Energy Systems Improvement based on Endogenous and Exogenous Exergy Destruction

vorgelegt von
M.Phil.
Solange Kelly
aus Trinidad und Tobago

Von der Fakultät III - Prozesswissenschaften
der Technischen Universität Berlin
zur Erlangung des akademischen Grades
Doktor der Ingenieurwissenschaften
- Dr.-Ing. -

genehmigte Dissertation

Promotionsausschuss:

Vorsitzender: Prof. Dr. rer. nat. Frank Behrendt
Berichter: Prof. Dr.-Ing. Georgios Tsatsaronis
Berichter: Prof. Dr. Tetyana Morozyuk

Tag der wissenschaftlichen Aussprache: 04.03.2008

Berlin, 2008
D 83

Foreword

This research work was carried out at the department of Energietechnik and Umweltschutz at the Technische Univerität Berlin. The main objective of the research is to split the exergy destruction in components into its endogenous and exogenous parts and to apply such a concept to the improvement of thermal systems.

The famous poet John Donne once said, "no man is an island, entire of itself". Truly, this work would not have been successfully completed without the input from many to whom I wish to express my heart felt gratitude.

I would like to thank Professor George Tsatsaronis for his encouragement, patience and guidance throughout the duration of this research work. His kindness shown to me will always be etched in my mind.

Professor Tatiana Morosuk was always willing to provide assistance. Her pleasant disposition and unending energy provided momentum to my process, for which I am thankful.

I would like to also thank my colleagues at the Energietechnik and Umvweltschutz for having helped to make my stay in Berlin a home away from home.

Many thanks to the Deutscher Akademischer Austauschdienst (DAAD) for funding this research work.

I would like to thank my husband, Sheldon, for his abundant support, encouragement, sacrifice and faithfulness and finally my Heavenly Father who taught me that the integrity of one's process in arriving at an ordained end is just as, or even more important than the achievement of the end itself.

Berlin, March 2008

Solange Kelly

Abstract

One of the roles of Exergoeconomics is to provide energy system designers and operators with the information, necessary for the improvement of energy systems. It employs both economic principles and exergy concepts particularly taking into account the values of individual components' exergy destruction: the thermodynamic loss due to irreversibilities within a system's component.

The total exergy destruction occurring in a component is not only due exclusively to the component (*endogenous exergy destruction)* but is also caused by the inefficiencies of the remaining system components (*exogenous exergy destruction*). Hence care must be taken in using the total exergy destruction of a component when making decisions to optimize the overall energy system.

The understanding of *Exogenous and Endogenous Exergy Destruction* for any given component can further assist the engineer in deciding whether a subsystem or a structural adjustment is required in the optimization of the entire energy system.

With emphasis placed on process performance (i.e. the mutual interdependencies of the components within the system) as oppose to the final output, exogenous and endogenous exergy destruction analysis guarantees that the quality of the output is improved without compromising the performance of individual components.

Additionally, only a part of the exergy destruction in a component can be avoided (*avoidable exergy destruction*) since a system component is also imposed by a number of constraints including physical, technological and economical. Knowledge of the Exogenous and Endogenous exergy destruction together with an understanding of the (*unavoidable and avoidable exergy destruction*) can provide a realistic measure of the potential for optimising any energy system.

The thesis deals with the development of a concept for splitting the exergy destruction and the costs associated with the system components. This concept is then applied to improve three energy conversion plants: a simple gas turbine process, a cogeneration and an externally-fired combined cycle power system and the results compared to the improvement of these said plants using a conventional exergoeconomic analysis.

Contents

List of Figures

List of Tables

Nomenclature

Latin

$\dot{C}$	total exergy cost rate, US$/h
c	specific exergy cost, US$/ MW
c_p	specific heat capacity, kJ/kg.K
$\dot{E}$	Exergy destruction rate, MW
$\dot{E}$	exergy rate, MW
e	specific exergy, MW/kg
$\dot{m}$	mass flow rate, kg/s
p	pressure, Pa
$\dot{Q}$	heat rate, MW
R	gas constant, kJ/kg.K
T	temperature, K
$\dot{W}$	power, MW
w	specific work, kJ/kg
$\dot{Z}$	component cost flow rate, US$/h
Z	related to capital investment, US$

Abbreviations

AC	air compressor
CC	combustion chamber
CM	compressor
EV	evaporator
GT	expander
HE	heat exchanger
RS	real system
ST	steam turbine
TV	throttling valve

Greek

Δ	difference

η	isentropic efficiency, %
ε	exergetic efficiency, %
κ	specific heat ratio
λ	air fuel ratio

Superscripts

AV	avoidable
$\dot{E}$	exergy
EN	endogenous
EN,AV	endogenous avoidable
EN,UN	endogenous unavoidable
EX	exogenous
EX,AV	exogenous avoidable
EX,UN	exogenous unavoidable
ID	ideal
k	k-th component.
RS	real system
T	thermal mechanical
TH	theoretical
UN	unavoidable
MX	mexogenous

Subscripts

a	air
D	destruction
F	fuel (e.g. methane)
F	fuel (exergy)
fr	friction
g	combustion gases
k	k-th component
$others$	other components
P	product
s	isentropic
st	steam

1 Introduction

1.1 Need for Improving Energy Conversion Systems

The design of both efficient and cost effective energy conversion systems is an on-going challenge facing energy engineers. With the increasing need to reduce the impact of waste from these systems on the environment and an ever increasing global demand for energy, especially in developing countries, it is becoming extremely important to develop even more accurate and systematic approaches for improving the design of energy systems.

This thesis focuses on the evaluation and improvement of these said systems (i.e. the improvement of the total cost rate) through a more in depth understanding of the exergy being destroyed within the individual components of the systems. An engineering approach will be investigated for the purpose of splitting the exergy destruction in a component into its endogenous part or that part which is due totally to the irreversibilities of the component and its exogenous part which is due to the irreversibilities of the other components within the system. The system here encompasses both the design configuration of the components as well as the required operating conditions of the system, such as the required product output, operational pressure and required temperature settings. The results will then be applied to the exergoeconomic analysis with the aim of providing relevant additional information for energy system optimization.

1.2 Exergoeconomics: a tool for improving thermal plant processes

Exergoeconomics is a technique initiated since the 1930's and used for designing efficient energy conversion systems or the optimization of such systems. It combines the second law of thermodynamics with economics, in other words exergoeconomics combines exergy analysis and economic principles. Various exergoeconomic methodologies have been developed over the last 20 years. They include the Average Costing Method (AVCO), the Last In-First Out Method (LIFO) and the

Specific Exergy Costing (SPECO), the Exergetic Costing Method (EXCO), the Thermo-functional Analysis Method (TFA) and the Engineering Functional Analysis (EFA) [11–13].

With the combination of the understanding of both irreversibilities and economics, the cost of exergy destroyed in a plant's component becomes measurable. Such information would otherwise not be obtained with the use of a conventional energy analysis. Exergoeconomics, therefore, provides the plant designer or operator with information critical to the plant as costs due to thermodynamic inefficiencies are identified and evaluated and hence can be reduced, creating opportunities for the optimization of the system be it at the design phase or the operational phase.

1.3 Limitations of the current Exergoeconomic analysis

Exergy analysis is without a doubt a powerful tool for developing, evaluating and improving a thermal system, particularly when this analysis is part of an exergoeconomic analysis. However, the lack of a formal procedure in using the results obtained by an exergy analysis is one of the reasons for exergy analysis not being very popular among energy practitioners [11]. For example, the results of an exergy analysis can be used to determine the total exergy destruction within a component. However, the total exergy destruction occurring in a component is not only due exclusively to the component but is also caused by the inefficiencies within the remaining system components.

Hence, a formal procedure cannot be developed as long as the interactions among components of the overall system are not being taken properly into account.

1.4 Improving the current Exergoeconomic analysis

The aim of this research work is to develop a methodology for splitting the exergy destruction taking place in the components of energy conversion systems and to use the results for the purpose of enhancing the exergoeconomic analysis and evaluation of these systems. In Chapter 2, work relating to the subject area as well as current developments have been discussed. In Chapter 3, a new methodology developed by this author was presented. In addition, other approaches dealing with determining the interaction among components have been analyzed for both their accuracy and practicality for the splitting of exergy destruction into its endogenous and exogenous parts. In the proceeding chapters, the new methodology was applied to four thermal systems, including a refrigeration machine. The latter was done to highlight not only the accuracy of the proposed methodology but also to show the importance of splitting the exergy destruction in individual components to the improvement of thermal systems.

2 Literature Survey

2.1 Exergy Analysis

Exergy analysis is relevant in identifying and quantifying both the consumption of exergy used to drive a process(due to irreversibilities) and the exergy losses i.e. the transportation of exergy to the environment. These are the true inefficiencies and therefore can highlight the areas of improvement of a system. Exergy measures the material's true potential to cause change and the degree to which the material has been processed. Throughout the years such analysis have been extensively discussed and applied to a wide variety of thermal systems [1, 2, 9, 10, 19].

Three main considerations are addressed in an exergy analysis; the type of exergy, the exergy balance for each component and the exergy destruction for each component.

2.1.1 Types of Exergy

Two types of exergy are considered in this thesis: physical exergy, $\dot{E}^{PH}$, and chemical exergy, $\dot{E}^{CH}$.

1. Physical exergy, $\dot{E}^{PH}$
   The physical exergy of a system comprises of two forms of exergy: mechanical exergy, $\dot{E}^{M}$, which is dependent on the system pressure and thermal exergy $\dot{E}^{T}$, which is dependent on the system temperature. Physical exergy is defined by the expression:

$$\dot{E}^{PH} = \dot{E}^{T} + \dot{E}^{M}. \tag{2.1}$$

2. Chemical exergy, $\dot{E}^{CH}$
   The Chemical Exergy is equal to the maximum net work obtained when the pure substance or working fluid of a system existing at the environment state is brought into complete thermodynamic equilibrium with the environment of known chemical composition [2].

2.1.2 Exergy Balance

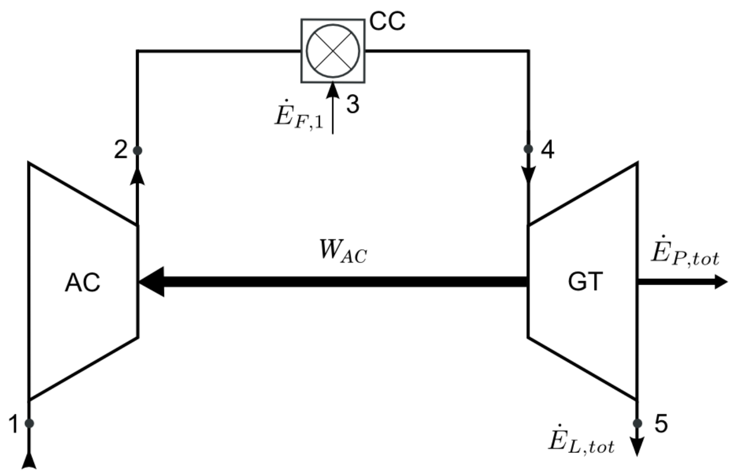


Figure 2.1: Simple gas turbine system.

The general exergy balance for any energy conversion plant, such as the simple gas turbine process shown in the Figure 2.1, can be defined as follows:

$$\dot{E}_{F,tot} - \dot{E}_{P,tot} - \dot{E}_{L,tot} = \dot{E}_{D,tot}. \tag{2.2}$$

where $\dot{E}_{F,tot} = \sum_{i=1}^{n} \dot{E}_{F,i}$; n being the number of fuel streams entering the system.

Equation (2.2) shows that the total amount of exergy destroyed, $\dot{E}_{D,tot}$, can be calculated based on all the incoming and outgoing exergy flows.

Exergy losses, $\dot{E}_{L,tot}$, should not be confused with exergy destruction. Exergy losses consist of exergy flowing to the surroundings, whereas exergy destruction indicates the loss of exergy inside the process boundaries due to irreversibilities.

An exergy balance formulated for the k-th component at steady state conditions can be written as:

$$\dot{E}_{F,k} = \dot{E}_{P,k} + \dot{E}_{D,k}. \tag{2.3}$$

Here it is assumed that the system boundaries used for all exergy balances are at the temperature T_0 of the reference environment, and, therefore, there are no exergy losses associated with the k-th component [13]. Exergy losses appear only at the level of the overall system.

The expression $\dot{E}_{P,k}/\dot{E}_{F,k}$ is defined as the exergetic efficiency of the k-th component, ε_k . The exergetic efficiency evaluates the true performance of the process within a component.

2.1.3 Exergy Destruction

All real processes are irreversible due to effects such as chemical reaction, heat transfer through a finite temperature difference, mixing of matter at different compositions or states, unrestrained expansion and friction [25]. An exergy analysis identifies the system components with the highest exergy destruction and the process that cause them. Efficiencies within a plant's component can then be improved by reducing the exergy being destroyed within the component. However, given present technical limitations, part of the exergy destruction and losses may be unavoidable, part may be due to the exergy destruction present in the other components within the thermal system , exogenous exergy destruction, and hence it may be worthwhile to improve the other components and not just the component with the highest exergy destruction.

It is therefore important to understand the genesis of the rate of exergy being destroyed in a component's process. Hence by splitting the exergy destruction within a component a more accurate solution concerning the improvement of the thermal system can be attained.

2.2 Splitting of Exergy Destruction

As stated before, the theory of splitting the exergy destruction allows for the further understanding of the exergy destruction values from an exergy analysis and hence improves the accuracy of the analysis, thereby facilitating the improvement of thermal systems. This research addresses four parts into which exergy destruction can be split and the implication of each part.

2.2.1 Unavoidable and Avoidable Exergy Destruction

At any given state of technological development, some exergy destruction within a system component will always be unavoidable due to physical and economic constraints [30].

Exergy destruction for the k-th component can therefore be further defined as

$$\dot{E}_{D,k} = \dot{E}^{AV}_{D,k} + \dot{E}^{UN}_{D,k}. \tag{2.4}$$

where $\dot{E}^{AV}_{D,k}$ represents the avoidable exergy destruction or that part of the exergy destruction that can be reduced whereas $\dot{E}^{UN}_{D,k}$ represents the unavoidable exergy destruction or that part of the exergy destruction which cannot be reduced.

The expression $(\dot{E}_D/\dot{E}_P)_k^{UN}$ is used to determine the unavoidable exergy destruction per unit of product exergy of component k. Based on work done in [6, 30], this expression is determined by selecting the best component possible in order to obtain the lowest exergy destruction rate that could be realized given the limitation of technology at the present time.

Hence for a similar component j of the same type as that of component k, in another system design, and with a value of the exergetic product, $\dot{E}_{Pj}$, the ratio $(\dot{E}_D/\dot{E}_P)_k^{UN}$ can be used to calculate the unavoidable exergy destruction in component j.

$$\dot{E}_{D,j}^{UN} = \dot{E}_{P,j}\left(\dot{E}_D/\dot{E}_P\right)_k^{UN}. \tag{2.5}$$

Equation (2.4) can then be used to determine $\dot{E}_{D,j}^{AV}$.

A modified expression for the exergetic efficiency was defined in [6, 30] as follows:

$$\varepsilon_k^{AV} = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k} - \dot{E}_{D,k}^{UN}} = 1 - \frac{\dot{E}_{D,k}^{AV}}{\dot{E}_{F,k} - \dot{E}_{D,k}^{UN}}. \tag{2.6}$$

2.2.2 Endogenous and Exogenous Exergy Destruction

The endogenous exergy destruction in the kth component, $\dot{E}_{D,k}^{EN}$, with exergetic efficiency(ε_k) operating in an energy conversion system is defined as that part of the entire exergy destruction within the component that is due only to the irreversibilities within the kth component when all remaining components operate in an ideal way.

The exogenous exergy destruction, $\dot{E}_{D,k}^{EX}$, is the remaining part of the entire exergy destruction in the kth component so that

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX}. \tag{2.7}$$

This understanding of endogenous and exogenous exergy destruction was illustrated by G. Tsatsaronis with use of the illustration shown in Figure 2.2 [23].

The total system consists of three components A, B and C. In this simple illustration, the product of one component is the fuel of the next component with the fuel of component A, being the fuel of the total system and the product from component B, $\dot{E}_{F,A}$, being the total product of the system $\dot{E}_{P,tot}$. In this analysis the total product of the system is kept constant.

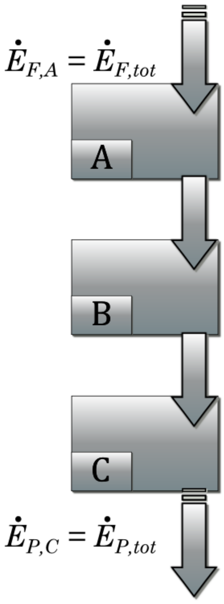


Figure 2.2: General case: step by step connection of elements.

For this case, there are no exergy losses at the level of the overall system. The exergy balance for this simple illustration can therefore be written as

$$\begin{aligned}
\dot{E}_{F,tot} &= \dot{E}_{P,tot} + \sum_k \dot{E}_{D,k} + \dot{E}_{L,tot} \\
&\quad where \\
\sum_k \dot{E}_{D,k} &= \dot{E}_{D,A} + \dot{E}_{D,B} + \dot{E}_{D,C} \\
&\quad and \\
\dot{E}_{L,tot} &= 0.
\end{aligned} \tag{2.8}$$

Hence, the exergy destruction for each component is defined as follows:

Component C

$$\begin{aligned}
\dot{E}_{D,C} &= \dot{E}_{P,C}\left(\frac{1}{\varepsilon_C} - 1\right) \\
&= \dot{E}_{P,tot}\left(\frac{1}{\varepsilon_C} - 1\right).
\end{aligned} \tag{2.9}$$

In Equation (2.9) it is clear that the exergy destruction within the component C is only dependent on the irreversibilities within the component itself, where $0 < \varepsilon_C < 1$. Hence the exergy destruction within this component is the endogenous exergy destruction of the component $\dot{E}_{D,C} = \dot{E}^{EN}_{D,C}$.

Component B

$$\dot{E}_{D,B} = \dot{E}_{P,tot}\varepsilon_C\left(\frac{1}{\varepsilon_B} - 1\right). \tag{2.10}$$

From Equation (2.10) it can be seen that the exergy destruction within component B is dependent on both the irreversibilities in component B and component C where $0 < \varepsilon_B < 1$. Therefore, there exist both endogenous and exogenous parts of the exergy destruction in this component. If the component C operated ideally i.e. $\varepsilon_C = 1$ or $\dot{E}_{D,C} = 0$, then the endogenous exergy of component B can be determined.

Component A

$$\dot{E}_{D,A} = \dot{E}_{P,tot}\varepsilon_C\varepsilon_B\left(\frac{1}{\varepsilon_A} - 1\right). \tag{2.11}$$

The exergy destruction in component A is dependent on the irreversibilties in component B and C as well as within component A, where $0 < \varepsilon_A < 1$. If the other components in the system, namely components C and B were to function ideally, i.e. $\varepsilon_B = \varepsilon_C = 1$ and $\dot{E}_{D,B} = \dot{E}_{D,C} = 0$ then the exergy destruction in component A can be determined.

When the exergy destruction within a component is dependent on the irreversibilities of the other components within the system, then its endogenous exergy destruction can be found when the other components operate ideally.

Until now work has not been done in the area of determining the endogenous and exogenous exergy destruction in power plants, though work has been done in splitting the exergy destruction rates in refrigeration machines, [16, 26, 28]. The method developed for determining the endogenous and exogenous exergy destruction in refrigeration systems is called the Thermodynamic method.

This method comprises of a methodological approach of introducing irreversibilities to an ideal thermodynamic cycle for the purpose of understanding the effect of irreversibilities in one component on the surrounding components, hence making the Thermodynamic method an appropriate tool in determining the endogenous exergy destruction within a component. Additional explanation and application of this method to refrigeration systems can be seen in the following references [16, 26, 28]. At the basis of the Thermodynamic method is the ability to define the ideal operation of a component, which can be a limitation for power plants, especially when considering components such as reactors (e.g. combustion chambers, fossil boilers).

2.3 Exergoeconomic Analysis and Evaluation

Exergoeconomic analysis is an effective tool used to evaluate the cost effectiveness of a thermal system, with the intent of improving the system. In other words exergoeconomic analysis assists in the understanding of the cost value associated with exergy destroyed in a thermal system and hence allows for the improvement of such system.

In an exergoeconomic analysis and evaluation, the cost rates associated with the rate of exergy destroyed in a component $\dot{C}_D$ together with the owning and operating costs of the components, $\dot{Z}$, within the system are used to calculate exergoeconomic factor f used in making key decisions concerning the improvement of the system. The exergoeconomic factor of the k-th component, f_k expresses the contribution of the capital cost to the sum of the capital cost and the cost of exergy destruction. It is calculated using the following equation:

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{C}_{D,k}}. \tag{2.12}$$

The concept of exergoeconomic analysis has been extensively discussed and applied to various thermal systems [2, 32], however, it will be also useful to know the costs $\dot{C}_D$ and $\dot{Z}$ associated with the various exergy destruction categories i.e. avoidable, unavoidable, endogenous and exogenous.

2.3.1 Splitting of the cost associated with exergy destruction

The avoidable $\dot{C}^{AV}_{D,k}$ and unavoidable $\dot{C}^{UN}_{D,k}$ cost rates are the cost rates associated with the avoidable and unavoidable exergy destruction respectively have been developed and applied to both a cogeneration and a combined power system [6, 30]. These rates along with the rates associated with the endogenous and exogenous exergy destruction parts are further discussed in Chapter 3.

2.3.2 Splitting of the investment costs , $\dot{Z}$

The unavoidable and avoidable investment cost is derived by first understanding the relationship between the investment cost and the exergy destruction (or exergetic efficiency) of a component. Such a relationship was developed in [30] and is shown in Figure 2.3. Here the operating and maintenance costs are assumed to be constant and independent of the selection of the design point for the component being considered. The shaded area illustrates the range of variation of the investment cost due to uncertainty and to multiple technical design solutions that might be available.

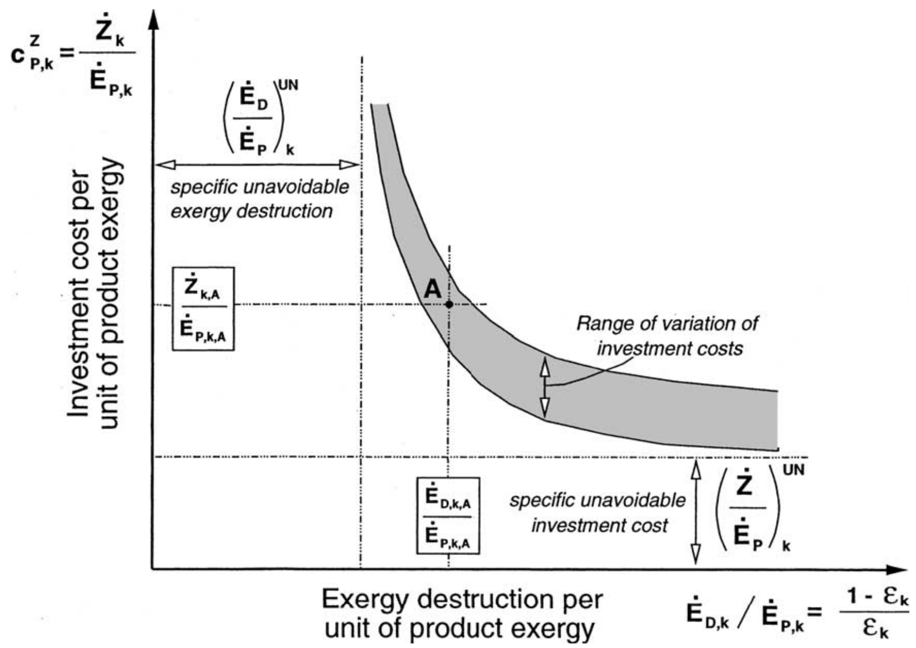


Figure 2.3: Expected relationship between investment cost and exergy destruction (or exergetic efficiency) for the k-th component of a thermal system. [30]

As this figure shows, the investment cost per unit of product exergy $\dot{Z}_k/\dot{E}_{P,k}$ increases with decreasing exergy destruction per unit of product exergy $\dot{E}_{D,k}/\dot{E}_{P,k}$ or with increasing efficiency. This is the normal cost behavior exhibited by most components. Technological limitations will only permit a minimum exergy destruction or maximum exergetic efficiency to be attained in the component and as stated before this minimum exergy destruction that can be attained is the unavoidable exergy destruction for the component. The figure also shows that the investment cost per unit of product exergy decreases as the exergy destruction increases, however the component will eventually attain the highest possible exergy destruction or the lowest possible exergetic efficiency. The investment cost per product would then reach its lowest possible value per product. This lowest possible value is referred to as the unavoidable investment cost per product $(\dot{Z}/\dot{E}_P)^{UN}_k$.

In [30] the concept of avoidable and unavoidable exergy destruction and costs was applied to an externally fired combined power plant. It was found that the recommendations with respect to the improvement of the cost effectiveness of the overall plant could be made with increased certainty when such splitting of both costs and exergy destruction is used. Like the cost of exergy destruction, the investment cost can be further split into endogenous and exogenous parts all of which will be addressed in the proceeding chapters.

3 Methodology: Determining Endogenous and Exogenous Exergy Destruction

3.1 The Engineering or "graph" Method

The Engineering or "graph" method was developed by this author as a means of splitting the exergy destruction in energy conversion systems. This method will now be systematically explained with the aid of a simple gas turbine system shown in Figure 3.1.

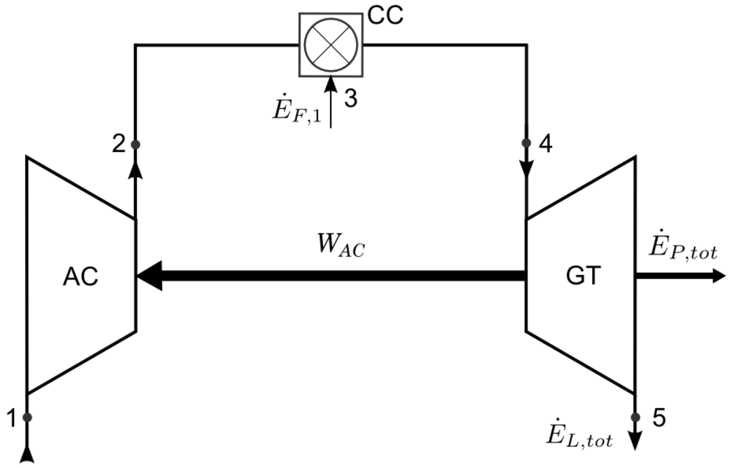


Figure 3.1: Simple gas turbine system.

