The Thesis Entitled

EXERGOECONOMIC OPTIMIZATION OF AN INDUSTRIAL AQUA AMMONIA VAPOUR ABSORPTION REFRIGERATION UNIT

Submitted to

The Maharaja Sayajirao University of Baroda, Vadodara

in fulfillment for the award of degree of

Doctor of Philosophy
in
MECHANICAL ENGINEERING

Submitted by

Vijaykumar Kundanlal Matawala

(ROLL NO-407)

DEPARTMENT OF MECHANICAL ENGINEERING
FACULTY OF TECHNOLOGY AND ENGINEERING
THE MAHARAJA SAYAJIRAO UNIVERISITY OF BARODA
VADODARA
March-2012

CERTIFICATE

This is to certify that the thesis entitled

EXERGOECONOMIC OPTIMIZATION OF AN INDUSTRIAL AQUA AMMONIA VAPOUR ABSORPTION REFRIGERATION UNIT

submitted by

Vijaykumar Kundanlal Matawala

In the fulfillment of the requirement for the award of

"DOCTOR OF PHILOSOPHY" IN MECHANICAL ENGINEERING
OF
THE MAHARAJA SAYAJIRAO UNIVERSITY OF BARODA

is a bonafide work carried out by him under my supervision and guidance. The matter embodied in the thesis, to the best of my knowledge, has not been submitted previously for the award of any other degree or diploma of any institute or university.

Guide:
Dr P Prabhakaran
Professor, Mech. Engg. Dept.,
Faculty of Tech & Engg.
The M. S. University of Baroda.

Head
Mechanical Engg. Dept.
Faculty of Tech & Engg.
The M. S. University of Baroda.

Dean
Faculty of Tech & Engg.
The M. S. University of Baroda

CERTIFICATE OF THESIS APPROVAL

Thesis entitled "Exergoeconomic Optimization of an Industrial Aqua Ammonia Vapour Absorption Refrigeration Unit" by Vijaykumar Kundanlal Matawala is approved for the award of 'Doctor of Philosophy in Mechanical Engineering' by The Maharaja Sayajirao University of Baroda.

Internal Referee	**External Referee**

ABSTRACT

The objective of the present research work is to develop a user-friendly method of exergoeconomic optimization to predict the cost effectiveness of an energy intensive thermal system such as Aqua Ammonia Vapour Absorption Refrigeration (AAVAR) system and suggest ways of improving the cost effectiveness from both thermodynamic and economic points of view.

In recent years, the exergy analysis (Second law analysis) is getting more popularity in analyzing thermal systems, as it provides information about the losses qualitatively as well as quantitatively along with their location. Exergetic optimization improves the performance of thermal system through increasing the exergetic efficiency. This improvement, however, is accompanied by an increase in capital investment of the system. Hence, thermal systems should be optimized from both thermodynamic and economic points of view. In this regard, exergoeconomic optimization is a better tool as it combines the thermodynamic analysis with the economic principles. Here, appropriate costs are assigned to the thermodynamic inefficiencies of the system components through meaningful Fuel-Product-Loss definition and also estimates the parameters like the exergy destruction and the exergy loss which add to the hidden cost. For maximum exergoeconomic efficiencies, these costs are to be minimized. Therefore, it is necessary to correlate the exergy with cost value. It can be carried out through exergoeconomic analysis.

The exergoeconomic analysis improves the performance of the thermal system which is associated with the increase in investment and operation & maintenance cost. Thus the cost optimization problem involves the maximization of thermodynamic performance and minimization of investment cost. These are the contradictory disciplines. The exergoeconomic concept combines them together and develops effective tool for design the thermal system with higher efficiency and lower cost.

The literature survey has also identified a number of exergoeconomic optimization models. However, most of them are either complex to translate or incomplete in their availability in open literature. Thermoeconomic Evaluation and Optimization (TEO) method suggested by Tsatsaronis and his associates is a user friendly

method which needed some alterations to suite to the optimization of a given thermal system.

The exhaustive literature survey on exergoeconomic optimization of various thermal systems reveals that the researchers have shown little interest towards AAVAR system. The probable reason behind this may be the less popularity of AAVAR system compared to vapour compression refrigeration system. Moreover, they are less capital intensive compared to other thermal intensive systems like power plants. However, AAVAR system used in huge chemical industries needs greater attention as slight modification in the system parameters brings substantial savings in energy and production cost. Considering this important observation, the present research work on the optimization of an industrial brine chilling unit using AAVAR system is undertaken.

The system considered in the present research work is a single effect AAVAR system with a cooling capacity of 800 TR located at a large fertilizer industry at Bharuch, Gujarat. The existing system uses steam as heat source generated in the independent boiler. This AAVAR system is exergoeconomically optimized to reduce the product cost through Exergoeconomic Evaluation and Optimization method. The method follows iterative procedure based on the local optimization of the sub systems. Being iterative in nature, the method requires engineering judgments and critical evaluations in each iterative step to take decisions on the change of decision variables.

The fertilizer industry is having gas turbine power plant with a capacity of 8 MW. The waste heat of exhaust gas is utilized in HRSG to generate steam thus forming a GT-HRSG plant. The steam thus generated is subsequently utilized in chemical process. The AAVAR system is simulated in combination with gas turbine power plant considering steam generated at HRSG as heat source for AAVAR system instead of the steam from independent boiler. This first option of the combining the steam generated at HRSG of GT-HRSG plant system with AAVAR is optimized exergoeconomically. Considerable reduction in the cooling cost of AAVAR system is observed with this option.

The major electricity demand of the Fertilizer plant is met by the captive steam turbine power plant of 50 MW capacity located in the plant premises. The thermodynamic cycle of the steam turbine power plant is Regenerative Rankine Cycle. The tapped steam from the steam turbine is considered as a heat source option for

AAVAR system. This second option of combining steam generated from the captive steam power plant with AAVAR is optimized exergoeconomically. It is found that the cooling cost of AAVAR system is further reduced by about 50%.

CONTENTS

Appendices

LIST OF FIGURES

LIST OF TABLES

Nomenclature

A	Heat transfer surface area (m^2)
B	Constant in cost equation
$\dot{C}$	Cost flow rate (₹/hr)
$\dot{c}$	Cost per unit exergy (₹/kJ)
$\dot{E}$	Exergy flow rate (kW)
f	Exergoeconomic factor (%)
h	Enthalpy
hr	hour
I	Total Capital Investment
i	Interest rate
$\dot{m}$	Mass flow (kg/sec)
M	Molecular weight
n	Plant life in years
p	pressure
$\dot{Q}$	Heat transfer rate (kW)
r	Escalation rate/Relative cost difference (%)
$\dot{R}$	O & M cost invariable to optimization
₹	Rupees (Indian currency)
sec	Second
S	Entropy
$\dot{s}$	Rate of entropy generation
t	Temperature (°C)
r	Rectifier

$\dot{W}$	Work done
x	Mole fraction
y	Mole fraction with water as liquid
$\dot{Z}$	Levelized investment cost of component
1-34	Stations in the absorption cycle

Subscript

a	Absorber/air
c	Condenser
D	Exergy Destruction
CE	Condenser Evaporator
e	Evaporator/Exit
eff	Effective
F	Fuel
g	Generator
i	Inlet
k	kth component of the system
L	Exergy Loss
m	Motor
p	Pump
s	Steam
m	Motor
p	Product
COP	Coefficient of performance

s	Steam
T	Turbine
Tot	Total system
w	Work

Superscript

CI	Capital investment
OM	Operation and maintenance
OPT	Optimum

Greek letter

α	Exponent
β	Capital recovery factor
χ	Effectiveness of the heat exchanger
ε	Exergetic efficiency (%)
γ	Percentage O&M cost
η	Isentropic efficiency
τ	Annual hours of system operation
ω	Coefficient expressing variable Levelized O&M cost for the
Δ	Difference
λ	Air Fuel ratio

Abbreviations

APH	Air Pre Heater
AC	Air Compressor
CC	Combustion Chamber
CFC	Chloro fluoro carbon
CRF	Capital Recovery Factor
EDM	Exergy Destruction Method
EGM	Entropy Generation Minimization
FD	Forced Draught
GT	Gas Turbine
GCV	Gross Calorific Value
HCFC	Hydro fluoro carbon
HRSG	Heat recovery steam generator
ID	Induced Draught
LHV	Lower Heating Value
M&S	Marshal & Swift
OBF	Objective function
PEC	Purchase equipment cost
PC	Pre-cooler
RHX	Refrigerant heat exchanger
SHX	Solution heat exchanger
TCI	Total capital investment
TEO	Thermoeconomic Evaluation and Optimization
TR	Ton of refrigeration
VAR	Vapour absorption refrigeration
VCR	Vapour Compression Refrigeration

Chapter 1

Introduction

Classical thermodynamics provides the concept of energy, energy transfer by heat and work, energy balance, entropy and entropy balance and calculations of thermodynamic properties at equilibrium. The second law of thermodynamics enhances an energy balance by calculating the true thermodynamic value of an energy career and real thermodynamic inefficiencies and losses from the process and system. Exergy is the maximum useful work attainable from an energy career under the given environmental conditions. The exergy of an energy carrier is a thermodynamic property that depends on both the state of the carrier being considered and the state of the environment. It expresses the maximum capability of the energy carrier to cause changes. Thus, exergy is closely related to the economic value of the carrier because users pay for the potential of energy to cause changes. When costs are assigned to energy carriers, exergy should serve as a basis in the costing process.

Conventionally, first law analysis gives only energy utilization scenario in terms of conservation of energy. But it cannot provide the information regarding the losses both qualitatively and quantitatively and cannot find the location of these losses. These limitations force us to go for exergy analysis based on second law of thermodynamics. Exergy is not a conserved property but some of it is destroyed in the real process. Exergy analysis gives uniform base for comparison of various thermodynamic processes. This analysis proves the information regarding losses including their location qualitatively and quantitatively. This information can be used for further improvement in the design and operation of the system. By locating the exergy destruction, the system performance can be improved by improving the exergetic efficiency of the component and the system.

Unlike total energy, a part of the total exergy supplied to a system is irreversibly destroyed in all real processes. This exergy destruction is the direct result of the irreversibilities in a system and usually represents the largest part of 'energy waste'. The other part of 'energy waste' is the exergy loss, i.e. the exergy associated with a material or energy stream rejected to the environment (e.g. flue gas, cooling water and heat loss). The exergy loss is associated with the design engineer's decision to not further use the exergy of a stream in a given system. Malfunctioning and off-design performance of a component usually increases both the exergy destruction and the exergy losses in a system. A part of the exergy destruction and exergy loss is dictated by considerations involving costs, environmental impact, availability, and operability of the system being considered. In a truly optimized system, the entire amount of exergy destruction and loss is justifiable through these considerations. Both exergy destruction and exergy loss are identified through an exergy analysis (second-law analysis).

The term exergy was coined by Rant as a new word for 'work capability' used previously by Bosnjakovic. This term has gained general acceptance in all countries except the United States where the parallel use of the terms exergy and availability (available energy) continues to contribute to some misconceptions and confusion surrounding the exergy method. However, the word exergy finally prevailed in the United States too. Although the method of exergy is often considered to be a new method for analyzing energy systems, the underlying fundamentals were introduced in the last century following the mathematical formulation of the second law of thermodynamics. As outlined in the critical historical review by Tsatsaronis [1], the earliest contributions to the exergy concept are due to Clausius, Tait, Thomson, Gibbs and Maxwell. This early work, as well as the subsequent developments by Gouy, Stodola, Goudenough and Darrieus generated interest in exergy.

The modern development of exergy analysis was initiated by Bosnjakovic in Europe and Keenan in the United States. The classical slogan "Fight the Irreversibilities" by Bosnjakovic marks the beginning of this development. In the1950s and 1960s, contributions to the exergy concept were also made by Rant, Grassmann, Brodyansky, Bruges, Tribus, Obert, Gaggioli, Evans, Baehr, Fratzscher, Szargut, Petela and Knoche, among the others. During this period, the exergy balance and its graphical presentation, the calculation of the exergy of fuels and the definition of reference states for calculating the chemical exergy were introduced. In addition, several definitions of exergetic efficiencies and the first exergy-analysis applications to industrial processes and plants were presented. At the same time, the method of exergy analysis was introduced into the textbooks of thermodynamics. In the last twenty years, the annual number of worldwide exergy applications to various systems and processes has increased exponentially.

Formerly the term 'thermoeconomics' has been used to indicate an appropriate combination of exergetic and economic analysis in which the cost is assigned to the exergy (not the energy) content of an energy carrier (exergy costing). In parallel, however, the term 'thermoeconomic analysis' has been used by others to report conventional thermodynamic analyses based only on the first law of thermodynamics and economic analyses conducted separately from the thermodynamic ones and without the consideration of exergy or exergy costing. But ‘thermo', is a derivative of the Greek word for heat and is used in most major languages. Thus, thermoeconomics does not imply exergy costing or exergy economics, but a combination of heat and economics.

The idea of using exergy for costing purposes was initiated by Keenan in 1932. His suggestion was not to apply exergy costing, but to use exergy for appropriately apportioning costs to the electric power and steam that were produced in a cogeneration plant. He pointed out that the economic value of steam and electricity lay in their exergy not energy. In 1949, Benedict presented the costing of exergy destruction in an air-separation plant and the use of these costs for 'optimal design'.

Along with the thermodynamic analysis, economic analysis gives the information regarding fixed cost e.g. investment cost, running cost, and operation and maintenance cost. In most of the cases, the overall cost of the system will increase with the increase in the system exergetic efficiency and capacity. Thus thermodynamic improvement in a system is accompanied by an increase in the economic cost. Therefore the system should be optimized between these two conflicting requirements. In this regard, thermoeconomic analysis evolved which bring thermodynamic and economic parameters in to one common platform and combines thermodynamic analysis with economic analysis. As discussed above, exergy analysis is preferred for thermodynamic analysis; the newly evolved field is called exergoeconomic analysis.

The development of thermoeconomics was initiated in the late 1950s by Tribus and Evans at the University of California, Los Angeles, and by Obert and Gaggioli at the University of Wisconsin, Madison. Tribus and Evans were applying the exergy concept to desalination processes when they introduced the word thermoeconomics, developed the idea of assigning costs to the exergy unit of streams, and formulated cost balances at the component level of energy systems. Obert and Gaggioli applied exergy costing to the optimal selection of steam piping and its insulation. In Europe, Bergmann and Schmidt assigned costs to the exergy destruction in each component of a steam power plant in a study dealing with optimization of feed water heaters. Fratzscher and Kloiditz referred to the early work of Evans and Tribus applied exergy costing to the design of a regenerative heat exchanger. Szargut used exergy costing in a cogeneration plant. In 1970 EI-Sayed and Evans marked the introduction of rigorous calculus methods of optimization in thermoeconomics. In 1980, Evans [2] thermoeconomically isolated the components of thermal system from each other and expressed the interactions in terms of essential or useful energy which he called “Essergy” and described all economic interaction by Lagrange multiplier. This approach, though it has not yet yielded the expected practical results, continues to show promise.

In 1985, Tsatsaronis and Winhold [3] coined the term 'exergoeconomics' to give a more precise combination of an exergy analysis with an economic analysis. The thermodynamic and economic analyses do not have to be combined in the more general field of thermoeconomics, whereas in exergoeconomics, they are integrated through exergy costing. Consequently, exergoeconomics is a part of thermoeconomics. A complete thermoeconomic analysis consists of (a) a detailed exergy analysis, (b) an economic analysis conducted at the component level of the energy system being analyzed, (c) exergy costing and (d) an exergoeconomic evaluation of each system component. The objectives of an exergoeconomic analysis are:

- To identify the location, magnitude and source of the real thermodynamic losses (energy waste) in an energy system (exergy destruction and exergy losses).
- To calculate the cost associated with the exergy destruction and exergy losses.

- To assess the production costs of each product (output) in an energy-conversion system that has more than one product.
- To facilitate feasibility and optimization studies during the design phase for an energy system, as well as process improvement studies for an existing system.

The exergoeconomic methods help in the system improvement using thermodynamic as well economic points of view by simultaneous modeling of thermodynamic and economic aspects of the system and its components. These methods are based on optimization technique, which search all possible solutions for the optimum design and operation of the system and its components. Just like the exergoeconomic analysis, exergoeconomic optimization also combines thermodynamic and economic aspects. For thermodynamic optimization based on exergetic consideration, two methodologies are identified, entropy generation minimization method and exergy destruction method.

The objective in the application of the entropy generation minimization (EGM) method is to find design in which the entropy generation is minimal. A minimum entropy generation design characterizes a system with minimum destruction of exergy. This method consists of dividing the system in to sub systems those are in local (or internal) thermodynamic equilibrium. Entropy is generated at the boundaries between sub systems, as heat and mass flow through the boundaries. Using these flow rates, the total rate of entropy generation is calculated in relation to the physical characteristics of the systems. The total entropy generation is then monitored and minimized by properly varying the physical characteristics of the systems.

In exergy destruction method (EDM), the exergy balance is to be carried out which states that the total exergy increase or decrease within the system boundary plus the exergy destruction within the same boundary equals the difference between the total exergy transfers in and out across the boundary. The exergy transfer across the boundary includes exergy transfer associated with the transfer of heat, work and mass entering and leaving the boundary across the boundary. This method gives the idea about the exergy loss and exergy destruction in the components. After analyzing all the components individually, the overall system performance can be estimated. Then by varying the system parameters, system can be optimized.

In exergoeconomics, a system and its components are thermodynamically based on exergy as well as economically analyzed to formulate an objective function which would satisfy the thermodynamic and the economic objectives of the system simultaneously. The thermodynamic objective is to maximize the exergetic efficiencies of the components and the system, and the economic objective is to minimize the investment cost, operation and maintenance cost of the system. Thus the objective of thermoeconomics is to obtain the compromise between these two competing objectives. In this methodology, appropriate costs are assigned to the thermodynamic inefficiencies of the system components through some meaningful fuel-product definition. For maximum exergetic efficiencies, these costs are to be minimized. The overall objective function for the system is defined so as to minimize the summation of the costs associated with the thermodynamic inefficiencies and other economic costs.

1.1 Exergoeconomic Methodologies

Many exergoeconomic analysis and optimization methodologies are developed in the last few decades by various researchers. They can be listed as below:

- Thermoeconomic Evaluation and Optimization Method
- Exergetic Cost Theory
- Thermoeconomic Functional Analysis
- Autonomous Method
- Structural Method
- Evolutionary Programming Method
- Extended Exergy Accounting Method
- Exergetic Production Cost Method

The detailed information regarding the development of these methodologies is given in the subsequent chapter on literature survey. In each methodology, mentioned above, has specific field of application for which it provides proven and efficient solution. All these methodologies are based on local optimization of the components after separation of the system components and then finding the overall solution for the whole system.

Exergoeconomic analysis and optimization, thus, is a very important step before the design, installation and commissioning of any energy intensive process plant. Nowadays, it has become an integral part of the plant design procedure for any thermal, chemical or petro-chemical process plants. A number of analysis and optimization models are suggested by various investigators in the recent past to carry out exergoeconomic analysis on thermally intensive systems such steam power plants, gas turbine power plants, combined cycle power plants, refrigeration systems such as vapour compression and vapour absorption systems, cryo-generation plants, internal combustion engines, hydrogen combustion process etc. Amongst the various thermal systems used for the development of analysis and optimization tools, vapour absorption refrigeration system seems to be attractive as it is a heat energy intensive system. Thus, an industrial AAVAR system used to generate chilled brine for industrial application in a fertilizer plant is exergoeconomically optimized using a hybrid method developed based on various methods suggested by earlier investigators.

1.2 Organization of the Thesis

The thesis is divided in to seven chapters. In the present chapter, the general area of exergoeconomic optimization is introduced and the development in the area of exergy and exergoeconomic analyses and optimization are briefly discussed. Chapter 2 gives an extensive review of open literature in the area of exergy analysis and optimization and exergoeconomic analysis and optimization. Based on the review, the current research area and problem are identified. A large capacity brine chilling unit working with AAVAR system of a large fertilizer plant with independent boiler as a heat source is selected as a case for developing the exergoeconomic optimization method. As the other option for heat source, steam generated at HRSG of gas turbine power plant and tapped steam from regenerative steam turbine power plant is selected. It is followed by the objectives of the present research work. The description of the AAVAR system, gas turbine power plant with HRSG and regenerative steam turbine power plant is given in Chapter 3. The exergoeconomic analysis chosen for the present optimization study is mainly based on Thermoeconomic Evaluation and Optimization (TEO) method introduced by Tsatsaronis and is described in Chapter 4. Chapter 5 presents the exergoeconomic optimization of VAR system. This includes the comparison of exergy destruction method (EDM) and entropy generation minimization method (EGM) of exergy analysis of the thermal system. There are three options available to develop the exergoeconomic optimization method through case studies. The first option is using steam generated in an independent dedicated boiler and supplied as energy source to VAR system. Next section gives the exergoeconomic optimization of the AAVAR system through TEO method and the optimum configuration of the system with respect to minimum cooling cost using the steam from independent dedicated boiler as a heat source. In Chapter 6, other two options as heat source for AAVAR system are analyzed. The second option is to use exhaust gases (waste heat) from the available gas turbine power plant as heat energy source to VAR system. The exergoeconomic optimization of the system through TEO method for the optimum configuration of the combined system with respect to minimum cost of power generation by the gas turbine and corresponding cost of cooling generated by the VAR system using the second option of exhaust gases (waste heat) from the available gas turbine power plant as heat source is described in subsequent section. The next section gives the exergoeconomic optimization through TEO method and the optimum configuration of the system with respect to minimum cost of power generated by the steam turbine for the third option of using tapped steam from the steam turbine of a regenerative type steam turbine power plant as heat source. At the end, both the alternative options viz. exhaust gas from gas turbine and tapped steam from steam turbine are compared. The conclusions derived from the present study are given in Chapter 7.

Chapter 2

Review of Literature

This chapter is concerned with review of literature on exergy and exergoeconomic analysis carried out on various thermal systems. The review is carried out primarily to develop an exergoeconomic tool for optimization of a brine chilling unit working on AAVAR system drawing heat energy from a dedicated boiler of a large fertilizer plant. The chapter is organized in three sections. The literature on various investigation using two types of exergy analysis, viz., entropy generation minimization method and exergy destruction method are separately given in Section 2.1. The section deals with the exergy analysis of various thermal systems in general and refrigeration systems in particular. Section 2.2 deals with the short discussion on various models developed by earlier investigators on exergoeconomic optimization. Based on the review, scope for the present investigation is identified and given in Section 2.3.

2.1 Exergy Analysis of Thermal Systems

The literature on exergy analysis using entropy generation minimization method (EGM) employed on various thermal systems such as vapour absorption refrigeration systems, vapour compression refrigeration systems, transcritical carbon dioxide refrigeration cycle, adsorption cycles, heat exchangers, radial fin geometry thermal energy storage systems, power plant, compression ignition engine and hydrogen combustion process are reviewed in Section 2.1.1. Section 2.1.2 deals with review of literature on exergy analysis using exergy destruction method (EDM) employed on various thermal systems. Both the sections are further divided in to two based on the literature on (i) refrigeration systems such as vapour absorption, vapour compression and transcritical refrigeration cycle etc. and (ii) other thermal systems such as heat exchangers, thermal storage systems, power plant, compression ignition engine etc.

2.1.1 Entropy Generation Minimization (EGM) Method

The objective in the application of the entropy generation minimization method is to find the design in which the entropy generation is minimum. A minimum entropy generation design characterizes a system with minimum destruction of exergy. This method consists of dividing the system in to sub systems which are in local (or internal) thermodynamic equilibrium. Entropy is generated at the boundaries between sub systems, as heat and mass flow through the boundaries. Using these flow rates, the total rate of entropy generation is calculated in relation to the physical characteristics of the systems. The total entropy generation is then monitored and minimized by properly varying the physical characteristics of the systems.

2.1.1.1 Refrigeration Systems

In 1995, Bejan et al. [4] applied the entropy generation minimization technique to absorption refrigeration system and found the way of dividing a finite supply of heat exchanger surface between the three heat exchangers of the refrigeration plant namely generator, condenser and evaporator for maximizing the cooling load at evaporator.

Talbi et al. [5], in 2000, carried out modeling, thermodynamic simulation and second law analysis through entropy generation minimization method for LiBr/Water VAR system. They quantified the irreversibility of each component of the chiller to determine the potential for each component to contribute to overall system exergy efficiency. The second law analysis of thermodynamics was applied and showed that the absorber, solution heat exchangers, and condenser have the most potential to improve chiller exergy efficiency.

In 2004, Ezzine et al. [6] carried out similar studies for ammonia-water double-effect, double-generator VAR system. In the same year, Adewusi et al. [7] carried out second law based thermodynamic analysis of AAVAR system. The entropy generation at each salient point and the total entropy generation S_{tot} of all the system components as well as the coefficient of performance of the system are calculated from the thermodynamic properties of the working fluids at various operating conditions. The results showed that the two stage system has a higher S_{tot} and COP, while the single-stage system has a lower S_{tot} and COP. This controversy is explained with respect to the performance results for both single and two-stage systems.

Kaynakli [8], in 2008, applied the entropy generation minimization method to the coil absorber of the LiBr absorption refrigeration system and determined the variation of the second law efficiency with cooling water flow rate, solution flow rate, cooling water temperature and solution concentration. The influence of absorber performance parameters is examined on the basis of the first and second laws of thermodynamics for parallel and counter-current types. In this regard, the heat and mass transfer, the second law efficiency, the magnitude and place of exergy losses in two types of absorbers are estimated and discussed comprehensively. The results showed that increasing the cooling water flow rate and decreasing the cooling water inlet temperature increase the heat and mass transfer, and decrease the second law efficiency. The effect of the solution concentration on the efficiency in general is small whereas the irreversibility for the counter-current mode is greater than that of the parallel-current mode.

In 2002, Yumrutas et al. [9] used entropy generation minimization method for the exergy analysis of vapour compression refrigeration (VCR) system using ammonia as refrigerant, and investigated the effects of the evaporating and condensing temperatures on the pressure losses, the exergy losses, the second law of efficiency, and the coefficient of performance (COP) of the cycle. It is found that the evaporating and condensing temperatures have strong effect on the exergy losses in the evaporator and condenser and on the second law of efficiency and COP of the cycle but little effects on the exergy losses in the compressor and the expansion valve. The second law efficiency and the COP increases, and the total exergy loss decreases with decreasing temperature difference between the evaporator and refrigerated space and between the condenser and outside air.

In 2005, Yang et al. [10] performed comparative study for the transcritical carbon dioxide refrigeration cycles with a throttling valve and with an expander, using entropy generation minimization method. The effects of evaporating temperature and outlet temperature of gas cooler on the optimal heat rejection pressure, the coefficients of performance, the exergy losses, and the exergy efficiencies are investigated. In order to identify the amounts and locations of irreversibility within the two cycles, exergy analysis is employed to calculate the entropy change and irreversibility through the Guy-Stodola's law, to analyze the thermodynamics process in each component. It is found that in the throttling valve cycle, the largest exergy loss occurs in the throttling valve.

Sarkar et al. [11], in 2009, carried out exergy analyses, with entropy generation minimization method, of evaporator and gas cooler of a CO_2 based transcritical heat pump for combined cooling and heating, employing water as the secondary fluid. Optimization of heat exchanger tube diameter and length and effect of design parameters on overall system performance is also presented. It is observed that higher heat transfer coefficient can be achieved by reducing the diameter only to a limited extent due to rapid increase in pressure drop. The minimum possible diameter depends on mass flow rate (capacity) and division of flow path. The right combination of optimum diameter and length depends on the number of passes, capacity and operating parameters. It is to be noticed that due to higher pressure drop occurring in the evaporator compared to the gas cooler, zero temperature approach is attained before the optimum length is reached in case of the evaporator. Presented results are helpful in choosing the effective heat exchanger size in terms of diameter, length and number of passes.

2.1.1.2 Other Thermal Systems

Other thermal systems that are analyzed by various investigators using EGM are adsorption cycles, heat exchangers, radial fin geometry thermal energy storage systems, power plant, compression ignition engine, hydrogen combustion process. They are reviewed in this section

Adsorption Cycles

Pons [12], in 1996, developed the second law analysis of the adsorption cycles with thermal regeneration. The different heat transports between heat transfer fluid and adsorbent, between adsorbent and condenser/evaporator heat sources, and between heat transfer fluid and heat sources are analyzed. The entropy balance is then completely established. Consistency between the first law and second law analysis is verified by the numerical values of the entropy productions. The optimal operation of an adsorber is then described, and the study of those optimal conditions leads to some correlation between the different internal entropy productions.

Heat Exchangers

In 1997, Cornelissen et al. [13] carried out an exergetic optimization of a heat exchanger by combining entropy generation minimization method and the life cycle analysis (LCA). The methodology in the LCA includes the effects of all the phases of the production, use and recycling on the environment by using only one criterion, to minimize the life cycle irreversibilities due to frictional pressure drops and the temperature difference between the hot and cold stream and irreversibilities due to the production of the materials and the construction of the heat exchanger associated with the delivery of domestic hot water while the other factors like pollution of air and water, noise etc are neglected. The analysis gives the design conditions of the heat exchangers which lead to the lowest life cycle irreversibility.

