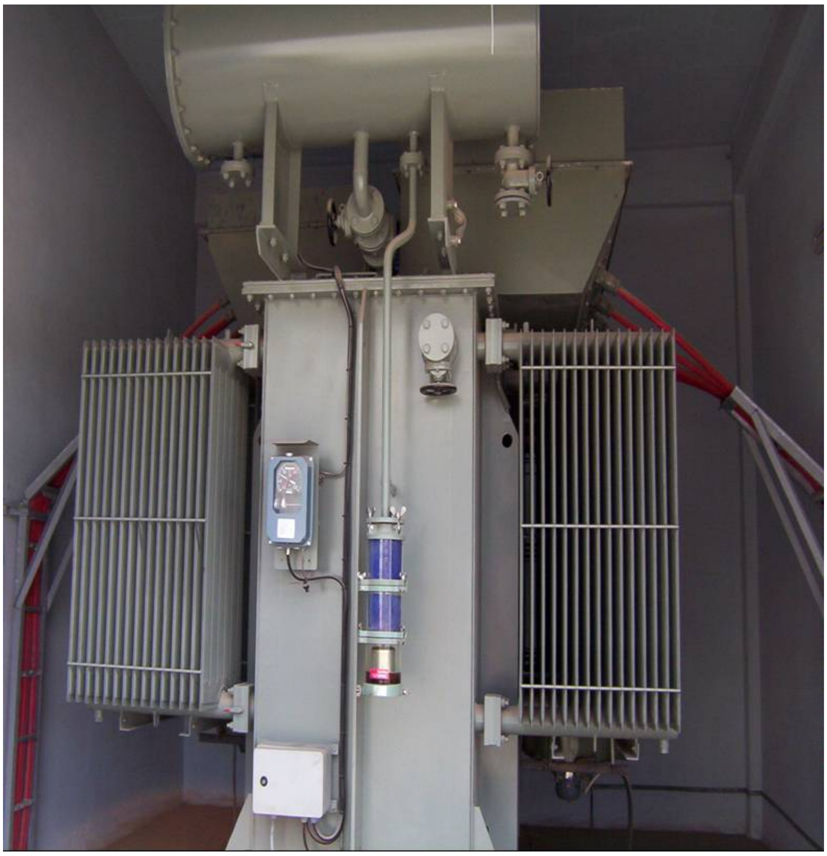

Figure 7.8 The Alstom transformers in SANKO DEPC

7.5 Charge Air and Exhaust Gas Systems

The air route: The air required for combustion of the fuel in the cylinder is drawn in axially by the compressor wheel of the turbocharger. This is done either using the intake sound damper with dry air filters or using the intake casing. Using the energy transmitted by the exhaust flow on the turbine wheel of the turbocharger the air is compressed and thus heated. The air of high energy (charge air) is field over a sliding sleeve and the double diffuser into the diffuser casing. The diffuser reduces the flow speed to the benefit of pressure. The air is cooled in the two stage charge air cooler fitted in the casing. In this way the cylinder is filled with the greatest possible mass of air. This is carried out using the charge pipe which consists of elements connected elastically with each other [175].

The exhaust route: The exhaust leaves the cylinder head on the opposite side to the charge pipe. It is collected in the exhaust manifold and fed to the turbine side of the turbocharger. Thermo elements in the cylinder heads both before and after the turbocharger are used for monitoring the temperature. The exhaust manifold consists of cylinder – length elements. The connection to the cylinder Head is made by using a clamping connection. To connect with one another and to the turbocharger corrugated tube compensators are used. The exhaust gases flow radially away from the turbine wheel [175].

Condensed water: On the casing of the charge air cooler and at the start of the charge pipe there are connected condensation water pipes. Any water occurring is led through the float wave [175].

Jet Assists: The "Jet Assists" acceleration device is fed by the 30-bar compressed air system. The flow of air is fed to the compressor casing and directed to the compressor wheel through bore holes distributed around the outsides. In this way the volume of air is increased and the turbocharger accelerated which results in the desired increase in charge pressure. The pressure and through put are set using the reducing valve and the choke cover. Control guarantees that sufficient air is available for starting procedures [175].

Charge air blower: The charge air blower is used to improve the partial load performance of the engine. When the butterfly valve is open, charge air flows through the blower pipe into exhaust pipe. This leads to an increase in turbine performance and a resulting increase in the charge pressure. The valve is activated using a control cylinder impinged with control air. The charge air relief device is used in the operation of stationary engines with excess load and it is also controlled using a butterfly valve or by a spring loaded valve. The device is used to limit the charge air pressure and the ignition pressure. The excess charge air is blown into the machine room. There is no connection here to exhaust pipe [175].

7.5.1 Turbocharger Principle Used In Engine

In reciprocating engines, engine can be equipped with a turbocharger and this process results in more air and fuel in the combustion chamber during the cycle, and the resulting net indicated work is increased. Higher intake pressures increases all pressures through the cycle, and increased air and fuel give greater heat gain in process. When air is compressed to a higher pressure by a turbocharger, the temperature is also increased due to compressive heating. This would increase air temperature at the start of the compression stroke, which in turn raises all temperatures in the remaining cycle. This can cause self-ignition and knocking problems in the latter part of compression or during combustion. For this reason, engine compressors can be equipped with an intercooler to again lower the compressed incoming air temperature. Intercoolers are heat exchangers which often use outside air as the cooling fluid (176).

The turbocharger consists of two machines, a turbine and a compressor which are mounted on a common shaft. The exhaust gas from the diesel engine flow through the gas inlet casing and nozzle ring to the turbine wheel. The turbine uses the energy contained in the exhaust gas to drive the compressor, whereby the compressor draws in fresh air, compresses it before being forced into the cylinders.

The exhaust gas exits the turbocharger via the gas outlet casing. The turbocharger is gastight. The air which is necessary for the operation of the diesel engine and which is compressed in the turbocharger is drawn through the suction branch or the silencer into the compressor wheel. It then passes through diffuser and leaves the turbocharger through the volute of the air outlet housing. The rotor runs in two radial plain bearings which are located in the bearing bush between the compressor casing and turbine casing. The axial thrust bearing is on the compressor side. The plain bearings are connected to a central lubricating oil feed in which the oil is supplied by the oil system of the engine (177). The oil outlet is always at the lowest point of the bearing casing. Side and front views of turbocharger NR34S in the Sanko Cogeneration Facility is given in Figures 7.9 and 7.10 respectively.

7.6 Fuel Oil System

The fuel is fed from a free–standing pump through a filter into the distributor pipe. From here a supply pipe branches to each fuel injection pump with a stop cock. The return of excess fuel is carried out through the manifold which is also connected through return pipes with stop cooks to the injection pumps. In this way each individual pump can be blocked from the fuel inlet and removed with out the whole pipe system having to be drained. The excess fuel flows back over the pressure control valves at the end of the manifold to the mix container.

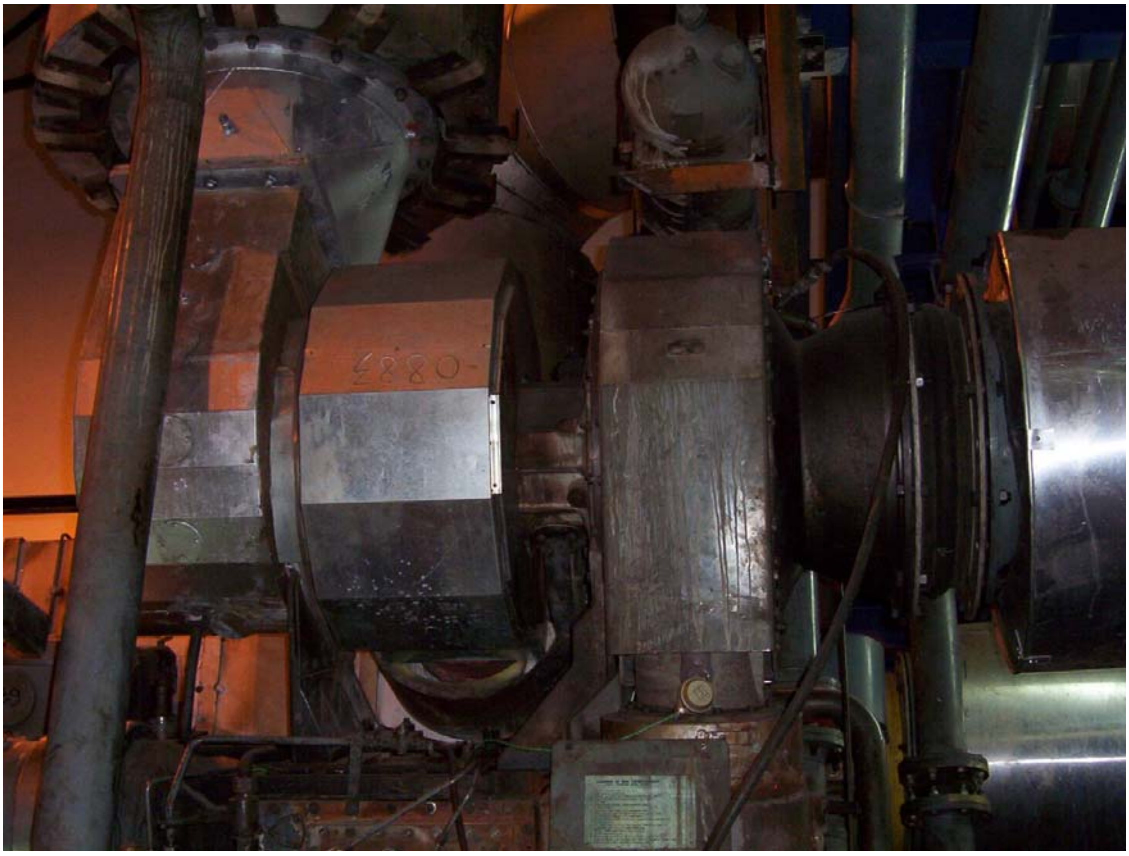

Figure 7.9 Side view of turbocharger NR34S in SANKO DEPC

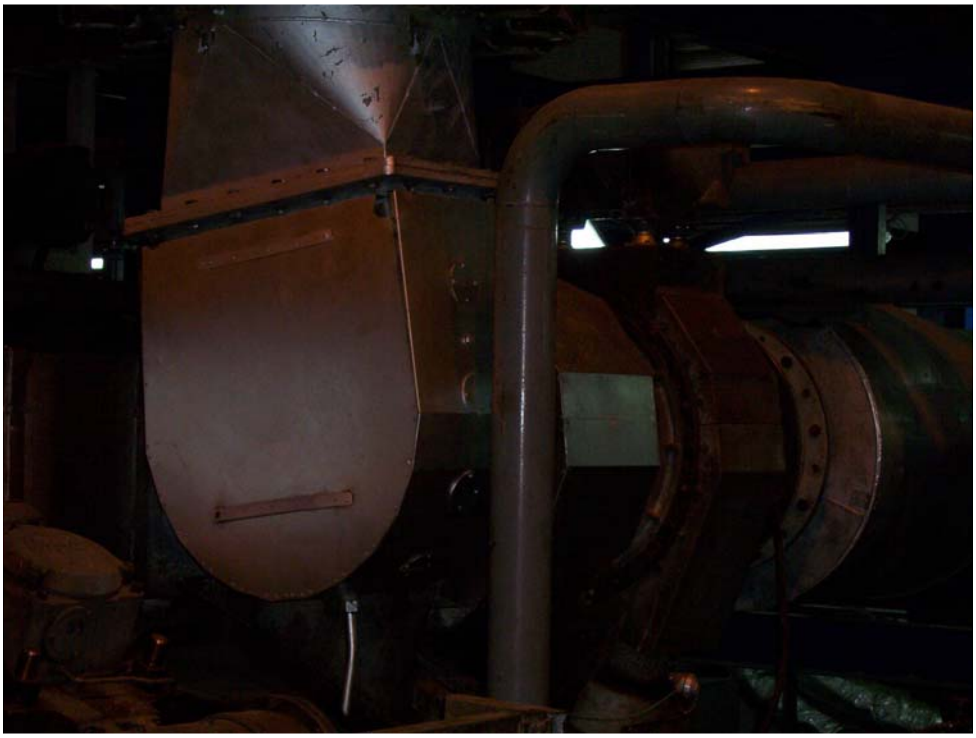

Figure 7.10 Front view of turbocharger NR34S in SANKO DEPC

This arrangement means that pre-heated fuel can be pumped around to warn the pipe system and the fuel injection pumps before starting the engine (see Figure 7.11 and 7.12). The heat pipe for the heavy oil mode arranged between the distributor and the manifold is also used for compensating heat losses. The heat return pipes serve the purpose of heating the leakage fuel pipe. The fuel injection pumps feed the fuel in the injection valves. The leakage fuel running from the injection valves and fuel injection pumps is collected in the leakage collector pipe and fed to the manifold at the foot of the fuel injection pumps [175].

Figure 7.11 The fuel forwarding module (FFM) in SANKO DEPC

Fuel oil preheating / pump ability: Using a state of–the–art final preheated a heavy fuel oil outlet temperature of 135ºC will be obtained at 8 bar from saturated steam. Higher temperatures involve the risk of increased residue formation in the preheated resulting in a reduction of the heating power and thermal overloading of the heavy fuel oil. This causes new asphalt to form, i.e. decrease of quality [175].

Figure 7.12 The fuel separator of fuel forwarding module in SANKO DEPC

Injection viscosity: The fuel pipes from the final preheated outlet up to injection valve must be insulated adequately ensuring that a temperature drop will be limited to max. 4 ºC. Only then can the prescribed injection viscosity of max.144mm^2 / s be achieved with a heavy fuel oil of a reference viscosity of 700 mm^2 / s=c_{st} / 50ºC. If a heavy fuel oil of a lower reference viscosity is used an injection viscosity of 112 mm^2/s should be aimed at, ensuring improved heavy fuel oil atomization and consequently heavy fuel oil combustion in the engine with less residues. The transfer pump is to be rated for a heavy fuel oil viscosity of up to 1000 mm^2/s. The pump ability of the heavy fuel oil also depends on the pour point. The design of the bunkering system must permit heating up of the fuel oil to approx. 10ºC above its pour points [175]. Table 7.2 gives the determination of heavy fuel oil (HFO) as a function of viscosity.

Table 7.2 Determination of HFO temperature as a function of viscosity [175]

Specified Injection viscosity, mm^2 / s	Required heavy fuel oil temperature, ºC
Minimum 12	126
Maximum 14	119

7.7 Lubrication Oil System

A lubrication oil pump (fitted on the engine or external) sucks the lubrication oil out of the storage container and squeezes it through filters, cooler and a pressure regulating valve to the distributing pipe situated between the cylinder rows (Figure 7.13 and 7.14). The oil led away from the pressure regulating valve flows back to the storage container in an overflow pipe. The main bearings are supplied with oil from the distributing pipe via a receptacle and a bore leading beyond this in the cylinder housing.

The oil flows through slanting bores in the crankshaft to the connecting rod bearings and from there through bores in the connecting rods to the gudgeon pin bearings and continues in to the cooling spaces of the piston. It flows freely from the piston in to the crankcase. The first main bearing between the coupling flange and the control drive is supplied with oil from the distributing pipe and through the distributing pipe. Oil flows on from the distributing pipe through the branch conduits to the intermediate wheel bearings and to the spray nozzle of the meshing. In addition branch conduits lead to the camshaft thrust bearings and, where this are present to the injection timing adjusting device. The pipe to the governor drive is connected to the distributing pipe of cylinder A1. In addition there is one pipe connected to the distributing pipe leading to the corresponding cylinder of series A and series B. The followings branch out from these distributing pipes per cylinder; the pipe to the injection pump drive, two pipes to the camshaft bearings, the pipe the cam followers for the valve and injection pump drives and the pipe to the rocker arms in the cylinder head with a short branch conduit to the injection pump. In addition branch conduits lead from the pipes to the bores for the crankshaft bearings bolts in the cylinder crankcase. This achieves damping of the vibrations in the long bearing bolt. In the upper area of the cylinder crankcase the oil flows out of these bores again and flows freely into the crankcase.

Figure 7.13 Lubrication oil cooler (LOC) in SANKO DEPC

The oil flowing from the rocker arm bearings collects on the respective cylinder head and flows through the push – road protecting tube into the camshaft through and from there back into the crankcase. If the turbochargers are arranged on the coupling end, supply pipe also branches out of from the distributing pipe. By means of the pressure regulating valve the oil pressure for the turbocharger is reduced. If the turbocharger is to be fitted on the opposite side to the coupling the oil supply pipe is also situated there. The oil sump serves as a collecting tank for the lubrication oil dripping from the bearing points. On the coupling end and the opposite side to the coupling on the front surface, drainage pipes are connected through which the oil flows into the storage container [175].

Figure 7.14 Lubrication oil filter system in SANKO DEPC

7.8 Cooling Water System

In order to maintain the lowest possible thermal stresses, the equipments of the diesel engine and through a separate system, the fuel injection valves must be cooled. The charge air heated up in the turbocharger is cooled down again using the charge air cooler. This serves the purpose of increasing the air mass required for combustion. Prepared fresh water is used for cooling. The charge air coolers are also cooled using fresh water. With two–stage charge air coolers the first stage has engine cooling water flow through and the second stage has fresh water from the low temperature circulation (see Fig. 7.15 and 7.16) [175].

Cooling Water System – Engine: The cylinder water is supplied to the engine by means of to distributing pipe. It is arranged in the V chamber of the engine. The supply pipe leads from this pipe to the backing ring of each cylinder. The water is carried upwards round the upper part of the cylinder liner and the top land ring in the backing ring [175].

Figure 7.15 Intercooler (IC) in SANKO DEPC

Figure 7.16 Air – water radiator units in SANKO DEPC

Quality Requirements for Engine Cooling Water: The engine cooling water like the fuel and lubricating oil is a medium which must be carefully selected, treated and controlled. Otherwise corrosion, erosion and cavitations may occur on the walls of the cooling system in contact with water and deposits may from. Deposits impair the heat transfer and may result in thermal overload on the components to be cooled. The treatment with an anti–corrosion agent has to be effected before the first commissioning of the plant. During subsequent operations the concentration specified by the engine manufacturer must always be ensured. In particular, this applies if a chemical additive is used.

7.9 Waste Heat Recovery Boiler System

The waste heat recovery system is designed to gain the heat from exhaust gases and to use this heat for steam generation (Figure 7.17). The waste heat recovery system is more significant for cogeneration plants since, they consume more amount of steam for industrial processes. On the other hand the steam generated at the mobile power plants is consumed for preheating of heavy fuel oil, for office heating and rarely for driving of small steam turbines. But the main aim at steam generation is to preheat the HFO.

The steam generation capacity of each boiler unit is 2700 kilogram steam per hour. The outside of boilers are insulated to prevent heat loses. The passage of the flue gas after boiling process is controlled by a pneumatically operated rotary bypass damper; damper opens and flue gas pass through the boiler stack. There is a second pneumatically operated bypass damper at the middle height of stack, which directs the flue gas passage through to $DeSO_x$ or directly to the atmosphere without being desulphurized. The quality of feed water at the boilers is very important for protection of steam quality and boilers' life. So feed water must be well treated and must include right dosage of chemicals. The condensed steam is collected in condense water tank, here it is filtered and mixed with necessary chemicals then let into steam boilers [175].

Figure 7.17 Waste heat recovery boiler unit in SANKO DEPC

Other characteristic of the boiler include:

- Boilers installed in vertical position
- Waste heat of exhaust gas from turbine used for heating water
- Total steam production of the cogeneration system is 8.1 ton/hr at nominal conditions.
- Gas taken into the $DeSO_x$ after boiler
- Each engine - generator set has one steam boiler
- Water for steam is treated by automatic water treatment unit and chemicals at water treatment unit.
- Steam generated is used for heat need of auxiliary equipments in the power house and textile production facility of Sanko Inc.
- Exhaust gas temperature before boilers is nearly 300 ºC , where after boilers is about 60 ºC that is below legal limits defined by emission standards for power plants in 2004 regulation by Ministry of Environment and Forestry.

7.10 Description of Exhaust Flow in the Diesel Engine Powered Cogeneration Plant

The exhaust gases leaving the engine flow through the turbine of the turbocharger unit to produce the necessary shaft work for the compressor. The exhaust gases leaving the turbine is sent to the $DeNO_x$ (denitrification) unit in which the NO_x emission is lowered to acceptable legal values by spraying urea solution onto the exhaust gases. Then, the exhaust gases enter the boiler unit to transfer heat to the condensate return and make-up water to produce saturated steam for preheating of streams in the auxiliary equipments such as fuel forwarding module and fuel and lubrication oil tanks. Finally, the exhaust gases flow through a DeSOx (desulphurization) unit before being exhausted to the atmosphere.

Figure 7.18 Exhaust gas treatment unit in SANKO DEPC

7.10.1 Denitrification Unit-$DeNO_x$

The main aim of the denitrification unit is to treat NO_x exhaust emissions as N_2 and water. During burning of fuel the molecules including nitrogen reacts with oxygen in the intake air and produce NO. If the amount of oxygen is less, then instead of NO, the product will be N_2. This process is the basic of the denitrification principle by improving the quality of combustion. For each engine-generator set one $DeNO_x$ unit is installed. Casing of the units are insulated to prevent heat transfer. NO_x emissions are treated into the nitrogen (N_2) and water (H_2O) by Selective Catalytic Reduction (SCR) process. The SCR reaction is catalyzed by vanadium/titanium catalysts at temperatures in the range 300-500ºC as shown below

$$NO_2 + NO + NH_3 \rightarrow 2N_2 + 3H_2O \tag{7.1}$$

The catalytic reactors are made of special ceramics shaped in honeycomb modules. Ammonia (NH_3) obtained from urea in liquid form is used as redactor element. The exhaust gas at 300-450ºC is transferred from cylinders to $DeNO_x$ units by insulated exhaust ducts. Exhaust gases at high temperature pass through the catalytic reactors, here NH_3 is injected on the exhaust gases, and so NO_x react with NH_3 and produce nitrogen (N_2) and water (H_2O). These two final products are natural materials and have no harmful effect to the environment. The total solid urea consumption of $DeNO_x$ unit is about 260 kg/hr, where total NO_x mass flow is 402 kg/hr. The treated flue gas should include NO_x at a total amount less than 800 mg/Nm^3 [142].

7.10.2 Desulphurization Unit-$DeSO_x$

From process point of view the proposed flue gas desulphurization system is referred to as "limestone – gypsum" process, based on the use of common limestone ($CaCO_3$) in powder form as reagent and production gypsum as by-product. The limestone consumption is nearly 750 kg/h. The waste gas purification process is performed for each gas stream into a scrubber with integrated quencher where SO_2 and other pollutants (SO_3 - HCI) are removed by washing liquid which is a suspension of limestone in water. The waste gas enters laterally above the bottom of the scrubber into the quenching zone of the scrubber. The waste air is cooled down to the saturation temperature in the quencher zone. From here the waste gas flows vertically upwards to the subsequent nozzle lever where the pollutants absorption, as well as some dust separation takes place. During the absorption of SO_2, SO_3 and HF, hardly soluble deposits such as $CaSO_4$, H_20, $CaSO_3$, and CaF_2 are formed, and suspended in the scrubbing solution. In order to prevent larger deposits in the scrubbing liquid tank, an agitator is installed in the scrubber sump. This serves at the same time for dispersing the oxidation air. The reaction that takes place during the desulphurization process is

$$CaCO_{3\,(s)} + SO_{2\,(g)} + 0.5H_2O \rightarrow CaSO_3 \,.\, 0.5H_2O + CO_{2\,(g)} \tag{7.2}$$

When the oxidation with air take place

$$CaSO_{3(s)} + 0.5H_2O_{(s)} + 1.5H_2O \rightarrow CaSO_4 \,.2H_2O_{\,(s)} \tag{7.3}$$

Due to the continuous recirculation and subsequent formation of some sulfites and sulfates a portion of re-circulated liquid is bled off to a dewatering system that provide a reduction of moisture down to 15% residual water content, in order to handle the gypsum removal. The scrubbing and cooling liquid suspension is accumulated in the scrubber lower sump from where is taken and re-circulated by suitable wear-resistant recirculation pumps. The treated flue gas should include SO_2 at a total amount less than 2400 mg/Nm^3 [142].

7.11 Conclusions

In this chapter, detailed technical information about auxiliary equipments and main flows through the system such as air, fuel oil, lubrication oil, cooling water and exhaust gases of Sanko DEPC facility is presented by using vendor quotations and contractor's guides. Data for the system operation is based on the plant description presented in this chapter. Exhaust emission characteristics of the facility are presented in Chapter 9 with some important environmental concerns and also with emission allocation methodologies.

CHAPTER 8

RESULTS AND DISCUSSION

8.1 Introduction

Combining the second law of thermodynamics with economics (i.e. exergoeconomics) using energy or available energy (i.e. exergy) for cost purposes provides a powerful tool for systematic study and optimization of complex energy systems. Its goal is to mathematically combine in a single model, the first or second law of thermodynamic analysis with the economic factors. Numerous studies have been undertaken to conduct energy, exergy and exergoeconomic analyses and optimization of thermal systems [1-130].

For combined heat and power (CHP) systems, i.e. cogeneration, these studies can be classified into three main groups: (i) energy and exergy analyses of CHP systems, (ii) exergoeconomic analysis of CHP systems and (iii) exergoeconomic optimization of CHP systems. There are numerous approaches in literature for exergy cost analysis. The SPECO (Specific Exergy Costing) method is used in this study. The most known application of the SPECO method is CGAM (a predefined and simple problem of optimization which was named after the first initials of the participating investigators: C. Frangopoulos, G. Tsatsaronis, A. Valero, M. von Spakovsky) problem [78].

In this chapter, exergy and exergoeconomic analysis and optimization of Sanko diesel engine powered cogeneration (DEPC) system is conducted using the methodologies described in earlier chapters. The results are obtained and discussed.

8.2 Description of Sanko DEPC Plant Operation

The schematic of Sanko diesel engine powered cogeneration (DEPC) plant is shown in Fig. 8.1. This plant has a total installed electricity and steam generation capacities of 25.3 MW and 8.1 tons/hr, respectively. The plant is located in the southeastern Turkey, the city of Gaziantep. It started to produce power and steam in 2002. The electricity is generated by three, diesel engine actuated generator sets each having two turbochargers. Each diesel engine–generator set in the plant produces 8.44 MW electricity and 2.7 tons/h saturated steam at 8 bars and 170°C. The schematic diagram of this plant for one engine set is shown in Figure 8.1 where only one turbocharger is demonstrated. The engine is four-stroke compression ignition engine with 18 cylinders in a V configuration. Heavy fuel oil is used as the fuel. The permissible annual electricity production is 217 GWh and the annual fuel consumption is nearly 45,000 tons at designed operating conditions.

When the engine starts, air is charged to the compressor of the turbocharger unit. The turbocharger consists of a turbine and a compressor, which are mounted on a common shaft. The shaft power obtained by the operation of diesel engine is transferred to the generator for electricity production. The exhaust gases leaving the engine flow through the turbine of the turbocharger unit to produce the necessary shaft work for the compressor. The air leaving the compressor is cooled by water in an intercooler before air enters the engine cylinders. The exhaust gases leaving the turbine enter the exhaust gas boiler unit to transfer heat to the feed water to produce steam for manufacturing facilities in the factory and for preheating of streams in the auxiliary equipments such as fuel forwarding module (FFM) and fuel oil in daily usage tank (FDT). The exhaust gas leaving the boiler is sent to the $DeSO_x$ (desulphurization) unit in which the SO_x emission is lowered to the acceptable legal values. After the $DeSO_x$ unit, the exhaust gases are released to the atmosphere.

The water used in the plant is distributed by the collectors to the exhaust gas boiler, air-water radiator (AWR) and the flue gas treatment unit. High temperature water (HT) from AWR with the low temperature water (LT) first enters through the intercooler (IC) for the cooling of compressed air before entering the engine. High temperature water from intercooler enters through the diesel engine for the cooling application of engine jacket while low temperature water from intercooler entering the lubrication oil cooler (LOC) for the cooling of lubrication oil from engine and then both unmixed water returns back to the air-water radiator unit. Oil is used for lubrication and cooling purposes of the engine components. The fuel used in the cogeneration system is stored in daily usage tanks and is preheated by steam before entering fuel forwarding module. Finally, it is injected to the engine cylinders through the nozzle in the fuel forwarding module.

In this application, July 2005 data of the plant is used (see Table 8.1). The reference (i.e. dead state) temperature and pressure of air are taken as the actual ambient conditions (30°C and 94 kPa). Other assumptions include:

a. The DEPC system operation is in the steady-state.
b. The ideal gas principles are applied to both air and exhaust gases.
c. The combustion reaction in diesel engine is complete.
d. The kinetic and potential energy changes are negligible.
e. Because the state of water in the exhaust is generally vapor in internal combustion engines, the lower heating value (LHV) of the fuel is used.

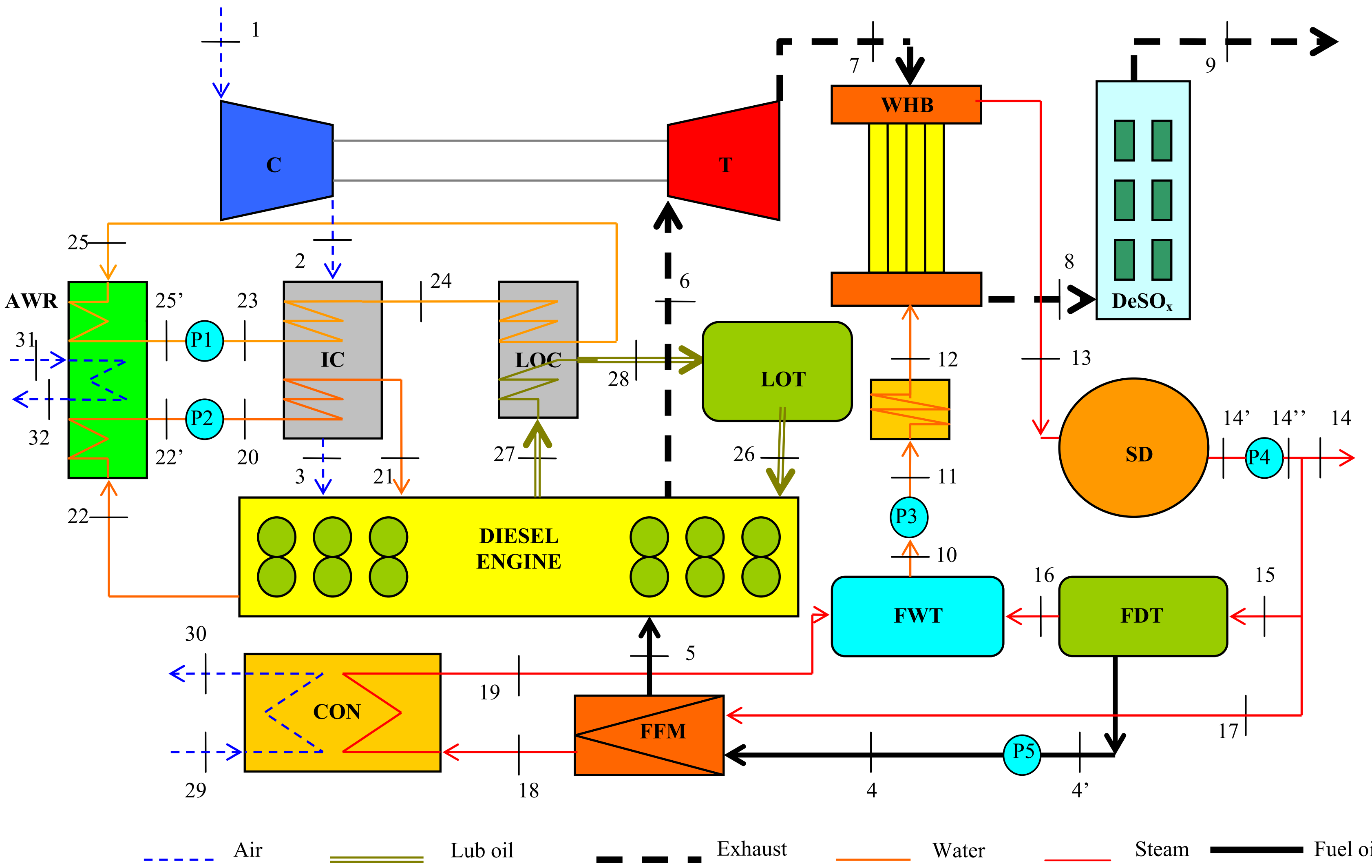

Figure 8.1 The schematic of Diesel Engine Powered Cogeneration (DEPC) plant

C: Compressor, T: Turbine, WHB: Waste Heat Boiler, DeSO$_x$: Desulphurization, AWR: Air-Water Radiator, IC: Intercooler, WH: Water Heater, LOC: Lubrication Oil Cooler, LOT: Lubrication Oil Tank, SD: Steam Drum, FWT: Feed Water Tank, FDT: Fuel oil Day Tank, CON: Condenser, FFM: Fuel Forwarding Module, P: Pump

Table 8.1 Plant data, thermodynamic properties, and exergies in the plant with respect to state points in Figure 8.1. Values are for one engine set only.

State No	Fluid	Pressure P (bar)	Temperature T (°C)	Mass flowrate, $\dot{m}$ (kg/s)	Enthalpy, h (kJ/kg)	Entropy, s (kJ/kg°C)	Specific exergy, ψ (kJ/kg)	Exergy rate, $\dot{E}$ (kW)
0	Air	1.00	30.0	-	303.5	5.712	0.0	-
0	Water	1.00	30.0	-	125.1	0.434	0.0	-
0	Fuel oil	1.00	30.0	-	-	-	0.0	-
0	Lub oil	1.00	30.0	-	-	-	0.0	-
1	Air	1.00	30.0	18.4	303.5	5.712	0.0	0.0
2	Air	2.90	172	18.4	446.9	5.794	118.40	2179
3	Air	2.80	53.7	18.4	313.7	5.492	76.80	1413.1
4’	Fuel oil	1.90	81.0	0.46	87.2	0.254	5.87	2.70
4	Fuel oil	4.30	81.0	0.46	103.5	0.316	7.75	3.57
5	Fuel oil	5.10	137.5	0.46	231.5	0.654	33.34	15.34
6	Exhaust	2.40	451	17.0	739.2	6.357	240.20	4083.4
7	Exhaust	0.80	302	17.0	580.9	6.428	60.42	1027.1
8	Exhaust	0.75	247	17.0	523.8	6.342	29.33	499
9	Exhaust	1.190	53.7	17.0	327.3	5.738	16.02	272.3
10	Water	2.40	88.0	0.75	326.1	1.050	14.55	11.0
11	Water	8.00	88.0	0.75	326.5	1.049	15.11	11.3
12	Water	8.00	95.0	0.75	397.9	1.248	26.31	20.0
13	Water	8.00	170	0.75	718.6	2.041	106.80	80.1
14’	Water	4.90	170	0.75	698.0	1.980	88.80	66.6
14’’	Water	8.00	170	0.75	718.6	2.041	106.80	80.1
14	Water	8.00	170	0.55	718.6	2.041	106.80	58.7
15	Water	8.00	170	0.05	718.6	2.041	106.80	5.4
16	Water	7.80	70.0	0.05	293.0	0.953	10.84	0.6
17	Water	8.00	170	0.15	718.6	2.041	106.80	16.0
18	Water	4.70	60.0	0.15	250.9	0.829	6.20	1.0
19	Water	4.60	50.0	0.15	209.1	0.702	3.01	0.5
20	HT Water	3.10	71.7	30.0	299.7	0.974	11.21	336.3
21	HT Water	3.00	81.0	30.0	338.7	1.085	16.37	491.1
22	HT Water	2.80	88.0	30.0	368.1	1.167	21.02	630.7
22’	HT Water	2.70	71.7	30.0	299.7	0.974	11.17	335.1
23	LT Water	3.10	38.4	47.2	160.5	0.549	0.69	32.5
24	LT Water	3.00	44.9	47.2	187.6	0.635	1.68	79.5
25	LT Water	2.90	50.3	47.2	210.2	0.706	2.20	104.0
25’	LT Water	2.70	38.4	47.2	160.4	0.550	0.65	30.7
26	Lub oil	4.20	63.0	20.0	65.7	0.206	3.33	66.6
27	Lub oil	3.00	78.3	20.0	97.7	0.299	7.05	141.0
28	Lub oil	2.90	59.7	20.0	58.9	0.186	2.71	54.2
29	Air	1.00	30.0	0.30	303.5	5.712	0	0.0
30	Air	0.99	51.0	0.30	324.6	5.781	0.26	0.1
31	Air	1.00	30.0	267.0	303.5	5.712	0	0.0
32	Air	0.90	47.0	267.0	320.0	5.765	0.41	110.7

8.3 Chemical Exergy Analysis of Heavy Fuel Oil

In diesel engine powered cogeneration systems, most commonly used fuel is heavy fuel oil. Heavy fuel oil is known as complex mixtures of hydrocarbon compounds, ranging from light, volatile, short-chained organic compounds to heavy, long-chained, branched compounds. Several studies on the chemical and physical characteristics of heavy fuel oils and on handling problems have been reported in the literature [178-181]. Since the composition of these fuel oils is very complex, their characterization is generally limited to certain specifications such as API gravity (which is a measure of how heavy or light a petroleum liquid is compared to water), viscosity, distillation temperature, and flash point [182].

The composition of the heavy fuel oil depends on the origin of the crude as well as the refining scheme. These specifications are not indicative of the oil composition. Since heavy fuel oils contain a very large number of different hydrocarbon compounds, their composition is generally specified in terms of three main classes:

- paraffinic
- aromatic
- naphtenic

The aromatics can include some very heavy compounds in the molecular weight range of 1000 to 20,000 which are known as asphaltenes. Gel permation chromatography (GPC) can be used to characterize heavy fuel oils into asphaltenic, paraffinic and light aromatic groups. According to GPC, the asphaltenes are compounds with linear molecular sizes greater than n-$C_{44}H_{80}$. The paraffins and light aromatics are those compounds with linear molecular sizes between n-$C_{44}H_{80}$ and $C_{12}H_{26}$ and smaller than n-$C_{12}H_{26}$, respectively [182-184].

Fuel oil no.6 is also called Bunker C or residual. It is the residual from crude oil after the light oils, gasoline, naphta, fuel oil no.1, and fuel oil no.2 have been fractioned off. Fuel oil no.6 can be blended directly to heavy fuel oil or made into asphalt. It is limited to commercial and industrial uses where sufficient heat is available to fluidize the oil for pumping and combustion [185].

In Table 8.2, chemical composition of heavy fuel oil no.6 by volume is given according to the varying sulfur content [181].