For any ideal system producing a constant supply of product $\dot{E}_{P,tot}$, the exergy balance can be written as

$$\dot{E}^{ID}_{F,tot} = \dot{E}_{P,tot} + \dot{E}^{ID}_{L,tot}$$

or

$$\dot{E}_{P,tot} = \dot{E}^{ID}_{F,tot} - \dot{E}^{ID}_{L,tot}. \tag{3.1}$$

The superscript ID refers to the ideal operation of the overall system. If just one component (the k-th component) in the system is imperfect, additional exergetic resources $\Delta\dot{E}^{k}_{F,tot}$ need to be supplied and the loss increases by $\Delta\dot{E}^{k}_{L,tot}$. Equation (3.1) then becomes

$$\left(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{k}_{F,tot}\right) - \left(\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{k}_{L,tot}\right) = \dot{E}_{P,tot} + \dot{E}_{D,k}. \tag{3.2}$$

Since exergy destruction takes place in component k only, the value $\dot{E}_{D,k}$ is equivalent to the endogenous exergy destruction of the component k, i.e. in this particular case $\dot{E}_{D,k} = \dot{E}^{EN}_{D,k}$. So Equation (3.2) becomes:

$$\left(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{k}_{F,tot}\right) - \left(\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{k}_{L,tot}\right) = \dot{E}_{P,tot} + \dot{E}^{EN}_{D,k}. \tag{3.3}$$

When there is exergy destruction in every component, as in the case of a real system, (superscript RS), the following equation is obtained:

$$\left(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{RS}_{F,tot}\right) - \left(\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{RS}_{L,tot}\right) = \dot{E}_{P,tot} + \dot{E}_{D,k} + \dot{E}_{D,others}. \tag{3.4}$$

where $\Delta\dot{E}^{RS}_{F,tot}$ and $\Delta\dot{E}^{RS}_{L,tot}$ represent the increases in the exergy of fuel required and in the exergy loss, respectively, as a result of the exergy destructions in all components. As the other components within the system approach ideal operation, $\dot{E}_{D,others}$ tends to zero, and their respective exergetic efficiencies, ε, approaches 100%.

Considering the impact of this limit of $\dot{E}_{D,others}$ on the LHS of Equation (3.4), we get

$$\begin{aligned} &\lim_{\dot{E}_{D,others}\to 0} \left[\left(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{RS}_{F,tot}\right) - \left(\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{RS}_{L,tot}\right)\right] \to \left[\left(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{k}_{F,tot}\right) - \left(\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{k}_{L,tot}\right)\right] \\ &\qquad and \\ &\lim_{\dot{E}_{D,others}\to 0} \dot{E}_{D,k} \to \dot{E}^{EN}_{D,k}. \end{aligned} \tag{3.5}$$

Hence by plotting $(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{RS}_{F,tot}) - (\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{RS}_{L,tot}) - \dot{E}_{P,tot}$ vs. $\dot{E}_{D,others}$, the value of $\dot{E}_{D,k}$ at ε_k can be obtained at the intercept where $\dot{E}_{D,others} = 0$.

Since the endogenous exergy destruction is a function of the component's exergetic efficiency, the exergetic efficiency of the component must be kept constant while $\dot{E}_{D,others}$ is being varied. Straight lines are obtained when $\dot{E}_{D,others}$ is varied, as shown in Figure 3.2. The proof of this linear dependence is shown in Appendix A.

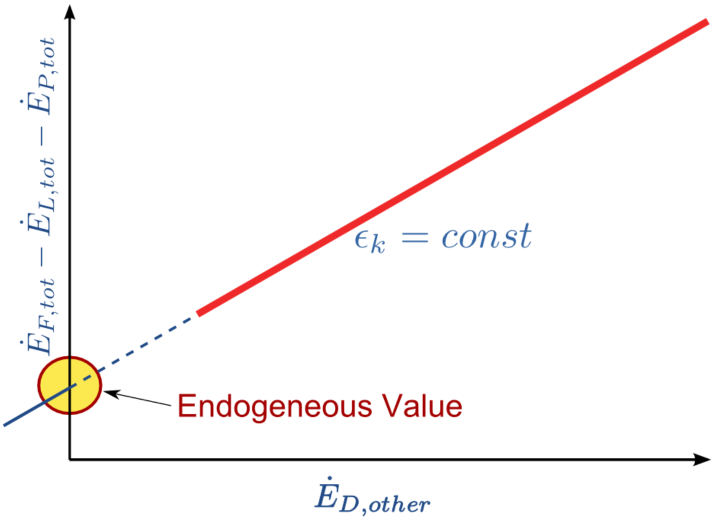


Figure 3.2: Attributes of the plot obtained from the Engineering Method.

The dotted line extension of the straight line indicates the values obtained if it were possible to reduce the exergy destruction in all components with the exception of the k-th component to zero. For some components such as a combustion chamber and a throttling valve, it is impossible to achieve ideal operations, because it is difficult to define an ideal process associated with such components. In addition, in some systems, it may be impossible for all the components to operate at ideal conditions and still maintain the required system product output.

An additional way of proving that the intercept of the plot $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot}$ vs. $\dot{E}_{D,others}$ with the vertical axis does indeed represent the endogenous exergy destruction within the k-th component can be developed with the use of partial derivatives.

Recall that the general equation for a real system in which exergy destruction is occurring in all the components is given as follows:

$$\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot} = \dot{E}_{D,k} + \dot{E}_{D,others}. \tag{3.6}$$

Equation (3.6) can be re-written in the form

$$\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX} + \dot{E}_{D,others}. \tag{3.7}$$

where $\dot{E}_{D,k}^{EN}$ and $\dot{E}_{D,k}^{EX}$ are the endogenous and exogenous exergy destruction parts of the total exergy destruction occurring in the k-th component, $\dot{E}_{D,k}$.

by differentiating Equation (3.7) with respect to $\dot{E}_{D,others}$, the following equation is achieved:

$$\frac{\delta(\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot})}{\delta \dot{E}_{D,others}} = \frac{\delta \dot{E}_{D,k}^{EN}}{\delta \dot{E}_{D,others}} + \frac{\delta \dot{E}_{D,k}^{EX}}{\delta \dot{E}_{D,others}} + 1. \tag{3.8}$$

It is important to note that $\delta \dot{E}_{D,others}$ implies that the exergy destruction in each component is changing.

Since $\dot{E}_{D,k}^{EN}$ is independent of $\dot{E}_{D,others}$ then $\frac{\delta \dot{E}_{D,k}}{\delta \dot{E}_{D,others}}$ Equation (3.8) becomes

$$\frac{\delta(\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot})}{\delta \dot{E}_{D,others}} = \frac{\delta \dot{E}_{D,k}^{EX}}{\delta \dot{E}_{D,others}} + 1. \tag{3.9}$$

The expression $\frac{\delta(\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot})}{\delta \dot{E}_{D,others}}$ represents the gradient, m, of the graph shown in figure 3.2. Hence (3.9) becomes

$$m = \frac{\delta \dot{E}_{D,k}^{EX}}{\delta \dot{E}_{D,others}} + 1. \tag{3.10}$$

where $1 < m$. Note when $m = 1$, $\frac{\delta \dot{E}_{D,k}^{EX}}{\delta \dot{E}_{D,others}} = 0$, implying that all the exergy destruction within the k-th component is endogenous.

Equation (3.10) can be re-written as

$$(m-1)\delta \dot{E}_{D,others} = \delta \dot{E}_{D,k}^{EX}. \tag{3.11}$$

by integrating (3.11), we get;

$$(m-1)\dot{E}_{D,others} = \dot{E}_{D,k}^{EX} + c. \tag{3.12}$$

when $\dot{E}_{D,others} = 0$, $\dot{E}_{D,k}^{EX} = 0$ hence (3.12) passes through the origin so the constant $c = 0$, therefore

$$(m-1)\dot{E}_{D,others} = \dot{E}_{D,k}^{EX}. \tag{3.13}$$

by substituting Equation (3.13) in Equation (3.6) the following equation is obtained

$$\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot} = \dot{E}_{D,k}^{EN} + m\dot{E}_{D,others}. \tag{3.14}$$

It is important that $\delta \dot{E}_{D,others}$ is not very small when evaluating $\dot{E}_{D,k}^{EN}$, as small changes can be comparable to already inherent errors in the simulation software used in evaluating energy system.

3.1.1 The endogenous curve

For any given component within a system its endogenous value can be determined at various exergetic efficiencies. A graph of the latter can be obtained as shown in Equation (3.6). The graph also

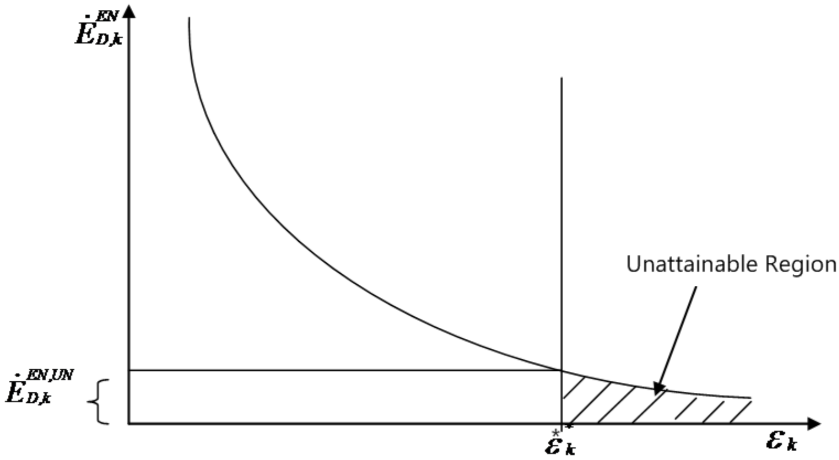


Figure 3.3: The endogenous curve.

shows the unavoidable endogenous exergy destruction and the corresponding exergetic efficiency as discussed in Chapter 2.

3.2 Application to various types of components

The operation of each component as well as the overall plant structure must be taken into consideration when developing procedures to determine the endogenous exergy destruction of a component. Based on the theory of the Engineering Method there are two questions that should be asked for each component of an energy conversion system:

- What must be considered in reducing the exergy destruction within a component when determining the endogenous exergy destruction of another component k within the given system?
- What must be considered when keeping the exergetic efficiency of a component constant during the determination of its endogenous exergy destruction within the given system?

In this section various components analyzed in this research work are examined in order to address the above mentioned questions. It is assumed that all components function adiabatically.

3.2.1 A coupled air compressor and expander

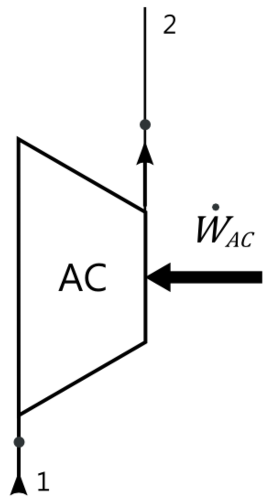


Figure 3.4: Schematic diagram of an Air Compressor.

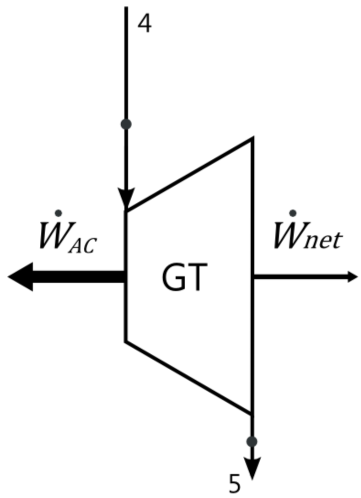


Figure 3.5: Schematic diagram of an Expander.

Considering Figure 3.4 and 3.5, the air compressor (AC) is driven by the power supplied by the expander (GT). The mass flow rate of the working fluids flowing through the air compressor and the expander are $\dot{m}_a$ and $\dot{m}_g$ respectively. The constant net power output required is $\dot{W}_{net}$ where

$$\begin{aligned} \dot{W}_{net} &= \dot{W}_{GT} - \dot{W}_{AC} \\ &= \dot{m}_g w_{GT} - \dot{m}_a w_{AC}. \end{aligned} \tag{3.15}$$

where w_{GT} and w_{AC} represent the specific work for the expander and air compressor respectively.

In general, $\dot{m}_g = \dot{m}_a + \dot{m}_f$ where $\dot{m}_f$ is the fuel supplied to the combustion chamber.

$$\dot{m}_a = \frac{\dot{W}_{net} - \dot{m}_f w_{GT}}{w_{GT} - w_{AC}} \tag{3.16}$$

where

$$w_{AC} = \eta_{AC} \left(\frac{\kappa_a}{\kappa_a - 1} \right) R_a T_1 \left[r_{p,AC}^{\kappa_a - 1/\kappa_a} \right] = h_2 - h_1 \tag{3.17}$$

and

$$w_{GT} = \eta_{GT} \left(\frac{\kappa_g}{\kappa_g - 1} \right) R_g T_4 \left[r_{p,GT}^{\kappa_g - 1/\kappa_g} \right] = h_4 - h_5. \tag{3.18}$$

η and r_p represent the isentropic efficiency and pressure ratio of the components and κ and R are the specific heat ratio and the gas constant respectively of the working fluids.

The exergy destruction in the air compressor is defined as follows:

$$\dot{E}_{D,AC} = \dot{m}_a w_{AC}(1 - \varepsilon_{AC}). \tag{3.19}$$

Substituting Equation (3.16) into Equation (3.19) , the expression for $\dot{E}_{D,AC}$ becomes

$$\dot{E}_{D,AC} = \frac{\dot{W}_{net} - m_f w_{GT}}{w_{GT} - w_{AC}} w_{AC}(1 - \varepsilon_{AC}). \tag{3.20}$$

From Equation (3.20) it is clear that the exergy destruction within the air compressor depends on its specific work w_{AC} as well as the specific work of the expander w_{GT}. Since w_{AC} or rather η_{AC} and $r_{p,AC}$ are parameters associated with the air compressor, they must be held constant during the analysis of this component. The exergy destruction within the expander approaches zero as w_{GT} approaches the specific isentropic work of the expander $w_{s,GT}$. It is important to note that an exergy destruction value of zero can be attained for various values of $w_{s,GT}$. For this reason $w_{s,GT}$ must be specified when determining the endogenous exergy destruction in the air compressor.

The exergetic efficiency of the air compressor is defined as follows:

$$\varepsilon_{AC} = \frac{w_{AC} - T_0(s_2 - s_1)}{w_{AC}}. \tag{3.21}$$

Now w_{AC} is already specified when examining this component, in addition, referring to Equation (3.21), it is clear that the expression $(s_2 - s_1)$ must also remain constant so that a constant exergetic efficiency ε_{AC} can be maintained during the analysis of this component. The exergy destruction for the expander is defined as follows:

$$\dot{E}_{D,GT} = \left(\dot{m}_a + \dot{m}_f\right) w_{GT} \left(\frac{1}{\varepsilon_{GT}} - 1\right). \tag{3.22}$$

Substituting for $\dot{m}_a$, the expression for $\dot{E}_{D,GT}$ becomes

$$\dot{E}_{D,GT} = \left(\frac{\dot{W}_{net} - \dot{m}_f w_{AC}}{w_{GT} - w_{AC}}\right) w_{GT} \left(\frac{1}{\varepsilon_{GT}} - 1\right). \tag{3.23}$$

Like the air compressor the exergy destruction in the expander depends on its specific work and the specific work of the air compressor. The endogenous exergy destruction value of this component is achieved when all other components within the system, namely the air compressor and the combustion chamber, have an exergy destruction value of zero. Like the expander, when the exergy destruction in the air compressor is zero, the value of its specific work w_{AC} will be equal to the value of its specific isentropic work $w_{s,AC}$. As stated before, the value of $w_{s,AC}$ can vary with the component having no exergy destruction taking place within it. Hence when evaluating the endogenous exergy destruction of the expander, it is important that $w_{s,AC}$ be kept constant as well as the specific work of the expander. The exergetic efficiency of the expander is defined as follows:

$$\varepsilon_{GT} = \frac{w_{GT}}{w_{GT} - T_0(s_4 - s_5)}. \tag{3.24}$$

Like the air compressor, the expression $(s_4 - s_5)$ must remain constant so that a constant exergetic efficiency can be maintained during the analysis of this component.

3.2.2 Combustion Chamber

So far the importance of the specific isentropic work of the air compressor and the expander in determining their endogenous exergy destruction has been discussed. The next component to be investigated is the combustion chamber. The exergy destruction in the combustion chamber is largely due to the chemical reaction taking place during the combustion process. However, other significant contributors to its exergy destruction include the initial mixing of the air and fuel at different temperatures and the mixing of the excess air and the gas formed at the end of the combustion process. Equation (3.25) provides a definition for the exergy destruction taking place within the combustion chamber shown in Figure 3.1:

$$\dot{E}_{D,CC} = \dot{E}_3 - (\dot{E}_4 - \dot{E}_2)$$

or

$$\dot{E}_{D,CC} = \dot{m}_f e_f - ((\dot{m}_a + \dot{m}_f)e_4 - \dot{m}_a e_2). \tag{3.25}$$

where e_f is the specific exergy of the fuel used in the combustion process and e_2 and e_4 are the specific exergy at the inlet and outlet of the combustion chamber respectively.

For this component the air-fuel ratio, λ, will be introduced where $\lambda = \dot{m}_a/\dot{m}_f$. So Equation (3.25) becomes

$$\dot{E}_{D,CC} = \dot{m}_f e_f - (\dot{m}_f(\lambda + 1)e_4 - \lambda \dot{m}_f e_2). \tag{3.26}$$

In determining the endogenous exergy destruction in the combustion chamber, $\dot{E}_{D,CC}$, the variables

$\dot{m}_f$, λ, e_4 and e_2 are evaluated at the point when all other components within the system are operating with no exergy destruction taking place within them.

In reducing $\dot{E}_{D,CC}$ in order to determine the endogenous exergy destruction of another component within the given system, the following should be noted:

- The variable T_2 is usually limited by the preceding component. As in the case of the simple cycle shown in Equation (3.25), if the air compressor is being examined then e_2 is fixed, e_2 will also be fixed when the air compressor is operating under ideal conditions. In such a case, $\dot{E}_{D,CC}$ can be reduced by increasing e_4. The latter can be done by increasing T_4 (which will result in a reduction of λ) or, if an isochoric combustion chamber is being used in the design, T_4 can remain constant while the pressure at which the combustion process takes place is varied. In this case, the increase in e_4 will be largely attributed to the reduction of the entropy generation taking place during the process.
- In the case where the variables T_4 and p_4 are limited by the requirements of the design of the proceeding component or the fact that the proceeding component is the component being examined then based on Equation (3.26), reducing e_2 may seem feasible. Increasing the isentropic efficiency of the air compressor leads to a reduction of e_2 but this is also accompanied by an increase in the exergy destruction in the combustion chamber and hence the engineering method cannot be applied.
- When it is not possible to vary the inlet and outlet temperatures and pressures of the combustion process due to design requirements as in the case of the simple cycle shown in Figure 3.1, a *reversible adiabatic heater* will be used to assist in determining the endogenous exergy destruction of the other components in the system.

The Reversible Adiabatic Heater

As mentioned previously, it is not always possible to reduce the exergy destruction in a combustion chamber by increasing its outlet temperature. In this research work, the use of a reversible adiabatic heater, RAH, was proposed.

The heater is applied in series with the combustion chamber, see Figure 3.6. The exergy balance for Figure 3.6 can be written as follows:

$$\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot} = \dot{E}_{D,AC} + \dot{E}_{D,CC} + \dot{E}_{D,GT}. \tag{3.27}$$

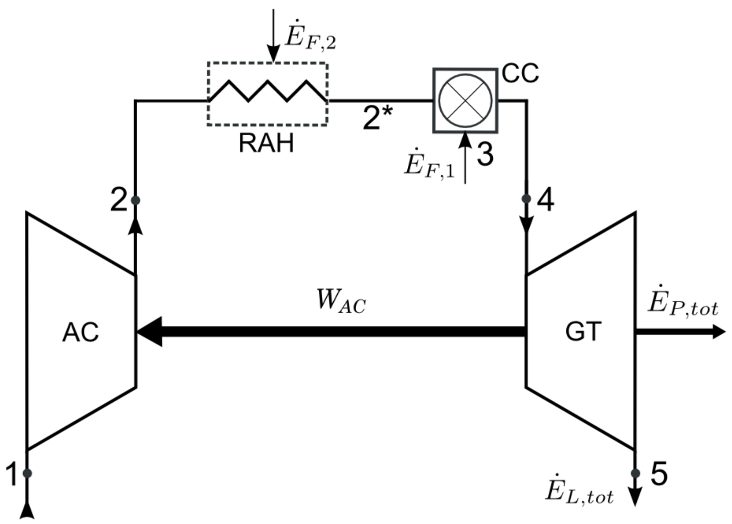


Figure 3.6: Simple gas turbine with RAH.

note that $\dot{E}_{D,RAH} = 0$. By expanding Equation (3.27), the following equation is obtained

$$\underbrace{(\dot{E}_{2^*} - \dot{E}_2)}_{\text{additional fuel due to the RAH, } \dot{E}_{F,2}} + \dot{E}_{F,1} - \dot{E}_{L,tot} - \dot{E}_{P,tot} = \dot{E}_{D,AC} + \underbrace{\dot{E}_{F,1} - (\dot{E}_4 - \dot{E}_{2^*})}_{\dot{E}_{D,CC}} + \dot{E}_{D,GT}$$

or

$$\underbrace{(\dot{E}_{2^*} - \dot{E}_2)}_{\text{additional fuel due to the RAH}} + \dot{E}_{F,1} - \dot{E}_{L,tot} - \dot{E}_{P,tot} = \dot{E}_{D,AC} + \underbrace{\dot{E}_{F,1} - (\dot{E}_4 - \dot{E}_2) + (\dot{E}_{2^*} - \dot{E}_2)}_{\dot{E}_{D,CC}} + \dot{E}_{D,GT}. \tag{3.28}$$

From Equation (3.28) we see that the introduced expression due to the additional fuel, $(\dot{E}_{2^*} - \dot{E}_2)$, on the LHS of the equation also appears on the RHS of the equation. In this way the exergy of the fuel, $\dot{E}_{F1}$, is varied and the additional fuel, $(\dot{E}_{2^*} - \dot{E}_2)$, is compensated for when the endogenous exergy destruction of the air compressor and the expander is being determined. The application of this method will be shown in the subsequent chapters.

The following example shown in Figure 3.7 was used to demonstrate the legitimacy of introducing a RAH in a system, in that, it does not affect the endogenous exergy destruction of the component being examined. In this example, the endogenous exergy destruction of the air compressor (AC) is being determined. A RAH is not really required since when the temperature of the exiting cold stream of the air preheater (APH) increases the exergy destruction in both the APH and the combustion chamber reduces simultaneously, i.e. there is no conflict. T_4 must be held constant. The expander can be set to ideal operation and the engineering method can then be used to determine the endogenous exergy destruction of the AC.

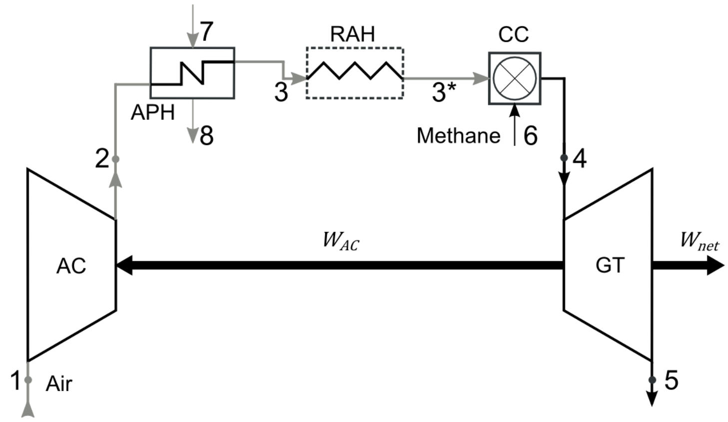


Figure 3.7: Power system used in verifying the use of the RAH.

A RAH is then introduced in the system in series with the combustion chamber to help reduce the exergy destruction in the combustion chamber. In both cases the endogenous exergy destruction rate obtained for the air compressor was the same, 3.67MW.

The results show that there is a negligible net effect in using the RAH to assist in reducing the exergy destruction of component within the system. Again it is important to state that when examining the component which the RAH assists, that the RAH be removed. Hence in this case the RAH cannot be used when evaluating the endogenous exergy destruction rate in the combustion chamber.

3.2.3 Expander (Uncoupled e.g. steam turbine)

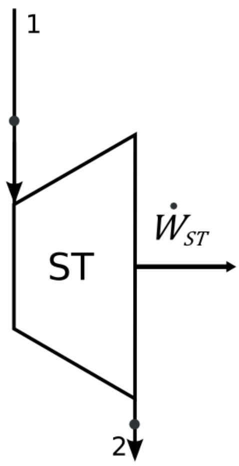


Figure 3.8: Schematic diagram of a Steam Turbine.

For the case where the expander is not coupled with an air compressor such as a steam turbine (see

Figure 3.8), the exergy destruction can be defined as follows:

$$\dot{E}_{D,ST} = \dot{m}_w w_{ST} \left(\frac{1}{\varepsilon_{ST}} - 1 \right). \tag{3.29}$$

Again it is clear that $\dot{E}_{D,ST}$ approaches zero when w_{ST} approaches its isentropic value, $w_{s,ST}$. Hence $w_{s,ST}$ must be specified in reducing the exergy destruction in this component.

The exergetic efficiency of this component is dependent on its specific work as seen from Equation (3.30). Hence when this component is being analyzed, i.e. when $\dot{E}^{EN}_{D,ST}$ is being determined, w_{ST} must be kept constant, additionally so must the entropy change in the component $(s_2 - s_1)$.

$$\varepsilon_{ST} = \frac{w_{ST}}{w_{ST} - T_0(s_2 - s_1)}. \tag{3.30}$$

3.2.4 Heat Exchanger

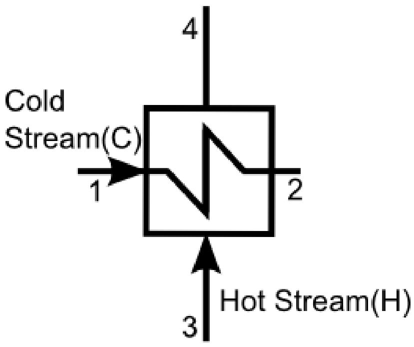


Figure 3.9: Schematic diagram of a heat exchanger.

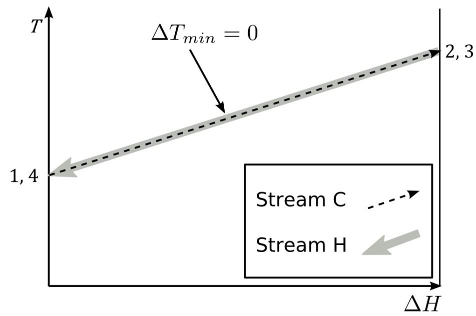


Figure 3.10: T-Q profile of an ideal heat exchanger.