In 2007, Gupta et al. [14] carried out second law analysis of cross flow heat exchangers in the presence of non-uniformity of flow by developing the analytical model for exergy destruction. Entropy generation due to finite temperature difference and due to fluid friction is calculated and thereby the rise in the irreversibility is found. Their results bring out the reason behind the maximum entropy paradox in heat exchangers, the proper perspective of exergy destruction and the consequent optimization of cross flow heat exchangers from the second law viewpoint.

Taufiq et al. [15], in 2007, found the optimal thermal design of radial fin geometry having the heat interaction by convection, through the second law analysis using entropy generation minimization technique. The analysis involves the achievement of a balance between the entropy generation due to heat transfer and entropy generation due to fluid friction. The entropy generation rate is discussed and optimum thickness for fin array is determined on the basis of entropy generation minimization subjected to the global constraint. In addition, the influence of cost parameters on the optimum thickness of fin array is also considered. It has been found that the increase in cross flow fluid velocity will enhance the heat transfer rate that will reduce the heat transfer irreversibility.

In 2008, Wang et al. [16] applied the entropy generation method on the irreversibility of rotary air preheater in thermal power plant. Through the exergy analysis, the relationship between the efficiency of the thermal power plant and the total process of irreversibility in the rotary air preheater is built up. The major contributions of the entropy generation rate compared to the total irreversibility expressed in the entropy are identified: the entropy generation rate by heat transfer between air and gas, the entropy generation rate by the mixing of the exhaust gas with ambient, and the entropy generation rate by the pressure loss caused by friction. The various parameters like rotor height, channel ratio, leakage factor, leakage factor distribution and flow rate are considered as decision variables and by parametric variation, their effect on entropy generation rate and exergetic efficiency is analyzed.

Thermal Storage Systems

In 1999, Zubair et al. [17] applied the entropy generation minimization method to a sensible heat thermal energy storage system. They calculated the irreversibilities in the system in terms of entropy generated and appropriate monetary values are attached to the irreversible losses caused by the finite temperature difference heat transfer and pressure drop in the system. Including the other cost, they developed a new cost function called cost rate number and tried to minimize the cost by optimization.

Erek et al. [18], in 2008, used entropy generation minimization technique to analyze a latent heat storage system (around a cylindrical tube of shell and tube heat exchanger) during charging process. The numerical model of heat transfer fluid, pipe wall and phase change material for different parameters (shell radius and pipe length, *Re* number, inlet temperature of fluid etc.) is solved and extensive parametric studies are conducted to investigate how the solidification fronts, heat stored, heat transfer rates, entropy generation number and exergy efficiency change with time.

Heat Recovery Steam Generators

In 2007, Butcher et al. [19] carried out exergy analysis for waste heat recovery based power generation system. The temperature profiles across the heat recovery steam generator (HRSG), net work output, second law efficiency and entropy generation number are simulated for various operating conditions. The effect of pinch point on the performance of HRSG and on entropy generation rate and second law efficiency are also investigated. They observed that the second law efficiency of the HRSG and power generation system decreases with increasing pinch point. Moreover they observed that the first and second law efficiency of the power generation system varies with exhaust gas composition and with oxygen content in the gas. Their results provides the information on the role of gas composition, specific heat and pinch point influence on the performance of a waste heat recovery based power generation.

Compression Ignition Engine Using Biodiesel as Fuel

In 2007, Azoumah et al. [20] used entropy generation minimization method to optimize the performance of a compression ignition engine using bio-fuels such as cotton seed and palm oils, pure or blended with diesel for different engine loads. The previous studies involving engine using bio-fuels have evaluated their performance based on their brake power, brake thermal efficiency, brake specific fuel consumption (BSFC) and gas emissions analysis. By doing so, thermal pollution is ignored and the real performance of the engines regarding the second law of thermodynamics is overlooked. Therefore the entropy change due to the dumping of the waste heat in to the environment is also considered and a trade-off zone of engine loads (60% and 70% of the maximum load) was established between the gas emissions (NO and CO_2) and the exergy efficiency for optimal performance of the CI engine.

Hydrogen Combustion

In 2008, Rakopoulos et al. [21] suggested that during combustion of hydrogen, the reaction is a combination of two relatively simple molecules into a more complicated one. While hydrocarbon combustion, during which relatively complex molecules are destroyed and a multitude of lighter fragments is produced in a process that obviously generates large amounts of entropy. Therefore, hydrogen and methane mixture is provided and exergy analysis is carried out in which the entropy generation is tested as a function of hydrogen content of the fuel. It is observed that with increasing hydrogen content, the irreversibility produced during combustion decreases as a percentage of total injected fuel availability, and the second-law efficiency increases.

Table 2.1 summarizes the various investigations reviewed. Amongst the various refrigeration systems analyzed by investigators, focus is found to be on vapour absorption systems as they are heat energy intensive systems.

Table 2.1 Summary of Investigations on Various Thermal Systems Using EGM

System	Investigators	Year	Remarks
Refrigeration Systems			
Vapour absorption refrigeration (VAR) systems	Bejan A, Vargas J V C & Sokolov M	1995	Found the way of dividing a finite supply of heat exchanger surface between generator, condenser and evaporator for maximizing the cooling load at evaporator.
	Talbi M M & Agnew B	2000	Quantified the irreversibility of each component of the chiller to determine the potential for each component to contribute to overall exergy efficiency of LiBr system.
	Ezzine N B, Barhoumi M, Mejbri K, Chemkhi S & Bellagi A	2004	Quantified the irreversibility of each component of the chiller to determine the potential for each component to contribute to overall exergy efficiency of aqua ammonia double effect absorption chiller.
	Adewusi S A & Zubair S M	2004	Showed that, for two stage aqua ammonia system has a higher entropy generation and coefficient of performance(COP), while the single-stage system has a lower entropy generation and COP
	Kaynakli O	2008	Determined the variation of the exergetic efficiency with cooling water flow rate, solution flow rate, cooling water temperature and solution concentration for coil absorber of the LiBr absorption system.
Vapour compression refrigeration(VCR)system	Yumrutas R, Kunduz M and Kanoglu M	2002	Investigated the effects of the evaporator and condenser temperatures on various losses and on the COP of the system
Transcritical carbon dioxide refrigeration cycle (heat pump)	Yang J L, Ma YT, Li M X & Guan H Q	2005	Investigated the effects of evaporating temperature and outlet temperature of gas cooler on the optimal heat rejection pressure, COP, the exergy losses, and the exergy efficiencies.
	Sarkar J, Bhattacharyya S & Gopal M R	2009	Presented CO_2 heat pump for combined cooling and heating application, optimization of heat exchanger tube diameter and length and presented the effect of design parameters on overall system performance.

Other Thermal Systems			
Adsorption cycle	Pons M	1996	Described the optimal operation of an adsorption cycle and presented optimal conditions leading to correlations between different internal entropy productions
Thermal energy storage system	Zubair S M and Al-Naglah M A	1999	Calculated irreversibility in a sensible heat storage system in terms of entropy generated and attached appropriate monetary values to the irreversible losses and tried to minimize the cost by optimization.
	Erek A and Dincer I	2008	Analyzed a latent heat storage system during charging process
Heat exchanger	Cornelissen R L and Hirs G	1997	Combined entropy generation minimization method and the life cycle analysis (LCA) and found the design conditions of the heat exchangers which lead to the lowest life cycle irreversibility.
	Gupta A and Das S K	2007	Calculated entropy generation due to finite temperature difference and due to fluid friction and found increase in the irreversibility for cross flow heat exchanger.
	Wang H Y, Zhao L L, Zhou Q T, Xu Z G and Kim H T	2008	Examined irreversibility in the rotary air preheater and analysed its effect on the efficiency of thermal power plant.
Radial fin geometry	Taufiq B N, Masjuki H H, Mahlia T M I, Saidur R, Faizul M S & Mohamad E N	2007	Found the optimal thermal design of radial fin geometry having the heat interaction by convection.
Heat Recovery Steam Generator	Butcher	2007	Presented, for waste heat recovery plant, the effect of pinch point on the performance of HRSG. Observed decrease in the second law efficiency and power generation rate for HRSG with increase in pinch point.
C I Engine using biodiesel as fuel	Azoumah Y, Blin J and Daho T	2007	Analyzed the performance of CI engine using various types of bio-fuels for various load conditions.
Hydrogen combustion process	Rakopoulos C D, Scott M A, Kyritsis D C and Giakoumis E G	2008	Showed that, with increasing hydrogen content, the irreversibility produced during combustion decreases as a percentage of total injected fuel availability, and the second-law efficiency increases.

2.1.2 Exergy Destruction Method (EDM)

An exergy balance states that the total exergy increase or decrease within the system boundary plus the exergy destruction within the same boundary equals the difference between the total exergy transfers in and out across the boundary. The exergy transfer across the boundary includes exergy transfer associated with the transfer of heat, work and mass entering and leaving the boundary across the boundary. Exergy destruction method is based on the above observations. The various studies reported are reviewed in this section.

2.1.2.1 Refrigeration & Air-conditioning Systems

This section is devoted to the review of various studies carried out on various refrigeration systems, air-conditioning systems and combined refrigeration and other thermal systems using exergy destruction method.

Vapour Absorption Refrigeration Systems

In 1986, Alvares et al. [22, 23] simulated the AAVAR system and carried out exergy analysis using exergy destruction method. They tried to analyze the effect of generator temperature and evaporator pressure on exergetic COP and tried to optimize the system.

In 1990, Karakas et al. [24] carried out Second-Law analysis of Solar Absorption Cooling Cycles using LiBr/Water and Ammonia/Water as working Fluids. Ataer et al. [25], in 1991, studied the irreversibilities in components of AAVAR system like condenser, evaporator, absorber, generator, pump, expansion valves, mixture heat exchanger and refrigerant heat exchanger, by second law analysis. Pressure losses between the generator and condenser, and the evaporator and absorber are taken into consideration. The dimensionless exergy loss of each component, the exergetic coefficient of performance, the coefficient of performance and the circulation ratio are given graphically for different generator, evaporator, condenser and absorber temperatures. They concluded that the evaporator and absorber of the absorption refrigeration system are the components in which, within the given operating conditions, high exergy loss is observed and they should be modified to give a better system performance. For each condenser, absorber and evaporator temperature, there is a generator temperature at which the dimensionless total exergy loss of the system is a minimum. At this point, the COP and Exergetic Coefficient of Performance of the system are at a maximum. Consequently the results of the second law analysis can be used to identify the less efficient components of the system and also to modify them. Moreover, the suitability of the selected components can be judged by this analysis. The second law analysis may be a good tool for the determination of the optimum working conditions of such systems.

In 1995, Aphornratana et al. [26] provided the second law method as applied to a single-effect LiBr/Water VAR system. Exergy at each salient point is found and exergy analysis of each component is carried out and found the effect of variation of various parameters like generator temperature, solution heat exchanger effectiveness, solution circulation rate, evaporator temperature, and condenser temperature is analyzed.

In 1998, Ravikumar et al. [27] carried out exergy analysis of double effect LiBr/water VAR system. He showed the exergy variation across the individual component with respect to generator temperature and found the second generator more effective.

In 2004, Kilic et al. [28] developed mathematical model using exergy analysis for LiBr-Water VAR system. The effect of main system temperatures on the performance parameters of the system, irreversibilities in the thermal process and non-dimensional exergy loss of each component are analyzed. The results show that the performance of the absorption refrigeration system increases with increasing generator and evaporator temperatures, but decreases with increasing condenser and absorber temperatures.

In 2005, Sencan et al. [29] carried out exergy analysis of single-effect LiBr/water VAR system. Exergy loss, enthalpy, entropy, temperature, mass flow rate and heat rate in each component of the system are evaluated. They concluded that the condenser and evaporator heat loads and exergy losses are less than those of the generator and absorber. This is due to the heat of mixing in the solution, which is not present in pure fluids.

In 2008, Morosuk et al. [30] suggested a new approach to the exergy analysis of VAR machines. Exergy destruction in a component can be split into endogenous and exogenous parts. The endogenous part of exergy destruction, associated only with the irreversibility occurring within the component when all other components operate in an ideal way and the component being considered operates with its current efficiency. The exogenous part of exergy destruction within the component is caused by the irreversibility that occurs in the remaining components. These splitting enable engineers working in system optimization to estimate the exergy destruction in a component caused by the component itself on one hand and by the remaining components on the other hand. This information can be used to decide whether engineers should focus on the component being considered or on the remaining system components, in order to effectively improve the overall performance. Again the exergy destruction in a component can be divided in unavoidable and avoidable parts. The exergy destruction rate that cannot be reduced due to technological limitations such as availability and cost of materials and manufacturing methods is the unavoidable part of the exergy destruction. The remaining part represents the avoidable part of the exergy destruction. Thus, splitting the exergy destruction into unavoidable and avoidable parts in the component provides a realistic measure of the potential for improving the thermodynamic efficiency of a component. A conventional exergy analysis (without splitting the exergy destruction) would suggest that components should be improved in the following order: absorber (40.4%), generator (39.2%), condenser (16.4%) and evaporator (1.2%).But the information provided through the splitting of the exergy destruction shows that 65.8 % of the total exergy destruction within the absorption refrigeration system is unavoidable.

In 2008, Gomri et al. [31] carried out Second law analysis of double effect LiBr/water VAR system using exergy destruction method. It is observed that that the performance of the system increases with increasing low pressure generator (LPG) temperature, but decreases with increasing high pressure generator (HPG) temperature. The highest exergy loss occurs in the absorber and in the HPG, which therefore makes the absorber and HPG the most important components of the double effect refrigeration system.

Similar type of analysis of single effect and double effect LiBr/water VAR system is carried out by Gomri [32] in 2009. It is concluded that the COP of double effect system is approximately twice the COP of single effect system but the exergetic efficiency of double effect system increase slightly compared to the exergetic efficiency of single effect system. It is found that for each condenser and evaporator temperature, there is an optimum generator temperature where the total change in exergy of the single effect and double effect absorption refrigeration systems is minimum. At this point the COP and exergetic efficiency of the systems become maximum.

It should be noted that most of the studies are focussed on LiBr/ water VAR systems and few studies are reported on AAVAR systems.

Vapour Compression Refrigeration Systems

In 1987, Mastrullo et al. [33] conducted exergetic analysis of multi stage VCR systems using R12. Plant exergetic efficiencies, equipment irreversibility, and their sensitivity to main system parameters are evaluated for several typical component arrangements. The use of a flash tank for separation, desuperheater, and with a subcooling coil seemed the best solution.

In 1988, Kumar et al. [34] explained the method of carrying out an exergetic analysis on a vapour compression refrigeration system using R11 and R12 as refrigerants. Exergy-Enthalpy diagrams are presented for these two refrigerants which facilitate the analysis. The procedure to calculate the various losses occurring in different components, as well as the coefficient of performance and the exergetic efficiency of the refrigeration cycle, has been explained by means of a numerical example. In 1993, Lohlein et al. [35] did the exergy analysis of low temperature refrigerators for large scale cooling system and check variety of arrangements of component.

In 2002, Aprea et al. [36] compared VCR systems with R22 and R407C on the base of exergetic analysis and found that the overall exergetic performance of the plant working with R22 is consistently better.

In 2003, Srinivasan at el, [37] carried out exergetic analysis of carbon dioxide VCR cycle using the new fundamental equation of state and prepared temperature v/s exergy chart and enthalpy v/s exergy chart. There exist upper and lower bounds for the high cycle pressure for a given set of evaporating and pre-throttling temperatures. The maximum possible exergetic efficiency for each case was determined. Empirical correlations for exergetic efficiency and COP, valid in the range of temperatures studied, are obtained and the exergy losses are quantified. In 2004, Fartaj et al. [38] analyzed transcritical CO_2 refrigeration cycle. By exergy loss analysis they showed that the compressor and the gas cooler exhibit the largest non-idealities within the system.

In 2008, Dopazo et al. [39] has analyzed a cascade refrigeration system with CO_2 and NH_3 as working fluids in the low and high temperature stages, respectively using exergy destruction method. After calculating exergy flow at inlet and outlet of all the components, exergetic efficiency of the components and system is evaluated. The effect of parametric variation of various decision variables on COP and exergetic efficiency is found and subsequently the exergetic optimization of the system is carried out.

In 2009, Mafi et al. [40] exegetically analyzed the multistage cascade low temperature refrigeration systems having closed cycle propylene and ethylene systems, through exergy destruction method. Propylene refrigeration is utilized at several temperature levels to cool and heat the feed in the initial fractionation sections of the plant while the ethylene refrigeration is utilized at several temperature levels to cool the feed in the cryogenic section of the plant. The equations of exergy destruction and exergetic efficiency for the main system components such as heat exchangers, compressors and expansion valves are developed and combining them expression for minimum work requirement for the refrigeration systems is developed. It shows that the minimum work depends only on the properties of incoming and outgoing process streams cooled or heated with refrigeration system and the ambient temperature.

Air-Conditioning Systems

In 2002, Bilgen et al. [41] carried out exergy analysis of air conditioner system. The irreversibilities due to heat transfer and friction have been considered. The coefficient of performance based on the first law of thermodynamics as a function of various parameters, their optimum values, and the efficiency and coefficient of performance based on exergy analysis have been derived. Based on the exergy analysis, a simulation program has been developed to simulate and evaluate experimental systems. The simulation of a domestic heat pump air conditioner is then carried out using experimental data. It is found that COP based on the first law varies from 7.40 to 3.85 and the exergy efficiency from 0.37 to 0.25 both a decreasing function of heating or cooling load. The exergy destructions in various components are determined.

In 2009, Wei et al. [42] applied the exergy destruction method to Variable Air Volume type Air Conditioning system. Exergy of air volume is calculated by considering the humidity and partial pressure of water vapour in the air. Exergy efficiency is calculated considering the equivalent CO_2 emissions due to the generation of electricity used by the VAV system. It is found that the exergy efficiency is only 2-3% of potential work that can be developed by using these energy sources as supplied for satisfying the environmental thermal conditions for human occupancy and indoor air quality.

Combined Refrigeration and Other Thermal Systems

In 2006, Vidal et al. [43] performed exergy analysis for the combined power and refrigeration cycle, also known as Goswami cycle, in which the AAVAR system generates cooling and expansion of refrigerant take place in vapour turbine generates power. Through analysis, it is showed that the solar collectors or waste heat can be the best heat sources to operate the cycle.

In 2008, Dai et al. [44] proposed combined power and refrigeration cycle, which combines the Rankine cycle and the ejector refrigeration cycle, produces power output and refrigeration output simultaneously. An exergy analysis is performed and exergy destruction in each component is calculated. A parameter optimization is achieved by means of genetic algorithm to reach the maximum exergy efficiency. The results show that the biggest exergy loss due to the irreversibility occurs in heat addition processes, and the ejector causes the next largest exergy loss. It is also shown that the turbine inlet pressure, the turbine back pressure, the condenser temperature and the evaporator temperature have significant effects on the turbine power output, refrigeration output and exergy efficiency of the combined cycle.

In 2008, Khaliq [45] analyzed trigeneration system generating electricity, process heat and cooling effect. In the system, gas turbine cycle is combined with heat recovery steam generator and LiBr/water VAR system. The exhaust gas from gas turbine is supplied to heat recovery steam generator to generate process steam. The gas coming out of HRSG is supplied to the generator of LiBr/water VAR system to produce cooling effect. Applying the exergy destruction method, the effect of overall pressure ratio, turbine inlet temperature, pressure drop in combustor and heat recovery steam generator, and evaporator temperature on the exergy destruction in each component is investigated. It is observed that maximum exergy is destroyed during the combustion and steam generation process; which represents over 80% of the total exergy destruction in the overall system. In 2009, Kelly et al. [46] applied the same concept to the VCR system and gas turbine system.

2.1.2.2 Other Thermal Systems

Various investigations carried out using EDM on other thermal systems such as various power plants, solar energy based systems, thermal storages systems, heat exchangers, bio mass gasifier, combustion process, cooling tower boilers and fuel cells are discussed in this section.

Power Generation Systems

Nag et al. [47] presented first and second law analysis of a combined cycle power plant using pressurized circulating fluidized beds for partial gasification and combustion of coal. They evaluated the effects of pressure ratio and peak cycle temperature ratio of the gas cycle and the lower saturation pressure of the steam cycle on the overall performance of the combined plant.

In 2006, Hamed et al. [48] applied the exergy destruction method to power/water cogeneration plant. They calculated the exergy destruction in each component and calculated the appropriate cost of it. They allocated the cost of exergy destruction and cost of various components to water and electricity production appropriately which he called direct cost allocation method and found minimum product cost.

In 2008, Abusoglu et al. [49] applied the exergy destruction method to the diesel engine powered cogeneration systems generating electricity and steam. After defining the fuel and product in terms of exergy flow for each component of the system and calculated the exergetic efficiency of them. It is observed that total exergy destruction in the engine is mostly due to the highly irreversible combustion process in the engine, heat losses from engine and friction.

In 2008, Rakopoulos [50] developed the zero-dimensional, multi-zone, thermodynamic combustion model based on second law analysis, by dividing the burned gas into several distinct zones, in order to account for the temperature and chemical species stratification developed in the burned gas during combustion, for the prediction of spark ignition (SI) engine performance and nitric oxide (NO) emissions. Total exergy including thermal, mechanical and chemical for fuel at each salient point is calculated. By applying the exergy balance method, the exergetic efficiency for each zone is calculated. By changing the air fuel ratio, it is revealed that the crucial factor determining the thermodynamic perfection of combustion in each burned zone is the level of the temperatures at which combustion occurs in the zone, with minor influence of the whole temperature history of the zone during the complete combustion phase.

In 2008, Som et al. [51] did the exhaustive review of the exergy analysis of combustion system. They defined the exergetic efficiency and rate of irreversibility for combustion system and using both method of exergy analysis; exergy balance and entropy generation minimization carried out the exergetic analysis of combustion system and compared the combustion of solid, liquid and gaseous fuel. They found that the major source of irreversibilities is the internal thermal energy exchange associated with high temperature gradients caused by heat release in combustion reactions and to reduce the exergy destruction in the combustion process, the irreversibility should be reduced through proper control of physical processes and chemical reactions resulting in a high value of flame temperature but lower values of temperature gradients within the system and optimum condition can be determined.

In 2008, Kanoglu et al. [52] performed exergy analysis of a binary geothermal power plant using exergy destruction method. Exergy destruction throughout the plant is quantified and illustrated using an exergy diagram, and compared to the energy diagram. The exergy and energy efficiencies are calculated for the entire plant and for the individual plant components. The sites of exergy destruction are identified and quantified. Also, the effects of turbine inlet pressure and temperature and the condenser pressure on exergy and energy efficiencies, the net power output and the brine reinjection temperature are investigated and the trends are explained.

In 2009, Aljundi [53] did energy and exergy analysis of a steam power plant in Jordan using exergy destruction method. The performance of the plant was estimated by a component wise modelling and a detailed break-up of energy and exergy losses. It shows that the thermal efficiency (26%) is low compared to modern power plants because this efficiency was not based on the specific heat input to the steam; rather, it was based on the lower heating value of the fuel to incorporate the losses occurring in the furnace-boiler system due to energy lost with hot gases, incomplete combustion, etc. It is observed that maximum exergy destruction is there in boiler and maximum exergy loss in condenser.

Kamate et al. [54], in 2009, analyzed cogeneration power plants in sugar industries through exergy destruction method for various steam inlet condition. The results shows that, at optimal steam inlet conditions of 61 bar and 475ºC, the backpressure steam turbine cogeneration plant perform with energy and exergy efficiency of 0.863 and 0.307 and condensing steam turbine plant perform with energy and exergy efficiency of 0.682 and 0.260, respectively. Boiler is the least efficient component and turbine is the most efficient component of the plant.

Solar Energy Based Systems

In 2007, Gunerhan et al. [55] analyzed the solar water heating systems for building applications. The system consists of namely a flat plate solar collector, a heat exchanger (storage tank) and a circulating pump. In the analysis, the exergy destruction method is used and irreversibility as per the guy stodola's law is introduced. Using exergy and irreversibility, few parameters such as fuel depletion ratio, relative irreversibility, productivity lack, exergetic factor and exergetic improvement potential are defined. Exergy destructions (irreversibilities) as well as exergy efficiency relations are determined for each of the system components and the whole system. Exergy efficiency correlations for the solar collector are presented to determine its exergetic performance. The effect of varying water inlet temperature to the collector on the exergy efficiencies of the Solar Water Heating system components is investigated and presented in the form of an exergy efficiency curve similar to the thermal efficiency.

In 2009, Celma et al. [56] analyzed solar drying process through exergy analysis. Using the first law of thermodynamics, energy analysis was carried out to estimate the amounts of energy gained from solar air heater and the ratio of energy utilization of the drying chamber. Also, applying the second law, exergy analysis was developed to determine the type and magnitude of exergy losses during the solar drying process. It was found that exergy losses took place mainly during the second day, when the available energy was less used.

In 2008, Zhai et al.[57] carried out the exergy analysis using exergy destruction method, of a small scale hybrid solar heating, chilling and power generation system, including parabolic trough solar collector with cavity receiver, a helical screw expander and silica gel-water adsorption chiller and the power generation cycle at lower temperature level. It is found that both the main energy and exergy loss take place at the parabolic trough collector. The economical analysis in terms of cost and payback period has been carried out using life cycle cost analysis

(LCCA). In LCCA, costs are grouped into three categories, capital expense for equipment and installation, operation & maintenance and fuel costs, and costs are incurred in demolition and disposal of the system while the equipment can have some salvage value. It is observed that the payback period is about 18 years in the present energy price condition.

In 2010, Gupta et al. [58] applied exergy analysis to the direct steam generation solar–thermal power plant in which steam is generated in solar collector and expanded in steam turbine to generate power. It is found that the maximum exergy loss is in the solar collector field while in other plant components it is small. The application of exergy destruction method is found for conventional power plant either based on gas cycle or on steam cycle.

In 2010, Baghernejad et al. [59] carried out exergy analysis, by the same method, of an integrated solar combined cycle system. They identified the causes of exergy destruction and their numerical values for the various components like combustor, collector, heat exchangers, pump and turbines.

In 2008, Torchia-Nunez et al. [60] presented a steady-state and transient theoretical exergy analysis of a solar still, focused on the exergy destruction in the components of the still: collector plate, brine and glass cover. The energy balance for each component resulting in three coupled equations where three parameters—solar irradiance, ambient temperature and insulation thickness are studied. The energy balances are solved to find temperatures of each component; these temperatures are used to compute energy and exergy flows. It is observed that the irreversibilities produced in the collector account for the largest exergy destruction, whereas irreversibility rates in the brine and in the glass cover are negligible. For the same exergy input a collector, brine and solar still exergy efficiency are calculated.

In 2009, chow et al. [61] proposed the exergy analysis of photovoltaic thermal collector with and without glass cover. From the exergy analysis, the increase of PV cell efficiency, packing factor, water mass to collector area ratio, and wind velocity are found favourable to go for an unglazed system, whereas the increase of on-site solar radiation and ambient temperature are favourable for a glazed system.

In 2009, Farahat et al. [62] exegetically optimized flat plate solar collector through exergy destruction method. After calculating the exergy flows, losses and exergetic efficiency of flat plate solar collector, the exergetic optimization is carried out under given design and operating conditions and the optimum values of the mass flow rate, the absorber plate area and the maximum exergy efficiency have been found.

Various researchers [63 to 68] applied this tool to find the losses in the various component of power plant and tried to improve the performance of the same.

Miscellaneous Thermal Systems

In 2001, Yilmaz et al. [69] carried out exergy analysis of heat exchanger using EGM and EDM and compared the output of both the methods.

In 2007 Talens et al. [70] suggested the use of Exergy Flow Analysis as an environmental assessment tool to account wastes and emissions, determine the exergetic efficiency, compare substitutes and other types of energy sources: all useful in defining environmental and economical policies for resource use with the example of process of bio-diesel production. The results show that the production process has a low exergy loss. The exergy loss is reduced by using potassium hydroxide and sulphuric acid as process catalysts and can further be minimized by improving the quality of the used cooking oil.

In 2007, Ptasinski et al. [71] carried out exergetic analysis of biomass gasification plant. They compared different types of bio-fuels for their gasification efficiency and benchmark against gasification of coal. In order to quantify the real value of the gasification process exergy-based efficiencies, defined as the ratio of chemical and physical exergy of the synthesis gas to chemical exergy of a bio-fuel, are proposed.

In 2008, Toonssen et al. [72] applied the exergy destruction method to hydrogen production plant based on biomass gasification. Three types of gasification processes are compared, battelle gasification process, the fast internal circulating fluidized bed gasifier (FICFB) and the Blaue Turm gasification process. The processes are compared on the basis of exergetic efficiency and fond that FICFB gasification process is less efficient.

In 2008, Rashidi et al. [73] applied the exergy destruction method for exergy analysis to the hybrid molten carbonate fuel cell system. A parametric study is performed to examine the effect of varying operating pressure, temperature and current density on the performance of the system. Thermodynamic irreversibilities in each component of the system are determined. An overall energy efficiency, exergy efficiency, bottoming cycle energy efficiency and stack energy efficiency are calculated. The results demonstrate that increasing the stack pressure decreases the over potential losses and, therefore, increases the stack efficiency.