Table 8.2 Composition of fuel oil no.6 with varying sulfur content [181]

Content	Low S–Fuel oil no.6	Medium S–Fuel oil no.6	High S–Fuel oil no.6
C, %	85.99	86.46	85.45
H, %	11.29	10.98	10.35
N, %	0.43	0.43	0.35
S, %	0.53	0.93	2.33
O, %	1.24	0.67	0.92
Moisture, %	0.50	0.50	0.50
Ash, %	0.02	0.03	0.10

For power production and cogeneration applications in Turkey, heavy fuel oil no.6 with high sulfur content is used and for this, waste exhaust gas treatment units such as desulphurization and denitrification should be installed in order to reduce emissions at levels below legal limits.

8.3.1 The combustion of asphaltenes - $C_{44}H_{80}$

The composition of asphaltenes, $C_{44}H_{80}$, is given in the third column of Table 8.2 as ing high sulfur content. We assume that heavy fuel oil no.6 with the given chemical composition is burned with stoichiometric amount of moist air in the diesel engine. We note that the moisture in the air does not react with anything; it simply shows up as additional H_2O in the products. Therefore, for simplicity, we will balance the combustion equation by using dry air and then add the moisture later to both sides of the equation. Considering one kmol of fuel,

$$(0.959C_{44}H_{80} + 0.0035N_2 + 0.0092O_2 + 0.0233S + 0.005H_2O) + \boldsymbol{a_{th}}(O_2 + 3.76N_2) \rightarrow$$

$$\boldsymbol{x}CO_2 + \boldsymbol{y}H_2O + \boldsymbol{z}SO_2 + \boldsymbol{w}NO + 3.76\, \boldsymbol{a_{th}}N_2$$

The unkownn coefficients in the above equation are determined from mass balances on various elements, and the amount of moisture that accompanies with 4.76 a_{th} can be used to calculate the partial pressure of the moisture in the air for the relative humidity of 10% and saturation pressure of water at 30ºC in July 2005 in Gaziantep. Assuming ideal gas behavior, the number of moles of the moisture in the air is

$$N_{v,\mathrm{air}} = \left(\frac{P_{v,\mathrm{air}}}{P_{\mathrm{total}}}\right) N_{\mathrm{total}} \tag{8.1}$$

The balanced combustion equation is obtained by substituting the coefficients determined and adding the number of moles of the moisture in the air to both sides of the equation:

$$(0.959C_{44}H_{80} + 0.0035N_2 + 0.0092O_2 + 0.0233S + 0.005H_2O) + \mathbf{63.752}(O_2 + 3.76N_2)$$

$$+ \mathbf{1.32}H_2O \rightarrow \mathbf{43.82}CO_2 + \mathbf{41.161}H_2O + \mathbf{0.0233}SO_2 + \mathbf{0.007}NO + \mathbf{239.71}N_2$$

This reaction is an approach of the stoichiometric combustion reaction of heavy fuel oil no.6 with high sulfur content. The dew-point temperature of the products is the temperature at which the water vapor in the products starts to condense as the products are cooled. Again, assuming ideal gas behavior, the partial pressure of the water vapor in the combustion (exhaust) gases is

$$P_{v,\mathrm{prod}} = \left(\frac{N_{v,\mathrm{prod}}}{N_{\mathrm{prod}}}\right) P_{\mathrm{prod}} \tag{8.2}$$

Thus, the dew-point temperature is found to be as 49.8ºC. If the combustion process were achieved with dry air instead of moist air, the products would contain less moisture, and the dew-point temperature in this case would be 49.2ºC.

8.3.2 The combustion of aromatics - $C_{12}H_{26}$

The combustion reaction of $C_{12}H_{26}$ with air as heavy fuel oil no.6 with high sulfur content is considered in a similar manner to the combustion of asphaltenes. Thus, the combustion reaction with determined coefficients and moist air can be written as

$$(0.959C_{12}H_{26} + 0.0035N_2 + 0.0092O_2 + 0.0233S + 0.005H_2O) + \mathbf{17.760}(O_2 + 3.76N_2)$$

$$+ \mathbf{0.368}H_2O \rightarrow \mathbf{11.508}CO_2 + \mathbf{12.84}H_2O + \mathbf{0.0233}SO_2 + \mathbf{0.007}NO + \mathbf{66.78}N_2$$

The dew-point temperature is calculated as 52.0ºC. This value would be 51.5ºC in the case of dry air.

8.3.3 Approximation Method for Chemical Exergy of Heavy Fuel oil no.6

Szargut and Styrylska [186] proposed a statistical method for representing the chemical exergy of fuels. If the elemental composition of fuel is known, then the correlation formula developed for complex technical fuels can be used for the calculation of the chemical exergy of fuel. The nitrogen content in these fuels is much lower than the values in organic compounds used for determination of the regression equations. Thus, it can be neglected in the chemical exergy calculations. The chemical exergy of heavy fuel no.6 can be obtained from the equation developed as [186]

$$\frac{\psi_{\mathrm{fuel}}}{q_{\mathrm{LHV}}} = 1.0401 + 01728\frac{z_{\mathrm{H_2}}}{z_{\mathrm{C}}} + 0.0432\frac{z_{\mathrm{O_2}}}{z_{\mathrm{C}}} + 0.2169\frac{z_{\mathrm{S}}}{z_{\mathrm{C}}}\left(1 - 2.0628\frac{z_{\mathrm{H_2}}}{z_{\mathrm{C}}}\right) \quad (8.3)$$

where z values are the volume fractions of elements in chemical composition of fuel. The ratio in the Equation (8.3) is calculated as **1.0659** by using the defined chemical composition of heavy fuel oil no.6 with high sulfur content in Table 8.2. This value is nearly equal to the approach of Brzustowski and Brena [187].

8.3.4 Specific Heat Calculations of Combustion (Exhaust) Gases

The flow exergy of an ideal gas is given by [160]

$$\psi = (h - h_0) - T_0(s - s_0) \quad (8.4)$$

or $$\psi = C_{p,av}(T - T_0) - T_0\left[C_{p,av}\ln\frac{T}{T_0} - R\ln\frac{P}{P_0}\right] \quad (8.5)$$

In some cases, exhaust gases are considered as air and exergy values are taken equal to the air at specified conditions. This is a source of error. Here, starting from the combustion reaction of heavy fuel oil no.6 (i.e., asphaltenes and aromatics are two distinct types of it and considered as two different reactions), average specific heat values of exhaust gases are calculated considering the composition of exhaust gases [160].

(i) Specific heat calculation of the exhaust gas from the combustion of $C_{44}H_{80}$

The exhaust gas mixture of the combustion reaction of asphaltenes contains:

N_{CO2}	: 43.820 kmol
N_{H20}	: 41.161 kmol
N_{SO2}	: 0.0233 kmol
N_{NO}	: 0.007 kmol
N_{N2}	: 239.71 kmol
	+ ----------------
$N_{mixture}$	: 324.721 kmol

Molecular mass of exhaust gas contents:

m_{CO2}	: 44.0 kg/ kmol
m_{H20}	: 18.0 kg/ kmol
m_{SO2}	: 64.0 kg/ kmol
m_{NO}	: 30.0 kg/kmol
m_{N2}	: 28.0 kg/ kmol
	+------------------
$m_{mixture}$	: 184.0 kg/kmol

In the combustion reaction above, the masses of component gases are calculated as

m_{CO2}	: 44.0 kg/ kmol × 43.82 kmol = 19281.1 kg
m_{H20}	: 18.0 kg/ kmol × 41.161 kmol = 741.0 kg
m_{SO2}	: 64.0 kg/ kmol × 0.0233 kmol = 1.491 kg
m_{NO}	: 30.0 kg/kmol × 0.007 kmol = 0.21 kg
m_{N2}	: 28.0 kg/ kmol × 239.71 = 6711.9 kg
	+------------------
$m_{mixture}$	= 9382.7 kg

Then, mass fractions of each element in the exhaust can be evaluated as

$$m_{fCO2} = \frac{1928.1}{9382.7} = 0.21 \rightarrow C_{p@303K} = 0.846\, kJ/kgK$$
$$m_{fH2O} = \frac{741.0}{9382.7} = 0.079 \rightarrow C_{p@303K} = 1.8723\, kJ/kgK$$
$$m_{fSO2} = \frac{1.491}{9382.7} = 0.00016 \rightarrow C_{p@303K} = 0.624\, kJ/kgK$$
$$m_{fNO} = \frac{0.21}{9382.7} = 0.0000224 \rightarrow C_{p@303K} = 1.0\, kJ/kgK$$
$$m_{fN2} = \frac{6711.9}{9382.7} = 0.72 \rightarrow C_{p@303K} = 1.039\, kJ/kgK$$

Specific heat values of elements are obtained from the tabulated values [160]. Then, the specific heat value of the exhaust gas from the combustion reaction of asphaltenes with air is obtained as 1.073 times specific heat of the air by using the relation [160],

$$\sum_{i=1} m_{fi}.C_{pi} \tag{8.6}$$

(ii) Specific heat calculation of the exhaust gas from the combustion of $C_{12}H_{26}$

Using the same procedure, the specific heat value of the exhaust gas from the combustion reaction of aromatics with air is obtained as 1.081 times specific heat of the air.

8.4 Diesel Engine Operating and Performance Characteristics

For the engine M.A.N. 18V 32/40, the cylinder bore is B=320 mm, piston stroke is S=400 mm, and the engine speed is N=750 rpm. The other characteristics of the engine and various engine performance parameters are defined below.
The crank offset a is given as

$$a = S/2 \tag{8.7}$$

Average piston speed is

$$U_{\mathrm{p}} = 2SN \tag{8.8}$$

Average piston speed for all engines will normally be in the range of 5 to 20 m/sec, with large diesel engines on the low end and high-performance automobile engines on the high end [175,176,188]. Displacement, or displacement volume V_d is the volume displaced by the piston as it travels from bottom dead center (BDC) to top dead center (TDC)

$$V_{\mathrm{d}} = V_{\mathrm{BDC}} - V_{\mathrm{TDC}} \tag{8.9}$$

Displacement can be given for one cylinder or the entire engine. For one cylinder

$$V_{\mathrm{d}} = \frac{\pi}{4} B^2 S \tag{8.10}$$

For an engine with N_c number of cylinders:

$$V_{\mathrm{d}} = \frac{\pi}{4} B^2 S N_{\mathrm{c}} \tag{8.11}$$

Typical values for engine displacement range from 0.1 cm^3 for small model airplanes to about 8 L for large automobiles to much larger numbers for large ship engines. The displacement of a modern average automobile engine is about two to three liters [176].

Minimum cylinder volume occurs when the piston is at TDC and called the clearance volume V_C

$$V_{\mathrm{C}} = V_{\mathrm{TDC}} = V_{\mathrm{d}} - V_{\mathrm{BDC}} \tag{8.12}$$

The compression ratio is defined as

$$R_{\mathrm{C}} = \frac{V_{\mathrm{BDC}}}{V_{\mathrm{TDC}}} = \frac{(V_{\mathrm{C}} + V_{\mathrm{d}})}{V_{\mathrm{C}}} \tag{8.13}$$

Modern spark ignition (SI) engines have compression ratios R_C of 8 to 11, while compression ignition (CI) engines have compression ratios in the range 12 to 24.

Engines with superchargers or turbochargers usually have lower compression ratios than naturally aspirated engines [176].

Work is the output of any heat engine, and in a reciprocating I.C engine, this work is generated by the gases in the combustion chamber of the cylinder. Force due to gas pressure on the moving piston generates the work in an internal combustion engine cycle

$$W = \int PdV \tag{8.14}$$

It is convenient to analyze engine cycles per unit mass of gas m within the cylinder. To do so, volume V is replaced with specific volume υ and work is replaced with specific work:

$$w = \frac{W}{m}, \qquad v = \frac{V}{m}, \qquad w = \int Pdv \tag{8.15}$$

Specific work is equal to the area under the process lines on the P-υ coordinates of indicator diagram. The areas shown in the indicator diagram gives the work inside the combustion chamber and called as indicated work W_i, but the work delivered by crankshaft is less than indicated one due to mechanical friction and parasitic loads of the engine W_f. Actual work available at the crankshaft is called brake work

$$W_b = W_i - W_f \tag{8.16}$$

The ratio of brake work to indicated work defines the mechanical efficiency of an engine. Mechanical efficiencies will be on the order of 75 % to 95 %,

$$\eta_m = \frac{W_b}{W_i} \tag{8.17}$$

An average or mean effective pressure (mep) is defined by:

$$mep = \frac{w}{\Delta v} = \frac{W}{V_d} \tag{8.18}$$

Mean effective pressure is a good parameter to compare engines for design or output because it is independent of engine size and /or speed. Various mep can be defined by using different work terms. If brake work is used brake mean effective pressure is obtained:

$$bmep = \frac{W_\mathrm{b}}{V_\mathrm{d}} \tag{8.19}$$

Indicated work gives indicated mean effective pressure:

$$imep = \frac{W_\mathrm{i}}{V_\mathrm{d}} \tag{8.20}$$

Typical maximum values of bmep for naturally aspirated SI engines are in the range of 850 to 1050 kPa. For CI engines, they are 700 to 900 kPa for naturally aspirated engines and 1000 to 1200 kPa for turbocharger engines [175,176].

Torque is a good indicator of an engine's ability to do work. It is defined as force acting at a moment distance and has units of N-m. Torque is related to work by:

$$2\pi\tau = W_\mathrm{b} = (bmep)\frac{V_\mathrm{d}}{n} \tag{8.21}$$

$$\tau = (bmep)\frac{V_\mathrm{d}}{4\pi} \quad \text{for four stroke cycle} \tag{8.22}$$

Power $\dot{W}$ is defined as the rate of work of the engine. If n is the number of revolutions per cycle and N is the engine speed, then

$$\dot{W} = W\frac{N}{n} \tag{8.23}$$

Other characteristic parameters for an engine are

$$\text{Specific power} \quad SP = \frac{\dot{W}_\mathrm{b}}{A_\mathrm{p}} \tag{8.24}$$

$$\text{Output per displacement} \quad OPD = \frac{\dot{W}_\mathrm{b}}{V_\mathrm{d}} \tag{8.25}$$

Specific volume $SV = \dfrac{V_{\mathrm{d}}}{\dot{W}_{\mathrm{b}}}$ (8.26)

Specific weight $SW = \dfrac{(Weight_{\mathrm{engine}})}{\dot{W}_{\mathrm{b}}}$ (8.27)

Energy input to an engine Q_{in} comes from the combustion of a hydrocarbon fuel. Air is used to supply the oxygen needed for this chemical reaction. For combustion reaction to occur, the proper relative amounts of air (oxygen) and fuel must be present. Air-fuel ratio (AF) and fuel–air ratio (FA) are parameters used to describe the mixture ratio:

$$AF = \frac{m_{\mathrm{a}}}{m_{\mathrm{f}}} = \frac{\dot{m}_{\mathrm{a}}}{\dot{m}_{\mathrm{f}}} \tag{8.28}$$

$$FA = \frac{m_{\mathrm{f}}}{m_{\mathrm{a}}} = \frac{\dot{m}_{\mathrm{f}}}{\dot{m}_{\mathrm{a}}} \tag{8.29}$$

where m_a = mass of air, $\dot{m}_{\mathrm{a}}$ = mass flow rate air, m_f = mass of fuel, $\dot{m}_{\mathrm{f}}$ = mass fuel rate of fuel. The ideal stoichiometric AF for many gasoline type hydrocarbon fuels is very close to 15, with combustion possible for value in the range 6 to 19 [175,176,189].

Equivalence ratio is defined the actual ratio of fuel–air to ideal or stoichiometric fuel–air:

$$\Phi = \frac{(FA)_{\mathrm{act}}}{(FA)_{\mathrm{stoich}}} = \frac{(AF)_{\mathrm{stoich}}}{(AF)_{\mathrm{act}}} \tag{8.30}$$

Fuel consumption of an engine is calculated from the consumption per power generated; this is also called specific fuel consumption (sfc) and can be derived for all types of works. The specific fuel consumption equations for brake power and indicated power are given as

$$bsfc = \frac{\dot{m}_{\mathrm{f}}}{\dot{W}_{\mathrm{b}}} \tag{8.31}$$

$$isfc = \frac{\dot{m}_{\mathrm{f}}}{\dot{W}_{\mathrm{i}}} \tag{8.32}$$

In order to measure or comment on an engine performance, different types of efficiencies related with engine parameters must be known. These efficiencies are: Combustion efficiency η_c is defined as the fraction of fuel which burns. It has values in the range 0.95 to 0.98 and it is given as

$$\eta_{\mathrm{c}} = \frac{\dot{Q}_{\mathrm{in}}}{Q_{\mathrm{HV}}\dot{m}_{\mathrm{f}}} \tag{8.33}$$

where Q_{in} is the rate of net heat input to the engine and Q_{HV} is the lower heating value of the fuel. Thermal efficiency is defined as

$$\eta_{\mathrm{th}} = \frac{W}{Q_{\mathrm{in}}} = \frac{\dot{W}}{\dot{Q}_{\mathrm{in}}} = \frac{\dot{W}}{\dot{m}_{\mathrm{f}}\dot{Q}_{\mathrm{HV}}\eta_{\mathrm{c}}} \tag{8.34}$$

Thermal efficiency can be given as indicated or brake depending on whether indicated power or brake power is used. Engines can have indicated thermal efficiencies in the range of 50% to 60% with brake thermal efficiency usually about 30%. Some large slow CI engines can have brake thermal efficiencies greater than 50% [176,188]. Fuel conversion efficiency is defined as

$$\eta_{\mathrm{f}} = \frac{\dot{W}}{\dot{m}_{\mathrm{f}}Q_{\mathrm{HV}}} \tag{8.35}$$

Volumetric efficiency is defined as

$$\eta_{\mathrm{v}} = \frac{n\dot{m}_{\mathrm{a}}}{\rho_{\mathrm{a}}V_{\mathrm{d}}N} \tag{8.36}$$

where ρ_a is the density of atmospheric air. Volumetric efficiency is a measure of how much air is ingested into the engine and it could be greater than one for turbocharged engines.

Using the equations given in this section and plant data in Tables 8.1, various engine operating and performance characteristics are calculated. The results are listed in Table 8.3. Certain engine parameters, which are not specified by the manufacturer, are calculated. These include specific power (6217 kW/m^2), specific volume (64.34 m^3/kW), specific weight (8.67 ton/MW), output per displacement (14.58 kW/L), volumetric efficiency (1.29), and thermal efficiency (0.47). All of these are very important parameters for the thermodynamic performance of internal combustion engines and they can be used for the performance comparison of similar engines. The values are typical of stationary engines. Perhaps, the most important result is high thermal efficiency, which is 47 percent. Note that the average thermal efficiencies are 30 to 40 percent for diesel automobile engines and 25 to 35 percent for gasoline automobile engines. This can be explained by the fact that the stationary engines operate at their optimum values with a constant engine speed. The entire operation is optimized to minimize the fuel consumption for a given power output. The inherent limitations such as space, weight, complexity, and maintenance in the design of automobile engines are not crucial for stationary large engines.

Typical thermal efficiencies are 30 to 40 percent for steam and gas turbine power plants and close to 50 percent for combined cycle power plants. Typical values are under 15 percent and for geothermal power plants and under 10 percent for solar power plants. It is clear that the diesel engine power plant operates at high thermal efficiency and consequently the cost of electricity should be low compared to other power systems.

8.4.1 Ideal Diesel Cycle Analysis of the Engine

The engine is compression ignition engine and it may be represented by an ideal diesel cycle, which is an air-standard cycle. Various assumptions used in an air-standard cycle include the followings: The gas mixture in the cylinder is treated as air for the entire cycle, and ideal gas properties of air are used in the analysis. The real open cycle is changed into a closed cycle. The combustion process is replaced with a heat addition process. This process takes place at constant pressure in ideal diesel cycle. Intake and exhaust strokes are assumed to be constant pressure. Compression and expansion are approximated by isentropic processes. Exhaust blowdown is replaced by a constant-volume heat rejection process. In ideal diesel

Table 8.3 Calculated engine operating and performance characteristics

Cylinder Diameter, D	320	mm
Cylinder Bore, B	320	mm
Stroke, S	400	mm
Crank Offset, a	200	mm
Number of Cylinders, N_c	18	-
Piston Area, A_P	0.08	m^2
Compression Ratio, r_c	12.37	-
Displacement Volume, V_d	0.03217	m^3
Clearance Volume, V_c	0.00283	m^3
Cylinder Volume, V	0.035	m^3
Air Flow Rate, $\dot{m}_a$	18.4	kg/s
Fuel Flow Rate, $\dot{m}_f$	0.46	kg/s
Air-Fuel Ratio, AF	40.0	-
Lubrication Oil Flow Rate	20.0	kg/s
Exhaust Flow Rate	17.0	kg/s
Piston Mean Speed, U_p	10.0	m/s
Engine Speed, N	750	rpm
Heating Value, Q_{HV}	42,700	kJ/kg
Break Power, $\dot{W}_b$	8440	kW
Break Power per Cylinder	468.9	kW
Brake Mean Effective Pressure, bmep	2490	kPa
Torque, τ	114.6	Nm
Brake Specific Fuel Consumption, bsfc	184.0	g/kWh
Specific Power, SP	6217	kW/m^2
Specific Volume, SV	0.06434	m^3/MW
Specific Weight, SW	8.67	ton/MW
Output per Displacement, OPD	14.58	kW/L
Combustion Efficiency, η_c	0.98	-
Volumetric Efficiency, η_v	1.29	-
Thermal Efficiency, η_{th}	0.47	-

cycle, the following relationships can be used based on the processes shown in Figure 8.2 [176].

Process 6-1 Constant pressure intake stroke

$$w_{6\text{-}1} = P_0(v_1 - v_6) \tag{8.37}$$

Process 2-3 Isentropic compression

$$T_2 = T_1 \left(r_\mathrm{c}\right)^{k-1} \tag{8.38}$$

$$P_2 = P_1 \left(r_\mathrm{c}\right)^{k} \tag{8.39}$$

$$\dot{m}_{mixture} = \dot{m}_\mathrm{exhaust} = \dot{m}_\mathrm{fuel} + \dot{m}_\mathrm{air} \tag{8.40}$$

$$m_\mathrm{mixture} = \frac{P_1 \times V_1}{R \times T_1} \tag{8.41}$$

$$V_1 = V_\mathrm{d} + V_\mathrm{c} \tag{8.42}$$

$$V_2 = \frac{V_1}{r_\mathrm{c}} = \frac{0.117}{14.4} = V_\mathrm{TDC} \tag{8.43}$$

$$w_{1-2} = \frac{m_\mathrm{mixture} \times R \times \left(T_2 - T_1\right)}{\left(1-k\right)} \tag{8.44}$$

Process 3-4 Constant pressure heat input

$$Q_{2-3} = Q_\mathrm{in} = \dot{m}_\mathrm{f} \times Q_\mathrm{HV} \times \eta_C \tag{8.45}$$

$$Q_{2-3} = Q_\mathrm{in} = \dot{m}_\mathrm{m} \times C_\mathrm{P} \times \left(T_3 - T_2\right) \tag{8.46}$$

$$V_3 = \frac{T_3 \times V_2}{T_2} \tag{8.47}$$

$$T_4 = T_3 \times \left(\frac{V_3}{V_4}\right)^{k-1} \tag{8.48}$$

$$P_4 = P_3 \times \left(\frac{V_3}{V_4}\right)^{k} \tag{8.49}$$

$$w_{3-4} = \frac{m_\mathrm{mixture} \times R \times \left(T_4 - T_3\right)}{\left(1-k\right)} \tag{8.50}$$

$$w_{\text{net}} = w_{3-4} - w_{1-2} \tag{8.51}$$

$$\dot{W}_{\text{net}} = w_{\text{net}} \times N / n \tag{8.52}$$

$$Q_{4-5} = Q_{\text{out}} = \dot{m}_{\text{m}} \times C_{\text{v}} \times (T_1 - T_4) \tag{8.53}$$

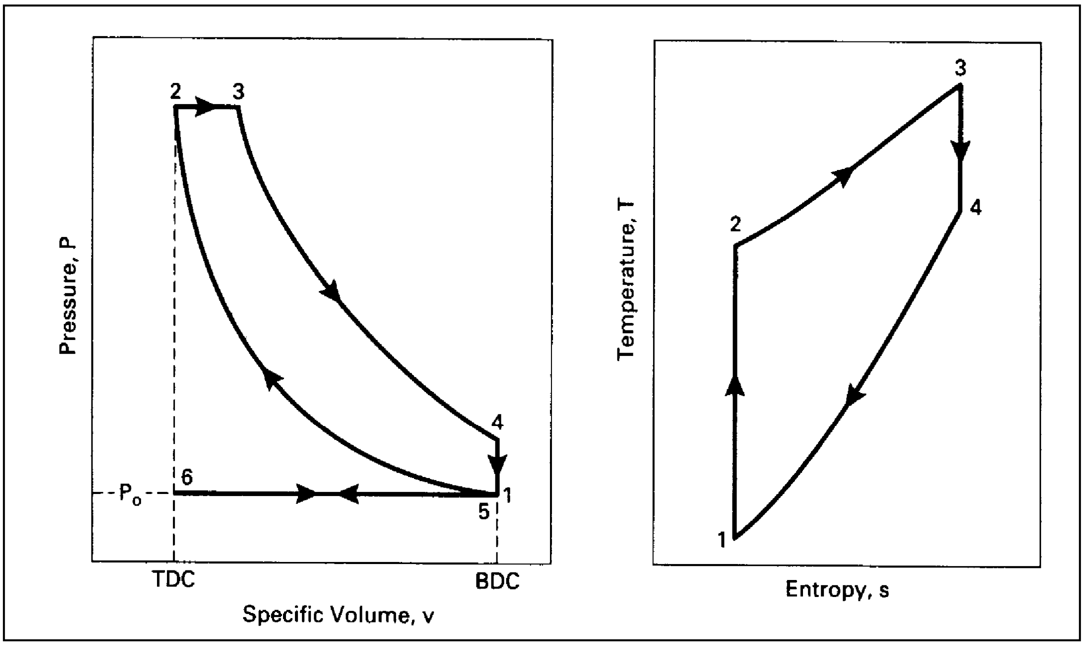

Figure 8.2 P-v and T-s diagrams of ideal diesel cycle [176]

<u>Process 5-6 Constant pressure exhaust stroke</u>

$$w_{5-6} = P_0 \times (V_6 - V_5) = P_0 \times (V_6 - V_1) \tag{8.54}$$

Thermal efficiency:

$$(\eta_{\text{t}})_{\text{DIESEL}} = \frac{|w_{\text{net}}|}{|q_{\text{in}}|} = 1 - \left(\frac{|q_{\text{out}}|}{|q_{\text{in}}|} \right) \tag{8.55}$$

$$(\eta_{\text{t}})_{\text{DIESEL}} = 1 - \left[\frac{(T_4 - T_1)}{k(T_3 - T_2)} \right] \tag{8.56}$$

$$(\eta_{\text{t}})_{\text{DIESEL}} = 1 - \left(\frac{1}{r_{\text{c}}} \right)^{k-1} \left[\frac{(\beta^k - 1)}{k(\beta - 1)} \right] \tag{8.57}$$

$$\beta = \frac{V_3}{V_2} = \frac{v_3}{v_2} = \frac{T_3}{T_2} \tag{8.58}$$

The input data for the diesel engines of the cogeneration plant and average air properties as

P_1=280 kPa, T_1=326.7 K, V_1=0.035 m^3, N=750 rpm, r_c=12.37, n=2, k=1.35, C_v=0.821 kJ/kgK, C_p=1.108 kJ/kgK, $\dot{m}_{air}$=18.4 kg/sec, $\dot{m}_{fuel}$=0.46 kg/sec

Using Equations (8.37) through (8.58), the results of ideal diesel engine analysis are obtained as

P_2=8353 kPa, T_2=788 K, $\dot{m}_{mixture} = \dot{m}_{exhaust}$=0.1046 kg, V_2=0.002832 m^3, T_3=1719 K, V_3=0.00618 m^3, T_4=937 K, P_4=803 kPa, w_{net}=291.5 kJ/kg, $\dot{Q}_{in}$=19,249 kW, $\dot{W}_{net}$=11,435 kW, η_{diesel} = 0.594

The important results are that the net power is 11,435 kW and the thermal efficiency is 59.4%. These are indicated power and indicated thermal efficiency values. The actual brake power output from the engine was obtained to be about 8440 kW and the actual brake thermal efficiency to be 47%. The difference between the actual brake values and ideal cycle indicated values are due to the mechanical inefficiencies as the power is transferred from inside the cylinder to the crankshaft and the assumptions used in the analysis of ideal diesel cycle.

8.5 Energy and Exergy Relations for Plant Components

Energy and exergy relations for the components of the plant are provided. The relations are based on general formulations provided in Chapter 4 and include mass, energy, and exergy balances as well as exergy destructions and exergy efficiencies. State numbers refer to Figure 8.1.

Compressor

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{\text{air}} \tag{8.59}$$

$$\dot{W}_{\text{comp,act,in}} = \dot{m}_1\left(h_2 - h_1\right) \tag{8.60}$$

$$\eta_{\text{comp}} = \frac{w_s}{w_a} = \frac{h_{2,s} - h_1}{h_2 - h_1} \tag{8.61}$$

$$\dot{W}_{\text{comp,rev,in}} = \dot{m}_1(\psi_2 - \psi_1) \tag{8.62}$$

$$\psi_2 - \psi_1 = \left(h_2 - h_1\right) - T_0 \cdot \left(s_2 - s_1\right) \tag{8.63}$$

$$s_2 - s_1 = s_2^0 - s_1^0 - R\, ln \frac{P_2}{P_1} \tag{8.64}$$

$$\dot{E}_{\text{comp,dest}} = \dot{W}_{\text{comp,act,in}} - \dot{W}_{\text{comp,rev,in}} \tag{8.65}$$

$$\varepsilon_{\text{comp}} = \frac{\dot{W}_{\text{comp,rev,in}}}{\dot{W}_{\text{comp,act,in}}} \tag{8.66}$$

Intercooler

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_{\text{air}} \tag{8.67}$$

$$\dot{m}_{23} = \dot{m}_{24} = \dot{m}_{\text{LT-Water}} \tag{8.68}$$

$$\dot{m}_{20} = \dot{m}_{21} = \dot{m}_{\text{HT-Water}} \tag{8.69}$$

$$\dot{m}_2\left(h_2 - h_3\right) = \dot{m}_{21}\left(h_{21} - h_{20}\right) + \dot{m}_{24}\left(h_{24} - h_{23}\right) \tag{8.70}$$

$$\dot{E}_{\text{IC,dest}} = \dot{m}_2\left(\psi_2 - \psi_3\right) - \dot{m}_{21}\left(\psi_{21} - \psi_{20}\right) - \dot{m}_{24}\left(\psi_{24} - \psi_{23}\right) \tag{8.71}$$

$$\psi_{21} - \psi_{20} = \left(h_{21} - h_{20}\right) - T_0\left(s_{21} - s_{20}\right) \tag{8.72}$$

$$\psi_{24} - \psi_{23} = \left(h_{24} - h_{23}\right) - T_0\left(s_{24} - s_{23}\right) \tag{8.73}$$

$$\psi_2 - \psi_3 = \left(h_2 - h_3\right) - T_0\left(s_2 - s_3\right) \tag{8.74}$$

$$s_2 - s_3 = s_2^0 - s_3^0 - R\, ln \frac{P_2}{P_3} \tag{8.75}$$

$$\varepsilon_{\text{IC}} = \frac{\left[\dot{m}_{21}\left(\psi_{21} - \psi_{20}\right) + \dot{m}_{24}\left(\psi_{24} - \psi_{23}\right)\right]}{\dot{m}_2\left(\psi_2 - \psi_3\right)} \tag{8.76}$$

Lubrication oil cooler

$$\dot{m}_{24} = \dot{m}_{25} = \dot{m}_{\text{LT-Water}} \tag{8.77}$$

$$\dot{m}_{27} = \dot{m}_{28} = \dot{m}_{\text{Lub-Oil}} \tag{8.78}$$

$$\dot{m}_{25}(h_{25} - h_{24}) = \dot{m}_{27}(h_{27} - h_{28}) \tag{8.79}$$

$$\dot{E}_{\text{LOC,dest}} = \dot{m}_{25}(\psi_{25} - \psi_{24}) - \dot{m}_{27}(\psi_{27} - \psi_{28}) \tag{8.80}$$

$$\psi_{27} - \psi_{28} = (h_{27} - h_{28}) - T_0(s_{27} - s_{28}) \tag{8.81}$$

$$\psi_{25} - \psi_{24} = C_{\text{av}}(T_{25} - T_{24}) - T_0 C_{\text{av}} ln\frac{T_{25}}{T_{24}} \tag{8.82}$$

$$\varepsilon_{\text{LOC}} = \frac{\dot{m}_{27}(\psi_{27} - \psi_{28})}{\dot{m}_{25}(\psi_{25} - \psi_{24})} \tag{8.83}$$

Diesel engine

$$\eta_{\text{engine}} = \frac{\dot{W}_{\text{net}}}{\dot{m}_f q_{\text{LHV}}} \quad \text{(Net power/Fuel energy)} \tag{8.84}$$

$$\dot{E}_{\text{engine,dest-1}} = (\dot{E}_f + \dot{E}_3 + \dot{E}_5 + \dot{E}_{21} + \dot{E}_{26}) - (\dot{E}_6 + \dot{E}_{22} + \dot{E}_{27} + \dot{W}_{\text{net}}) \tag{8.85}$$

$$\varepsilon_{\text{engine-1}} = \frac{\dot{E}_6 + \dot{E}_{22} + \dot{E}_{27} + \dot{W}_{\text{net}}}{\dot{E}_f + \dot{E}_3 + \dot{E}_5 + \dot{E}_{21} + \dot{E}_{26}} \quad \text{(Total exergy out/Total exergy in)} \tag{8.86}$$

$$\dot{E}_{\text{engine,dest-2}} = \dot{E}_f - \dot{W}_{\text{net}} \tag{8.87}$$

$$\varepsilon_{\text{engine-2}} = \frac{\dot{W}_{\text{net}}}{\dot{E}_f} = \frac{\dot{W}_{\text{net}}}{\dot{m}_f \psi_{\text{fuel}}} \quad \text{(Net power/Fuel exergy)} \tag{8.88}$$

Turbine

$$\dot{m}_6 = \dot{m}_7 = \dot{m}_{\text{exhaust}} \tag{8.89}$$

$$\dot{W}_{\text{turb,act,out}} = \dot{m}_6(h_6 - h_7) \tag{8.90}$$

$$\eta_{\text{turb}} = \frac{h_6 - h_7}{h_6 - h_{7s}} \tag{8.91}$$

$$\dot{W}_{\text{turb,rev,out}} = \dot{m}_6 \cdot (\psi_6 - \psi_7) \tag{8.92}$$

$$\psi_6 - \psi_7 = (h_6 - h_7) - T_0 \cdot (s_6 - s_7) \tag{8.93}$$

$$\dot{E}_{\text{turb,dest}} = \dot{W}_{\text{turb,rev,out}} - \dot{W}_{\text{turb,act,out}} \tag{8.94}$$

$$\varepsilon_{\text{turb}} = \frac{\dot{W}_{\text{turb,act,out}}}{\dot{W}_{\text{turb,rev,out}}} \tag{8.95}$$

Waste heat boiler

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_{\text{exhaust}} \tag{8.96}$$

$$\dot{m}_{12} = \dot{m}_{13} = \dot{m}_{\text{water}} \tag{8.97}$$

$$\dot{m}_7(h_7 - h_8) = \dot{m}_{13}(h_{13} - h_{12}) \tag{8.98}$$

$$\dot{E}_{\text{WHB,dest}} = \dot{m}_7(\psi_7 - \psi_8) - \dot{m}_{13}(\psi_{13} - \psi_{12}) \tag{8.99}$$

$$\psi_7 - \psi_8 = (h_7 - h_8) - T_0 \cdot (s_7 - s_8) \tag{8.100}$$

$$\psi_{13} - \psi_{12} = (h_{13} - h_{12}) - T_0 \cdot (s_{13} - s_{12}) \tag{8.101}$$

$$\varepsilon_{\text{WHB}} = \frac{\dot{m}_{13}(\psi_{13} - \psi_{12})}{\dot{m}_7(\psi_7 - \psi_8)} \tag{8.102}$$

Fuel oil day tank

$$\dot{m}_4 = \dot{m}_{\text{Fuel-Oil}} \tag{8.103}$$

$$\dot{m}_{15} = \dot{m}_{16} = \dot{m}_{\text{water}} \tag{8.104}$$

$$\dot{m}_4(h_4 - h_{0-\text{FO}}) = \dot{m}_{15}(h_{15} - h_{16}) \tag{8.105}$$

$$\dot{E}_{\text{FDT,dest}} = \dot{m}_{15}(\psi_{15} - \psi_{16}) - \dot{m}_4(\psi_4 - \psi_{0-\text{FO}}) \tag{8.106}$$

$$\psi_{15} - \psi_{16} = (h_{15} - h_{16}) - T_0(s_{15} - s_{16}) \tag{8.107}$$

$$\psi_4 - \psi_{0-\text{FO}} = C_{\text{av}}(T_4 - T_0) - T_0 C_{\text{av}} ln \frac{T_4}{T_0} \tag{8.108}$$

$$\varepsilon_{\text{FDT}} = \frac{\dot{m}_4(\psi_4 - \psi_{0-\text{FO}})}{\dot{m}_{15}(\psi_{15} - \psi_{16})} \tag{8.109}$$

Fuel forwarding module

$$\dot{m}_{17} = \dot{m}_{18} = \dot{m}_{\text{water}} \tag{8.110}$$

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_{\text{fuel-oil}} \tag{8.111}$$

$$\dot{m}_{17}(h_{17} - h_{18}) = \dot{m}_5(h_5 - h_4) \tag{8.112}$$

$$\dot{E}_{\text{FFM,dest}} = \dot{m}_{17}(\psi_{17} - \psi_{18}) - \dot{m}_5(\psi_5 - \psi_4) \tag{8.113}$$

$$\psi_5 - \psi_4 = (h_5 - h_4) - T_0(s_5 - s_4) \tag{8.114}$$

$$\psi_{17} - \psi_{18} = (h_{17} - h_{18}) - T_0(s_{17} - s_{18}) \tag{8.115}$$

$$\varepsilon_{\text{FFM}} = \frac{\dot{m}_5(\psi_5 - \psi_4)}{\dot{m}_{17}(\psi_{17} - \psi_{18})} \tag{8.116}$$