The exergy destruction taking place in the heat exchanger, shown in Figure 3.9, the purpose of which is to supply heat to the cold stream, is given as:

$$\dot{E}_{D,HE} = (\dot{E}_2 - \dot{E}_1) \left(\frac{1}{\varepsilon_{HE}} - 1 \right) \tag{3.31}$$

or

$$\dot{E}_{D,HE} = (\dot{E}_3 - \dot{E}_4) \left(1 - \varepsilon_{HE} \right). \tag{3.32}$$

This exergy destruction $\dot{E}_{D,HE}$ is primarily due to heat transfer and friction and can also be written as:

$$\dot{E}_{D,HE} = T_0\dot{Q}\left(\frac{T_{H,a} - T_{C,a}}{T_{H,a}T_{C,a}}\right) + \dot{E}_{D,fr,C} + \dot{E}_{D,fr,H}. \tag{3.33}$$

where $T_{H,a}$ and $T_{C,a}$ represent the average temperatures of the hot and cold streams respectively. $\dot{E}_{D,fr}$ is the exergy destruction due to friction and is defined by the following equation:

$$\dot{E}_{D,fr} = \frac{-T_0\dot{m}\int_i^e v dp}{T_a}. \tag{3.34}$$

where v and dp represent the specific volume of and the pressure drop across the heat exchanger [2].

Equation (3.34) shows that $\dot{E}_{D,fr}$ is negligible, when dp along each stream is zero.

The $T - \Delta H$ profile in Figure 3.10 shows the temperature profiles of the working fluids when the operation is ideal. The figure shows that when ΔT_{min} , the minimum temperature difference of the streams is zero and the heat capacities of each stream (which determines the stream profiles) are equal, i.e $\dot{C}_{p,H} = \dot{C}_{p,C}$ the component operates ideally.

It is important to note that temperature value at each state point is sometimes limited by the demands of the system design on the heat exchanger and the equality of heat capacities between the two streams is not a normal occurrence in the operation of the heat exchanger (i.e. $\dot{C}_{p,H} \neq \dot{C}_{p,C}$).

The exergetic efficiency of the heat exchanger is defined as follows:

$$\varepsilon_{HE} = \frac{\dot{E}_2 - \dot{E}_1}{\dot{E}_3 - \dot{E}_4} \tag{3.35}$$

or

$$\varepsilon_{HE} = \frac{\dot{Q} - T_0\Delta\dot{S}_C}{\dot{Q} - T_0\Delta\dot{S}_H}. \tag{3.36}$$

When $E_{D,HE}^{EN}$ is being determined, ε_{HE} must be held constant.

3.3 Additional guidelines in plotting the graph , $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot}$ vs. $\dot{E}_{D,others}$

The following are some additional guidelines employed when obtaining points for the engineering method:

- Before reducing the exergy destruction in the other components, set the pressure drops in these components to zero with the exception of the component under study.
- The exergy destruction in the other components must be reduced in such a way that at the point where $\dot{E}_{D,others} = 0$, the exergy destruction in each individual component $\dot{E}_{D,1}, \cdots, \dot{E}_{D,n-1}$ (where n is the number of components in the system) must all be zero. This can be achieved by selecting the number of points one desires to plot and then dividing this number by the starting exergy destruction value in each component. The result will then indicate the rate at which the exergy destruction in each component must be reduced.
- When the system is large, it is better to concentrate on reducing the exergy destruction in the components with the highest exergy destruction rates.
- The reversible adiabatic heater should only be used when conflicts arise in reducing the exergy destruction of two adjacent components. Here the word conflict implies that the reduction in the exergy destruction in one component will result in the increase in exergy destruction in the adjacent component.

3.4 Other approaches considered

Other approaches were considered for splitting the exergy destruction into its endogenous and exogenous parts. One of these approaches include the application of a symbolic mathematical method proposed by Valero and Torres in [33]. This method was developed to show the effects of the connection of components on the thermal system behavior and the interaction among components. Such an approach was applied to a simple gas turbine system and discussed in Chapter 8.

At the heart of the problem for improving thermal systems with the use of endogenous and exogenous exergy destruction is defining an ideal reactor (e.g. combustion chambers, fossil boilers etc.). Other proposals therefore include the formulation of a definition for an ideal reactor. These definitions usually assume that the energy balance or the mass balance in the reactors can be ignored, which compromises the integrity of the method to produce accurate results. Examples of such methods are also shown in Chapter 8 and the advantages and disadvantages of each method are discussed.

3.5 Determining the Avoidable and Unavoidable Exergy Destruction

The avoidable and unavoidable exergy destruction for each component used in this study was calculated based on work done in [6, 30].

The unavoidable exergy destruction of each component $\dot{E}^{UN}_{D,k}$, which as previously stated cannot be reduced due to technological and process limitations, was first evaluated. Equation (2.4), $\dot{E}_{D,k} = \dot{E}^{AV}_{D,k} + \dot{E}^{UN}_{D,k}$, was then used to determine the avoidable part of the exergy destruction.

3.5.1 Evaluating $\dot{E}^{UN}_{D,k}$

In determining this value the "best" available component possible was selected based on assumptions made. These assumptions include determining the most important thermodynamic parameters of the k-th component, which are essential to reducing the exergy destruction within the component to a minimum value. Such as the isentropic efficiency which is key to reducing the exergy destruction within an air compressor or an expander. The ratio $(\dot{E}_D/\dot{E}_P)^{UN}_k$ is then calculated and used to determine $\dot{E}^{UN}_{D,k,A}$, the unavoidable exergy destruction of the same component operating in a system, A, with an exergetic product, $\dot{E}_{P,k,A}$, where from Equation (2.5)

$$\dot{E}^{UN}_{D,k,A} = \dot{E}_{P,k,A} \left(\frac{\dot{E}_D}{\dot{E}_P} \right)^{UN} . \tag{3.37}$$

In subsection 3.5.2, the methodology used in calculating $(\dot{E}_D/\dot{E}_P)^{UN}_k$ for each component used in this thesis will be examined.

3.5.2 Air compressor, expander and steam turbine

The largest technically achievable values of the pressure ratio and the isentropic efficiency were selected. These assumptions made for each cycle are shown in the subsequent chapters.

3.5.3 Combustion chamber and fossil boiler

As mentioned before, chemical reaction is the most significant source of exergy destruction in a combustion chamber. For evaluating the unavoidable exergy destruction of this component a high combustion reactant temperature at the inlet and the lowest technically meaningful value of the air-fuel ratio was selected.

3.5.4 Heat exchanger

For calculating the unavoidable exergy destruction in heat exchangers, the smallest technically attainable value for the minimum temperature difference was used while the heat was being transferred at a high temperature level.

3.6 Analyzing the various parts of the exergy destruction

After having split the total exergy destruction occurring in a component into its four categories, namely endogenous, exogenous, avoidable and unavoidable parts, the task left to be done will be to evaluate how the different categories of the exergy destruction can be combined and used to provide meaningful information. Table 3.1 lists the four categories obtained and how the information within each category can be used in the iterative exergoeconomic optimization.

3.6.1 Evaluating each categorized segment in Table 3.1

There are four categorized segments for exergy destruction obtained

- endogenous-avoidable, $\dot{E}_{D,k}^{EN,AV}$
- endogenous-unavoidable, $\dot{E}_{D,k}^{EN,UN}$
- exogenous-avoidable, $\dot{E}_{D,k}^{EX,AV}$
- exogenous- unavoidable, $\dot{E}_{D,k}^{EX,UN}$.

Only having information of the total values of each of the four categories is not sufficient to evaluate the values of each segment. Additional information about any one of the segments is required.

In this research work the endogenous-unavoidable exergy destruction was evaluated. This is determine by first finding the value of ε_k^* which corresponds to the ratio $(\dot{E}_D/\dot{E}_P)_k^{UN}$. In general, ε_k is defined as follows:

$$\varepsilon_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} = \frac{\dot{E}_{P,k}}{\dot{E}_{P,k} + \dot{E}_{D,k}} = \frac{1}{1 + (\dot{E}_{D,k}/\dot{E}_{P,k})}. \tag{3.38}$$

Hence ε_k^*, which is the maximum exergetic efficiency obtained for the component when it is operating with a minimum exergy destruction rate, was defined as follows:

$$\varepsilon_k^* = \frac{1}{1 + (\dot{E}_D/\dot{E}_P)_k^{UN}}. \tag{3.39}$$

Using this exergetic value and the endogenous-exergetic efficiency curve (see Figure 3.3), the corresponding endogenous exergy destruction value which is also the endogenous-unavoidable exergy destruction value can be found. The values of the exergy destruction in all the other segments can then be calculated.

Table 3.1: The concepts of endogenous-exogenous and avoidable-unavoidable exergy destruction being applied to the k-th component of an energy conversion system.

	Endogenous	Exogenous
Avoidable	can be reduced through an improvement of the efficiency of the k-th component	can be reduced by a structural optimization of the overall system or by improving the efficiency of the remaining components
Unavoidable	cannot be reduced because of technical and process limitations for the k-th component	cannot be reduced because of technical and or process limitations in other components of the overall system for the given structure

3.7 Splitting the cost rates (stream and the investment cost rates)

As additional information is gained from the splitting of the exergy destruction within a component so too can additional useful information be gained by the splitting of the costs associated with each stream within the system and the owning and operating costs of each component.

3.7.1 Avoidable and Unavoidable cost

The concept of the avoidable and unavoidable exergy destruction was discussed in Chapter 2. The cost rates associated with the avoidable and unavoidable exergy destruction are further defined by Equations (3.40) and (3.41) respectively,

$$\dot{C}_{D,k}^{AV} = c_{F,k}\dot{E}_{D,k}^{AV} \tag{3.40}$$

$$\dot{C}_{D,k}^{UN} = c_{F,k}\dot{E}_{D,k}^{UN}. \tag{3.41}$$

where $c_{F,k}$ is the average cost per exergy unit of fuel. Additional information on $c_{F,k}$ can be found in the following references [2, 6]. The cost associated with the total rate of exergy destruction in a component can therefore be defined as,

$$\dot{C}_{D,k} = c_{F,k}\dot{E}_{D,k} = \dot{C}_{D,k}^{AV} + \dot{C}_{D,k}^{UN}. \tag{3.42}$$

In references [2] and [5] the unavoidable investment costs per unit of product exergy $(\dot{Z}/\dot{E}_P)_k^{UN}$ was determined by considering an extremely inefficient version of the k-th component. Referring to Figure 2.3 in Chapter 2, this unavoidable investment cost at any given design point A, $\dot{Z}_{k,A}^{UN}$, is defined as follows:

$$\dot{Z}_{k,A}^{UN} = \dot{E}_{P,k,A}\left(\frac{\dot{Z}}{\dot{E}}\right)_k^{UN}. \tag{3.43}$$

and the avoidable cost $\dot{Z}_{k,A}^{AV}$ is defined as

$$\dot{Z}_{k,A}^{AV} = \dot{Z}_{k,A} - \dot{Z}_{k,A}^{UN}. \tag{3.44}$$

A modified exergoeconomic factor f_k^{AV} was then introduced [6, 30]

$$f_k^{AV} = \frac{\dot{Z}_k^{AV}}{\dot{Z}_k^{AV} + \dot{C}_{D,k}^{AV}}. \tag{3.45}$$

3.7.2 Endogenous and Exogenous costs

The costs associated with the endogenous and exogenous parts of the exergy destruction are defined as follows,

$$\dot{C}_{D,k}^{EN} = c_{F,k}\dot{E}_{D,k}^{EN} \tag{3.46}$$

$$\dot{C}_{D,k}^{EX} = c_{F,k}\dot{E}_{D,k}^{EX}. \tag{3.47}$$

The cost of exergy destruction for the four categorized segments of the exergy destruction are defined as follows:

$$\dot{C}_{D,k}^{EN,AV} = c_{F,k}\dot{E}_{D,k}^{EN,AV} \tag{3.48}$$

$$\dot{C}_{D,k}^{EN,UN} = c_{F,k}\dot{E}_{D,k}^{EN,UN} \tag{3.49}$$

$$\dot{C}_{D,k}^{EX,AV} = c_{F,k}\dot{E}_{D,k}^{EX,AV} \tag{3.50}$$

$$\dot{C}_{D,k}^{EX,UN} = c_{F,k}\dot{E}_{D,k}^{EX,UN}. \tag{3.51}$$

The endogenous owning and operating cost rate of the k-th component, $\dot{Z}_k^{EN}$ is defined as the cost rate associated with the component if all the other components in the system were to function ideally whereas the exogenous owning and operating cost rate, $\dot{Z}_k^{EX}$ is the additional cost invested into the k-th due to the exergy destruction in the other components in the system. A detailed procedure for determining these values can be found in Appendix D. The total investment cost of the k-th component can therefore be defined as,

$$\dot{Z}_k = \dot{Z}_k^{EN} + \dot{Z}_k^{EX}. \tag{3.52}$$

3.7.3 Additional splitting of the investment cost

So far the four main categories in which the investment cost has been split has been discussed. Like the splitting of the exergy destruction, this cost can be further split into four categorized segments, namely,

- endogenous-avoidable investment cost, $\dot{Z}_k^{EN,AV}$
- endogenous-unavoidable investment cost, $\dot{Z}_k^{EN,UN}$
- exogenous-avoidable investment cost, $\dot{Z}_k^{EX,AV}$

- exogenous- unavoidable investment cost, $\dot{Z}_k^{EX,UN}$.

The endogenous- unavoidable investment cost was first determined and used to calculate the other costs.

3.7.4 Endogenous Unavoidable Investment Cost , $\dot{Z}^{EN,UN}$

The endogenous unavoidable investment cost is determined in the same manner as the endogenous exergy destruction. The component understudy is maintained at the assumed design parameter while the other components approach ideal operation. $(\dot{Z}_k/\dot{E}_P)_k^{EN,UN}$ for the k-th component is calculated at the point where all the other components operate ideally. The endogenous unavoidable investment cost for the k-th component in the original thermal plant, $\dot{Z}_k^{EN,UN}$, is calculated as follows:

$$\dot{Z}_k^{EN,UN} = \dot{E}_{P,k}^{EN} \left(\frac{\dot{Z}_k}{\dot{E}_{P,k}} \right)^{EN,UN} . \tag{3.53}$$

The equations used to calculate the other investment cost categories are as follows,

$$\dot{Z}_{AC}^{EN,AV} = \dot{Z}_{AC}^{EN} - \dot{Z}_{AC}^{EN,UN} \tag{3.54}$$

$$\dot{Z}_{AC}^{EX,UN} = \dot{Z}_{AC}^{UN} - \dot{Z}_{AC}^{EN,UN} \tag{3.55}$$

$$\dot{Z}_{AC}^{EX,AV} = \dot{Z}_{AC}^{EX} - \dot{Z}_{AC}^{EX,UN} \tag{3.56}$$

3.8 Summary of splitting exergy destruction and cost in a component

Figures 3.11 and 3.12 summarize the splitting of the exergy destruction and investment cost of the k-th component into its avoidable/unavoidable and endogenous/exogenous parts. This theory will be further developed in the proceeding chapters and applied to various energy conversion systems in order to illustrate its importance in improving the quality of the conclusions obtained from an exergoeconomic evaluation.

3.9 Advanced Exergoeconomic Analysis and Evaluation

An advanced exergoeconomic evaluation of a system is then defined as an evaluation based on the following parameters [29]:

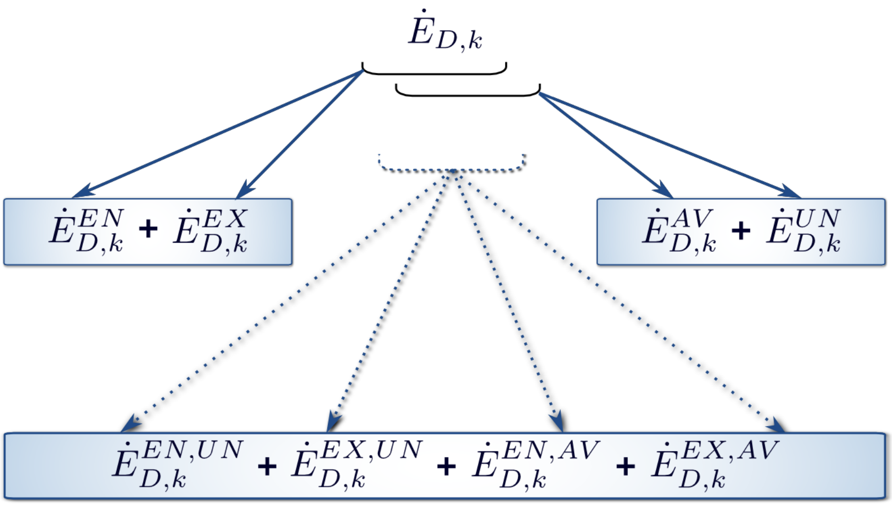


Figure 3.11: A diagrammatic representation of the splitting of the exergy destruction of a component k into its endogenous/exogenous and unavoidable/avoidable parts [24].

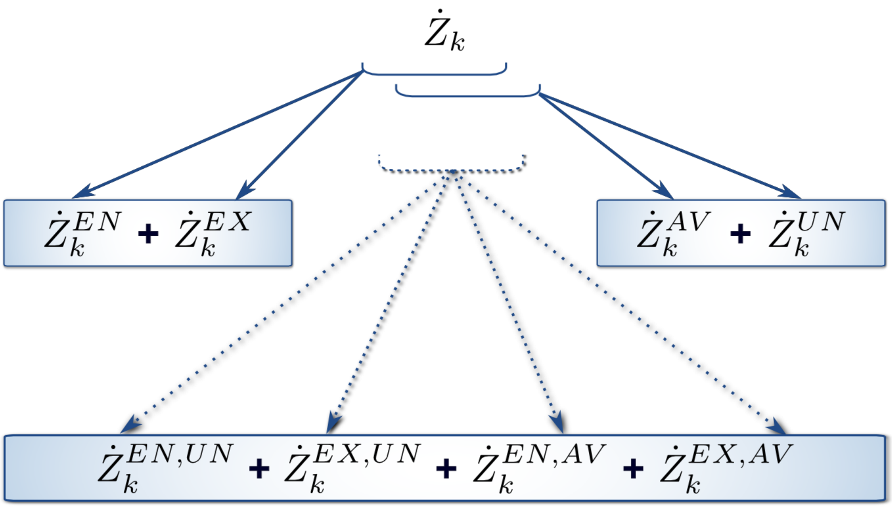


Figure 3.12: A diagrammatic representation of the splitting of the investment cost of a component k into its endogenous/exogenous and unavoidable/avoidable parts [24].

- The endogenous avoidable, $\dot{E}_{D,k}^{EN,AV}$ and the exogenous avoidable, $\dot{E}_{D,k}^{EX,AV}$ exergy destruction and the associated costs $\dot{C}_{D,k}^{EN,AV}$ and $\dot{C}_{D,k}^{EX,AV}$ respectively.
- The endogenous avoidable investment cost $\dot{Z}_{D,k}^{EN,AV}$
- The modified exergetic efficiency, $\varepsilon_k^{EN,AV}$ where,

$$\varepsilon_k^{EN,AV} = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k} - \dot{E}_{D,k}^{UN} - \dot{E}_{D,k}^{EX,AV}} = 1 - \frac{\dot{E}_{D,k}^{EN,AV}}{\dot{E}_{F,k} - \dot{E}_{D,k}^{UN} - \dot{E}_{D,k}^{EX,AV}}. \tag{3.57}$$

- The modified exergoeconomic factor, $f_k^{EN,AV}$ where,

$$f_k^{EN,AV} = \frac{\dot{Z}_k^{EN,AV}}{\dot{Z}_k^{EN,AV} + c_{F,k}\dot{E}_{D,k}^{EN,AV}}. \tag{3.58}$$

Plant components with a large value of the sum $\dot{Z}_k^{EN,AV} + \dot{C}_{D,k}^{EN,AV}$ should be improved first. The comparison of the $\varepsilon_k^{EN,AV}$ values indicate the potential for improving the thermodynamic performance of the k-th component, with a low $\varepsilon_k^{EN,AV}$ value indicating a high potential for improvement. The exergoeconomic factor, $f_k^{EN,AV}$ is used to identify the major cost source (capital investment or cost of exergy destruction) associated with the system component. If the $f_k^{EN,AV}$ value is high, it should be investigated as to whether it is cost effective to reduce the capital investment at the expense of the component's efficiency. If the $f_k^{EN,AV}$ value is low the component's efficiency should be improved by increasing the component's capital investment that is endogenous and avoidable.

In [17], the authors applied the idea of splitting both the exergy destruction and the investment costs in the endogenous/exogenous and avoidable/unavoidable parts to refrigeration machines and showed that such an application produced better conclusions and indeed enhanced the conventional exergoeconomic evaluation.

The proceeding chapters show the application of the engineering "graphical" method to selected thermal systems for the purpose of splitting the exergy destruction into its exogenous and endogenous parts. The thermal systems include:

- A refrigeration system
- A simple gas turbine system
- A cogeneration system

- A combined system

The results from the last three systems were used in carrying out an advanced exergoeconomic evaluation for improving the design of the systems.

4 Application I: Simple Refrigeration Machine

4.1 Application I: Simple Refrigeration Machine

The Engineering Method was applied to a simple refrigeration machine comprising of an evaporator (EV) with exergetic efficiency ε_{EV} , a compressor (CM) with exergetic efficiency ε_{CM} , a condenser (CD) with exergetic efficiency ε_{CD} and a throttling valve (TV) with exergetic efficiency ε_{TV}. Figure 4.1 shows a schematic diagram of the machine. The working fluid is R717 and the required cooling capacity is 100kW.

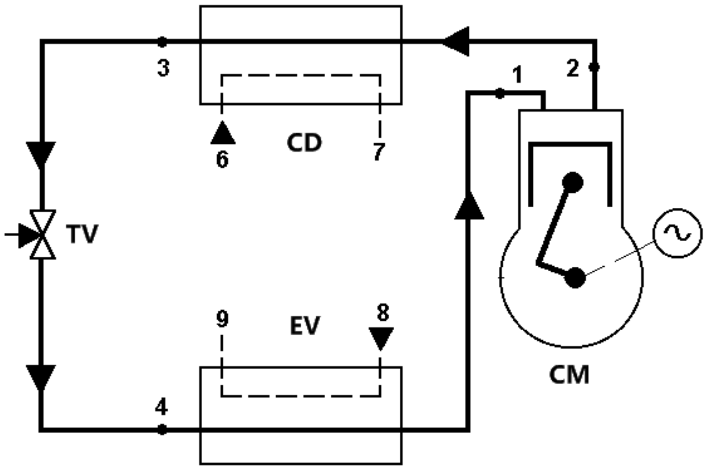


Figure 4.1: Simple refrigeration machine.

Energy and Exergy Analysis

Table 4.1 and 4.2 list the parameters used and the results of the exergy analysis for the system.

Table 4.1: Parameters of each stream of the simple refrigeration machine.

Stream	$\dot{m}$ (kg/s)	T (K)	p (bar)
1	0.0962	248.15	1.515
2	0.0962	456.65	1.555
3	0.0962	313.15	1.555
4	0.0962	248.15	1.515
6	14.18	293.15	1.013
7	14.18	303.15	1.013
8	9.921	258.15	1.013
9	9..21	268.15	1.013

Table 4.2: Exergy Analysis of the refrigeration machine.

Component k	$\dot{E}_{F,k}$(kW)	$\dot{E}_{P,k}$(kW)	$\dot{E}_{D,k}$(kW)	ε_k(%)
CM	43.12	37.36	5.76	87.00
CD	15.08	5.90	9.18	39.00
TV	23.14	18.99	4.15	82.00
EV	18.77	13.42	5.35	72.00

4.2 Procedure for Examining each Component

The engineering method was applied to each component to determine its endogenous exergy destruction, with the exception of the throttling device, since it is impossible to maintain a constant exergetic efficiency in this device while varying the exergy destruction in the other components. For the heat exchangers (i.e. the evaporator and the condenser) the exergy destruction was reduced by adjusting the minimum temperature difference between the two streams flowing through them. The isentropic efficiency was changed in order to vary the exergy destruction within the compressor.

4.3 Results for the simple refrigeration machine

The graphical results obtained for each component are shown in the plots, Figure 4.2 to 4.4.

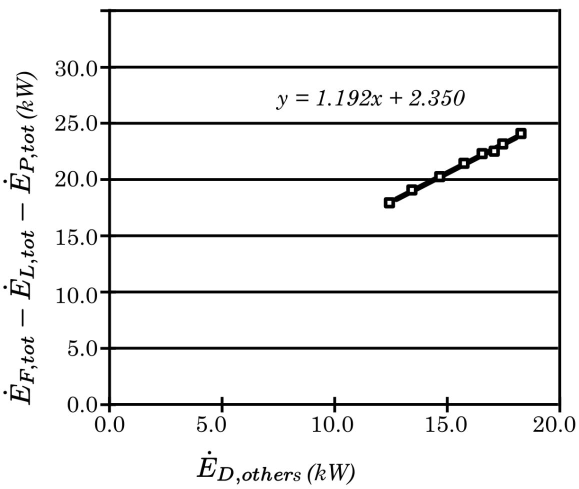


Figure 4.2: Plot showing the results of the engineering method used in analyzing the compressor.

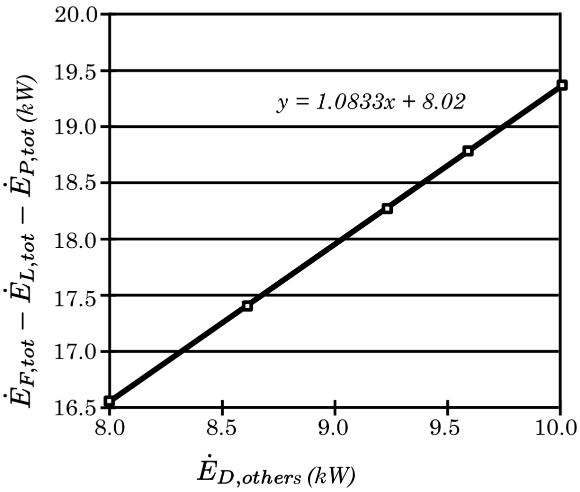


Figure 4.3: Plot showing the results of the engineering method used in analyzing the condenser.

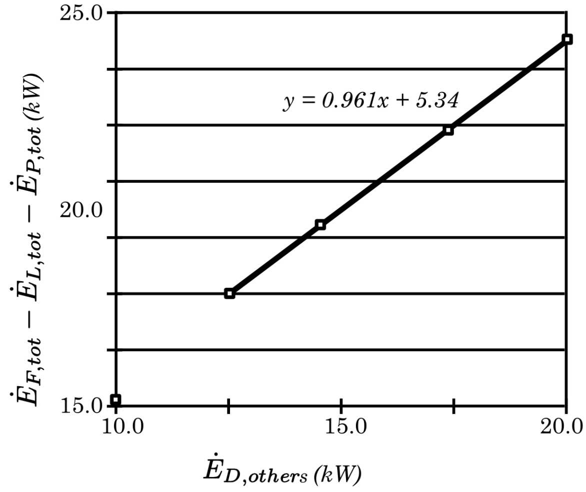


Figure 4.4: Plot showing the results of the engineering method used in analyzing the evaporator.