In 2008, Obara et al. [74] investigated the exergy flow and exergy efficiency of a proton-exchange-membrane (PEM) fuel cell was investigated using exergy destruction method. The exergetic efficiency of the system was calculated and the effect of change in the environment temperature on the exergetic efficiency was analyzed.

In 2008, Wang et al. [75] investigated the possibility of waste heat recovery in cement industry from the preheater exhaust and clinker cooler exhaust gases in cement plant. For this task, single flash steam cycle, dual-pressure steam cycle, organic rankine cycle and the kalina cycle are identified. The exergy analysis is examined, and a parameter optimization for each cogeneration system is achieved by means of genetic algorithm to reach the maximum exergy efficiency. The optimum performances for different cogeneration systems are compared under the same condition. The results show that the exergy losses in turbine, condenser, and heat recovery vapour generator are relatively large, and reducing the exergy losses of these components could improve the performance of the cogeneration system. Compared with other systems, the Kalina cycle could achieve the best performance in cement plant.

In 2008, Muangnoi et al. [76] analyzed the influence of the ambient temperature and humidity on the performance of a counter flow wet cooling tower according to the second law, exergy destruction method. The air is considered as the mixture of air and steam and proposed the method of calculation of its exergy. Exergy analysis then has been carried out for investigating the cooling tower performance with various inlet air conditions, relative humidity and dry bulb temperature, while the water side condition is kept constant. The similar result in terms of required dry air flow rate, exergy change of water and that of air, exergy destruction and second law efficiency were obtained for the various inlet air conditions.

In 2010, Saidur et al. [77] proposed the exergy analysis of industrial boiler. They showed that the total exergy destruction in the boiler is equal to the sum of exergy destruction in combustion zone and that in evaporation zone.

From the above review, it is seen that both the exergy analysis, viz. EGM and EDM are used to analyse various thermal systems. There are a number of studies carried out on vapour absorption refrigeration systems with LiBr/water and aqua ammonia as absorbent-refrigerant fluids. However, studies on AAVAR systems are relatively few, in spite of the fact that both are heat energy intensive systems and equally popular. Table 2.2 gives the summary of the various investigations on different system using exergy destruction method.

Table 2.2 Summary of Investigations on Various Thermal Systems Using EDM

System	Investigator	Year	Remark
Refrigeration & Air-conditioning Systems			
Vapour Absorption Refrigeration (VAR) System	Alvares et al.	1986	Analyzed the effect of generator temperature and evaporator pressure on exergetic COP and tried to optimize the aqua ammonia vapour absorption system
	Karakas et al.	1990	Performed exergetic analysis and found various losses in LiBr/ water and AAVAR system using solar energy as heat source
	Ataer et al.	1991	Studied the irreversibilities in the components of AAVAR system
	Aphornratana et al	1995	For single effect LiBr VAR system, studied the effect of various parameters on the performance of the system through exergy analysis.
	Ravikumar et al	1998	Checked the effect of generator temperature for double effect Li Br vapour absorption system
	Kilic et al	2004	Analyzed the effect of main system temperatures on the performance parameters of the system, irreversibilities in the thermal process and non-dimensional exergy loss of each component of LiBr VAR system.
	Sencan et al	2005	Evaluated the exergy loss, enthalpy, entropy, temperature, mass flow rate and heat rate in each component of the LiBr/water VAR system.
	Morosuk et al.	2008	Identified endogenous and exogenous parts of exergy destruction in absorption system .Estimated the exergy destruction in a component caused by the component itself on one hand and by the remaining components on the other hand
	Gomri et al.	2008	Analyzed the effect of effect of temperature of LPG and HPG on the performance of LiBr/water VAR system.
	Gomri et al.	2009	Compared the COP of single effect and double effect LiBr/water VAR system.
Vapour compression refrigeration system	Mastrullo et al.	1987	Evaluated, for multi stage VCR system, plant exergetic efficiencies, equipment irreversibility, and their sensitivity to main system parameters.

	Kumar et al.	1988	Compared the exergetic performance of VCR, using R11 and R12 as refrigerants
	Lohlein et al.	1993	Checked variety of arrangements of component in low temperature refrigerators through exergy analysis.
	Aprea et al.	2002	Compared the performance of the system with refrigerant R22 and R407C.
	Srinivasan at el	2003	Analyzed the effect of gas cooler pressure on the exergetic efficiency of the system.
	Fartaj at el	2004	Analyzed transcritical CO_2 refrigeration cycle and showed that the compressor and the gas cooler exhibit the largest non-idealities within the system.
	Dopazo et al.	2008	Analyzed a cascade refrigeration system with CO_2 and NH_3 as working fluids in the low and high temperature stages, respectively
	Mafi et al.	2009	Analysed cascade refrigeration and shows that the minimum work depends only on the properties of incoming and outgoing process streams cooled or heated with refrigeration system and the ambient temperature.
Air-Conditioning Systems			
Air conditioning system	Bilgen et al.	2002	Compared the performance based on first law and second law.
	Wei et al.	2009	Analyzed the Variable Air Volume type Air Conditioning system.
Combined Refrigeration and Other Thermal Systems			
Cogen Plant	Vidal et al	2006	For power/refrigeration plant, showed that the solar collectors or waste heat can be the best heat sources to operate the cycle.
	Dai et al.	2008	For combined, power and refrigeration cycle, which combines the rankine cycle and the ejector refrigeration cycle, performed exergy analysis
	Khaliq	2008	Analyzed trigeneration system generating electricity, process heat and cooling effect.
	Kelly et al.	2009	Applied the concept of exergy splitting to VCR system and gas power plant suggested by Morosuk
Other Thermal Systems			
Power generation system	Nag et al.	1995	Evaluated the effects of pressure ratio and peak cycle temperature ratio of the gas

			cycle and the lower saturation pressure of the steam cycle on the overall performance of the combined plant.
	Hamed et al.	2006	Calculated the exergy destruction in each component of power/water cogen plant and calculated the appropriate cost of it.
	Abusoglu et al.	2008	Calculated the exergetic efficiency for diesel engine powered cogeneration systems generating electricity and steam.
	Rakopoulos	2008	Developed the zero-dimensional, multi-zone, thermodynamic combustion model based on second law analysis.
	Som et al	2008	compared the combustion of solid, liquid and gaseous fuel using both methods
	Kanoglu et	2008	Analyzed the effect of turbine inlet pressure and condenser pressure, on the exergetic performance of binary geothermal power plant.
	Aljundi	2009	Analyzed steam turbine based power plant.
	Kamate et al.	2009	Identified the optimal steam inlet condition to steam turbine exergetic point of view.
Solar Energy Based Systems			
Solar heating system	Gunerhan et al.	2007	Investigated the effect of various parameters on the performance of the system.
	Celma et al.	2009	Analyzed solar drying process and estimated the exergy losses.
Solar trigeneration system	Zhai et al.	2008	For solar heating, chilling and power generation system, found the exergy loss in various components.
Solar power plant	Gupta et al.	2010	Analyzed direct steam generation solar power planr
	Baghernejad et al.	2010	Identified exergy destruction in the various components of solar combine cycle system
Solar still	Torchia-Nunez et al.	2008	Steady-state and transient theoretical exergy analysis of a solar still, focused on the exergy destruction in the components
Solar collector	chow et al.	2009	Analyzed photovoltaic thermal collector with and without glass cover.
	Farahat et al	2009	optimized flat plate solar collector.
Miscellaneous Thermal Systems			

Heat exchanger	Yilmaz et al.	2001	Compared EGM and EDM method
Gasifier	Ptasinski et al.	2007	Compared different types of bio-fuels for their gasification efficiency
	Toonssen et al.	2008	
Fuel cell	Rashidi et al.	2008	For the hybrid molten carbonate fuel cell, examined the effect of varying operating pressure, temperature and current density on the performance of the system.
	Obara et al.	2008	Investigated the exergy flow and exergy efficiency of a proton-exchange-membrane fuel cell.
Cooling tower	Muangnoi et al.	2008	Analyzed the influence of ambient temperature and humidity on the performance of the system.
Boiler	Saidur et al.	2010	showed that the total exergy destruction in the boiler is equal to the sum of exergy destruction in combustion zone and that in evaporation zone

2.1.3 Comparison between EGM and EDM

Table 2.3 gives comparison of EGM and EDM. It is seen that both the methods are useful in predicting the quality and quantity of energy utilized in thermal systems. However, it is reported that EGM poses difficulties in combining both exergy and cost analysis. Exergoeconomic analysis thus, needs the exergy balance and exergy destruction method (EDM) along with a suitable cost analysis.

Table 2.3 Comparison of the Exergy Analysis Methods

Method	Overview of Method	Remark
EGM	• Individual components are identified. • For each component, entropy associated with inlet and outlet flow is calculated. • Entropy change is multiplied with environment temperature and exergy destruction is calculated. • Loss in each components are found	• All components can be analyzed. • Only destruction can be found but fuel and product cannot be identified. • Difficult to combine with Exergoeconomic analysis
EDM	• For each component, exergy flow at inlet and outlet is calculated. • Difference in exergy at inlet and outlet is considered as fuel or product as per the application. • Unaccounted exergy is considered as exergy destruction.	• Components like throttle valve cannot be analyzed. • For some components, difficult to identify fuel and product. More than one component to be combined as a single component. • Easy to combine with exergoeconomic analysis.

2.2 Exergoeconomic Analysis

The exergy analysis yields the desired information for a complete evaluation of the design and performance of an energy system from the thermodynamic viewpoint. However, one still needs to know how much the exergy destruction in a plant component costs the plant operator. Knowledge of this cost is very useful in improving ('optimizing') the cost effectiveness of the plant. Exergy not only is an objective measure of the thermodynamic value of an energy carrier but also is closely related to the economic value of the energy carrier, because users pay only for the useful part of energy. Exergy costing is based on the notion that exergy is the only rational basis for assigning costs to energy carriers and to 'energy waste' (exergy destruction and exergy losses, respectively). Thus, exergy costing uses costs per exergy unit. Exergoeconomics is based on exergy costing and is usually applied at the plant-component level. This section will deal with the exergoeconomic analysis carried out by various investigators.

One of the ways to apply exergy costing is to charge throughout the plant for exergy destruction and exergy losses at a uniform cost per exergy unit equal to the average cost per exergy unit of the fuel of the total plant. This approach, however, does not consider that the importance of exergy destruction and exergy loss, from both the thermodynamic and economic viewpoints, depends on the relative position of the sub system where the exergy destruction occurs within the total plant. For example, one MW of exergy destruction rate in the low-pressure steam turbine affects the cost of electricity more than an exergy destruction rate of one MW in the boiler of a steam power plant. Therefore, more sophisticated approaches to exergy costing are required.

In 1981, Shiran et al. [78] tried to apply thermodynamic analysis based on first law and economic analysis in combination to AAVAR system and tried to optimize the system thermoeconomically.

In 1982, London [79] tried to relate exergy and economy using the case of steam power plant. Starting with recognition of the individual internal and relevant external irreversibility thermodynamic arguments are used to formulate both entropy and energy measures in terms of operating conditions. The energy measures, lead to economic pricing relating to system energy expenditure and sometimes system energy rating penalties. The analysis loop is closed by considerations relating to the reduction of the individual irreversibility in terms of trade-off factor. The available energy or exergy analysis provides an answer to the overall costs of the collective internal irreversibility.

In 1989, Duarte et al. [80] tried to analyze optimal working conditions for an absorber heat transfer analysis of the LiBr/water theoretical cycle. They showed that the optimal circulation ratio will give the maximum profit for a given investment. The cost of each piece of equipment is an increasing function of its heat exchange surface area, which is proportional to the heat transfer. The surface area depends also upon the design characteristics of the equipment, but the product (UA) is only a function of LMTD and heat transfer rate. A relation between the temperature uplift and heat transfer driving force Δt^h (temperature change for hot fluid) is suggested by them. When Δt^h diminishes, the circulation ratio increases but, the LMTD diminishes, and therefore the parameter (UA) increases which would increase proportionally the equipment cost. It can therefore be deduced that the optimal value of Δt^h must be calculated from a careful economical evaluation.

In 1996, Saghiruddin et al. [81] applied the energy costing to VAR system and using three types of working fluid and three type of energy source. The costs of the sources of energy estimated for the solar collector areas and volume flow rates of biogas and LPG, for a typical operating condition in the absorption cycles with respect to generator temperature by plotting the variation of cost against the generator temperature and found minimum corresponding cost for optimum generator temperature iteratively.

In 2003, Zhang et al. [82] carried out thermoeconomic optimization of small size central air conditioner. They carried out exergy analysis on the basis of entropy generation minimization and defined cost function to be minimized.

In 1998, Kim et al. [83], proposed another method for exergoeconomic analysis of thermal system with the application to power plant. They derived general cost balance equation which can be applied to any component of the system. The exergy of the material stream decomposed in to thermal mechanical and chemical exergy flows and an entropy production flow. A unit exergy cost is assigned to them. Then the set of equations for the unit cost of various exergies is obtained by applying the cost balance to each component of the system and to each junction. The monetary evaluations of various exergy cost, as well as production cost of electricity are obtained by solving the set of equations. The lost cost of each component also can be obtained.

In 2004, Kwaka et al. [84] carried out Exergetic and thermoeconomic analysis of a phosphoric acid fuel cell plant. The above publications are the individual efforts and not following any systematic methodology. In the following section, few standard exergoeconomic methods are explained. In each method, first the refrigeration and air conditioning system under analysis is discussed. Subsequently the other systems are discussed.

2.2.1. Thermoeconomic Evaluation and Optimization (TEO) Method

In 1985, Tsatsaronis et al. [3] proposed thermoeconomic optimization of thermal system. After calculating mass, energy and exergy balance for total system, levelized investment and operating cost of each component of coal fired power plant (economic analysis) is carried out. Then the cost of the exergy unit of each process flow stream is calculated. Marginal exergy unit cost for fuel and product of each plant component is calculated. Then the cost of the exergy losses in each plant component is calculated.

Refrigeration and Air conditioning system

In 1997, Cammarata et al. [85] used the same approach to optimize the air conditioning system. The thermodynamic model is stated according to recent formulations of exergy for moist air streams, while the economic model is based on cost balance equations and real cost data for mechanical equipment. The objective function to minimize includes decision variables such as fresh to total air rate, coefficient of performance for the chiller, inlet temperature of water for the cooling and the heating coils, and temperature difference of the same streams.

In 2004, Morosuk et al. [86] tried to combine TEO method with pinch technology and applied to low temperature refrigeration system (Cryogenic system) generating temperature between -100 °C to -150 °C.

In 2005, Leo et al. [87] applied TEO method to commercial aircraft environmental control system and the cost per unit of exergy of the conditioning stream entering the cabin has been obtained for a range of the aircraft engine bleed pressure values. A minimum cost has been found at a pressure close to the nominal bleed pressure.

Using this approach, in 2003, Misra et al. [88] carried out exergoeconomic analysis and optimized single effect LiBr/water VAR system. In 2005, they extended the analysis for double-effect LiBr/water VAR system [89] and in 2006, AAVAR machine [90].

Power generating system

In 1993, Tsatsaronis [3] suggested the improved method for exergy costing in which a average cost (monetary) value is assigned to each material and energy stream in the energy-conversion system (process) being considered. This value represents the total cost required to produce the stream. He presented a case study of an exergoeconomic analysis of power plant.

In 2002, Tsatsaronis et al. [92], identified avoidable and unavoidable exergy destruction and associated cost and applied to cogeneration system. In 2006, Cziesla et al. [93] continued this work and applied to an externally fired combined cycle power plant. For each plant component, avoidable and unavoidable exergy destructions and investment costs are calculated.

In 2007, Notario et al. [94] suggested that the thermal component of exergy ϕ^{TM} should be divided in to two parts. $\phi^{TM} = \phi^{tb} + \phi^{ta}$ as shown in Fig. 2.1. Where ϕ^{tb} represents the work produced (or required) in the isentropic expansion (or compression) process at ambient pressure, zero velocity and zero potential energy level. This part of the thermo mechanical exergy is known as the thermo baric component, and its energetic quality is comparable to that of work and ϕ^{ta} represents an exclusively thermal-type exergetic component at the ambient pressure p_a called thermo ambient component. To reversibly convert this into work is a highly complex task. The economic values of these two components are clearly different, and the thermo baric component has a higher value than the thermo ambient component in the case of thermal engines. The economic evaluation of both thermo mechanical exergy components is carried out for the case of combine cycle power plant.

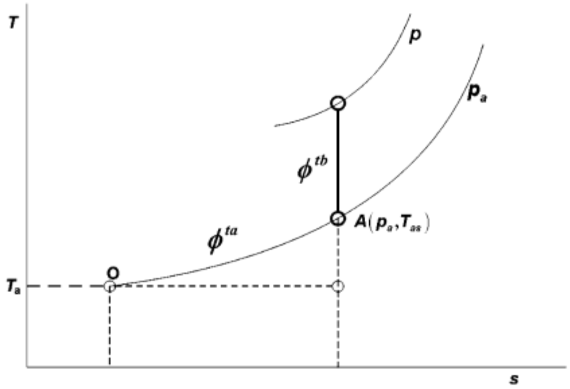


Fig. 2.1: Division of Thermo Mechanical Exergy into Thermo Baric and Thermo Ambient Components [93]

In 2009, Abusoglu et al [95, 96] used the SPECO method to find specific exergy cost to analyze diesel engine powered cogeneration plant. After calculating specific exergy cost, a detailed exergoeconomic analysis is carried out using TEO method. It is observed that the specific unit exergetic cost of the power produced by the plant is 10.3 $/GJ.

Power plant in combination with other system

In 2006, Tsatsaronis et al. [97] carried out exergoeconomic optimization of a novel zero-emission process generating hydrogen and electric power. In 2009, Modesto et al. [98] used TEO method for exergoeconomic optimization of the power generation system using blast furnace and coke oven gas in a steel mill.

In 2010, Cortes et al. [99] tried to combine exergoeconomic methodology with pinch technology and applied to a cogeneration power plant with pulp and paper mill. They proposed improvements to the plant, through the modification of the operating conditions and the implementation of new equipment. The location of the new equipment was determined using pinch technology and the exergoeconomic optimization was carried out through TEO method.

Gasifier

Subsequently Lazzaretto et al. [100] proposed this methodology systematically considering the case of gasifier, by taking a systematic record of all exergy additions to and removals from all the exergy streams of the system and the costs are calculated by applying basic principles from business administration. Thus, a direct link between the definitions of fuel and product for a component and the corresponding costing equations is established. In particular, the paper shows how to obtain detailed definitions of exergetic efficiencies using separate forms of exergy (thermal, mechanical and chemical) and how, according to these definitions, to conduct an evaluation of costs associated with all the exergy streams entering and leaving a system component. For this case, the cost equations are presented in a general matrix form.

Boiler

In 2010, Ozdemir et al. [101] applies TEO method to a fluidized-bed coal combustor (FBCC) of steam power plant. He considered ventilation fan, FBCC, HRSG, cyclone, economizer, aspiration fan, pump and chimney as sub systems and calculated the exergy destruction in each part and cost of steam generated.

2.2.2 Theory of Exergetic Cost

In 1993, Lozano et al. [102] proposed theory of exergetic cost. It is based on the concept of; a higher irreversibility in a sub system always implies higher consumption of resources of the plant if the products remain constant. It is essential to link the variation in the local irreversibility to the increase of resources consumed. The plant will be defined as a set of sub systems or units linked to each other and to the environment by another set of matter, heat, and work flows. The relationship between the flows and sub systems is set up through the incidence matrix. After developing the thermodynamic and cost model, the exergetic and thermoeconomic cost balance equations are formulated with the help of fuel product definition.

Refrigeration and Air conditioning system

In the year 1997, Accadia [103] analyzed the vapour compression refrigeration plant using exergetic cost theory. In 2002, Misra et al. [104] applied theory of exergetic cost to the LiBr/water VAR system.

Power generation system

In 2006, Modesto et al. [105] applied same method to the power plant, generating power from the waste heat of a steel mill plant. The system was assessed by means of two thermoeconomic methodologies, Theory of Exergetic Cost and Thermoeconomic Functional Analysis; exergetic and monetary costs of power production were calculated and compared to the respective values of the current system.

In 2007, Aguilar [106] analyzed steam turbine using exergetic cost theory. In the first part, the relationship between entropy and enthalpy modifications due to stage malfunctions are developed to evaluate their economic impact, based on the cost of irreversibility in a system. The second part of the work, they determined and evaluated the degree of entropy generation and power loss in a steam turbine stage (nozzle-bucket), under the detection of specific malfunctions such as roughness, seal and leak clearances, erosion, and sedimentation. Then the computation of exergy-cost and economic-cost of local products and fuels in steam turbines is carried out.

Engine

In 2006, Sala et al. [107] analyzed container housed engine using theory of exergetic cost and optimized the system to calculate the minimum cost of the electricity and useful heat energy produced by the engine. As an output, identified the exergy losses and areas of improvement.

2.2.3 Engineering Functional Analysis (EFA)

Refrigeration and Air conditioning system

In 1986, Wall [108], analyzed heat pump, used autonomous method, also known as EFA method. His decision variables are the efficiencies of the compressor, the condenser, the evaporator, and the electric motor. The system is completely defined apart from the decision variables, each set of which determines a state of the system. The exergy flows and exergy losses are also determined for each component. The objective is to minimize the cost for a given amount of produced heat. The cost includes both the operating (electricity) cost and the capital cost. The operating cost increases if the investments decrease and vice versa. The income from the product (heat) and a given required value of the profit sets an upper limit for the total cost of the system. In 2004, Al-Otaibi et al. [109], followed similar procedure to optimize the vapour compression refrigeration system.

Power generation system

In 2003, Rosen M A et al. [110] examine the relations between thermodynamic losses and capital costs for devices in several modern coal-fired power plants, and suggest possible generalizations in the relation between thermodynamic losses and capital costs. They compared the performance of power plants operating on various fuels.

2.2.4 Thermoeconomic Functional analysis

In 1987, Frangopoulos [111] suggested Thermoeconomic Functional analysis (TFA), is a method for optimal design or improvement of complex thermal systems. Each unit has a particular quantified function (purpose or product). The distribution of functions establishes inter-relations between units or between the system and the environment and leads to a functional diagram of the system. The optimization minimizes the total cost of owning and operating the system, subject to constraints revealed by the functional diagram and analysis.

Power generation system

In 1996, Frangopoulos et al. [112] applied thermoeconomic functional analysis to the cogeneration system of the refinery and optimized the system.

In 1990, a group of concerned specialists in the field (**C**. Frangopoulos, **G**. Tsatsaronis, **A.** Valero, and **M.** von Spakovsky) [113 to 117] decided to compare their methodologies by solving a predefined and simple problem of optimization: the CGAM problem, which was named after the first initials of the participating investigators. The objective of the CGAM problem is to show how the methodologies are applied, what concepts are used and what numbers are obtained in a simple and specific problem. In the final analysis, the aim of the CGAM problem is the unification of thermoeconomic methodologies. This comparison is not a competition among methodologies. Each methodology has specific fields of applications for which it provides proven and efficient solutions.

2.2.5 Structural Method

As cited by Kotas [118] the structural method was developed by Beyer and is proved as one of the best method for system optimization. This method is using the unitary costs of exergy loses. Structural coefficients are used in the study of the system structure, for the optimization of plant components and product pricing in multi-product plants.

Refrigeration and Air conditioning system

In 2004, Selbas [119] optimized vapour compression refrigeration cycle with subcooling and superheating. The final equation of irreversibility was applied to condenser, evaporator, subcooler and super heater. On the common plot, variation of irreversibility v/s area and cost v/s area was plotted. The point of intersection is the optimum area of the heat exchanger. In 2004, Accadia et al. [120] optimized condenser in a vapour compression heat pump using the same method.

In 2005, Misra et al [121] applied the structural theory to LiBr/water absorption chiller. Their analysis reveals that the capital cost of the optimum configuration is increased by about 33.3% from the base case; however, the additional cost is well compensated by reduced fuel cost. This is possible because of reduction of plant irreversibilities by about 47.2%. In 2006, Kizilkan et al. [122] used this technique for thermoeconomic optimization of a LiBr/water VAR system.

Power generation system

In 1999, Erlach et al. [123] tried to apply Last in First out (LIFO) approach to the structural method in application with combined cycle power plant. The LIFO approach is a cost accounting method used to calculate the cost of the fuel and product of the components, which are defined as the resources consumed to generate these fuels and products, respectively. The costs are calculated from a linear equation system consisting of the cost balances of the components, which can be obtained from the physical structure of the system and some auxiliary equations required for the components that have more than one outlet flow. The auxiliary equations express the fuel and product distribution given by the thermoeconomic model.

In 2002, Torres et al. [124, 125] improved the structural theory by computing the additional fuel consumption as the sum of both the irreversibilities and the malfunction costs of the gas turbine power plant components. It will be able to quantify the effect of a component malfunction in the other components of the plant. The key of the proposed method is the construction of the malfunction/dysfunction table which contains the information related with the plant inefficiencies and their effects on each component and on the whole plant.

In 2006, Zhang et al. [126] applied a cost analysis method based on structural method to a pulverized coal fired power plant. An exergy analysis is performed to calculate the exergy and negentropy of the flows. Then, a thermoeconomic model of the plant is defined based on the functionality of each component using the fuel–product definition. The distribution of the resources throughout the plant and the costs of all flows in the production structure can be calculated by solving a set of equations including the thermoeconomic model of the plant. Three thermoeconomic variables are defined for improving the exergy cost equations in the structural theory of thermoeconomics.

In 2006, Valero et al. [127] explained the structural theory in detail with the example of gas power plant. They explained the difference in average and marginal cost, Building of productive structure, thermoeconomic model and cost estimation in first part. In Part II [128] they developed the mathematical formulations of three applications of the thermoeconomic analysis methodology described in Part I: the operation diagnosis study, including new concepts that helps to separate different contributions to those inefficiencies; the local optimization process in case of special conditions for the whole plant, and the benefit maximization (a direct application of the exergy costs accounting analysis). The operation diagnosis, which is the most complex and sophisticated application, is presented with the help of an example: the co-generation plant, as it was described in Part-I.

In 2006, Zhang et al. [129] presented a progressive separation procedure of the induced effects for power plant system diagnosis based on structural theory and symbolic thermoeconomics. The malfunction/dysfunction analysis and the fuel impact analysis of the structural theory as well as an improved induced malfunction evaluation method, which is composed of a progressive separation procedure, are applied to a coal fired power plant. First, the dysfunctions induced by the malfunctions are separated by the malfunction/dysfunction analysis from the irreversibility increases in the components. The effects of the malfunctions on each component and the whole plant are also evaluated by using the fuel impact analysis. Then, the induced malfunctions generated by the inefficiencies in the other components are separated from the remaining irreversibility increases (i.e. malfunctions) by using the induced malfunction evaluation method proposed by them. This method enables evaluation of the induced malfunctions in thermoeconomic diagnosis applications. After separating the induced effects, the real anomalies can be quantified and localized.

Power generation system in combination with other system

In 2008, Deng et al. [130] used structural theory for the exergoeconomic analysis of gas-fired micro-trigeneration system, which uses a small-scale generator set driven by a gas engine and small-scale adsorption chillier, generating electricity, cooling and heating effect. A comparison between the method of conventional exergy analysis and exergy cost analysis is presented. The result reflects that the structural theory is a powerful and effective tool for performance evaluation of complex system, and also proves that the micro-trigeneration system is efficient in utilizing the low-grade waste heat.

Heat exchanger

In 2002, Accadia et al. [131] determined the optimal configuration of a heat exchanger with a two-phase refrigerant using exergoeconomics by structural method. In their paper, the irreversibility due to heat transfer across the stream-to-stream temperature-difference and to frictional pressure-drops is calculated as a function of two design variables: the inner-tube's diameter and the saturation temperature of the refrigerant, on which the heat-exchange area directly depends. Then, a cost function is introduced, defined as the sum of two contributions: the amortization cost of the condenser under study and the operating cost of the conventional electric-driven heat-pump in which this component will have to work. The latter contribution is directly related to the overall exergy destruction rate in the plant, whereas the amortization cost mainly depends on the heat-exchange area. So, design optimization of the device can be performed by minimizing this cost function with respect to the selected design variables. The Coefficient of Structural Bond is used in the optimization to relate the local irreversibility in the condenser to the overall exergy destruction rate in the heat-pump plant. For a commercial heat-exchanger, the design improvements needed to obtain a cost-optimal configuration are investigated. The results show that significant improvements can be obtained with respect to devices based on conventional values of the design parameters.

2.2.6 Evolutionary programming (EP)

Evolutionary programming, originally conceived by Fogel [132] in 1960, is a stochastic optimization strategy similar to genetic algorithms (GA), from which it differs in its emphasizing the behavioral linkage between parents and their offspring, rather than seeking to emulate specific genetic operators as observed in nature. EP is a powerful method of optimization when other techniques such as gradient descent or direct analytical discovery are not possible. For EP, a fitness landscape can be characterized in terms of variables, and that there is an optimum solution (or multiple such optima) in terms of those variables. In 2007, Sahoo [132] applied this methodology to cogeneration plant and in the same year Koch et al. [133] used the same method for Optimization of combined cycle power plants.