Condenser

$$\dot{m}_{29} = \dot{m}_{30} = \dot{m}_{\text{surr-air}} \tag{8.117}$$

$$\dot{m}_{18} = \dot{m}_{19} = \dot{m}_{\text{water}} \tag{8.118}$$

$$\dot{m}_{18}(h_{18} - h_{19}) = \dot{m}_{30}(h_{30} - h_{29}) \tag{8.119}$$

$$\dot{E}_{\text{cond,dest}} = \dot{m}_{18}(\psi_{18} - \psi_{19}) - \dot{m}_{30}(\psi_{30} - \psi_{29}) \tag{8.120}$$

$$\psi_{18} - \psi_{19} = (h_{18} - h_{19}) - T_0(s_{18} - s_{19}) \tag{8.121}$$

$$\psi_{30} - \psi_{29} = (h_{30} - h_{29}) - T_0(s_{30} - s_{29}) \tag{8.122}$$

$$\varepsilon_{\text{cond}} = \frac{\dot{m}_{30}(\psi_{30} - \psi_{29})}{\dot{m}_{18}(\psi_{18} - \psi_{19})} \tag{8.123}$$

*Pump*1

$$\dot{m}_{25'} = \dot{m}_{23} = \dot{m}_{\text{LT-Water}} \tag{8.124}$$

$$\dot{W}_{\text{pump1,act,in}} = \dot{m}_{25'}(h_{23} - h_{25'}) \tag{8.125}$$

$$\eta_{\text{pump,1}} = \frac{h_{23,\text{s}} - h_{25'}}{h_{23} - h_{25'}} \tag{8.126}$$

$$\dot{W}_{\text{pump1,rev,in}} = \dot{m}_{25'}(\psi_{23} - \psi_{25'}) \tag{8.127}$$

$$\psi_{23} - \psi_{25'} = \left(h_{23} - h_{25'}\right) - T_0\left(s_{23} - s_{25'}\right) \tag{8.128}$$

$$\dot{E}_{\text{pump1,dest}} = \dot{W}_{\text{pump1,act,in}} - \dot{W}_{\text{pump1,rev,in}} \tag{8.129}$$

$$\varepsilon_{\text{pump,1}} = \frac{\dot{W}_{\text{pump1,rev,in}}}{\dot{W}_{\text{pump1,act,in}}} \tag{8.130}$$

<u>*Pump*2</u>

$$\dot{m}_{22'} = \dot{m}_{20} = \dot{m}_{\text{HT-Water}} \tag{8.131}$$

$$\dot{W}_{\text{pump2,act,in}} = \dot{m}_{22'}\left(h_{20} - h_{22'}\right) \tag{8.132}$$

$$\eta_{\text{pump,2}} = \frac{h_{20,\text{s}} - h_{22'}}{h_{20} - h_{22'}} \tag{8.133}$$

$$\dot{W}_{\text{pump2,rev,in}} = \dot{m}_{22'}\left(\psi_{20} - \psi_{22'}\right) \tag{8.134}$$

$$\psi_{20} - \psi_{22'} = \left(h_{20} - h_{22'}\right) - T_0\left(s_{20} - s_{22'}\right) \tag{8.135}$$

$$\dot{E}_{\text{pump2,dest}} = \dot{W}_{\text{pump2,act,in}} - \dot{W}_{\text{pump2,rev,in}} \tag{8.136}$$

$$\varepsilon_{\text{pump,2}} = \frac{\dot{W}_{\text{pump2,rev,in}}}{\dot{W}_{\text{pump2,act,in}}} \tag{8.137}$$

<u>*Pump*3</u>

$$\dot{m}_{10} = \dot{m}_{11} = \dot{m}_{\text{water}} \tag{8.138}$$

$$\dot{W}_{\text{pump3,act,in}} = \dot{m}_{10}\left(h_{11} - h_{10}\right) \tag{8.139}$$

$$\eta_{\text{pump,3}} = \frac{h_{11,\text{s}} - h_{10}}{h_{11} - h_{10}} \tag{8.140}$$

$$\dot{W}_{\text{pump3,rev,in}} = \dot{m}_{11}\left(\psi_{11} - \psi_{10}\right) \tag{8.141}$$

$$\psi_{11} - \psi_{10} = \left(h_{11} - h_{10}\right) - T_0\left(s_{11} - s_{10}\right) \tag{8.142}$$

$$\dot{E}_{\text{pump3,dest}} = \dot{W}_{\text{pump3,act,in}} - \dot{W}_{\text{pump3,rev,in}} \tag{8.143}$$

$$\varepsilon_{\text{pump,3}} = \frac{\dot{W}_{\text{pump3,rev,in}}}{\dot{W}_{\text{pump3,act,in}}} \tag{8.144}$$

*Pump*4

$$\dot{m}_{14'} = \dot{m}_{14} = \dot{m}_{\text{steam}} \tag{8.145}$$

$$\dot{W}_{\text{pump4,act,in}} = \dot{m}_{14'}\left(h_{14} - h_{14'}\right) \tag{8.146}$$

$$\eta_{\text{pump},4} = \frac{h_{14,\text{s}} - h_{14'}}{h_{14} - h_{14'}} \tag{8.147}$$

$$\dot{W}_{\text{pump4,rev,in}} = \dot{m}_{14'}\left(\psi_{14} - \psi_{14'}\right) \tag{8.148}$$

$$\psi_{14} - \psi_{14'} = \left(h_{14} - h_{14'}\right) - T_0\left(s_{14} - s_{14'}\right) \tag{8.149}$$

$$\dot{E}_{\text{pump4,dest}} = \dot{W}_{\text{pump4,act,in}} - \dot{W}_{\text{pump4,rev,in}} \tag{8.150}$$

$$\varepsilon_{\text{pump},4} = \frac{\dot{W}_{\text{pump4,rev,in}}}{\dot{W}_{\text{pump4,act,in}}} \tag{8.151}$$

*Pump*5

$$\dot{m}_{4'} = \dot{m}_{4} = \dot{m}_{\text{fuel}} \tag{8.152}$$

$$\dot{W}_{\text{pump5,act,in}} = \dot{m}_{4'}\left(h_{4} - h_{4'}\right) \tag{8.153}$$

$$\eta_{\text{pump},5} = \frac{h_{4,\text{s}} - h_{4'}}{h_{4} - h_{4'}} \tag{8.154}$$

$$\dot{W}_{\text{pump5,rev,in}} = \dot{m}_{4'}\left(\psi_{4} - \psi_{4'}\right) \tag{8.155}$$

$$\psi_{4} - \psi_{4'} = \left(h_{4} - h_{4'}\right) - T_0\left(s_{4} - s_{4'}\right) \tag{8.156}$$

$$\dot{E}_{\text{pump5,dest}} = \dot{W}_{\text{pump5,act,in}} - \dot{W}_{\text{pump5,rev,in}} \tag{8.157}$$

$$\varepsilon_{\text{pump},5} = \frac{\dot{W}_{\text{pump5,rev,in}}}{\dot{W}_{\text{pump5,act,in}}} \tag{8.158}$$

Air-Water Radiator

$$\dot{m}_{31} = \dot{m}_{32} = \dot{m}_{\text{air,AWR}} \tag{8.159}$$

$$\dot{m}_{25} = \dot{m}_{25'} = \dot{m}_{\text{LT-Water}} \tag{8.160}$$

$$\dot{m}_{22} = \dot{m}_{22'} = \dot{m}_{\text{HT-Water}} \tag{8.161}$$

$$\dot{m}_{31}(h_{32} - h_{31}) = \dot{m}_{25}(h_{25'} - h_{25}) + \dot{m}_{22}(h_{22'} - h_{22}) \tag{8.162}$$

$$\dot{E}_{\mathrm{AWR,dest}} = \dot{m}_{31}(\psi_{32} - \psi_{31}) - \dot{m}_{25}(\psi_{25'} - \psi_{25}) - \dot{m}_{22}(\psi_{22'} - \psi_{22}) \tag{8.163}$$

$$\psi_{32} - \psi_{31} = (h_{32} - h_{31}) - T_0(s_{32} - s_{31}) \tag{8.164}$$

$$\psi_{25'} - \psi_{25} = (h_{25'} - h_{25}) - T_0(s_{25'} - s_{25}) \tag{8.165}$$

$$\psi_{22'} - \psi_{22} = (h_{22'} - h_{22}) - T_0(s_{22'} - s_{22}) \tag{8.166}$$

$$s_{32} - s_{31} = s_{32}^0 - s_{31}^0 - R\,ln\frac{P_{31}}{P_{32}} \tag{8.167}$$

$$\varepsilon_{\mathrm{AWR}} = \frac{[\dot{m}_{22}(\psi_{22'} - \psi_{22}) + \dot{m}_{25}(\psi_{25'} - \psi_{25})]}{\dot{m}_{31}(\psi_{32} - \psi_{31})} \tag{8.168}$$

8.6 Results and Discussion for Energy and Exergy Analysis

The DEPC plant is divided into fifteen components/subsystems as shown in Figure 8.1. The temperature, pressure, and mass flow rate data and certain exergy evaluations of the plant according to the nomenclature shown in Figure 8.1 are presented in Table 8.1. The energy and exergy calculations are done using commercial software with built-in thermodynamic property functions for a variety of substances [191] (see Appendix 1). In defining the exergy flow through the subsystems, fuel and product terms must be identified. The product represents the desired result produced by the component (i.e. subsystem) whereas the fuel represents the resources expended to generate the product. Both the product and the fuel are expressed in terms of exergy and definitions of the exergies of the fuels $\dot{E}_{\mathrm{F}}$ and the exergies of products $\dot{E}_{\mathrm{P}}$ for the components of the plant.

The chemical exergy of the fuel is determined using Equation 8.3. For one engine set with two turbochargers, exergy input is calculated as 20,919 kW. Then the total exergy input of fuel by considering three diesel engine systems to this cogeneration plant becomes 62,757 kW. In Table 8.4, energy and exergy analyses results of the plant are given for one engine set.

The exergy assessment of the plant is given schematically in Figure 8.3. The rates of exergy destructions of the components of the plant as compared with total fuel exergy input are given in Figure 8.4. We note the followings from these results:

- 40.4% of the exergy entering the plant is converted to electrical power and 5% of this power is used for parasitic load in the plant to drive auxiliary components in the plant. The net steam production of the plant represents only 0.3% of the total exergy input. The remaining 59.3% of the exergy input is destroyed. This corresponds to 37,246 kW, which is the total exergy destruction in the plant.

- The exergetic efficiency of the plant is determined to be 40.7%. The exergy destruction in the diesel engines of the cogeneration plant accounts about 46.0% of the total exergy input and 79.5% of the total exergy destruction in the plant. The exergy destruction in the engine is mostly due to the highly irreversible combustion process, heat losses from the engine and friction.

- The exergetic efficiencies of the compressor and turbine of the turbocharger are 82.6% and 88.1%, respectively. These values indicate sufficient exergetic performance for the turbocharger.

- The exergetic efficiencies of the waste heat boiler and condenser are calculated as 11.4% and 16.6%, respectively making them the least efficient components of the plant. The intercooler and air-water radiator have the exergetic efficiencies of 26.3% and 70%. Exergy destructions in these heat exchange units in the plant are mainly due to the high average temperature (for intercooler) and mass flow rate (for air-water radiator) differences between the two unmixed fluid streams.

- The percent of exergy loss associated with lubrication oil cooler is low. This is due to the cooling of lubrication oil by using low temperature water [160].

Table 8.4 Energetic and exergetic analyses results for the subsystems in the plant. The values are for one engine set only.

C: Compressor, T: Turbine, WHB: Waste Heat Boiler, $DeSO_x$: Desulphurization, IC: Intercooler, LOC: Lubrication Oil Cooler, CON: Condenser, FFM: Fuel Forwarding Module, P: Pump.

	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{E}_F$ (kW)	$\dot{E}_P$ (kW)	$\dot{E}_D$ (kW)	y^* (%)	y (%)	ε (%)
C	0.00	2640	2640	2180	460.0	3.80	2.200	82.6
IC	2452	0.00	766.0	201.8	564.20	4.67	2.700	26.3
LOC	790.2	0.00	86.8	24.5	62.30	0.52	0.300	63.0
AWR	4402.6	0.00	369.0	110.7	258.3	2.14	1.230	70.0
DE	0.00	8440	22,905	13,295	9610	79.49	45.94	40.4
T	0.00	2692	3056	2692	364.0	3.01	1.740	88.1
WHB	970.8	0.00	528.1	60.1	468.0	3.87	2.240	11.4
FDT	21.3	0.00	4.8	3.6	1.20	0.009	0.006	79.1
FFM	70.2	0.00	15.0	11.8	3.20	0.009	0.020	87.4
CON	6.3	0.00	0.5	0.1	0.40	0.00	0.001	16.6
P1	0.00	0.95	0.95	0.94	0.01	0.00	0.000	98.9
P2	0.00	0.61	0.61	0.59	0.02	0.00	0.000	96.7
P3	0.00	0.43	0.43	0.30	0.13	0.00	0.000	97.0
P4	0.00	33.20	33.20	13.50	19.70	0.23	0.000	40.67
P5	0.00	7.50	7.50	0.87	6.63	0.00	0.000	11.6
$DeSO_x$	272.3	0.00	272.3	0.00	272.30	2.25	0.010	-
Overall System	80.1	8440	20,919	8520	12,090	100.0	57.79	40.7

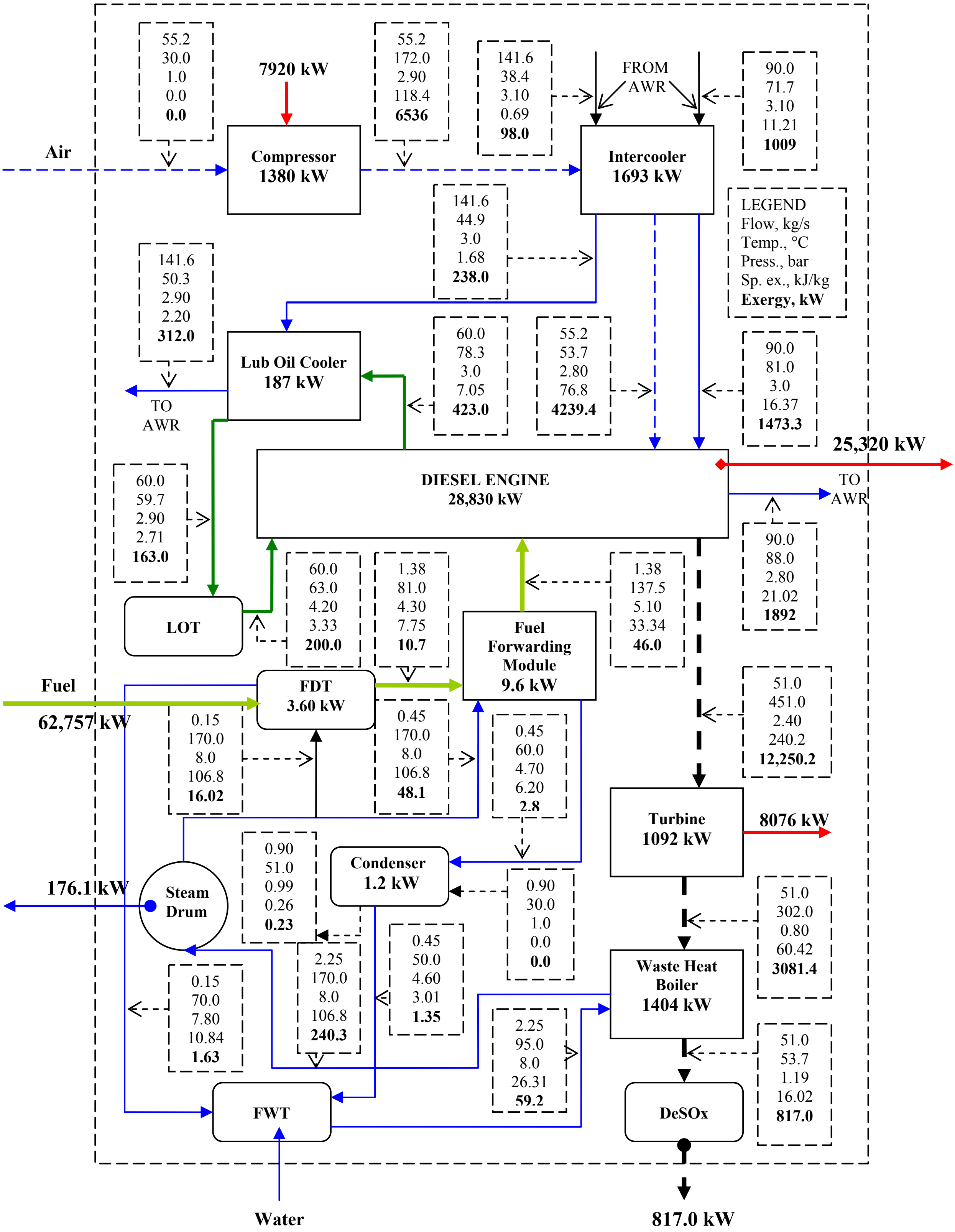

Figure 8.3 Exergy flow diagram of Sanko DEPC.

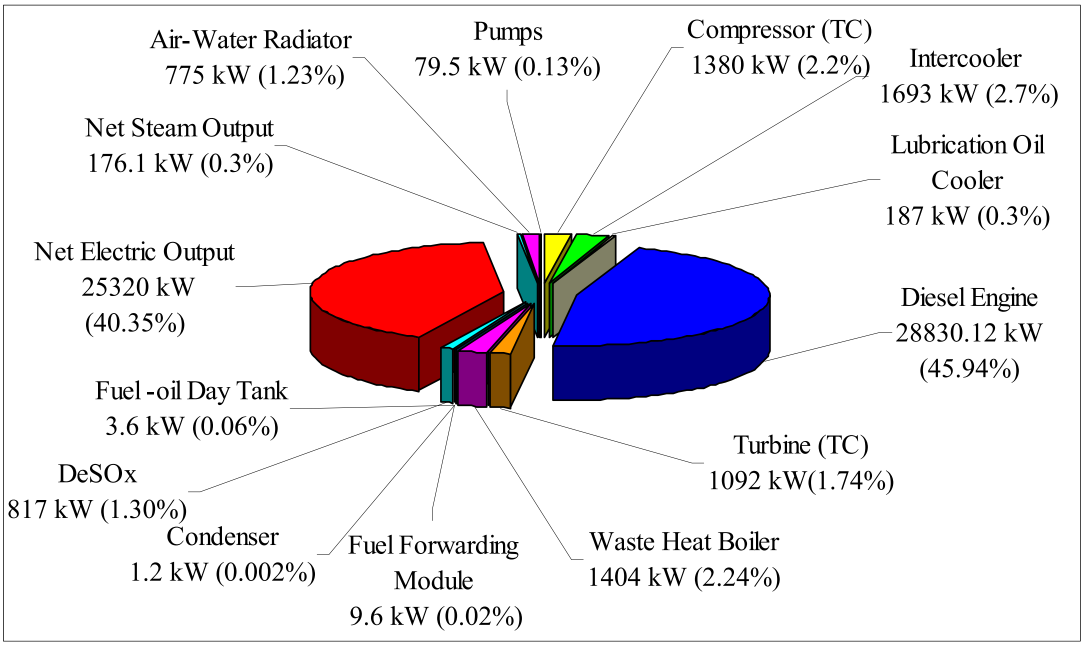

Figure 8.4 The rate of exergy destructions of the components of the plant as compared with the fuel exergy input. Electrical and steam outputs are also shown.

8.7 Performance Assessment Parameters

The relations for various performance assessment parameters of cogeneration are provided in Chapter 4. The fuel utilization efficiency (FUE) of the overall plant is determined to be 44.6%. This value is high compared to thermal efficiencies of power plants whose sole purpose is the production of electricity. In diesel engine cogeneration plants, the main product is electricity and the steam may be called as "byproduct". The thermal efficiency of the diesel engine defined as the power output over the fuel energy input is calculated to be 43.0%.

Power to heat ratio (PHR) of the plant is calculated to be 143.8. When this value is compared with the corresponding PHR values of conventional gas turbine (i.e. 1.2), and gas engine (i.e. 1.1) [192] cogeneration plants, it gives us a vague idea about the cogeneration system: whether the analyzed system is close to a thermal power plant (a large value of PHR as in the case of DEPC system presented) or to an ordinary boiler (a small value of PHR). The PHR, when employed alone, is not a decisive parameter for performance assessment of cogeneration plants. However, it is valuable in understanding the behavior of other performance parameters of any thermal power plant in which the primary goal is to produce electricity rather than process heat [193]. The exergetic efficiency of the plant is determined to be 40.6%. When calculating the exergetic efficiency of the overall plant, the input exergy is taken to be the chemical exergy of the fuel. The exergy of the fresh treated water at the inlet of the waste heat boiler is negligible. The exergetic efficiency of the diesel engine itself is 40.4%.

The variation of FUE and PHR as a function of the process steam output is given in Figure 8.5. As the process steam output increases, FUE increases and PHR decreases. Higher values of FUE make the plant thermodynamically more efficient while the higher values of PHR makes the plant economically more feasible.

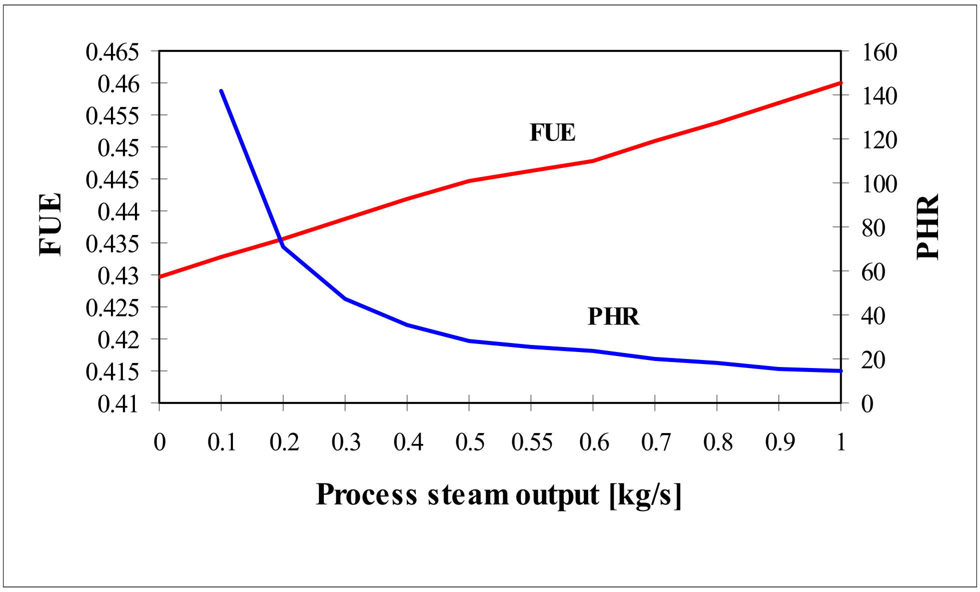

Figure 8.5 Variation of fuel utilization efficiency (FUE) and power to heat ratio (PHR) as a function of process steam output for Sanko DEPC system

8.8 Economic Analysis

In order to calculate the cost rates of the plant, the economic data are obtained from the actual vendor quotations of the company. The DEPC plant is supplied as packaged system and cost allocation among its components (i.e. subsystems) is not separately quoted. However, to obtain more accurate results from thermoeconomic analysis, the subsystems are considered as separate and cost allocation of subsystems and the other expenditures are obtained from the energy manager of the company and the contractor of the DEPC system.

All cost items including fuel are considered to increase with the general inflation rate which is taken as 5% per year. Design and construction of DEPC started in January, 2000 and lasted for 2 years. The economic life of DEPC is considered as 25 years that is from January 1, 2002 to December 31, 2027. The system life for tax purposes is 20 years. It is also considered that when the system is retired, the cost of removal will be equal the salvage value, resulting in zero salvage.

The average capacity factor for the cogeneration system is considered as 85% which means that the system will operate at full load 7446 hours out of the total available 8760 hours per year. The company has 30 labor positions which are required for operation and maintenance (O&M) at an average labor rate of \$5.00 per hour. The average number of working hours per labor position is 2482 h per year. Thus the annual direct labor costs are 372.3×10^3 US dollars. Based on these numbers, the annual fixed O&M costs and the annual variable O&M costs at full capacity are calculated by management to be 500×10^3 and 100×10^3 USD, respectively. The annual variable O&M costs at the given capacity factor of 85% are 85×10^3 USD.

To calculate the corresponding costs for the first year of operation, these costs are escalated at a nominal escalation rate of 5% per year to the middle of 2002. The escalated annual values for the fixed and variable O&M costs are 732×10^3 and 124.5×10^3 USD, respectively. The cost of heavy fuel oil is \$3.00 per GJ of lower heating value. Lower heating value of heavy fuel oil is 42.7 MJ/kg. The fuel mass flow rate of total system is 1.38 kg/s. The annual fuel cost is then

$$\begin{aligned} FC &= (\$0.003\,/\text{MJ}) \times (42.7\text{ MJ/kg}) \times (1.38\text{ kg/s}) \times (7446\text{ h/yr}) \times (3600\text{ s/h}) \\ &= 4.74 \times 10^6 \text{ USD/yr} \end{aligned}$$

To calculate the annual fuel costs for the first year of commercial operation (mid-2002 dollars), we escalate this number at an annual escalation rate of 5% to the middle of 2002:

Table 8.5 The total capital investment for the Sanko DEPC (all costs are expressed in thousands of mid-2000 dollars and for including three engine sets)

I. Fixed Capital Investment	
A. Direct Costs	
1. Onsite Costs	
Purchased – equipment costs	
* Compressor (Turbocharger)	850
* Intercooler	300
* Air-Water Radiator	300
* Lubrication Oil Cooler	100
* Diesel Engine	5,400
* Turbine (Turbocharger)	850
* Waste Heat Boiler	300
* Fuel Oil Day Tank	100
* Fuel Forwarding Module	1,000
* Condenser	50
* Pumps	50
* Other plant equipment	700
Total Purchased Equipment Cost (PEC)	**10,000**
Purchased equipment installation (30% of PEC)	3,000
Piping (30% of PEC)	3,000
Instrumentation and controls (10% of PEC)	1,000
Electrical equipment and materials (10% of PEC)	1,000
Total Onsite Costs	**18,000**
2. Offsite Costs	
Land	300
Civil, structural, and architectural work (15% of PEC)	1,500
Service facilities (25% of PEC)	2,500
Total Offsite Costs	**4,300**
TOTAL DIRECT COSTS	**22,300**
B. Indirect Costs	
Engineering and supervision (5% of DC)	1,115
Construction costs and contractor's profit (10% of DC)	2,230
Sum	25,645
Contingency (10% of the above sum)	2,564
Total Indirect Costs	**5,909**
FIXED CAPITAL INVESTMENT	**28,209**
II. Other Outlays	
Startup costs	741
Working capital	1,562
Costs of licensing, research and development	0
Allowance for funds used during construction (AFUDC)	4,515
Total Other Outlays	**6,818**
TOTAL CAPITAL INVESTMENT	**35,027**

$$\mathrm{FC} = (4.74 \times 10^{6}) \times (1.05)^{4} = 5.76 \times 10^{6} \text{ USD}$$

The starts up costs for a cogeneration plant are the sum of the following unescalated costs (SUC): (a) one month of fixed O&M costs, (b) one month of variable O&M costs at full load, (c) one week of full load fuel, (d) 2% of the plant facilities investment. Then,

$$\begin{aligned} SUC &= \frac{(0.5 \times 10^{6})}{12} + \frac{(0.1 \times 10^{6}) \times 0.85}{12} + \frac{(5.76 \times 10^{6})}{52} + (0.02) \times (27.909 \times 10^{6}) \\ &= 0.718 \times 10^{6} \text{ USD} \end{aligned}$$

After escalation with the annual escalation rate of 5%,

$$SUC = (0.718 \times 10^{6}) \times (1.05)^{3} = 0.831 \times 10^{6} \text{ USD}$$

Similarly, the working capital is the sum of the unescalated expenses representing two months of fuel and variable operating costs at full load, and three months of labor costs plus a contingency of 25% of the total of the above three items. Neglecting the variable operating costs for the cogeneration system, we obtain

$$\mathrm{WC} = \left[(5.76 \times 10^{6})/6 + (0.3723 \times 10^{6})/4\right] \times (1.25) = 1.316 \times 10^{6} \text{ USD}$$

or after escalation to the end of 2001 (or at the beginning of 2002)

$$\mathrm{WC} = (1.316 \times 10^{6}) \times (1.05)^{3} = 1.523 \times 10^{6} \text{ USD}$$

The plant facilities investment (PFI) for this DEPC is given by vendor as 27.909×10^{9} USD, which is the difference between fixed-capital investment and land costs. According to the allocation of plant facilities investment plan of the firm, 50% of this amount must be escalated at an annual rate of 5% to the middle of 2000, whereas remaining 50% of PFI must be escalated with the same rate to the middle of 2001.

The average cost of money in the investment of the DEPC depends on the fractions of the total capital requirement financed through debt which comes from borrowing capital, for instance, by selling bonds, preferred stock and common stock which come from equity financing and on the required return on each type of financing. In this study, the total capital requirement of the DEPC is covered using debt (20%), preferred stock (30%) and common stock (50%). The minimum acceptable returns on investment are 10%, 15%, and 20%, respectively, and the average annual rate for the cost of money is calculated as

$$i = (0.10)\times(0.20) + (0.15)\times(0.30) + (0.20)\times(0.50) \cong 0.17 \quad (17\%)$$

Allowance for funds during construction (AFUDC) is calculated separately for each type of financing using the corresponding returns on investment and given from Table 8.6a to 8.6d. AFUDC for the land and start up costs are given in Table 8.6e.

Table 8.6a Evaluation of the Plant-Facilities Investment (PFI) used during construction (end-2001 values) of the DEPC (All values are given in thousands of dollars)

Design and Construction Year	Calendar Year	In Mid – 1998 Dollars (PFI)	Amount of Escalation (PFI)	Escalated Investment (PFI)
1	2000	13,954.5	2,930.4	16,885
2	2001	13,954.5	4,619.0	18,574
Totals		27,909	7,549.4	35,459
		(A)	(B)	(C)

Table 8.6b Evaluation of the Common Equity and Allowance for Funds used during construction (AFUDC) (end 2001 values) of the DEPC (All values are given in thousands of dollars)

Design and Construction Year	Calendar Year	Escalated Investment (PFI)	Escalated Investment (Common Equity)	AFUDC
1	2000	16,885	8,442.5	1,688.5
2	2001	18,574	9,287.0	1,857.5
Totals		35,459	17,730	3,546.0
		(C)	(D)	(E)

Table 8.6c Evaluation of the Preferred Equity and AFUDC (end 2001 values) of the DEPC (All values are given in thousands of dollars)

Design and Construction Year	Calendar Year	Escalated Investment (PFI)	Escalated Investment (Preferred Equity)	AFUDC
1	2000	16,885	5,065.5	760.0
2	2001	18,574	5,572.2	836.0
Totals		35,459	10,638	1,596.0
		(C)	(F)	(G)

Table 8.6d Evaluation of the Debt and AFUDC (end 2001 values) of the DEPC (All values are given in thousands of dollars)

Design and Construction Year	Calendar Year	Escalated Investment (PFI)	Escalated Investment (Debt)	AFUDC
1	2000	16,885	3,377.0	338.0
2	2001	18,574	3,714.8	371.5
Totals		35,459	7,092	710.0
		(C)	(H)	(I)

Table 8.6e AFUDC for the land and start up costs (end 2001 values) (All values are given in thousands of dollars)

	Calendar Year	Escalation Values	AFUDC (Common Equity)	AFUDC (Preferred Equity)	AFUDC (Debt)
Land Cost	Jan. 1, 2000	346.1	69.0	52.0	35.0
Start up Cost	July 1, 2001	1,505.4	301.0	226.0	151.0
			3,546.0	1,596.0	710.0
Total			3,916.0	1,874.0	896.0

The total amount of AFUDC is 6.686×10^6 end-2001 USD. The AFUDC value given in Table 8.4 is obtained by discounting the end -2001 AFUDC to the middle of 1998 using the average discounting rate of 17%.

The levelized annual fuel costs in current dollars $(FC_{L,cu})$ can be calculated using the related formulas which are given in Chapter 5 as

$$k_F = \frac{1+0.05}{1+0.17} = 0.8974$$

$$CRF_{cu} = \frac{0.17 \times (1.17)^{25}}{(1.17)^{25} - 1} = 0.1734$$

$$FC_{L,cu} = \frac{5.76 \times 10^6}{1.05} \times \frac{0.8974 \times (1-(0.8974)^{25}) \times (0.1734)}{(1-0.8974)} = 13.440 \times 10^6 \text{ USD}$$

Similarly, the levelized annual operating and maintenance costs $(OMC_{L,cu})$ are

$$k_{OM} = \frac{1+0.05}{1+0.17} = 0.8974$$

$$\mathrm{OMC_{L,cu}} = \frac{0.857\times10^{6}}{1.05}\times\frac{0.8974\times(1-(0.8974)^{25})\times(0.1734)}{(1-0.8974)} = 1.298\times10^{6}\,\mathrm{USD}.$$

The levelized annual carrying charges $(\mathrm{CC_{L,cu}})$ are then calculated as

$$\mathrm{CC_{L,cu}} = 5.732\times10^{6}\,\mathrm{USD}$$

Since the DEPC plant presented in this study produces steam as a "byproduct" and it is a small fraction of the electrical output, we consider only electrical output as the product. Thus, its unit cost (MPUC) can be calculated directly from the annual total revenue requirement (TRR), the annual total value of the byproducts (BPV) produced in the same plant, and the main product quantity (MPQ) [51]

$$\mathrm{MPUC} = \frac{\mathrm{TRR}-\mathrm{BPV}}{\mathrm{MPQ}} \tag{8.116}$$

However, it is sold for 25 year levelized value, which is 0.1 USD/kg. The levelized total annual value of steam can be obtained as

$$\mathrm{RWC_{L}} = 0.03043\times10^{6}\,\mathrm{USD}$$

The electrical energy developed by this actual DEPC per year is

$$\mathrm{MPQ} = (25{,}320\ \mathrm{kW})\times(7446\ \mathrm{h/yr}) = 188.53\times10^{6}\ \mathrm{kWh/yr}$$

The levelized unit cost of electricity for the 25 year period can be calculated as

$$\mathrm{MPUC_{L}} = \frac{20.5\times10^{6}-0.03043\times10^{6}}{188.53\times10^{6}} = 0.1085\ \$/\mathrm{kWh} = 10.85\ \mathrm{cents/kWh}$$

This value represents the levelized cost for a 25 year period assuming average annual nominal escalation rates for the fuel costs and the O&M expenses of 5% for the entire economic life of the cogeneration system. We should also keep in mind that the levelized costs are not directly comparable to actual costs at any given year of plant operation. In a conventional economic analysis of a thermal system that generates more than one product we need to know the selling prices of all but one product in order to calculate the cost associated with this product. In other words, a conventional economic analysis does not provide criteria for apportioning the carrying charges, fuel costs, and O&M expenses to the various products generated in the same system. In Table 8.7, the levelized cost values of the carrying charges and expenditures of the DEPC system are given. The annual total revenue requirement (TRR) is equal to the sum of the costs of carrying charges (CC), fuel, raw water and O&M. Levelization including entire economic life of the plant is performed by using plant's economic data and related formulations given in Chapter 5 of this study and in literature [47,51,68]. Levelized cost rate of the fuel is calculated to be 1806 \$/h and that of the raw water to be 4.1 \$/h (see Table 8.8). The purchased equipment costs, the hourly levelized costs of capital investment, the operating and maintenance costs, and the total costs of the components of the plant are given in Table 8.8. The levelized values in this table are obtained using the values in Table 8.7.

Table 8.7 The annual levelized cost values of carrying charges and expenditures

Levelized Costs	Carrying Charges Cost $\mathbf{CC_L}$	Fuel Cost $\mathbf{FC_L}$	Raw Water Cost $\mathbf{RWC_L}$	Operating and Maintenance $\mathbf{OMC_L}$	Total Revenue Requirement $\mathbf{TRR_L}$
$\times 10^3$ US\$	5732	13,440	30.43	1298	20,500

Table 8.8 The cost rates associated with first capital investment and O&M costs for the subcomponents of the DEPC plant

Component	**PEC** ($\times 10^3$ \$)	$\mathbf{Z_k^{CI}}$ (\$/h)	$\mathbf{Z_k^{OM}}$ (\$/h)	$\mathbf{Z_k^{T}}$ (\$/h)
Compressor (TC)	850	65.5	14.8	80.3
Intercooler	300	23.1	5.2	28.3
Lubrication Oil Cooler	100	7.7	1.7	9.4
Air-Water Radiator	300	23.1	5.2	28.3
Diesel Engine	5400	416.0	94.2	510.2
Turbine (TC)	850	65.5	14.8	80.3
Waste Heat Boiler	300	23.1	5.2	28.3
Fuel Oil Day Tank	100	7.7	1.7	9.4
Fuel Forwarding Module	1,000	77.0	17.4	94.4
Condenser	50	3.9	0.9	4.8
Pumps	50	3.9	0.9	4.8
$DeSO_x$	500	38.5	8.7	47.2
Other Plant Equipments	200	15.4	3.5	18.9
Total Purchased Equipment Cost (PEC)	**10,000**	**770.5**	**174.5**	**945**

8.9 Thermoeconomic Analysis

Thermoeconomics assess the cost of consumed resources, money and system irreversibilities in terms of the overall production process. It helps to point out how resources are used more effectively in order to save them. Monetary costs express the economic effect of inefficiencies and are used to improve the cost effectiveness of production processes. Assessing the cost of the flow streams and processes in a plant helps to understand the process of cost formation, from the input resources to final products [56].

In this study, specific exergy costing (SPECO) method is used to obtain and understand the cost formation structure of the plant. Exergetic cost rates balances and corresponding auxiliary equations of the plant are given in Chapter 5. Since the level at which the cost balances are formulated (i.e. aggregation level) affects the results of the thermoeconomic analysis, the lowest possible aggregation level is set. Exergetic cost rate balances and corresponding auxiliary equations are formulated for each subsystem of the diesel engine powered cogeneration plant. Auxiliary equations are found by applying $\dot{\mathrm{F}}$ and $\dot{\mathrm{P}}$ principles.

Exergetic cost rate balances and corresponding auxiliary equations for each subsystem of DEPC are obtained by SPECO method and are given in the following equations (Eqs. 8.169 through 8.207). Solving the linear system consisting of related thermoeconomic equations given in these equations, we can obtain the cost flow rates and the unit exergetic costs associated with each stream of the plant. These results are given in Table 8.9.