Table 4.3 summarizes the results of splitting the exergy destruction rate for each component into its endogenous and exogenous parts. Equation (2.7), $\dot{E}_{D,k} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX}$, was used to calculate the exogenous values.

Table 4.3: Endogenous and exogenous exergy destruction rate in each component of the simple refrigeration machine.

Component k	ε_k(%)	$\dot{E}_{D,k}$(kW)	$\dot{E}_{D,k}^{EN}$(kW)	$\dot{E}_{D,k}^{EX}$(kW)
CM	87.00	5.76	2.35	3.41
CD	39.00	9.18	8.02	1.16
TV	82.00	4.15	-	-
EV	72.00	5.35	5.34	0.01

The results were also compared against those obtained using the thermodynamic method discussed in chapter 2. Table 4.4 highlights this comparison.

Table 4.4: Comparison of the endogenous exergy destruction values obtained for the simple refrigeration machine using the engineering and the thermodynamic methods

			Endogenous Exergy Destruction (kW)		
Component	Exergetic Efficiency(%)	Exergy Destruction(kW)	Thermodynamic Method	Engineering Method	difference (%)
Compressor (CM)	87.0	5.76	2.53	2.35	7.1
Condenser (CD)	39.0	9.18	8.11	8.02	1.1
Throttle Valve (TV)	82.0	4.15	1.74	Cannot be determined as ε_{TV} cannot be held constant while the other components vary	–
Evaporator (EV)	72.0	5.35	5.24	5.34	1.9

The results obtained from both methods are quite similar. Such an outcome further validates the engineering method as a legitimate procedure for splitting the exergy destruction in a component into its endogenous and exogenous parts.

An advanced exergoeconomic analysis of the simple refrigeration machine can be found in [17].

5 Application II: Simple Gas Turbine System

5.1 Application II: Simple Gas Turbine System

The Engineering Method was applied to a simple power system. The power plant comprised of an air compressor (AC) with exergetic efficiency ε_{AC}, a combustion chamber (CC) with exergetic efficiency, ε_{CC} and an expander (GT) with exergetic efficiency, ε_{GT}. The combustion gases enter the expander at 1230 K and the system produces power at a constant rate 30 MW.

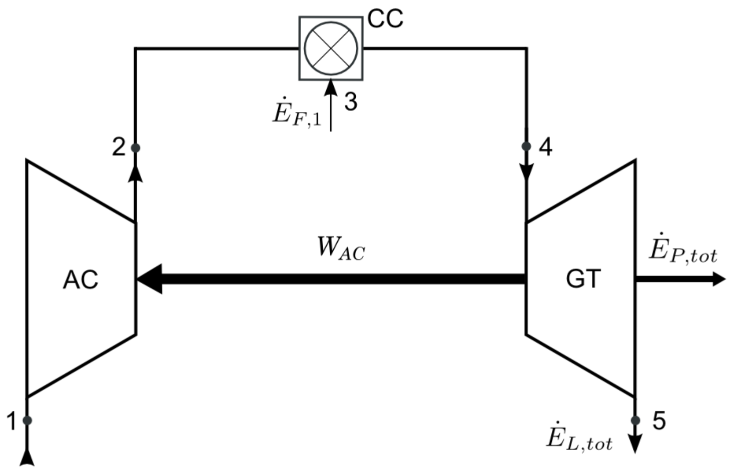


Figure 5.1: Simple Gas Turbine System.

5.1.1 Energy and Exergy Analysis

Tables 5.1 and 5.2 list the parameters used and the results of the exergy analysis for the simple gas turbine system.

Table 5.1: Parameters of each stream of the simple power system.

Stream	$\dot{m}$ (kg/s)	T (K)	p (bar)
1	140.70	298.15	1.013
2	140.70	635.02	10.13
3	2.16	298.15	12.00
4	142.86	1230.00	10.13
5	142.86	763.58	1.013

Table 5.2: Exergy Analysis of the simple gas turbine system.

Component k	$\dot{E}_{F,k}$(MW)	$\dot{E}_{P,k}$(MW)	$\dot{E}_{D,k}$(MW)	ε_k(%)
Air Compressor (AC)	48.92	44.04	4.88	90.02
Combustion Chamber (CC)	112.13	69.73	42.39	62.18
Expander (GT)	84.24	79.80	4.44	94.72

5.2 Procedure for Examining each Component

The procedure used to examine each component was done in accordance with that described in chapter 3.

5.2.1 Air Compressor

The reversible adiabatic heater (RAH) was used in series with the combustion chamber as shown in Figure 3.6.

Figures 5.2 a and 5.2 b illustrate on a T-s diagram the adjustments made to the components for determining the endogenous exergy destruction in an air compressor with an exergetic efficiency of ε_{AC} . In both cases the exergy destruction in the combustion chamber (process 2*-4) is changed by varying the exergy supplied by the RAH. In Figure 5.2 a the exergy destruction in the expansion process 4-5, is varied by adjusting the efficiency of the expander whereas in Figure 5.2 b the isentropic efficiency of the expander is maintained at 100% (i.e. $\dot{E}_{D,GT} = 0$). The arrows indicate the direction of the adjustments made to the reversible adiabatic heater.

Results

The results from both procedures are shown in the plots Figure 5.3 a and 5.3 b.

The plots show that the endogenous exergy destruction in the air compressor is approximately 4.2 MW at an exergetic efficiency of 90.02%. There is a 2% difference between the results obtained

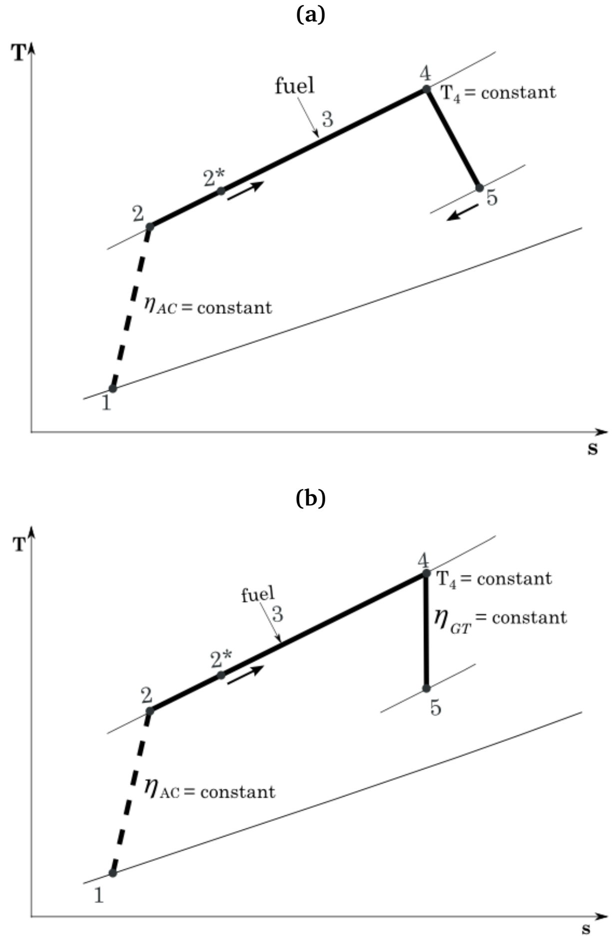


Figure 5.2: T-s diagram of the simple cycle illustrating the two procedures for analyzing the AC

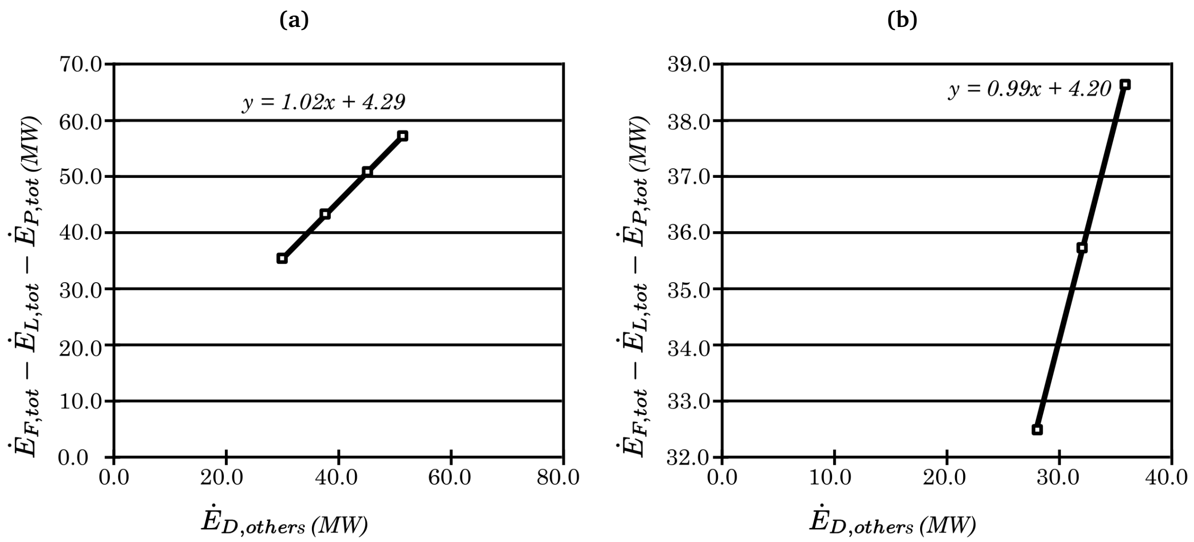


Figure 5.3: Plots showing the results for the AC in accordance with the procedure shown in Figure 5.2(a) and 5.2(b) respectively.

from the two procedures, which can be attributed to the fact that when the exergy destruction is being reduced in both the expander and the combustion chamber one has to ensure that at the point when $\dot{E}_{D,others} = 0$, the exergy destruction in each component is also zero, (i.e. $\dot{E}_{D,CC} = 0$ and $\dot{E}_{D,GT} = 0$) (refer to section 3.3). Sometimes this was a bit difficult to maintain and thus an error was introduced as a result of this.

Referring to Table 5.2 an endogenous exergy destruction of 4.2 MW also suggests that 0.68 MW of the exergy destroyed in this component is exogenous. In analyzing the air compressor, using the procedure described in Figure 5.2 b it was discovered that as the exergy destruction in the combustion chamber reduces the exergy destruction in the air compressor increases suggesting that the exogenous effect of the combustion chamber is actually negative. The latter appears to be true on a "thermodynamic" level as well since as the exergy destruction in the combustion chamber decreases the fuel required by the combustion chamber also reduces, when this happens the mass flow rate of air must increase in order to maintain a constant total product, $\dot{W}_{NET}$.

The expander on the other hand has a "positive" effect on the air compressor in that as the exergy destruction in the expander reduces the exogenous exergy destruction in air compressor also reduces. The net result of the exogenous exergy destruction of both the combustion chamber and the expander on the air compressor in this case is quite small, 0.68 MW.

5.2.2 Expander

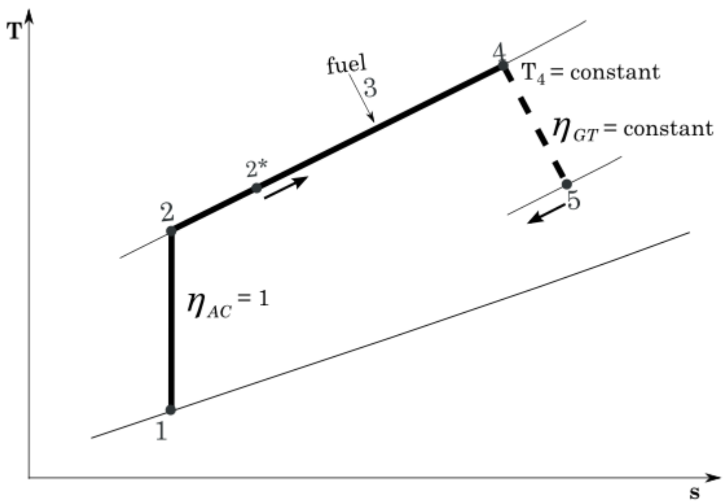


Figure 5.4: T-s diagram of the simple cycle illustrating the procedure for analyzing the GT.

Figure 5.4 illustrates the procedure used in determining the endogenous exergy destruction for the expander. Here the isentropic efficiency of the air compressor was held constant at 100% (i.e. $\dot{E}_{D,AC} = 0$). Again the exergy destruction in the combustion chamber is adjusted by varying the fuel supplied to the reversible adiabatic heater.

Results

The results of the procedure is shown in Figure 5.5.

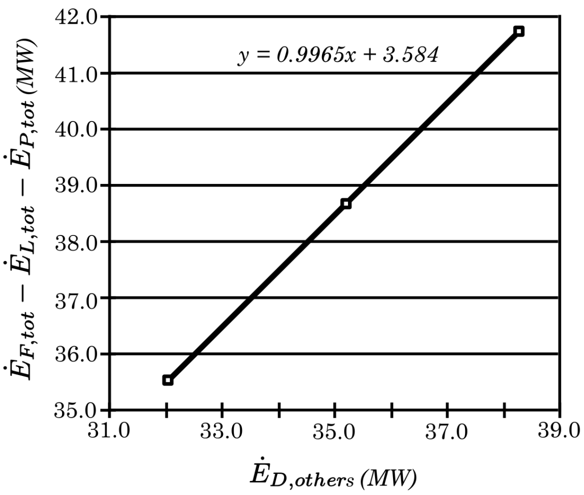


Figure 5.5: Plot showing the results of the procedure used in analyzing the GT.

Figure 5.5 shows that at an exergetic efficiency of 94.72% the endogenous exergy destruction of the expander is 3.584 MW with an exogenous value of 0.856 MW. Here it was also discovered that the combustion chamber contributed "negatively" to the exogenous exergy destruction in the expander.

5.2.3 Combustion Chamber

Figure 5.6 illustrates on a T-s diagram the adjustments made to the components for determining the endogenous exergy destruction in the combustion chamber. Note that for this procedure the reversible adiabatic heater cannot be used. The combustion chamber is analyzed by fixing the exergetic efficiency of the air compressor to 1 (i.e. $\dot{E}_{D,AC} = 0$). This will result in a new inlet temperature to the combustion chamber, T_2'. T_4 will then have to be increased to a new position T_4* until the required value ε_{CC} is achieved. The exergy destruction in the expander is gradually reduced by increasing its isentropic efficiency.

The endogenous exergy destruction of the combustion chamber is independent of the magnitude of the specific isentropic work of both the air compressor and the expander. Hence the endogenous exergy destruction occurring between T_2 and T_4 is equivalent to the endogenous exergy destruction

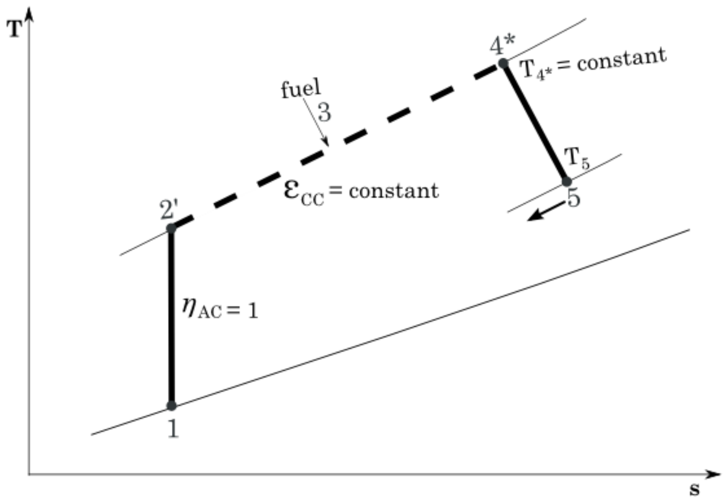


Figure 5.6: T-s diagram of the Simple Cycle illustrating the procedure for analyzing the CC.

between T_2' and T_4* providing the exergetic efficiency remains the same. To prove the latter, the endogenous exergy destruction was evaluated at different inlet and outlet temperatures with the exergetic efficiency of the combustion chamber being kept constant. The results are shown in Table 5.3.

Lastly, the endogenous exergy destruction can be directly achieved by simulating isentropic conditions for both the air compressor and the expander with any simulation software package.

Figure 5.7 shows the data obtained when analyzing the combustion chamber with exergetic efficiency of 62.18%, here $\dot{E}^{EN}_{D,CC} = 27.85$ MW.

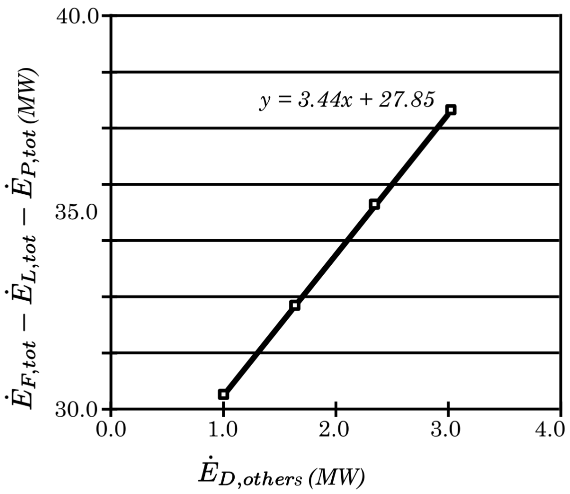


Figure 5.7: Plot showing the results of the procedure used in analyzing the CC

The various procedures mentioned and the corresponding results obtained are shown in Table 5.3.

Again these values were taken when the combustion chamber operated at an exergetic efficiency of 62.18%. The endogenous values obtained by the various methods are quite close. In the last two

Table 5.3: Results of the Endogenous Exergy Destruction in the CC.

Procedure	Endogenous value obtained (MW)
Plot: $\varepsilon_{AC} = 100\%$, ε_{GT} is varied, $T_1 = 298.15K$, $T_{2'} = 569.31K$, $T_{4*} = 1313.4K$	27.85
Simulation: Isentropic Compression and Expansion simulated using GateCycle© $T_1 = 298.15K$, $T_2' = 569.31K$, $T_{4*} = 1313.4K$	28.54
Simulation: Isentropic Compression and Expansion simulation using GateCycle©. Inlet and outlet temperatures vary. $T_{1*} = 313.77$, $T_2 = 596.19$, $T_4 = 1230K$	28.54

procedures shown in the table, GateCycle©was used to directly determine $\dot{E}^{EN}_{D,CC}$. It is important to note here that the limitation of the GateCycle© software to achieve an adiabatic isentropic expansion will have created a slightly higher endogenous exergy destruction value (2.5% error) from the value determined using the engineering method. Only a maximum isentropic expansion efficiency of 99% was achieved using GateCycle©.

5.3 Summary of Results

The Table 5.4 summarizes the results of the endogenous exergy destruction for each component within the simple system. Equation (2.7), $\dot{E}_{D,k} = \dot{E}^{EN}_{D,k} + \dot{E}^{EX}_{D,k}$, was used to determine the exogenous exergy destruction of each component.

Table 5.4: Endogenous and Exogenous exergy destruction values for the Simple System.

Component k	ε_k(%)	$\dot{E}_{D,k}$(MW)	$\dot{E}^{EN}_{D,k}$(MW)	$\dot{E}^{EX}_{D,k}$(MW)
Air Compressor (AC)	90.02	4.88	4.20	0.68
Combustion Chamber (CC)	62.19	42.39	27.85	14.54
Expander (GT)	94.72	4.44	3.584	0.856

5.4 Relationship between endogenous exergy destruction and exergetic efficiency

To facilitate the ease of determining the endogenous exergy destruction of each component operating at any exergetic efficiency in the simple gas turbine system configuration, graphical plots were created. These graphs were created by plotting the endogenous exergy destruction versus exergetic efficiency. Figure 5.8 to 5.10 show the plots obtained for each component of the simple gas turbine system.

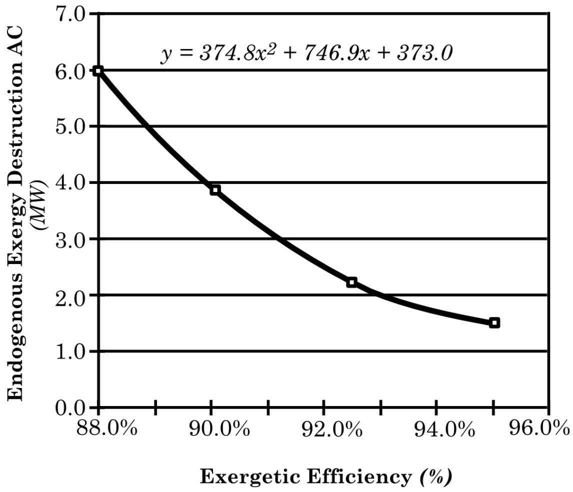


Figure 5.8: Endogenous Exergy Destruction Curve for the AC.

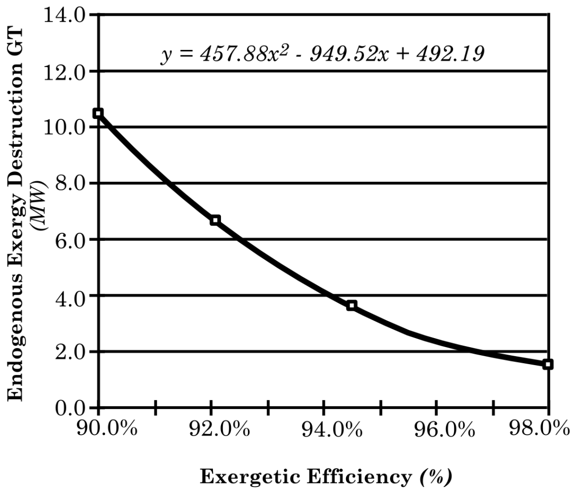


Figure 5.9: Endogenous Exergy Destruction Curve for the GT.

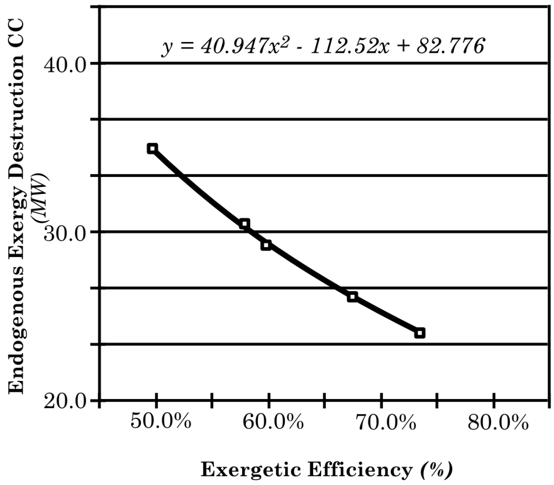


Figure 5.10: Endogenous Exergy Destruction Curve for the CC.

5.5 Splitting II: Avoidable and Unavoidable Exergy Destruction

The exergy destruction in the components of the simple gas turbine system was then split into its avoidable and unavoidable parts.

The unavoidable exergy destruction for each k-th component was found using the unavoidable exergy destruction per product $(\dot{E}_D/\dot{E}_P)_k^{UN}$ calculated in [30]. Table 5.5 shows the calculated $(\dot{E}_D/\dot{E}_P)_k^{UN}$ for each component and the assumptions made. Table 5.6 shows the unavoidable and

Table 5.5: Unavoidable exergy destruction per product exergy in each component.

Component k	$\left(\frac{\dot{E}_D}{\dot{E}_P}\right)_k^{UN}$	Assumptions
Air Compressor	0.054	Purchased equipment cost of the compressor becomes infinite when isentropic efficiency is 90%
Combustion Chamber	0.367	• high temperature of the reactants (811K fuel and 1000K/1360K for air) • High outlet temperature 1773 K/2998K • Adiabatic combustion
Expander	0.021	Purchased equipment cost of the expander becomes infinite when isentropic efficiency is 92%

avoidable exergy destruction calculated for each component of the simple gas turbine system.

Table 5.6: Avoidable and unavoidable exergy destruction in each component.

Component k	$\dot{E}_{P,k}$ (MW)	$\dot{E}_{D,k}$ (MW)	$\dot{E}^{UN}_{D,k}$ (MW)	$\dot{E}^{AV}_{D,k}$ (MW)
Air Compressor (AC)	44.04	4.88	2.38	2.51
Combustion Chamber (CC)	69.73	42.39	25.59	16.80
Expander (GT)	79.80	4.44	2.15	2.29

5.6 Combining the various types of Exergy Destruction

Table 3.1 in chapter 3 was created for each component and shown in Table 5.7 to Table 5.9. The

Table 5.7: Splitting of the exergy destruction in the air compressor.

Exergy destruction category	$\dot{E}^{EN}_{D,AC}$ (MW)	$\dot{E}^{EX}_{D,AC}$ (MW)
$\dot{E}^{AV}_{D,AC}$ (MW)	2.45	0.05
$\dot{E}^{UN}_{D,AC}$ (MW)	1.75	0.63

results indicate that at least 50% of the exergy destroyed in the air compressor is endogenous-avoidable and can be reduced through the improvement of the component's efficiency. At least 48% of the totally exergy destroyed is unavoidable. Over 50% of the exergy destroyed in the combustion

Table 5.8: Splitting of the exergy destruction in the combustion chamber.

Exergy destruction category	$\dot{E}^{EN}_{D,CC}$ (MW)	$\dot{E}^{EX}_{D,CC}$ (MW)
$\dot{E}^{AV}_{D,CC}$ (MW)	5.47	11.33
$\dot{E}^{UN}_{D,CC}$ (MW)	22.38	3.21

chamber is due to the irreversibilities within the component itself and cannot be reduced. Such results were expected as a large part of the exergy destroyed is due to the chemical combustion process taking place within the component. Approximately 60% of the exergy destroyed in the expander is endogenous.

A further examination of the exogenous exergy destruction in each component will now be carried out. The exogenous exergy destruction of the k-th component can be divided into the contribution by each component within the system. With the use of the engineering method, the exergetic efficiency of two components within the system can be held constant while the exergy destruction of the other component is gradually reduced. The part of exogenous exergy destruction in the k-th component that is caused by the exergy destruction in r-th component $\dot{E}^{EX,r}_{D,k}$ can then be determined, at the

Table 5.9: Splitting of the exergy destruction in the gas turbine.