In Europe, this methodology is known as Genetic Algorithm. In 1998, Cammarata et al. [134] formulated the objective function, the sum of the capital, and the operational and maintenance costs, of a district heating network using exergoeconomic concepts and minimized the objective function using genetic algorithm (GA) and found the suitability of this concept. In 2002, Toffolo et al. [135] used the multi-objective evolutionary algorithm for simultaneous exergetic and economic optimization of the CGAM problem. In 2008, Caputo et al. [136] carried out economic optimization of heat exchanger using genetic algorithm and showed that significant cost reductions are feasible with respect to traditionally designed exchangers.

2.2.7 EEA method

In 2001, Sciubba [137] realized that exergy analysis, though completely satisfactory from a thermodynamic point of view, has always been regarded as unable to determine real design optima, and therefore its use has been associated with customary monetary cost-analysis. He suggested the concept of Extended Exergy Accounting (EEA). He defined the term invested exergy. It is equal to the sum of the 'non-energetic' externalities (Labor and Capital) used in the construction and operation of the plant in which the product is generated. For any product, it is then conceivable to define an 'extended exergy' as the sum of the physical exergy and the proper portion of the invested exergy that can be assigned to the stream under consideration. Using this approach, he analyzed the gas turbine based cogeneration system.

In 2007, Verda et al. [138] proposed procedure, based on this method with the aim of locating and quantifying malfunctions in the system. The value of malfunction can be correlated with economic value.

2.2.8 Exergetic Production Cost method (EPC)

In 2003, Silveira et al. [139,140] proposed method, and applied to power plant. The developed technique has as objective the minimum (optimal) total operating costs of the cogeneration plant (EPC), assuming a fixed rate of electricity production and process steam. The operating cost of each component of the plant is calculated. Taking the example of pump, the cost of the outflow stream is the sum of inflow stream cost, pump work cost and investment and maintenance cost. Similarly the operation cost for other components are found. The objective function to be minimized is the EPC. The operating parameters selected to be optimized are temperature and pressure leaving the boiler for systems with steam turbine, and pressure ratio, flow rate and temperature of exhaust gases for systems with gas turbine. Those parameters are selected because of their influence over the power generated and the purchase costs of the components. The EPC equation is formulated as a function of these operating parameters. The function minimization is done with the help of computer program. Using similar method, in 2003, Camargo et al. [141] carried out thermoeconomic analysis of an evaporative desiccant air conditioning system.

2.2.9 Graphical Method

In 2002, Can et al. [142] suggested that the previous methodologies are tedious and proposed graphical solution considering the case of condenser type of heat exchanger. In this method, first a diagram for the investment and exergy loss expenses is drawn and then the Optimum Operation Point is determined by intersection of the investment line and operation line.

2.2.10 Input-Output Method

In the year 1994, Alvarado et al. [143], presented and called input-output exergoeconomic optimization, applied to the cogeneration plant (the CGAM problem). The selection of sub systems or components to be optimized is dictated by two parameters: the exergetic efficiency and the elasticity, the latter measuring the variation of global system efficiency with efficiency change of the sub system, both evaluated for the given operational conditions. The proposed method gives results that agree quite closely with those of the authors of the CGAM problem.

In 2009, Abusoglu et al. [144] discussed various thermoeconomic methodologies considering the case of CGAM problem and exergoeconomic analysis and optimization of combined heat and power production are presented. In the same year, Lazzaretto [145] presented a comparison between thermoeconomic and energy analyses (TA) algebra using the case of cogeneration system. In energy analysis, he included all human and natural inputs, energy, material, human labour and information and the combination of all these is called as emergy and application of it is called emergy analysis (EMA). He concluded that the TA is based on the idea that the monetary cost and exergy cost are conserved at system (or component) level. Unlike TA, the EMA attention is not on system (or components) but on system (or component) products. The idea is to take a record of all the energy that was previously required to generate each of the system (or component) products, separately.

In 2010, Kim [146] presented a new thermoeconomic methodology for energy systems, the wonergy method. In this methodology, various energies, including enthalpy and exergy, can be integrated with ‘‘wonergy’’, a portmanteau of ‘‘worth’’ and ‘‘energy’’ and is defined as an energy that can equally evaluate the worth of each product, and worth is not an absolute number but a relative concept. He applied this method to CGAM problem and compared with various conventional methodologies.

In 2010, Miguel et al. [147] carried out Cost optimization of the design of combined heat, cooling and power (CHCP) systems under legal constraints. An optimization model is developed, using mixed integer linear programming (MILP), to determine the preliminary design of CHCP systems with thermal storage. The objective function to be minimized is the total annual cost. Taking into account the legal constraints imposed on cogeneration systems in Spain, the optimization model is applied to design a system

The above review of literature shows that the exergoeconomic optimization is having the major field of application in power plant as they are having huge capital investment. Minor variation in the working parameter also brings drastic change capital investment and operating cost. For other thermal systems, especially refrigeration systems, limited applications are observed. Due to ozone layer depletion and global worming problem, the VAR systems are getting more and more popularity. But VAR systems are heat intensive and facing a problem of huge capital investment. Just like power plant, the optimization of VAR system by variation in working parameters can bring major reduction in investment and operating cost. The major publications are in the field of LiBr/water VAR system but very few for AAVAR system. Moreover all publications are found with independent boiler as heat source. The literature review of publications related with gas and steam turbine power plant reveals that the optimization of gas turbine power plant with HRSG can bring appreciable reduction in the cost of electricity as well the cost of steam generation [114]. Similarly, the exergoeconomic optimization of steam turbine power plant with regeneration [139,140] can bring reduction in the cost of electricity and shows that the cost of steam tapped from steam turbine will be less than the cost of steam generated in the independent boiler.

2.3 Problem Formulation

From the review of various investigations on exergoeconomic analyses on thermal systems, it can be seen that, many optimization methods are in use. The summary of all the methods is given in Table 2.4. All these methods are independent and used as per the application. It is further observed that almost all of them are stand-alone methods developed by the respective investigators and the complete details are not readily available from the published literature for the user to adapt them with ease. Thermoeconomic Evaluation and Optimization Method suggested by Tsatsaronis [1], however, is found to be the most user friendly method.

As the method is available in open literature, it is decided to analyse AAVAR System using this method. From the reported investigations of Misra et al. [90], it is observed that the components like throttling valve and expansion valve have to be excluded from the analysis when TEO method is employed. However, as the exergy destruction in these components cannot be avoided and must be included in the overall system analysis, it is necessary to find suitable scheme for the purpose. Due to the limitation of the exergy destruction method that Misra et al. have adopted in their analysis, the actual performance of condenser and evaporator cannot be judged. Therefore, the present proposed research study will include Entropy Generation

Minimization Method in Thermoeconomic Evaluation and Optimization (TEO) method so as to take care of the limitations. It is expected that the proposed study will be more general in nature.

TEO method suitably modified using Entropy Generation Minimization (EGM) method is proposed to be employed to optimize the existing AAVAR system with independent boiler as a heat source in the fertilizer industry for minimum cooling cost to be achieved. There are two different heat sources, viz., steam generated at HRSG of gas power plant and tapped steam from steam power plant, available as alternative energy options in the fertilizer industry which can be incorporated to AAVAR system. It is proposed that these options be examined using the optimization method developed. Then, the cooling costs of AAVAR system with different heat source options be compared and optimum cost be decided.

2.4 Objectives of the Research

The following are the objectives for the present research work:

- To develop the user friendly method for exergy analysis of thermal system systems by combining the EDM method and EGM method of exergy analysis.
- To carryout exergy analysis of AAVAR system, Gas Turbine Power Plant and Steam Turbine Power Plant and to identify losses in the various components of these systems.
- To carryout exergoeconomic optimization of AAVAR system with independent boiler as heat source and to decide optimum cost for cooling generated at evaporator.
- To identify alternative options as heat source for AAVAR system other than independent boiler.
- To carryout exergoeconomic optimization of AAVAR system and GT-HRSG jointly using steam generated at HRSG as heat source.
- To carryout exergoeconomic optimization of AAVAR system and steam turbine power plant jointly using tapped steam from steam turbine as heat source.
- To identify most economical heat source for AAVAR system from the options available in the fertilizer industry.

Table 2.4: Summary of Investigations on Exergoeconomic Optimization Methods

System	Type of system	Investigator	Year	Exergoeconomic method used	remark
Refrigeration and Air conditioning systems					
Absorption Refrigeration System	LiBr-Water	Misra R D et al. (India)	2002	Exergetic Cost Theory	
	LiBr-Water	Misra R D et al. (India)	2003	TEO	
	LiBr-Water Double effect	Misra R D et al. (India)	2005	TEO	
	LiBr-Water	Misra R D et al. (India)	2005	Structural Method	
	Ammonia Water	Misra R D et al. (India)	2006	TEO	
	LiBr-Water	Kizilkan O et al. (Turkey)	2007	Structural Method	
Vapour Compression Refrigeration system		Accadia M D et al. (Italy)	1998	Exergetic Cost Theory	
	R-134a as refrigerant	Otaibi D A et al. (Saudi Arabia)	2004	Engineering Functional Analysis	
		Selbas R et al. (Turkey)	2004	Structural Method	Subcooling and Superheating are considered
Air Conditioning System	Heat Pump	Wall G (Sweden)	1985	Engineering Functional Analysis	
	Re circulation type	Cammarata G et al.	1997	TEO	Average cost

		(Italy)			approach
	Central A/C Small size	Zhang G Q et al. (China)	2003	Engineering Functional Analysis	
	Desiccant A/C	Camargo J R et al. (Brazil)	2003	Exergetic Production cost method	
Power generating systems					
Power Plant	Steam power Plant	Tsatsaronis G Winhold M (Germany)	1984	TEO	LIFO approach
	Steam power plant	Tsatsaronis G (Germany)	1993	TEO	Average cost approach
	Steam power plant	Lozano M A et al. (Spain)	1993	Exergetic Cost Theory	
	Combine Cycle Power Plant	Notario P P Leo T J (Spain)	2005	Converted economy in to equivalent exergy	Assigned different cost to thermal and mechanical exergy
	Combine Cycle Power Plant	Cziesla F Tsatsaronis G (Germany)	2006	TEO	Average cost approach
Power plant in combination with other system					
Cogeneration Power plant	Heat and power generation	Silveira J L at el. (Brazil)	2003	Exergetic Production cost method	
	Electricity + Hydrogen	Tsatsaronis G Kapanke K (Germany)	2006	TEO	Average cost approach

	Power/water cogeneration plant	Hamed O A (Saudi Arabia)	2006	Structural Method	
	Cogeneration plant	Modesto M et al. (Brazil)	2006	Exergetic Cost Theory Thermoeconomic functional Analysis	Gas from blast furnace and coke oven gas Compared ECT and TFA
	Cogeneration plant	Valero A et al. (Spain)	2006	Structural Method	Compared average and Marginal cost
	Cogeneration Plant	Sahoo P K (India)	2008	Evolutionary Programming	
Miscellaneous system					
Steam Turbine		Aguilar A Z et al. (Mexico)	2007	Exergetic Cost Theory	Exergy cost of turbine steam emission is found
Engine	Container housed type	Sala J M et al. (Spain)	2006	Exergetic Cost Theory	
Heat Exchanger	Counter flow	Soylemez M S (Turkey)	2000	Saving function is developed in terms of exergy and area And optimized the system	Based on thermodynamics fundamentals
	Parallel flow				
	Single fluid (Ratio of heat capacity rate = 0)				
	Tube in Tube type (Two Phase Refrigerant)	Accadia M D et al. (Italy)	2002	Structural Method	

Table 2.5 Comparison of Various Exergoeconomic Optimization Methods

Sr. No.	Method	Overview of the Method	Remark
1	Thermoeconomic Evaluation and Optimization Method	• Based on local optimization of sub systems • Follow iterative procedure • Consider actual component level penalty function.	• Heat dissipative devices like throttle valve can not be analyzed. • No computer programming is required • User friendly
2	Exergetic Cost Theory	• Cost are calculated for the system as a whole	• Using the negentropy concept, heat dissipative devices can be analyzed.
3	Thermoeconomic Functional Analysis	• Exergy and negentropy flows are determined • Functional diagram is prepared • Standard cost functions used for all components • Cost flows are solved using Lagarangian multiplier • Objective function, to be minimized, is formulated	• Cost functions for the components like throttle valve are not defined • Preparation of functional diagram involves individual judgment • Solution of Lagarangian multiplier needs high level of computer skill • Decision of constrains & decision variable depends on individual
4	Autonomous Method	• Objective function, to be minimized, is defined • Cost model of each equipment is used • system product, input, same at branch and junctions are determined • secondary product handled by branch • Junction represents mixing of two fluid • Functional diagram is prepared • Using the internal economy, Lagarangian associated with the objective function is	• similar to TFA • judgment of branch and junction involve individual decision • functional diagram is complicated • solution of Lagrange is difficult

		formulated • Other algorithms are also possible for solution	
5	Structural Method	• For the plant component, change in irreversibility with respect to decision variable is modeled if output is constant • Due to change in irreversibility, change in cost is decided • Differentiating the cost model with decision variable and equating to zero, optimum condition is obtained.	• Selection of decision parameter is critical • Output is fixed. In case of variable output method is not applicable
6	Evolutionary Programming Method	• Global search technique • Used for single objective function	• Simple method • Sound knowledge of programming and Gaussian random variable is required
7	Extended Exergy Accounting Method	• Extension of the previous method. • Apart from physical and chemical exergy, recycling exergy necessary for ideal zero impact disposal of the equipment should be added. • Labor and capital cost converted in the exergy term called added exergy and included in the total exergy	• Cost optimization is similar to Structural method
8	Exergetic Production Cost Method	• Evaluation of exergy input and output for each unit • Exergetic manufacturing cost function formulated • Thermoeconomic functional diagram prepared • Selection of exergetic incremental linked to the input and output of each unit.	• Exergetic manufacturing cost function given by manufacturer, not available for all type of equipment. • Thermoeconomic functional diagram involves individual judgments.

The above review of literature shows that exergy analysis and exergoeconomics are very useful analytical tools in energy engineering for improving the design, operation and maintenance of energy systems. Both exergy analysis and thermoeconomics may be considered to be general and objective methodologies for analyzing and optimizing energy systems. An exergy analysis identifies the location, cause and magnitude of the real thermodynamic losses (exergy destruction and exergy loss). A thermoeconomic evaluation identifies the location and cause of the cost sources, calculates their magnitude and compares their effects on the costs of the products. All this information, complemented by the engineer's intuition and judgment, assists in the improvement of the efficiency and reduction of the product costs in energy systems by identifying the required changes in structure and parameter values much faster than traditional approaches. Decisions about the design, operation and repair or replacement of equipment are facilitated.

The advantage of thermoeconomics is that it replaces an expensive and subjective search for cost reduction with an objective, well informed, systematic and, therefore, shorter search in which all of the cost sources are properly identified and evaluated. The savings in both engineering and computer time are significant. In particular, application of thermoeconomic analysis to new energy system concepts and complex plants (especially those with important chemical reactions) results in significant savings in design costs and costs of plant products. Compared with engineers using the traditional energy and economic analyses, those applying thermoeconomics develop, in general, a better understanding of the performance of energy systems and the interactions between performance and economics, as well as more confidence in their ability to improve energy systems

Chapter 3

Aqua Ammonia VAR system

This chapter is concerned with the description of the brine chilling unit consisting of an 800 TR AAVAR system which is a part of Gujarat Narmada Valley Fertilizer Company (GNFC) Bharuch, Gujarat, a large fertilizer plant located at Bharuch, Gujarat State, India. The company manufactures many products related with chemical fertilizer. For the manufacturing of chemical fertilizer, large quantities of chilled brine at specified temperature is needed as one of the raw materials. As a number of sources of heat energy are available at GNFC plant, it is economical to use vapour absorption refrigeration system instead of a conventional vapour compression refrigeration system for the purpose of chilling as the vapour absorption system needs heat energy for its operation instead of electricity for vapour compression refrigeration system. Since there are a number of sources of heat energy available as under utilized, it is desirable to evaluate the techno-economic viability of using the heat energy source. Hence, the existing AAVAR system using an independent and dedicated boiler unit generating steam as heat source is considered first. Section 3.1 describes the various components of the system. Section 3.2 describes the other options of the heat energy sources considered during the techno-economic evaluation of the plant. The online steady state data during a normal operation of the brine chilling unit using AAVAR system with independent boiler as the source of steam (fuel), steam generated at HRSG of the gas turbine power plant and tapped steam from steam power plant are given in Section 3.3.

3.1 System Description

The basic components of AAVAR system consist of brine chilling unit and condensing unit. The brine chilling unit consists of pre-cooler-1, pre-cooler-2 and evaporator whereas the condensing unit includes solution heat exchangers, ammonia condenser, rectifier-generator assembly, aqua ammonia solution pump, expansion valve and pressure reducing valve and ammonia absorber. The working fluid (refrigerant) in the AAVAR system is ammonia and the absorbent is water having strong chemical affinity with ammonia. Fig. 3.1 gives the plant layout of the brine chilling unit of the fertilizer plant. The component-wise description of the chilling unit and condensing unit of AAVAR system as referred in Fig. 3.1 is given below:

3.1.1 Chilling Unit

The chilling unit consists of pre-cooler-1 (PC1), pre-cooler-2 (PC2), and evaporator. The incoming brine at a pressure and temperature of 1.01 bar and 24.7 °C, respectively is cooled at constant pressure in two stages while passing through PC1 and PC2.

In the fertilizer industry, the liquid ammonia is manufactured at two pressure levels, 2.3 bar saturated and 4 bar saturated in different plants. The liquid ammonia at 4 bar saturated is used as coolant in PC1. During evaporation, ammonia absorbs latent heat of evaporation (process 31 to 32) and chills the brine from 24.7°C to 15.9°C (process 29 to 30). Similarly, the liquid ammonia at 2.3 bar saturated evaporates in PC2 (process 33 to 34) further chilling the brine from 15.9°C to 5.7°C (Process 30 to 21). The chilled brine from PC2 enters the evaporator of AAVAR system where the remaining chilling is carried out from 5.7°C to -1.7°C. The evaporated ammonia from both the pre-coolers is utilized in the various chemical processes of the fertilizer plant. Thus, the AAVAR system is designed to chill brine from 5.7 °C to - 1.7°C.

3.1.2 Condensing Unit

The condensing unit consists of refrigerant heat exchangers (RHX 06 and RHX 05), ammonia absorber (E04), solution pump, solution heat exchanger (SHX E02), throttle valve (V111), generator, rectifier-reflux assembly and ammonia condenser (E03).

The ammonia vapour is liquefied (process 5-6) rejecting its latent heat in condenser E03 using cooling water (process 23-24) supplied by cooling tower No. 34/02. The high pressure liquid ammonia from condenser is collected in receiving tank D01. High pressure liquid ammonia collected in the receiving tank is partially supplied to the rectifier as a reflux and remaining liquid ammonia flows through RHX 06 and RHX 05 before expanded through an expansion valve V208 to evaporator pressure at E07.

About ninety per cent of the liquid ammonia absorbs heat energy from the brine during its flow in the evaporator E07 and flows out as vapour through RHX 06 (process 10-11), while the rest is un-evaporated and gets collected at the bottom of the evaporator and flows through RHX 05 (process 12-13). Both the liquid and vapour ammonia from RHX 06 and RHX 05, respectively are used to subcool (processes 6-7 and 7-8) the incoming high pressure liquid ammonia from the condenser E03 before expanding in V208 (process 8-9). The introduction of RHX E06 and RHX E05 helps to increase the refrigerating effect at the expense of addition of heat energy to the ammonia vapour entering the absorber E04 from the evaporator E07.

Vaporized ammonia from E07 after absorbing heat energy while passing through RHX 06 and RHX 05 is absorbed by the weak solution of ammonia in the absorber (E04) Since the absorption process is exothermic, heat should be removed from the absorber to maintain a specified low temperature in the absorber so that high chemical affinity between the ammonia vapour and weak absorption solution can be maintained so as to form a strong solution of ammonia. For the cooling purpose, water from the cooling tower 34/02 is used as a coolant.

The strong solution of aqua ammonia is pumped through solution heat exchanger (SHX E02) to generator. SHX E02 is used to recover heat from the weak aqua ammonia solution returning from the generator. The heat energy of the weak solution is used (process 2-16) to preheat the strong solution from the absorber (process 15-1). The pressure of weak aqua ammonia solution from solution heat exchanger (E02) is reduced to that of the absorber (E04) through a throttle valve (V111) (process 16 to 17). Rectifier-generator assembly is used to separate ammonia vapour at high pressure so that the heat energy absorbed from evaporator can be rejected at the condenser to be used again in the evaporator to absorb heat energy from brine.

The heat is added to the strong aqua ammonia solution in the generator (E01) during condensation of steam generated in the independent boiler No. 38/02. The condensate coming out of the generator is supplied back to the boiler (process 19-20). As a result, the strong aqua ammonia solution boils and ammonia vapour is relieved from the mixture and leaves the weak aqua ammonia solution in the generator. The weak aqua ammonia solution returns to the absorber through the solution heat exchanger.

Ammonia vapour leaving generator (E01) enters the rectifier (C01) which is installed between generator and condenser at station 3. Since the difference in the boiling point between ammonia and water (absorbent) is not very large, some amount of water is likely to evaporate along with ammonia in the generator. When this water vapour passes through the expansion device, freezes in to ice and chock the line. In the rectifier, some quantity of liquefied refrigerant in the condenser (E03) utilizes as a reflux at station 5 to condense the water vapour part and reduces the carryover of the absorbent in to the condenser. It acts as a desuperheater and condensed water vapour returns to the generator and improve the ammonia concentration in the flow to the condenser. It also helps in maintaining the required temperature of the refrigerant entering the condenser.

It should be noted that there are a large number of utility components and control systems in the AAVAR system for the brine chilling as can be seen in Fig. 3.1. However, they are not heat energy consuming or energy producing systems and hence not considered in the present analysis. Therefore, the schematic of the brine chilling unit using AAVAR system is given in Fig. 3.2 which illustrates the essential components for the purpose of techno-economic analysis.

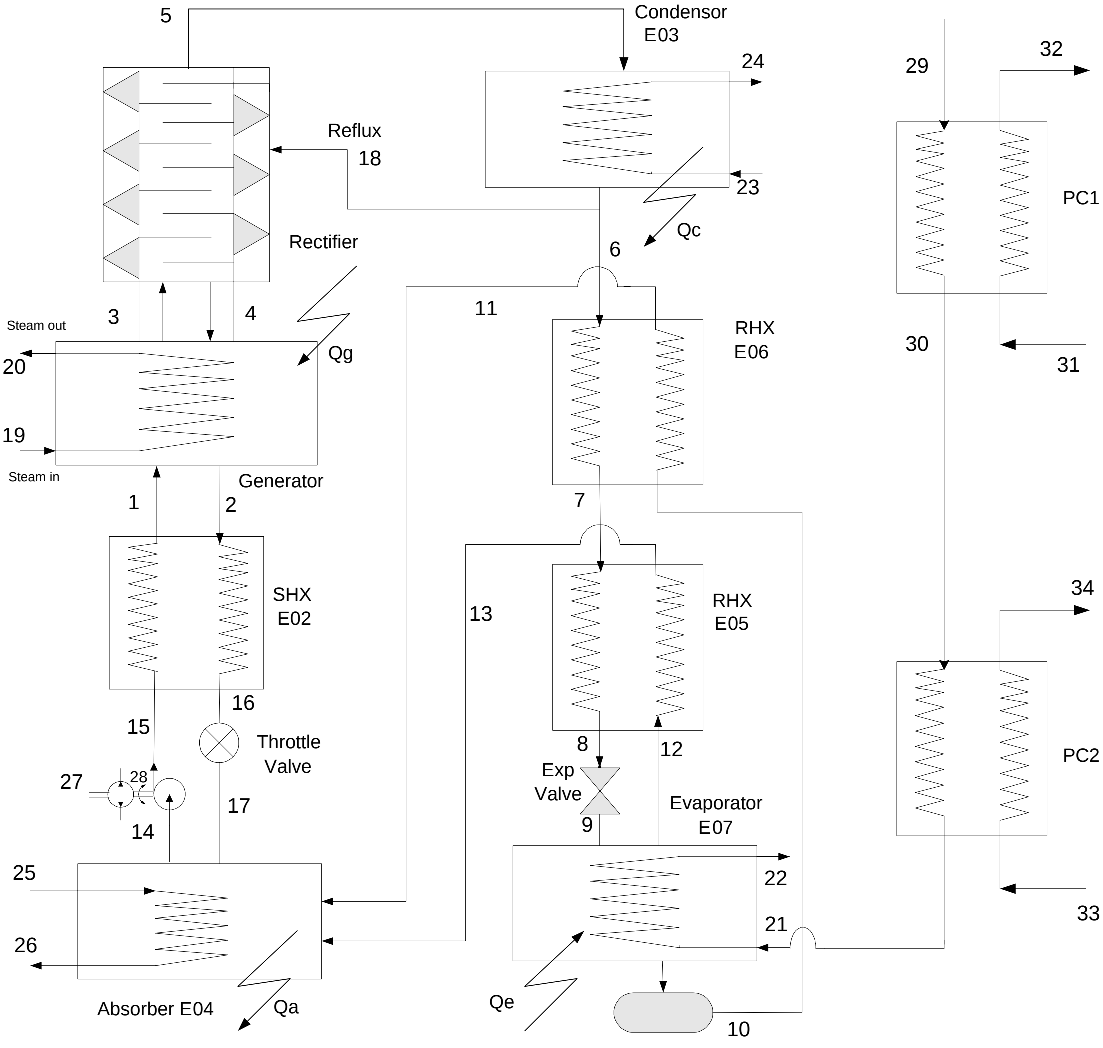


Fig. 3.2 Simplified Brine Chilling Unit with AAVAR System

3.2 Other Options of Heat Energy Sources

It should be recalled that the existing source of heat energy for the operation of 800 TR AAVAR system is steam from a dedicated independent boiler. In order to analyze the effect of the source of heat energy on the cooling cost to be estimated through the application of a proposed unified exergoeconomic optimization scheme, two other existing different schemes for the source of heat energy are also identified.

3.2.1 Steam from GT-HRSG Plant as Heat Source

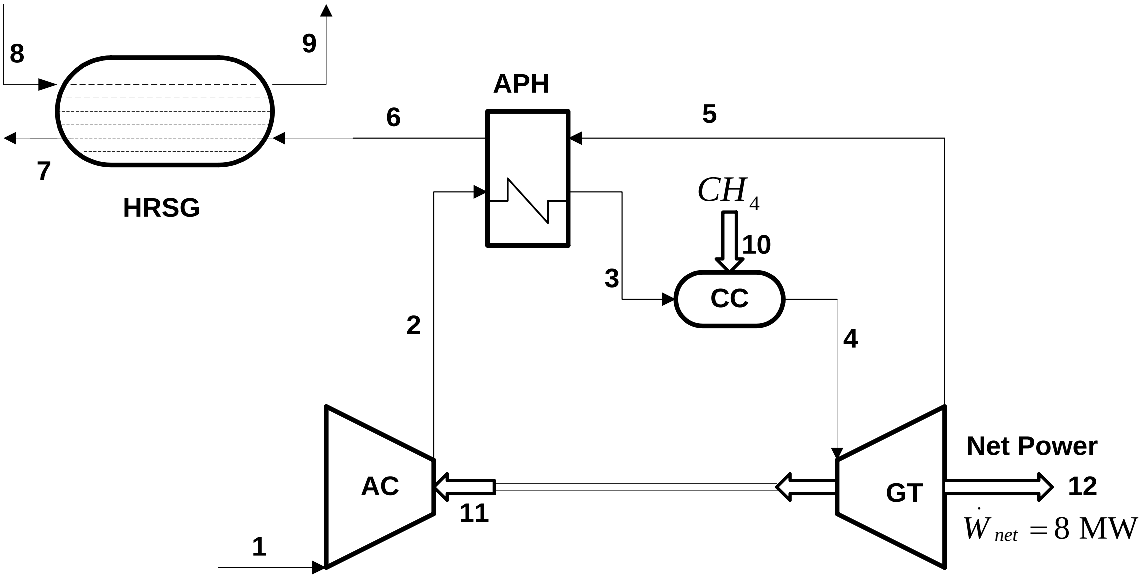


Fig. 3.3 Gas Turbine Power Plant

AC-Air compressor, **GT**- Gas turbine, **APH**- Air Preheater, **CC**- Combustion Chamber, **HRSG** - Heat Recovery Steam generator.

1- Atmospheric air	**6** – Exhaust gas from APH
2 – Compressed air	**7** – Waste gas
3 – Preheated air	**8** – feed water
4- Combustion product	**9** – Steam generated
5 – Exhaust gas from turbine	**10** – Fuel supply

The first option for replacing the existing heat energy source of steam from the independent boiler for the AAVAR system for brine chilling unit is the steam from the existing 8 MW combined gas turbine power plant with heat recovery steam generator (GT-HRSG). GT-HRSG utilizes natural gas as fuel for its operation. “The waste heat energy” from the exhaust gases from gas turbine power plant is utilized in the generation of steam at a pressure of 33 bar at the existing HRSG (station 9). If the steam generation in HRSG is carried out at 15 bar saturated instead of 33 bar saturated, then this steam will be useful as input heat energy (known as fuel) in AAVAR system. The 8 MW gas turbine power plant with HRSG is shown in Fig. 3.3. The figure is self explanatory.