<u>*Compressor*</u>

$$\dot{C}_{\mathrm{W_{COMP}}} + \dot{Z}_{\mathrm{COMP}} = \dot{C}_2 - \dot{C}_1 \tag{8.169}$$

$$\dot{C}_1 = 0 \ (\dot{E}_1 = 0) \tag{8.170}$$

<u>*Intercooler*</u>

$$\left(\dot{C}_{20} - \dot{C}_{21}\right) + \left(\dot{C}_{23} - \dot{C}_{24}\right) + \dot{Z}_{\mathrm{IC}} = \dot{C}_3 - \dot{C}_2 \tag{8.171}$$

$$\frac{\dot{C}_{21}-\dot{C}_{20}}{\dot{E}_{21}-\dot{E}_{20}}=\frac{\dot{C}_{24}-\dot{C}_{23}}{\dot{E}_{24}-\dot{E}_{23}} \quad \text{(P)} \tag{8.172}$$

Lubrication Oil Cooler

$$\left(\dot{C}_{24}-\dot{C}_{25}\right)+\dot{Z}_{\mathrm{LOC}}=\dot{C}_{28}-\dot{C}_{27} \tag{8.173}$$

$$\frac{\dot{C}_{28}}{\dot{E}_{28}}=\frac{\dot{C}_{27}}{\dot{E}_{27}} \tag{8.174}$$

$$c_{24}=c_{25} \tag{8.175}$$

Diesel Engine

$$\dot{C}_{3}+\dot{C}_{5}+\left(\dot{C}_{21}-\dot{C}_{22}\right)+\left(\dot{C}_{26}-\dot{C}_{27}\right)+\dot{Z}_{\mathrm{DE}}=\dot{C}_{6}+\dot{C}_{\mathrm{W-ELECTRIC}} \tag{8.176}$$

$$\frac{\dot{C}_{22}}{\dot{E}_{22}}=\frac{\dot{C}_{21}}{\dot{E}_{21}} \quad \text{(F)} \tag{8.177}$$

$$\frac{\dot{C}_{26}}{\dot{E}_{26}}=\frac{\dot{C}_{27}}{\dot{E}_{27}} \quad \text{(F)} \tag{8.178}$$

Turbine

$$\dot{C}_{6}-\dot{C}_{7}+\dot{Z}_{\mathrm{TURBINE}}=\dot{C}_{\mathrm{W_{TURBINE}}} \tag{8.179}$$

$$c_{6}=c_{7} \tag{8.180}$$

Waste Heat Boiler

$$\left(\dot{C}_{7}-\dot{C}_{8}\right)+\dot{Z}_{\mathrm{WHB}}=\dot{C}_{13}-\dot{C}_{12} \tag{8.181}$$

$$\frac{\dot{\mathrm{C}}_{12}}{\dot{\mathrm{E}}_{12}}=\frac{\dot{\mathrm{C}}_{13}}{\dot{\mathrm{E}}_{13}} \quad \text{(P)} \tag{8.182}$$

$$c_{7}=c_{8} \tag{8.183}$$

DeSO$_x$ Unit

$$\left(\dot{C}_{9}-\dot{C}_{8}\right)+\dot{Z}_{\mathrm{DeSOx}}=0 \tag{8.184}$$

$$c_{9}=0 \tag{8.185}$$

Fuel Oil Day Tank

$$\left(\dot{C}_{15} - \dot{C}_{16}\right) + \dot{Z}_{\mathrm{FDT}} = \dot{C}_4 - \dot{C}_{0,\mathrm{FO}} \tag{8.186}$$

$$\frac{\dot{C}_{15}}{\dot{E}_{15}} = \frac{\dot{C}_{16}}{\dot{E}_{16}} \text{ (F)} \tag{8.187}$$

Fuel Forwarding Module

$$\left(\dot{C}_{17} - \dot{C}_{18}\right) + \dot{Z}_{\mathrm{FFM}} = \dot{C}_5 - \dot{C}_4 \tag{8.188}$$

$$\frac{\dot{C}_{17}}{\dot{E}_{17}} = \frac{\dot{C}_{18}}{\dot{E}_{18}} = \frac{\dot{C}_{16}}{\dot{E}_{16}} \text{ (F)} \tag{8.189}$$

Condenser

$$\left(\dot{C}_{29} - \dot{C}_{30}\right) + \dot{Z}_{\mathrm{CON}} = \dot{C}_{19} - \dot{C}_{18} \tag{8.190}$$

$$\frac{\dot{C}_{18}}{\dot{E}_{18}} = \frac{\dot{C}_{19}}{\dot{E}_{19}} \text{ (P)} \tag{8.191}$$

$$c_{29} = 0 \tag{8.192}$$

Pump1

$$\dot{C}_{W_{\mathrm{PUMP1}}} + \dot{Z}_{\mathrm{P1}} = \dot{C}_{23} - \dot{C}_{25'} \tag{8.193}$$

$$\frac{\dot{C}_{23}}{\dot{E}_{23}} = \frac{\dot{C}_{25'}}{\dot{E}_{25'}} \text{ (P)} \tag{8.194}$$

Pump2

$$\dot{C}_{W_{\mathrm{PUMP2}}} + \dot{Z}_{\mathrm{P2}} = \dot{C}_{20} - \dot{C}_{22'} \tag{8.195}$$

$$\frac{\dot{C}_{20}}{\dot{E}_{20}} = \frac{\dot{C}_{22'}}{\dot{E}_{22'}} \text{ (P)} \tag{8.196}$$

Pump3

$$\dot{C}_{W_{\mathrm{PUMP3}}} + \dot{Z}_{P3} = \dot{C}_{11} - \dot{C}_{10} \tag{8.197}$$

$$\frac{\dot{C}_{10}}{\dot{E}_{10}} = \frac{\dot{C}_{11}}{\dot{E}_{11}} \text{ (P)} \tag{8.198}$$

<u>*Pump*4</u>

$$\dot{C}_{W_{\text{PUMP4}}} + \dot{Z}_{\text{P4}} = \dot{C}_{14} - \dot{C}_{14'} \qquad (8.199)$$

$$\frac{\dot{C}_{14}}{\dot{E}_{14}} = \frac{\dot{C}_{14'}}{\dot{E}_{14'}} \ \text{(P)} \qquad (8.200)$$

<u>*Pump*5</u>

$$\dot{C}_{W_{\text{PUMP5}}} + \dot{Z}_{\text{P5}} = \dot{C}_{4} - \dot{C}_{4'} \qquad (8.201)$$

$$\frac{\dot{C}_{4}}{\dot{E}_{4}} = \frac{\dot{C}_{4'}}{\dot{E}_{4'}} \ \text{(P)} \qquad (8.202)$$

<u>*Air-Water Radiator*</u>

$$\left(\dot{C}_{25'} - \dot{C}_{25}\right) + \left(\dot{C}_{22'} - \dot{C}_{22}\right) + \dot{Z}_{\text{AWR}} = \dot{C}_{32} - \dot{C}_{31} \qquad (8.203)$$

$$\frac{\dot{C}_{25'} - \dot{C}_{25}}{\dot{E}_{25'} - \dot{E}_{25}} = \frac{\dot{C}_{22'} - \dot{C}_{22}}{\dot{E}_{22'} - \dot{E}_{22}} \ \text{(P)} \qquad (8.204)$$

$$c_{31} = 0 \qquad (8.205)$$

<u>*DEPC*</u>

$$\dot{C}_{1} + \dot{C}_{5} + \dot{C}_{10} + \dot{Z}_{\text{DEPC}} = \dot{C}_{14} + \dot{C}_{15} + \dot{C}_{9} + \dot{C}_{\text{W-ELECTRIC}} \qquad (8.206)$$

$$\frac{\dot{C}_{14}}{\dot{E}_{14}} = \frac{\dot{C}_{15}}{\dot{E}_{15}} = \frac{\dot{\mathrm{C}}_{12}}{\dot{E}_{12}} \ \text{(P)} \qquad (8.207)$$

The exergetic cost parameters of the plant components are given in Table 8.10. These parameters indicate the performance of system components on a rational exergetic cost basis. The cost associated with other plant equipments must be allocated to the two product streams. For simplicity, we may divide the cost rate associated with other plant equipment (18.9 \$/h) equally between steam and net power and obtain adjusted cost rates

$$\dot{C}_{\text{electric}} = 2820 + \frac{18.9}{2} = 2829.5 \text{ \$/h}$$

$$\dot{C}_{\text{steam}} = 64.13 + \frac{18.9}{2} = 73.58 \text{ \$/h}$$

Table 8.9 The exergy flow rates, cost flow rates and the unit exergy costs associated with each stream of Sanko DEPC plant with 25.32 MW electricity and 8.1 tons/h steam production. State numbers refer to Figure 8.1

State no	$\dot{E}$ (kW)	$\dot{C}$ ($/h)	c ($/GJ)
1	0.0	0.0	0.0
2	6537	1706.2	24.20
3	4239.3	1109.0	24.22
4'	62,768	1821.0	2.70
4	62,768	1821.0	2.70
5	62,803	1933.3	2.85
6	12,250	723.0	5.50
7	3081.3	182.0	5.50
8	1497	47.20	3.0
9	817.0	0.0	0.0
10	33.0	12.01	33.71
11	34.0	12.05	33.71
12	60.0	21.85	33.71
13	240.3	87.51	33.71
14	176.1	64.13	33.71
14'	199.8	72.76	33.71
14''	240.3	87.51	33.71
15	16.2	6.41	36.64
16	1.8	0.71	36.52
17	48.0	19.0	36.70
18	3.0	1.20	37.03
19	1.5	0.60	37.03
20	1008.9	9.80	2.70
21	1473.3	14.31	2.70
22	1892.1	18.40	2.70
22'	1005.3	9.78	2.70
23	97.5	5.25	17.50
24	238.5	12.84	5.0
25	312.0	16.80	5.0
25'	92.1	4.96	5.0
26	200.0	4.25	5.90
27	424.0	9.0	2.0
28	163.0	3.46	2.0
29	0.0	0.0	0.0
30	0.3	0.03	24.20
31	0.0	0.0	0.0
32	332.7	123.2	24.20
$\dot{W}_{\text{compressor}}$	6540	1626.0	23.02
$\dot{W}_{\text{turbine}}$	9168	2277.3	23.02
$\dot{W}_{\text{pump}}$	1.26	4.80	353.0
$\dot{W}_{\text{plant}}$	25,320	2820.0	10.31

Table 8.10 The unit exergetic costs of fuels and products, relative exergetic cost difference, exergoeconomic factor, cost rate of exergy destruction, and total investment cost rate for the plant components

Component	$c_{f,k}$ (\$/GJ)	$c_{p,k}$ (\$/GJ)	r (%)	f (%)	$\dot{D}_D$ (\$/h)	$\dot{Z}^T$ (\$/h)
Compressor	23.02	24.22	4.95	41.30	120.22	80.30
Intercooler	17.50	24.20	27.70	35.20	106.70	28.30
Lubrication oil cooler	5.00	2.00	60.00	73.60	3.37	9.40
Air-water radiator	5.00	2.70	46.00	55.60	95.46	28.30
Diesel engine	2.85	10.31	72.40	63.30	296.0	510.20
Turbine	5.50	23.02	76.10	79.00	21.62	80.30
Waste heat boiler	5.50	33.71	83.70	50.45	27.80	28.30
Fuel oil day tank	36.52	2.85	92.20	95.20	47.33	9.40
Fuel forwarding module	36.70	2.85	92.20	98.60	1.27	94.40
Condenser	37.03	24.20	34.65	96.70	0.16	4.80
Pumps	353.0	33.71	90.45	81.00	1.14	4.80
$DeSO_x$	3.0	-	-	25.40	138.82	47.20

We note the followings from the exergoeconomic results of this power plant as listed in Table 8.9 and Table 8.10:

- The exergetic cost rate and the specific unit exergetic cost of the fuel entering the plant are 1806 \$/h and 2.70 \$/GJ, respectively. The corresponding values of these costs for the diesel engines are 1933.3 \$/h and 2.85 \$/GJ.
- The capital investment cost, the operating and maintenance costs, and the total cost of the DEPC system are found to be 770.5 \$/h, 174.5 \$/h and 945 \$/h, respectively.
- The net electrical power output of the plant is 25.32 MW. The exergetic cost rate and the specific unit exergetic cost of the power produced by the plant are 2844 \$/h and 10.31 \$/GJ.
- The steam output of the system is 8.1 tons/h at 170°C and 8 bars. The exergetic cost rate and specific unit exergetic costs of the steam are 87.73 \$/h and 33.71 \$/GJ, respectively.
- Exergetic cost rates difference between electric and steam production outputs is very high for this DEPC system. This is directly proportional to the exergy allocation of fuel between steam and electric outputs. As indicated in Table 8.10, diesel engine is the most exergy destructive component of the plant. The

exergoeconomic factor of the diesel engine is determined to be 63.3%. Depending on total dominate effect of the highest investment and exergetic destruction cost rates of the engine itself, a decrease of the exergy destruction could be cost effective even if this would increase the investment cost for the diesel engine.

- The exergy unit cost is highest for the pump work since all exergy available at the exit of the pump is supplied by mechanical power which is the most expensive "fuel" in the system. The exergoeconomic factor is rather high (81%) due to low initial investment and exergetic destruction cost rates.
- Exergoeconomic factors for fuel forwarding module, fuel oil day tank, and condenser are 98.6%, 95.2%, and 96.7%, respectively. Thus, it is expected that the cost effectiveness of the entire system can be improved by reducing the total capital investment and O&M costs for these components.
- The exergetic cost value is 23.02 $/GJ for the compressor work and 5.50 $/GJ for the exhaust gas stream. This difference makes the exergetic destruction cost rate for the compressor dominant in the exergoeconomic factor. The exergoeconomic factor of the compressor is 41.3%, which is rather low compared with the other components. This value is 79.0% for the turbine. Thermoeconomic improvement of the turbocharger unit can be achieved by a decrease of the total effect of the initial investment and destruction cost rates.
- Waste heat boiler unit involves an exergetic destruction rate of 1404 kW (see Table 8.4). The specific cost of this unit is due to considering the exhaust gases as fuel. The exergetic unit cost of steam is inversely proportional to the exergetic efficiency of this unit. Because of the low mass flow rate of saturated steam as compared to exhaust gases in the waste heat boiler, exergetic destruction cost rates involve high weighing factor in the denominator of exergoeconomic factor relation. Cost effectiveness for this unit can be achieved by reducing exergy destruction.
- The exergoeconomic factors of intercooler and air-water radiators are 35.2% and 55.60%, respectively. This low exergoeconomic factor of intercooler is the second lowest value among the plant components and it is due to high value of the exergetic destruction cost rate compared with low total investment cost.

- The relative cost difference for lubrication oil cooler is determined to be 60.0%, which is the second lowest value after the compressor. It also has a low destruction cost rate of 3.37 \$/h. It maybe suggested that a decrease in the investment cost may improve the cost effectiveness of the system.
- Desulphurization ($DeSO_x$) unit has the lowest exergoeconomic factor among the components of the plant. This is expected since it is the most destructive unit in the plant. This unit is necessary to keep the emissions below legal limits but the more efficient processes may be achieved. This problem points out the need for environmentally optimized systems together with thermoeconomical ones.

8.10 Thermoeconomic Optimization

The objective function expresses the optimization criterion as a function of dependent and independent variables. For the DEPC system, the objective function can be written as

$$\textit{minimize}\ \dot{C}_{\mathrm{P,total}} = \dot{C}_{\mathrm{F,total}} + \dot{Z}_{\mathrm{total}}^{\mathrm{CI}} + \dot{Z}_{\mathrm{total}}^{\mathrm{OM}} \tag{8.208}$$

The variables, the total cost rate of fuel $\dot{C}_{\mathrm{F,total}}$, the total cost rate of capital investment $\dot{Z}_{\mathrm{total}}^{\mathrm{CI}}$, and the total cost rate of O&M costs $\dot{Z}_{\mathrm{total}}^{\mathrm{OM}}$, are functions of decision variables. In this study, we minimize the total cost rate associated with the product $\dot{C}_{\mathrm{P,total}}$ instead of the cost per unit of product exergy c_{P}. As stated in Chapter 6 in detail, the cost optimal exergetic efficiency approach is used for an isolated system component and the following constraints are used for the existing DEPC system: The net power produced by Sanko DEPC system is 25.32 MW and the steam generated is 8.1 tons/hr (2.25 kg/sec) at 8 bars. Specifically, for every system component we must use equations 6.1 and 6.2 as constraints that is for a *k*th component to be optimized: $\dot{E}_{\mathrm{P,k}}$= constant and $c_{\mathrm{F,k}}$= constant. Also, for each component of the DEPC plant we must calculate

- an optimum value of the relative cost difference $r_{\mathrm{k}}^{\mathrm{opt}}$ (the actual value of r_{k} is always greater than $r_{\mathrm{k}}^{\mathrm{opt}}$),

- an optimum value of the exergetic efficiency $\varepsilon_{\mathrm{k}}^{\mathrm{opt}}$,
- an estimate of the costs of the electricity and steam which are the total plant products, based on the given total costs of the heavy fuel oil and the calculated costs of the DEPC plant components. That is,

$$minimize\ \dot{C}_{\mathrm{W_{net},DE}} = \dot{C}_3 + \dot{C}_5 + \left(\dot{C}_{21} - \dot{C}_{22}\right) + \left(\dot{C}_{26} - \dot{C}_{27}\right) - \dot{C}_6 + \dot{Z}_{\mathrm{DE}} \quad (8.209)$$

$$minimize\ \dot{C}_{13} = \left(\dot{C}_7 - \dot{C}_8\right) + \dot{C}_{12} + \dot{Z}_{\mathrm{WHB}} \quad (8.210)$$

- the cost of the total plant exergy losses $\dot{D}_{\mathrm{D,total}}$, and its optimal value $\dot{D}_{\mathrm{D,total}}^{\mathrm{opt}}$.

For an existing system such DEPC plant of this study, performance evaluation and optimization procedure are parallel to what it may be considered as "performance improvement" and "searching a good solution" for the overall system rather than to find a global optimum. Moreover, for such a system, total capital investment costs and operating and maintaining costs are taken as sunk costs which may not be included in the exergetic cost rate balances. However, by using the SPECO method appropriately in this study, thermoeconomic analysis and optimization include all investment data for each components of the cogeneration system neglecting any simplification in such a manner. In the following section, major plant components which have an effect on the optimizing the plant exergy destruction cost rates and the cost of the electricity and steam are optimized by using the iterative thermoeconomical procedure.

8.10.1 Compressor

In order to optimize the exergetic efficiency and the cost of the compressor destruction, from Equation 8.169, we may write

$$minimize\ \dot{C}_2 = \dot{C}_1 + \dot{C}_{\mathrm{W,COMP}} + \dot{Z}_{\mathrm{COMP}} \quad (8.211a)$$

or

$$minimize\, c_{p,2}\dot{E}_2 = c_1\dot{E}_1 + c_{f,COMP}\dot{E}_{W,COMP} + \dot{Z}_{COMP} \quad (8.211b)$$

The compressor pressure ratio P_2/P_1 is taken as the decision variable and it is required to be given in the range between $1.10 \le P_2 / P_1 \le 3.85$ for the operational restrictions of turbocharger unit in the system. Also, for cost reasons, the maximum value of compressor isentropic efficiency is less than 90% ($\eta_c = 0.80$ for this study). In Table 8.11, dependent variables are given during the searching process for thermoeconomically optimal range of the compressor. The base case value (i.e. 2.90) is the actual pressure ratio of the DEPC system at full load. In this study, not only the estimated convergent range but also all possible variations of cost structure of the subsystems and corresponding exergetic efficiencies and destruction cost rates are taken into account for a general perspective.

Table 8.11 Dependent variables obtained during the searching procedure of the appropriate thermoeconomical optimal range of compressor of the DEPC system

Variable	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7
P_2/P_1	1.10	1.50	1.75	2.00	2.25	2.50	2.90 base case
Fixed parameters $c_{f,comp}$ (\$/GJ) $\dot{E}_{P,comp}$ (kW)	23.02 6536	23.02 6536	23.02 6536	23.02 6536	23.02 6536	23.02 6536	23.02 6536
ε_{comp}	0.2385	0.4264	0.5198	0.6007	0.6720	0.7358	0.8260
$c_{p,comp}$ (\$/GJ)	10.08	15.33	17.95	20.21	22.20	23.98	24.22
Δr_{comp} (%)	-61.20	-38.36	-26.97	-17.16	-8.51	-0.78	0.00
$\dot{E}_{D,comp}$ (kW)	6036	4547.4	3807.6	3166	2601	2095	1380
$\dot{D}_{D,comp}$ (\$/h)	500.22	251.0	246.0	230.3	207.9	173.6	120.22

As shown in the Table 8.11, in the first seven cases, up to the base case value, exergetic efficiency increases remarkably with the decreasing value of the exergy destruction. This, as expected, makes the product cost value higher but the exergetic destruction cost rate lower than the previous case. Again, up to the actual base case value, because of the reason that the exergetic efficiency term has the higher weighing factor in the definition of relative cost difference relation (Equation 5.37), pressure ratios less than 2.90 have some negative effects on the destruction cost side.

The total investment cost of the components in the system including compressor may be considered as constant for the requirement of the existing system.

Table 8.11 continued

Variable	Case 8	Case 9	Case 10	Case 11	Case 12	Case 13	Case 14
P_2/P_1	3.00	3.25	3.50	3.60	3.70	3.80	3.85
Fixed parameters $c_{f,comp}$ (\$/GJ) $\dot{E}_{P,comp}$ (kW)	23.02 6536	23.02 6536	23.02 6536	23.02 6536	23.02 6536	23.02 6536	23.02 6536
ε_{comp}	0.8463	0.8948	0.9397	0.9568	0.9734	0.9895	0.9974
$c_{p,comp}$ (\$/GJ)	27.07	28.43	29.69	30.16	30.63	31.08	31.30
Δr_{comp} (%)	12.64	18.60	24.02	26.10	28.11	30.1	31.02
$\dot{E}_{D,comp}$ (kW)	1212	828.0	474.0	336.0	204.0	78.0	18.0
$\dot{D}_{D,comp}$ (\$/h)	100.44	68.62	39.30	27.84	16.91	6.46	1.50

In the remaining seven cases, more attention must be paid to exergetic destruction rates and corresponding cost rates which may be far from optimum range, since large deviations from the best range can be observed thermoeconomically. On the other hand, relative cost difference is positively affected when we take the components destruction cost into account. In Figure 8.6, product cost values are given with respect to the variations of pressure ratios and corresponding results of the exergetic efficiencies of the compressor. As shown in Figure 8.6 together with Table 8.11, optimum thermoeconomic range lies between 2.50 and 3.50 for pressure ratio. Optimum value for the destruction cost of the component must be searched in this range even when the product cost value is higher than the base case. In Figure 8.7, destruction cost rate values are given with respect to the selected pressure ratios and corresponding exergetic efficiencies of compressor.

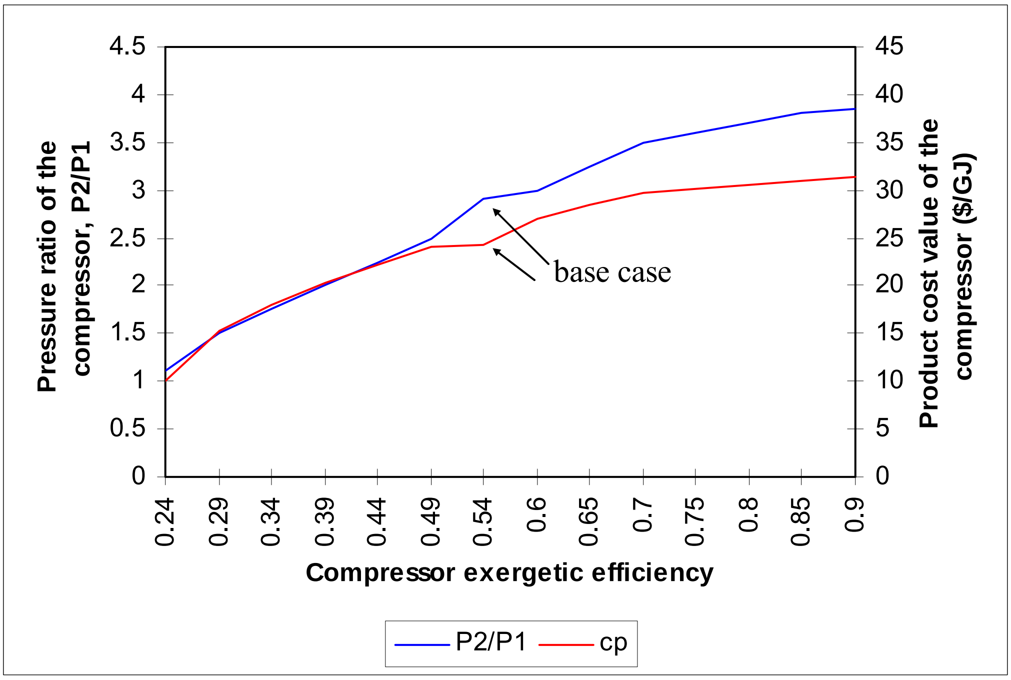

Figure 8.6 Variation of product cost values of compressor with respect to different pressure ratios of the compressor and corresponding calculated exergetic efficiencies

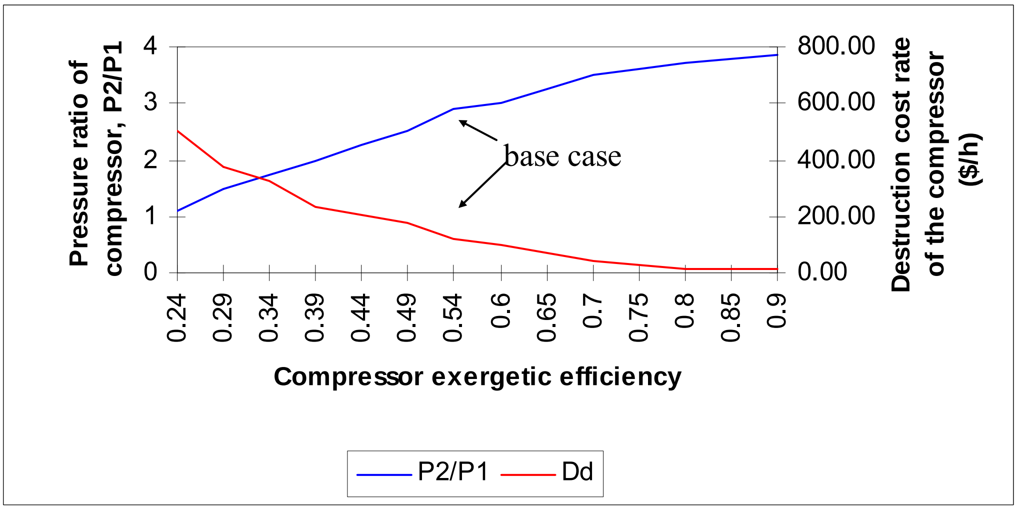

Figure 8.7 Variation of destruction cost rate of the compressor with respect to different pressure ratios of the compressor and corresponding calculated exergetic efficiencies

It is found that the optimum pressure ratio $(P_2 / P_1)^{opt}$ equals to 3.12. Using the first optimization approach given in Chapter 6, we keep the cost rate of the fuel and product exergy value of the compressor constant and calculate the optimal values of the exergetic efficiency and destruction cost rate of the component at this optimum pressure ratio value as ε_{comp}^{opt}=0.8697 and $\dot{D}_{D,comp}^{opt}$=85.5 $/h. Figure 8.8 shows the iterations of the pressure ratio of the compressor in the range between case 6 and case 10 as given in Table 8.12, and variation of destruction cost rates and the corresponding values of exergetic efficiencies through optimization process. Figure 8.9 shows variation of relative cost difference and product cost with respect to the corresponding exergetic efficiencies of the compressor through iterations of the optimization process.

Table 8.12 Dependent variables obtained during the iterations of the pressure ratio for the optimization procedure of exergetic efficiency and corresponding destruction cost rate of the compressor of DEPC system

Variable	Case 6 →	1st Iteration	2nd Iteration	2nd Iteration	1st Iteration	Case 10 ←
P_2/P_1	2.50	3.045	**3.118**	**3.118**	3.160	3.50
Fixed parameters $c_{f,comp}$ ($/GJ) $\dot{E}_{P,comp}$ (kW)	23.02 6536	23.02 6536	**23.02** **6536**	**23.02** **6536**	23.02 6536	23.02 6536
ε_{comp}	0.7358	0.8553	**0.8697**	**0.8697**	0.8778	0.9397
$c_{p,comp}$ ($/GJ)	23.98	27.33	**27.73**	**27.73**	27.96	29.69
Δr_{comp} (%)	-0.78	18.7	**20.5**	**20.5**	21.5	24.02
$\dot{E}_{D,comp}$ (kW)	2095	1146	**1032**	**1032**	968.0	474.0
$\dot{D}_{D,comp}$ ($/h)	173.6	95.0	**85.5**	**85.5**	80.2	39.30

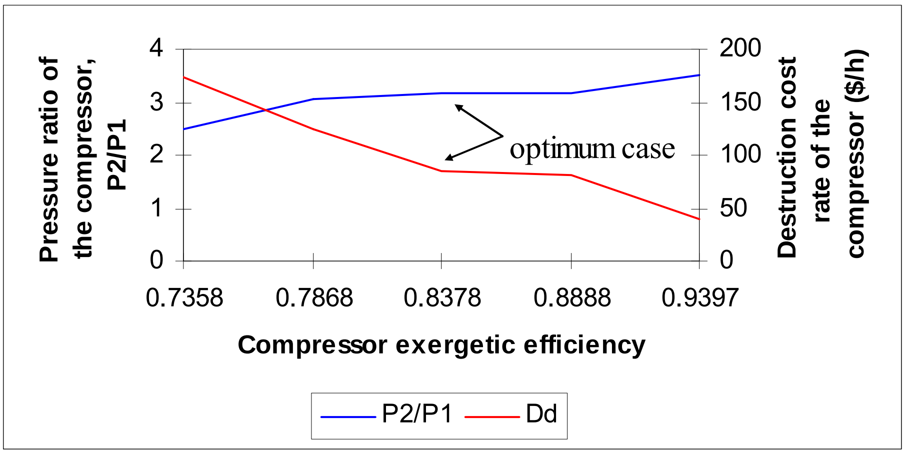

Figure 8.8 Variation of destruction cost rate of the compressor with respect to the iterated pressure ratios and corresponding calculated exergetic efficiencies of the compressor through optimization process

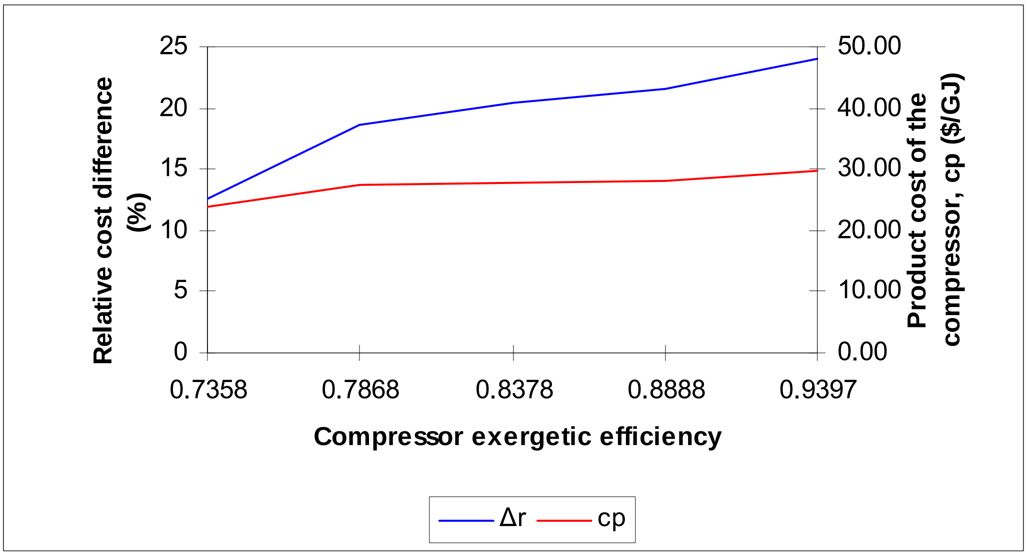

Figure 8.9 Variation of relative cost difference and product cost with respect to the corresponding exergetic efficiencies of the compressor through iterations of the optimization process

8.10.2 Intercooler

Intercooler in the DEPC system is a heat exchanger network. The cost rate balance of the intercooler is given in Equation 8.171. In order to optimize the exergetic efficiency and the destruction cost rate of the intercooler we use thermoeconomical isolation approach and minimize the product cost of the component as

$$minimize \;\; \dot{C}_3 = \left(\dot{C}_{20} - \dot{C}_{21}\right) + \left(\dot{C}_{23} - \dot{C}_{24}\right) + \dot{C}_2 + \dot{Z}_{\mathrm{IC}} \qquad (8.212a)$$

or

$$minimize \;\; c_3\dot{E}_3 = \left(c_{20}\dot{E}_{20} - c_{21}\dot{E}_{21}\right) + \left(c_{23}\dot{E}_{23} - c_{24}\dot{E}_{24}\right) + c_2\dot{E}_2 + \dot{Z}_{\mathrm{IC}} \qquad (8.212b)$$

In this system, according to their different mass flow rates, high-temperature (HT) Water and low-temperature (LT) Water can be separated into two different heat exchanging processes in intercooler by using weighing factor. Thus, the decision variables in this component are taken as inlet temperatures of two water streams from AWR unit as T_{20} and T_{23}. First process takes place between hot air from compressor unit and HT water from AWR unit. The maximum temperature difference in the first process is 100.3ºC. Therefore, hot air from the compressor of the turbocharger unit cannot be cooled by more than 100.3ºC (to 71.7ºC). When we take HT Water inlet temperature to the intercooler as decision variable, the optimum range can be determined in the range of 71.7 ºC and 86.5ºC. In Table 8.13, exergetic efficiencies, destruction cost rates and corresponding product costs calculated during the iterations in the first heat exchanging process are given for the intercooler at the optimum pressure ratio of the compressing process.

Table 8.13 Dependent variables obtained during the iterations of the HT Water inlet temperature for the optimization procedure of exergetic efficiency and corresponding destruction cost rate of the intercooler-1 of DEPC system

Variable	⟶	1st Iteration	2nd Iteration	3rd Iteration
P_2/P_1 **T_{20} (°C)**	3.118 71.7(min)	3.118 73.05	3.118 73.43	**3.118** **76.53**
Fixed parameters $c_{f,IC-1}$ ($/GJ) $\dot{E}_{P,IC-1}$ (kW)	2.70 1817	2.70 1817	2.70 1817	**2.70** **1817**
ε_{IC-1}	0.4571	0.4442	0.4404	**0.4035**
$c_{p,IC-1}$ ($/GJ)	2.55	2.63	2.67	**3.10**
Δr_{IC-1} (%)	-5.55	-2.60	-1.11	**14.8**
$\dot{E}_{D,IC-1}$ (kW)	986.4	1009	1017	**1084**
$\dot{D}_{D,IC-1}$ ($/h)	9.60	9.82	9.90	**10.54**

Table 8.13 continued

Variable	3rd Iteration	2nd Iteration	1st Iteration	⟵
P_2/P_1 **T_{20} (°C)**	**3.118** **76.53**	3.118 78.44	3.118 80.85	3.118 86.5(max)
Fixed parameters $c_{f,IC-1}$ ($/GJ) $\dot{E}_{P,IC-1}$ (kW)	**2.70** **1817**	2.70 1817	2.70 1817	2.70 1817
ε_{IC-1}	**0.4035**	0.3618	0.2865	0.1654
$c_{p,IC-1}$ ($/GJ)	**3.10**	3.34	4.28	5.23
Δr_{IC-1} (%)	**14.8**	23.4	58.5	93.7
$\dot{E}_{D,IC-1}$ (kW)	**1084**	1160	1296	1516
$\dot{D}_{D,IC-1}$ ($/h)	**10.54**	11.30	12.60	14.74

As shown in Table 8.13, the optimum value of HT Water inlet temperature and corresponding optimum values of exergetic efficiency and destruction cost rate of the first heat exchanging process in the intercooler are T_{20}=76.5ºC, $\varepsilon_{\text{IC-1}}^{\text{opt}} = 0.4035$, and $\dot{D}_{\text{D,IC-1}}^{\text{opt}} = 10.54\ \$/\text{h}$.

The second heat exchanging process in the intercooler is between the air from the first section and LT water. The maximum temperature difference in the second process is 103.6ºC. Therefore, hot air from the first section of the intercooler unit cannot be cooled by more than 103.6ºC (to 38.4ºC). When we take LT Water inlet temperature to the intercooler as decision variable, the optimum range becomes 38.4ºC and 61.5ºC. In Table 8.14, exergetic efficiencies, destruction cost rates and corresponding product costs calculated during the iterations in the second heat exchanging process are given for the intercooler at the optimum pressure ratio of the compressing process and the optimum inlet temperature of the HT Water from first heat exchanging process in the intercooler. Thus, in the intercooler network, optimum values of exergetic efficiency and destruction cost rate are determined to be $\varepsilon_{\text{IC}}^{\text{opt}} = 0.6388$ and $\dot{D}_{\text{D,IC-1}}^{\text{opt}} = 77.24\ \$/\text{h}$, respectively by using the optimal values determined through the iterations in Table 8.13 and 8.14.

Figures 8.10 and 8.11 show variations of destruction cost rates of the intercooler with respect to the HT and LT Water streams temperatures, respectively, and corresponding calculated exergetic efficiencies of the intercooler network through optimization process. In table 8.13, the second section of the intercooler in which heat exchange process exists between hot air and low temperature water, is the component with the largest Δr values. As the decision variable, T_{23} is varied further it becomes apparent that some limits have been reached which do not allow for further significant improvements in the intercooler network. In addition, any attempt to reduce the relative cost difference value for this component may result in an increase in the heat transfer area of the heat exchanger or an increase of the air-fuel ratio of the engine which may cause incomplete combustion.