Exergy destruction category	$\dot{E}^{EN}_{D,GT}$ (MW)	$\dot{E}^{EX}_{D,GT}$ (MW)
$\dot{E}^{AV}_{D,GT}$ (MW)	2.69	0.08
$\dot{E}^{UN}_{D,GT}$ (MW)	0.90	0.78

same time the exogenous exergy destruction in the r-th component that is caused by the exergy destruction in the k-th component, $\dot{E}^{EX,r}_{D,k}$ can also be found . It was also observed that the total exogenous exergy destruction in the k-th component was higher than the sum of the contribution of all the other components to its exogenous exergy destruction value, i.e.,

$$\dot{E}^{EX}_{D,k} > \sum_{\substack{r=1 \\ r\neq k}}^{n-1} \dot{E}^{EX,r}_{D,k}$$

where n is the total number of components within the system. The difference is caused by the combined effect of the exergy destruction of all the other components within the system on the k-th component. In [29] this combined effect was termed *mexogenous* exergy destruction, $\dot{E}^{MX}_{D,k}$, where

$$\dot{E}^{MX}_{D,k} = \dot{E}^{EX}_{D,k} - \sum_{\substack{r=1 \\ r\neq k}}^{n-1} \dot{E}^{EX,r}_{D,k}. \tag{5.1}$$

The splitting of the exogenous exergy destruction for each component of the simple gas turbine system is shown in Table 5.10.

Table 5.10: Splitting of the exogenous exergy destruction within each component of the simple gas turbine system.

k-th Component	$\dot{E}^{EX}_{D,k}$ (MW)	r-th Component	$\dot{E}^{EX,r}_{D,k}$ (MW)
Air Compressor (AC)	0.68	CC	-0.565
		GT	1.000
		mexogenous	0.245
Combustion Chamber (CC)	14.54	AC	4.320
		GT	5.990
		mexogenous	4.230
Expander (GT)	0.856	AC	1.016
		CC	-0.194
		mexogenous	0.034

The exogenous exergy destruction values provide the net effect of the other components within the system on the k-th component. Table 5.10 shows that the effect of the exergy destruction of the combustion chamber on both the air compressor and the expander is negative, implying that as the exergy destruction in the combustion chamber reduces, the exergy destruction in the air compressor and the expander increases or vice versa. As the combustion chamber improves i.e. its exergy destruction decreases, less fuel is required for the combustion process. However additional air will be required in order to satisfy the net work output demand on the system, thus increasing the exergy destruction in both the air compressor and the expander. It is important to note again that the exergy destruction in the combustion chamber was reduced with the use of the reversible adiabatic heater, RAH.

Care must be taken when drawing conclusions based on the total exogenous exergy destruction values only. For example, for a small exogenous exergy destruction value, it may be misleading to conclude that the other components have little effect on the exergy destruction in the air compressor. More accurate conclusions can be drawn by further splitting the exogenous exergy destruction values.

Additionally, one can also conclude from Table 5.10 that the effect of the exergy destruction in the air compressor and the expander on the exergy destruction in the combustion chamber is higher than the endogenous exergy destruction in the air compressor and the expander. This clearly proves the importance of the efficiency of the turbomachines (air compressor and expander) to the overall system efficiency.

5.7 Splitting the Cost of Exergy Destruction

The cost rate of exergy destruction, $\dot{C}_{D,k}$, was calculated for each component of the simple cycle and is shown in Table 5.11. All economic assumptions made can be found in Appendix C and all equations used in calculating the specific costs can be found in Appendix E.

Table 5.11: Cost rate of exergy destruction for each component of the simple power system.

Component k	$\dot{C}_{D,k}$ (US$/h)
Air compressor	138.89
Combustion chamber	584.50
Expander	117.17

The cost rate of exergy destruction associated with each exergy destruction category is listed in Tables 5.12 to 5.14. These costs were calculated using equations (3.48) to (3.51).

Table 5.12: Cost rate of exergy destruction per category for the air compressor.

Air compressor	$\dot{C}^{EN}_{D,AC}$ (US$/h)	$\dot{C}^{EX}_{D,AC}$ (US$/h)
$\dot{C}^{AV}_{D,AC}$	69.82	1.44
$\dot{C}^{UN}_{D,AC}$	49.65	17.99

Table 5.13: Cost rate of exergy destruction per category for the combustion chamber.

Combustion chamber	$\dot{C}^{EN}_{D,CC}$ (US$/h)	$\dot{C}^{EX}_{D,CC}$ (US$/h)
$\dot{C}^{AV}_{D,CC}$	75.39	156.25
$\dot{C}^{UN}_{D,CC}$	308.60	44.26

Table 5.14: Cost rate of exergy destruction per category for the expander.

Expander	$\dot{C}^{EN}_{D,GT}$ (US$/h)	$\dot{C}^{EX}_{D,GT}$ (US$/h)
$\dot{C}^{AV}_{D,GT}$	70.89	2.10
$\dot{C}^{UN}_{D,GT}$	23.61	20.58

5.8 Splitting the investment cost , $\dot{Z}_k$

The investment cost rate calculated for each component is shown in Table 5.15.

To further assist in accurately appropriating costs to the components the investment cost was also split. Appendix D details the procedure for splitting the investment cost for each component. All assumptions used by this author as well as from other references relating to the purchased costs of each component can be found in Appendix C.

5.8.1 Unavoidable Investment Cost , $\dot{Z}^{UN}$

Table 5.16 lists the design values used for determining the unavoidable investment cost for the components of the simple cycle. For the turbomachines (i.e. the air compressor and the expander),

Table 5.15: Investment cost rate for each component of the simple power system.

Component k	$\dot{Z}_k(US\$/h)$
Air compressor	57.03
Combustion chamber	5.23
Expander	48.59

the lowest possible isentropic efficiencies are selected whereas for the combustion chamber, the lowest pressure drop is selected while maintaining the required outlet temperature.

Table 5.16: Assumptions made in determining the unavoidable investment cost.

Component	Design Parameters	Design Parameters for inefficient components
Air Compressor	$\eta_{AC} = 0.8$	$\eta = 0.70$
Combustion Chamber	$\Delta P = 0.001$	$\Delta P = 0.05$
Expander	$\eta_{GT} = 0.88$	$\eta_{GT} = 0.70$

The splitting of the investment cost rate for each component is listed in Tables 5.17 to 5.20.

Table 5.17: Splitting of the investment cost rate in the four major segments.

Component k	$\dot{Z}^{EN}_{D,k}$ (US\$/h)	$\dot{Z}^{EX}_{D,k}$ (US\$/h)	$\dot{Z}^{AV}_{D,k}$ (US\$/h)	$\dot{Z}^{UN}_{D,k}$ (US\$/h)
Air compressor	48.96	8.07	30.84	26.19
Combustion chamber	3.26	1.97	5.12	0.11
Expander	40.32	8.27	36.96	11.62

5.9 Exergoeconomic Evaluation

The conventional exergoeconomic analysis was applied to the simple power system and the results as shown in Table 5.21 with the aid of the following parameters:

- The Capital cost associated with each component, $\dot{Z}$
- The costs rate of exergy destruction $\dot{C}_{D,k}$ for each component.
- The relative cost difference, r_k, where $r_k = (c_{P,k} - c_{F,k})/c_{F,k} = \dot{Z}_k/c_{F,k}\dot{E}_{P,k} + (1 - \varepsilon_k)/\varepsilon_k$ and $c_{F,k}$ and $c_{P,k}$ are the average cost per exergy unit of fuel and product respectively.
- The exergoeconomic factor f_k where $f_k = \dot{Z}_k/(\dot{Z}_k + \dot{C}_{D,k})$

Table 5.18: Investment cost rate per category for the air compressor.

Air compressor	$\dot{Z}^{EN}_{D,AC}$ (US$/h)	$\dot{Z}^{EX}_{D,AC}$ (US$/h)
$\dot{Z}^{AV}_{D,AC}$	25.93	4.91
$\dot{Z}^{UN}_{D,AC}$	23.03	3.16

Table 5.19: Investment cost rate per category for the combustion chamber.

Combustion chamber	$\dot{Z}^{EN}_{D,CC}$ (US$/h)	$\dot{Z}^{EX}_{D,CC}$ (US$/h)
$\dot{Z}^{AV}_{D,CC}$	3.20	1.92
$\dot{Z}^{UN}_{D,CC}$	0.06	0.05

Table 5.20: Investment cost rate per category for the expander.

Expander	$\dot{Z}^{EN}_{D,GT}$ (US$/h)	$\dot{Z}^{EX}_{D,GT}$ (US$/h)
$\dot{Z}^{AV}_{D,GT}$	31.10	5.86
$\dot{Z}^{UN}_{D,GT}$	9.22	2.41

Table 5.21: Results of the conventional exergoeconomic analysis.

Component	$c_{F,k}$ (MW)	$c_{P,k}$ (MW)	r_k (%)	ε_k (%)	f_k (%)	$\dot{C}_{D,k}$ (US$/h)	$\dot{Z}_k$ (US$/h)	$\dot{C}_{D,k} + \dot{Z}_k$ (US$/h)
Air compressor	0.0079	0.0091	15.64	90.02	29.11	138.89	57.03	195.92
Combustion chamber	0.0038	0.0062	61.34	62.18	0.89	584.50	5.23	589.73
Expander	0.0073	0.0079	7.88	94.72	29.31	117.17	48.59	165.76

From the conventional exergoeconomic analysis, the sum $\dot{C}_{D,k} + \dot{Z}_k$ indicates that for the purpose of improving the system, emphasis should be placed on the combustion chamber followed by the air compressor.

Given a low contribution of the investment cost to the total operating cost of the combustion chamber f_{CC} and a high relative cost difference r_{CC}, one can also conclude that this component can be improved by reducing the exergy being destroyed within it. However such analysis does not consider the technical limitation in reducing the exergy being destroyed or whether an improvement in the efficiency of the remaining components will result in a lowering of the cost of exergy destruction. An advanced exergoeconomic evaluation is therefore required.

5.10 Advanced Exergoeconomic evaluation

In section 3.9 the parameters used in performing an exergoeconomic analysis and evaluation on thermal systems were given. The results of such an analysis for the simple power system are shown in Table 5.22.

Table 5.22: Advanced exergoeconomic analysis of the simple power system.

Component	$\varepsilon_k^{EN,AV}$ (%)	$f_k^{EN,AV}$ (%)	$\dot{C}_{D,k}^{EN,AV}$ (US$/h)	$\dot{Z}_k^{EN,AV}$ (US$/h)	$\dot{C}_{D,k}^{EN,AV} + \dot{Z}_k^{EN,AV}$ (US$/h)
Air compressor	94.72	27.08	69.82	25.93	95.75
Combustion chamber	92.73	4.07	75.39	3.20	78.59
Expander	96.74	30.49	70.89	31.10	101.99

The results of the endogenous/avoidable part of the total cost, $\dot{C}_{D,k}^{EN,AV} + \dot{Z}_k^{EN,AV}$ indicates that more emphasis should be placed on the expander and the air compressor with very little emphasis placed on the combustion chamber for the improvement of the overall system. The reason being is that most of the exergy being destroyed in the combustion chamber is endogenous and unavoidable due to current technical limitations.

The factor $f_k^{EN,AV}$ which considers the contribution of the endogenous/avoidable investment cost to the total endogenous/avoidable costs demonstrates that only 27.08% of the total endogenous/avoidable costs associated with the air compressor is investment cost. Thus, an improvement of the cost effectiveness of the air compressor can be achieved by improving its isentropic efficiency, which is currently 80%.

These results are closer to reality than the results obtained without splitting the exergy destruction or the investment cost.

6 Application III: Cogeneration Power Plant System

6.1 Application III: Cogeneration Plant

The Engineering Method was also applied to a cogeneration system. The system selected was the CGAM cogeneration system [32], which was designed to deliver 30 MW of electrical power and 14 kg/s of saturated steam at 20 bar (see Figure 6.1).

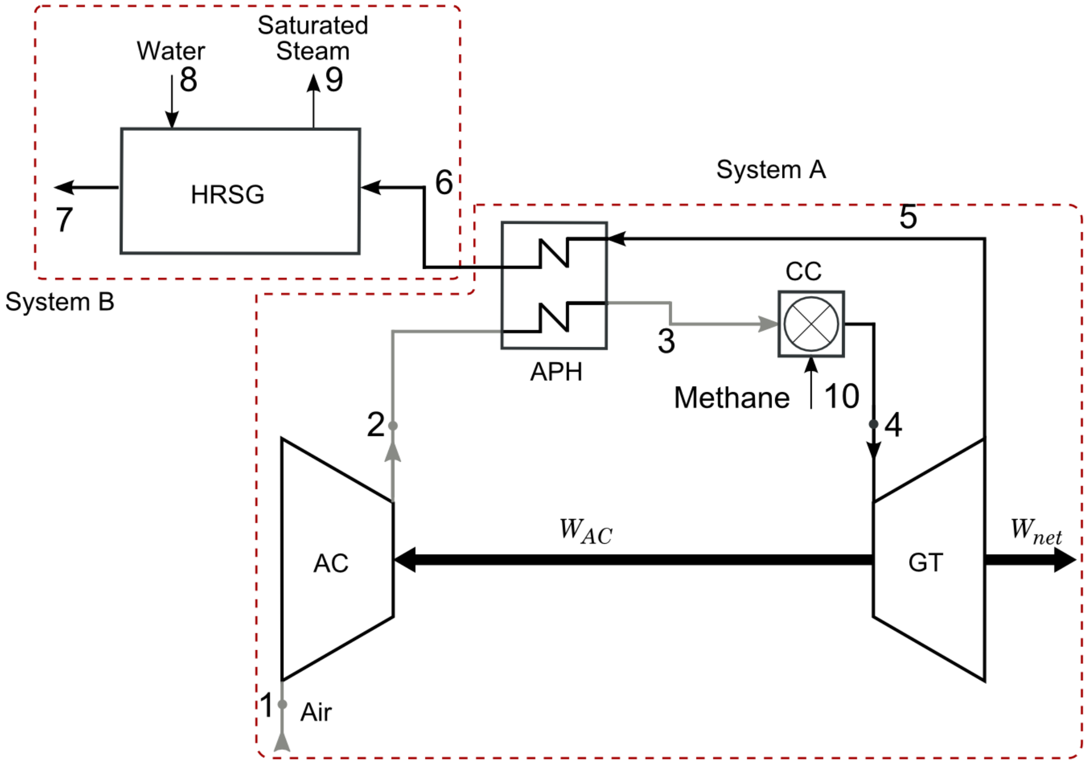


Figure 6.1: Schematic diagram of the CGAM cogeneration system.

The system consists of an air compressor (AC), an air preheater (APH), a combustion chamber (CC), and an expander (GT) which all form the power generating part of the system (system A) and a heat recovery steam generator (HRSG) which represents the second product of the system; 14 kg/s saturated steam at 20 bar (system B). Like all other applications used in this research work, the products are always kept constant.

In this system it is important to note that since there is no direct feed back of system B to system A, when sufficient energy is being supplied by system A to system B, the thermodynamic analysis of system B can be conducted independently of system A. The exergy at state point 6 then represents the exergy loss of system A.

The Tables 6.1 and 6.2 list the parameters used and the results of the exergy analysis for the cogeneration system.

Table 6.1: Parameters of each stream of the cogeneration system.

Stream	$\dot{m}$ (kg/s)	T (K)	p (bar)
1	93.149	298.15	1.01
2	93.149	612.21	10.13
3	93.149	850.00	9.62
4	94.922	1520.02	9.14
5	94.922	1010.84	1.10
6	94.922	793.46	1.07
7	94.922	432.974	1.01
8	14.000	298.15	20.00
9	14.000	482.53	20.00
10	1.773	298.15	12.00

Table 6.2: Exergy Analysis of the cogeneration system.

Component k	$\dot{E}_{F,k}$(MW)	$\dot{E}_{P,k}$(MW)	$\dot{E}_{D,k}$(MW)	ε_k(%)
Air Compressor (AC)	30.134	28.039	2.095	93.05
Air Pre-heater (APH)	16.618	13.799	2.819	83.04
Combustion Chamber (CC)	91.779	61.270	30.509	66.76
Expander (GT)	63.972	60.914	3.058	95.22
Heat Recovery Steam Generator (HRSG)	19.478	12.754	6.724	65.48

6.2 Procedure for Examining each Component

Like the simple power system, procedures for reducing the exergy destruction in each component of the cogeneration system were developed. In this system the temperatures $T_1 = 298.15K$, $T_3 = 850K$ and $T_4 = 1520K$ as well as the pressure ratio of the air compressor $r_p = 10$ are considered fixed system parameters.

The reversible adiabatic heater (RAH) was used in series with the combustion chamber as shown in Figure 6.2.

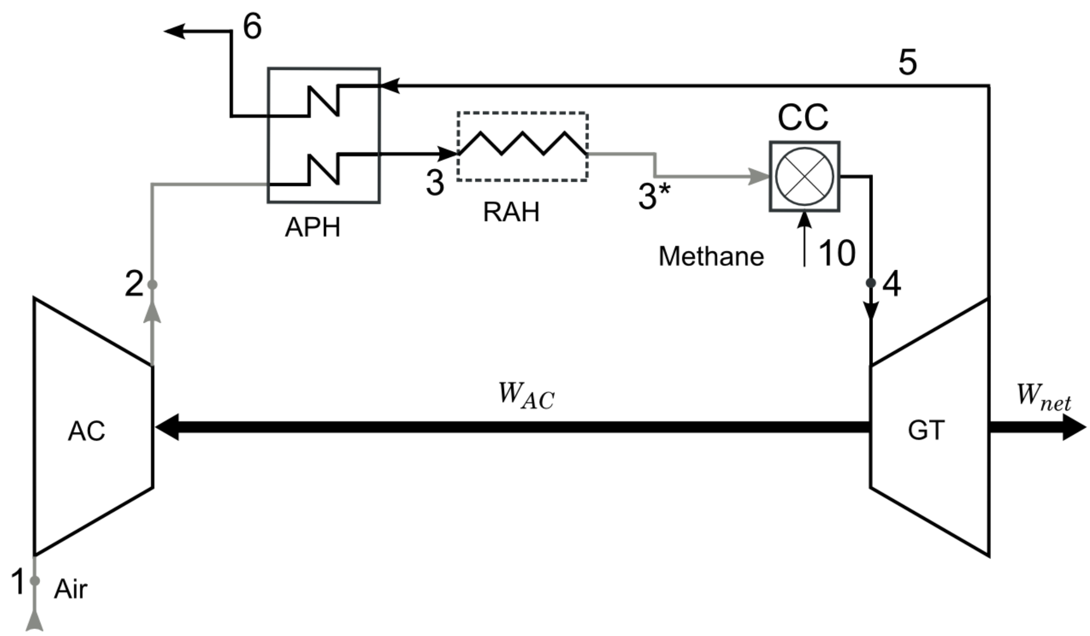


Figure 6.2: Position of the RAH in the Cogeneration System.

6.2.1 Air Compressor

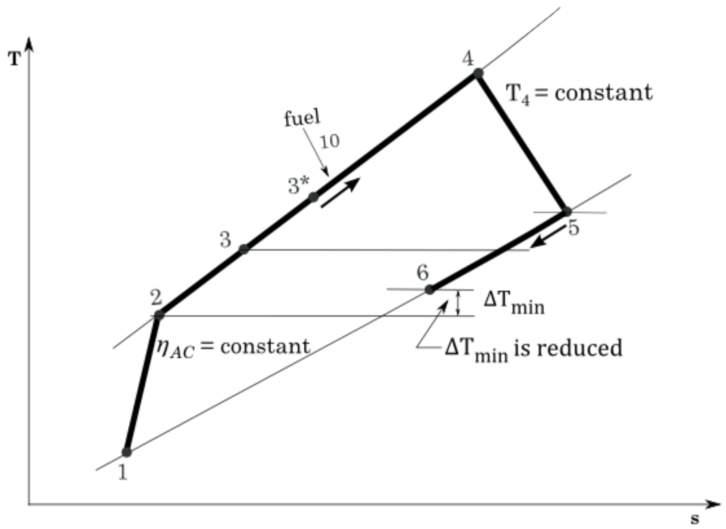


Figure 6.3: T-s diagram of the cogeneration system illustrating the procedure for analyzing the AC.

In Figure 6.3 the arrows illustrate the adjustments made to the components for determining the endogenous exergy destruction in an air compressor with an exergetic efficiency of ε_{AC} . Here the exergy destruction in the combustion chamber (process 3*-4) is changed by varying the exergy supplied to the RAH. The exergy destruction in the expansion process 4-5, is varied by adjusting the efficiency of the expander whereas and the exergy destruction in the air pre-heater is reduced by reducing the minimum temperature difference between the cold and hot streams.

Results

The results are shown in the plots of Figure 6.4 and 6.5. The results show that the endogenous

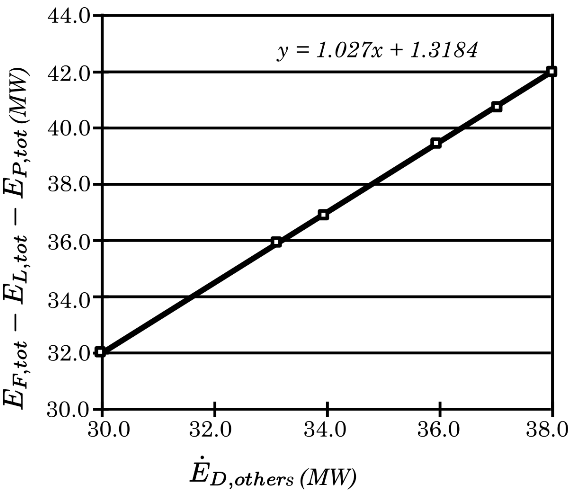


Figure 6.4: Plot showing the results of the procedure used in analyzing the AC

exergy destruction in the air compressor where $\varepsilon_{AC} = 93.05\%$ is approximately 1.32 MW.

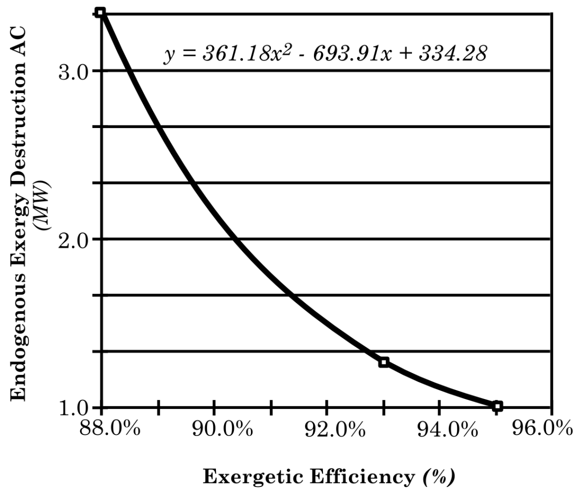


Figure 6.5: Endogenous Exergy Destruction Curve for the AC.

6.2.2 Air Preheater

Figure 6.6 illustrates the procedure used in determining the endogenous exergy destruction for the air preheater. The isentropic efficiencies of both the air compressor and expander were varied. The temperature of the material stream leaving the air preheater was kept constant since this was a requirement of this component for the system. Again the exergy destruction in the combustion

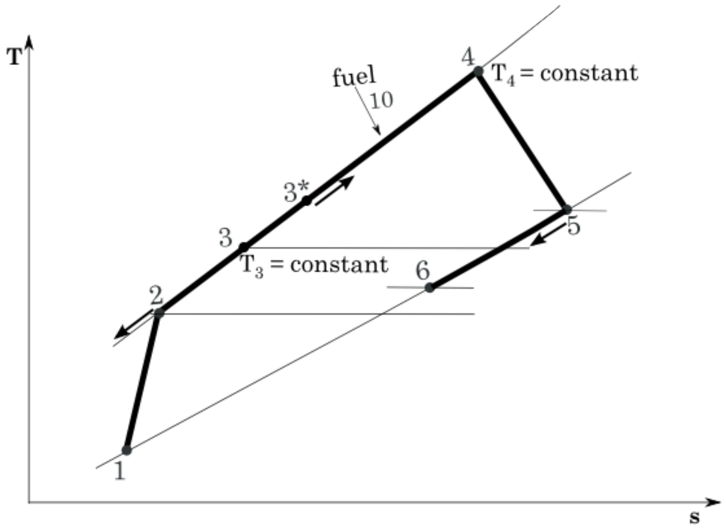


Figure 6.6: T-s diagram of the cogeneration system illustrating the procedure for analyzing the APH.

chamber is adjusted by varying the fuel supplied by the reversible adiabatic heater and the exergetic efficiency of the air pre-heater is kept constant.

Results

The result of the procedure is shown in Figure 6.7 and the endogenous exergy destruction curve for this component is shown in Figure 6.8.

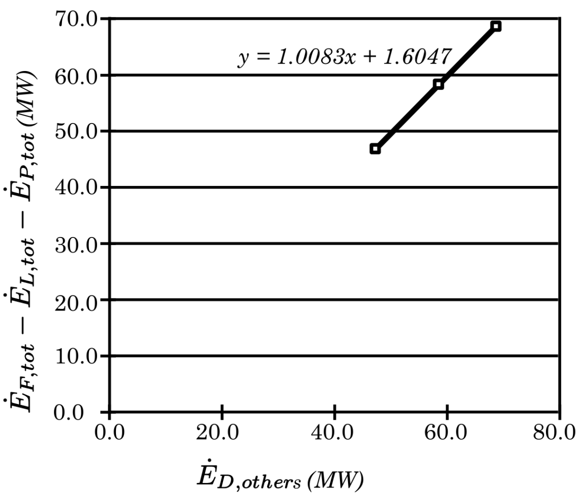


Figure 6.7: Plot showing the results of the procedure used in analyzing the APH

Figure 6.7 shows that at an exergetic efficiency of 88.3% the endogenous exergy destruction of the air pre-heater is 1.6047 MW. The endogenous exergy destruction curve was used to evaluate the endogenous exergy destruction in the APH at the exergetic efficiency of 83.04% from the original cycle. The corresponding endogenous exergy destruction of the latter was found to be 2.42 MW.

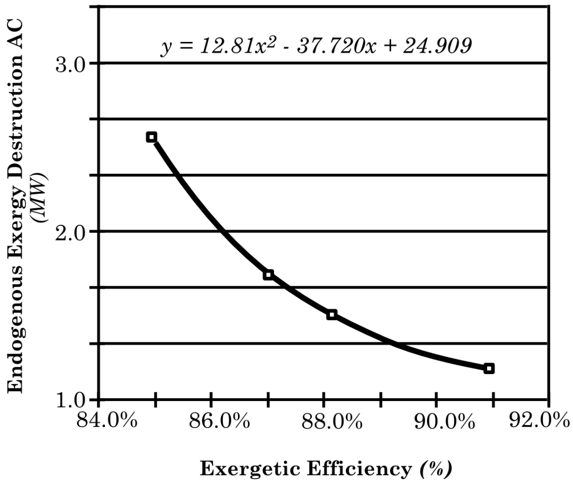


Figure 6.8: Endogenous Exergy Destruction Curve for the APH.