3.2.2 Tapped Steam from Steam Power Plant as Heat Source

The fertilizer industry also operates a condensing type 50 MW regenerative type steam power plant for its captive power requirement. The steam turbine is having three steam tappings for regeneration purpose at 17 bar, 7 bar and 4 bar. The exhaust steam from the turbine is condensed in the condenser and the condensate is sent to the open feed water heater where the steam from the tap at 4 bar is utilized for heating purpose by direct contact. The resulting feed water is supplied to the second feed water heater which is a closed one in which the steam from the tap at 7 bar is used for heating purpose. After the heating is completed in the closed feed water heater, the feed water is further heated using another closed feed water heater in which the tapping steam at 17 bar is used. After the heating is completed in the heater, the feed water is sent back to the boiler.

Thus, the second option of the heat energy source for the AAVAR system can be the tapping steam from a steam turbine of the existing 50 MW steam power plant. Therefore, it is proposed to increase the steam flow rate from the tapping at 17 bar by 3.2 kg/s and throttle it to 15 bar before sent to the generator of the vapour absorption refrigeration system. The combined system of AAVAR system and the proposed heat energy source from steam power plant is to be exergoeconomically analyzed for minimum cost of power generation and tapping steam for the brine chilling plant. Fig. 3.4 gives the schematic of the Steam power plant considered for the second option of the heat energy source.

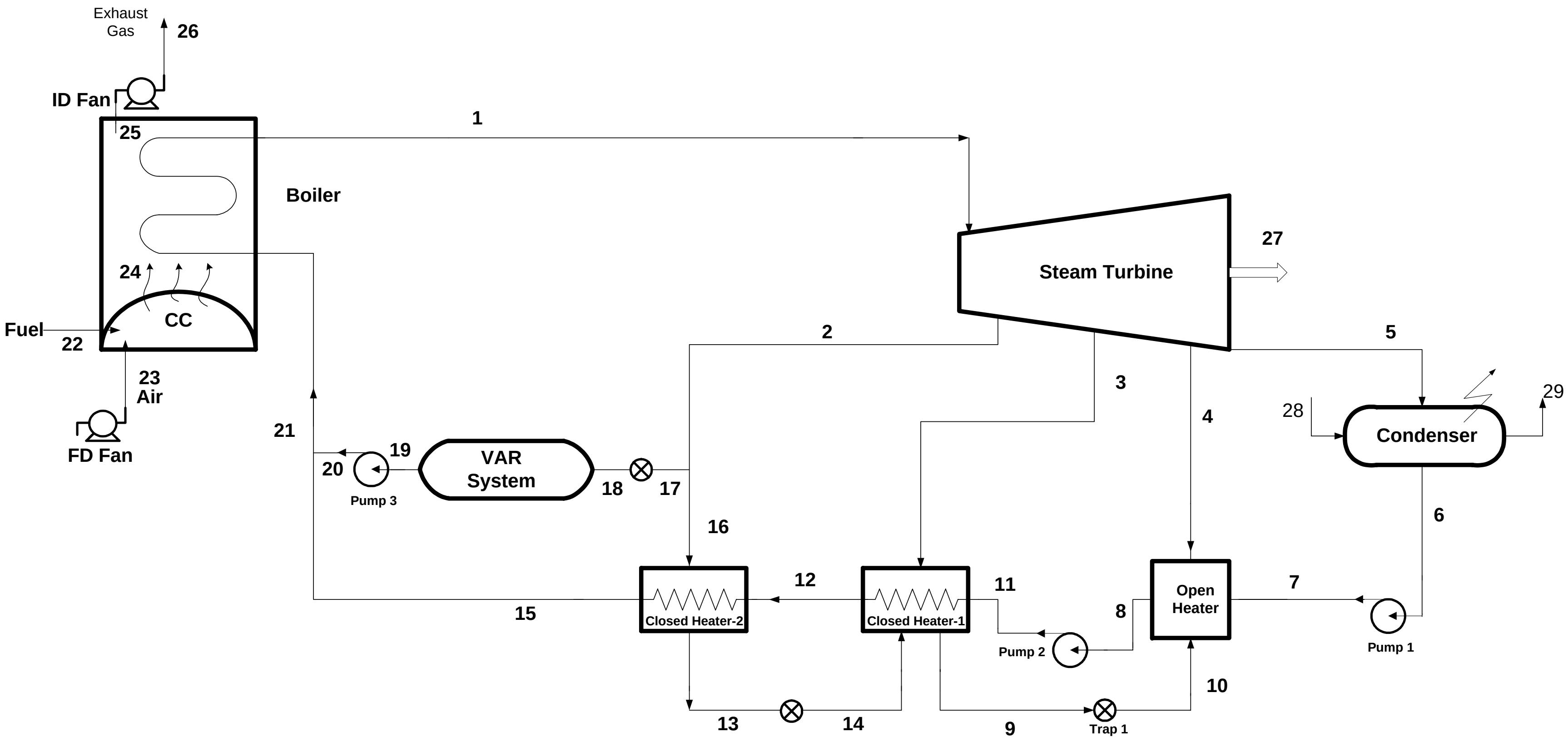


Fig. 3.4 Steam Power Plant

CC-Combustion Chamber, **ID-** Induced Draught, **FD-** Forced Draught, **VAR-** Vapour Absorption refrigeration System.

3.3 Steady State Online Data

The steady state online data during the normal operation of all the three systems are collected from control panel as well as some design data are collected. As the panel data are not complete for the purpose of exergoeconomic analysis, the necessary missing data are generated through system simulation, the details of which are given in Chapter 5.

3.3.1. Online Data for Brine Chilling Unit

Table 3.1. Steady State Online Data for Brine Chilling Unit

Stations	Mass Flow rate kg/sec	Pressure bar	Temp. °C	NH_3 Concentration % wt
1	--	18.90	--	0.270
2	--	18.90	140	0.150
3	--	18.90	140	0.990*
4	--	18.90	100*	
5	--	18.90	60	0.998
6	--	18.00	40	0.998
7	--	18.00	--	0.998
8	--	18.00	--	0.998
9	--	1.90	--	0.998
10	--	1.90	-20	0.970
11	--	1.90	--	0.970
12	--	1.90	-20	1.000
13	--	1.90	--	1.000
14	--	1.77	40	0.270
15	--	18.90	--	0.270
16	--	18.90	--	0.150
17	--	1.77	--	0.150
18	--	18.00	40	0.998
19	--	15.00	198.30	--
20	--	15.00	198.3	--
21	125.00	1.01	5.4	--
22	125.00	1.01	-1.7	--
23	88.06	1.01	33	--
24	88.06	1.01	--	--
25	125	1.01	33	--
26	125	1.01	--	--
27	--	--	--	--
28	--	--	--	--
29	125	1.01	24.70	--
30	125	1.01	--	--
31	2.56	4.00	-1.89	--
32	2.56	4.00	6.40	--
33	2.85	2.3	-15.62	--
34	2.85	2.3	12.50	--

For the chilling unit using vapour absorption refrigeration system the online data are given in Table 3.1. The additional data known are given in Table 3.2.

*Assumed Data

Table 3.2. Additional Data for Brine Chilling Unit

• Steam Condition inlet	15 bar saturated
• Steam condition exit	15 bar,0.27 % dry
• System Capacity	Q_e = 800 TR
• Liquid ammonia passed from PC-1	9.2 tons/hr at 4 bar saturated
• Liquid ammonia passed from PC-2	10.3 tons/hr at 2.3 bar saturated

3.3.2. Online Data for GT-HRSG

The available data for GT-HRSG are given in Table 3.3 and additional data are given in Table 3.4.

Table 3.3. Steady State Online Data for GT-HRSG

Stations	Flow rate m^3/hr	Pressure bar	Temp. °C
1	--	1.013	25
2	--	--	--
3	--	--	--
4	--	--	1247
5	--	1.099	--
6	--	--	--
7	--	1.013	--
8	--	15	25
9	--	15	--
10	--	25	12
11	--	--	--
12	--	--	--

Table 3.4. Additional Data for GT-HRSG

- *Air Compressor*
  - Pressure Ratio : 10 : 1
- *Air Preheater*

  - Pressure drop: 3% on gas side and 5% on the air side
  - Effectiveness: 0.75 (Considered for base case)
- *Heat Recovery Steam Generator*
  - Pressure drop: 5 % on gas side
- *Combustion Chamber*
  - Pressure drop: 5 %
- *Gas turbine*
  - Power Out-Put Rating: 8 MW
  - Speed: 14045 RPM

3.3.3. Online Data for Steam Power Plant

Table 3.5. Steady State Online Data for Steam Power Plant

Stations	Flow rate kg/sec	Pressure bar	Temp. ºC
1	--	96	500
2	--	17	--
3	--	7	--
4	--	4	--
5	--	0.1	--
6	--	--	--
7	--	--	--
8	--	2	--
9	--	6	--
10	--	4	--
11	--	135	--
12	--	134	154
13	--	15	--
14	--	7	--
15	--	133	190
16	--	--	--
17	3.2	--	--
18	--	15	--
19	--	15	--
20	--	133	--
21	--	--	--
22	--	--	--
23	--	--	--
24	--	--	1227
25	--	--	160
26	--	--	--
27	--	--	--
28	2555	1.01	33
29	2555	1.01	--

The online data available for steam power plant in the form of independent variables whose values are specified are kept fixed are given in Table 3.5 and additional data are given in Table 3.6.

Table 3.6. Additional Data for Steam Power Plant

- *Boiler*
    - FD fan draught: 472 mmWC
    - ID fan draught: 230 mmWC
    - Gas side pressure drop in the boiler 170 mm WC approximately.
- *Steam turbine*
    - Rated output: 50 MW
- *Surface condenser*
    - Cooling water flow: 92 m^3/hr

Chapter 4

Exergoeconomic Optimization Methodology

This chapter is devoted to the description of the various steps involved in the development of a unified exergoeconomic optimization method to be adopted for the combination of heat energy source and the AAVAR system for minimum cooling cost. The methodology presented in this chapter is proposed to be applied to the existing large industrial AAVAR system dedicated for the brine chilling and compare the same with that applied to the other two options of heat energy source available with the fertilizer industry (as described in Chapter 3) to identify the best option with minimum cooling cost. In this context, a unified approach of combining the Thermoeconomic Evaluation and Optimization (TEO) method by Tsatsaronis [1] along with the Entropy Generation Minimization (EGM) is proposed. The methodology for exergoeconomic optimization of thermal system using the unified approach can be divided in three steps viz. exergy analysis, exergoeconomic evaluation and exergoeconomic optimization. The following sections deal with them one by one.

4.1 Exergy Analysis

The exergy analysis gives the idea about the thermodynamic inefficiencies produced in a particular process quantitatively as well as qualitatively. This inefficiency increases the cost of the final product. When the system interacts with another system and is allowed to come to equilibrium, gives work as output. Exergy can be defined as the maximum amount of theoretical useful work obtainable when the state of a system comes to the state of the environment. Thus, exergy is a measure of the departure of the state of a system from that of an environment (reference state). Pressure p_0 and temperature T_0 represent environment which is modelled as reference. In the present work, the value of p_0 and T_0 are taken as 1 atm and 25°C, respectively. Exergy is not a conserved property as some of it is destroyed due to irreversibilities. Hence it is reasonable to use exergy as a basis for thermodynamic analysis. In the absence of magnetic, electrical, nuclear and surface tension effect, the total exergy of the system is considered to be consists of four components, viz. physical exergy, kinetic exergy, potential exergy and chemical exergy.

$$\dot{E} = \dot{E}^{PH} + \dot{E}^{KN} + \dot{E}^{PT} + \dot{E}^{CH} \tag{4.1}$$

The sum of kinetic, potential and physical energies is also referred as thermo physical exergy. The physical exergy is given by the following equation.

$$\dot{E}^{PH} = \dot{m}\left[\left(h-h_0\right)-T_0(s-s_0)\right] \tag{4.2}$$

Considering the system at rest with respect to environment, total exergy becomes the sum of physical and chemical exergy.

$$\dot{E} = \dot{E}^{PH} + \dot{E}^{CH} \tag{4.3}$$

After calculating exergy at each station, exergy analysis is carried out either using exergy destruction method or entropy generation minimization method.

4.1.1 Exergy Destruction Method (EDM)

The thermal system under consideration for the analysis is supplied with some exergy inputs (fuel exergy, $\dot{E}_F$) derived from some energy source. In the process of conversion, these exergy inputs transform in to some exergy output (product exergy, $\dot{E}_P$) and some exergy be destroyed (exergy destruction, $\dot{E}_D$) and remaining is loss of exergy ($\dot{E}_L$). For the exergy analysis, it is necessary to define product and fuel for each component and for the overall system. The product is defined according to the purpose of owning and operating the component under consideration and fuel represents the resources consumed in generating the product. Fuel and product are expressed in terms of exergy. Exergy destruction is the amount of exergy lost due to irreversibilities and can not be used anywhere. The exergy loss is the amount of exergy that is wasted from the system under consideration, but can be useful to other system. The exergy destruction can be calculated by the exergy balance.

$$\dot{E}_{D,k} = \dot{E}_{F,k} - \dot{E}_{P,k} - \dot{E}_{L,k} \tag{4.4}$$

The exergetic efficiency of a component or system, ε is the percentage of the fuel exergy ($\dot{E}_{F,k}$) found in the product exergy ($\dot{E}_{P,k}$).

$$\varepsilon = \dot{E}_{P,k} / \dot{E}_{F,k} = 1 - [(\dot{E}_{D,k} + \dot{E}_{L,k}) / \dot{E}_{F,k}] \tag{4.5}$$

After calculating the exergy destruction and exergy losses for each component of the system, they are related to the fuel exergy of the component, total exergy supplied to the system and total exergy destruction in the system using exergy destruction and exergy loss ratios. The first exergy destruction ratio, $Y_{D,k}$ compares the exergy destruction in the kth component with total exergy supplied to the system using the following:

$$Y_{D,k} = \dot{E}_{D,k} / \dot{E}_{F,tot} \tag{4.6}$$

The second exergy destruction ratio, $Y^*_{D,k}$ compares the exergy destruction in the kth component with total exergy destruction in the system.

$$Y^*_{D,k} = \dot{E}_{D,k} / \dot{E}_{D,tot} \tag{4.7}$$

The two exergy destruction ratios are useful for comparisons among various components of the same system. The first exergy destruction ratio can also be invoked for comparisons among similar components of different systems using the same, or closely similar, fuels.

The exergy loss ratio, $Y_{L,k}$ is defined as the ratio between the exergy loss in the kth component and the total exergy supplied to the system.

$$Y_{L,k} = \dot{E}_{L,k} / \dot{E}_{F,tot} \tag{4.8}$$

The purpose of the exergy analysis is to identify the sources of the thermodynamic inefficiencies and to find the direction of improvement in the overall efficiency of the system through design changes.

4.1.2 Entropy Generation Minimization Method

The objective in the application of the entropy generation minimization (EGM) method is to find design in which the entropy generation is minimum. A minimum entropy generation design characterizes a system with minimum destruction of available work (exergy). In case of refrigeration plant, the minimum entropy generation rate is equivalent to maximum refrigeration load or minimum power input. This method consists of dividing the system in to sub systems those are in local (or internal) thermodynamic equilibrium. Entropy is generated at the boundaries between sub systems, as heat and mass flow through the boundaries. Using these flow rates, the total rate of entropy generation is calculated in relation to the physical characteristics of the systems. The total entropy generation is then monitored and minimized by properly varying the physical characteristics of the systems.

$$\dot{S}_g = \dot{m}_e s_e - \dot{m}_i s_i - \frac{Q_g}{T_{steam}} \tag{4.9}$$

$$I_g = T_0 \dot{S}_g \tag{4.10}$$

The EGM method is useful for the components like throttle valve and expansion valve where fuel and product can not be defined.

4.2 Exergoeconomic Analysis

Exergy analysis, in the previous section gives the quantitative values of the exergy destruction and the exergy loss in the transformation of fuel exergy to the product exergy in the system. The cost of the product depends upon the cost of fuel and cost of exergy destruction and losses. By reducing the exergy destruction and losses, the fuel requirement can be reduced and exergetic efficiency can be increased. As a result, the cost of exergy input and losses are decreased if the unit exergy cost is constant. This improvement of the system accompanies with additional investment cost. Thus, the main objective of the design engineer is to get best possible configuration to have lowest product cost by optimizing the system using exergoeconomics.

The exergoeconomic concept based cost minimization methodology calculates the economic costs of all the internal flows and products of the system by formulating exergoeconomic cost balances. The system is then exergoeconomically evaluated to identify the effects of design variables on costs and thereby enables to suggest values of design variables that would make the overall system cost-effective. Based on these suggestions put forward by Bejan et al. [155], the optimization of the system is carried out through an iterative procedure. This information is made available through the formulation of cost balance equations. The cost rate associated with the product of the system, $\dot{C}_{p,tot}$ is the total rate of expenditure made to generate it, i.e. the summation of the fuel cost rate $\dot{C}_{F,tot}$ and the cost rate associated with the total capital investment $\dot{Z}_{tot}^{TCI}$ and operation and maintenance $\dot{Z}_{tot}^{OM}$ of the system.

$$\dot{C}_{p,tot} = \dot{C}_{F,tot} + \dot{Z}_{tot} \tag{4.11}$$

$$\dot{Z}_{tot} = \dot{Z}^{TCI} + \dot{Z}^{OM} \tag{4.12}$$

4.2.1 Exergy Costing

In exergoeconomics, it is assumed that each exergy stream of the system is associated with the cost rate. Exergy costing involves cost balance formulations for each component separately. A cost balance applied to the kth component of a system shows that the sum of the cost rates associated with all leaving exergy streams equals the sum of cost rates of all entering exergy streams, the appropriate charge due to total capital investment (TCI) and operation and maintenance (O&M) expenses. The cost balance equation for the component receiving heat and generating power would be

$$\sum_e \dot{C}_{e,k} + \dot{C}_{w,k} = \dot{C}_{q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k \tag{4.13}$$

Expressing the costs in terms of cost per unit exergy ($c = \dot{C} \big/ \dot{E}$),

$$\sum_{e} (c_{e,k} \, \dot{E}_e)_k + c_{w,k} \, \dot{W}_k = c_{q,k} \, \dot{E}_{q,k} + \sum_{i} (c_{i,k} \, \dot{E}_i)_k + \dot{Z}_k \tag{4.14}$$

For a system with 'n' number of components with 'm' exergy streams (n<m), 'm' number of cost balance equations are required to calculate the cost flow rates of all the streams. For such a system, 'n' number of cost balance equations corresponding to the number of components can be developed by using Eqs. 4.13 and 4.14. The remaining '(m-n)' number of auxiliary equation can be developed through following principles.

1. When the product definition for a component involves a single exergy stream, the unit cost of this leaving stream can be calculated from the cost balance. The auxiliary relations are formulated for the remaining leaving exergy streams that are used in the definition of fuel or in the definition of exergy loss associated with the component being considered.
2. When the product definition for a component involves m leaving exergy streams, '(m-1)' auxiliary relations referring to these product streams must be formulated. In the absence of information about the production process of each of m streams, it may be assumed that each unit of exergy is supplied to each product stream at the same average cost.
3. When the fuel definition for a component involves the difference between the entering and leaving states of the same stream of matter the average cost per unit exergy remains constant for this stream. This cost changes only when exergy is supplied to the stream, which then becomes part of the product definition.

Once the cost rates ($\dot{C}_i$) associated with each stream in a system are known, the cost of fuel ($\dot{C}_F$) and the product ($\dot{C}_P$) for each of the components are obtained by using exergetic fuel and product relationship.

The term $\dot{Z}_k$ can be obtained by calculating the TCI cost and O&M costs associated with the kth component and then computing the levelized values of these costs per unit time (year, hour, second) of system operation. The variables ($c_{e,k}, c_{w,k}, c_{q,k}, c_{i,k}$) are the cost per unit exergy of the exergy streams associated with the k^{th} component. In analyzing the component, it is assumed that the cost per unit exergy of all entering streams is known as these are either the product streams of other components or the fuel streams of the overall system. The cost of fuel stream would be the purchase cost of that stream. The unknown variables can be calculated by solving the cost balance equations for all the components.

4.2.2 Economic Analysis

Economic analysis of the system includes calculation of total capital investment which includes purchased equipment cost (PEC) of all the components, installation cost, material cost, instrumentation and control cost etc and O&M costs. For incorporating these costs in the exergoeconomic cost balance equations, they are converted to levelized cost.

4.2.2.1 Estimation of TCI

The capital needed to purchase the land, build all necessary facilities and purchase and install the required machinery and equipment for a system is called fixed capital investment (FCI). The TCI is the sum of the FCI and other outlays as explained in Table 4.1 with detailed breakdown of TCI.

Purchased Equipment Cost (PEC)

The estimation of PEC can be obtained through vendor’s quotation. In case of unavailability of these, PEC can be obtained from the cost estimating chart or mathematical correlations with respect to equipment size when all available cost data are plotted against equipment size. The plot will be the straight line which is represented by the following equation.

$$C_{PE,Y} = C_{PE,W}\left(\frac{X_Y}{X_W}\right)^{\alpha} \tag{4.15}$$

Eq. 4.15 allows the purchase cost of an equipment item ($C_{PE,Y}$) at a given capacity or size (as represented by the variable X_Y) to be calculated when the purchase cost of the same equipment at a different capacity or size (expressed by X_W) is known. In the absence of other cost information, an exponent value of 0.6 may be used as suggested by Bejan et al. [155]

Table 4.1: Break down of TCI

- I Fixed capital investment (FCI)
  - A Direct cost (DC)
    - 1 Onsite costs (ONSC)
      - Purchased equipment cost (PEC)
      - Purchased equipment installation (20-90% of PEC)
      - Piping (10-70% of PEC)
      - Instrumentation and control (6-40% of PEC)
      - Electrical equipment and material (10-15% of PEC)
    - 2 Off-site costs (OFSC)
      - Land (0-10% of PEC)
      - Civil, structural and architectural work (15-90% of PEC)
      - Service facilities (30-100 % of PEC)
  - B Indirect cost (IC)
    - 1 Engineering and supervision (25-75% of PEC)
    - 2 Construction cost with contractors profit (15% of DC)
    - 3 Contingencies (20% of FCI)
- II Other outlays
  - A Startup cost (5-12% of FCI)
  - B Working capital (10-20% of TCI)
  - C Cost of licensing
  - D Allowance for funds used during construction

4.2.2.2 Cost Index

Once the PECs are known from the above method, they must be brought to the reference year, i.e. the year used as the base for the cost calculations [156,157]. It is because all data are historical and costs do change with time. This is done with the aid of appropriate cost index using the following relation.

$$\text{Cost at the reference year} = \text{original cost} \times \left(\frac{\text{cost index for the reference year}}{\begin{array}{l}\text{cost index for the year when the}\\ \text{original cost was obtained}\end{array}} \right) \tag{4.16}$$

Cost Index, an inflation indicator, is used to correct the cost of equipment, material, labour and supplies to the date of estimate. Existing cost indicators include the following: Chemical engineering plant cost index (based on construction costs for chemical plant listed in the Journal of Chemical Engineering), Marshal & Swift (M&S) equipment cost index (based on construction cost for various chemical process industries, listed in Journal of Chemical Engineering and in Oil and Gas Journal), Nelson Ferrar Refinery Cost Index (based on construction costs in the petroleum industry) and Engineering News Record (ENR) Construction Cost Index (based on general industrial construction, published in Engineering News Record). For thermal design project, the use of M&S cost index is recommended by Bejan et al. [155]. The capital recovery factor (β) is defined by the following equation

$$\beta = \left(\frac{i_{eff}(1+i_{eff})^n}{(1+i_{eff})^n - 1} \right) \left(\frac{1}{\tau} \right) h^{-1} \tag{4.17}$$

Where ‘n’ is the plant life which is considered to be 30 years, τ is the number of hours of operation per year which is taken as 8000 and i_{eff} effective annual rate of return which is taken as 10 % per year. Hence β is found to be 0.1061.

The Operation and maintenance (O&M) cost (γ) excluding fuel is assumed to be 1.092 percent of the investment cost for each component as suggested by Tsatsaronis et al. [114]. Under these assumptions, the cost flow rate (₹/hr) associated with levelized O&M cost ($\dot{Z}_k$) for the kth component is calculated from the following relation

$$\dot{Z}_k = \frac{(\beta+\gamma)*TCI_k}{\tau} \tag{4.18}$$

Where TCI_k is the total capital investment for the kth component.

4.2.2.3 Fuel Cost

The fuel cost obtained from the vendor is to be updated to the processing year. It is done with the help of the economic term escalation rate (r_n).

$$\text{Fuel cost at the reference year} = \begin{pmatrix}\text{Fuel cost at the original year}\\ \text{at which it is available}\end{pmatrix} \times (1+r_n)^n \tag{4.19}$$

Where 'r_n' is escalation rate and 'n' is the difference between year at which the cost is available and the processing year.

4.2.3 Exergoeconomic Evaluation

After introducing the cost rates associated with the fuel ($\dot{C}_F$) and product ($\dot{C}_P$), one can define cost per unit exergy of fuel and product for the kth component, $c_{F,k}$ and $c_{p,k}$, respectively as follows:

$$c_{F,k} = \frac{\dot{C}_{F,k}}{\dot{E}_{F,k}} \tag{4.20}$$

$$c_{p,k} = \frac{\dot{C}_{p,k}}{\dot{E}_{p,k}} \tag{4.21}$$

In the cost balance equation for a component, there is no cost term directly associated with the exergy destruction and exergy loss. They are hidden cost and can be defined as

$$\dot{C}_{D,k} = c_{F,k}\,\dot{E}_{D,k} \tag{4.22}$$

$$\dot{C}_{L,k} = c_{F,k}\,\dot{E}_{L,k} \tag{4.23}$$

The relative cost difference (r_k) for the kth component is defined by

$$r_k = \frac{c_{p,k} - c_{F,k}}{c_{F,k}} \tag{4.24}$$

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

$$r_k = \frac{c_{F,k}(\dot{E}_{D,k} + \dot{E}_{L,k}) + (\dot{Z}_k^{TCI} + \dot{Z}_k^{OM})}{c_{F,k}\,\dot{E}_{p,k}} \tag{4.25}$$

$$r_k = \frac{1-\varepsilon_k}{\varepsilon_k} + \frac{(\dot{Z}_k^{TCI} + \dot{Z}_k^{OM})}{c_{F,k}\,\dot{E}_{p,k}} \tag{4.26}$$

The cost sources in a component may be grouped in two categories, viz., non exergy related cost due to TCI and O&M and exergy related cost due to exergy destruction and exergy loss and the relative significance of each category can be determined by the exergoeconomic factor f_k defined for component k by

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + (\dot{C}_{D,k} + \dot{C}_{L,k})} \tag{4.27}$$

f_k is the ratio of non exergy related cost to the total cost. A low value of it for a major component suggests that cost saving in the entire system might be achieved by improving the component efficiency (reducing the exergy destruction) even if the capital investment for this component will increase. On the other hand, a high value of this factor suggests a decrease in the investment cost of the component at the expense of its exergetic efficiency. The system can be evaluated with the help of exergoeconomic variables given in Eqs. 4.20 to 4.27.

Bejan et al. [155] has suggested the following methodology for exergoeconomic evaluation:

1. Consider design changes initially for the components for which the value of the sum ($\dot{Z}_k + \dot{C}_{D,k}$) is high.
2. Pay particular attention to the components with a high relative cost difference r_k, especially when the cost rates $\dot{Z}_k$ and $\dot{C}_{D,k}$ are high.
3. Use the exergoeconomic factor f_k to identify the major cost source (capital investment or exergy destruction).
   a. If the f_k value is high, investigate whether it is cost effective to reduce the capital investment at the expense of component efficiency.
   b. If the f_k value is low, try to improve the component efficiency by increasing the capital investment.
4. Eliminate any sub processes that increase the exergy destruction or exergy loss without contributing to the reduction of capital investment or fuel cost for other component.
5. Consider improving the exergetic efficiency of the component if it has low exergetic efficiency or large value of the rate of exergy destruction, the exergy destruction ratio or exergy loss ratio.

4.3 Exergoeconomic Optimization

Optimization means the modification of the structure and the design parameters of a system to minimize the total levelized cost of the system product under the given boundary conditions. The objective of the exergoeconomic optimization is to minimize costs including costs owing to thermodynamic inefficiencies. The objective function expresses the optimization criteria as a function of dependent and independent variables.

$$Minimize\ \dot{C}_{p,tot} = \dot{C}_{F,tot} + \dot{Z}^{TCI}_{tot} + \dot{Z}^{OM}_{tot} \tag{4.28}$$

$\dot{C}_{p,tot}$ is total cost rate associated with the product instead of the cost rate per unit exergy of product $\dot{c}_p$, since the exergy flow rate of the product $\dot{E}_P$ is constant. In this approach the cost optimal exergetic efficiency can be obtained for a component isolated from the remaining system components. The optimization approach is based on the following assumptions.