Table 8.14 Dependent variables obtained during the iterations of the LT Water inlet temperature for the optimization procedure of exergetic efficiency and corresponding destruction cost rate of the intercooler-2 of DEPC system

Variable	→	**1st Iteration**	**2nd Iteration**	**3rd Iteration**	**4th Iteration**	**5th Iteration**	**6th Iteration**
P_2/P_1	3.118	3.118	3.118	3.118	3.118	3.118	**3.118**
T_{20} (°C)	76.53	76.53	76.53	76.53	76.53	76.53	**76.53**
T_{23} (°C)	50.72	44.0	47.3	49.1	50.0	50.5	**50.72**
Fixed parameters							
$c_{f,IC-2}$ ($/GJ)	17.50	17.50	17.50	17.50	17.50	17.50	17.50
$\dot{E}_{P,IC-2}$ (kW)	2422.4	2422.4	2422.4	2422.4	2422.4	2422.4	2422.4
$\varepsilon_{IC\text{-}2}$	0.1269	0.1772	0.2061	0.2215	0.2292	0.2334	**0.2353**
$c_{p,IC-2}$ ($/GJ)	16.57	18.43	19.76	21.02	21.65	22.44	**23.21**
$\Delta r_{IC\text{-}2}$ (%)	-5.31	5.32	12.90	20.11	23.71	28.23	**32.63**
$\dot{E}_{D,IC-2}$ (kW)	2115.0	1993.1	1923.1	1886.0	1867.2	1857.0	**1852.4**
$\dot{D}_{D,IC-2}$ ($/h)	76.14	71.8	69.3	68.0	67.2	66.9	**66.7**

Table 8.14 continued

Variable	**6th Iteration**	**5th Iteration**	**4th Iteration**	**3rd Iteration**	**2nd Iteration**	**1st Iteration**	←
P_2/P_1	**3.118**	3.118	3.118	3.118	3.118	3.118	3.118
T_{20} (°C)	**76.53**	76.53	76.53	76.53	76.53	76.53	76.53
	50.72	51.2	51.5	52.2	53.5	56.1	61.5(max)
Fixed parameters							
$c_{f,IC-2}$ ($/GJ)	17.50	17.50	17.50	17.50	17.50	17.50	17.50
$\dot{E}_{P,IC-2}$ (kW)	2422.4	2422.4	2422.4	2422.4	2422.4	2422.4	2422.4
$\varepsilon_{IC\text{-}2}$	**0.2353**	0.2394	0.2419	0.2478	0.2586	0.2800	0.3232
$c_{p,IC-2}$ ($/GJ)	**23.21**	24.01	24.86	25.12	25.97	27.32	31.44
$\Delta r_{IC\text{-}2}$ (%)	**32.63**	37.20	42.06	43.54	48.40	56.11	79.65
$\dot{E}_{D,IC-2}$ (kW)	**1852.4**	1842.5	1836.4	1822.1	1796.0	1744.1	1639.5
$\dot{D}_{D,IC-2}$ ($/h)	**66.7**	66.3	66.1	65.6	64.6	63.0	59.0

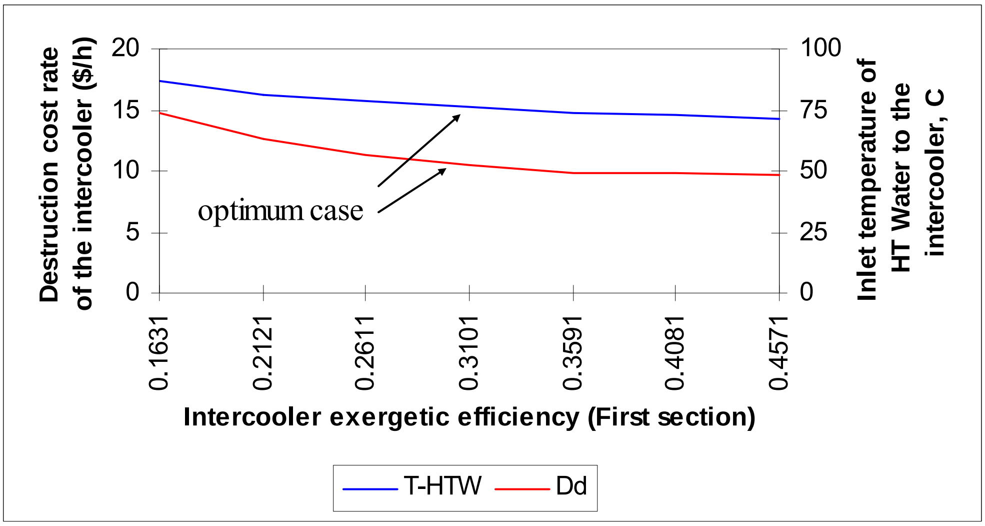

Figure 8.10 Variation of destruction cost rate of the intercooler (first section) with respect to the iterated HT Water inlet temperatures and corresponding calculated exergetic efficiencies of the intercooler through optimization process

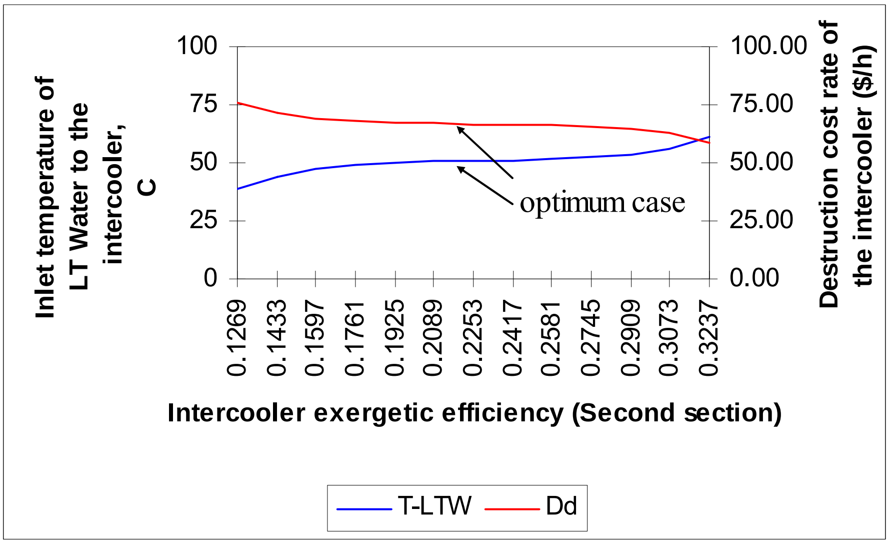

Figure 8.11 Variation of destruction cost rate of the intercooler (second section) with respect to the iterated LT Water inlet temperatures and corresponding calculated exergetic efficiencies of the intercooler through optimization process

8.10.3 Turbine

The cost rate balance of the turbine is given in Equation 8.179. By minimizing the product cost term, we can obtain the optimum values of the exergetic efficiency and the corresponding destruction cost rate of the component:

$$minimize\ \dot{C}_{W_{TURBINE}} = \dot{C}_6 - \dot{C}_7 + \dot{Z}_{TURBINE} \tag{8.213a}$$

or

$$minimize\ c_{W_{TURBINE}} \dot{E}_{W_{TURBINE}} = c_6\dot{E}_6 - c_7\dot{E}_7 + \dot{Z}_{TURBINE} \qquad (8.213b)$$

The decision variable in turbine is selected as the exit temperature of exhaust gas through the turbine T_7. The base case value of the exit temperature is 302ºC. The iterations performed for the optimization of turbine is given in Table 8.15. The turbine of the turbocharger unit has the smallest Δr values in the plant. The optimum values found are $\varepsilon_{Turb}^{opt} = 0.8775$ and $\dot{D}_{D,Turb}^{opt} = 29.70$ \$/h. These values are obtained at T_7=303ºC, which is the closest value to the corresponding base case value (302ºC). This means that turbine is the best performed component in the plant according to the decision variables selected. In this performance improvement study, it may be suggested that a newly designed turbine may increase the investment cost of the plant. However, it is not necessary to consider the effects of new investment opportunities of any plant component since this step may cause a significant change in the objective function value (but also significant improvement in the optimization study). In Figures 8.12 and 8.13, variations of destruction cost rates and product cost of the turbine at the iterated values of the exhaust exit temperature with respect to the corresponding calculated exergetic efficiencies of the turbine are given.

Table 8.15 Dependent variables obtained during the iterations of the turbine exhaust outlet temperature for the optimization procedure of exergetic efficiency and corresponding destruction cost rate of the turbine of DEPC system

Variable	⟶	**1st Iteration**	**2nd Iteration**	**3rd Iteration**	**4th Iteration**	**5th Iteration**	**6th Iteration**
P_2/P_1	**3.118**	**3.118**	**3.118**	**3.118**	**3.118**	**3.118**	**3.118**
T_{20} (°C)	**76.53**	**76.53**	**76.53**	**76.53**	**76.53**	**76.53**	**76.53**
T_{23} (°C)	**50.72**	**50.72**	**50.72**	**50.72**	**50.72**	**50.72**	**50.72**
T_7 (°C)	267(min)	277	284.3	290	298	300	**303**
Fixed parameters							
$c_{f,Turb}$ (\$/GJ)	5.50	5.50	5.50	5.50	5.50	5.50	**5.50**
$\dot{E}_{P,Turb}$ (kW)	8076	8076	8076	8076	8076	8076	**8076**
ε_{Turb}	0.9919	0.9616	0.9388	0.9206	0.8944	0.8877	**0.8775**
$c_{p,Turb}$ (\$/GJ)	5.53	5.30	5.0	4.93	4.78	4.66	**4.60**
Δr_{Turb} (%)	0.54	-3.64	-9.10	-10.4	-13.10	-15.30	**-16.40**
$\dot{E}_{D,Turb}$ (kW)	99.23	470.4	750.0	973.0	1294.0	1376.0	**1500**
$\dot{D}_{D,Turb}$ (\$/h)	2.00	9.31	14.90	19.30	25.62	27.24	**29.70**

Table 8.15 continued

Variable	6th Iteration	5th Iteration	4th Iteration	3rd Iteration	2nd Iteration	1st Iteration	←
P_2/P_1 **T_{20} (°C)**	**3.118** **76.53** **50.72** **303**	**3.118** **76.53** **50.72** 305	**3.118** **76.53** **50.72** 307	**3.118** **76.53** **50.72** 311	**3.118** **76.53** **50.72** 314.3	**3.118** **76.53** **50.72** 320	**3.118** **76.53** **50.72** 327(max)
Fixed parameters $c_{f,Turb}$ (\$/GJ) $\dot{E}_{P,Turb}$ (kW)	**5.50** **8076**	5.50 8076	5.50 8076	5.50 8076	5.50 8076	5.50 8076	5.50 8076
ε_{Turb}	**0.8775**	0.8707	0.8638	0.8498	0.8381	0.8173	0.7910
$c_{p,Turb}$ (\$/GJ)	**4.60**	4.52	4.50	4.34	4.25	4.07	3.85
Δr_{Turb} (%)	**-16.40**	-17.82	-18.20	-21.09	-22.72	-26.0	-30.0
$\dot{E}_{D,Turb}$ (kW)	**1500**	1584.0	1668.5	1840.0	1983.3	2238.0	2560.3
$\dot{D}_{D,Turb}$ (\$/h)	**29.70**	31.40	33.04	36.43	39.30	44.31	50.7

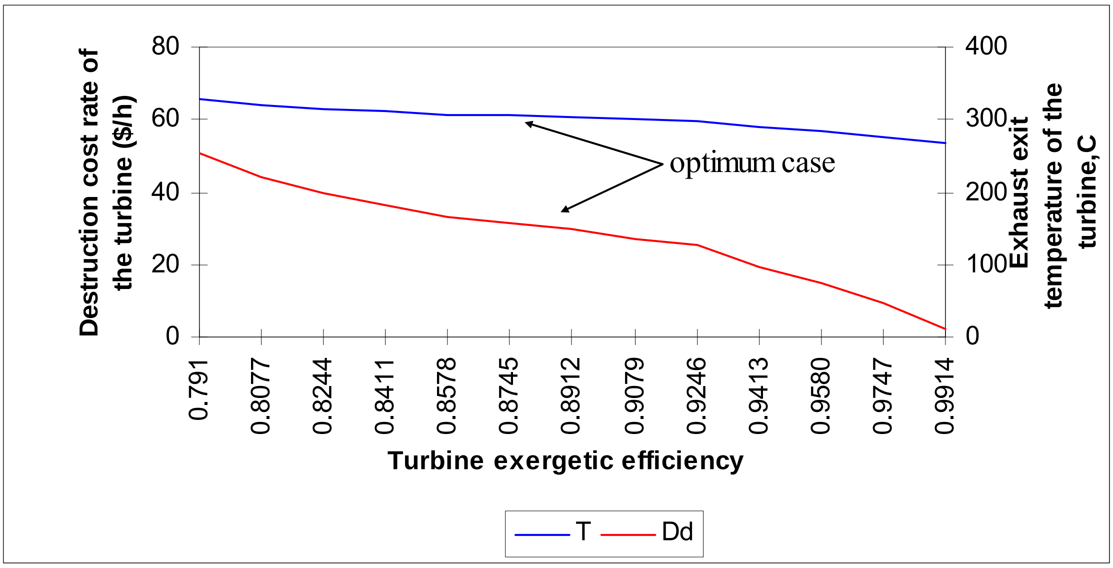

Figure 8.12 Variation of destruction cost rate of the turbine with respect to iterated turbine exhaust exit temperatures and corresponding calculated exergetic efficiencies of the turbine through optimization process

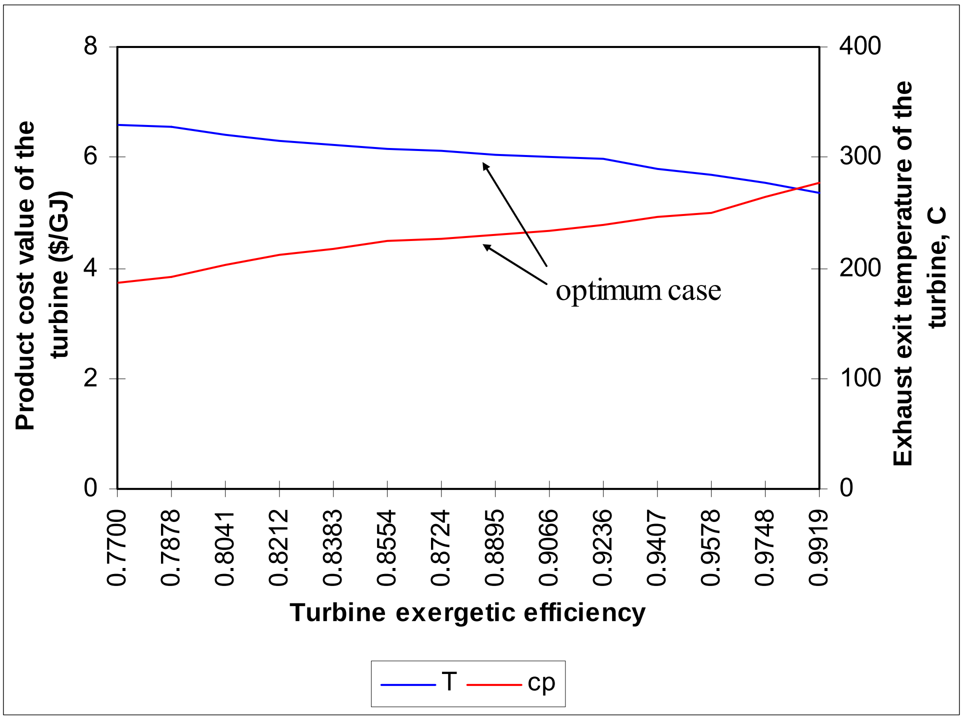

Figure 8.13 Variation of product cost values of turbine with respect to iterated exhaust exit temperatures of the turbine and corresponding calculated exergetic efficiencies

8.10.4 Waste Heat Boiler

The thermoeconomic cost balance equation for waste heat boiler (WHB) is given in Equation 8.181. By minimizing the product cost rate, i.e. the cost rate of the steam produced, we can optimize the destruction cost rate and the exergetic efficiency of the WHB unit. The objective function of this subcomponent is written as

$$minimize\, \dot{C}_{13} = (\dot{C}_7 - \dot{C}_8) + \dot{C}_{12} + \dot{Z}_{WHB} \tag{8.214a}$$

or

$$minimize\, c_{13}\dot{E}_{13} = (c_7\dot{E}_7 - c_8\dot{E}_8) + c_{12}\dot{E}_{12} + \dot{Z}_{WHB} \tag{8.214b}$$

The relative cost difference of WHB unit is among the higher ones in the system. This is because of the low mass flow rate of saturated steam as compared to exhaust gas in the unit. Since in thermoeconomic optimization assumptions product exergy value is a constraint, we may increase the exergetic efficiency (thus reduce the relative cost difference) of the component by decreasing the inlet temperature of feed water or by decreasing the exhaust exit temperature through WHB. This is more cost effective than the first option since after WHB unit, exergy of the stream is destructed in $DeSO_x$ unit. Because of the low mass flow rate of steam, the second option has no any improvement on the exergetic efficiency of the component. Therefore, we must search the optimum solution for WHB in the lower bounds of inlet temperatures of feed water relative to the base case instead of any changing exhaust temperature (see Table 8.16). As shown in Table 8.16, optimal values for exergetic efficiency and destruction cost rate of the WHB unit are $\varepsilon_{\mathrm{WHB}}^{\mathrm{opt}} = 0.1249$ and $\dot{D}_{\mathrm{D,WHB}}^{\mathrm{opt}} = 27.01$ \$/h, respectively. These values are very close to those at actual base case when compared to other subcomponents in the DEPC system. In Figures 8.14 and 8.15, variations of destruction cost rates and product cost of the waste heat boiler at the iterated values of the feed water inlet temperature with respect to the corresponding calculated exergetic efficiencies of the waste heat boiler unit are given.

Table 8.16 Dependent variables obtained during iterations of the feed water inlet temperature to the waste heat boiler for the optimization procedure of exergetic efficiency and corresponding destruction cost rate of the waste heat boiler of DEPC system

Variable	$\longrightarrow$	**1st Iteration**	**2nd Iteration**	**3rd Iteration**	**3rd Iteration**	**2nd Iteration**	**1st Iteration**	$\longleftarrow$
P_2/P_1	**3.118**	3.118	3.118	**3.118**	**3.118**	3.118	3.118	**3.118**
T_{20} (°C)	**76.53**	76.53	76.53	**76.53**	**76.53**	76.53	76.53	**76.53**
T_{23} (°C)	**50.72**	50.72	50.72	**50.72**	**50.72**	50.72	50.72	**50.72**
T_7 (°C)	**303**	303	303	**303**	**303**	303	303	**303**
T_{12} (°C)	60 (min)	74	81.1	**81.6**	**81.6**	81.2	84.4	95 (max)
Fixed parameters								
$c_{\mathrm{f,WHB}}$ (\$/GJ)	5.50	5.50	5.50	**5.50**	**5.50**	5.50	5.50	5.50
$\dot{E}_{\mathrm{P,WHB}}$ (kW)	240.3	240.3	240.3	**240.3**	**240.3**	240.3	240.3	240.3
$\varepsilon_{\mathrm{WHB}}$	0.1399	0.1310	0.1254	**0.1249**	**0.1249**	0.1253	0.1225	0.1123
$c_{\mathrm{p,WHB}}$ (\$/GJ)	32.11	32.63	33.01	**33.02**	**33.02**	33.01	33.11	33.81
Δr_{WHB} (%)	82.0	82.52	82.90	**82.91**	**82.91**	82.90	83.0	83.70
$\dot{E}_{\mathrm{D,WHB}}$ (kW)	1325.1	1339.0	1347.5	**1348.2**	**1348.2**	1347.6	1352	1367.7
$\dot{D}_{\mathrm{D,WHB}}$ (\$/h)	26.10	26.62	27.0	**27.01**	**27.01**	27.0	27.10	27.80

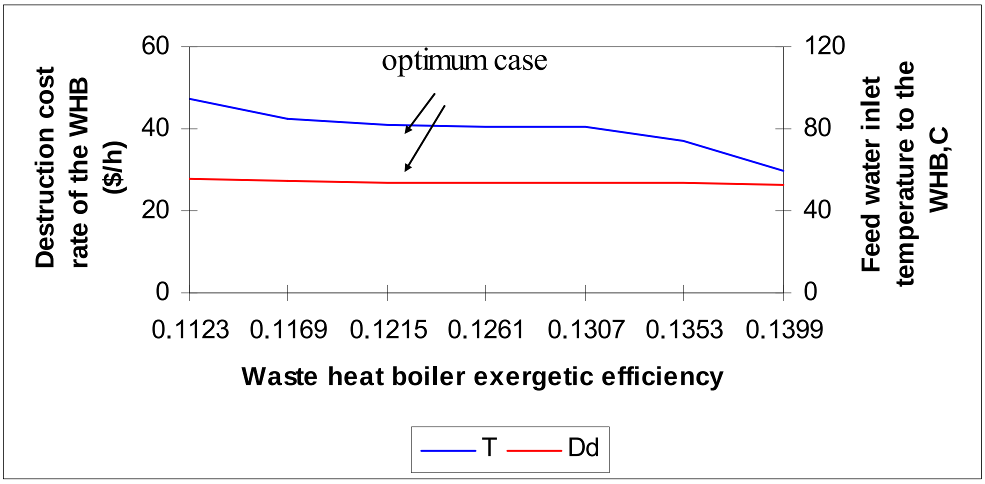

Figure 8.14 Variation of destruction cost rate of the waste heat boiler with respect to iterated feed water inlet temperatures and corresponding calculated exergetic efficiencies of the WHB through optimization process

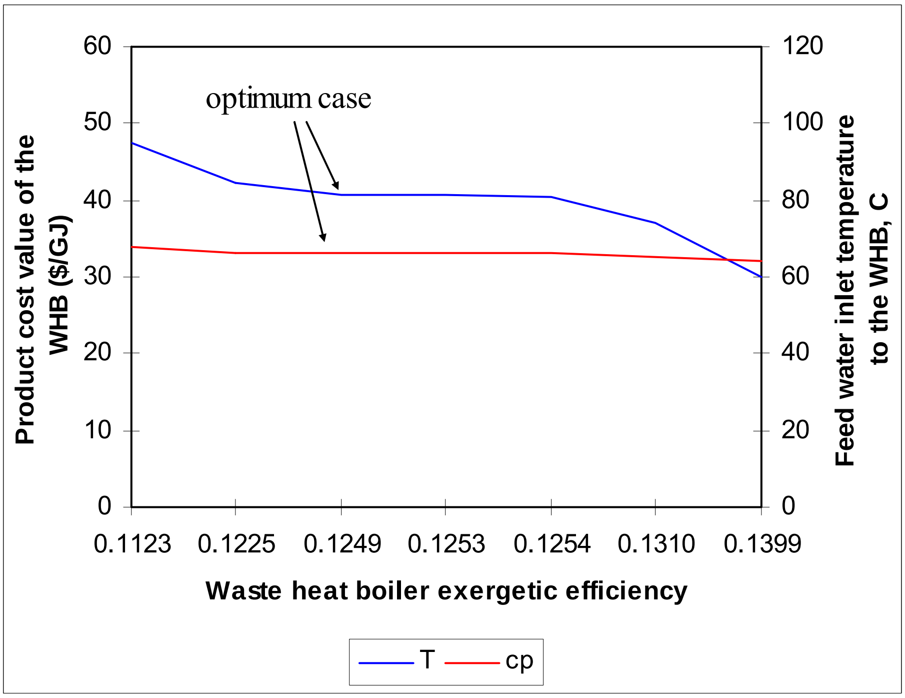

Figure 8.15 Variation of product cost values of waste heat boiler with respect to the iterated feed water inlet temperatures of the WHB and corresponding calculated exergetic efficiencies through optimization process

8.10.5 Lubrication Oil Cooler

In the base case, the temperature of low temperature water (LTW) at the inlet of intercooler is 44.9ºC. In the optimization of intercooler, the decision variable of the component is selected as LTW temperature at the inlet of intercooler. This changes temperature of LTW (dependent variable) at the exit of intercooler. This affects the temperature of the LTW at the inlet of lubrication oil cooler unit, and thus the exergetic efficiency. The optimum temperature of LTW at the exit of lubrication oil cooler unit may be searched in the range of 61.5 ºC – 65.4ºC considering the corresponding exergetic efficiencies. Using the thermoeconomical cost balance equation from Equation 8.173, we can optimize the destruction cost rate of the lubrication oil cooler and the exergetic efficiency. Thus,

$$\textit{minimize}\, \dot{C}_{28} = \dot{C}_{27} + \left(\dot{C}_{24} - \dot{C}_{25}\right) + \dot{Z}_{\mathrm{LOC}} \tag{8.215a}$$

or

$$\textit{minimize}\, c_{28}\dot{E}_{28} = c_{27}\dot{E}_{27} + \left(c_{24}\dot{E}_{24} - c_{25}\dot{E}_{25}\right) + \dot{Z}_{\mathrm{LOC}} \tag{8.216b}$$

From Table 8.17, optimal values of exergetic efficiency and destruction cost rate are $\varepsilon_{\mathrm{LOC}}^{\mathrm{opt}} = 0.6952$ and $\dot{D}_{\mathrm{D,LOC}}^{\mathrm{opt}} = 1.31$ \$/h, respectively for lubrication oil cooler. The optimal values for this component are obtained to be outside of the actual base case range. This is because when we fixed optimal values obtained of the other subcomponents through optimization process, the working conditions of LOC unit does not satisfy the real range of the process and should be redesigned. The inlet and exit temperatures of lubrication oil are kept constant and possible increments of temperature between inlet and outlet cases of LTW through LOC unit are taken into account by checking the exergetic efficiency range for the component (i.e., 0.1-0.99). In Figures 8.16 and 8.17, variations of destruction cost rates and product cost of the lubrication oil cooler at the iterated values of LTW exit temperatures with respect to the corresponding calculated exergetic efficiencies of the LOC unit are given.

Table 8.17 Dependent variables obtained during iterations of the LT water exit temperature for the optimization procedure of exergetic efficiency and corresponding destruction cost rate of the lubrication oil cooler of DEPC system

Variable	⟶	1st Iteration	2nd Iteration	3rd Iteration	4th Iteration
P_2/P_1	**3.118**	3.118	3.118	3.118	**3.118**
T_{20} (°C)	**76.53**	76.53	76.53	76.53	**76.53**
T_{23} (°C)	**50.72**	50.72	50.72	50.72	**50.72**
T_7 (°C)	**303**	303	303	303	**303**
T_{12} (°C)	**81.6**	81.6	81.6	81.6	**81.6**
T_{25} (°C)	61.5	63.1	63.7	64	**64.1**
Fixed parameters					
$c_{f,LOC}$ ($/GJ)	5.0	5.0	5.0	5.0	**5.0**
$\dot{E}_{P,LOC}$ (kW)	163.0	163.0	163.0	163.0	**163.0**
ε_{LOC}	0.1506	0.4811	0.6089	0.6736	**0.6952**
$c_{p,LOC}$ ($/GJ)	3.43	5.32	6.98	7.43	**7.56**
Δr_{LOC} (%)	-31.4	6.4	39.6	48.6	**51.2**
$\dot{E}_{D,LOC}$ (kW)	202.2	123.5	93.10	77.7	**72.5**
$\dot{D}_{D,LOC}$ ($/h)	3.70	2.22	1.68	1.40	**1.31**

Table 8.17 continued

Variable	4th Iteration	3rd Iteration	2nd Iteration	1st Iteration	⟵
P_2/P_1	**3.118**	3.118	3.118	3.118	**3.118**
T_{20} (°C)	**76.53**	76.53	76.53	76.53	**76.53**
T_{23} (°C)	**50.72**	50.72	50.72	50.72	**50.72**
T_7 (°C)	**303**	303	303	303	**303**
T_{12} (°C)	**81.6**	81.6	81.6	81.6	**81.6**
T_{25} (°C)	**64.1**	64.2	64.3	64.6	65.4
Fixed parameters					
$c_{f,LOC}$ ($/GJ)	**5.0**	5.0	5.0	5.0	5.0
$\dot{E}_{P,LOC}$ (kW)	**163.0**	163.0	163.0	163.0	163.0
ε_{LOC}	**0.6952**	0.7170	0.7387	0.8044	0.9821
$c_{p,LOC}$ ($/GJ)	**7.56**	8.07	8.32	8.76	9.85
Δr_{LOC} (%)	**51.2**	61.4	66.4	75.2	97.0
$\dot{E}_{D,LOC}$ (kW)	**72.5**	67.35	62.2	46.6	4.26
$\dot{D}_{D,LOC}$ ($/h)	**1.31**	1.21	1.12	0.84	0.08

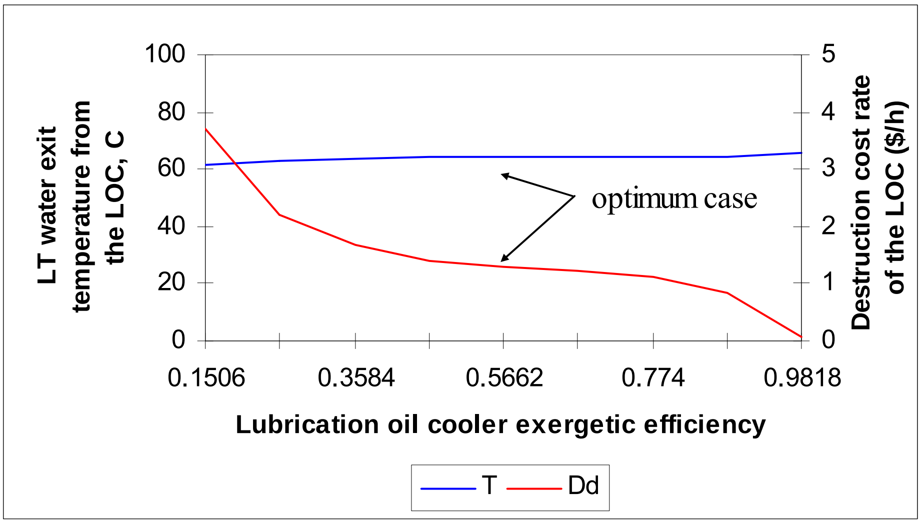

Figure 8.16 Variation of destruction cost rate of the lubrication oil cooler with respect to iterated LT water exit temperatures and corresponding calculated exergetic efficiencies of the LOC unit through optimization process

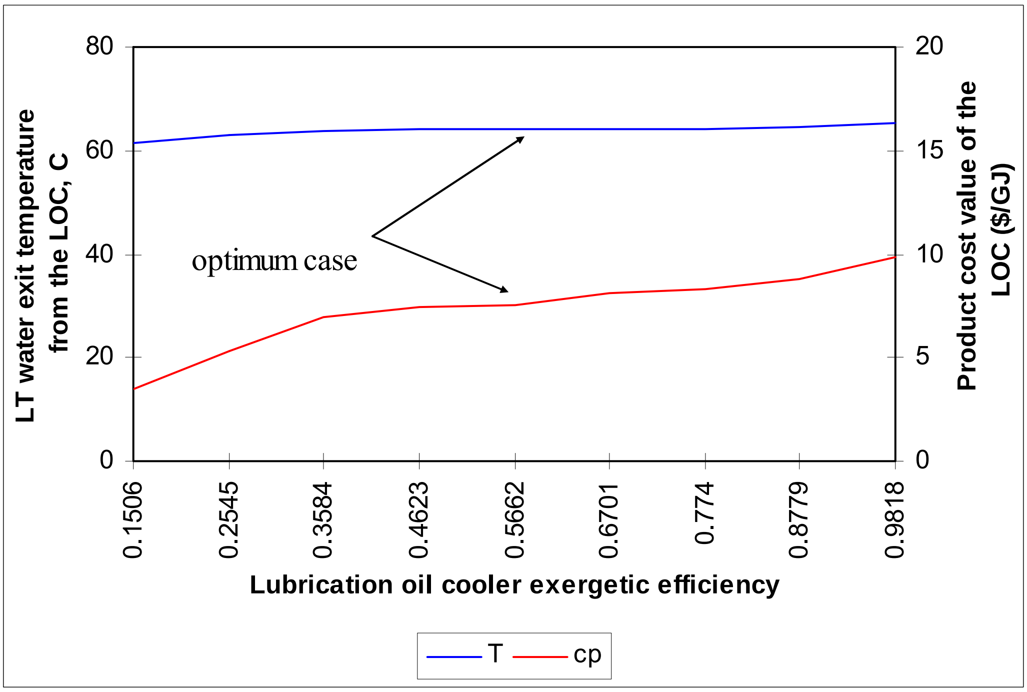

Figure 8.17 Variation of product cost values of lubrication oil cooler with respect to iterated LT water exit temperatures of the LOC unit and corresponding calculated exergetic efficiencies through optimization process

8.10.6 Air–Water Radiator

Air-water radiator in the DEPC system is a heat exchanger network. The cost rate balance of the air-water radiator is given in Equation 8.203. In order to optimize the exergetic efficiency and the destruction cost rate of the air-water radiator, we use thermoeconomical isolation approach and minimize the product cost of the component as

$$minimize\ \dot{C}_{32} = \left(\dot{C}_{25'} - \dot{C}_{25}\right) + \left(\dot{C}_{22'} - \dot{C}_{22}\right) + \dot{Z}_{\mathrm{AWR}} + \dot{C}_{31} \tag{8.216a}$$

or

$$minimize\ c_{32}\dot{E}_{32} = \left(c_{25'}\dot{E}_{25'} - c_{25}\dot{E}_{25}\right) + \left(c_{22'}\dot{E}_{22'} - c_{22}\dot{E}_{22}\right) + c_{31}\dot{E}_{31} + \dot{Z}_{\mathrm{AWR}} \tag{8.216b}$$

In this system, according to their different mass flow rates, high-temperature (HT) water and low-temperature (LT) water can be separated into two different heat exchanging processes in air-water radiator by using weighing factor. Thus, the decision variables in this component are taken as inlet temperatures of low-temperature water stream from intercooler unit and high-temperature water stream from diesel engine as T_{25} and T_{22}, respectively. First process takes place between environment air and HT water from diesel engine. The maximum temperature difference in the first process is 58.8ºC. Therefore, high-temperature water from the engine through air-water radiator cannot be cooled by more than 58.8ºC (to 30ºC). When we take HT water exit temperature as the decision variable, the optimum range can be determined in the range of 71.7 ºC and 87.0ºC. In Table 8.18, exergetic efficiencies, destruction cost rates and corresponding product costs calculated during the iterations in the first heat exchange process are given for the air-water radiator at the optimum pressure ratio of the compressing process.

As shown in Table 8.18, the optimum value of HT water exit temperature and corresponding optimum values of exergetic efficiency and destruction cost rate of the first heat exchanging process in the air-water radiator are $T_{22'}$=74.4ºC, $\varepsilon_{\mathrm{AWR\text{-}1}}^{\mathrm{opt}} = 0.6264$, and $\dot{D}_{\mathrm{D,AWR\text{-}1}}^{\mathrm{opt}} = 6.12\ \$/\mathrm{h}$.

Table 8.18 Dependent variables obtained during the iterations of the HT water exit temperature for the optimization procedure of exergetic efficiency and corresponding destruction cost rate of the air-water radiator-1.

Variable	⟶	1st Iteration	2nd Iteration	3rd Iteration
P_2/P_1 **$T_{22'}$ (°C)**	3.118 71.7(min)	3.118 72.34	3.118 73.65	**3.118** **74.42**
Fixed parameters $c_{f,AWR-1}$ ($/GJ) $\dot{E}_{P,AWR-1}$ (kW)	2.70 1005.3	2.70 1005.3	2.70 1005.3	**2.70** **1005.3**
$\varepsilon_{AWR\text{-}1}$	0.8611	0.7567	0.6823	**0.6264**
$c_{p,AWR-1}$ ($/GJ)	24.20	24.00	22.87	**20.23**
$\Delta r_{AWR\text{-}1}$ (%)	0.888	0.887	0.882	**0.866**
$\dot{E}_{D,AWR-1}$ (kW)	865.7	760.7	685.9	**629.7**
$\dot{D}_{D,AWR-1}$ ($/h)	8.41	7.39	6.67	**6.12**

Table 8.18 continued

Variable	3rd Iteration	2nd Iteration	1st Iteration	⟵
P_2/P_1 **$T_{22'}$ (°C)**	**3.118** **74.42**	3.118 76.54	3.118 81.76	3.118 87.0(max)
Fixed parameters $c_{f,AWR-1}$ ($/GJ) $\dot{E}_{P,AWR-1}$ (kW)	**2.70** **1005.3**	2.70 1005.3	2.70 1005.3	2.70 1005.3
$\varepsilon_{AWR\text{-}1}$	**0.6264**	0.5671	0.5088	0.4409
$c_{p,AWR-1}$ ($/GJ)	**20.23**	18.56	16.93	14.40
$\Delta r_{AWR\text{-}1}$ (%)	**0.866**	0.855	0.841	0.813
$\dot{E}_{D,AWR-1}$ (kW)	**629.7**	570.1	511.5	443.2
$\dot{D}_{D,AWR-1}$ ($/h)	**6.12**	5.54	4.97	4.31

The second heat exchange process in the air-water radiator is between the air from the first section and LT water from the intercooler. The maximum temperature difference in the second process is 103.6ºC. Therefore, LT water from the intercooler unit cannot be cooled by more than 18.4ºC (to 38.4ºC). When we take LT water exit temperature from the air-water radiator as decision variable, the optimum range becomes 38.4ºC and 48.7ºC. In Table 8.19, exergetic efficiencies, destruction cost rates and corresponding product costs calculated during the iterations in the second heat exchanging process are given for the air-water radiator at the optimum pressure ratio of the compressing process and the optimum exit temperature of the HT water from first heat exchange process in the air-water radiator. Thus, in the AWR network, optimum values of exergetic efficiency and destruction cost rate due to the second heat exchange process are determined to be $\varepsilon_{\text{AWR}-2}^{\text{opt}} = 0.4271$ and $\dot{D}_{\text{D,AWR-2}}^{\text{opt}} = 1.69\ \$/\text{h}$, respectively, through the iterations in Table 8.18.

Figures 8.18 and 8.19 show variations of destruction cost rates of the intercooler with respect to the HT and LT water streams temperatures, respectively, and corresponding calculated exergetic efficiencies of the air-water radiator through optimization process.

Table 8.19 Dependent variables obtained during the iterations of the LT water exit temperature for the optimization procedure of exergetic efficiency and corresponding destruction cost rate of the air-water radiator-2.