6.2.3 Combustion Chamber

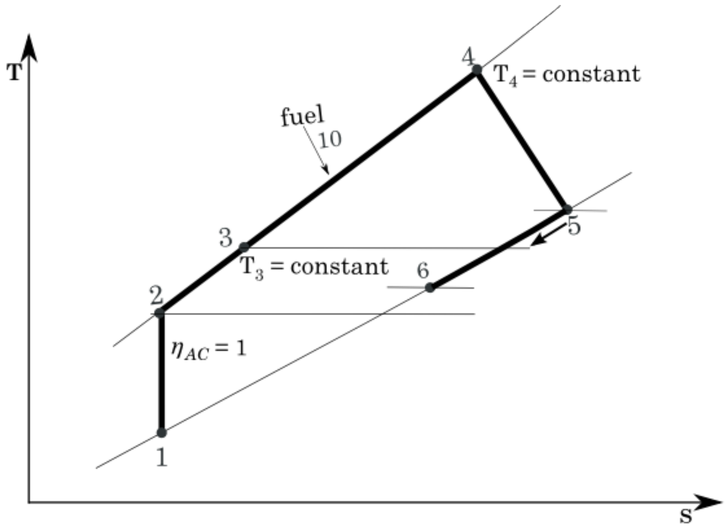


Figure 6.9: T-s diagram of the Cogeneration System illustrating the procedure for analyzing the CC.

Figure 6.9 illustrates the adjustments made to the components for determining the endogenous exergy destruction in a combustion chamber with an exergetic efficiency , note again, that for this procedure, the reversible adiabatic heater cannot be used. The combustion chamber is analyzed by fixing the exergetic efficiency of the air compressor to 1 (i.e. $\dot{E}_{D,AC} = 0$). The exergy destruction in the expander is gradually reduced by increasing its isentropic efficiency, this procedure also reduces the exergy destruction in the APH.

The graph in Figure 6.10 shows the data obtained when analyzing the combustion chamber with exergetic efficiency of 66.76%. From the plot it can be seen that the endogenous exergy destruction of

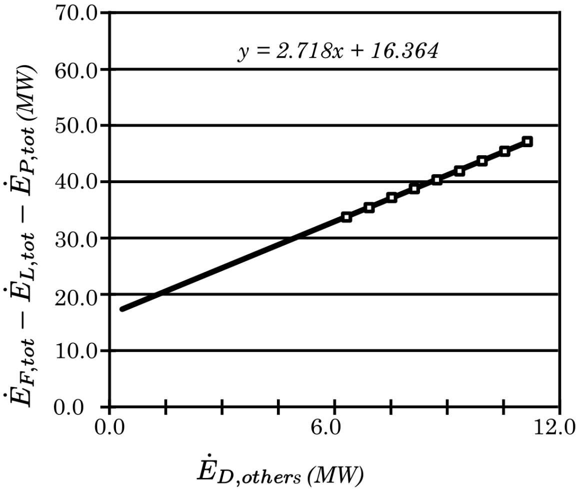


Figure 6.10: Plot showing the results of the procedure used in analyzing the CC

the combustion chamber $\dot{E}^{EN}_{D,CC} = 16.364MW$ at an exergetic efficiency of 66.76%. The endogenous exergy destruction curve is shown in Figure 6.11

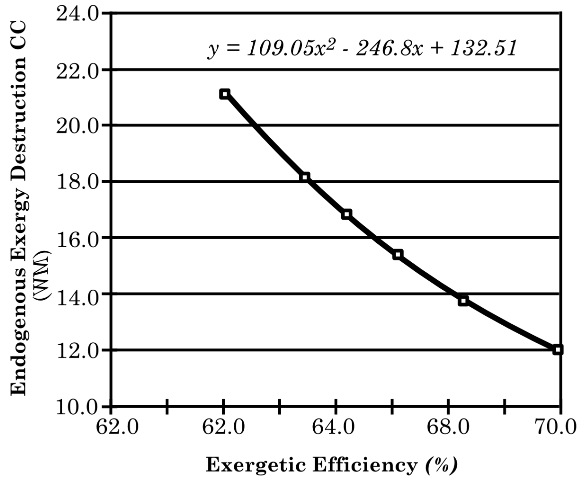


Figure 6.11: Endogenous Exergy Destruction Curve for the CC.

6.2.4 Expander

The adjustments made to the components for determining the endogenous exergy destruction in the expander with an exergetic efficiency, ε_{GT}, is shown in Figure 6.12. The exergy destruction in the combustion chamber is reduced by increasing the exergy supplied by the reversible adiabatic heater.

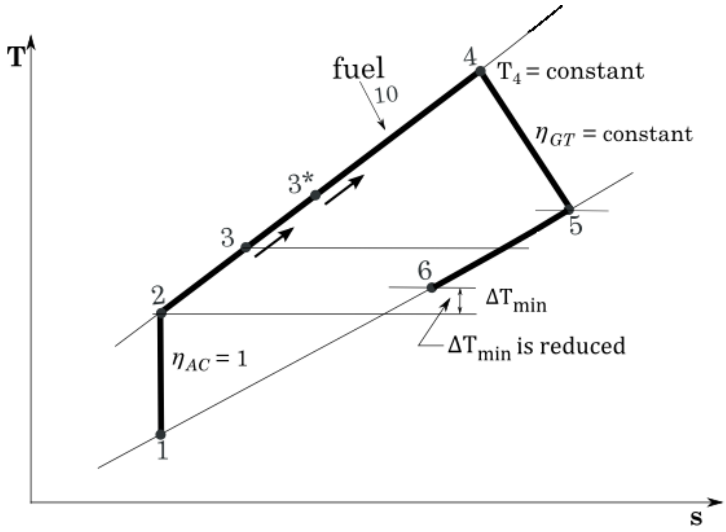


Figure 6.12: T-s diagram of cogeneration system illustrating procedure for analyzing the GT.

The exergetic efficiency of the air compressor is maintained at 100% (i.e. $\dot{E}_{D,AC} = 0$). ΔT_{min} of the APH is gradually reduced while T_4 and the η_{GT} is kept constant.

The results are shown in Figure 6.13 and the endogenous exergy destruction curve for this component is shown in Figure 6.14.

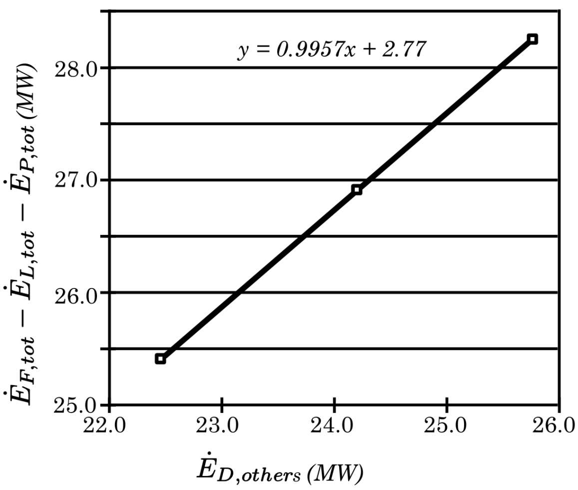


Figure 6.13: Plot showing the results of the procedure used in analyzing the GT

From the plot it can be seen that the endogenous exergy destruction of the expander at an exergetic efficiency of 95.22% is 2.77MW.

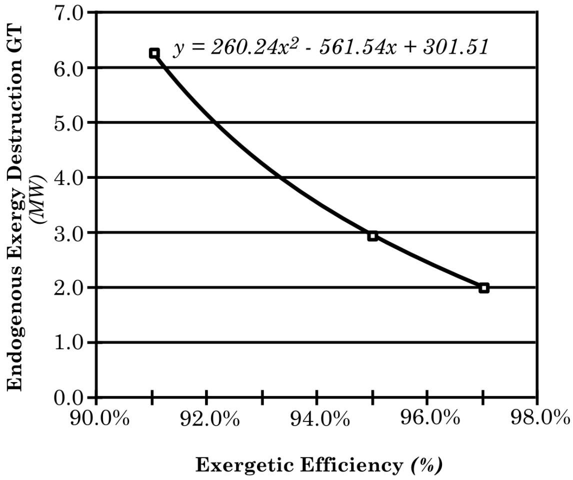


Figure 6.14: Endogenous Exergy Destruction Curve for the GT.

6.2.5 Heat Recovery Steam Generator

The exergy of the product of the HRSG, $\dot{E}_{P,HRSG}$ (system B) must be constant in order to deliver a constant supply of steam at the given requirements. Based on the procedure requirements for the engineering method, the exergetic efficiency of the HRSG will have to remain constant during the process of determining its endogenous value. This therefore implies that $\dot{E}_{D,HRSG}$ will have to be constant as well. However, it is apparent that the exergy destruction in system A affects the term T_6 and, therefore, the exergy destruction in the HRSG. In such a case the engineering approach cannot be applied and a different approach should be used. For this analysis, however, the entire exergy destruction in the HRSG will be taken as endogenous, with the change in exergy destruction in system A affecting only the exergy loss of the entire system .

6.3 Summary of Results

Table 6.3 summarizes the results of the endogenous exergy destruction for each component within the cogeneration power plant.

Table 6.3: Endogenous and Exogenous exergy destruction values for the cogeneration power plant.

Component k	ε_k(%)	$\dot{E}_{D,k}$(MW)	$\dot{E}_{D,k}^{EN}$(MW)	$\dot{E}_{D,k}^{EX}$(MW)
Air Compressor (AC)	93.05%	2.095	1.3184	0.7766
Air Pre-heater (APH)	83.04%	2.819	2.42	0.40
Combustion Chamber (CC)	66.76%	30.509	16.364	14.145
Expander (GT)	95.22%	3.058	2.77	0.288
Heat Recovery Steam Generator (HRSG)	65.48%	6.724	6.724	0.000

6.4 Splitting II: Avoidable and Unavoidable Exergy Destruction

Table 6.4 shows the calculated $(\dot{E}_D/\dot{E}_P)_k^{UN}$ for each component and the assumptions made. All assumptions were taken from reference [30].

Table 6.4: Unavoidable exergy destruction per product exergy in each component.

Component (k)	$(\dot{E}_D/\dot{E}_P)_k^{UN}$	Assumptions
AC	0.054	Purchased equipment cost of the compressor becomes infinite when isentropic efficiency is 90%
APH	0.0164	A minimum temperature of 10K was assumed
CC	0.367	• high temperature of the reactants (811K fuel and 1000K/1360K for air) • High outlet temperature 1773 K/2998K • Adiabatic combustion
GT	0.021	Purchased equipment cost of the expander becomes infinite when isentropic efficiency is 92%
HRSG	0.345	A minimum temperature of 10K was assumed

Table 6.5 shows the unavoidable and avoidable exergy destruction calculated for each component of the cogeneration system.

Table 6.5: Avoidable and unavoidable exergy destruction in each component.

Component k	$\dot{E}_{P,k}$(MW)	$\dot{E}_{D,k}$(MW)	$\dot{E}_{D,k}^{UN}$(MW)	$\dot{E}_{D,k}^{AV}$(MW)
AC	28.039	2.095	1.514	0.581
APH	13.799	2.819	0.226	2.593
CC	61.270	30.509	16.359	14.150
GT	60.914	3.058	1.645	1.413
HRSG	12.754	6.724	4.400	2.324

6.5 Splitting the Exergy Destruction

Table 6.6 shows the values obtained from splitting the exergy destruction in each component of the CGAM power system.

Table 6.6: Splitting of the exergy destruction in the components of the CGAM power system.

k-th Component	Exergy destruction categories (MW)							
	$\dot{E}_{D,k}^{EN}$	$\dot{E}_{D,k}^{EX}$	$\dot{E}_{D,k}^{AV}$	$\dot{E}_{D,k}^{UN}$	$\dot{E}_{D,k}^{EN,AV}$	$\dot{E}_{D,k}^{EN,UN}$	$\dot{E}_{D,k}^{EX,AV}$	$\dot{E}_{D,k}^{EX,UN}$
AC	1.32	0.78	0.58	1.51	0.277	1.04	0.304	0.474
APH	2.42	0.40	2.59	0.23	2.223	0.20	0.370	0.029
CC	16.36	14.15	14.15	16.36	10.713	5.65	3.437	10.708
GT	2.77	0.29	1.41	1.64	1.301	1.47	0.113	0.175
HRSG	6.72	0.00	2.32	4.40	2.324	4.400	0.000	0.000

6.6 Exergoeconomic Evaluation

The conventional exergoeconomic analysis was applied to the cogeneration system and the results are shown in Table 6.7.

Table 6.7: Results of the conventional exergoeconomic analysis for the CGAM cogeneration system.

Component	$c_{F,k}$ (MW)	$c_{P,k}$ (MW)	r_k (%)	f_k (%)	$\dot{C}_{D,k}$ (US$/h)	$\dot{Z}_k$ (US$/h)	$\dot{C}_{D,k} + \dot{Z}_k$ (US$/h)
AC	0.0076	0.0088	14.87	49.75	57.56	56.99	114.55
APH	0.0070	0.0086	23.97	14.78	70.68	12.26	82.94
CC	0.0038	0.0058	50.32	1.04	420.66	4.40	425.06
GT	0.0070	0.0075	8.17	38.58	76.67	48.16	124.83
HRSG	0.0070	0.0110	58.02	9.13	168.59	16.93	185.52

The cost rate of exergy destruction $\dot{C}_{D,k}$ associated with each exergy destruction category is listed in Table 6.8.

Table 6.8: The splitting of the cost of exergy destruction in selected components of the CGAM.

k-th Component	Cost of exergy destruction categories (US$/h)							
	$\dot{C}_{D,k}^{EN}$	$\dot{C}_{D,k}^{EX}$	$\dot{C}_{D,k}^{AV}$	$\dot{C}_{D,k}^{UN}$	$\dot{C}_{D,k}^{EN,AV}$	$\dot{C}_{D,k}^{EN,UN}$	$\dot{C}_{D,k}^{EX,AV}$	$\dot{C}_{D,k}^{EX,UN}$
AC	36.21	21.35	15.96	41.60	7.62	28.59	8.34	13.01
APH	60.68	10.00	65.01	5.67	55.73	4.95	9.28	0.72
CC	225.63	195.03	195.10	225.56	147.71	77.91	47.39	147.65
GT	69.45	7.22	35.44	41.24	32.61	36.84	2.82	4.40
HRSG	168.59	0.00	58.27	110.32	58.27	110.32	0.00	0.00

All economic assumptions and cost equations pertaining to this cogeneration system are given in Appendix C and E.

6.7 Splitting the $\dot{Z}_k$ value

The investment cost was split into avoidable and unavoidable endogenous parts.

6.7.1 Unavoidable Investment Cost , $\dot{Z}^{UN}$

Table 6.9 lists the design values used for determining the unavoidable investment cost for the components of the CGAM cycle.

Table 6.9: Assumptions made in determining the unavoidable investment cost for the CGAM.

Component	Parameter	Design Parameters	Parameters used in determining unavoidable costs
AC	η_{AC}	0.86	0.7
APH	ΔT_{min}	160.83K	400K
CC	ΔP	0.05	0.10
GT	η_{GT}	0.86	0.7
HRSG (parameter applied to the economizer section of the HRSG)	ΔT_{min}	60K	500K

Table 6.10 summarizes the splitting of the investment costs which were calculated in accordance with the procedures given in Appendix D.

Table 6.10: The splitting of the investment costs of selected components in the CGAM.

k-th Component	Cost of exergy destruction categories (US$/h)							
	$\dot{Z}^{EN}_{D,k}$	$\dot{Z}^{EX}_{D,k}$	$\dot{Z}^{AV}_{D,k}$	$\dot{Z}^{UN}_{D,k}$	$\dot{Z}^{EN,AV}_{D,k}$	$\dot{Z}^{EN,UN}_{D,k}$	$\dot{Z}^{EX,AV}_{D,k}$	$\dot{Z}^{EX,UN}_{D,k}$
AC	39.93	17.06	46.92	10.07	33.90	6.03	13.02	4.03
APH	7.02	5.24	4.83	7.43	4.38	2.64	0.44	4.79
CC	2.37	2.04	2.55	1.86	1.23	1.14	1.32	0.72
GT	43.29	4.87	32.17	15.99	29.57	13.73	2.60	2.27
HRSG	16.93	0.00	3.89	13.03	13.03	3.89	0.00	0.00

6.7.2 Exergoeconomic analysis vs. advanced exergoeconomic analysis

The results from the conventional exergoeconomic analysis was compared with the results from the advanced exergoeconomic analysis and shown in Table 6.11.

Table 6.11: Comparison of results between the conventional exergoeconomic analysis and the advanced exergoeconomic analysis.

k-th Component	Exergoeconomic analysis (without splitting)				Advanced exergoeconomic analysis (with endogenous avoidable)		
	$\dot{Z}_k + \dot{C}_{D,k}$ (US$/h)	ε_k (%)	r_k (%)	f_k (%)	$\dot{Z}_k^{EN,AV} + \dot{C}_{D,k}^{EN,AV}$ (US$/h)	$\varepsilon_k^{EN,AV}$ (%)	$f_k^{EN,AV}$ (%)
AC	114.55	93.05	14.87	49.75	41.52	99.02	81.65
APH	82.94	83.04	23.97	14.78	60.11	86.13	7.29
CC	425.06	66.76	50.32	1.04	148.94	85.12	0.83
GT	124.83	95.22	8.17	38.58	62.18	97.91	47.55
HRSG	185.52	65.48	58.02	9.13	71.30	84.59	18.28

Table 6.11 shows the values of both the conventional exergoeconomic variables and the advanced exergoeconomic variables for each component.

Based on the sum $\dot{Z}_k + \dot{C}_{D,k}$ and $\dot{Z}_k^{EN,AV} + \dot{C}_{D,k}^{EN,AV}$ the combustion chamber should be modified first, followed by the heat recovery steam generator and then the gas expander. Both analyses differ, however, when it comes to the air compressor. The conventional exergoeconomic analysis indicates that the air preheater should be improved last whereas the advanced exergoeconomic analysis suggests that the air compressor should be improved last. The latter makes more sense given the additional information obtained in splitting the exergy destruction of the components. In the air preheater approximately 85% of its exergy destruction is endogenous as compared to 63% for the air compressor.

Of the three components targeted to be improved first, the expander has the lowest potential for improvement whereas the heat recovery steam generator has the highest as indicated by the $\varepsilon_k^{EN,AV}$ values.

The low $f_k^{EN,AV}$ value in the combustion chamber indicates that the avoidable endogenous cost associated with this component is due to the high cost of exergy destruction. An increase in its exergetic efficiency will improve the effectiveness of the CGAM plant. This can be done by reducing the pressure drop taking place during its operation, increasing the temperature T_3 (or the amount of heat transfer in the air preheater) and preheating the fuel.

7 Application IV: Combined Power Plant

The fourth application of the Engineering Method was an externally fired combined cycle power plant. The system was developed by the US Department of Energy and analyzed by Cziesla,Tsatsaronis and Gao [6]. For the case shown in Figure 7.1, the plant is designed to deliver 130 MW of power. The working fluid in the gas turbine is air. Compressed air is heated in two high-temperature heat exchangers (C2 and C3). The air leaving the heat exchanger C3 has reached the inlet temperature of the gas turbine (1510 K).

At the turbine outlet the hot air stream is split into two parts. One part is used as pre-heated air in a coal fired combustion chamber which uses coal of a specific exergy of 27.648MJ/kg and which, for safety reasons, operates slightly below atmospheric pressure. The combustion gases exit the combustion chamber at 2273 K. The other part of the exiting hot air stream is directly supplied to a clean heat recovery steam generator.

The bottoming cycle consists of two non-reheat heat recovery steam generators (HRSG) where steam is generated at three pressure levels. A more detailed description of the EFCC can be found in reference [6].

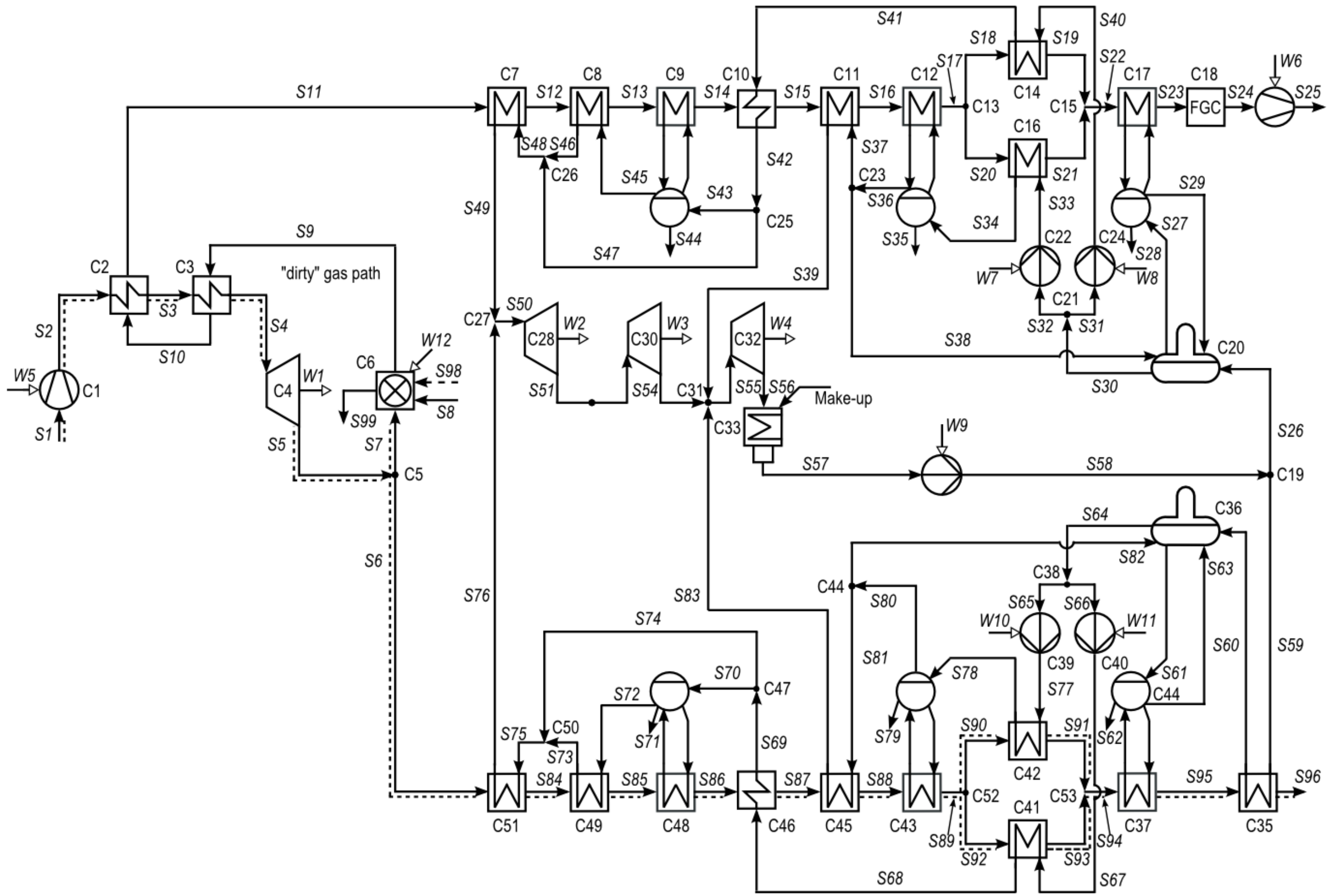


Figure 7.1: Combined Power Plant

Energy and Exergy Analysis

Tables 7.1 to 7.3 list some of the stream information used and the results of the exergy analysis for the simple power plant system.

Table 7.1: Parameters of each stream of the EFCC.

Stream	$\dot{m}$ (kg/s)	T (K)	p (bar)
1	265.73	288.15	1.013
2	265.73	667.34	15.246
3	265.73	1130.00	14.941
4	265.73	1510.00	14.642
5	265.73	861.55	1.043
7	121.85	861.55	1.043
9	131.32	2273.00	0.991
10	131.32	1574.69	0.971
11	131.32	707.76	0.952
13	131.32	683.07	0.949
14	131.32	596.03	0.944
43	8.03	536.52	117.958
44	0.16	571.09	83.414
45	7.87	571.03	83.345
55	37.77	463.14	5.000
56	37.77	309.31	0.060
78	6.04	424.21	21.335
79	0.12	425.50	5.069
80	5.91	424.99	5.000
88	143.87	521.87	1.027
89	143.87	436.99	1.024
98	10.46	288.15	–
99	0.99	1373.00	–

7.1 Splitting I: Exogenous and Endogenous exergy destruction

7.1.1 Air Compressor , $C1$

In this case, $\dot{W}_{net}$ where $\dot{W}_{net} = \dot{W}_1 + \dot{W}_2 + \dot{W}_3 + \dot{W}_4 - \dot{W}_5 - \dot{W}_6 - \dot{W}_7 - \dot{W}_8 - \dot{W}_9 - \dot{W}_{10} - \dot{W}_{11} - \dot{W}_{12}$, is a bit more complex as that defined in equation 3.14. This suggests that the specific work of all the turbomachines all contribute to form the required product of $\dot{W}_{net}$. It also suggests that

$$\dot{E}^{EN}_{D,AC} = \int \left(w_{n(n=1-12)}, \dot{W}_{net}, \dot{m}_a, \dot{m}_g, \dot{m}_{ST}, \dot{\varepsilon}_{AC} \right). \tag{7.1}$$

Table 7.2: Exergy rate for the electric power of the EFCC.

No.	$\dot{E}$ (MW)
W_1	202.81
W_2	6.816
W_3	9.293
W_4	22.315
W_5	104.26
W_6	2.267
W_7	0.02
W_8	0.12
W_9	0.065
W_{10}	0.017
W_{11}	0.282
W_{12}	0.753

Table 7.3: Exergy Analysis of the EFCC.

Component k	$\dot{E}_{F,k}$(MW)	$\dot{E}_{P,k}$(MW)	$\dot{E}_{D,k}$(MW)	ε_k(%)
Air Compressor (C1)	104.26	97.60	6.66	93.62
Heat Exchanger 1 (C2)	102.98	91.16	11.82	88.52
Heat Exchanger 2 (C3)	104.41	93.04	11.37	89.11
Heat Exchanger 1 & 2 (C2 & C3)	207.39	184.19	23.19	88.82
Expander (C4)	211.00	202.81	8.197	96.12
Fossil Boiler ($C6$)	289.83	201.52	88.31	69.53
Steam turbine ($C32$)	25.35	22.32	3.035	88.03
Evaporator (C48)	16.19	13.52	2.66	83.55

Hence, in analyzing this component, the specific work of the turbomachines was kept constant and was set to operate at an isentropic efficiency = 1 with a constant entering temperature. Pressure drops in all other components were removed.

The RAH was used to reduce the exergy destruction and increase the exergetic efficiency in the fossil boiler.

In all heat exchangers, the following requirements were satisfied:

- $\Delta T_{min} \rightarrow 0$
- $T_{H,a} - T_{C,a} \rightarrow 0$

7.1.2 Heat exchangers , $C2$ and $C3$

These components were grouped together to form one component (as demonstrated in the figure below).