1. The exergy flow rate of the product $\dot{E}_p$ and the unit cost of fuel c_F remain constant for the kth component to be optimized.

   $\dot{E}_{p,k} = \text{constant}$

   $c_{F,k} = \text{constant}$

2. For every component, it is expected that the investment cost increases with increasing capacity and increasing exergetic efficiency of the component. Therefore TCI$_k$ for the kth component can be approximated by the following relation. [155]

   $$TCI_k = B_k\left(\frac{\varepsilon_k}{1-\varepsilon_k}\right)^{n_k} \dot{E}_{p,k}^{m_k} \tag{4.29}$$

   Where $\dot{E}_{p,k}$ is the exergy rate of the product for the kth component and ε_k is the component's exergetic efficiency. The term $[\varepsilon_k/(1-\varepsilon_k)]$ expresses the effect of exergetic efficiency (thermodynamic performance) while the term $\dot{E}_{p,k}^{m_k}$ expresses the effect of capacity (component size) on the value of TCI$_k$. Eq. 4.29 is valid within a certain range of design conditions for the kth component. Within that range, the parameter B_k and the exponents n_k are constants, and can be calculated based on cost data through curve fitting technique. For simplicity, the value of m_k can be assumed equal to the scaling exponent α for the respective equipment as explained in Eq. 4.15 and suggested by Bejan et al. [155].

3. Usually a part of the O&M cost depends on the total investment cost and another part on the actual production rate. Then the annual O&M cost for the kth component can be represented by

   $$Z_k^{OM} = \gamma_k(TCI_k) + \omega_k \tau \dot{E}_{p,k} + R_k \tag{4.30}$$

   In this equation, γ_k is a coefficient that accounts for the part of the fixed O&M cost depending on the TCI$_k$ associated with the kth component, ω_k is a constant accounts for the variable O&M cost associated with the kth component and denotes the O&M cost per unit of product exergy. τ is the average annual time of plant operation at the nominal load and R_k includes all the remaining O&M cost that are independent of the TCI and exergy of the product.

4. The economic analysis of the system is simplified by neglecting the effect of financing, inflation, taxes, insurance and construction time and by considering the start-up cost, working capital and the cost of licensing, research and development together with the total capital investment. The annual carrying charge associated with the kth component can then be obtained by multiplying the TCI for this component by the capital recovery factor β.

$$Z_k^{CI} = \beta(TCI_k) \tag{4.31}$$

The above assumptions form the cost model. The total annual costs, excluding fuel cost, associated with the kth component are obtained by combining Eqs. 4.30 and 4.31.

$$Z_k = Z_k^{CI} + Z_k^{OM} = (\beta + \gamma_k)(TCI_k) + \omega_k \tau \dot{E}_{p,k} + R_k \tag{4.32}$$

The corresponding cost rate $\dot{Z}_k$ is obtained by dividing Eq. 4.32 by annual hours of operation τ.

$$\dot{Z}_k = \frac{(\beta + \gamma_k)}{\tau}(TCI_k) + \omega_k \dot{E}_{p,k} + \frac{R_k}{\tau} \tag{4.33}$$

Inserting the value of TCI_k from Eq. 4.29

$$\dot{Z}_k = \frac{(\beta + \gamma_k)B_k}{\tau}\left(\frac{\varepsilon_k}{1-\varepsilon_k}\right)^{n_k} \dot{E}_{p,k}^{m_k} + \omega_k \dot{E}_{p,k} + \frac{R_k}{\tau} \tag{4.34}$$

The objective function to be minimized expresses the cost per exergy unit of product for the kth component

$$\text{Minimize } c_{p,k} = \frac{c_{F,k}\dot{E}_{F,k} + \dot{Z}_k}{\dot{E}_{p,k}} \tag{4.35}$$

From Eqs. 4.5 and 4.34, this objective function may be expressed as

$$Minimize\ c_{p,k} = \frac{c_{F,k}}{\varepsilon_k} + \frac{(\beta+\gamma_k)B_k}{\tau \dot{E}_{p,k}^{1-m_k}}\left(\frac{\varepsilon_k}{1-\varepsilon_k}\right)^{n_k} + \omega_k + \frac{R_k}{\tau \dot{E}_{p,k}} \tag{4.36}$$

The values of parameters β, γ_k, B_k, τ, ω_k and R_k remain constant during optimization process and so $c_{p,k}$ varies only with ε_k. Thus the optimization problem reduces to the minimization of Eq. 4.36 subject to constrain explained in assumption 1. The minimum cost per unit exergy of product can be obtained by differentiating Eq. 4.36 and setting the derivative to zero.

$$\frac{dc_{p,k}}{d\varepsilon_k} = 0$$

The resulting cost optimal exergetic efficiency is

$$\varepsilon_k^{OPT} = \frac{1}{1+F_k} \tag{4.37}$$

Where

$$F_k = \left(\frac{(\beta+\gamma_k)B_k n_k}{\tau c_{F,k} \dot{E}_{p,k}^{1-m_k}}\right)^{\frac{1}{(n_k+1)}} \tag{4.38}$$

Eqs. 4.37 and 4.38 show that the cost optimal exergetic efficiency increases with increasing cost per unit exergy of fuel $c_{F,k}$, increasing annual number of hours of system operation τ, decreasing capital recovery factor β, decreasing fixed O&M cost factor γ_k and decreasing cost exponent n_k. From Eq. 4.37

$$F_k = \frac{1-\varepsilon_k^{OPT}}{\varepsilon_k^{OPT}} \tag{4.39}$$

From Eqs. 4.5 and 4.39

$$F_k = \left(\frac{\dot{E}_{D,k} + \dot{E}_{L,k}}{\dot{E}_{p,k}}\right)^{OPT} \tag{4.40}$$

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

$$(\dot{E}_{D,k}+\dot{E}_{L,k})^{OPT}=\dot{E}_{p,k}\,F_k=\dot{E}_{p,k}\left(\frac{1-\varepsilon_k^{OPT}}{\varepsilon_k^{OPT}}\right) \tag{4.41}$$

Eq. 4.36 can be reduced by neglecting the last two terms as

$$\text{Minimize } c_{p,k}=c_{F,k}\left(1+\frac{\dot{E}_{D,k}+\dot{E}_{L,k}}{\dot{E}_{p,k}}\right)+\frac{(\beta+\gamma_k)B_k}{\tau\,\dot{E}_{p,k}^{\,1-m_k}}\left(\frac{\dot{E}_{p,k}}{\dot{E}_{D,k}+\dot{E}_{L,k}}\right)^{n_k} \tag{4.42}$$

By differentiating Eq. 4.42, with respect to $(\dot{E}_{D,k}+\dot{E}_{L,k})$ and setting the derivative to zero, the relation between cost optimal values of the cost rates can be expressed by $c_{F,k}(\dot{E}_{D,k}+\dot{E}_{L,k})$ and $\dot{Z}_k$

$$n_k=\frac{c_{F,k}(\dot{E}_{D,k}+\dot{E}_{L,k})}{\dot{Z}_k^{OPT}} \tag{4.43}$$

From Eqs. 4.22 and 4.23

$$n_k=\frac{(\dot{C}_{D,k}+\dot{C}_{L,k})^{OPT}}{\dot{Z}_k^{OPT}} \tag{4.44}$$

Thus, the cost exponent n_k expresses the ratio between the cost optimal rates associated with the exergy destruction and exergy loss and cost optimal rates associated with capital investment. From this equation, the expressions for cost optimal values of the non fuel related cost rate $\dot{Z}_k$, the relative cost difference r_k and the exergoeconomic factor f_k can be obtained as shown below.

$$\dot{Z}_k^{OPT}=c_{F,k}\,\dot{E}_{p,k}\,\frac{F_k}{n_k} \tag{4.45}$$

$$r_k^{OPT}=\left(\frac{n_k+1}{n_k}\right)F_k \tag{4.46}$$

$$f_k^{OPT}=\frac{1}{1+n_k} \tag{4.47}$$

In the present optimization problem, though the main goal is to obtain the optimum value of the product cost, the cost of exergy destruction and the cost of exergy loss also have to be minimum. Therefore, the objective function for the overall system can be defined as

$$\text{Minimize OBF}=\dot{C}_{p,tot}+\dot{C}_{D,tot}+\dot{C}_{L,tot} \tag{4.48}$$

Therefore $\dot{C}_{p,tot}$ is to be optimized. From Eq. 4.21

$$\dot{C}_{p,tot}=c_{p,tot}E_{p,tot} \tag{4.49}$$

$c_{p,tot}$ can be optimized using Eq. 4.29. To solve this equation for the local optimum condition, the parameter B_k and the exponents n_k and m_k are to be evaluated. They are calculated based on the cost data for each component. The value of m_k can be assumed equal to the scaling exponent α explained in Eq. 4.15 and can be referred from Bejan et al. [155].

Then, based on thermodynamic and the cost data, the variation of the exergetic product ($\dot{E}_{p,k}$), the exergy destruction ($\dot{E}_{D,k}$) and the TCI_k with respect to the exergetic efficiency corresponding to variation in local decision variable can be generated. From the generated data for each component, $(TCI_k/\dot{E}_{p,k}^{m_k})$ can be plotted against $(\dot{E}_{p,k}/\dot{E}_{D,k})$ which is equivalent to $[\varepsilon_k/(1-\varepsilon_k)]$. By curve fitting technique, the equivalent power law can be found and the required value of B_k and n_k for each component can be determined.

After calculating the values of constants B_k and n_k, the cost optimal values of the exergetic efficiency ε_k^{OPT} the relative cost difference r_k^{OPT}, total exergy loss $(\dot{E}_{D,k}+\dot{E}_{L,k})^{OPT}$, the capital investment Z_k^{OPT} and thermoeconomic factor f_k^{OPT} can be calculated from Eqs. 4.37 to 4.47.

The optimization procedure using the above approach is an iterative one that aims at finding out a better solution for the system, unlike conventional optimization procedure, where the aim is to calculate the global optimum. In the iterative optimization procedure, the following thermoeconomic variables are defined to facilitate the decision-making:

$$\Delta\varepsilon_k = 100 \times \frac{\varepsilon_k - \varepsilon_k^{OPT}}{\varepsilon_k^{OPT}} \tag{4.50}$$

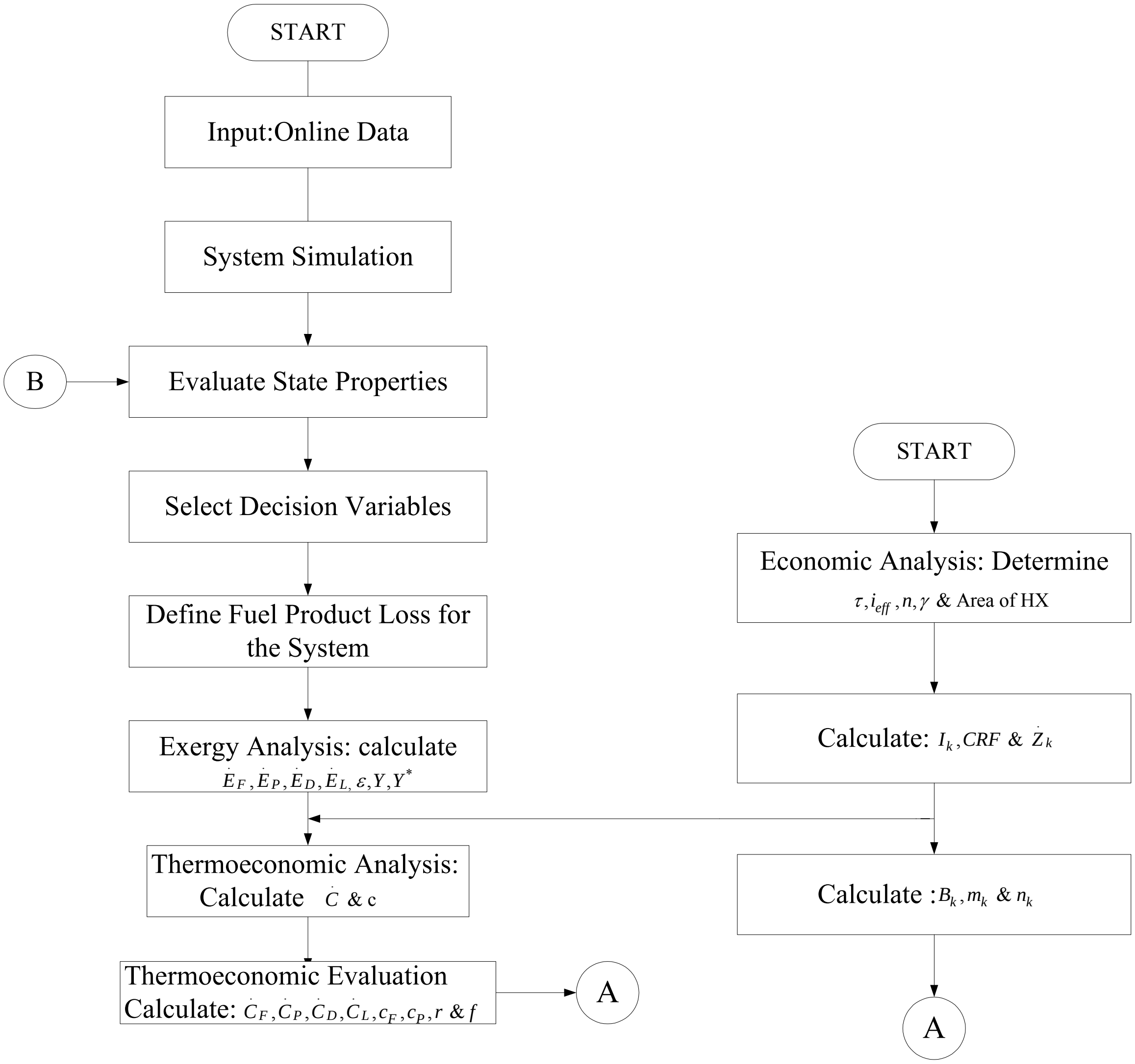
$$\Delta r_k = 100 \times \frac{r_k - r_k^{OPT}}{r_k^{OPT}} \tag{4.51}$$

The value of $\Delta\varepsilon_k$ and Δr_k express the respective relative deviation of actual values from optimal values. In the iterative optimization procedure, engineering judgments and critical evaluations are used in deciding on the changes made to the decision variables from one iterative step to the next. Also, while taking the decision on the changes of the decision variables, the practical limitations of the system, mentioned earlier, are also considered. The criteria followed in decision-making on the changes of the decision variables from one iterative step to the next are as follows

- Calculation of $\Delta\varepsilon_k, \Delta r_k, \Delta f_k, \dot{C}_{p,tot}$ and $\dot{C}_{D+L}$ variables for a change in one decision variable in a certain step, while keeping other decision variables constant.
- Examination of its effects on the exergoeconomic variables.
- If the effect is positive, i.e., $\dot{C}_{p,tot}$ and $\dot{C}_{D+L}$ has reducing trend, then in the next iterative step this variable becomes a candidate for a similar change, otherwise, this variable remains unchanged in the next iterative step.
- Repetition of the above three steps for the other decision variables.

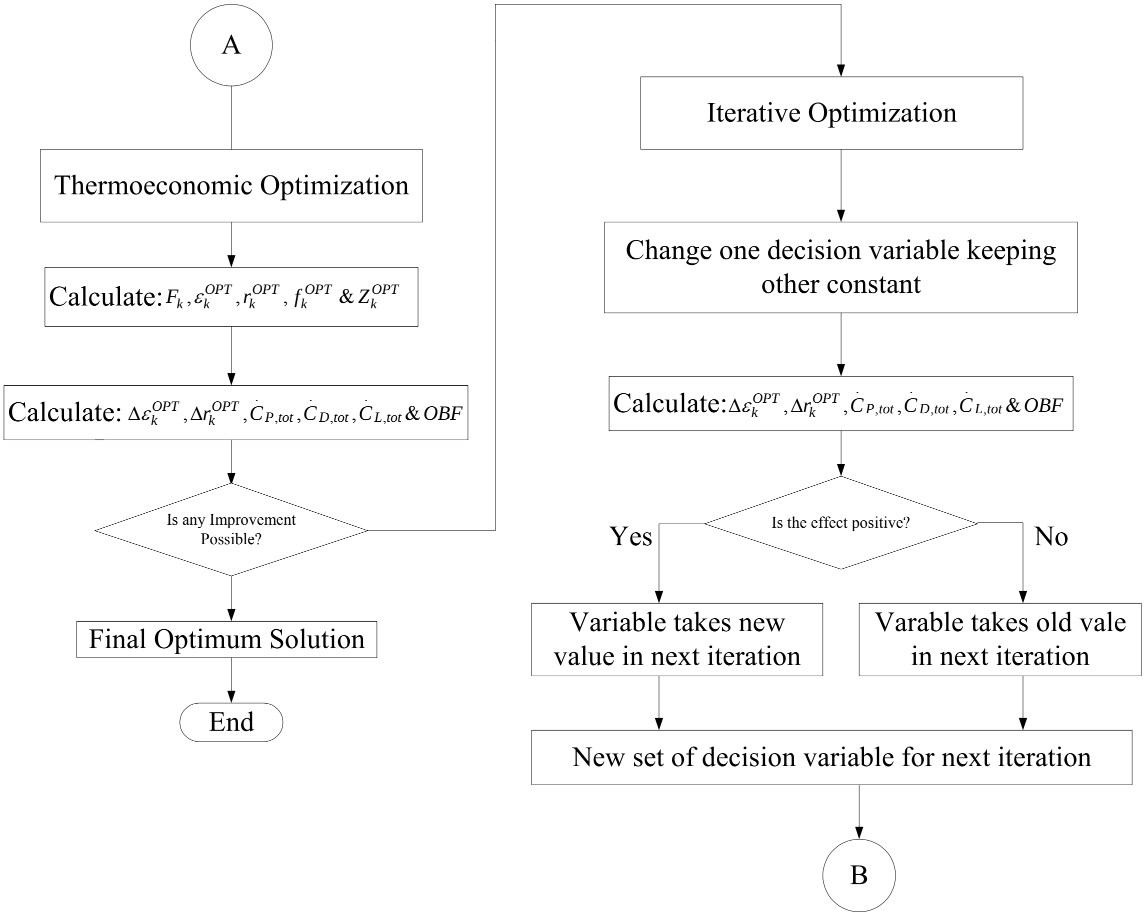
4.4 Unified Approach for Exergoeconomic Optimization

A unified approach of combining the TEO method by Tsatsaronis [1] along with the iterative procedure by Bejan et al.[155] is presented in the above sections dealing with the three steps proposed, viz. exergy analysis, exergoeconomic evaluation and exergoeconomic optimization. A computer code written in EES software based on the unified approach is developed. The flow chart of the same is given in Fig. 4.1.



(a) Exergoeconomic Analysis **(b) Economic Analysis**

Fig. 4.1 Flow Chart for Unified Approach of Exergoeconomic Optimization
(Continued)



(c) Thermoeconomic Optimization

Fig. 4.1 Flow Chart for Unified Approach of Exergoeconomic Optimization

The flow chart is described in the following section

Module 1:

Step 1: Input the online data from control panel

Step 2: Simulate the system through energy balance and mass balance

Step 3: Evaluate the state properties and Exergy at each station (Eqs. 4.1 to 4.3)

Step 4: Select decision variables

Step 5: Define Fuel, Product and Loss for every component

Step 6: Exergy analysis: Calculate $\dot{E}_F, \dot{E}_P, \dot{E}_D, \dot{E}_{L,} \varepsilon, Y$ and Y^* using Eqs. 4.4 to 4.10

Step 7: Thermoeconomic analysis: Calculate $\dot{C}$ & c using the values of I_k, CRF & $\dot{Z}_k$ from step 2 of module 2 and Eqs.4.13, 4.14 & 4.19

Step 8: Thermoeconomic evaluation: Calculate $\dot{C}_F, \dot{C}_P, \dot{C}_D, \dot{C}_L, c_F, c_P, r \,\&\, f$ using Eqs. 4.20 -4.27

Module- 2

Step 1: Economic analysis: Consider the value of $\tau, i_{eff}, n \,\&\, \gamma$ as input and calculate area of heat exchanger using eq. 4.15 and PEC from cost model for other components

Step 2: Using step1, calculate I_k, CRF & $\dot{Z}_k$ using Eqs 4.16- 4.18

Step 3: Calculate the values of B_k, m_k & n_k using Eq. 4.29

Module 3

Step 1: Thermoeconomic optimization: Calculate $F_k, \varepsilon_k^{OPT}, r_k^{OPT}, f_k^{OPT} \,\&\, Z_k^{OPT}$ using step 8 of module 1 and step 3 of module2 and Eqs.4.37 to 4.45

Step 2: Calculate $\Delta\varepsilon_k^{OPT}, \Delta r_k^{OPT} \,\&\, OBF$ using Eqs 4.48-4.51

Step 3: If any improvement possible? Go to iterative optimization

Step 4: Change one decision variable keeping other constant

Step 5: Calculate $\dot{C}_{P,tot}, \dot{C}_{D,tot}, \dot{C}_{L,tot} \,\&\, OBF$ for the new value of variable

Step 6: Calculate $\Delta\varepsilon_k^{OPT}$ and Δr_k^{OPT} and check whether the effect is positive or negative

Step 7: If the effect is positive, the variable takes new value in next iteration and reach final optimum solution.

Step 8: If the effect is negative, the variable takes old value in the next iteration then go to Step 3 of module 1 and determine new set of decision variable.

Step 9: By more iteration, reach the final optimum solution.

Chapter 5

Exergoeconomic Optimization of Existing System

The unified scheme of exergoeconomic optimization using a combination of TEO method, suggested by Tsatsaronis and an exergy analysis using EGM and EDM approach is given in chapter 4. A flow chart depicting the various steps involved in the unified scheme is also given. This scheme is now employed to carry out the Exergoeconomic Optimization of an industrial AAVAR system in which economy with existing heat source is analyzed.

5.1 Exergy Analysis

For the case of an industrial 800 TR AAVAR system used for brine chilling in which the steam generated in an independent boiler (boiler No. 38/02) is used as heat source (fuel), the cooling cost for the cooling generated at the evaporator of the AAVAR system is optimized and presented in this chapter. The following are the various steps involved in the cooling cost optimization.

5.1.1 System Simulation

Using the governing equations for the processes undergone in each component considering mass, energy and concentration balances [148,149,150], AAVAR system simulation is carried out with the help of EES software [151]. For estimating the properties such as enthalpy, entropy and specific volume of aqua ammonia solution circulated in the condensing unit, the inbuilt subroutine of EES software is used. They are then compared with the properties provided by Ziegler et al. [152] and Patek et al. [153].

The simulation is necessary as data needed at number of stations are not available online during normal operation of the plant. This may be due to the reason that they are not critical for the day to day monitoring and operation of the plant. The simulation model along with the fundamental equation is given in Appendix A1. For the simulation of the AAVAR system, following assumptions are taken.

1. The system is in a steady state.
2. The temperatures of the component in the generator, condenser, evaporator and absorber are constant and uniform.
3. The generator and rectifier pressures are equal.
4. The pressure losses in the pipe between the rectifier and the condenser, and between the RHX 05 and absorber are expressed as $\Delta p/p_{out} = 0.05$ and $\Delta p/p_{out} = 0.075$, respectively [90].
5. The condenser pressure and evaporator pressures are the equilibrium pressures corresponding to the temperature and concentration in the condenser and in the evaporator, respectively.
6. The refrigerant vapour concentration at station 3 is 0.99 and the temperature at station 4 is 100°C.

As mentioned earlier, it should be noted that the online data available from the AAVAR plant (refer Table 3.1) is not complete for the purpose of exergy analysis due to the reason that the data needed for the smooth functioning and monitoring of the plant operation does not need all of them. Hence few data not available at certain stations are to be worked out which one cannot measure due to obvious reasons of the impossibility of introducing measurement systems in the plant. Therefore, a system simulation should be carried out to generate the missing data. Table 5.1 gives the data not available through online measurements generated through simulation using EES solver and already available data from the online monitoring. Table 5.1 also gives the estimated values of enthalpy and entropy at stations 1 to 18 using the solver.

The AAVAR system uses steam generated in the independent boiler No. 38/02 as heat source. The boiler supplies saturated steam at 15 bar at a flow rate of 3.14 kg/sec.

The properties of steam at given pressure and corresponding saturation temperature are estimated using inbuilt subroutine of EES software at stations 19 and 20.

Table 5.1 Generated Online Data for AAVAR System

Stations	mass flow rate kg/sec	Pressure. bar	Temperature. ºC	NH_3 Concen tration % wt	Enthalpy kJ/kg	Entropy kJ/kgK
1	18.28	18.90	107.70	0.270	292.80	1.3760
2	15.70	18.90	140.00	0.150	497.30	1.7800
3	2.62	18.90	140.00	0.990*	1576.00	4.8730
4	0.04	18.90	100.00*	0.448	216.70	1.2430
5	3.17	18.90	60.00	0.998	1338.00	4.2330
6	2.59	18	40.00	0.998	189.30	0.6580
7	2.59	18	35.87	0.998	168.90	0.5924
8	2.59	18	11.65	0.998	52.97	0.2018
9	2.59	1.90	-19.90	0.998	52.97	0.2415
10	0.24	1.90	-20.00	0.970	-113.60	-0.3284
11	0.24	1.90	-19.22	0.970	111.00	0.5523
12	2.35	1.90	-20.00	1.000	1243.00	4.9140
13	2.35	1.90	35.16	1.000	1371.00	5.3710
14	18.28	1.77	40.00	0.270	-3.58	0.5254
15	18.28	18.90	40.14	0.270	-1.50	0.5261
16	15.70	18.90	60.58	0.150	154.50	0.8593
17	15.70	1.77	60.90	0.150	154.50	0.8648
18	0.58	18	40.00	0.998	189.30	0.6580

(* Assumed data)

At evaporator, the cooling generated is utilized in chilling the brine. The temperature of brine at inlet and exit are measured online. The properties of brine at stations 21 and 22 are estimated with the help of EES software

Condenser and absorber are supplied cooling water from common cooling tower No. 34/02 at 33°C. The cooling water flow rate through condenser is 317 m^3 /hr. (The equivalent mass flow rate of cooling water through condenser is 88.06 kg/sec). After absorbing the heat from ammonia vapour in the condenser, the cooling water is heated to

42.8°C and returns to cooling tower. The volume flow rate of cooling water through absorber is 453 m^3/hr. (The equivalent mass flow rate is 125 kg/sec). In the absorber, the ammonia vapour from evaporator and weak solution from generator are mixed. The mixing process is exothermic and the released heat is absorbed by cooling water in the absorber. The cooling water at the absorber exit is heated to 43.30°C and returns to cooling tower. The temperature of cooling water at condenser and absorber exit is calculated from energy balance as explained in Appendix A2. Considering the cooling water and brine at atmospheric pressure, the properties of cooling water (stations 23 to 26) are estimated using EES software and given in Table 5.2.

After estimating the properties at various stations of AAVAR system, properties of working fluids at various stations of pre-cooler-1 and pre-cooler-2 are estimated. Pre-cooler-1 cools the incoming brine at a flow rate of 125 kg/sec from the process plant and enters the evaporator of AAVAR at a temperature of 24.7°C. The fertilizer industry is manufacturing many products. Ammonia, one among them, is used as raw material and is manufactured at two pressure levels viz. 4 bar and 2.3 bar in different manufacturing plants in saturated liquid form. Saturated liquid ammonia at 4 bar enters the shell side of the pre-cooler-1 at a steady rate of 9.2 ton per hour and evaporates while absorbing latent heat from the brine. The exit temperature of brine is estimated through energy balance across Pre-cooler-1 and found to be 15.9°C.The ammonia leaving the pre-cooler-1 is heated up to 6.4°C and consumed in the fertilizer plant.

The brine from pre-cooler-1 then enters the tube side of pre-cooler-2, which is a shell and tube type heat exchanger, at 15.9°C. Saturated ammonia at 2.3 bar at the steady rate of 10.3 ton per hour enters the shell side and evaporates while absorbing latent heat from the brine and cools the brine up to 5.4°C. During the heat exchange process, the ammonia at exit is heated up to 12.5°C and consumed in the fertilizer plant. The properties of ammonia at stations 31 to 34 are estimated at given temperature and pressure using inbuilt subroutine of EES software and given in Table 5.2. The temperature of brine at exit of Pre-cooler-1 is estimated through energy balance at Pre-cooler-1 as explained in Appendix A2 and its properties at station 30 is estimated using EES software and given in Table 5.2.