Variable	⟶	**1st Iteration**	**2nd Iteration**	**3rd Iteration**	**4th Iteration**	**5th Iteration**	**6th Iteration**
P_2/P_1	3.118	3.118	3.118	3.118	3.118	3.118	**3.118**
$T_{22'}$ (°C)	74.42	74.42	74.42	74.42	74.42	74.42	**74.42**
$T_{25'}$ (°C)	38.40	39.11	39.87	40.54	41.32	42.05	**43.48**
Fixed parameters							
$c_{\text{f,AWR}-2}$ (\$/GJ)	5.0	5.0	5.0	5.0	5.0	5.0	5.0
$\dot{E}_{\text{P,AWR}-2}$ (kW)	219.9	219.9	219.9	219.9	219.9	219.9	219.9
$\varepsilon_{\text{AWR-2}}$	0.4850	0.4778	0.4657	0.4530	0.4412	0.4366	**0.4271**
$c_{\text{p,AWR}-2}$ (\$/GJ)	20.23	19.87	18.67	17.56	16.33	15.70	**14.54**
$\Delta r_{\text{AWR-2}}$ (%)	0.753	0.748	0.732	0.715	0.694	0.682	**0.656**
$\dot{E}_{\text{D,AWR}-2}$ (kW)	106.7	105.1	102.4	99.6	97.0	96.0	**93.9**
$\dot{D}_{\text{D,AWR}-2}$ (\$/h)	1.92	1.89	1.84	1.79	1.75	1.73	**1.69**

Table 8.19 continued

Variable	6^{th} Iteration	5^{th} Iteration	4^{th} Iteration	3^{rd} Iteration	2^{nd} Iteration	1^{st} Iteration	←
P_2/P_1 **$T_{22'}$ (°C)** **$T_{25'}$ (°C)**	**3.118** **74.42** **43.48**	3.118 74.42 44.66	3.118 74.42 45.49	3.118 74.42 46.37	3.118 74.42 47.10	3.118 74.42 48.05	3.118 74.42 48.7(max)
Fixed parameters $c_{f,AWR-2}$ (\$/GJ) $\dot{E}_{P,AWR-2}$ (kW)	5.0 219.9	5.0 219.9	5.0 219.9	5.0 219.9	5.0 219.9	5.0 219.9	5.0 219.9
ε_{AWR-2}	**0.4271**	0.4172	0.4068	0.3961	0.3863	0.3776	0.3680
$c_{p,AWR-2}$ (\$/GJ)	**14.54**	13.43	12.79	11.69	10.56	9.54	8.88
Δr_{AWR-2} (%)	**0.656**	0.628	0.609	0.572	0.527	0.476	0.437
$\dot{E}_{D,AWR-2}$ (kW)	**93.9**	91.7	89.5	87.1	84.9	83.0	80.9
$\dot{D}_{D,AWR-2}$ (\$/h)	**1.69**	1.65	1.61	1.57	1.53	1.49	1.46

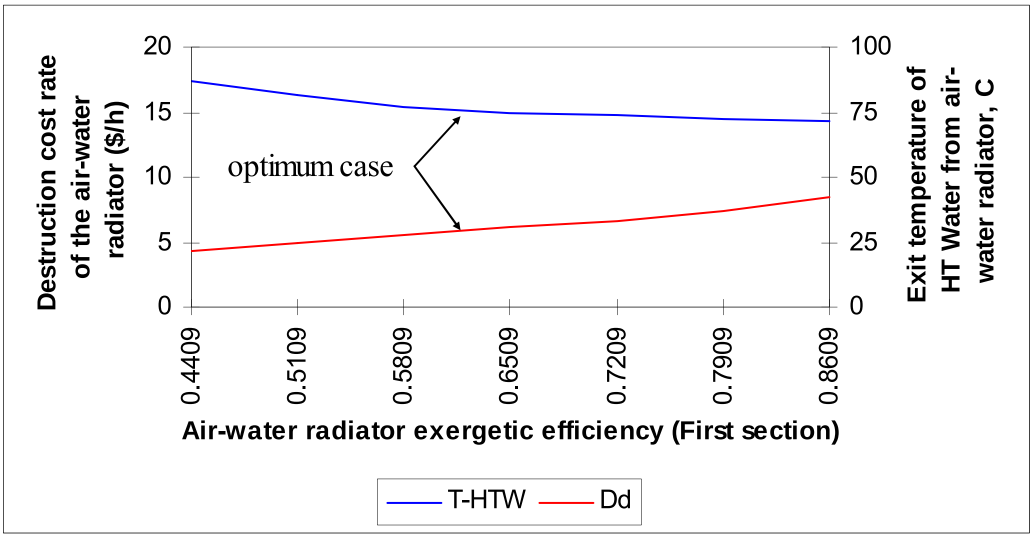

Figure 8.18 Variation of destruction cost rate of the air-water radiator (first section) with respect to the iterated HT water exit temperatures and corresponding calculated exergetic efficiencies of the air-water radiator through optimization process

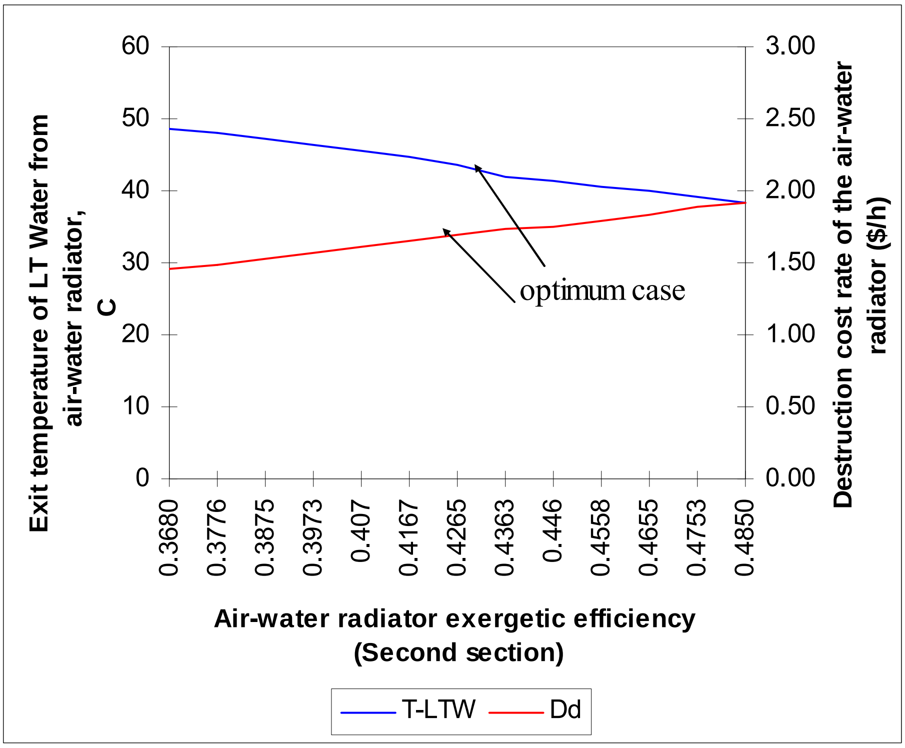

Figure 8.19 Variation of destruction cost rate of the air-water radiator (second section) with respect to the iterated LT water exit temperatures and corresponding calculated exergetic efficiencies of the air-water radiator through optimization process

In the optimization of DEPC system considered, two types of optimization are considered: thermodynamic and thermoeconomic. The objective of the thermodynamic optimization is to maximize the exergetic efficiency and thus to minimize the exergy destruction of the cogeneration system. In the thermoeconomic optimization, the objective is to minimize the destruction cost rate of the components, by doing this we can optimize the product cost values of components in the system. The thermodynamic optimization is based on the model explained in Chapter 4, and developed formulations for the subcomponents of the DEPC plant in section 8.5 of this chapter, whereas the thermoeconomic optimization employs both thermodynamic model and the exergy based economic model presented in Chapters 5 and 6.

Table 8.20 gives the values of the decision variables and selected parameters for the thermodynamically optimal case (TO) and the thermoeconomically cost optimal case (CO). For comparison, values for the actual base working conditions are also presented. As expected, the thermodynamic optimum is obtained at the nearly maximum values of the decision variables, compressor pressure ratio P_2/P_1, isentropic efficiencies of compressor and turbine of the turbocharger, exit temperatures of LT water from intercooler and lubrication oil cooler at states 23 and 25 respectively, and the minimum values of temperature of the exhaust gas at the exit of turbine, and the temperature of feed water at the inlet of waste heat boiler unit.

Table 8.20 Values of the decision variables and selected parameters for the actual base working condition, thermodynamically optimal thermoeconomically cost optimal cases[a]

Parameter	Base Case	Thermodynamically Optimal Case	Cost Optimal Case
P_2/P_1	2.90	3.50	3.12
T_{20} (°C)	71.7	72.0	76.5
$T_{22'}$ (°C)	71.7	76.5	74.4
T_{23} (ºC)	38.4	61.5	50.7
T_7 (ºC)	302	277	303
T_{12} (ºC)	95.0	60.0	81.6
T_{25} (ºC)	50.3	65.4	64.1
$T_{25'}$ (°C)	38.4	44.6	43.5
η_c	0.80	0.88	0.82
η_t	0.85	0.90	0.85

[a] The optimal values of the decision variables given for thermodynamically and thermoeconomically optimal cases are not unique. The same values of the maximum exergetic efficiency and the minimum overall cost rate may be obtained through other combinations of the values of the decision variables. Besides, many different sets of the decision variables values may lead to nearly optimal values of the objective function.

In Table 8.21, the values of three important exergy-related variables introduced in Chapter 5 are listed for the overall DEPC system and each of the system components. These are the rate of exergy destruction $\dot{E}_{\mathrm{D,k}}$, exergy destruction ratio $\frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{F,total}}}$, and the exergetic efficiency ε_{k}. In Table 8.21, we see that the overall exergetic efficiency of the TO case of the DEPC plant when diesel engine is not taken into account is 93.94%. When diesel engine is taken into consideration, the corresponding value is 48.0%. The corresponding values for the base case and CO case are 88.15% and 91.11%, respectively when diesel engine is not taken into account. Table 8.21 also shows that the component values for the exergy destruction rate and the exergy destruction ratio, which are generally smaller in the TO case than in other two cases.

Table 8.21 Exergy destruction $\dot{E}_{\mathrm{D,k}}$, exergy destruction ratio $\frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{F,total}}}$, and exergetic efficiency ε_{k} for the *k*th component of the Sanko DEPC plant for the actual base working condition, thermodynamically optimal and thermoeconomically cost optimal cases

Component	**Base Case**			**Thermodynamically Optimal Case (TO)**			**Thermoeconomically Cost Optimal Case (CO)**		
	$\dot{E}_{\mathrm{D,k}}$ (kW)	$\frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{F,total}}}$ (%)	ε_{k} (%)	$\dot{E}_{\mathrm{D,k}}$ (kW)	$\frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{F,total}}}$ (%)	ε_{k} (%)	$\dot{E}_{\mathrm{D,k}}$ (kW)	$\frac{\dot{E}_{\mathrm{D,k}}}{\dot{E}_{\mathrm{F,total}}}$ (%)	ε_{k} (%)
Compressor	1380	2.20	82.60	474.0	0.756	93.90	1032	1.64	86.90
Intercooler	1693	2.70	26.30	243.2	0.400	78.03	400	0.64	63.88
Lubrication oil cooler	187.0	0.30	63.0	4.26	0.0007	98.21	72.5	0.12	69.52
Air-water radiator	775.0	1.23	30.0	456.3	0.73	58.78	383.4	0.61	65.36
Turbine	1092	1.74	88.1	470.4	0.75	96.16	1500	2.40	87.75
Waste heat boiler	1404	2.24	11.4	1325.1	2.11	14.0	1348	2.15	12.49
Fuel oil day tank	3.60	0.006	79.1	3.60	0.006	79.1	3.60	0.006	79.1
Fuel forwarding module	3.20	0.002	87.4	3.20	0.002	87.4	3.20	0.002	87.4
Condenser	1.20	0.001	16.6	1.20	0.001	16.6	1.20	0.001	16.6
Pumps	79.47	0.0013	62.1	6.40	0.0001	95.0	15.37	0.0002	88.0
$DeSO_x$	817.0	0.010	-	817.0	0.010	-	817.0	0.010	-
Total Plant Comp.	**7435.5**	**10.43**	**88.15**	**3804.7**	**4.77**	**93.94**	**5576.3**	**7.58**	**91.11**
Diesel Engine	28,830	45.94	40.4	28,830	45.94	40.4	28,830	45.94	40.4
Total DEPC	**36,266**	**56.37**	**42.21**	**32,635**	**50.71**	**48.00**	**34,406.3**	**53.52**	**45.18**

The values of these exergy destruction variables for the overall DEPC system are lower in TO case than in other two cases. The constraints on the values of the decision variables presented in the optimization study above and in Table 8.21 limit the maximum value of $\varepsilon_{\text{total}}$ that can be obtained in practice. For example, when the thermodynamically optimal values are used for the remaining variables, the maximum value of $\varepsilon_{\text{total}}$ would be obtained for P_2/P_1 to be greater than 3.50, which according to the thermodynamic model of Section 8.4 exceeds the maximum allowed value. Therefore, for this actual DEPC system, the thermodynamic optimum is obtained at the boundary points not only with respect to isentropic efficiencies of compressor and turbine or LT water temperatures but also with respect to the pressure ratio of the compressor, exhaust gas generated by diesel engine, and power produced by the engine.

Table 8.22 provides the costs obtained in the thermoeconomic optimization. The costs calculated for the thermodynamically optimum case and previously calculated base case costs are presented in this table.

Table 8.22 Calculated costs for the actual base case, thermodynamically optimal case and cost optimal case

Parameter	**Base Case**	**TO Case**	**CO Case**
Total cost flow rate ($/h)	2932	7359	2316
Cost of electricity (¢/kWh)	8.90	21.36	6.70
Cost of steam (¢/kg)	5.22	10.04	4.50

Comparing the cost data of the cost optimal and thermodynamically optimal cases shows striking differences: the total cost flow rate and the cost of electricity are significantly higher in the TO case than in the CO case. These differences result mainly from the investment cost of the diesel engine, waste heat boiler unit and $DeSO_x$ unit that are very high in the TO design. Figure 8.20 shows the variations of exergetic efficiencies of subcomponents in the DEPC system with respect to the three identical cases presented in Table 8.21.

In the optimization studies of cogeneration systems, the total cost associated with the thermodynamically optimal cases, as in the present study, are sometimes significantly higher. Accordingly, studies focusing only on the thermodynamically optimal performance for the improvement of an existing system can lead to gross misevaluations and skewed decision making.

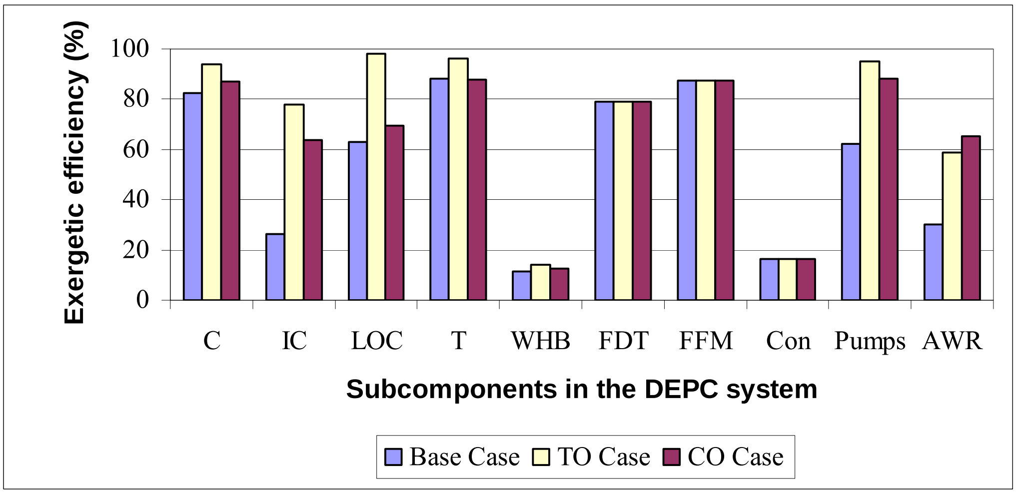

Figure 8.20 Variations of exergetic efficiencies of subcomponents in the DEPC system with respect to the base case, thermodynamically optimal and cost optimal cases

8.11 Accuracy of Measurements in DEPC System

Measurements are done for each operating condition in Sanko DEPC. Measurements are carried out for the purpose of verifying the declaration of power, engine speed and fuel consumption. Valid measurements are carried out at least two times. A measurement is considered to be valid if the variations of the engine brake torque and engine speed values in relation to the settings of the operating values do not exceed $\pm 2\ \%$. The variation of the power output during this period did not exceed $\pm 3\ \%$. The control system of the plant is calibrated due to the ISO-3046 standards. According to these standards there are some corrections factors which have to be used for calibrations and accuracy. The main aim is to keep the tolerance limit for the measurements at overall the plant within the $\pm 3\ \%$.

Table 8.23 Uncertainty of the measured quantities in DEPC plant

Measured Quantities in DEPC plant	Unit	Uncertainty (%)
Temperature	°C	± 2
Pressure	bar	± 2
Mass flowrate	kg/s	± 1
Power	kW	± 3

8.12 Conclusions

Energy, exergy, and thermoeconomic analysis and thermoeconomic optimization of Sanko DEPC system are performed. The iterative methodology of exergy based economic optimization is used. In the iterative optimization procedure we use the variables relative cost difference Δr and exergetic efficiency ε with the corresponding optimal values obtained through the optimization procedure. The effects of changes in the decision variables selected on relative cost difference, exergetic efficiency and destruction cost rate can provide suggestions for the design changes that need to be considered in the next optimization step.

CHAPTER 9

ENVIRONMENTAL IMPACTS AND EMISSION ASSESSMENT OF SANKO DEPC PLANT

9.1 Introduction

Air pollution is becoming a great concern in Turkey. Air pollution from energy utilization in the country is due to the combustion of coal, lignite, petroleum, natural gas, wood, and agricultural and animal wastes. On the other hand, owing mainly to the rapid growth of primary energy consumption and the increasing use of domestic lignite, SO_2 emissions, in particular, have increased rapidly in recent years in Turkey. The major source of SO_2 emissions is the power sector, contributing more than 50% of the total emissions [146].

Turkey, being a member of the OECD (Organization for Economic Cooperation and Development), was initially listed in Annexes-I and II of the UNFCCC (United Nations Framework Convention on Climate Change) in 1992 [142]. Under the convention Annex I countries have to take steps to reduce emissions and Annex II countries have to take steps to provide financial and technical assistance to developing countries. Following number of negotiations in 2001, Turkey was finally removed from the list of Annex II countries but remained in the list of Annex I countries with an accompanying footnote specifying that Turkey should enjoy favorable conditions considering differentiated responsibilities. This led to an official acceptance of the UNFCCC by the Turkish Grand National Assembly in October 2003, followed by its enactment in May 2004 [142,194].

One of the major important aspects of heavy fuel oil fired diesel cogeneration systems is environmental. Combustion of lower quality fuels like heavy fuel oil that is rich in sulfur and asphaltene in compression ignition engines causes an increased emission of harmful pollutants: CO_2, SO_2, NO_x, hydrocarbons, particulate matters (PM) and volatile organic compounds (VOC). It is well known that carbon dioxide (CO_2) contributes to the greenhouse effect considerably which causes global warming [195]. More than 95% of the sulfur content in the fuel transforms into sulfur dioxide (SO_2) which is very harmful to humans, animals and plants, either directly or indirectly, and when combines with water vapor in the atmosphere, it causes the so-called acid rain. The combustion process of heavy fuel oil in DEPC also produces nitrogen oxides (NO_x) where more than 95% are NO, the remaining part being NO_2. In the atmosphere, the combination of NO with oxygen under the influence of ultraviolet rays, transforms into NO_2, and this, either in NO_2 form or in N_2O form is very harmful to living organisms directly. Also N_2O has a greater influence, about 200-300 times more, than CO_2 in the greenhouse effect [196].

When cogeneration systems are considered as a major interest for future electricity supply, the allocation of pollutant emissions and costs between the two products is an important question for the researchers, organizations, and governments. There have been several studies in literature to determine how to locate emissions among the products of the cogeneration. Phylipsen and Blok [197] provided some methods to allocate energy and CO_2 emissions in cogeneration systems to the electrical and thermal products. Strickland and Nyboer [198,199] adapted six calculations of fuel allocation to the thermal and electrical products of a cogeneration system from Phylipsen and Blok [197]. They stated that distributing emissions to the electricity and thermal energy could be calculated depending on different criteria such as product amounts, products exergy, and economical values of products.

Kaarsberg et al. [200,201] suggested an integrated analysis for combined heat and power production to save energy and reduce emissions. In the analysis, they used the difference between the heat rate of conventional fossil fuel fired systems and the net heat rate of cogeneration systems. Stenhede [202] described some methods for how to use and convert energy for utility and industry by internal combustion engine powered cogeneration and how to keep emissions low. Cardu and Baica [203] generated a methodology to analyze a thermopower plant ecologically and considered the harmful effects of all toxic flue gases (i.e. CO_2, SO_2, and NO_x) as a single entity instead of CO_2 only. Also, they defined the relation of the quantities of harmful gas emissions per unit of useful energy produced.

Villela and Silveira [204] analyzed the environmental impact resulting from the natural gas and diesel cogeneration plants following previous work of Cardu and Baica. For this, they considered major flue gases and particulate matter emission separately. Voorspools and D'haseleer [205] simulated and compared two different situations in order to determine energy savings and greenhouse gas emissions in the buildings in Belgium by using the difference method approach. Erdem et al. [206] studied cogeneration and separate heat and power systems, and investigated limiting conditions providing emission reduction by using the same method as Sevilgen et al. [207] who compared the environmental effects of cogeneration and conventional systems. Rosen [208-210] described and compared some selected methods in literature for allocating emissions for cogeneration systems and presented exergy values for typical commodities encountered in cogeneration. Rosen [210] classified the methods used in the allocation of emissions of cogeneration as follows: efficiency methods, work potential methods and heat content methods. He emphasized that more research is needed in this area.

In this chapter, emission characteristics of the DEPC plant and exhaust gas treatment units in the plant are described. Allocation of emission methodologies presented by Phylipsen et al. [6] and Strickland and Nyboer [7,8] are developed and applied to the emissions of diesel engine powered cogeneration plant (DEPC). In literature, these methodologies were generally applied based on energy evaluations. In this study, allocation of emissions of cogeneration plants based on both energy and exergy are performed.

9.2 Emission Characteristics of Heavy Fuel Oil Fired Diesel Engine Powered Cogeneration

Heavy fuel oil has a high share in fossil fuel consumption especially for diesel engine powered cogeneration applications. Two major categories of heavy fuel oil are distillate oils and residual oils. Distillate oils are more volatile and less viscous than residual oils. They have negligible nitrogen and ash contents and usually contain less than 0.3% sulfur (by weight) [178]. Distillate oils are used mainly in domestic and small commercial applications, and include kerosene and diesel fuels. Being more viscous and less volatile than distillate oils, the heavier residual oils may need to be heated for ease of use and to facilitate proper atomization for combustion. Residual oils are used mainly in industrial applications, especially in power production facilities [179,182]. In the utilization of residual heavy fuel oils for industrial power production, two major problems arise: hazardous emissions and depletion of fuel reserves in the world. Cogeneration which generates heat and power simultaneously from same fuel supply may be one of the most appropriate methods to address these concerns [201,211].

Carbon dioxide (CO_2), methane (CH_4), and nitrous oxide (N_2O) emissions are all produced during fuel oil combustion. Nearly all of the fuel carbon in fuel oil is converted to CO_2 during combustion process. This conversion is relatively independent of firing configuration. Although the formation of CO acts to reduce CO_2 emissions, the amount of CO produced is insignificant compared to amount of CO_2 produced. Formation of N_2O during the combustion process is governed by a complex series of reactions and its formation is dependent upon many factors [141,142]. Formation of N_2O is minimized when combustion temperatures are kept high (above 800ºC) and excess air is kept minimum (less than 1%). Methane emissions vary with the type of fuel and firing configuration, but are the highest during period of incomplete combustion or low-temperature combustion, such as the start-up or shut-down cycle for oil-fired combustion units. Typically, conditions that favor the formation of N_2O also favor emissions of CH_4 [131].

The main air pollutants related to the power production and use of energy is sulfur oxides (SO_x) – in particular sulfur dioxide (SO_2), - nitrogen oxides (NO_x) and suspended particulates. Sulfur oxides (SO_x) emissions are generated during oil combustion from the oxidation of sulfur contained in the fuel. The emissions of SO_x from conventional combustion systems are predominantly in the form of SO_2. On average more than 95% of the fuel sulfur is converted to SO_2, about 1 to 5% is further oxidized to sulfur trioxide (SO_3), and 1 to 3% is emitted as sulfur particulate [212]. SO_3 readily reacts with water vapor (both in the atmosphere and in flue gases) to form a sulfuric acid mist.

Oxides of nitrogen (NO_x) formed in combustion processes are due either to thermal fixation of atmospheric nitrogen in the combustion air (i.e. thermal NO_x), or the conversion of chemically bound nitrogen in the fuel (i.e. fuel NO_x). The term NO_x refers to the composite of nitric oxide (NO) and nitrogen dioxide (NO_2). For most fossil fuel combustion systems, over 95% of the emitted NO_x is the form of nitric oxide. Nitrous oxide (N_2O) is not included in NO_x but has recently received increased interest because of atmospheric effects [181].

Particulate emissions maybe categorized as either filterable or condensable. Filterable particulate matter emissions depend predominantly on the grade of fuel fired. Combustion of lighter distillate oils results in significantly lower PM formation than does combustion of heavier residual oils. In general, filterable PM emissions depend on the completeness of combustion as well as on the oil ash content [213]. The PM emitted by distillate oil-fired combustion processes primarily comprises carbonaceous particles resulting from incomplete combustion of oil and is not correlated to the ash or sulfur content of the oil. However, PM emissions from residual oil burning are related to the oil sulfur content. This is because low-sulfur residual oil (heavy fuel oil no.6), either from naturally low-sulfur crude oil or desulphurized by several processes, exhibits substantially lower viscosity and reduced asphaltene, ash, and sulfur contents, which results better atomization and more complete combustion [178,179,182].

In Chapter 7, the flue gas treatment units in Sanko DEPC are explained in detail. In Table 9.1 and 9.2, emission content of exhaust gas after the flue gas treatment units ($DeNO_x$ and $DeSO_x$) and total key indicative emission amounts for the DEPC plant are given.

Table 9.1 Emission amounts of exhaust gas after the flue gas treatment

Type of fuel	Fuel oil no 6
Power produced (MW)	25.32 MW
Height of stack (from ground) (m)	21
Height of stack (from roof) (m)	6
Reference O_2 (%)	15
Exit exhaust temperature (ºC)	53.7
Velocity of exhaust gas at the exit (m/s)	8.1±0.1
Cross sectional area of stack (m^2)	6.154
Volume flowrate of exhaust gas at the exit (m^3/h)	123,464
Particulate matter concentration (mg/m^3)	134.92
Particulate matter concentration (mg/m^3, at 15% O_2)	98.99±25.55
Particulate matter emission (kg/h)	16.7256±3.6669
CO concentration (mg/m^3)	101.67
CO concentration (mg/m^3, at 15% O_2)	74.70±8.40
CO emission (kg/h)	12.5507±1.4898
SO_2 concentration (mg/m^3)	40.95
SO_2 concentration (mg/m^3, at 15% O_2)	30.09±3.40
SO_2 emission (kg/h)	5.0673±0.6036
NO concentration (mg/m^3)	42.41
NO emission (kg/h)	5.2356
NO_2 concentration (mg/m^3)	83.0704
NO_2 emission (kg/h)	10.2556±3.5846
CO_2 concentration (%)	6.3

Table 9.2 Total key indicative emission amounts

Type of Emission	**Total Emission Amounts** (kg/h)	**Limit Values**[142,213] (kg/h)
CO	12.5507±1.4898	1000
SO_2	5.0673±0.6036	60
NO_x	15.5000±3.5846	40
Particulate Matter	16.7256±3.6669	15

9.3 Allocation of Emissions of DEPC

Following Phylipsen et al. [197], Strickland and Nyboer [198,199], and Rosen [208-210], major portion of works in literature [197-199,205-210] adapted main six methodologies in order to allocate energy, exergy, and CO_2 emissions in cogeneration systems to the electrical and thermal products. In this study, the methodology introduced mainly by Phylipsen et al. [197] is applied for *CO*, NO_x,

SO_x and Particulate Matter (PM) emissions for the environmental impact characteristics of diesel engine powered cogeneration. In the following, main equations for the allocation of emissions of cogeneration systems are presented.

9.3.1 Allocation of Emissions Based on Energy

This method is given as one of the simple methods of allocation of emissions. It accounts only for the quantity of the energy produced, not the quality of it. Thus, the amounts of emissions based on energy content allocated to electrical and heat productions can be given, respectively as

$$E_{e,i} = \left(\frac{\dot{W}_{net}}{\dot{W}_{net} + \dot{Q}_{net}} \right) \varphi_i \tag{9.4}$$

$$E_{h,i} = \left(\frac{\dot{Q}_{net}}{\dot{W}_{net} + \dot{Q}_{net}} \right) \varphi_i \tag{9.5}$$

where $\dot{W}_{net}$ and $\dot{Q}_{net}$ are the net power and heat productions of cogeneration system, respectively and φ is the amount of specific emission released per unit electricity production in cogeneration system and "i" represents emission type (i.e. SO_2, NO_x, CO, CO_2, and PM)

9.3.2 Allocation of Emissions Based on Exergy

Allocation based on exergy content accounts for the quality of the energy [199]. The amounts of emissions based on exergy content allocated to electrical and heat productions are given, respectively as

$$X_{e,i} = \left(\frac{\dot{W}_{net}}{\dot{W}_{net} + \beta \dot{E}_{Q_{net}}} \right) \varphi_i \tag{9.6}$$

$$X_{h,i} = \left(\frac{\beta \dot{E}_{Q_{net}}}{\dot{W}_{net} + \beta \dot{E}_{Q_{net}}} \right) \varphi_i \tag{9.7}$$

where β is the ratio of exergy to energy content of heat produced. It is noted that electrical (power) energy and electrical exergy are equivalent. For the heat produced, corresponding thermal exergy can be written as

$$\dot{E}_{\mathrm{Q_{net}}} = \dot{Q}_{\mathrm{net}}\tau \tag{9.8}$$

where τ represents the exergetic temperature factor [210] and it is given as

$$\tau = \left(1 - \frac{T_0}{T}\right) \tag{9.9}$$

In the above equation, T represents the temperature at which heat, $\dot{Q}$ crosses the system boundary and T_0 is the temperature of the reference environment. The choice of reference environment for determining exergy quantities is very important and can affect the results. In practice, selecting the reference environment similar to the actual environment is common application; however other reference environments can also be used related with the process.

9.3.3 Allocation of Emissions Based on Economic Value

This method is originally given in terms of conventional energy based economic analysis. In this chapter, it is given in terms of exergy based economic analysis approach [199,210]. Thus, the amounts of emissions based on exergoeconomic values of power and heat produced by the cogeneration plant can be defined respectively as

$$C_{\mathrm{e,i}} = \left(\frac{c_{\mathrm{e}}\dot{W}_{\mathrm{net}}}{c_{\mathrm{e}}\dot{W}_{\mathrm{net}} + c_{\mathrm{h}}\beta\dot{E}_{\mathrm{Q_{net}}}}\right)\varphi_{\mathrm{i}} \tag{9.10}$$

$$C_{\mathrm{h,i}} = \left(\frac{c_{\mathrm{h}}\beta\dot{E}_{\mathrm{Q_{net}}}}{c_{\mathrm{e}}\dot{W}_{\mathrm{net}} + c_{\mathrm{h}}\beta\dot{E}_{\mathrm{Q_{net}}}}\right)\varphi_{\mathrm{i}} \tag{9.11}$$

where c_{e} and c_{h} are the exergetic cost values of power and heat produced respectively.

9.3.4 Allocation of Emissions Based on The Incremental Fuel Consumption to Power Production

In this method, emissions are allocated in proportion to the fuel division among the power and heat produced by the cogeneration while considering power generation to be a byproduct of the thermal energy production process. Then, the amount of emissions based on incremental fuel consumption to power production in terms of power and heat produced by cogeneration system can be defined, respectively as

$$F_{\text{ee,i}} = \left[1 - \frac{\dot{Q}_{\text{net}}}{\left(\dot{W}_{\text{net}} + \dot{Q}_{\text{net}}\right)\eta_{\text{boiler}}}\right]\left(\frac{\dot{m}_{\text{fuel}_{\text{pp}}}}{\dot{m}_{\text{fuel}_{\text{pp}}} + \dot{m}_{\text{boiler}}}\right)\varphi_{\text{i}} \tag{9.12}$$

$$F_{\text{eh,i}} = \left[\frac{\dot{Q}_{\text{net}}}{\left(\dot{W}_{\text{net}} + \dot{Q}_{\text{net}}\right)\eta_{\text{boiler}}}\right]\left(\frac{\dot{m}_{\text{fuel}_{\text{pp}}}}{\dot{m}_{\text{fuel}_{\text{pp}}} + \dot{m}_{\text{boiler}}}\right)\varphi_{\text{i}} \tag{9.13}$$

where η_{boiler} is the thermal efficiency of hypothetical boiler that would have been used in the production of heat energy as produced by cogeneration system. Instead of energy terms, we can use corresponding exergetic ones as an alternative evaluation method as

$$X_{F\,\text{ee,i}} = \left[1 - \frac{\beta\dot{E}_{\text{Q}_{\text{net}}}}{\left(\dot{W}_{\text{net}} + \beta\dot{E}_{\text{Q}_{\text{net}}}\right)\varepsilon_{\text{boiler}}}\right]\left(\frac{\dot{m}_{\text{fuel}_{\text{pp}}}}{\dot{m}_{\text{fuel}_{\text{pp}}} + \dot{m}_{\text{boiler}}}\right)\varphi_{\text{i}} \tag{9.14}$$

$$X_{F\,\text{eh,i}} = \left[\frac{\beta\dot{E}_{\text{Q}_{\text{net}}}}{\left(\dot{W}_{\text{net}} + \beta\dot{E}_{\text{Q}_{\text{net}}}\right)\varepsilon_{\text{boiler}}}\right]\left(\frac{\dot{m}_{\text{fuel}_{\text{pp}}}}{\dot{m}_{\text{fuel}_{\text{pp}}} + \dot{m}_{\text{boiler}}}\right)\varphi_{\text{i}} \tag{9.15}$$

where $\varepsilon_{\text{boiler}}$ is the exergetic efficiency of hypothetical boiler.

9.3.5 Allocation of Emissions Based on The Incremental Fuel Consumption to Heat Production

This method is similar to the previous one, except that heat production by cogeneration system is considered as byproduct. Then, the amount of emissions based on incremental fuel consumption to heat production in terms of power and heat produced by the cogeneration can be obtained, respectively as

$$F_{\mathrm{he,i}} = \left[\frac{\dot{W}_{\mathrm{net}}}{\left(\dot{W}_{\mathrm{net}} + \dot{Q}_{\mathrm{net}}\right)\eta_{\mathrm{pp}}}\right]\left(\frac{\dot{m}_{\mathrm{boiler}}}{\dot{m}_{\mathrm{fuel_{pp}}} + \dot{m}_{\mathrm{boiler}}}\right)\varphi_{\mathrm{i}} \tag{9.16}$$

$$F_{\mathrm{eh,i}} = \left[1 - \frac{\dot{W}_{\mathrm{net}}}{\left(\dot{W}_{\mathrm{net}} + \dot{Q}_{\mathrm{net}}\right)\eta_{\mathrm{pp}}}\right]\left(\frac{\dot{m}_{\mathrm{boiler}}}{\dot{m}_{\mathrm{fuel_{pp}}} + \dot{m}_{\mathrm{boiler}}}\right)\varphi_{\mathrm{i}} \tag{9.17}$$

where η_{pp} is the energy efficiency of the power plant that would have been used in the production of same amount of power as produced by cogeneration system. With the same way as previous method, corresponding exergetic terms can be replaced by energy terms in the above equations as

$$X_{F\,\mathrm{he,i}} = \left[\frac{\dot{W}_{\mathrm{net}}}{\left(\dot{W}_{\mathrm{net}} + \dot{E}_{\mathrm{Q_{net}}}\right)\varepsilon_{\mathrm{pp}}}\right]\left(\frac{\dot{m}_{\mathrm{boiler}}}{\dot{m}_{\mathrm{fuel_{pp}}} + \dot{m}_{\mathrm{boiler}}}\right)\varphi_{\mathrm{i}} \tag{9.18}$$

$$X_{F\,\mathrm{hh,i}} = \left[1 - \frac{\dot{W}_{\mathrm{net}}}{\left(\dot{W}_{\mathrm{net}} + \dot{E}_{\mathrm{Q_{net}}}\right)\varepsilon_{\mathrm{pp}}}\right]\left(\frac{\dot{m}_{\mathrm{boiler}}}{\dot{m}_{\mathrm{fuel_{pp}}} + \dot{m}_{\mathrm{boiler}}}\right)\varphi_{\mathrm{i}} \tag{9.19}$$

where ε_{pp} is the exergetic efficiency of hypothetical power plant.

9.3.6 Allocation Based on A Shared Emission Savings Between Power and Heat

This allocation method shares the emissions among the products of cogeneration facility in a particular way that can be considered as a middle way for the concepts used in previous two emission allocation methods. The amount of emissions based on shared emission savings between power and heat produced by the cogeneration in terms of energy can be obtained, respectively as

$$F_{\mathrm{e,i}} = \left(\frac{\dfrac{\dot{W}_{\mathrm{net}}}{\eta_{\mathrm{pp}}}}{\dfrac{\dot{W}_{\mathrm{net}}}{\eta_{\mathrm{pp}}} + \dfrac{\dot{Q}_{\mathrm{net}}}{\eta_{\mathrm{boiler}}}} \right) \varphi_{\mathrm{i}} \tag{9.20}$$

$$F_{\mathrm{h,i}} = \left(\frac{\dfrac{\dot{Q}_{\mathrm{net}}}{\eta_{\mathrm{boiler}}}}{\dfrac{\dot{W}_{\mathrm{net}}}{\eta_{\mathrm{pp}}} + \dfrac{\dot{Q}_{\mathrm{net}}}{\eta_{\mathrm{boiler}}}} \right) \varphi_{\mathrm{i}} \tag{9.21}$$

Corresponding exergetic relations can be written as

$$X_{\mathrm{Fe,i}} = \left(\frac{\dfrac{\dot{W}_{\mathrm{net}}}{\varepsilon_{\mathrm{pp}}}}{\dfrac{\dot{W}_{\mathrm{net}}}{\varepsilon_{\mathrm{pp}}} + \dfrac{\beta \dot{E}_{\mathrm{Q\,net}}}{\varepsilon_{\mathrm{boiler}}}} \right) \varphi_{\mathrm{i}} \tag{9.22}$$

$$X_{\mathrm{Fh,i}} = \left(\frac{\dfrac{\beta \dot{E}_{\mathrm{Q\,net}}}{\varepsilon_{\mathrm{boiler}}}}{\dfrac{\dot{W}_{\mathrm{net}}}{\varepsilon_{\mathrm{pp}}} + \dfrac{\beta \dot{E}_{\mathrm{Q\,net}}}{\varepsilon_{\mathrm{boiler}}}} \right) \varphi_{\mathrm{i}} \tag{9.23}$$

9.4 Results and Discussion

Cogeneration systems have different characteristics in terms of electricity production capacity, fuel type, heat-power ratio, overall efficiency, heat quality, investment costs, etc. These differences mostly result from different characteristics of cycles and fuel used. Emissions released to the environment for combined heat and power systems are varied according to the electrical efficiency, fuel type and exhaust gas treatment technologies. In Table 9.3, specific amounts of emissions of different cogeneration systems are given. Also, specific emission amounts for the DEPC plant are given in this table using the plant data given in Table 9.2. According to the fuel types used in the boiler, amount of emissions is given in Table 9.4 [214].