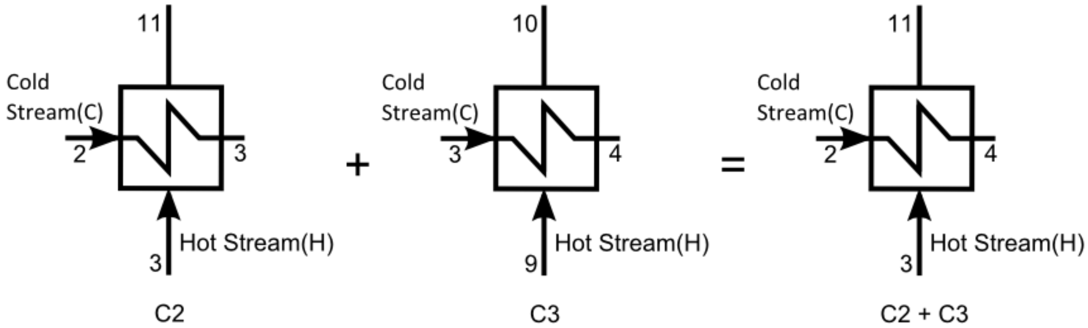


Figure 7.2: Grouped heat exchangers, $C2$ and $C3$.

This combined component was examined with its original pressure drop of 2% across both the hot and cold stream. During the process of determining $\dot{E}^{EN}_{D,C2/C3}$, the turbomachines, with the exception of the air compressor $C1$, were again operating at an isentropic efficiency of 1 and at a constant inlet temperature and pressure. The isentropic efficiency of this the air compressor was gradually increased (i.e. T_2 was reduced).

The exergy destruction in all other components was reduced in accordance with section 3.2.

7.2 Summary of Results

Table 7.4 summarizes the results of the endogenous exergy destruction for each component within the combined power plant.

Table 7.4: Endogenous and Exogenous exergy destruction values for the combined power plant.

Component k	ε_k(%)	$\dot{E}_{D,k}$(MW)	$\dot{E}^{EN}_{D,k}$(MW)	$\dot{E}^{EX}_{D,k}$(MW)
Air Compressor ($C1$)	93.62	6.66	5.64	1.02
Heat Exchangers ($C2$ & $C3$)	88.82	23.19	19.27	3.92
Expander ($C4$)	96.12	8.20	7.53	0.67
Fossil Boiler ($C6$)	69.53	88.31	57.84	30.47
Steam Turbine ($C32$)	88.03	3.04	1.29	1.74
Evaporator ($C48$)	83.55	2.66	1.64	1.03

Additional results obtained from splitting the exergy destruction into its endogenous and exogenous parts can be found in Appendix E.

7.3 Splitting II: Avoidable and Unavoidable Exergy Destruction

The avoidable and unavoidable exergy destruction in the components were calculated and shown in Table 7.5. The unavoidable exergy destruction per product $(\dot{E}_D/\dot{E}_P)^{UN}_k$ calculated by in accordance with the methodology discussed in reference [6]. All assumptions made can be found in Appendix E.

Table 7.5: Avoidable and unavoidable exergy destruction in each component.

Component k	$(\dot{E}_D/\dot{E}_P)^{UN}_k$	$\dot{E}_{P,k}$(MW)	$\dot{E}_{D,k}$(MW)	$\dot{E}^{UN}_{D,k}$(MW)	$\dot{E}^{AV}_{D,k}$(MW)
Air Compressor ($C1$)	0.0306	97.60	6.66	2.99	3.67
Heat Exchangers ($C2$ & $C3$)	–	184.19	23.19	2.54	20.65
Expander ($C4$)	0.0288	202.81	8.197	5.84	2.36
Fossil Boiler ($C6$)	0.3000	201.52	88.31	60.46	27.85
Steam Turbine ($C32$)	0.1035	22.32	3.035	2.31	0.73
Evaporator ($C48$)	0.136	13.52	2.66	1.84	0.82

$\dot{E}^{UN}_{D,C2\&C3}$ for the combined component ($C2$&$C3$) was determined using $(\dot{E}_D/\dot{E}_P)^{UN}_k$ values for the individual components.

7.4 Summarizing the splitting of the exergy destruction

The exergy destruction was then split into the four categorized segments and Table 7.6 summarizes the total splitting of the exergy destruction within each component.

Table 7.6: The splitting of the exergy destruction in selected components in the EFCC.

k-th Component	Exergy destruction categories (MW)							
	$\dot{E}^{EN}_{D,k}$	$\dot{E}^{EX}_{D,k}$	$\dot{E}^{AV}_{D,k}$	$\dot{E}^{UN}_{D,k}$	$\dot{E}^{EN,AV}_{D,k}$	$\dot{E}^{EN,UN}_{D,k}$	$\dot{E}^{EX,AV}_{D,k}$	$\dot{E}^{EX,UN}_{D,k}$
Air Compressor ($C1$)	5.64	1.02	3.67	2.99	3.24	2.40	0.43	0.58
Heat Exchangers ($C2$ & $C3$)	19.27	3.92	20.65	2.54	17.48	1.80	3.18	0.74
Expander ($C4$)	7.53	0.67	2.36	5.84	2.13	5.39	0.22	0.45
Fossil Boiler ($C6$)	57.84	30.47	27.85	60.46	16.08	41.76	11.77	18.70
Steam Turbine ($C32$)	1.29	1.74	0.73	2.31	0.28	1.01	0.44	1.30
Evaporator ($C48$)	1.64	1.03	0.82	1.84	0.58	1.06	0.25	0.78

7.5 Splitting of the costs associated with the EFCC

Both the cost of exergy destruction and the investment costs were also split and the results placed in the Table 7.8 and 7.9 respectively. The specific cost factors used in calculating the latter are shown in Table 7.7. All economic assumptions made and cost equations used can be found in Appendix B, D and E.

Table 7.7: Specific costs and Total costs of selected components of the EFCC.

k-th Component	c_F ($/MJ)	c_P ($/MJ)	C_D ($/h)	$\dot{Z}$ ($/h)
Air Compressor ($C1$)	0.0100	0.0112	238.74	201.40
Heat Exchanger ($C2$)	0.0034	0.0060	142.97	241.50
Heat Exchanger ($C3$)	0.0034	0.0068	137.54	525.78
Expander ($C4$)	0.0086	0.0100	254.29	728.64
Fossil Boiler ($C6$)	0.0025	0.0044	779.15	615.36
Steam Turbine ($C32$)	0.0102	0.0150	111.88	269.09
Evaporator ($C48$)	0.0082	0.0109	78.95	52.44

Table 7.8: The splitting of the cost of exergy destruction in selected components of the EFCC.

k-th Component	Cost of exergy destruction categories, (US$/h)							
	$\dot{C}_{D,k}^{EN}$	$\dot{C}_{D,k}^{EX}$	$\dot{C}_{D,k}^{AV}$	$\dot{C}_{D,k}^{UN}$	$\dot{C}_{D,k}^{EN,AV}$	$\dot{C}_{D,k}^{EN,UN}$	$\dot{C}_{D,k}^{EX,AV}$	$\dot{C}_{D,k}^{EX,UN}$
Air Compressor ($C1$)	202.27	36.47	131.62	107.13	116.04	86.23	15.58	20.89
Heat Exchangers ($C2$ & $C3$)	233.12	47.40	249.80	30.72	211.33	21.79	38.47	8.93
Expander ($C4$)	233.54	20.75	73.09	181.20	66.23	167.31	6.87	13.88
Fossil Boiler ($C6$)	510.33	268.82	245.75	533.40	141.90	368.43	103.85	164.97
Steam Turbine ($C32$)	47.62	64.26	26.74	85.14	10.41	37.20	16.33	47.93
Evaporator ($C48$)	48.54	30.41	24.41	54.55	17.06	31.47	7.34	23.07

Table 7.9: The splitting of the investment costs of selected components in the EFCC.

k-th Component	Investment cost categories, (US$/h)							
	$\dot{Z}_{D,k}^{EN}$	$\dot{Z}_{D,k}^{EX}$	$\dot{Z}_{D,k}^{AV}$	$\dot{Z}_{D,k}^{UN}$	$\dot{Z}_{D,k}^{EN,AV}$	$\dot{Z}_{D,k}^{EN,UN}$	$\dot{Z}_{D,k}^{EX,AV}$	$\dot{Z}_{D,k}^{EX,UN}$
Air Compressor ($C1$)	170.44	30.95	49.56	151.84	42.14	128.31	7.43	23.53
Heat Exchangers ($C2$ & $C3$)	252.31	514.97	330.84	436.44	219.46	32.86	111.39	403.58
Expander ($C4$)	654.71	73.93	569.79	158.85	549.25	105.45	20.54	53.40
Fossil Boiler ($C6$)	366.90	248.46	322.51	292.85	189.42	177.49	133.09	115.36
Steam Turbine ($C32$)	80.37	188.72	178.64	90.45	58.68	21.69	119.96	68.76
Evaporator ($C48$)	16.45	35.99	8.89	43.55	4.18	12.28	4.71	31.27

7.6 Exergoeconomic and advanced exergoeconomic evaluation

A comparison was done between the conventional exergoeconomic analysis and the advanced exergoeconomic analysis on the system. The results are shown in Table 7.10.

Table 7.10: Comparison of results between the conventional exergoeconomic analysis and the advanced exergoeconomic analysis.

k-th Component	Exergoeconomic analysis (without splitting)				Advanced exergoeconomic analysis (with endogenous avoidable)		
	$\dot{Z}_{D,k} + \dot{C}_{D,k}$ (US$/h)	ε_k (%)	r_k (%)	f_k (%)	$\dot{Z}_{D,k}^{EN,AV} + \dot{C}_{D,k}^{EN,AV}$ (US$/h)	$\varepsilon_k^{EN,AV}$ (%)	$f_k^{EN,AV}$ (%)
Air Compressor ($C1$)	440.14	93.62	12.57	45.76	158.18	96.79	26.64
Heat Exchangers ($C2$ & $C3$)	1047.79	88.82	45.15	73.23	430.80	91.33	50.94
Expander ($C4$)	982.93	96.12	15.62	74.13	615.48	98.96	89.24
Fossil Boiler ($C6$)	1394.51	69.53	78.43	44.13	331.32	92.61	57.17
Steam Turbine ($C32$)	380.97	88.03	46.31	70.63	69.09	98.76	84.93
Evaporator ($C48$)	131.39	83.55	32.76	39.91	21.24	95.89	19.68

In the EFCC power plant, analysis was only performed on components with the highest exergy destruction rates. Table 7.10 shows the values from both exergoeconomic analyses.

The conventional exergoeconomic analysis indicates that the fossil boiler ($C6$) and the heat exchangers($C2$ & $C3$) should be improved first, however, consideration of the avoidable endogenous costs, $\dot{Z}_{D,k}^{EN,AV} + \dot{C}_{D,k}^{EN,AV}$, indicates that the expander and the heat exchangers should be improved first.

The comparison of the modified exergetic efficiencies, $\varepsilon_k^{EN,AV}$, identifies the heat exchangers ($C2$ & $C3$) as having the largest potential for improving their thermodynamic performance. Though the expander has the highest avoidable endogenous cost, the potential of improving this component is very low.

The $f_k^{EN,AV}$ values indicate that the cost effectiveness of the EFCC power plant might be improved by reducing the capital investment for the expander and the steam turbine ($C32$). The air compressor ($C1$) has a relatively low $f_k^{EN,AV}$ value since the avoidable endogenous costs associated with this component is due to the avoidable-endogenous cost of exergy destruction. An increase in the exergetic efficiency of this component might reduce the overall product cost of the plant even if the cost associated with the capital cost investment will increase.

8 Comparison of the Engineering Method with other Approaches

The engineering method is actually the first method which can be used to accurately determine the endogenous and exogenous exergy destruction rates associated with a component. Though other approaches could be applied to determine such interaction they either have assumptions which may undermine the integrity of the results obtained or cannot at this time be applied to all thermal systems. Some of these alternative methods developed in this research work will now be discussed. All methods were applied to the simple gas turbine described in chapter 5.

Alternative Method 1: Use of symbolic mathematics proposed in [33]

This method uses an algebraic formula which defines the exergy destruction of the total system. The main assumption behind this method is that the product of the previous system becomes the fuel of the subsequent system. The latter allows one to create a fuel/product or <FP> matrix which can be used to mathematically define the exergy destruction in each component.

The total exergy destruction of any given system is defined as follows:

$$\dot{E}_{D,system} = \sum_{i}^{n} \dot{E}_{D,i} = {}^{t}\mathbf{U}\mathbf{E}_{\mathbf{D,system}} = {}^{t}\mathbf{U}(\mathbf{K}_D - \mathbf{U}_D)\mathbf{P}. \tag{8.1}$$

where

$\dot{E}_{D,system}$	exergy destruction rate of the entire system (MW)
i	i-th component within the system
n	number of components within the system
$\dot{E}_{D,i}$	exergy destruction rate of the i-th component (MW)
${}^{t}\mathbf{U}$	transpose unit vector
$\mathbf{E}_{\mathbf{D,system}}$	matrix vector of the exergy destruction rate of the entire system
$\mathbf{K}_D$	inverse diagonal matrix containing the exergetic efficiency of each component
$\mathbf{P}$	matrix vector of the product exergy from each component

Applying Equation (8.1) to the simple power plant, the following equations for the exergy destruction in each component were obtained

$$\dot{E}_{D,AC} \approx \frac{(\varepsilon_{AC}-1)\left(\dot{E}_1+y_3\dot{W}_{net}\right)}{-1+y_3} \tag{8.2}$$

$$\dot{E}_{D,CC} \approx \frac{(1-\varepsilon_{CC})\left(\dot{E}_1\left(-1+y_2\varepsilon_{AC}\varepsilon_{GT}\right)+\left(-1+y_2y_3\varepsilon_{AC}\varepsilon_{CC}\right)\dot{W}_{net}\right)}{y_2\left(-1+y_3\right)\varepsilon_{CC}\varepsilon_{GT}} \tag{8.3}$$

$$\dot{E}_{D,GT} \approx \frac{(\varepsilon_{GT}-1)\left(\dot{E}_1+\dot{W}_{net}\right)}{\left(-1+y_3\right)\varepsilon_{GT}}. \tag{8.4}$$

where

y_2 Portion of exergy available for processing in the turbine, $1-\dot{E}_5/\dot{E}_4$

y_3 Portion of total turbine output that is used in driving the shaft, $\dot{W}_{AC}/(\dot{W}_{AC}+\dot{E}_{P,tot})$

y_2 and y_3 are assumed constant and calculated at the design operating conditions of the plant.

The endogenous exergy destruction in each component is found by setting the exergetic efficiency of the other components in the system to 1.

Table 8.1 compares the results obtained by applying the engineering method and the symbolic algebraic method to the simple gas turbine. The results of the symbolic mathematical method imply that

Table 8.1: Comparison of the endogenous exergy destruction values obtained by applying the symbolic algebraic method and the engineering method to the simple cycle.

			Endogenous Exergy Destruction (MW)		
Component	Exergetic Efficiency(%)	Exergy Destruction(MW)	Engineering Method	Symbolic Algebraic Method	difference (%)
Air Compressor (CM)	90.02	4.88	4.20	4.88	16.3
Combustion Chamber(CC)	62.19	42.39	27.85	35.05	25.9
Expander(GT)	94.72	4.44	3.584	4.40	22.6

the endogenous exergy destruction in both the air compressor and the expander is approximately equal to the total exergy destruction in these components. This corresponds to the expressions for the exergy destruction given in Equations (8.2) and (8.4). These expressions suggest that all the exergy destroyed in these two components are endogenous i.e. there is no exogenous exergy destruction. This however contradicts knowledge of the operation of actual power plant systems, since irreversibilities in any component will affect the mass flow rate of the material streams and hence impact the irreversibilities of other components in the system.

Another discrepancy occurs with the value obtained for the endogenous exergy destruction of the combustion chamber. As stated in chapter 4 this value can be easily achieved via a simulation software program, by setting the exergetic efficiency of both the air compressor and the expander to 1. In Table 5.3 this value from the simulation program was found to be 28.54 which indicates a 2% difference from the results obtained from the engineering method but 22.8% difference from the results obtained from the symbolic algebraic method.

At present it does not appear that this approach could provide useful results with respect to calculating the endogenous and exogenous exergy destruction. It cannot be excluded, however, that through further work on the selection of the constants employed in this approach, results of acceptable accuracy could be obtained.

Alternative Method 2: Mass balance method

In this method at ideal operation the energy balance of the combustion chamber is ignored since it cannot be fulfilled in the ideal case.

An exergy balance for the combustion chamber is as follows:

$$\left(\dot{m}_2 + \dot{m}_3\right) e_4 - e_2\dot{m}_2 = \dot{m}_3 e_3. \tag{8.5}$$

now both $\dot{m}_3$ and e_4 is a function of the air fuel ratio, λ

where

$$\lambda = \frac{\dot{m}_2}{\dot{m}_3}. \tag{8.6}$$

The ideal condition of the combustion chamber is therefore determined at the value λ which satisfies the equation

$$\dot{m}_3 = \frac{\dot{m}_2(e_4 - e_2)}{(e_3 - e_4)}. \tag{8.7}$$

note however the exit temperature T_4 is also kept constant.

Table 8.2 compares the results obtained by applying the thermodynamic method and the mass balance method to the simple power cycle. In this method the impact of ignoring the energy balance

Table 8.2: Comparison of the endogenous exergy destruction values obtained by applying the mass balance method and the engineering method to the simple cycle.

			Endogenous Exergy Destruction (MW)		
Component	Exergetic Efficiency(%)	Exergy Destruction(MW)	Engineering Method	Mass Balance Method	difference (%)
Air Compressor(CM)	90.02	4.88	4.20	3.740	11.00
Combustion Chamber(CC)	62.19	42.39	27.850	27.85	0.00
Expander(GT)	94.72	4.44	3.584	3.406	4.97

when defining an ideal combustion chamber on the value of the endogenous exergy is unknown.

Alternative Method 3: Exergy balance method

In this method an ideal combustion chamber is defined as one where the exergy balanced is fulfilled. In this case the ideal combustion chamber is defined as follows:

$$\dot{E}_2 + \dot{E}_{3*} = \dot{E}_4. \tag{8.8}$$

where $\dot{E}_{3*}$ is the exergy of the fuel required to satisfy Equation (8.8). Again the temperature of the material stream entering the expander is maintained at T_4.

Table 8.3 compares the results obtained by applying the engineering method and the exergy balance method to the simple power cycle.

Table 8.3: Comparison of the endogenous exergy destruction values obtained by applying the mass balance method and the thermodynamic method to the simple cycle.

			Endogenous Exergy Destruction (MW)		
Component	Exergetic Efficiency(%)	Exergy Destruction(MW)	Engineering Method	Exergy Balance Method	difference (%)
Air Compressor(CM)	90.02	4.88	4.20	4.30	2.40
Combustion Chamber(CC)	62.19	42.39	27.85	27.85	0.00
Expander(GT)	94.72	4.44	3.584	3.623	1.09

Table 8.4: Comparison of the endogenous exergy destruction values obtained by applying the equivalent component method and the engineering method to the simple cycle.

Component	Exergetic Efficiency(%)	Exergy Destruction(MW)	Endogenous Exergy Destruction (MW) Engineering Method	Equivalent Component Method	difference (%)
Air Compressor(CM)	90.02	4.88	4.20	4.00	4.80
Combustion Chamber(CC)	62.19	42.39	27.85	27.85	0.00
Expander(GT)	94.72	4.44	3.584	3.463	3.38

This method seems very similar to the engineering method and should be further investigated on larger systems.

Alternative Method 4: Equivalent Component method

In this method the ideal combustion chamber is approximated to an ideal heat exchanger. Hence Figure 3.1 will look more like Figure 8.1. All state point pressure and temperature values were maintained and the working fluid used throughout the cycle was air.

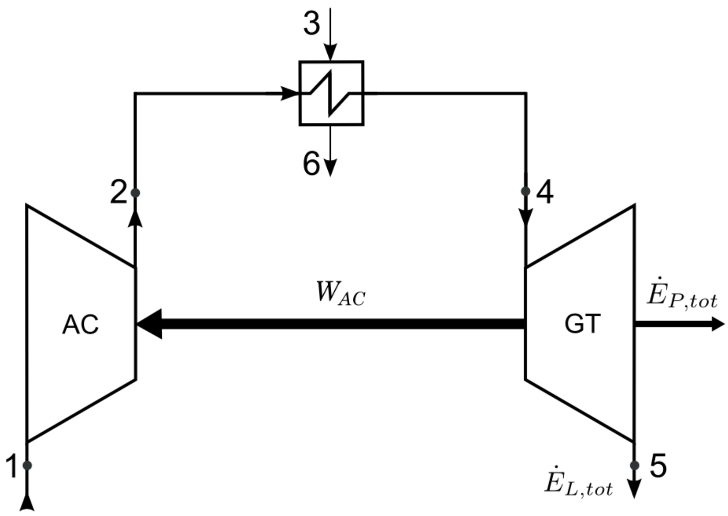


Figure 8.1: Schematic diagram of the simple cycle where the ideal combustion chamber is approximate to an ideal heat exchanger.

Table 8.4 compares the results obtained by applying the engineering method and the equivalent component method to the simple power cycle.

The impact of such an approximation on the endogenous exergy values seems very minimal however further investigations should be carried out on larger systems.

Table 8.5: Summary of the endogenous exergy destruction values obtained by applying various methods to the simple cycle.

Component	Exergetic Efficiency(%)	Exergy Destruction(MW)	Endogenous Exergy Destruction (MW) Engineering Method	Symbolic Algebraic Method	Mass Balance Method	Exergy Balance Method	Equivalent Component Method
Air Compressor (CM)	90.02	4.88	4.20	4.88	3.74	4.30	4.00
Combustion Chamber(CC)	62.19	42.39	27.85	35.05	27.85	27.85	27.85
Expander(GT)	94.72	4.44	3.584	4.40	3.406	3.623	3.463

The accuracy of the engineering method was also clearly seen when its results were compared to the results obtained from the thermodynamic method. In Chapter 4 both methods were applied to a refrigeration system and the results were quite similar. Table 8.6 lists the advantages and disadvantages of all approaches discussed.

Table 8.6: The advantages and disadvantages of each approach.

Approach	Advantage	Disadvantage
Engineering or "graph" method	An accurate procedure which can be applied to both simple and complex thermal systems.	Unable to determine the endogenous exergy destruction for dissipative devices such as throttle valves. A long and perhaps time consuming procedure.
Thermodynamic method	This method can be applied to all thermal systems.	The application to power plant systems has not as yet been demonstrated. In addition, this method requires the ideal operation of a component to be defined. The ideal operation of some components can only be approximately determined.
Other Approaches		
• Symbolic mathematics	• Solutions can be quickly obtained.	• The method does not use a simulation to determine the changing mass flow rates as the exergy destruction changes in the components. By operating only with exergy values, this approach needs additional arbitrary assumptions that greatly affect the obtained results, particularly in complex designs.
• Algebraic methods which define the ideal operations for reactors (e.g. combustion chambers and fossil boilers etc.)	• Solutions can be quickly determined.	• The assumptions made for these methods affect the accuracy of the results obtained. However the results obtained from the Exergy balance method were very close to the results obtained from the Engineering method.

9 Discussion and Conclusion

9.1 The Engineering or "graph" method

The engineering or "graph" method is a basic step by step method for determining the interactions among the components of a thermal system. It is founded on the basis of the exergy balance equation $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot} = \dot{E}_{D,k} + \dot{E}_{D,others}$. This method is based on the understanding that as the exergy destruction in the other components decreases or tends to zero the expression $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot}$ approaches the value of the endogenous exergy destruction of the k-th component, $\dot{E}_{D,k}$. This approach however requires that the exergetic efficiency of the k-th component be kept constant as the exergy destruction in the other components decreases. The latter is also a limitation for the application of such a method to dissipative components such as throttle devices, in which exergy is destroyed without having a productive effect. Keeping the exergetic efficiency constant also poses a limitation in cases where the product exergy of the component must be held constant. Such was the case with the HRSG in the CGAM system in chapter 6.

The method also employs the use of, what was termed in this thesis, a "reversible" adiabatic heater to address conflicts between components when reducing the exergy destruction, since the reduction of the exergy destruction in one component may cause an increase in the exergy destruction in an adjacent component.

9.1.1 Further work on the engineering method

The concept of the engineering method can be used to develop models for determining the endogenous and exogenous exergy destruction in thermal systems. Such models can greatly reduce the time spent in attaining the final results. There is a need to also apply this methodology to chemical processes in order to fully appreciate its usefulness.

9.2 Splitting the exergy destruction

The concept of splitting the exergy destruction for the most important components can further assist engineers in deciding whether an adjustment in a specified component or in the remaining components is required to improve the overall system.

In Chapter 5, the effect of the exergy destruction in each component on the exogenous exergy destruction of the component understudy was determined. It was shown that the exogenous exergy destruction of a component is also made up of a combined effect of the exergy destruction in all the components in the system; mexogenous exergy destruction. This further splitting of the exogenous exergy destruction helped identified components which play a major role in the overall system efficiency.

9.3 Advanced exergoeconomic analysis

The concept of splitting the exergy destruction rates of a component was then applied to the costs and factors of the conventional exergoeconomic analysis. These costs and factors were modified to produce what is now being termed an advanced exergoeconomic analysis. The advanced exergoeconomic analysis was applied to three power systems; a simple power system, a cogeneration power system and a combined power system. In each case the splitting of the exergy destruction and costs improved the results obtained from the conventional exergoeconomic analysis.

An advanced exergoeconomic evaluation focuses only on the endogenous avoidable inefficiencies and costs of each component within the thermal system. It utilizes a modified exergetic efficiency, $\varepsilon^{EN,AV}$ and exergoeconomic factor, $f^{EN,AV}$, and the endogenous-avoidable parts of the cost of exergy destruction, $\dot{C}_D^{EN,AV}$ and investment cost, $\dot{Z}_D^{EN,AV}$. The use of these variables allows for the simultaneous evaluation of a component and its interactions with the other components within the thermal system thereby increasing the certainty with which conclusions are obtained. It better assists the designer in the iterative cost minimization process better than the results obtained from the conventional analysis since a more comprehensive understanding of the inefficiencies within each component is known.

It is important to note that the results from an advanced exergoeconomic analysis and evaluation are based on estimated values of the current technological limitations and cost of a component.

Hence care must be exercised in selecting these estimated values so as not to significantly affect the conclusions drawn from the analysis.