Table 5.2 Generated Online Data for AAVAR System and Pre-coolers

Stations	mass flow rate kg/sec	Pressure. bar	Temperature. ºC	NH_3 Concentration % wt	Enthalpy kJ/kg	Entropy kJ/kgK
19	3.14	15.00	198.30	0	2791.00	6.4440
20	3.14	15.00	198.30	0	1370.00	3.4300
21	125.00	1.01	5.40	0	16.55	0.0600
22	125.00	1.01	-1.70	0	-5.19	-0.0191
23	88.06	1.01	33.00	0	138.30	0.4777
24	88.06	1.01	44.80	0	171.80	0.5856
25	125.00	1.01	33.00	0	138.30	0.4777
26	125.00	1.01	43.30	0	171.80	0.5856
27	---	---	---	---	---	---
28	---	---	---	---	---	---
29	125.00	1.01	24.70	0	76.36	0.2677
30	125.00	1.01	15.90	0	48.97	0.1743
31	2.56	4.00	-1.89	---	191.30	0.9681
32	2.56	4.00	6.40	---	1482.00	5.7240
33	2.85	2.3	-15.62	---	128.40	0.7312
34	2.85	2.3	12.50	---	1510.00	6.0830

5.1.2 Exergy Destruction Method (EDM) of Exergy Analysis

After estimation of properties at all the stations 1 to 34, the exergy flow at all the stations is calculated. Considering the system at rest with respect to environment, the total exergy becomes the sum of physical and chemical exergy. The physical exergy component associated with the work obtainable in bringing a matter from its initial state to a state that is in thermal and mechanical equilibrium with the environment is given by the Eq. 4.2. Chemical exergy associated with the fluid is calculated using the following:

$$\dot{E}_i^{CH} = \dot{m}_i \left[\left(\frac{x_i}{M_{NH_3}} \right) e^0_{CH,NH_3} + \left(\frac{1-x_i}{M_{H_2O}} \right) e^0_{CH,H_2O} \right] \quad (5.1)$$

Where $e^0_{CH,NH_3} = 341250$ kJ/kmol and $e^0_{CH,H_2O} = 3120$ kJ/kmol are the standard chemical exergy of ammonia and water, respectively and is given by Kotas [118]. $M_{NH_3} = 17$ kg/kmol and $M_{H_2O} = 18$ kg/kmol are molecular weight of ammonia and water, respectively. The exergy flow in terms of physical, chemical and total exergy at all stations is estimated using Eqs. 4.2, 5.1 and 4.3, respectively and is given in Table 5.3. For this purpose, the enthalpy and entropy available at thirty four stations in brine chilling unit estimated from the online data and given in Table 5.1 and Table 5.2 are used. A sample calculation is given in Appendix B1.

5.1.2.1 Definition of Fuel, Product and Loss for Various Processes

The calculated values of physical, chemical and total exergy given in Table 5.3 can now be used to define fuel, product and loss for the purpose of exergoeconomic analysis. Based on the definition of fuel, product and loss given in Section 4.1.1, the estimated values of exergy can be translated in to the fuel, the product and the loss for each component of the AAVAR system. Following few paragraphs are devoted for the definition with respect to each component of AAVAR.

Generator

The function of generator in AAVAR system is to separate ammonia vapour from the strong aqua ammonia solution by heating. This is accomplished by using steam generated in an independent boiler. In the generator, therefore, by adding heat energy to the strong aqua ammonia solution, ammonia vapour is separated from the solution to the greatest extent possible. The separated water along with residual ammonia as mixture flows back to absorber. As per exergy analysis point of view, the transfer of steam in to the generator is interpreted as transfer of exergy from steam. Thus the exergy gained by the ammonia vapour separated and the exergy of the leaving stream of aqua ammonia weak solution becomes the product for the generator.

Rectifier

In the rectifier, the exergy of reflux is used to increase the exergy of ammonia separated in the form of vapour (refrigerant) in the generator by condensing the water vapour as a carryover by evaporating some portion of reflux. So the concentration of the ammonia vapour can be increased by vaporizing liquid water droplets present in ammonia vapour. Therefore, the decrease in the exergy of reflux is the fuel and rise in the exergy of ammonia vapour is the product for rectifier.

Table 5.3 State Properties for AAVAR System

Stations	Mass flow rate kg/sec	Pres-sure. bar	Temp. ºC	NH_3 Concen tration % wt	Specific Enthalpy kJ/kg	Specific Entropy kJ/kgK	Chemical Exergy kW	Physical Exergy kW	Total Exergy kW
1	18.28	18.70	107.70	0.270	292.80	1.3760	101407	811.90	102219
2	15.70	18.70	140.00	0.150	497.30	1.7800	49574	1232.00	50806
3	2.62	18.70	140.00	0.990*	1576.00	4.8730	52180	1142.00	53321
4	0.04	18.70	100.00*	0.448	216.70	1.2430	346.9	1.89	348.7
5	3.17	18.70	60.00	0.998	1338.00	4.2330	63470	1257.00	64727
6	2.59	17.81	40.00	0.998	189.30	0.6580	51833	812.80	52645
7	2.59	17.81	35.87	0.998	168.90	0.5924	51833	810.60	52643
8	2.59	17.81	11.65	0.998	52.97	0.2018	51833	811.90	52645
9	2.59	1.90	-19.90	0.998	52.97	0.2415	51833	781.30	52614
10	0.24	1.90	-20.00	0.970	-113.60	-0.3284	4581	70.74	4652
11	0.24	1.90	-19.22	0.970	111.00	0.5523	4581	61.79	4643
12	2.35	1.90	-20.00	1.000	1243.00	4.9140	47214	229.10	47443
13	2.35	1.90	35.16	1.000	1371.00	5.3710	47121	208.50	47329
14	18.28	1.77	40.00	0.270	-3.58	0.5254	101407	29.90	101437
15	18.28	18.70	40.14	0.270	-1.50	0.5261	101407	64.65	101471
16	15.7	18.70	60.58	0.150	154.50	0.8593	49574	158.60	49733
17	15.7	1.77	60.90	0.150	154.50	0.8648	49574	132.90	49707
18	0.58	18.70	40.00	0.998	189.30	0.6580	11637	182.50	11820
19	3.14	15.00	198.30	0	2791.00	6.4440	36725	2744.00	39469
20	3.14	15.00	198.30	0	1370.00	3.4300	36725	1105.00	37830
21	125.00	1.01	5.40	0	16.55	0.0600	1.50E+06	283.50	1.50E+06
22	125.00	1.01	-1.70	0	-5.19	-0.0191	1.50E+06	510.40	1.50E+06
23	88.06	1.01	33.00	0	138.30	0.4777	274733	38.74	274772
24	88.06	1.01	44.80	0	171.80	0.5856	274733	152.50	274886
25	125.00	1.01	33.00	0	138.30	0.4777	392600	55.36	392655
26	125.00	1.01	43.30	0	171.80	0.5856	392600	217.90	392818
27	---	---	---	---	---	---	---	---	42.38
28	---	---	---	---	---	---	---	---	38.14
29	125.00	1.01	24.70	0	76.36	0.2677	1.50E+06	0.06	1.50E+06
30	125.00	1.01	15.90	0	48.97	0.1743	1.50E+06	55.33	1.50E+06
31	2.56	4.00	-1.89	---	191.30	0.9681	51388	829.5	52218
32	2.56	4.00	6.40	---	1482.00	5.7240	51388	503.7	51892
33	2.85	2.3	-15.62	---	128.40	0.7312	57210	945.5	58155
34	2.85	2.3	12.50	---	1510.00	6.0830	57210	336.8	57546

Heat Exchangers – SHX, RHX05 and RHX06

In the solution heat exchanger (SHX), the exergy of the strong aqua ammonia solution flowing to the generator is increased by transferring the exergy of weak aqua ammonia solution from the generator while flowing to the absorber. In the ammonia heat exchanger (RHX 05), the liquid ammonia leaving the condenser is sub cooled by the vapour leaving the evaporator and in (RHX 06) the liquid ammonia from the condenser is sub cooled by the separated aqua ammonia weak solution in the evaporator.

Absorber

Ammonia vapour (refrigerant) from the evaporator and the weak aqua ammonia solution (absorber) from generator are mixed in the absorber. In fact, the absorber acts as an absorber for ammonia vapour (refrigerant) in to liquid water and forms a strong aqua ammonia solution. The process of ammonia absorption is an exothermic reaction and heat released is dissipated in the cooling water circulated in cooling coils. Weak solution from generator and ammonia vapour from evaporator are considered as a fuel. The strong aqua ammonia solution flowing to the generator is considered as a product. The dissipated heat energy in to the cooling water is considered as loss. The exergy of the strong solution leaving the absorber is increased in the solution pump by transferring the exergy of the external work provided to the pump.

Condenser-Evaporator

The evaporator increases the exergy of the chilled brine by transferring the exergy from the refrigerant. In case of heat dissipative components like condenser, throttling valve, pressure reducing valve, the product cannot be defined. Therefore evaporator and condenser together are considered as single virtual component as suggested by Sahoo et al. [90]. Heat rejected in condenser in cooling water is considered as exergy loss. For expansion valve and pressure reducing valve, fuel and product cannot be decided but only exergy destruction can be calculated which is added in the exergy destruction of the overall system.

Pre-cooler-1 and Pre-cooler-2

As mentioned earlier, saturated liquid ammonia at two different pressure levels is used for pre-cooling the brine before entering the ammonia (refrigerant) evaporator. The latent heat of evaporation and heat required for superheating is absorbed from the brine coming from the process plant for brine chilling. Therefore, the change in exergy of the evaporating ammonia is considered as a fuel and the chilling of the brine is the product in both the pre-coolers.

Fig. 5.1 illustrates the schematic of the sub systems of AAVAR system for the purpose of exergoeconomic analysis and optimization. It should be noted that, conventionally, rectifier-generator assembly is a part of the condensing unit. Evaporator is one of the other main components along with condensing unit in the plant for conventional thermodynamic analysis. Since the product cannot be defined for condenser as mentioned earlier, the evaporator and condenser together are considered as a single virtual component for the purpose of exergoeconomic analysis. Table 5.4 gives the defined fuel, product and loss for each of the ten productive components considered for analysis, namely, generator, rectifier, condenser-evaporator assembly, solution pump, SHX, RHX 05, RHX 06, absorber, pre-cooler-1 and pre-cooler-2.

Table 5.4 Definition of Fuel, Product and Loss for Components of AAVAR System

Component	**Fuel ($\dot{E}_F$)**	**Product($\dot{E}_P$)**	**Loss($\dot{E}_L$)**
Generator	$\dot{E}_{19}-\dot{E}_{20}$	$\dot{E}_2+\dot{E}_3-\dot{E}_1-\dot{E}_4$	----
Rectifier	$\dot{E}_{18}-\dot{E}_4$	$\dot{E}_5-\dot{E}_3$	----
Condenser Evaporator. Assembly	$\dot{E}_5-\dot{E}_{18}-\dot{E}_6+\dot{E}_9-\dot{E}_{12}-\dot{E}_{10}$	$\dot{E}_{22}-\dot{E}_{21}$	$\dot{E}_{24}-\dot{E}_{23}$
SHX	$\dot{E}_2-\dot{E}_{16}$	$\dot{E}_1-\dot{E}_{15}$	----
RHX 05	$\dot{E}_{12}-\dot{E}_{13}$	$\dot{E}_8-\dot{E}_7$	----
RHX 06	$\dot{E}_{10}-\dot{E}_{11}$	$\dot{E}_6-\dot{E}_7$	----
Solution Pump	$\dot{W}_p$	$\dot{E}_{15}-\dot{E}_{14}$	----
Absorber	$\dot{E}_{11}+\dot{E}_{13}+\dot{E}_{17}$	$\dot{E}_{14}$	$\dot{E}_{26}-\dot{E}_{25}$
Pre-cooler-1	$\dot{E}_{31}-\dot{E}_{32}$	$\dot{E}_{30}-\dot{E}_{29}$	----
Pre-cooler-2	$\dot{E}_{33}-\dot{E}_{34}$	$\dot{E}_{21}-\dot{E}_{30}$	----
Overall System	$\dot{F}_G+\dot{F}_{sp}+\dot{F}_{pc1}+\dot{F}_{pc2}$	$\dot{P}_{CE}+\dot{P}_{pc1}+\dot{P}_{pc2}$	$\dot{E}_{24}-\dot{E}_{23}+\dot{E}_{26}-\dot{E}_{25}$

5.1.2.2 Exergetic Destruction, Loss and Efficiency

The exergy analysis of AAVAR system includes the calculations for the exergy destruction, $\dot{E}_D$, exergy destruction ratio, Y_D, the exergy loss, Y_L, the exergy loss ratio,

Y_D^* and the exergetic efficiency, ε for each productive components and the overall system and is carried out using Eqs. 4.4 to 4.8. Appendix B2 gives a sample calculation of the same. The calculated values of exergetic destruction, exergy loss and exergetic efficiency are listed for each productive components of AAVAR system in Table 5.5. The rate of exergy destruction in each of the components is pictorially represented in Fig. 5.2.

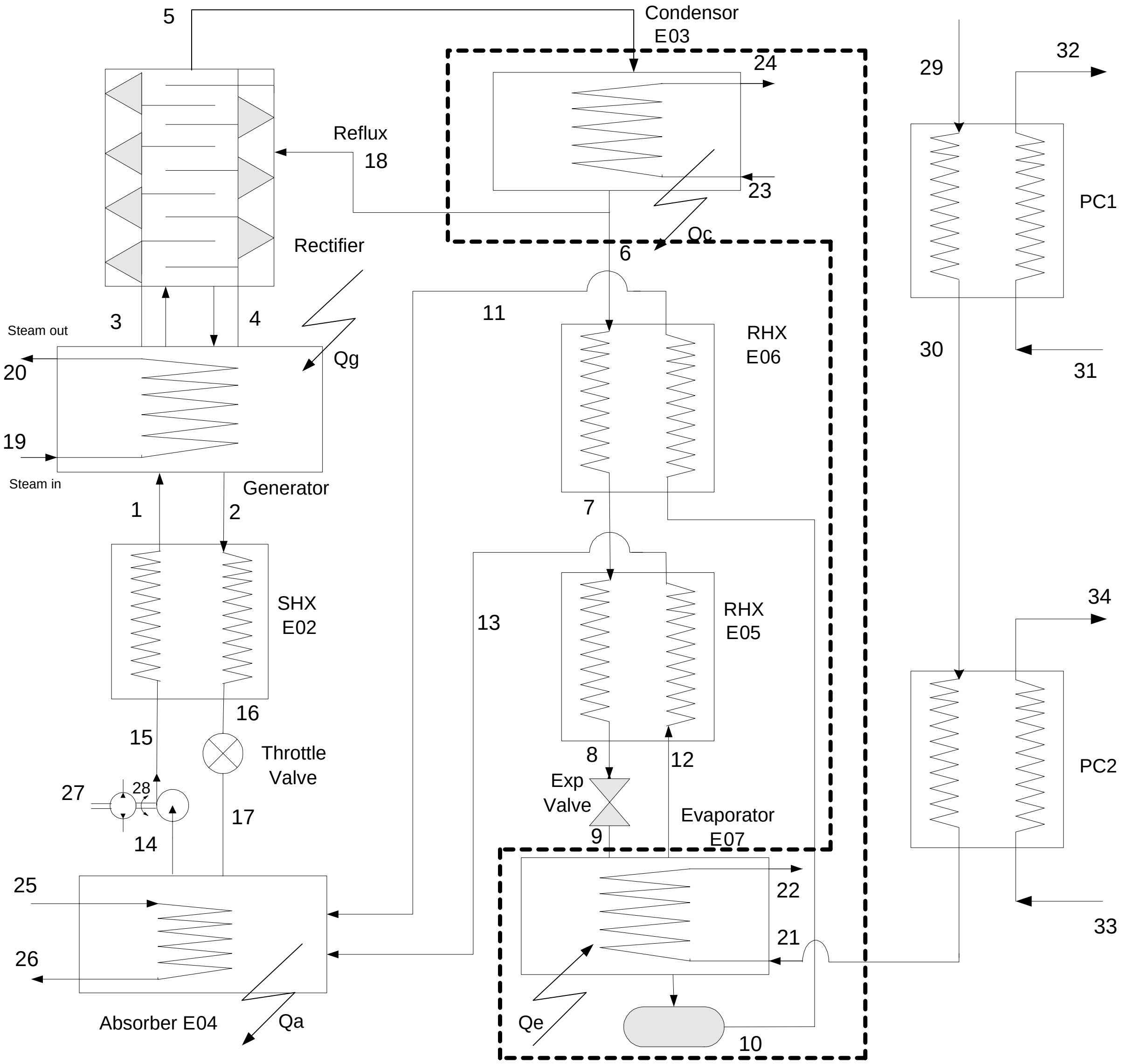


Fig. 5.1 Sub systems of AAVAR System for Exergoeconomic Optimization

5.1.2.3 Result and Discussion

Table 5.5 gives rate of exergy destruction, $\dot{E}_D$ and exergetic efficiency, ε for all the components of AAVAR system and Pre-cooler-1 and Pre-cooler-2 constituting the industrial brine chilling unit. It should be noted that Pre-cooler-1 and Pre-cooler-2 are excluded from components of AAVAR system and treated separately in exergy and exergoeconomic analysis.

Table 5.5 Exergetic Destruction, Loss and Efficiency Using EDM

Component	$\dot{E}_F$ **kW**	$\dot{E}_p$ **kW**	$\dot{E}_L$ **kW**	$\dot{E}_D$ **kW**	Y_D **%**	Y_L **%**	Y_D^* **%**	ε **%**
Generator	1640.00	1563.00	0	76.92	2.93	0	4.22	95.36
Rectifier	11536	11471	0	65.86	2.50	0	3.61	99.44
Cond.Evap.assly	784.30	226.90	114.50	443.71	16.96	4.38	24.35	28.93
SHX	1073.00	747.20	0	325.80	12.47	0	17.88	69.64
RHX 05	114.20	1.38	0	111.79	4.26	0	6.14	1.21
RHX 06	8.95	2.19	0	6.76	0.26	0	0.37	24.47
Solution pump	38.59	35.16	0	3.43	0.13	0	0.19	91.11
Absorber	101680	101437	163.60	78.68	3.01	6.26	4.32	99.76
Throttle Valve	---	---	---	26.63	1.00	---	1.46	---
Expansion Valve	---	---	---	30.76	1.18	---	1.68	---
Pre cooler-1	325.90	55.27	0	270.89	10.36	0	14.86	16.96
Pre cooler-2	608.70	228.20	0	380.80	14.57	0	20.90	37.49
Overall system	2613.19	510.37	278.10	1822.03	69.62	10.64	100	19.54

It is seen that 19.54 % of the exergy entering the system is converted to cooling effect which is the product of the system. The remaining exergy is either lost to the environment or destructed due to irreversibilities in the various components of the system. The rate of exergy destruction of the components of the system as compared with total fuel exergy input and net product (cooling produced) is given in Fig. 5.2 using Table 5.5. The total exergy supplied to the system is 2613.19 kW. Out of total exergy supplied as fuel, 19.54 % exergy is converted to useful product which is equivalent to 510.37 kW, 69.69 % exergy is destroyed which is equivalent to 1822.03 kW and remaining 10.64 % exergy is lost to environment which is equivalent to 278.1 kW.

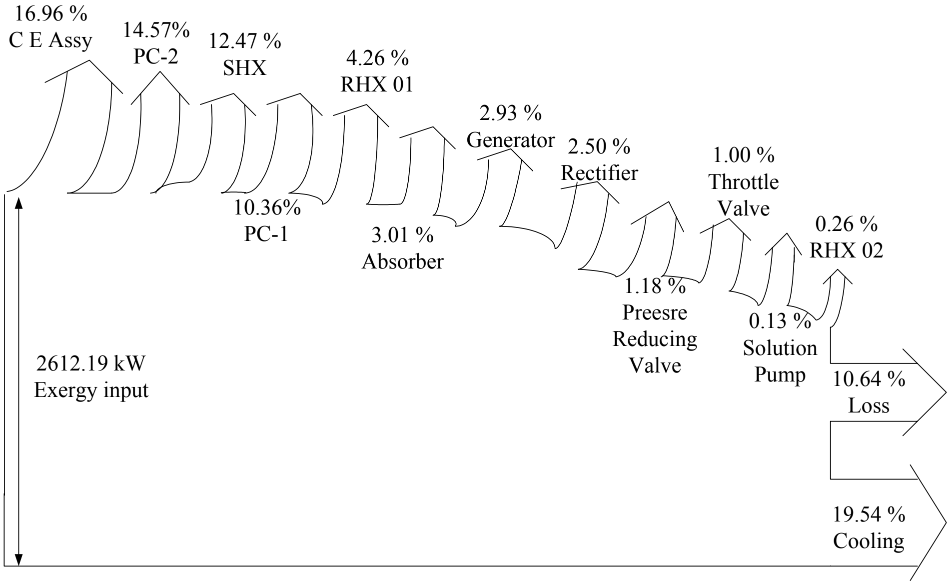


Fig. 5.2 Rate of Exergy Destruction of Various Components

The highest exergy destruction is found in condenser evaporator assembly. The total exergy destruction in this component is found to be 16.96 % of the total exergy supplied or 24.36 % of the total exergy destruction of the system. The reason for the highest exergy destruction may be attributed to the large temperature difference between the working fluid and brine in the evaporator and the working fluid and cooling water in the condenser.

The second highest exergy destruction is found to be in the pre-cooler-2 which amounts to be 380.5 kW and is equivalent to 20.93 % of the total exergy destruction and 14.57 % of the total exergy input. The effectiveness of pre-cooler-2 may be further increased by increasing the heat transfer area.

The third least efficient component in the system is solution heat exchanger having 69.64 % exergetic efficiency. This is due to the heat transfer across a high average temperature difference between the two unmixed streams in the heat exchanger. The improvement in the effectiveness of the solution heat exchanger may be possible by increasing heat transfer area.

The next highest exergy destruction is observed in the RHX 05 having the least exergetic efficiency of 1.21 % only. This heat exchanger is used to subcool the condensed refrigerant before entering in the evaporator. The high temperature difference between both the fluids and relatively very less mass flow rate of gaseous ammonia from the evaporator compared to the condensed ammonia from the condenser may be the reason for the poor exergetic efficiency. An increase in the convective heat transfer coefficient on gas side will improve the performance.

For RHX 06, the exergy destruction is less but the exergetic efficiency is quite low. The similar remedies can be used to improve the performance of this heat exchanger. The exergetic efficiencies of generator, rectifier, absorber and solution pump are 95.36 %, 99.44 %, 99.76 % and 91.11 %, respectively. It shows that the performance of these components is at the desired level. The expansion process in the expansion valve (V208) and pressure reducing process in throttle valve (V111) is irreversible and therefore, the exergy destruction in throttling valve and expansion valve is not avoidable.

The detailed exergy analysis of the large capacity industrial aqua ammonia vapour absorption refrigeration system presented here is well suited for finding the location, cause and true magnitude of the losses to be determined. This analysis enables for more effective utilization of energy resource and thereby having higher exergetic efficiency of the brine chilling unit using AAVAR system. The exergy analysis can now be extended for exergoeconomic analysis and exergoeconomic optimization which is a powerful tool to identify the entire cost source and designing the cost optimized AAVAR system

5.1.3 Entropy Generation Minimization (EGM) Method of Exergy Analysis

In this method of exergy analysis, the AAVAR system is divided in individual components (sub systems) considering them as an individual system. The entropy generation and the irreversibility defined based on Guoy-Stodola theorem at each of the sub system boundaries are calculated for all the components. Hence, the total irreversibility associated with the AAVAR system is estimated. The following are the component wise expressions for the entropy generation and irreversibility:

Generator

$$\dot{S}_g = \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_4 s_4 - \frac{Q_g}{T_{steam}} \tag{5.2}$$

$$I_g = T_0 \dot{S}_g \tag{5.3}$$

Rectifier

$$\dot{S}_r = \dot{m}_4 s_4 + \dot{m}_5 s_5 - \dot{m}_3 s_3 - \dot{m}_{18} s_{18} \tag{5.4}$$

$$I_r = T_0 \dot{S}_r \tag{5.5}$$

SHX

$$\dot{S}_{shx} = \dot{m}_1 s_1 + \dot{m}_{16} s_{16} - \dot{m}_2 s_2 - \dot{m}_{15} s_{15} \tag{5.6}$$

$$I_{shx} = T_0 \dot{S}_{shx} \tag{5.7}$$

Solution Pump

$$\dot{S}_{sp} = \dot{m}_{15} s_{15} - \dot{m}_{14} s_{14} \tag{5.8}$$

$$I_{sp} = T_0 \dot{S}_{sp} \tag{5.9}$$

Throttle Valve

$$\dot{S}_{tv} = \dot{m}_{17} s_{17} - \dot{m}_{16} s_{16} \tag{5.10}$$

$$I_{tv} = T_0 \dot{S}_{tv} \tag{5.11}$$

Absorber

$$\dot{S}_a = \dot{m}_{14} s_{14} - \dot{m}_{17} s_{17} - \dot{m}_{11} s_{11} - \dot{m}_{13} s_{13} + \frac{Q_a}{T_{atm}} \tag{5.12}$$

$$I_a = T_0 \dot{S}_a \tag{5.13}$$

Condenser

$$\dot{S}_c = \dot{m}_{18}\, s_{18} + \dot{m}_6\, s_6 - \dot{m}_5\, s_5 + \frac{Q_c}{T_{atm}} \tag{5.14}$$

$$I_c = T_0\, \dot{S}_c \tag{5.15}$$

RHX 06

$$\dot{S}_{rhx06} = \dot{m}_{11}\, s_{11} + \dot{m}_7\, s_7 - \dot{m}_6\, s_6 - \dot{m}_{10}\, s_{10} \tag{5.16}$$

$$I_{shx06} = T_0\, \dot{S}_{shx06} \tag{5.17}$$

RHX 05

$$\dot{S}_{rhx05} = \dot{m}_8\, s_8 + \dot{m}_{13}\, s_{13} - \dot{m}_7\, s_7 - \dot{m}_{12}\, s_{12} \tag{5.18}$$

$$I_{shx05} = T_0\, \dot{S}_{shx05} \tag{5.19}$$

Expansion Valve

$$\dot{S}_{ev} = \dot{m}_9\, s_9 - \dot{m}_8\, s_8 \tag{5.20}$$

$$I_{ev} = T_0\, \dot{S}_{ev} \tag{5.21}$$

Evaporator

$$\dot{S}_e = \dot{m}_{12}\, s_{12} + \dot{m}_{10}\, s_{10} - \dot{m}_9\, s_9 - \frac{Q_e}{T_{brine}} \tag{5.22}$$

$$I_e = T_0\, \dot{S}_e \tag{5.23}$$

Pre-cooler-1

$$\dot{S}_{pc1} = \dot{m}_{30}\, s_{30} + \dot{m}_{32}\, s_{32} - \dot{m}_{29}\, s_{29} - \dot{m}_{31}\, s_{31} \tag{5.24}$$

$$I_{pc1} = T_0\, \dot{S}_{pc1} \tag{5.25}$$

Pre-cooler-2

$$\dot{S}_{pc2} = \dot{m}_{21}\, s_{21} + \dot{m}_{34}\, s_{34} - \dot{m}_{30}\, s_{30} - \dot{m}_{33}\, s_{33} \tag{5.26}$$

$$I_{pc2} = T_0\, \dot{S}_{pc2} \tag{5.27}$$

Total System

$$\begin{aligned} \dot{S}_{tot} = {} & \dot{S}_{g} + \dot{S}_{r} + \dot{S}_{c} + \dot{S}_{ev} + \dot{S}_{shx} + \dot{S}_{rhx05} + \dot{S}_{rhx06} + \dot{S}_{sp} + \dot{S}_{ab} + \\ & \dot{S}_{ev} + \dot{S}_{tv} + \dot{S}_{pc1} + \dot{S}_{pc2} \end{aligned} \tag{5.28}$$

$$I_{tot} = T_0\, \dot{S}_{tot} \tag{5.29}$$

Eqs. 5.2 to 5.27 are used to estimate the entropy generation rate and irreversibility of processes in components. Eqs. 5.28 and 5.29 give, respectively total entropy generation rate and total irreversibility of the complete AAVAR system.

5.1.3.1 Results and Discussions

The outcome of the exergy analysis using EGM approach is given in Table 5.6. It is observed that the absorber is suffering from highest irreversibilities. This irreversibility may be due to irreversible mixing process of weak solution and ammonia vapour. The next highest irreversibility is observed in generator. It is recommended to increase the generator temperature. The components like condenser, evaporator, SHX, RHX05, RHX06, pre-cooler-1 and pre-cooler-2 are considered as simple heat exchanger and their irreversibilities can be reduced by improving their effectiveness. Using EGM method, the components like throttle valve and expansion valve can be analyzed, while EDM method cannot analyze components like expansion valve and throttle valve because fuel, product and loss cannot be defined for these components. The same problem is experienced for the components like condenser where product cannot be identified. Therefore, such components are analyzed by combining them with other components as an assembly where fuel, product and loss can be defined using EDM approach. For example, the combination of condenser and evaporator as a sub system helps in defining fuel, product and loss which in turn is used for exergoeconomic analysis. Considering the irreversibilities in throttle valve and pressure reducing valve equivalent to exergy destruction and combined with EDM.