Table 9.3 Specific amounts of emissions (φ) of different cogeneration systems [192,207]

Type of system	SO_2 (g/kWh)	NO_x (g/kWh)	CO (g/kWh)	CO_2 (g/kWh)	PM (g/kWh)
Steam turbine power plant	0.12	0.68	1.11	1130	0.11
Combined cycle	-	0.18	0.09	400	-
Gas turbine cogeneration	-	0.25	0.11	580	-
Diesel engine cogeneration (DEPC) *	0.20	0.612	0.5	500	0.66
Gas engine cogeneration	-	1.34	3.14	529.1	-

* Specific emission amounts for DEPC are obtained from the data given in Table 9.2

Table 9.4 Specific amounts of emissions (φ) of different fuel boosted boilers [214]

Fuel Type	SO_2 (g/kWh)	NO_x (g/kWh)	CO (g/kWh)	CO_2 (g/kWh)	PM (g/kWh)
Natural gas	0.02	0.93	0.07	201.92	0.01
Lignite coal	9.21	0.93	0.45	364.25	10.44
Heavy fuel oil	7.25	0.62	0.06	263.95	0.46

Once the amounts of fuels required to produce the same amount of electricity and thermal product (i.e. steam or hot water) are calculated, the allocation of released emission amounts of actual DEPC system according to the methodologies given in the previous section can be evaluated. The required data for the calculations is taken from Chapter 8.

In order to compare the emissions of the DEPC system with those of other cogeneration systems, average efficiency values that are taken from literature [206,207] are given in Table 9.5.

Table 9.5 First law efficiencies of cogeneration and combined heat and power systems [206,207]

Systems	η_e	η_t	β
Steam turbine power plant	0.334	-	-
Diesel engine power plant	0.4297*		
Combined cycle	0.510	-	-
Gas turbine cogeneration	0.325	0.400	1.2
DEPC*	0.407	0.2478	0.6
Boiler	-	0.850	-

* Values for DEPC are obtained from Chapter 8

In combined heat and power plant applications, fuel consumption increases depending on the separate production of electricity and thermal product (i.e. steam or hot water). Thus, the amount of fuel required producing the same amount of electricity in any power plant or combined heat and power system with the similar fuel type used in DEPC plant in terms of thermal efficiencies can be expressed as

$$\dot{m}_{\text{fuel}_{\text{pp}}} = \dot{m}_{\text{fuel}} \left(\frac{\eta_{\text{e,DEPC}}}{\eta_{\text{e,PP}}} \right) \tag{9.24}$$

Similarly, the amount of fuel required to produce the same amount of heat in the boiler with the same fuel type in terms of thermal efficiencies can be written as

$$\dot{m}_{\text{fuel}_{\text{boiler}}} = \dot{m}_{\text{fuel}} \left(\frac{\eta_{\text{t,DEPC}}}{\eta_{\text{boiler}}} \right) \tag{9.25}$$

In order to avoid inadequacy of using thermal efficiency (i.e. first law) terms in the analysis of fuel saving process and in the analysis of allocation of emissions based on the incremental fuel method, exergetic efficiency terms can be used for the assessment of the DEPC system versus combined heat and power systems or power plants. Thus the corresponding equations in terms of exergetic efficiencies can be expressed, respectively as

$$\dot{m}_{\text{fuel}_{\text{pp}}} = \dot{m}_{\text{fuel}} \left(\frac{\varepsilon_{\text{e,DEPC}}}{\varepsilon_{\text{e,PP}}} \right) \tag{9.26}$$

$$\dot{m}_{\text{fuel}_{\text{boiler}}} = \dot{m}_{\text{fuel}} \left(\frac{\varepsilon_{\text{t,DEPC}}}{\varepsilon_{\text{boiler}}} \right) \tag{9.27}$$

The amounts of fuel required for separate power and heat production in separate units (i.e. power plants and boilers), in terms of thermal efficiencies are found to be 1.307 kg/s and 0.402 kg/s, respectively. The corresponding amounts of fuels, in terms of exergetic efficiencies are 1.387 kg/s and 0.231 kg/s, respectively.

These results show that using separate units of power and heat production increase the fuel consumption by 23.8% in terms of thermal efficiencies, and by 17.3% in terms of exergetic efficiencies with respect to the DEPC system. The obtained results are used to calculate allocation of emissions of DEPC plant based on the incremental fuel consumption to power and heat production in terms of energy and exergy. The results of the calculations of the presented methodologies from the previous section are given through Tables 9.6 to 9.9.

Table 9.6 Allocations of emissions of DEPC based on energy, exergy, and exergoeconomic cost value

	E_e	E_h	X_e	X_h	C_e	C_h
SO_2 (g/kWh)	0.180	0.020	0.199	0.001	0.198	0.002
NO_x (g/kWh)	0.549	0.063	0.608	0.004	0.606	0.006
CO (g/kWh)	0.449	0.052	0.497	0.003	0.495	0.005
CO_2 (g/kWh)	448.5	51.50	496.5	3.50	495.0	5.00
PM (g/kWh)	0.592	0.068	0.655	0.005	0.653	0.007

Table 9.7 Allocation of emissions of DEPC based on the incremental fuel consumption to power production in terms of energy and exergy

	F_{ee}	F_{eh}	X_{Fee}	X_{Feh}
SO_2 (g/kWh)	0.134	0.066	0.163	0.037
NO_x (g/kWh)	0.411	0.201	0.500	0.112
CO (g/kWh)	0.336	0.164	0.408	0.092
CO_2 (g/kWh)	336.1	163.9	408.0	92.0
PM (g/kWh)	0.444	0.216	0.539	0.121

Table 9.8 Allocation of emissions of DEPC based on the incremental fuel consumption to heat production in terms of energy and exergy

	F_{he}	F_{hh}	X_{Fhe}	X_{Fhh}
SO_2 (g/kWh)	0.098	0.102	0.070	0.130
NO_x (g/kWh)	0.300	0.360	0.215	0.397
CO (g/kWh)	0.246	0.254	0.176	0.324
CO_2 (g/kWh)	246.0	254.0	176.0	324.0
PM (g/kWh)	0.324	0.336	0.232	0.428

Table 9.9 Allocation of emissions of DEPC based on a shared emission savings between power and heat in terms of energy and exergy

	F_e	F_h	X_{Fe}	X_{Fh}
SO_2 (g/kWh)	0.189	0.011	0.195	0.005
NO_x (g/kWh)	0.578	0.034	0.597	0.015
CO (g/kWh)	0.473	0.027	0.488	0.012
CO_2 (g/kWh)	473.0	27.0	488.0	12.0
PM (g/kWh)	0.624	0.036	0.644	0.016

Based on the results obtained through Table 9.6 to 9.9, we can clearly see that, the most rational and meaningful method of allocation of emissions of DEPC is based on exergetic efficiency evaluations. The allocation method based on energy and thermal efficiency is very straightforward and simple but it only focuses on quantitative amounts of energy and it may cause misleading results. On the other hand, the exergy method prevents neglecting the share of emissions allocated to electrical product, and allocates a lower portion of the emissions to the thermal product [51] as seen in Table 9.6. For example, when SO_2 is considered, the amount of SO_2 emission based on energy allocated to power and steam are obtained as 0.180 and 0.020 g/kWh, respectively. The corresponding values for exergy based calculations are 0.199 and 0.001 g/kWh, respectively. According to Chapter 8, useful steam output of the DEPC is 176.1 kW. The plant has a power to heat ratio of 143.8 and thus, exergy based allocation of emission of SO_2 can give a more meaningful and proper way of evaluation of emissions. Thus, the proper amounts of emissions can be allocated when they are obtained with a qualitative solution methodology (i.e. exergy) more than a quantitative one (i.e. energy). In Figure 9.1 and 9.2, comparisons of the allocation of emissions of DEPC to the power produced and steam generated in terms of energy and exergy are given according to the results presented in Table 9.6.

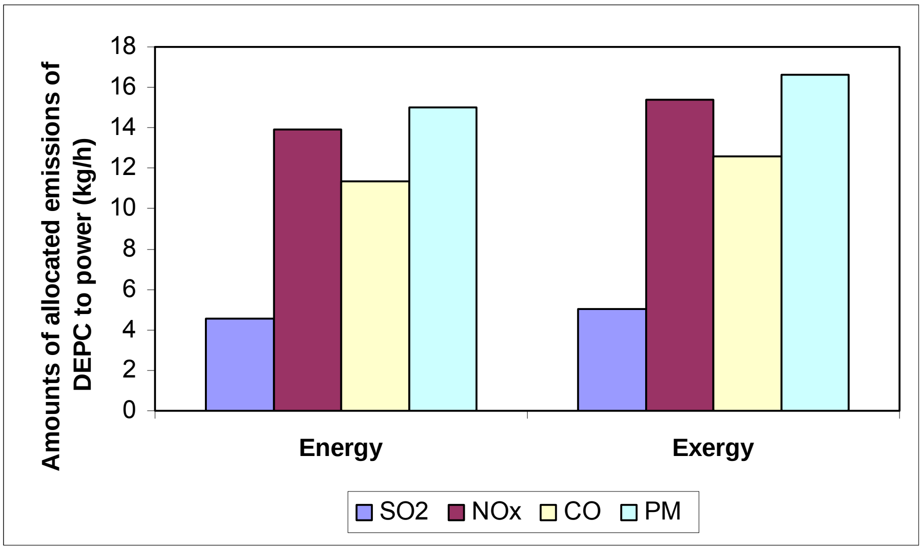

Figure 9.1 Comparison of the allocation of emissions of DEPC to the power production in terms of energy and exergy

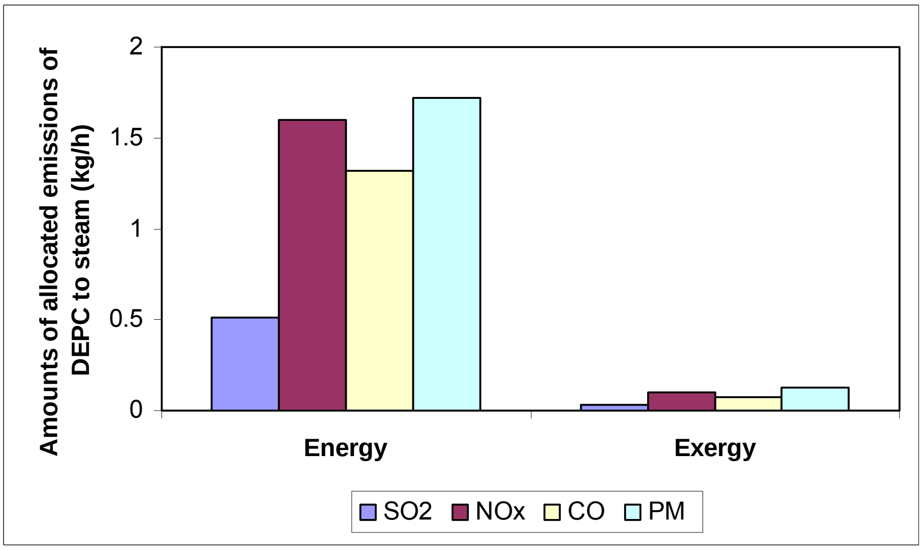

Figure 9.2 Comparison of the allocation of emissions of DEPC to the steam generation in terms of energy and exergy

The allocation method based on the economic value of products can be considered as an advantage for the owners of the cogeneration systems because they may sell power and thermal products separately. Since conventional economic analysis parameters such as raw material cost, fuel and investment costs, and operation and maintenance costs vary with time and location, levelized exergetic cost values make the owner of cogeneration system to allocate proper emission values to the unit cost of the products. Besides, the exergetic cost allocation is adjusted in terms of the weighing of the products in the cogeneration system analyzed in the previous chapter. Thus, using exergoeconomic cost values for the proper allocation of emissions is a rational evaluation approach as compared to using energy based economic parameters. In Figure 9.3, the allocation of emissions of DEPC to the power and steam productions in terms of exergoeconomic cost value is given based on the results given in Table 9.6.

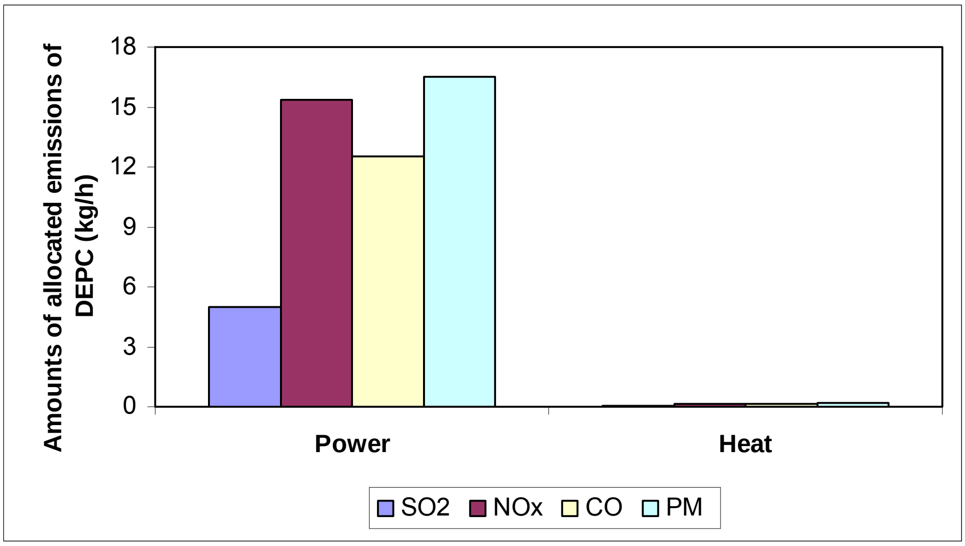

Figure 9.3 The allocation of emissions of DEPC to the power produced and steam generated based on the exergoeconomic cost value

The allocation of emissions based on the incremental fuel consumption to power produced in terms of energy estimates that the thermal efficiency of the generated heat product through cogeneration process is roughly similar to the thermal efficiency of the corresponding heat product via a separate process. Similarly, the allocation of emissions based on the incremental fuel consumption to thermal energy production in terms of energy assumes the energetic efficiency of the power production process through cogeneration is similar to the corresponding one via separate power production. Thus, both methods estimate erroneously that the emissions from one of the products of any cogeneration process is at the expense of the other. As a result, incremental-based allocations of emission methods are unfair with the reason explained for the usage of energy terms (see Tables 9.7 and 9.8). Instead of energy terms, corresponding exergetic ones can be used with more and meaningful results. In the DEPC system, power produced has more weight than the generated steam, and therefore allocation of emissions based on the incremental fuel consumption must be adjusted in a fair way by using the exergetic terms. Comparison of the allocation methods of emissions of DEPC based on the incremental fuel consumption are given through Figure 9.4 to Figure 9.5 based on the results given in Tables 9.7 and 9.8.

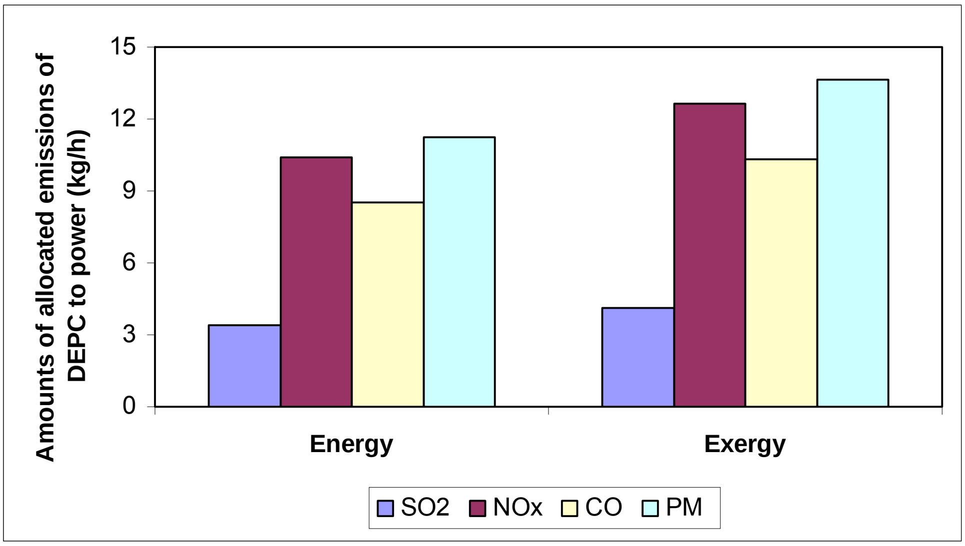

Figure 9.4 Comparison of the allocation of emissions of DEPC based on the incremental fuel consumption to power production based on power produced in terms of energy and exergy

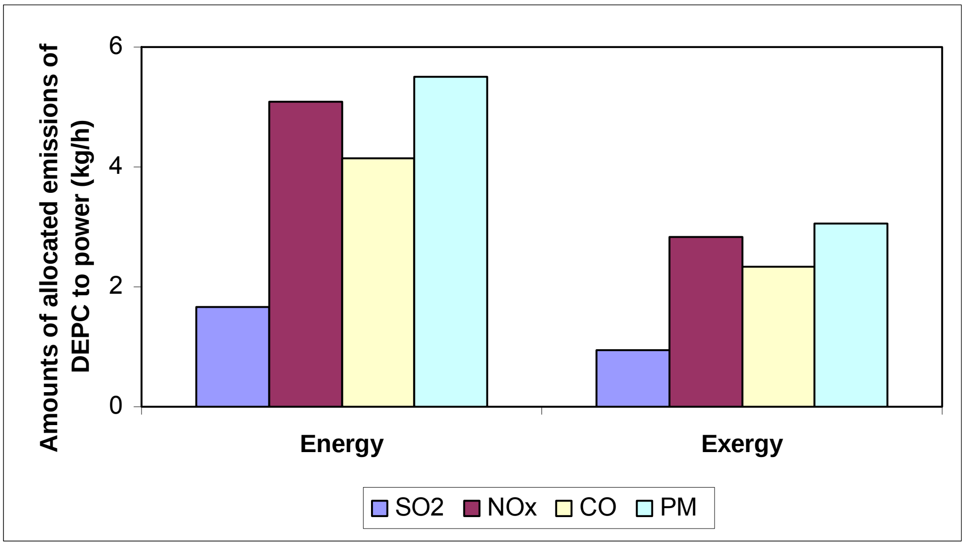

Figure 9.5 Comparison of the allocation of emissions of DEPC based on the incremental fuel consumption to power production based on steam generated in terms of energy and exergy

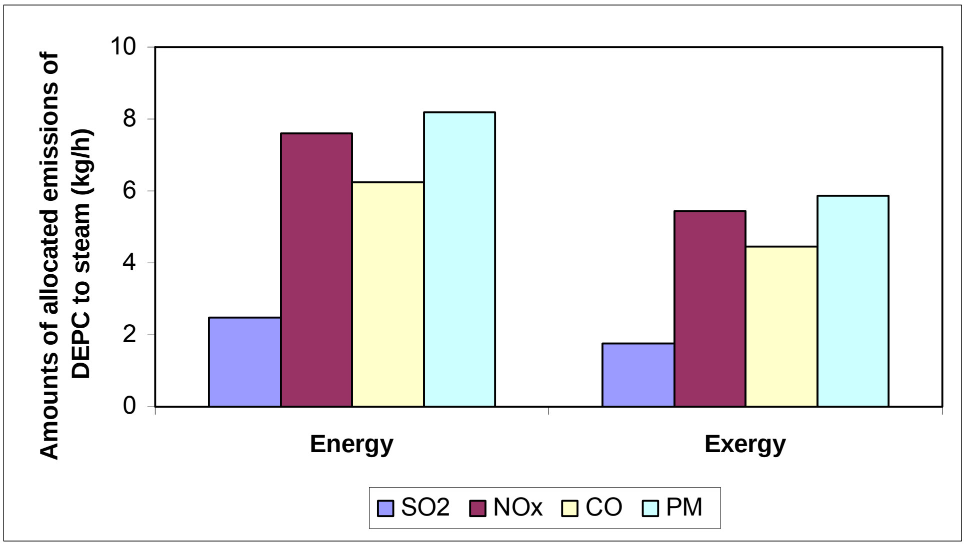

Figure 9.6 Comparison of the allocation of emissions of DEPC based on the incremental fuel consumption to steam generation based on power produced in terms of energy and exergy

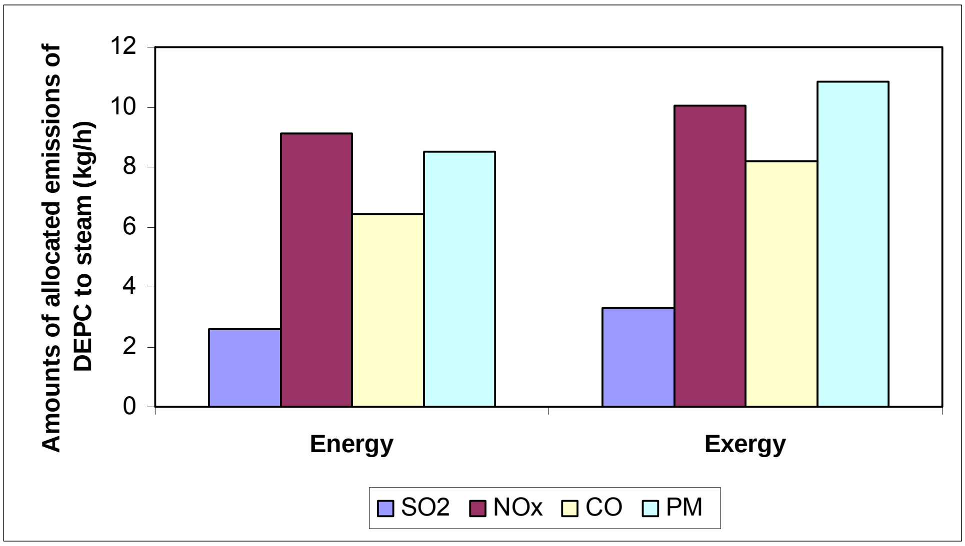

Figure 9.7 Comparison of the allocation of emissions of DEPC based on the incremental fuel consumption to steam generation based on steam generated in terms of energy and exergy

Allocation of emissions based on a shared emission savings method has a common problem with two incremental based emission allocation methods. In these three methods, independent devices for providing electrical energy (i.e. power plant) and thermal energy (i.e. boiler) are assumed to be introduced. The results obtained by using these three allocation methods are strongly dependent on the thermal efficiencies of these independent devices, and the values of thermal efficiencies can vary in a wide range depending on the specific production types and/or specific devices chosen. Consequently, these variations of efficiencies, either thermal or exergetic, cause the emission allocations obtained with these methods to vary over correspondingly wide ranges [208-210]. Allocations of emissions of DEPC to the power produced and steam generated based on the shared emission savings method are given Figures 9.8 and 9.9 respectively, in terms of energy and exergy.

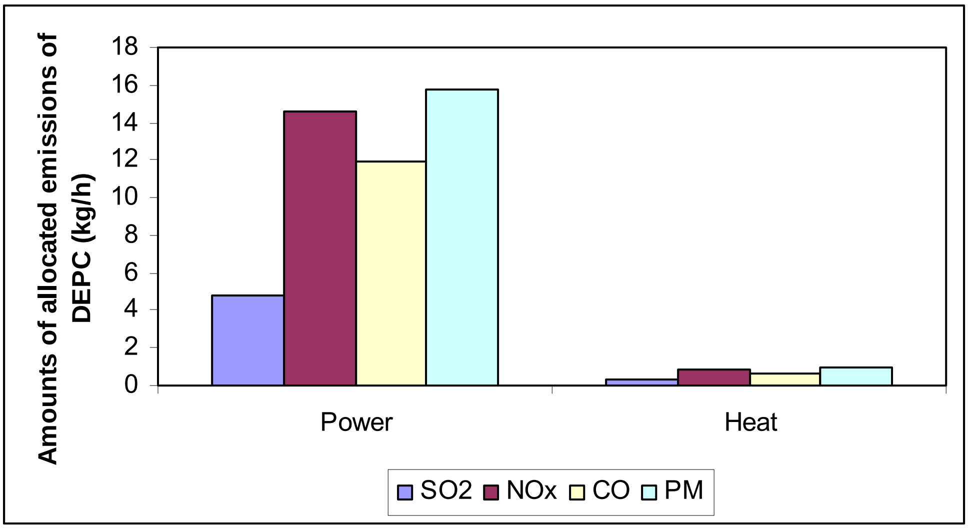

Figure 9.8 Allocation of emissions of DEPC based on a shared emission savings between power and heat in terms of energy

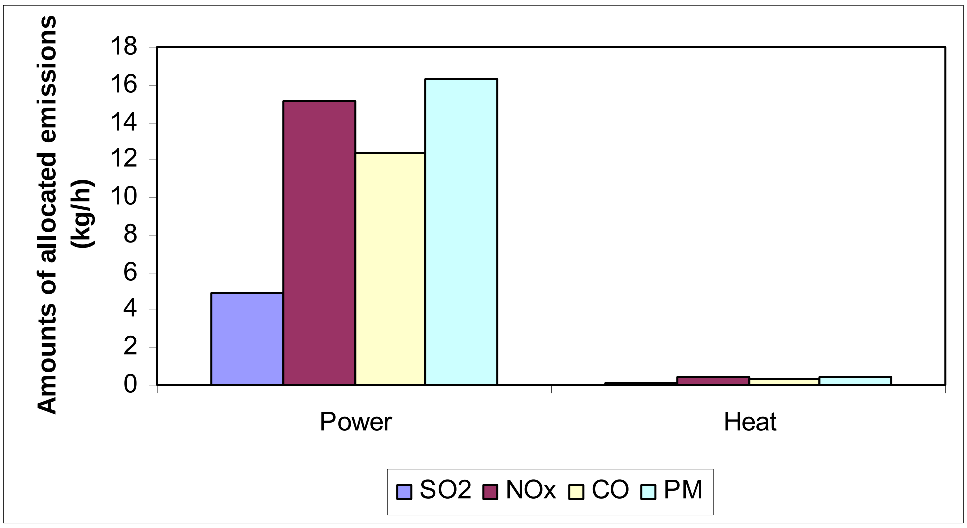

Figure 9.9 Allocation of emissions of DEPC based on a shared emission savings between power and heat in terms of exergy

In Figures 9.10 and 9.11, comparisons of the six main methodologies according to the allocation of CO_2 to the power produced and steam generated of DEPC system in terms of energy and exergy are given.

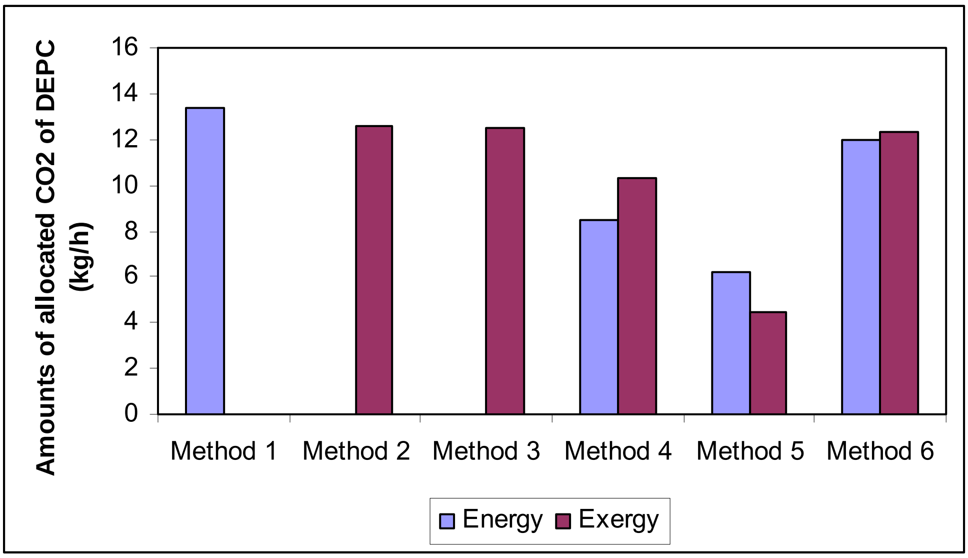

Figure 9.10 Comparison of the amounts of allocation of CO_2 emission of DEPC to the power produced with respect to six main methodologies

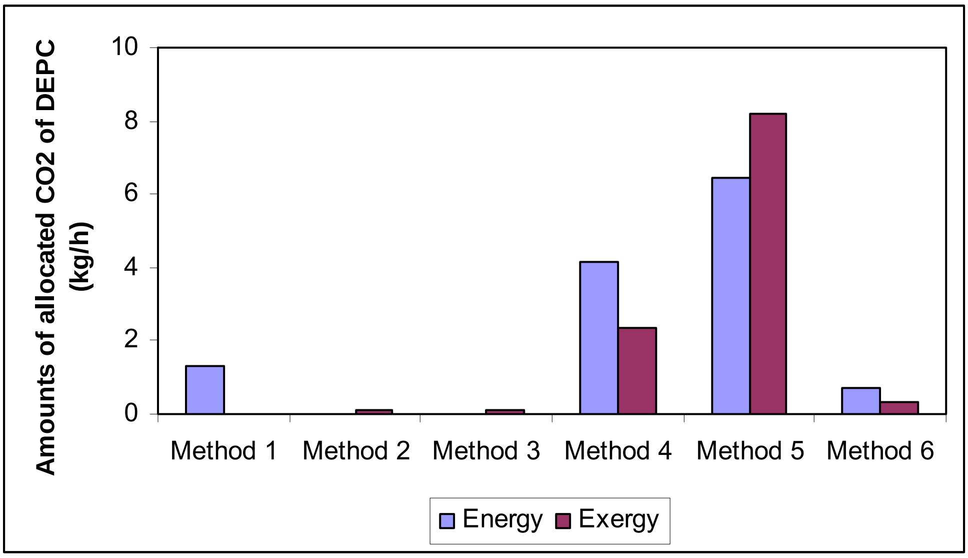

Figure 9.11 Comparison of the amounts of allocation of CO_2 emission of DEPC to the steam generated with respect to six main methodologies

9.5 Conclusions

In this chapter, exhaust emission characteristics of Sanko DEPC plant and the exhaust gas treatment unit in the facility are studied. Exhaust emission assessment is performed by using the six main allocations of emission methods presented in common literature. Methodologies are developed for exergy based analysis of the emissions of DEPC. In order to obtain a rigorous approach for the allocation of emissions of cogeneration systems and to remove the arbitrariness as a result of energy based methods, exergy based methodologies can be used as the most meaningful and accurate methods.

CHAPTER 10

CONCLUSIONS

This study is on thermoeconomic analysis and performance optimization of diesel engine powered cogeneration systems. The developed procedure and formulations are applied to an existing diesel cogeneration plant in Gaziantep, Turkey using actual operational and cost data. The plant has a total installed electricity and steam generation capacities of 25.3 MW and 8100 kg/hr, respectively.

Following conclusions can be drawn based on the analysis and the results obtained:

1. Thermodynamic relations of the plant and its subsystems/components are given in Chapter 8 based on the relations in Chapter 4. The temperature, pressure, and mass flow rate data and certain exergy evaluations of the plant are presented in Table 8.1. Energy and exergy calculations are done using commercial software with built-in thermodynamic property functions for a variety of substances (see Appendix 1). The total exergy input by the fuel for the entire cogeneration plant is 62,757 kW. In Table 8.4, energy and exergy analyses results of the plant are given for one engine set. The exergy assessment of the plant is given schematically in Figure 8.3. The rates of exergy destructions of the components of the plant as compared with total fuel exergy input are given in Figure 8.4.

2. 40.4% of the exergy entering the plant is converted to electrical power and about 5% of this power is used for parasitic load in the plant to drive auxiliary components. The net steam production of the plant represents only 0.3% of the total exergy input. The remaining 59.3% of the exergy input is destroyed. This corresponds to 37,246 kW, which is the total exergy destruction in the plant. The exergy destruction in the diesel engines of the cogeneration plant accounts about 46.0% of the total exergy input and 81.4% of the total exergy destruction in the plant. The exergy destruction in the engine is mostly due to the highly irreversible combustion process, heat losses from the engine, and friction. The exergetic efficiencies of the compressor and turbine of the turbocharger are 82.6% and 88.1%, respectively. These values indicate sufficient exergetic performance from the turbocharger. The exergetic efficiencies of the waste heat boiler and condenser are calculated as 11.4% and 16.6%, respectively making them the least efficient components of the plant. The intercooler has an exergetic efficiency of 26.3%. Exergy destructions in these heat exchange units in the plant are mainly due to the high average temperature difference between the two unmixed fluid streams. The percent of exergy loss associated with lubrication oil cooler is low. This is due to the cooling of lubrication oil by using low temperature water.

3. The fuel utilization efficiency (FUE) of the overall plant is determined to be 44.6%. This value is high compared to thermal efficiencies of power plants whose sole purpose is the production of electricity. In diesel engine cogeneration plants, the main product is electricity and the steam may be called as “byproduct”. The thermal efficiency of the diesel engine defined as the power output over the fuel energy input is calculated to be 43.0%. Power to heat ratio (PHR) of the plant is calculated to be 143.8. The exergetic efficiency of the plant is determined to be 40.6%. When calculating the exergetic efficiency of the overall plant, the input exergy is taken to be the chemical exergy of the fuel. The exergy of the fresh treated water at the inlet of the waste heat boiler is negligible. The exergetic efficiency of the diesel engine itself is 40.4%.

4. The economic data are obtained from the actual vendor quotations of the company. The DEPC plant is supplied as packaged system and cost allocation among its components (i.e. subsystems) is not separately quoted. However, to obtain more accurate results from thermoeconomic analysis, the subsystems are considered as separate and cost allocation of subsystems and the other expenditures are obtained from the energy manager of the company and the contractor of the DEPC system. All cost items including fuel are considered to increase with the general inflation rate which is taken as 5% per year. The economic life of DEPC is considered as 25 years that is from January 1, 2002 to December 31, 2027. The system life for tax purposes is 20 years. The average capacity factor for the cogeneration system is considered as 85% which means that the system will operate at full load 7446 hours out of the total available 8760 hours per year. The levelized total annual value of electricity and steam are found as 10.85 cents/kWh and 30.43×10^3 USD, respectively. The levelized cost values of the carrying charges and expenditures of the DEPC system are given in Table 8.7. Levelized cost rate of the fuel is calculated to be 1806 $/h and that of the raw water to be 4.1 $/h. The purchased equipment costs, the hourly levelized costs of capital investment, the operating and maintenance costs, and the total costs of the components of the plant are given in Table 8.8.

5. In this study, specific exergy costing (SPECO) method is used to obtain and understand the cost formation structure of the plant. Exergetic cost rates balances and corresponding auxiliary equations of the plant are given in Chapter 5. Exergetic cost rate balances and corresponding auxiliary equations are formulated for each subsystem of the plant. Auxiliary equations are found by applying F and P principles. Results obtained are given in Table 8.9. The exergetic cost parameters of the plant components are given in Table 8.10. These parameters indicate the performance of system components on a rational exergetic cost basis. The exergetic cost rate and the specific unit exergetic cost of the fuel entering the plant are found as 1806 $/h and 2.70 $/GJ, respectively. The corresponding values for the diesel engines are 1933.3 $/h and 2.85 $/GJ. The capital investment cost, the operating and maintenance costs, and the total cost of the DEPC system are found to be 770.5 $/h, 174.5 $/h and 945 $/h, respectively. The exergetic cost rate and the specific unit exergetic cost of the power produced by the plant are 2844 $/h and 10.31 $/GJ, respectively. The exergetic cost rate and specific unit exergetic costs of steam are 87.73 $/h and 33.71 $/GJ, respectively.

6. Exergetic cost rates difference between electric and steam production outputs is determined to be very high for this DEPC system. This is directly proportional to the exergy allocation of fuel between steam and electric outputs. Diesel engine is the most exergy destructive component of the plant. The exergoeconomic factor of the diesel engine is determined to be 63.3%. The exergy unit cost is highest for the pump work since all exergy available at the exit of the pump is supplied by mechanical power which is the most expensive "fuel" in the system. The exergoeconomic factor is rather high (81%) due to low initial investment and exergetic destruction cost rates. Exergoeconomic factors for fuel forwarding module, fuel oil day tank, and condenser are 98.6%, 95.2%, and 96.7%, respectively. The exergetic cost value is 23.02 $/GJ for the compressor work and 5.50 $/GJ for the exhaust gas stream. This difference makes the exergetic destruction cost rate for the compressor dominant in the exergoeconomic factor. The exergoeconomic factor of the compressor is 41.3%, which is rather low compared with the other components. This value is 79.0% for the turbine. Waste heat boiler unit involves an exergetic destruction rate of 1404 kW. The exergoeconomic factor of intercooler is 35.2%. The relative cost difference for lubrication oil cooler is determined to be 60.0%, which is the second lowest value after the compressor. It also has a low destruction cost rate of 3.37 $/h. Desulphurization ($DeSO_x$) unit has the lowest exergoeconomic factor among the components of the plant. This is expected since it is the most destructive unit in the plant.

7. For an existing system such as the DEPC system of this study, performance evaluation and optimization procedure may be considered as "performance improvement" and "searching a good solution" for the overall system rather than finding a global optimum. Since the optimal values of the decision variables given for thermodynamically and thermoeconomically optimal cases are not unique, the same values of the maximum exergetic efficiency and the minimum overall cost rate may be obtained through other combinations of the values of the decision variables. Besides, many different sets of the decision variables values may lead to nearly optimal values of the objective function. In the application of the iterative optimum procedure to the plant components, study presented follows the following procedure: (a) evaluation of detailed exergy analysis at the DEPC plant component level, (b) calculation of capital costs associated with each plant component, (c) an exergoeconomic analysis using en exergy based costing method (SPECO method in this study), which is as detailed and objective as possible by keeping aggregation level is low, and (d) evaluation of the effects of decision variables on selected exergoeconomic variables. The integration of cost and performance data for a given component permits the calculation of optimum design conditions when the exergy of the component product $\dot{E}_{\mathrm{P,k}}$ and the cost of the component fuel, $c_{\mathrm{F,k}}$ remain constant.