Appendices

A Proof of linear dependence between $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot}$ and $\dot{E}_{D,other}$

Consider the following theoretical process:

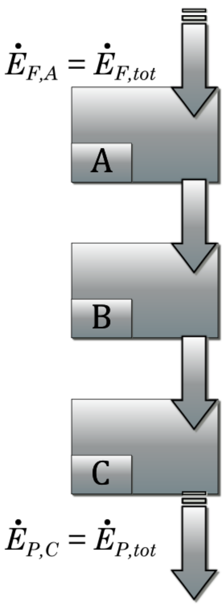


Figure A.1: Theoretical process.

where

$$\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot} = \overbrace{\frac{\dot{E}_{P_{tot}}}{\varepsilon_C \varepsilon_B}\left(\frac{1}{\varepsilon_A} - 1\right) \overbrace{+ \frac{\dot{E}_{P_{tot}}}{\varepsilon_C}\left(\frac{1}{\varepsilon_B} - 1\right) + \dot{E}_{P_{tot}}\left(\frac{1}{\varepsilon_C} - 1\right)}^{f_2(x)}}^{f_1(x)} \tag{A.1}$$

$$\text{let variable } x = \frac{1}{\varepsilon_C \varepsilon_B} \tag{A.2}$$

If linear dependence exists between two functions, $f_1(x)$ and $f_2(x)$, then any two selected values a_1 and a_2, where a_1 and a_2 are real numbers and

$$a_1 f_1(x) + a_2 f_2(x) = 0 \tag{A.3}$$

should not be both equal to zero [3].

Considering the theoretical process in Figure A.1 and Equation (A.3), and given that the endogenous exergy destruction in component A is being investigated, then, $f_1(x)$ represents $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot}$ and $f_2(x)$ represents $\dot{E}_{D,others}$ where,

$$f_1(x) = \frac{\dot{E}_{P_{tot}}}{\varepsilon_C \varepsilon_B}\left(\frac{1}{\varepsilon_A} - 1\right) + \frac{\dot{E}_{P_{tot}}}{\varepsilon_C}\left(\frac{1}{\varepsilon_B} - 1\right) + \dot{E}_{P_{tot}}\left(\frac{1}{\varepsilon_C} - 1\right) \tag{A.4}$$

$$f_2(x) = \frac{\dot{E}_{P_{tot}}}{\varepsilon_C}\left(\frac{1}{\varepsilon_B} - 1\right) + \dot{E}_{P_{tot}}\left(\frac{1}{\varepsilon_C} - 1\right) \tag{A.5}$$

Equations (A.4) and (A.5) in the form of Equation (A.3) becomes

$$\begin{aligned} & a_1 \frac{\dot{E}_{P_{tot}}}{\varepsilon_C \varepsilon_B}\left(\frac{1}{\varepsilon_A}\right) - a_1 \dot{E}_{P_{tot}} + a_2 \frac{\dot{E}_{P_{tot}}}{\varepsilon_C \varepsilon_B} - a_2 \frac{\dot{E}_{P_{tot}}}{1} = 0 \\ & \quad \Rightarrow a_1 \frac{\dot{E}_{P_{tot}}}{\varepsilon_A} x - a_1 \dot{E}_{P_{tot}} + a_2 \dot{E}_{P_{tot}} x - a_2 \dot{E}_{P_{tot}} = 0. \end{aligned} \tag{A.6}$$

Differentiating Equation (A.6), we get;

$$a_1 \frac{\dot{E}_{P_{tot}}}{\varepsilon_A} + a_2 \dot{E}_{P_{tot}} = 0 \tag{A.7}$$

or

$$a_1 \frac{1}{\varepsilon_A} + a_2 = 0. \tag{A.8}$$

Therefore, a_1 can be 1 and

a_2 can be $-1/\varepsilon_A$

Hence, there exists a combination of non zero values for a_1 and a_2. Therefore linear dependence exists.

B Investment Cost Equations

Table B.1: Equations for calculating the investment costs(I) of the components.

Component	Equation
Air Compressor(AC)	$I_{AC} = \left(\frac{C_{11}\dot{m}_a}{C_{12}-\eta_{AC}}\right)\left(\frac{P_{out}}{P_{in}}\right)\ln\left(\frac{P_{out}}{P_{in}}\right)$
Combustion Chamber(CC)	$I_{CC} = \left(\frac{C_{21}\dot{m}_a}{C_{22}-P_{out}/P_{in}}\right)\left[\exp(C_{23}T_{out} - C_{24})\right]$
Expander/Steam Turbine (GT/ST)	$I_{GT} = \left(\frac{C_{31}\dot{m}_g}{C_{32}-\eta_{GT}}\right)\ln\left(\frac{P_{in}}{P_{out}}\right)\left[\exp(C_{33}T_{in} - C_{34})\right]$
Heat Exchanger(HE)	$I_{HE} = \left(\frac{\dot{m}_g(h_{hot,in}-h_{hot,out})}{U(\Delta TLM)}\right)^{0.6}$
Evaporator(EV)	$I_{EV} = I_{o,EV}\left[\left(\frac{\dot{C}_{W,EV}}{U_{EV}A_{o,EV}}\right) - \ln(1-\varepsilon_{EV})\right]\frac{Q_{EV}}{T_0}$
Heat-Recovery Steam Generator (HRSG)	$I_{HRSG} = C_{51}\left[\left(\frac{\dot{Q}_{PH}}{(\Delta TLM)_{PH}}\right)^{0.8} + \left(\frac{\dot{Q}_{EV}}{(\Delta TLM)_{EV}}\right)^{0.8}\right] + C_{52}\dot{m}_{st} + C_{53}\dot{m}_g^{1.2}$

$\dot{m}_a$, $\dot{m}_g$, $\dot{m}_{st}$ are the mass flow rates of air, gas and steam respectively. $h_{hot,in}$ and $h_{hot,out}$ are the specific enthalpies of the inlet and outlet hot stream. ΔTLM is the log mean temperature difference. $\dot{C}_{W,EV} = U_{EV}A_{EV}$. $\dot{Q}_{PH}$ and $\dot{Q}_{EV}$ represents the rate of heat transfer in the preheater (economizer) and evaporator respectively. [32, 34]

Table B.2: Constants used for the investment cost equations for the components in the simple gas turbine.

Component	Equation
Air Compressor(AC)	$C_{11} = 73.00 \quad \$/(kg/s) \quad C_{12} = 0.9$
Combustion Chamber(CC)	$C_{21} = 1.50 \quad \$/(kg/s) \quad C_{22} = 1.00$ $C_{23} = 0.018 \quad \$/(K^{-1}) \quad C_{24} = 26.4$
Expander(GT)	$C_{31} = 245.00 \quad \$/(kg/s) \quad C_{32} = 0.92$ $C_{33} = 0.036 \quad \$/(K^{-1}) \quad C_{34} = 54.4$

Unit cost of methane is US$4/GJ-LHV

Table B.3: Constants used for the investment cost equations for the components in the cogeneration plant.

Component	Equation
Air Compressor(AC)	$C_{11} = 39.5 \quad \$/(kg/s) \quad C_{12} = 0.9$
Combustion Chamber(CC)	$C_{21} = 25.6 \quad \$/(kg/s) \quad C_{22} = 0.995$
	$C_{23} = 0.018 \quad \$/(K^{-1}) \quad C_{24} = 26.4$
Expander(GT)	$C_{31} = 266.3 \quad \$/(kg/s) \quad C_{32} = 0.92$
	$C_{33} = 0.036 \quad \$/(K^{-1}) \quad C_{34} = 54.4$
Heat Exchanger(HE)	$C_{41} = 2290 \quad \$/(m^{1.2}) \quad U = 0.9kW/(m^2K)$
Heat-Recovery Steam Generator	$C_{51} = 3650 \quad \$/(kW/K)^{0.8} \quad C_{52} = 11820 \quad \$/(kg/s)$
(HRSG)	$C_{53} = 658 \quad \$/(kg/s)^{1.2}$

Table B.4: Constants used for the investment cost equations for the components in the combined cycle plant.

Component	Equation
Air Compressor(AC)	$C_{11} = 94.11 \quad \$/(kg/s) \quad C_{12} = 0.9$
Fossil Boiler(FB)	$C_{21} = 0.0144 \quad \$/(kg/s) \quad C_{22} = 0.995$
	$C_{23} = 0.018 \quad \$/(K^{-1}) \quad C_{24} = 26.4$
Expander (GT)	$C_{31} = 1411.61 \quad \$/(kg/s) \quad C_{32} = 0.92$
	$C_{33} = 0.036 \quad \$/(K^{-1}) \quad C_{34} = 54.4$
Steam Turbine (ST)	$C_{31} = 8591.00 \quad \$/(kg/s) \quad C_{32} = 0.92$
	$C_{33} = 0.036 \quad \$/(K^{-1}) \quad C_{34} = 54.4$
Evaporator (EV)	$I_{o,EV} = 823.19 \quad \$/(m^2) \quad A_{o,EV} = 100(m^2K)$
Heat Exchanger 2(HE)	$C_{41} = 69227.80 \quad \$/(m^{1.2}) \quad U = 0.03kW/(m^2K)$
Heat Exchanger 3(HE)	$C_{41} = 346182.18 \quad \$/(m^{1.2}) \quad U = 0.03kW/(m^2K)$

Unit cost of coal is US$2.48/GJ-LHV

C Procedure for splitting the owning and operating cost rate , $\dot{Z}_k$

The investment cost of the k-th component can be defined as:

$$\dot{Z}_k = (\beta + \gamma)\,\frac{I_k}{\tau}. \tag{C.1}$$

where
β capital recovery factor 18.2%
γ operating and maintenance costs (excluding fuel costs) 1.092%
I_k investment cost for k-th component (see Table B.1- B.4)
τ annual system operation time at nominal capacity (8000 hours)

Example C.0.1 (Splitting the owning and operating cost rate of the air compressor in the simple power plant).

From Table B.1

$$I_{AC} = \left(\frac{C_{11}\dot{m}_a}{C_{12} - \eta_{AC}}\right)\left(\frac{P_2}{P_1}\right)\ln\left(\frac{P_2}{P_1}\right). \tag{C.2}$$

Determining the unavoidable/avoidable cost rates, $\dot{Z}_{AC}^{UN}$ and $\dot{Z}_{AC}^{AV}$

- determine the lowest possible available isentropic efficiency, $\eta_{AC,min}$
- simulate the simple power plant with the air compressor operating at $\eta_{AC,min}$ and all other components operating at their worst design parameters. Adjust the mass flow rate of air through the system $\dot{m}_a$ that the required power output will be fulfilled.
- determine the unavoidable investment cost I_{AC}^{UN} by substituting $\eta_{AC,min}$ and $\dot{m}_a$ into Equation (C.2)
- calculate the unavoidable product exergy of the air compressor $\dot{E}_{P,AC}^{UN}$

- using equation Equation (C.1) determine the unavoidable cost $\dot{Z}_{AC}^{UN}$ at the calculated $\dot{E}_{P,AC}^{UN}$.
- calculate the ratio $(\dot{Z}/\dot{E}_P)_{AC}^{UN}$
- $\dot{Z}_{AC}^{UN}$ is determined by the following equation $\dot{Z}_{AC}^{UN} = \dot{E}_{P,AC}(\dot{Z}/\dot{E}_P)_{AC}^{UN}$ where $\dot{E}_{P,AC}$ is the product exergy of the air compressor in the original cycle.
- $\dot{Z}_{AC}^{AV}$ is determined by $\dot{Z}_{AC}^{AV} = \dot{Z}_{AC} - \dot{Z}_{AC^{UN}}$

Determining the endogenous/exogenous cost rates, $\dot{Z}_{AC}^{EN}$ and $\dot{Z}_{AC}^{EX}$

- determine the mass flow rate of air through the system $\dot{m}_a$ at the point when all the other components are operating ideally and the air compressor is operating with its original exergetic efficiency, ε_{AC}.
- by substituting $\dot{m}_a$ in Equation (C.2), the equivalent investment cost value I_{AC} can be determined.
- $\dot{Z}_{AC}^{EN}$ can then be calculated using Equation (D.1).
- $\dot{Z}_{AC}^{EX}$ can be calculated using the following equation $\dot{Z}_{AC}^{EX} = \dot{Z}_{AC} - \dot{Z}_{AC}^{EN}$

Determining the endogenous/unavoidable, $\dot{Z}_{AC}^{EN,UN}$ cost rate

- determine the lowest possible available isentropic efficiency, $\eta_{AC,min}$
- using the engineering or "graph" method determine $\dot{E}_{D,AC}^{EN,UN}$ at $\eta_{AC,min}$ (i.e. at the corresponding $\varepsilon_{AC,min}$).
- calculate the corresponding $\dot{E}_{P,AC}^{EN,UN}$.
- substitute the corresponding $\dot{m}_a$ (at the point when $\dot{E}_{D,AC} = \dot{E}_{D,AC}^{EN,UN}$) into Equation (C.2) to determine the corresponding I_{AC} .
- using Equation (8.1) calculate the corresponding $\dot{Z}_{AC}^{EN,UN}$ at $\eta_{AC,min}$
- determine $\dot{Z}_{AC}^{EN,UN}$ for the original system parameters, where $\dot{Z}_{AC}^{EN,UN} = \dot{E}_{P,AC}^{EN}\left(\frac{\dot{Z}_{AC}}{\dot{E}_{P,AC}}\right)^{EN,UN}$ and $\dot{E}_{P,AC}^{EN}$ is the product exergy for the original system parameters.

Determining the endogenous/avoidable, $\dot{Z}_{AC}^{EN,AV}$,exogenous/unavoidable, $\dot{Z}_{AC}^{EX,UN}$ and exogenous/avoidable, $\dot{Z}_{AC}^{EX,AV}$ cost rates

- The following equations are used to determine $\dot{Z}_{AC}^{EN,AV}$, $\dot{Z}_{AC}^{EX,UN}$, $\dot{Z}_{AC}^{EX,AV}$ respectively:

i. $\dot{Z}_{AC}^{EN,AV} = \dot{Z}_{AC}^{EN} - \dot{Z}_{AC}^{EN,UN}$,

ii. $\dot{Z}_{AC}^{EX,UN} = \dot{Z}_{AC}^{UN} - \dot{Z}_{AC}^{EN,UN}$,

iii. $\dot{Z}_{AC}^{EX,AV} = \dot{Z}_{AC}^{EX} - \dot{Z}_{AC}^{EX,UN}$

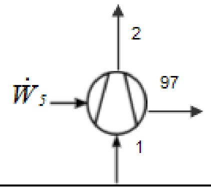
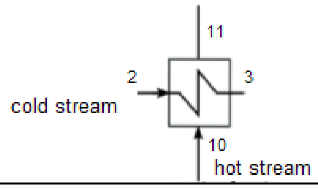
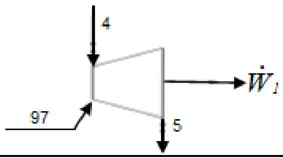
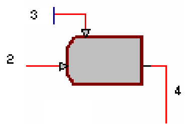
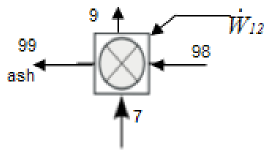
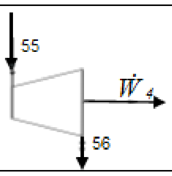
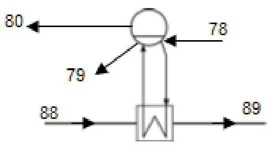
D Calculation of the material cost streams

To calculate the costs associated with each material stream in a thermal system the cost balances for each component must be formulated. A cost balance states that the total cost of the output streams in an energy conversion system must equal the total cost of the input streams plus the cost flow rate of each component $\dot{Z}_k$. Therefore for the i-th energy stream, $\dot{Z}_k + \sum \dot{C}_{i,in} = \sum \dot{C}_{i,out}$ where $\dot{C}_i = c_i \dot{E}_i$ with c_i being the cost per exergy associated with the i-th stream.

In cases where the number of material and energy transfer streams is lager than the number of components, additional equations are required to calculate the unknown variables. These equations are referred to as auxiliary equations. In determining these auxiliary equations, the F-rule and the P-rule are applied. Additional information on these rules can be found in references [2, 32].

Table D.1 lists the cost equations and auxiliary cost equations for selected components used in this research work.

Table D.1: Equations used in calculating the cost streams of selected components.

Component (k)	Schematic	Cost Balance	Additional information
Air compressor (AC)		$\dot{C}_1+\dot{C}_{W5}-\dot{C}_2-\dot{C}_{97}+\dot{Z}_{AC}=0$	air is supplied at zero cost, $c_1=0$. $c_F=\dot{C}_{W5}/W_5$ $c_P=(\dot{C}_2+\dot{C}_{97}-\dot{C}_1)/(\dot{E}_2+\dot{E}_{97}-\dot{E}_1)$
Heat exchanger (HE)		$\dot{C}_2+\dot{C}_{10}-\dot{C}_{11}-\dot{C}_3+\dot{Z}_{HE}=0$	$c_{10}=c_{11}=c_F$ $c_P=(\dot{C}_3-\dot{C}_2)/(\dot{E}_3-\dot{E}_2)$
Expander (GT)		$\dot{C}_4+\dot{C}_{97}-W_1-\dot{C}_5+\dot{Z}_{GT}=0$	One auxiliary equation required $c_5=(\dot{C}_4+\dot{C}_{97})/(\dot{E}_4+\dot{E}_{97})$ $c_F=(\dot{C}_4+\dot{C}_{97}-\dot{C}_5)/(\dot{E}_4+\dot{E}_{97}-\dot{E}_5)$ $c_P=\dot{C}_{W1}/W_1$
Combustion chamber (CC)		$\dot{C}_2+\dot{C}_3-\dot{C}_4+\dot{Z}_{CC}=0$	Combustion process in the chamber is complete; auxiliary equations are not required. $c_F=c_3$ $c_P=(\dot{C}_4-\dot{C}_2)/(\dot{E}_4-\dot{E}_2)$
Fossil Boiler (FB)		$\dot{C}_7+\dot{C}_{98}+W_{12}-\dot{C}_9-\dot{C}_{99}+\dot{Z}_{FB}=0$	$\dot{C}_{99}=0$ $c_F=(\dot{C}_{98}+\dot{C}_{W12})/(\dot{E}_{98}+W_{12})$ $c_P=(\dot{C}_9-\dot{C}_7)/(\dot{E}_9-\dot{E}_7)$
Steam turbine (ST)		$\dot{C}_{55}-W_4-\dot{C}_{56}+\dot{Z}_{ST}=0$	$c_{55}=c_{56}$ $c_F=(\dot{C}_{55}-\dot{C}_{56})/(\dot{E}_{55}-\dot{E}_{56})=c_{55}$ $c_P=\dot{C}_{W4}/W_4$
Evaporator (EV)		$\dot{C}_{78}+\dot{C}_{88}-\dot{C}_{79}-\dot{C}_{80}-\dot{C}_{89}+\dot{Z}_{EV}=0$	$c_{88}=c_{89}$, $c_{79}=0$ $c_F=(\dot{C}_{88}-\dot{C}_{89})/(\dot{E}_{88}-\dot{E}_{89})$ $c_P=(\dot{C}_{80}-\dot{C}_{78})/(\dot{E}_{80}-\dot{E}_{78})$

E Additional results and data for the EFCC power plant

Endogenous curve equations

Table E.1 lists the endogenous curve equations obtained for selected components of the EFCC power plant.

Table E.1: Endogenous curve equations for selected components of the EFCC.

Component	Endogenous curve equation
Air compressor (C1)	$416.03x^2 - 887.88x + 472.23$
Heat exchanger (C2 and C3)	$407.11x^2 - 941.04x + 533.93$
Expander (C4)	$103.66x^2 - 397.08x + 293.42$
Fossil Boiler (C6)	$-159.173ln(x) - 0.00297$
Steam Turbine (C32)	$0.2856x^2 - 11.405x + 11.11$
Evaporator (C48)	$24.26x^2 - 54.48x + 30.22$

Unavoidable investment cost parameters

Table E.2 lists the parameters used in calculating the unavoidable investment costs for selected components of the EFCC power plant.

Table E.2: Assumptions made in determining the unavoidable investment cost for the EFCC.

Component	Parameter	Design Parameters	Parameters used in determining unavoidable costs
Air compressor (C1)	η_{C1}	0.895	0.85
Heat exchanger (C2 and C3)	ΔT_{min}	425K	667.41K
Expander (C4)	η_{C4}	0.90	0.85
Fossil Boiler (C6)	ΔP	0.05	0.10
Steam Turbine (C32)	η_{C32}	0.88	0.75
Evaporator (C48)	ΔT_{min}	18K	35K

References

[1] BEJAN, A. *Advanced Engineering Thermodynamics*, 2d. ed. New York: John Wiley and Sons, 1997.

[2] BEJAN, A., TSATSARONIS, G., AND MORAN, M. *Thermal Design and Optimization*. Wiley-IEEE, 1996.

[3] CODDINGTON, E. *An Introduction to Ordinary Differential Equations*. Courier Dover Publications, 1989.

[4] CORPORATION, N. T. Turbine prices. http://www.gas-turbines.com/trader/kwprice.htm, 2007.

[5] CZIESLA, F., AND TSATSARONIS, G. Cost-effective design of thermal power plants using intelligent systems. In *ICIIS '99: Proceedings of the 1999 International Conference on Information Intelligence and Systems* (Washington, DC, USA, 1999), IEEE Computer Society, p. 262.

[6] CZIESLA, F., TSATSARONIS, G., AND GAO, Z. Avoidable thermodynamic inefficiencies and costs in an externally fired combined cycle power plant. *Energy(Oxford) 31*, 10-11 (2006), 1472–1489.

[7] GAGGIOLI, R., AND EL SAYED, Y. A critical review of second law costing methods-II; Calculus procedures. *Journal of Energy Resources Technology 111*, 1 (1989).

[8] IBRAHIM O. M., A. K. S. A. Optimum power of carnot and lorenz cycle. education in thermodynamics and energy systems. *Avanced Energy Systems 20* (1990), 91–96.

[9] KEENAN, J. Availability and irreversibility in thermodynamics. *British Journal of Applied Physics 2*, 7 (1951), 183–192.

[10] KOTAS, T. *The Exergy Method of Thermal Plant Analysis*. Butterworths, 1985.

[11] LAZZARETTO, A., AND TSATSARONIS, G. A general process-based methodology for exergy costing. *Proceedings of the ASME Advanced Energy Systems Division 36* (1996), 413–428.

[12] LAZZARETTO, A., AND TSATSARONIS, G. On the quest for objective equations in exergy costing. *Proceedings of the ASME Advanced Energy Systems Division. ASME. AES 37* (1997), 197–210.

[13] LAZZARETTO, A., AND TSATSARONIS, G. SPECO: A systematic and general methodology for calculating efficiencies and costs in thermal systems. *Energy 31*, 8-9 (2006), 1257–1289.

[14] MORAN, J. M., AND SHAPIRO, H. *Fundamentals of Engineering Thermodynamics*, 5d ed. New York: John Wiley and Sons, 2004.

[15] MOROSUK, T., MOROSUK, C., AND FEIDT, M. New proposal in the thermodynamic analysis of complex heat regeneration systems. *Energy 29*, 12-15 (2004), 2517–2535.

[16] MOROSUK, T., AND TSATSARONIS, G. The "cycle method" used in the exergy analysis of refrigeration machines: From education to research. In *ECOS-2006 Conference on "Efficiency, Costs, Optimization, Simulation and Environmental Impact of Energy Systems", Agia Pelagia, Crete, Greece, July 12-14* (2006).

[17] MOROSUK, T., AND TSATSARONIS, G. Exergoeconomic Evaluation of Refrigeration Machines based on Avoidable Endogenous and Exogenous Costs. In *Proceedings of the 20th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems (ECOS 2007)* (Padova, Italy, June 25-28 2007), L. A. Mirandola A., Arnas O., Ed., vol. I, pp. 1459–1467.

[18] PONS, M., AND NUMBER, R. Irreversibility in energy processes: Non-dimensional quantification and balance. *J. Non-Equilib. Thermodyn. 29* (2004), 157.

[19] SZARGUT, J., MORRIS, D., AND STEWARD, F. Exergy Analysis of Thermal, Chemical, and Metallurgical Processes. *Hemisphere Publishing Corporation, 79 Madison Avenue, New York, New York 10016, USA, 1988.* (1988).

[20] TSATSARONIS, G. Thermoeconomic analysis and optimization of energy systems. *Progress in Energy and Combustion Science 19*, 3 (1993), 227–257.

[21] TSATSARONIS, G. Exergoeconomics: Is It Only a New Name? *Chem. Eng. Technol. 19*, 163 (1996), 163–169.

[22] TSATSARONIS, G. Design optimization using exergoeconomics. *Thermodynamics and the Optimization of Complex Energy Systems, NATO Advanced Study Institute, Neptun, Romania* (1998), 394–410.

[23] TSATSARONIS, G. Strengths and limitations of exergy analysis. *A. Bejan and E. Mamut (eds.), Thermodynamic Optimization of Complex Energy System* (1999), 93–100.

[24] TSATSARONIS, G. Advanced Exergoeconomics-1. In *Proceedings of the 4th "European Congress Economics and management of energy in industry"* (Porto, Portugal, November, 27-30 2007). CD-ROM, 10 p.

[25] TSATSARONIS, G., AND CZIESLA, F. *Encyclopedia of Physical Science and Technology*, 3 ed., vol. 16. Academic Press, 2002, ch. Thermoeconomics, pp. 659–680.

[26] TSATSARONIS, G., KELLY, S., AND MOROSUK, T. Endogenous and exogenous destruction in thermal systems. In *Proceedings of the ASME International Mechanical Engineering Congress and Exposition, November 5-10* (Chicago, USA, 2006).

[27] TSATSARONIS, G., AND MORAN, M. Exergy-Aided Cost Minimization. *Energy Conversion and Management 38*, 15-17 (1997), 1535–1542.

[28] TSATSARONIS, G., MOROSUK, C., AND KELLY, S. Approaches for splitting the exergy destruction into endogenous and exogenous parts. In *Paper presented at the 5-th Workshop "Advances in Energy Studies"* (Porto Venere, Italy, September 12-16 2006).

[29] TSATSARONIS, G., AND MOROSUK, T. Advanced exergoeconomic evaluation and its application to compression refrigeration machines. In *Proceedings of the ASME International Mechanical Engineering Congress and Exposition* (Seattle, USA, November, 11-15 2007). CD-ROM, file 2007-41202.

[30] TSATSARONIS, G., AND PARK, M.-H. On avoidable and unavoidable exergy destructions and investment costs in thermal systems. *Energy Conversion and Management 43* (2002), 1259–1270.

[31] TSATSARONIS, G., AND PISA, J. Exergoeconomic evaluation and optimization of energy systems: application to the CGAM problem. *Energy(Oxford) 19*, 3 (1994), 287–321.

[32] Valero, A., Lozano, M., Serra, L., Tsatsaronis, G., Pisa, J., Frangopoulos, C., and Von Spakovsky, M. CGAM problem: definition and conventional solution. *Energy(Oxford) 19*, 3 (1994), 279–286.

[33] Valero, A., and Torres, C. Algebraic thermodynamic analysis of energy systems. *Proceedings of approaches to the design and optimization of thermal systems AES 7* (1988), 13–23.

[34] Wall, G. Thermoeconomic optimization of a heat pump system. *ENERGY. 11*, 10 (1986), 957–967.