8. In the iterative optimization procedure we use the variables the relative cost difference Δr and exergetic efficiency ε with the corresponding optimal values obtained through the optimization procedure. The effects of changes in the decision variables selected on the values of relative cost difference, exergetic efficiency and destruction cost rate, and the objective functions of components suggest the design changes that need to be considered in the next optimization step. Tables 8.20, 8.21 and 8.22 show the comparison of the thermodynamically and thermoeconomically optimal cases of DEPC plant with actual base case. Starting from the compressor of the turbocharger, major plant components are considered as thermoeconomically isolated and are searched for optimum range. When an optimal is found, it is fixed in the optimization of the next component. This make all plant equipments thermoeconomically at optimal level. However, this may not be "globally optimum" while it may be a good solution (or sometimes a good range) for performance improvement of existing system. In the actual base case, electricity stream cost is calculated as 10.31 \$/GJ or 8.90 ¢/kWh, which appears to be lower than that in gas and steam turbines presented in open literature. This is due to relatively low price for fuel oil and low investment cost for this DEPC plant. According to the thermoeconomical optimization study, electricity stream cost is determined to be 6.70 ¢/kWh (8.38 \$/GJ) whereas corresponding value of thermodynamically optimal case is 23.18 ¢/kWh (21.36 \$/GJ). The cost rate of steam for actual base case, thermodynamically optimum and thermoeconomically optimum cases are determined to be 5.22 ¢/kWh, 10.04 ¢/kWh, and 4.50 ¢/kWh, respectively. In thermoeconomic optimization studies, differences between the base case value and optimal value and between the two methods of optimization are expected.

9. Using the equations given in Chapter 8 and the plant data in Table 8.1, various engine operating and performance characteristics are calculated. The results are listed in Table 8.3. Certain engine parameters, which are not specified by the manufacturer, are calculated. These include specific power (6217 kW/m^2), specific volume (64.34 m^3/kW), specific weight (8.67 ton/MW), output per displacement (14.58 kW/L), volumetric efficiency (1.29), and thermal efficiency (0.47). These parameters can be used for the performance comparison of internal combustion engines. The values obtained are typical of stationary engines. Perhaps, the most important result is the high thermal efficiency(47 percent) compared to automobile engines. The typical brake thermal efficiencies are 30 to 40 percent for diesel automobile engines and 25 to 35 percent for gasoline automobile engines. This can be explained by the fact that the stationary engines operate at their optimum values with a constant engine speed. The entire operation is optimized to minimize the fuel consumption for a given power output. The inherent limitations such as space, weight, complexity, and maintenance in the design of automobile engines are not crucial for stationary large engines. Typical thermal efficiencies are 30 to 40 percent for steam and gas turbine power plants and close to 50 percent for combined cycle power plants. It is clear that the diesel engine power plant operates at high thermal efficiency and consequently the cost of electricity should be low compared to other power systems.

10. The indicated power and indicated thermal efficiency values are calculated as 11,435 kW and 59.4%, respectively. These values are obtained as a result of the ideal diesel cycle analysis given in Chapter 8. The actual brake power output from the engine is obtained to be about 8440 kW and the actual brake thermal efficiency to be 47%. The difference between the actual brake values and ideal cycle indicated values are due to the mechanical inefficiencies as the power is transferred from inside the cylinder to the crankshaft and the assumptions used in the analysis of ideal diesel cycle.

11. Heavy fuel oil has a high share in fossil fuel consumption especially for diesel engine powered cogeneration applications. One of the major important aspects of heavy fuel oil fired diesel cogeneration systems is environmental. Combustion of lower quality fuels like heavy fuel oil that is rich in sulfur and asphaltene in compression ignition engines causes an increased emission of harmful pollutants: CO_2, SO_2, NO_x, hydrocarbons, particulate matters (PM) and volatile organic compounds (VOC). In this thesis, exhaust emission characteristics of DEPC plant are also assessed, and allocation amounts of emissions of the DEPC system to the power produced and steam generated are calculated. Exhaust emission assessment is done following both energy and exergy based approaches. We determined that using combined power and heat production provide fuel savings of 23.8% in energy approach and 17.3% in exergy approach with respect to separate units of power and heat production.

12. It may be concluded that the exergetic and thermoeconomic performance analysis and optimization of diesel engine powered cogeneration systems can be used as a guide study to analyze and evaluate the exergoeconomic performance analysis and optimization of other cogeneration systems and power plants. The results of the present thesis study are also expected to give a new and original direction to engineers, scientists and energy policy makers in implementing energy planning studies and dictating the energy strategies as a potential tool in the light of exergy and exergy based economical methodologies.

13. The results of this thesis provides cogeneration plant investers, designers and engineers some key information: (*i*) Diesel engine powered cogeneration applications are characterized with a high power to heat ratio, and thus they should be used when power demand is high and heat demand is low. (*ii*) Exergy methods can be effectively used to analyze diesel cogeneration systems both thermodynamically and economically providing rational comparison to other cogeneration applications. (*iii*) Exergy methods can effectively be used for rational allocation of emissions from DEPC systems. (*iv*) Diesel cogeneration is a key technology for using heavy fuel oil and it can be cost-effective depending on the local cost of fuel-oil.

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APPENDIX

1. Thermodynamic Analysis of Sanko DEPC Plant by EES

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"SANKO DIESEL ENGINE COGENERATION SYSTEM : ACTUAL DATA JULY 2005 STUDY"
"ENERGY AND EXERGY ANALYSIS OF DIESEL COGENERATION SYSTEM"

"Ambient pressure and temperature are taken as 1 bar and 30 C - July 2005"

"1. COMPRESSOR OF THE TURBOCHARGER - STATES 1-2"

"Assumptions: 1. This is a steady flow process since there is no change with time at any point. 2. Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141 C and 3.77 MPa. 3. The kinetic and potential energy changes are zero. 4. There are two turbocharger sets for each diesel engine, in calculations one compressor of the two turbochargers are considered. Therefore mass flowrate of air for one engine is divided by two for each turbocharger, it is multiplied by two before entering engine "

"ENERGY - FIRST LAW ANALYSIS"

W_dot_compressor_actual=m_dot_1*(h_2-h_1) "[kW]" "Actual compressor power input to the compressor"
m_dot_2=m_dot_1 "[kg/s]" "Conservation of mass"
h_1=enthalpy(Air; T=T_1) "[kJ/kg]" "Enthalpy of the air at the inlet state of the compressor - i.e. fixed state"
h_2=enthalpy(Air; T=T_2) "[kJ/kg]" " Actual enthalpy of the air at the outlet state of the compressor"
W_dot_compressor_isentropic=eta_compressor*W_dot_compressor_actual "[kW]" "Isentropic power input to the compressor"

"EXERGY - SECOND LAW ANALYSIS"

e_1=(h_1-h_0_AIR)-T_0*(s_1-s_0_AIR)"[kJ/kg]" "exergy of the air at the inlet state, i.e. state 1, of the compressor per unit mass"
e_2=(h_2-h_0_AIR)-T_0*(s_2-s_0_AIR) "[kJ/kg]" "exergy of the air at the outlet state, i.e. state 2, of the compressor per unit mass"
E_dot_compressor=m_dot_1*(e_2-e_1) "[kW]" "Minimum possible power consumed by the compressor"
h_0_AIR=enthalpy(Air; T=T_0) "[kJ/kg]"
s_0_AIR=entropy(Air; T=T_0; P=P_0) "[kJ/kgK]"
s_1=entropy(Air; T=T_1; P=P_1) "[kJ/kgK]"
s_2=entropy(Air; T=T_2; P=P_2) "[kJ/kgK]"
epsilon_compressor=E_dot_compressor/W_dot_compressor_actual "Exergetic - second law efficiency of the compressor of the turbocharger"

"Data"
T_1=303 "[K]" : P_1=100 "[kPa]" : P_2=290"[kPa]" : eta_compressor=0,80 : m_dot_1=9,2 "[kg/s]" : R=0,287 "[kJ/kgK]" : T_0=303 "[K]" : P_0=100 "[kPa]" : T_2=445 "[K]"
```

"2. INTERCOOLER - STATES 2-3"

"Assumptions: 1. This is a steady flow process since there is no change with time at any point. 2. The kinetic and potential energies are negligible. 3. Heat losses from the system are negligible. 4. There is no work interaction. 5. Intercooler is made up of three identical multipasses counter flow heat exchanger."

"2a. Air - High Temperature (HT) Water Heat Exchange" "Mass flowrate of HT Water is divided by two for each turbocharger of one engine"

"ENERGY - FIRST LAW ANALYSIS"

Q_dot_actual_intercooler_1=m_dot_20*(h_21-h_20) "[kW]" "Heat gain by high temperature water"
h_20=enthalpy(Water; T=T_20;P=P_20) "[kJ/kg]"
h_21=enthalpy(Water; T=T_21;P=P_21) "[kJ/kg]"
h_22=enthalpy(Water; T=T_22;P=P_22) "[kJ/kg]"
s_22=entropy(Water; T=T_22; P=P_22) "[kJ/kgK]"
Q_dot_actual_intercooler_1=m_dot_3*(h_2-h_3a) "[kW]" "Heat lost by air"
m_dot_3=m_dot_2 "[kg/s]"
T_3a=temperature(Air; h=h_3a) "[K]"

"EXERGY - SECOND LAW ANALYSIS"

e_20=(h_20-h_0_WATER)-T_0*(s_20-s_0_WATER) "[kJ/kg]" "exergy of HTW at state 20 per unit mass"
e_21=(h_21-h_0_WATER)-T_0*(s_21-s_0_WATER) "[kJ/kg]" "exergy of HTW at state 21 per unit mass"
s_0_WATER=entropy(Water; T=T_0; P=P_0) "[kJ/kgK]"
h_0_WATER=enthalpy(Water; T=T_0; P=P_0) "[kJ/kg]"
s_20=entropy(Water; T=T_20; P=P_20) "[kJ/kgK]"
s_21=entropy(Water; T=T_21; P=P_21) "[kJ/kgK]"
e_3a=(h_3a-h_0_AIR)-T_0*(s_3a-s_0_AIR) "[kJ/kg]" "exergy of AIR at state 3a per unit mass"
s_3a=entropy(Air; T=T_3a; P=P_3a) "[kJ/kgK]"
delta_P_IC_1=P_2-P_3a "[kPa]"
epsilon_intercooler_1=(m_dot_20*(e_21-e_20))/(m_dot_3*(e_2-e_3a))

"Data"
T_20= 344,7 "[K]" : m_dot_20=15 "[kg/s]" : T_21=354"[K]" : P_20=310 "[kPa]" : P_21=300 "[kPa]" : delta_P_IC_1=5 "[kPa]" : T_22=361 "[K]" : P_22=280 "[kPa]"

"2b. Air - Low Temperature (LT) Water Heat Exchange" "Mass flowrate of LT Water is divided by two for each turbocharger of one engine"

"ENERGY - FIRST LAW ANALYSIS"

Q_dot_actual_intercooler_2=m_dot_23*(h_24-h_23) "[kW]" "Heat gain by low temperature water"
h_23=enthalpy(Water; T=T_23;P=P_23)
h_24=enthalpy(Water; T=T_24;P=P_24)
Q_dot_actual_intercooler_2=m_dot_3*(h_3a-h_3) "[kW]"
delta_P_IC_2=P_3a-P_3 "[kPa]"
Q_dot_actual_intercooler_total=Q_dot_actual_intercooler_1+Q_dot_actual_intercooler_2 "[kW]"

"EXERGY - SECOND LAW ANALYSIS"

e_23=(h_23-h_0_WATER)-T_0*(s_23-s_0_WATER) "[kJ/kg]" "exergy of LTW at state 23 per unit mass"
e_24=(h_24-h_0_WATER)-T_0*(s_24-s_0_WATER) "[kJ/kg]" "exergy of LTW at state 24 per unit mass"
s_23=entropy(Water; T=T_23; P=P_23) "[kJ/kgK]"
s_24=entropy(Water; T=T_24; P=P_24) "[kJ/kgK]"
e_3=(h_3-h_0_AIR)-T_0*(s_3-s_0_AIR) "[kJ/kg]" "exergy of AIR at state 3 per unit mass"
s_3=entropy(Air; T=T_3; P=P_3) "[kJ/kgK]"
epsilon_intercooler_2=(m_dot_23*(e_24-e_23))/(m_dot_3*(e_3a-e_3))
epsilon_intercooler=(m_dot_20*(e_21-e_20)+m_dot_23*(e_24-e_23))/(m_dot_3*(e_2-e_3))
"the second law - exergetic efficiency of the intercooler"

"Data"
T_23= 311,4 "[K]" : m_dot_23=23,6 "[kg/s]" : T_24=317,9 "[K]" : P_23=310 "[kPa]" : P_24=300 "kPa]": delta_P_IC_2=5 "[kPa]" : T_3=326,7 "[K]"

"3. LUBRICATION OIL COOLER - STATES 24 - 25 (LTW) ; STATES 27-28 (Lubrication Oil-LO)"

"ENERGY - FIRST LAW ANALYSIS"

{Q_dot_actual_luboilcooler=m_dot_24*(h_25-h_24) "[kW]" "Heat gain by low temperature water"}
h_25=enthalpy(Water; T=T_25;P=P_25) "[kJ/kg]"
m_dot_24=m_dot_23 "[kg/s]"
Q_dot_actual_luboilcooler=m_dot_27*C_p_27*(T_27-T_28) "[kW]"

"EXERGY - SECOND LAW ANALYSIS"

e_25=(h_25-h_0_WATER)-T_0*(s_25-s_0_WATER) "[kJ/kg]" "exergy of LTW at state 25 per unit mass"
s_25=entropy(Water; T=T_25; P=P_25) "[kJ/kgK]"
e_27= C_p_27*(T_27-T_0)-T_0*(C_p_27*ln(T_27/T_0)) "[kJ/kg]" "exergy of LO at state 27 per unit mass"
e_28=C_p_28*(T_28-T_0)-T_0*(C_p_28*ln(T_28/T_0)) "[kJ/kg]" "exergy of LO at state 28 per unit mass"
epsilon_luboilcooler=(m_dot_24*(e_25-e_24))/(m_dot_27*(e_27-e_28)) "the second law - exergetic efficiency of the lubrication oil cooler"
"C_p values for unused engine oil were taken from M. Necati Özışık's Heat Transfer Textbook"

"Data"
T_25=323,3 "[K]" : m_dot_27= 10 "[kg/s]" : C_p_27=2,124 "[kJ/kgK]" : C_p_28=2,046 : T_27=351,3 "[K]" : T_28=332,7 "[K]" : P_25=290 "[kPa]"

"4. DIESEL ENGINE"

"Assumptions: 1. The engine is compression ignition engine and it may be represented by an ideal diesel cycle, which is an air-standard cycle. 2. The gas mixture in the cylinder is treated as air for the entire cycle, and ideal gas properties of air are used in the analysis. 3. The real open cycle is changed into a closed cycle. 4. The combustion process is replaced with a heat addition process,this process takes place at constant pressure in ideal diesel cycle. 5. Intake and exhaust strokes are assumed to be constant pressure. 6. Compression and expansion are approximated by isentropic processes. 7. Exhaust blowdown is replaced by a constant volume heat rejection process."

"Process 1-2 Isentropic Compression Stroke - All valves closed"

```
T_1_d=T_3 "[K]"
P_1_d=P_3 "[kPa]"
r_c=V_1/V_2 "Compression ratio"
r_c=(V_c+V_d)/V_c "V_c : clearance volume of the cylinder, V_d : displacement volume of the cylinder"
V_1=V_c+V_d "[m3]" "V_1=V_BDC"
T_2_d=T_1_d*(r_c^(k-1)) "[K]"
P_2_d=P_1_d*(r_c^k) "[kPa]"
w_12_d=R*(T_2_d-T_1_d)/(1-k) "[kJ/kg]"
m_mix=P_1_d*V_1/(R*T_1_d) "[kg]"

"Data"
V_d=0,0322 "[m3]" : r_c=12,37 : k=1,35

"Process 2-3 Constant Pressure Heat Input - Combustion Stroke - All valves closed"

P_3_d=P_2_d "[kPa]"
Q_dot_in_d=m_dot_mix*C_p_23_d*(T_3_d-T_2_d) "[kW]"
Q_dot_in_d=m_dot_fuel*Q_LHV*eta_combustion"[kW]"
m_dot_mix=(2*m_dot_3)+m_dot_fuel "[kg/s]" "There are two turbochargers for one engine set, therefore we must multiply air flowrate by two"
w_23_d=P_2_d*(V_3-V_2) "[kJ/kg]"
beta=V_3/V_2
beta=T_3_d/T_2_d
C_p_23_d=specheat(Air; T=T_2_d) "[kJ/kgK]"

"Data"
m_dot_fuel= 0,460 "[kg/s]" : Q_LHV= 42700 "[kJ/kg]" : eta_combustion=0,98

"Process 3-4 Isentropic Power or Expansion Stroke - All valves closed"

T_4_d=T_3_d*((V_3/V_4)^(k-1)) "[K]"
V_4=V_1 "[m3]"
P_4_d=P_3_d*((V_3/V_4)^k) "[kPa]"
w_34_d=R*(T_4_d-T_3_d)/(1-k) "[kJ/kg]"

"Process 4-5 Constant Volume Heat Rejection - Exhaust Blowdown - Exhaust valve open and Intake valve closed"

Q_dot_out_d=m_dot_mix*C_v_45*(T_1_d-T_4_d) "[kW]"
C_v_45=specheat(Air; T=T_4_d) "[kJ/kgK]"

"Net power of the Diesel Engine"

W_dot_net_diesel=(w_12_d+w_23_d+w_34_d)*m_mix*N/2 "[kW]" "Indicated Power"
eta_thermal_diesel=W_dot_net_diesel/Q_dot_in_d "Indicated Thermal Efficiency"
N=750 "[rpm]" "Speed of engine"


"5. TURBINE OF THE TURBOCHARGER - STATES 6-7"

"Assumptions: 1. This is a steady flow process since there is no change with time at any point.  2. The kinetic and potential energy changes are zero. 3. There are two turbocharger sets for each diesel engine, in calculations one turbine of the each turbocharger are considered. 4. Exhaust gas is assumed as ideal gas and specific heat value of the exhaust gas is obtained as 1.063 times specific heat of the air."
```

"ENERGY - FIRST LAW ANALYSIS"

C_p_6=1,063*(specheat (Air; T=T_6)) "Specific heat of the exhaust gas at the inlet of the turbine"
W_dot_turbine_actual=m_dot_6*(h_6-h_7) "[kW]" "Actual turbine power output"
h_6=enthalpy(Air; T=T_6) "[kJ/kg]" "Enthalpy of the exhaust gas at the inlet state of the turbine - i.e. fixed state"
h_7=enthalpy(Air; T=T_7) "[kJ/kg]" " Actual enthalpy of the air at the outlet state of the turbine"
W_dot_turbine_isentropic=W_dot_turbine_actual/eta_turbine "[kW]" "Isentropic power output from turbine"

"EXERGY - SECOND LAW ANALYSIS"

e_6=(h_6-h_0_AIR)-T_0*(s_6-s_0_AIR) "[kJ/kg]" "exergy of exhaust gas at state 6 per unit mass"
e_7=(h_7-h_0_AIR)-T_0*(s_7-s_0_AIR) "[kJ/kg]" "exergy of exhaust gas at state 6 per unit mass"
s_6=entropy(Air; T=T_6; P=P_6) "[kJ/kgK]"
s_7=entropy(Air; T=T_7; P=P_7) "[kJ/kgK]"
E_dot_turbine=m_dot_6*(e_6-e_7) "[kW]" "Maximum possible power output of the turbine"
epsilon_turbine= W_dot_turbine_actual/E_dot_turbine "the second law -exergetic- efficiency of the turbine"

"Data"
T_6=724 "[K]": P_6=240 "[kPa]": m_dot_6=8,5 "[kg/s]": T_7=575 "[K]": P_7=80 "[kPa]": eta_turbine=0,85

"6. WASTE HEAT BOILER - STATES 7-8 (Exhaust Gas); STATES 12-13 (Feed Water)"

"Assumptions: 1. This is a steady flow process since there is no change with time at any point. 2. The kinetic and potential energies are negligible. 3. Heat losses from the system are negligible. 4. There is no work interaction. 5. Waste Heat Boiler (WHB) is made up of two identical multipasses counter flow heat exchanger. 6.Exhaust gas from two turbochargers set are considered for one engine. "

"ENERGY - FIRST LAW ANALYSIS"

Q_dot_actual_wasteheatboiler=m_dot_7*(h_7-h_8) "[kW]" "Actual loss of heat transfer rate from hot exhaust gas "
m_dot_7=2*m_dot_6 "[kg/s]"
{Q_dot_actual_wasteheatboiler=m_dot_12*(h_13-h_12) "[kW]" "Actual gain of of heat transfer rate by feed water"}
h_12=enthalpy(Water; T=T_12; P=P_12) "[kJ/kg]"
h_13=enthalpy(Water; T=T_13; P=P_13) "[kJ/kg]"

"EXERGY - SECOND LAW ANALYSIS"

e_8=(h_8-h_0_AIR)-T_0*(s_8-s_0_AIR) "[kJ/kg]" "exergy of exhaust gas at state 8 per unit mass"
h_8=enthalpy(Air; T=T_8)
e_12=(h_12-h_0_WATER)-T_0*(s_12-s_0_WATER) "[kJ/kg]" "exergy of feed water at state 12 per unit mass"
e_13=(h_13-h_0_WATER)-T_0*(s_13-s_0_WATER) "[kJ/kg]" "exergy of feed water at state 13 per unit mass"
s_8=entropy(Air; T=T_8; P=P_8) "[kJ/kgK]"
s_12=entropy(Water; T=T_12; P=P_12) "[kJ/kgK]"

s_13=entropy(Water; T=T_13; P=P_13) "[kJ/kgK]"
epsilon_wasteheatboiler=(m_dot_12*(e_13-e_12))/(m_dot_7*(e_7-e_8)) "the second law - exergetic efficiency of the waste heat boiler"

"Data"
T_12=368 "[K]" : P_12=800 "[kPa]" : m_dot_12=0,75 "[kg/s]" : T_13=443 "[K]" : P_13=800 "[kPa]" : P_8=75 "[kPa]" : T_8=520 "[K]"

"7. DESOX UNIT - STATES 8-9"

"DESOX Unit is used for the treatment of the sulphur of the exhaust gas. At the exit of the chimney of the DESOX unit exhaust gas temperature is in the limit of <61 degrees celcius. There is a large amount of heat loss from the DESOX unit to the atmosphere"

"ENERGY - FIRST LAW ANALYSIS"

Q_dot_DESOX_lost=m_dot_8*(h_8-h_9) "[kW]" "Heat lost from DESOX unit to the atmosphere"
m_dot_8=m_dot_7 "[kg/s]"
h_9=enthalpy(Air; T=T_9) "[kJ/kg]"

"EXERGY - SECOND LAW ANALYSIS"

e_9=(h_9-h_0_AIR)-T_0*(s_9-s_0_AIR) "[kJ/kg]" "exergy of exhaust gas at state 9 per unit mass"
s_9=entropy(Air; T=T_9; P=P_9) "[kJ/kgK]"
E_dot_DESOX=m_dot_8*(e_8-e_9) "[kW]" "exergy lost through DESOX unit"

"Data"
T_9=326,7 "[K]" : P_9=119 "[kPa]"

"8. FUEL - OIL DAY TANK (FDT) - STATES 15-16 (STEAM); STATES 0-4 (FUEL-OIL)"

"ENERGY - FIRST LAW ANALYSIS"

Q_dot_actual_fueloildaytank=m_dot_15*(h_15-h_16) "[kW]" "Actual heat lost from steam through FDT"
h_15=enthalpy(Water; T=T_15; P=P_15) "[kJ/kg]"
h_16=enthalpy(Water; T=T_16; P=P_16) "[kJ/kg]"
m_dot_4=m_dot_fuel "[kg/s]"

"EXERGY - SECOND LAW ANALYSIS"

e_4=C_p_4_FO*((T_4-T_0)-T_0*ln(T_4/T_0)) "[kJ/kg]" "exergy of fuel oil at state 4"
e_15=(h_15-h_0_WATER)-T_0*(s_15-s_0_WATER) "[kJ/kg]" "exergy of water at state 15"
e_16=(h_16-h_0_WATER)-T_0*(s_16-s_0_WATER) "[kJ/kg]" "exergy of water at state 16"
s_15=entropy(Water; T=T_15; P=P_15) "[kJ/kgK]"
s_16=entropy(Water; T=T_16; P=P_16) "[kJ/kgK]"
epsilon_fueloildaytank=(m_dot_4*(e_4-e_0_FO))/(m_dot_15*(e_15-e_16)) "exergetic - second law efficiency of FDT"

"Data"
m_dot_15=0,05 "[kg/s]" : T_0_FO=303 "[K]" : T_15=443 "[K]" : T_16=343 "[K]" : P_15=800 "[kPa]" : P_16=780 "[kPa]" : C_p_0_FO= 1,922 "[kJ/kgK]" : C_p_4_FO=2,135 "[kJ/kgK]" : T_4=354 "[K]" : P_4=430 "[kPa]" : e_0_FO=0 "[kJ/kg]"

"9. FUEL FORWARDING MODULE (FFM) - STATES 4-5 (FUEL-OIL) ; STATES 17-18 (STEAM)"

"ENERGY - FIRST LAW ANALYSIS"

Q_dot_actual_fuelforwardingmod=m_dot_17*(h_17-h_18) "[kW]" "Actual heat lost from steam through FFM"
h_17=h_15 "[kJ/kg]"
h_18=enthalpy(Water; T=T_18; P=P_18) "[kJ/kg]"
T_17=T_15 "[K]"
P_17=P_15 "[kPa]"
m_dot_5=m_dot_4

"EXERGY - SECOND LAW ANALYSIS"

e_17=e_15 "[kJ/kg]" "exergy of water at state 17"
e_18=(h_18-h_0_WATER)-T_0*(s_18-s_0_WATER) "[kJ/kg]" "exergy of water at state 18"
s_18=entropy(Water; T=T_18; P=P_18) "[kJ/kgK]"
e_5=C_p_5_FO*((T_5-T_0)-T_0*ln(T_5/T_0)) "[kJ/kg]" "exergy of fuel oil through FFM"
epsilon_fuelforwardingmodule=(m_dot_4*(e_5-e_4))/(m_dot_17*(e_17-e_18)) "exergetic - second law efficiency of FFM"

"Data"
T_18=333 "[K]" : P_18=470 "[kPa]" : T_5=410,5 "[K]" : m_dot_17=0,15 "[kg/s]" : C_p_5_FO=2,384 "[kJ/kgK]"

"10. CONDENSER - STATES 18-19 (WATER) ; 29-30 (AIR)"

"ENERGY - FIRST LAW ANALYSIS"

Q_dot_actual_condenser=m_dot_18*(h_18-h_19) "[kW]" "Actual heat lost from steam through Condenser"
h_19=enthalpy(Water; T=T_19; P=P_19) "[kJ/kg]"
h_29=enthalpy(Air; T=T_29) "[kJ/kg]"
h_30=enthalpy(Air; T=T_30) "[kJ/kg]"
m_dot_18=m_dot_17"[kg/s]"
T_29=T_0 "[K]"
P_29=P_0 "[kPa]"

"EXERGY - SECOND LAW ANALYSIS"

e_19=(h_19-h_0_WATER)-T_0*(s_19-s_0_WATER) "[kJ/kg]" "exergy of water at state 19"
s_19=entropy(Water; T=T_19; P=P_19) "[kJ/kgK]"
e_29=(h_29-h_0_AIR)-T_0*(s_29-s_0_AIR) "[kJ/kg]" "exergy of air at state 29"
s_29=entropy(Air; T=T_29; P=P_29) "[kJ/kgK]"
e_30=(h_30-h_0_AIR)-T_0*(s_30-s_0_AIR) "[kJ/kg]" "exergy of air at state 30"
s_30=entropy(Air; T=T_30; P=P_30) "[kJ/kgK]"
epsilon_condenser=(m_dot_29*(e_30-e_29))/(m_dot_18*(e_18-e_19)) "exergetic - second law efficiency of Condenser"

"Data"
T_19=323 "[K]" : P_19=460 "[kPa]" : T_30=324 "[K]" : P_30=99,5 "[kPa]" : m_dot_29=0,3 "[kg/s]"

"11. FEED WATER TANK (FWT)- STATE 19 (RETURN WATER) + STATE 16 (RETURN WATER) + STATE 10 (FEED WATER TO WATER HEATER)"

m_dot_10=m_dot_16+m_dot_19+m_dot_fw "[kg/s]"
m_dot_16=m_dot_15 "[kg/s]"
m_dot_19=m_dot_18 "[kg/s]"

"Data"
m_dot_fw=0,55 "[kg/s]"

"12. PUMP (from FWT to WATER HEATER)"

"Assumptions : 1. This is a steady flow process since there is no change with time at any point. 2. Heat transfer is negligible. 3. Frictional heating effects are disregarded. 4. There is no change in temperature across the pump. 5. There is no change in pipe diameters at the inlet and outlet of the pump. 6. There is no level difference between inlet and exit states of the pump. 7. Liquids can be treated as incompressible substances, which is v=constant."

"ENERGY - FIRST LAW ANALYSIS"

v_10=volume(Water; T=T_10; P=P_10)
W_dot_pump_in=m_dot_11*v_10*(P_11-P_10) "[kW]" "Power input to the PUMP 1 which is the pump between FWT and WATER HEATER"
m_dot_11=m_dot_10
T_10=T_11 "[K]"

"EXERGY - SECOND LAW ANALYSIS"

e_10=(h_10-h_0_WATER)-T_0*(s_10-s_0_WATER)"[kJ/kg]" "exergy of the water at state 10"
e_11=(h_11-h_0_WATER)-T_0*(s_11-s_0_WATER) "[kJ/kg]" "exergy of water at state 11"
E_dot_pump=m_dot_10*(e_11-e_10) "[kW]" "Minimum possible power consumed by the compressor"
h_10=enthalpy(Water; T=T_10; P=P_10) "[kJ/kg]"
s_10=entropy(Water; T=T_10; P=P_10) "[kJ/kgK]"
h_11=enthalpy(Water; T=T_11; P=P_11) "[kJ/kg]"
s_11=entropy(Water; T=T_11; P=P_11) "[kJ/kgK]"
epsilon_pump=E_dot_pump/W_dot_pump_in "Exergetic - second law efficiency of the pump"

"Data"
P_10=240 "[kPa]" : P_11=800 "[kPa]" : T_10=351 "[K]"

"13. ELECTRIC WATER HEATER"

Q_dot_electricwaterheater=m_dot_11*(h_12-h_11) "[kW]"

2. Thermoeconomic Cost Analysis of Sanko DEPC Plant by EES

"EXERGOECONOMIC ANALYSIS OF SANKO DIESEL ENGINE POWERED COGENERATION (DEPC) BY SPECO METHODOLOGY"

"1. COMPRESSOR EXERGETIC COST RATE BALANCE"

```
C_dot_W_COMP+Z_dot_COMP=C_dot_2-C_dot_1
c_W_COMP=278*(C_dot_W_COMP/E_dot_W_COMP)
c_2=278*(C_dot_2/E_dot_2)

"Auxiliary Relations"
C_dot_1=0

"Data"
Z_dot_COMP=80,3 "[$/h]" : E_dot_1=0 "[kW]" : E_dot_2=6536 "[kW]" :
E_dot_W_COMP=7920 "[kW]"


"2. INTERCOOLER EXERGETIC COST RATE BALANCE"

(C_dot_20-C_dot_21)+(C_dot_23-C_dot_24)+Z_dot_IC=C_dot_3-C_dot_2
c_3=278*C_dot_3/E_dot_3
c_20=278*C_dot_20/E_dot_20
c_21=278*C_dot_21/E_dot_21
c_23=278*C_dot_23/E_dot_23
c_24=278*C_dot_24/E_dot_24

"Auxiliary Relations"
(C_dot_21-C_dot_20)/(E_dot_21-E_dot_20)=(C_dot_24-C_dot_23)/(E_dot_24-E_dot_23)
c_2=c_3
c_20=c_21
c_23=c_24

"Data"
Z_dot_IC=28,3 "[$/h]": E_dot_3=4239,4 "[kW]" : E_dot_20=1008,9 "[kW]" : E_dot_21=1473,3
"[kW]" : E_dot_23=97,5 "[kW]" : E_dot_24=238,5 "[kW]"


"3. LUBRICATION OIL COOLER EXERGETIC COST RATE BALANCE"

(C_dot_24-C_dot_25)+Z_dot_LOC=C_dot_28-C_dot_27
c_25=278*C_dot_25/E_dot_25
c_27=278*C_dot_27/E_dot_27
c_28=278*C_dot_28/E_dot_28

"Auxiliary Relations"
C_dot_28/E_dot_28=C_dot_27/E_dot_27
c_24=c_25
c_27=c_28


"Data"
Z_dot_LOC=9,4 "[$/h]": E_dot_25=312 "[kW]" : E_dot_27=423 "[kW]" : E_dot_28=162,6
"[kW]"


"4. DIESEL ENGINE EXERGETIC COST RATE BALANCE"

C_dot_3+C_dot_5+(C_dot_21-C_dot_22)+(C_dot_26-
C_dot_27)+Z_dot_DE=C_dot_6+C_dot_W_ELECTRIC
c_5=278*C_dot_5/E_dot_5
c_6=278*C_dot_6/E_dot_6
c_22=278*C_dot_22/E_dot_22
c_26=278*C_dot_26/E_dot_26
c_w_electric=278*C_dot_W_ELECTRIC/25320
```

```
"Auxiliary Relations"
C_dot_22/E_dot_22=C_dot_21/E_dot_21
C_dot_26/E_dot_26=C_dot_27/E_dot_27
c_5=c_6

"Data"
Z_dot_DE=510,2 "[$/h]" : E_dot_5=46,02 "[kW]" : E_dot_6=12250,2 "[kW]" :
E_dot_22=1892,1 "[kW]" : E_dot_26=199,8 "[kW]"


"5. TURBINE EXERGETIC COST RATE BALANCE"

C_dot_6-C_dot_7+Z_dot_TURBINE=C_dot_W_TURBINE
c_W_TURBINE=278*C_dot_W_TURBINE/E_dot_W_TURBINE
c_7=278*C_dot_7/E_dot_7
{c_W_COMP=c_W_TURBINE}

"Auxiliary Relations"
c_6=c_7

"Data"
Z_dot_TURBINE=80,3 "[$/h]" : E_dot_7=3081,3 "[kW]" : E_dot_W_TURBINE=8076 "[kW]"


"6. WASTE HEAT BOILER EXERGETIC COST RATE BALANCE"

C_dot_7-C_dot_8+Z_dot_WHB=C_dot_13-C_dot_12
c_8=278*C_dot_8/E_dot_8
c_12=278*C_dot_12/E_dot_12
c_13=278*C_dot_13/E_dot_13

"Auxiliary Relations"
c_7=c_8
C_dot_12/E_dot_12=C_dot_13/E_dot_13

"Data"
Z_dot_WHB=28,3 "[$/h]" : E_dot_8=1497 "[kW]" : E_dot_12=60 "[kW]" : E_dot_13=240,3
"[kW]"


"7. DeSOx UNIT EXERGETIC COST RATE BALANCE"

C_dot_9-C_dot_8+Z_dot_DeSOx=0
c_9=278*C_dot_9/E_dot_9

"Auxiliary Relations"
c_9=0

"Data"
Z_dot_DeSOx=47,2 "[$/h]" : E_dot_9=817 "[kW]"


"8. FUEL OIL DAY TANK EXERGETIC COST RATE BALANCE"

C_dot_15-C_dot_16+Z_dot_FDT=C_dot_4-C_dot_FO
c_4=278*C_dot_4/E_dot_4
c_15=278*C_dot_15/E_dot_15
c_16=278*C_dot_16/E_dot_16

"Auxiliary Relations"
```

```
C_dot_15/E_dot_15=C_dot_16/E_dot_16


"Data"
Z_dot_FDT=9,4 "[$/h]" :  E_dot_4=10,71 "[kW]" : E_dot_15=16,2 "[kW]" : E_dot_16=1,8
"[kW]"


"9. FUEL FORWARDING MODULE EXERGETIC COST RATE BALANCE"

C_dot_17-C_dot_18+Z_dot_FFM=C_dot_5-C_dot_4
c_17=278*C_dot_17/E_dot_17
c_18=278*C_dot_18/E_dot_18

"Auxiliary Relations"
C_dot_17/E_dot_17=C_dot_16/E_dot_16
C_dot_18/E_dot_18=C_dot_17/E_dot_17


"Data"
Z_dot_FFM=94,4 "[$/h]" : E_dot_17=48 "[kW]" : E_dot_18=3  "[kW]"


"10. CONDENSER EXERGETIC COST RATE BALANCE"

C_dot_29-C_dot_30+Z_dot_CON=C_dot_19-C_dot_18
c_19=278*C_dot_19/E_dot_19
c_19=c_18


"Auxiliary Relations"
C_dot_29=0
C_dot_19/E_dot_19=C_dot_18/E_dot_18
c_30=0

"Data"
Z_dot_CON=4,8 "[$/h]" : E_dot_19=1,5  "[kW]" :   E_dot_29=0 "[kW]" : E_dot_30=0,3 "[kW]"


"11. PUMP EXERGETIC COST RATE BALANCE"

C_dot_W_PUMP+Z_dot_PUMP=C_dot_11-C_dot_10
c_10=278*C_dot_10/E_dot_10
c_11=278*C_dot_11/E_dot_11
c_W_PUMP=278*C_dot_W_PUMP/E_dot_PUMP

"Auxiliary Relations"
C_dot_10/E_dot_10=C_dot_11/E_dot_11
c_10=c_11

"Data"
Z_dot_PUMP=4,8    "[$/h]"    :    E_dot_10=33    "[kW]"    :          E_dot_11=33,9    "[kW]"    :
E_dot_PUMP=1,29 "[kW]"
```

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WORK EXPERIENCE

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2002-present	Gaziantep University	Research Assistant
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1994-1996	ASTHERM Limited Company - Gaziantep	Owner and Partner

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PUBLICATIONS

P1: International Journal Papers

1. Abusoglu A., Kanoglu M. (2008), First and Second Law Analysis of Diesel Engine Powered Cogeneration Systems. *Energy Conversion and Management*, **49**, 2026-2031.
2. Abusoglu A., Kanoglu M. (2008), Exergetic and thermoeconomic analyses of diesel engine powered cogeneration: Part 1: Formulations. *Applied Thermal Engineering*. (in press)
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P2: Symposium Papers

1. Abusoglu A., Kanoglu M. (2008), Dizel motorlu kojenerasyon sistemlerinin termoekonomik analizi. *4. Ege Enerji Sempozyumu, 21-23 Mayıs*, Bildiriler Kitabı, 681-694.
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3. Abusoglu A., Kanoglu M. (2006), Thermodynamic analysis of diesel engine powered cogeneration systems. *3. Otomotiv Teknolojileri Kongresi, 26-28 Haziran*, Bildiriler Kitabı.
4. **Workshop:** Summer course on exergy and its applications, 16-17 June 2008, TUBITAK – MAM, Gebze.

HOBBIES

Amateur classical guitar and electro guitar player, Non-fiction books, Psychotherapy and NLP readings, Amateur Go Game (player), Rock and Roll and Classical Music (listener), Amateur Movie Scenario Writings.