Table 5.6 Results of EGM Approach of Exergy Analysis

Component	EGM $I = T_0\Delta S_g$, kW	$I_R = \frac{I}{I_t}\%$
Generator	458.6	20.32
Rectifier	84.16	3.73
Condenser	265.50	5.60
Evaporator	200.50	8.88
SHX	325.80	14.43
RHX 05	19.19	0.85
RHX 06	11.14	0.49
Solution Pump	3.43	0.15
Absorber	533.10	23.62
Expansion Valve	30.76	1.36
Throttle Valve	26.04	1.15
Pre-cooler-1	149.10	6.61
Pre-cooler-2	288.80	12.80
Overall System	2257.12	100.00

From the above comparative study of two approaches of exergy analysis, viz. EGM and EDM, it can be concluded that the approach of EGM is quite convenient and useful to the extent of only up to exergy analysis. However, it is found difficult to combine it with economic analysis. Therefore, EDM method is followed in the present exergoeconomic analysis and optimization of AAVAR system.

5.2 Exergoeconomic Analysis

The essence of the economic analysis is the identification and inclusion of various cost heads incurred in the estimation of the total cost for the production. In the present case, the total cost involved in the cooling operation of brine consists of many cost heads. Thus, in general, the economic analysis of the system requires the estimation of levelized O & M cost of component $(\dot{Z}_k)$ and fuel cost rate $(\dot{C}_f)$. $\dot{Z}_k$ should be estimated for each component for brine chilling unit using TCI, β, γ and τ (Refer Eq.4.18). The fuel cost rate ($\dot{C}_f$) is governed by the source of heat energy used for the system. The estimation of $\dot{Z}_k$ and $\dot{C}_f$ are explained in the following section.

5.2.1 Levelized O & M cost ($\dot{Z}_k$)

To estimate $\dot{Z}_k$, TCI for each component should be estimated. The major components of the system such as generator, condenser, evaporator, absorber, SHX, RHX05 and RHX06 are considered as simple heat exchangers for the purpose of estimation of TCI. The costs of these heat exchangers are calculated based on weighted area method suggested by Peters et al. [158]. It should be noted that the data used by them are for the year 1990. Therefore, these costs are brought to the year 2009 with the help of M & S cost index as given in Section 4.2.2.3. The estimation of levelized O & M cost, ($\dot{Z}_k$) for various components are given below:

Generator

Generator used in the AAVAR is a 1-2 pass shell and tube heat exchanger with steam flowing through the tube made up of carbon steel and the strong aqua ammonia solution flowing through the shell with total heat transfer area of 517.4 m^2 (5570 ft^2). Fig. 5.3 gives purchased equipment cost for 1-2 shell and tube type heat exchanger and the equipment cost of the generator for the year 1990 is ₹ 1715000 ($35000). Using Table 4.1, the converted equipment cost for the year 2009 is estimated to be ₹ 19010000. Then, using Eq. 4.18, the levelized O&M cost for generator is found to be 278 ₹/hr as explained in Appendix C.

Rectifier

Rectifier is a packed tower made up of carbon steel similar to a cooling tower in construction. Fig. 5.4 gives the equipment cost of various tower constructions with respect to their diameter and height. The AAVAR system has the rectifier with diameter and height of 1.5 m (59 in) and 5 m (16.5 ft).

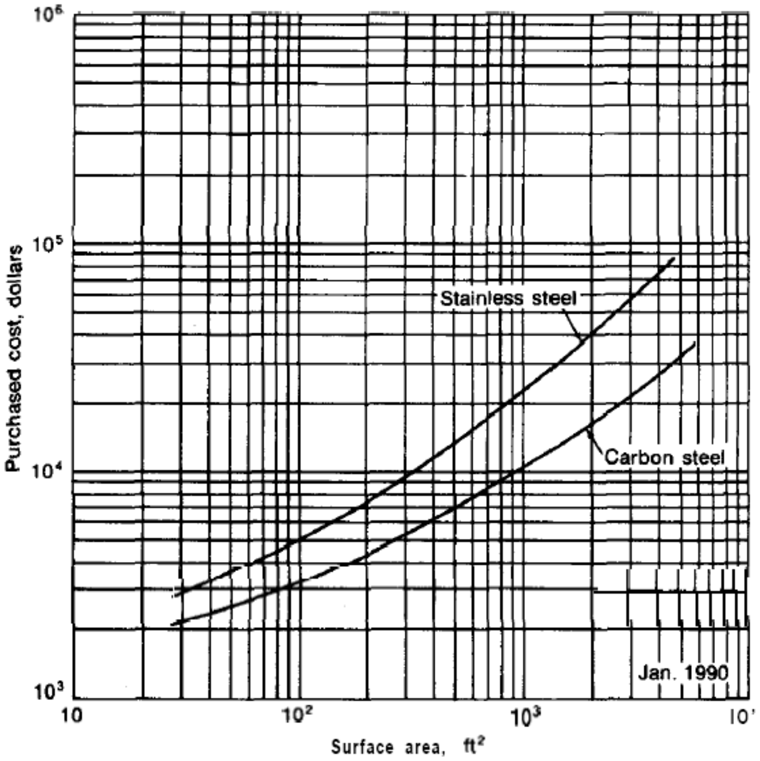


Fig. 5.3 Cost of U-Tube Type Shell & Tube Heat Exchanger [158]

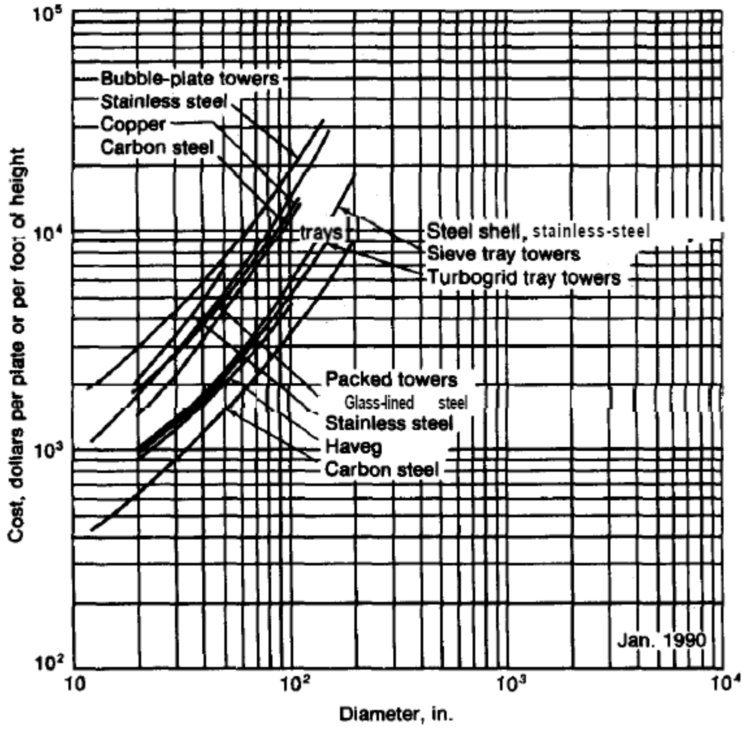


Fig. 5.4 Cost of Tower Including Installation [158]

Using Fig. 5.4, the purchase equipment cost for the year 1990 is found to be ₹ 1455300 (\$29700). The equivalent cost for the year 2009 and the levelized O&M cost of rectifier are estimated, respectively at ₹ 2356000 and 235.9 ₹/hr.

Condenser

Condenser of AAVAR system is 1-2 pass shell and tube type heat exchanger of carbon steel material having cooling water flowing through tube side and ammonia vapour condensing on the shell side with total heat transfer area of 615.3 m^2 (6624 ft^2). Using Fig. 5.3, the cost for the year 1990 is found to be ₹ 1862000 (\$38000). The equivalent cost for the year 2009 ₹ 2976636 the levelized O&M cost for condenser is found to be 301.9 ₹/hr.

Refrigerant Heat Exchanger (RHX 05)

RHX 05 is a shell and tube type heat exchanger with single shell and single tube pass having carbon steel tube with total heat transfer area of 273.1 m^2 (2940 ft^2) . Using Fig.5.5, the PEC for the year 1990 is ₹ 1127000 (\$23000). The equivalent cost for the year 2009 will be ₹ 1801648. Therefore, the levelized O&M cost for RHX 05 is found to be 182.7 ₹/hr.

Refrigerant Heat Exchanger (RHX 06)

RHX 06 is a finned tube type heat exchanger with ammonia vapour condensing on the shell side and cooling water flowing through the tube. The tubes are 18 feet long and made up of carbon steel with 1.25 in. square pitch arrangement with total heat transfer area of 146.2 m^2 (1574 ft^2). Using Fig. 5.6, the PEC for the year 1990 is ₹ 1225000 (\$25000) and the equivalent cost for the year 2009 is ₹ 1958313. Therefore, the levelized O&M cost for RHX 06 is found to be 198.5 ₹/hr.

Evaporator

Evaporator is a 1-2 pass shell and tube type heat exchanger with brine flowing through the tube and ammonia vaporizes in the shell with total heat transfer area of 1226 m^2 (13195 ft^2). Fig.5.3 gives the PEC with respect to the heat transfer area and carbon steel as tube material. The PEC for the year 1990 is ₹ 3822000 (\$78000). The levelized O&M cost estimated for the evaporator is found to be 619.7 ₹/hr.

Absorber

Absorber is also a 1-2 pass shell and tube heat exchanger with cooling water flowing through the tube. Ammonia vapour from evaporator enters from side and weak

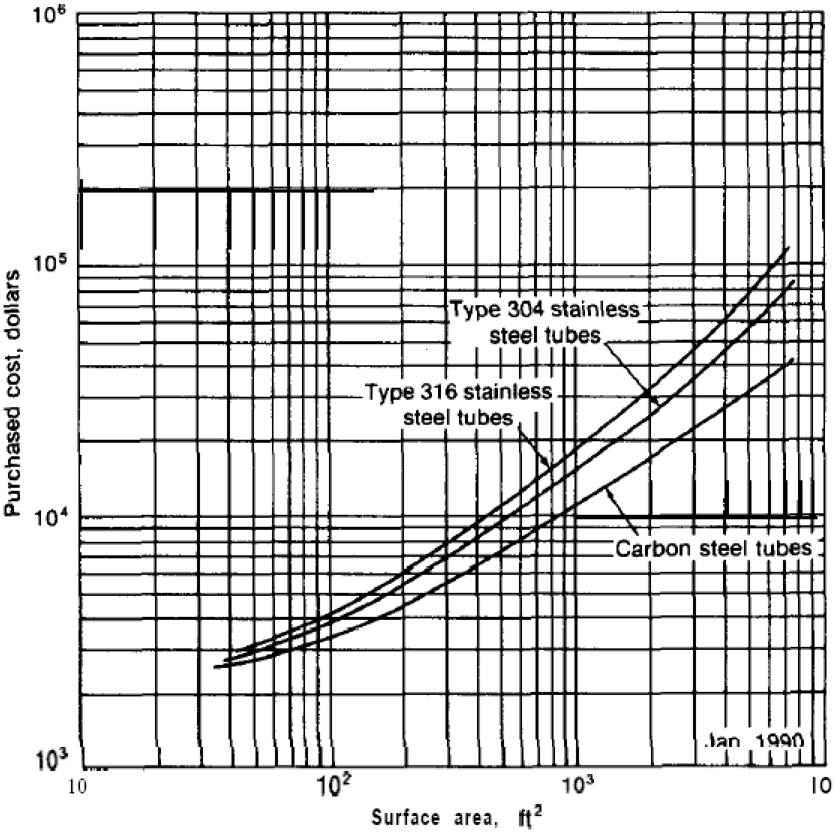


Fig. 5.5 Cost of Fixed Tube Sheet Type Heat Exchanger [158]

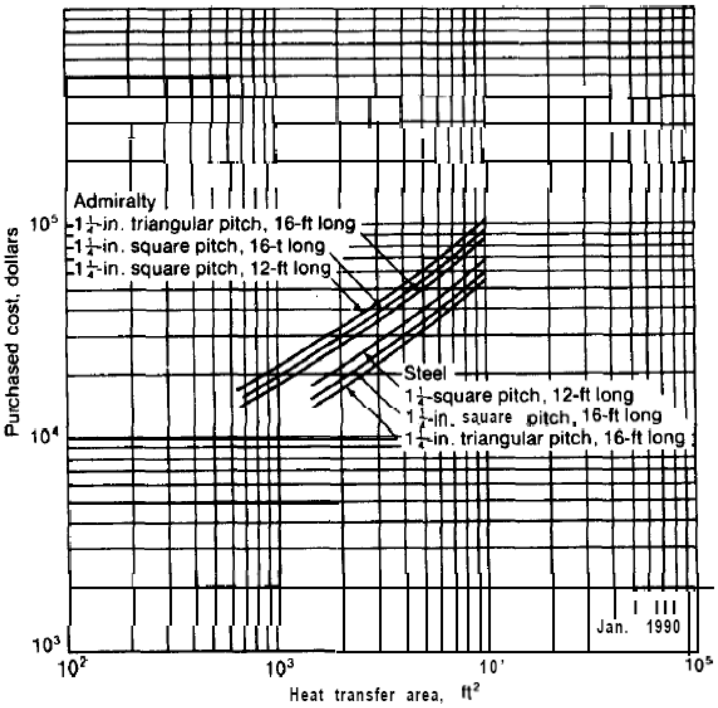


Fig. 5.6 Cost of Finned Tube Heat Exchanger with 1 in. Tube, 150 psi [158]

solution sprayed from the top in to the shell. The total heat transfer area in the absorber is 1772 m^2 (19072 ft^2). Using Fig.5.3, the PEC corresponding to the year 1990 is ₹ 5390000 ($110000) and the levelized O&M cost for absorber for carbon steel as tube material is found to be 874 ₹/hr.

Pump and Motor

Eqs. 5.30 and 5.31 give the relations for the estimation of equipment cost for pump and motor, respectively as suggested by Dentice d'Accadia [103]. The data provided along with the relation is for the year 1997. The M&S swift cost index for the year 1997 can be used for the estimation of the equipment cost for the year 2009.

$$Z_p = Z_{0p}\left(\frac{P_p}{P_{0p}}\right)^{m_p}\left(\frac{\eta_p}{1-\eta_p}\right)^{\eta_p} \tag{5.30}$$

Where Z_{0p} = $800, P_{0p} = 10 kW, m_p = 0.26, η_p = 0.5, P_p = 41.14 kW

$$Z_m = Z_{0m}\left(\frac{P_m}{P_{0m}}\right)^{m}\left(\frac{\eta_m}{1-\eta_m}\right) \tag{5.31}$$

Where Z_{0m} = $150, P_{0m} = 10 kW, m = 0.67, η_m = 0.8, P_m = 45.71 kW

The M&S swift cost index for the year 1997 is 1056.8. Using the above relations and data, the total levelized O&M cost for pump and motor $\dot{Z}_{pm}$ is found to be 19.38 ₹/hr.

Solution Heat Exchanger

SHX in AAVAR system is a double pipe heat exchanger having many hairpins made up of carbon steel with total heat transfer area of 1255 m^2 (13504 ft^2). Fig. 5.7 gives purchased equipment cost based on the heat transfer area for double pipe heat exchanger. Using the chart, the purchased equipment cost is ₹ 1125000 ($22959) in the year 1990. This cost is to be converted for the year 2009 for the purpose of the present analysis. The M&S cost index for the year 1990 was 915.1 while for the year 2009, it was 1462.9. Therefore, using Eq. 4.16, the purchased equipment cost for the year 2009 is estimated to be ₹ 1798000. Similarly, using Table 4.1, the other related costs of SHX are calculated and the TCI is estimated to be ₹ 12460000. Finally, using Eq. 4.18, the levelized O&M cost for SHX is found to be 182.2 ₹/hr.

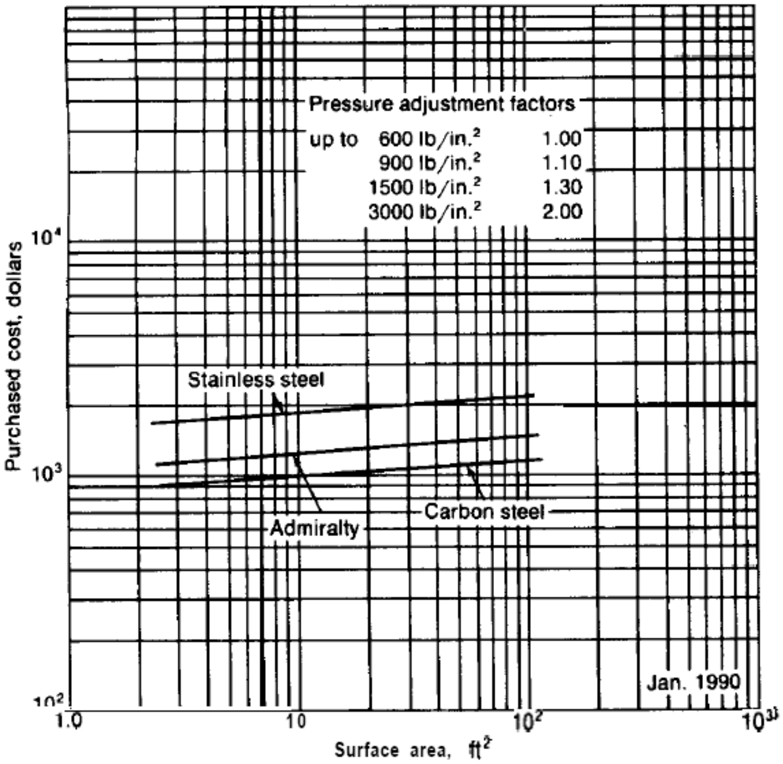


Fig. 5.7 Cost of Double Pipe Heat Exchanger [158]

Pre-cooler-1

Pre-cooler-1 is a fixed tube sheet type shell & tube heat exchanger having carbon steel as tube material. Brine flowing through the tube while ammonia vapour evaporates in the shell side with the total heat transfer area is at 81.43 m^2 (876.5 ft^2). Using Fig. 5.5, the PEC for the year 1990 is ₹ 49000 (\$10000) and the levelized O&M cost is found to be 79.44 ₹/hr.

Pre-cooler-2

Pre-cooler-2 is also a similar type of shell & tube heat exchanger with carbon steel as tube material. Brine flowing through the tube while ammonia vapour evaporates in the shell with the total heat transfer area is 248.80 m^2 (2678 ft^2). Using Fig. 5.5, the PEC for the year 1990 is ₹ 1029000 (\$21000) and the levelized O&M cost is found to be 166.7 ₹/hr.

The estimated values of $\dot{Z}_k$ for all the components of AAVAR including the pre-coolers 1 and 2 are given in Table 5.7. It can be seen that for each component of the

AAVAR system along with pre-coolers 1 and 2, a number of cost heads are involved in the estimation of TCI. TCI consists of FCI and Other Outlays. DC and IC constitute FCI, while Other Outlays consists of start up cost, working capital cost and allowance for funds. DC consists of on-site (ONSC) and off-site (OFSC) costs while IC consists of engineering & supervision, construction and contingency costs.

Table 5.7 Estimation of Levelized O & M Cost

Component	**PEC ₹ (1)**	**Installation Cost 45 % of PEC (2)**	**Piping Cost 66 % of PEC (3)**	**Instru. & Control cost 20 % of PEC (4)**	**Electrical Equipment 11 % of PEC (5)**	**On Site Cost (ONSC) 1+2+3+4+5**	**Land 10 % of PEC (6)**	**Civil Work 60 % of PEC (7)**	**Service 65 % of PEC (8)**	**Of Site Cost (OFSC) (6+7+8)**	**Direct Cost (DC) ONSC + ONFC**
Generator	2741639	1233738	1809482	548328	301580	6634767	274164	1644983	1782065	3701212	10335979
Rectifier	2326476	1046914	1535474	465295	255912	5630071	232648	1395886	1512209	3140743	8770814
Condenser	2976636	1339486	1964580	595327	327430	7203459	297664	1785982	1934813	4018459	11221918
Evaporator	6109937	2749472	4032558	1221987	672093	14786047	610994	3665962	3971459	8248415	23034462
SHX	1797732	808979	1186503	359546	197751	4350511	179773	1078639	1168526	2426938	6777449
RHX05	1801648	810742	1189088	360330	198181	4359989	180165	1080989	1171071	2432225	6792214
RHX06	1958313	881241	1292487	391663	215414	4739118	195831	1174988	1272903	2643722	7382840
Sol. Pump	191044	85970	126089	38209	21015	462327	19104	114626	124179	257909	720236
Absorber	8616579	3877461	5686942	1723316	947824	20852122	861658	5169947	5600776	11632381	32484503
Pre-cooler-1	783325	352496	516995	156665	86166	1895647	78333	469995	509161	1057489	2953136
Pre-cooler-2	1644983	740242	1085689	328997	180948	3980859	164498	986990	1069239	2220727	6201586

Continue

Table 5.7 Continue

Engg. & Supervision 30% of PEC (9)	**Construction Cost 15 % of DC (10)**	**Contingency 20 % of FCI (11)**	**Indirect Cost (IC) (9+10+11)**	**Fixed Capital Investment (DC+IC)**	**Startup Cost 10% of FCI (12)**	**Working Capital 15 % of TCI (13)**	**Allowance For Funds 10% of PEC (14)**	**Other Outlays (12+13+14)**	**TCI (FCI+Other Outlays)**	**Levelized O&M Cost ₹/hr ($\dot{Z}_k$)**
822492	1550397	1733027	4105916	14441895	1444189	2851809	274164	4570162	19012057	278
697943	1315622	1470597	3484162	12254976	1225498	2419963	232648	3878109	16133085	235.9
892991	1683288	1881572	4457851	15679769	1567977	3096249	297664	4961890	20641659	301.9
1832981	3455169	3862174	9150324	32184786	3218479	6355458	610994	10184931	42369717	619.7
539320	1016617	1136371	2692308	9469757	946976	1869972	179773	2996721	12466478	182.2
540494	1018832	1138846	2698172	9490386	949039	1874045	180165	3003249	12493635	182.7
587494	1107426	1237876	2932796	10315636	1031564	2037006	195831	3264401	13580037	198.5
57313	108035	120761	286109	1006345	100634	198721	19104	318459	1324804	19.38
2584974	4872675	5446657	12904306	45388809	4538881	8962826	861658	14363365	59752174	874
234998	442970	495150	1173118	4126254	412625	814802	78333	1305760	5432014	79.44
493495	930238	1039816	2463549	8665135	866513	1711085	164498	2742096	11407231	166.7

5.2.2 Fuel Cost

As mentioned earlier, the exergoeconomic analysis considers the steam used as the source of heat energy in AAVAR system as fuel. The cost of the fuel (steam) is taken as 322 ₹/1000 kg for the year 1990 as suggested by Peters et al. [158]. The fuel cost so obtained is to be updated to the processing year 2009. It can be carried out with the help of the economic term escalation rate (r_n) using Eq. 4.19. Thus, the fuel cost i.e., the cost rate associated with fuel (Steam) for the year 2009 is found to be 974 ₹ /1000 kg steam. However, from the cost data available from the fertilizer plant at GNSFC, Bharuch, Gujarat, the actual cost of steam for the generation of saturated steam at 15 bar is 900 ₹/1000 kg steam so the cost of steam considered is 0.9 ₹/kg steam. Similarly, the actual cost of generation of electricity from the captive power plant at GNSFC is 4 ₹/kWh, while the cost of purchased electricity from Gujarat Electricity Board is 6 ₹/kWh. It should be noted that all the above data are based on processing year 2009.

5.2.3 Cost flow

In Chapter 4, the principles for formulation of cost balance equations are explained (Refer Section 4.2.1). Applying the formulation of cost balance equations and the definition of fuel, product and loss (Refer Table 5.4); the exergoeconomic cost balance equations for each component of AAVAR system are formulated in the following forms:

Generator

As stated earlier, the purpose of the generator is to separate the refrigerant (ammonia vapour) from absorbent (water vapour) (stream 3, which is the stream passing through station 3) and the separated weak aqua ammonia solution returns back to absorber (stream 2). This is achieved by supplying ($\dot{E}_{19}-\dot{E}_{20}$) exergy of steam. The cost rate associated with steam ($\dot{C}_s$) is considered to be ₹ 900/1000 kg steam. The stream 2 and 3 are the product while stream 1 and 4 are the fuel for the generator.

$$c_1\dot{E}_1+c_4\dot{E}_4-c_2\dot{E}_2-c_3\dot{E}_3+\dot{C}_s+\dot{Z}_g=0 \qquad (5.32)$$

$$\frac{\dot{c}_3\dot{E}_3-(\dot{c}_1\dot{E}_1+\dot{c}_4\dot{E}_4)}{\dot{E}_3-(\dot{E}_1+\dot{E}_4)}=\frac{\dot{c}_2\dot{E}_2-(\dot{c}_1\dot{E}_1+\dot{c}_4\dot{E}_4)}{\dot{E}_2-(\dot{E}_1+\dot{E}_4)} \tag{5.33}$$

Here $\dot{c}$ is the unit exergy cost rate in terms of ₹/kJ

Rectifier

The purpose of the rectifier is to separate the unwanted water vapour from the ammonia vapour leaving the generator and going to the condenser (stream 5) by cooling it. The cooling of ammonia vapour which contains traces of water vapour is carried out by reflux (stream 18). Reflux is a portion of the condensed liquid refrigerant from the condenser flowing through RHX 05 and RHX 06 to evaporator. It is used to condense the traces of water vapour thus increase the quality of ammonia vapour going to the condenser. This results in the increase of the exergy of ammonia vapour coming out from rectifier. Stream 4 and 5 are the product while streams 3 and 18 are the fuel for rectifier.

$$\dot{c}_3\dot{E}_3+\dot{c}_{18}\dot{E}_{18}-\dot{c}_5\dot{E}_5-\dot{c}_4\dot{E}_4+\dot{Z}_r=0 \tag{5.34}$$

$$\frac{\dot{c}_5\dot{E}_5-(\dot{c}_3\dot{E}_3+\dot{c}_{18}\dot{E}_{18})}{\dot{E}_5-(\dot{E}_3+\dot{E}_{18})}=\frac{\dot{c}_4\dot{E}_4-(\dot{c}_3\dot{E}_3+\dot{c}_{18}\dot{E}_{18})}{\dot{E}_4-(\dot{E}_3+\dot{E}_{18})} \tag{5.35}$$

SHX

Exergoeconomic analysis point of view, SHX transfers the exergy of the weak aqua ammonia solution (stream 2) coming from the generator to the strong aqua ammonia solution going to the generator (stream 1). Stream 2 acts as a fuel in SHX and unit exergy cost for stream 2 remains same at inlet and outlet of SHX as there is no exergy addition in between.

$$\dot{c}_{15}\dot{E}_{15}-\dot{c}_1\dot{E}_1+\dot{c}_2\dot{E}_2-\dot{c}_{16}\dot{E}_{16}+\dot{Z}_{shx}=0 \tag{5.36}$$

$$\dot{c}_2=\dot{c}_{16} \tag{5.37}$$

RHX 06

The portion of the un-evaporated liquid ammonia from the evaporator is to be evaporated before being allowed to enter the absorber. In RHX 06, the evaporation is achieved by using the condensed refrigerant (liquid ammonia) flowing to the absorber. Therefore, the exergy of liquid ammonia (refrigerant) leaving the condenser (stream 6) is transferred to the un-evaporated ammonia (refrigerant) flowing from the evaporator to the absorber.

$$\dot{c}_6 \dot{E}_6 - \dot{c}_7 \dot{E}_7 + \dot{c}_{10} \dot{E}_{10} - \dot{c}_{11} \dot{E}_{11} + \dot{Z}_{rhx06} = 0 \tag{5.38}$$

$$\dot{c}_{10} = \dot{c}_{11} \tag{5.39}$$

RHX 05

In RHX 05, the exergy of liquid ammonia (refrigerant), leaving RHX 06 (stream 7) is transferred to the vapour ammonia (refrigerant) (stream 12) leaving the evaporator.

$$\dot{c}_7 \dot{E}_7 - \dot{c}_8 \dot{E}_8 + \dot{c}_{12} \dot{E}_{12} - \dot{c}_{13} \dot{E}_{13} + \dot{Z}_{rhx05} = 0 \tag{5.40}$$

$$\dot{c}_{12} = \dot{c}_{13} \tag{5.41}$$

Condenser

The high pressure ammonia vapour (refrigerant) from the rectifier (stream 5) is condensed in the condenser using cooling water circulated through the cooling tower as the cooling medium.

$$\dot{c}_5 \dot{E}_5 - \dot{c}_6 \dot{E}_6 - \dot{c}_{18} \dot{E}_{18} - \dot{C}_c + \dot{Z}_c = 0 \tag{5.42}$$

$$\dot{c}_5 = \dot{c}_6 \tag{5.43}$$

$$\dot{c}_{18} = \dot{c}_6 \tag{5.44}$$

Where, $\dot{C}_c$ is the cost rate associated with exergy loss from the condenser.

Absorber

The vapour ammonia from the evaporator flowing through RHX 05 and the evaporated vapour ammonia from RHX 06 are both absorbed by the weak aqua ammonia solution from the generator.

$$\dot{c}_{17} \dot{E}_{17} - \dot{c}_{14} \dot{E}_{14} + \dot{c}_{13} \dot{E}_{13} + \dot{c}_{11} \dot{E}_{11} - \dot{C}_a + \dot{Z}_a = 0 \tag{5.45}$$

Pressure Reducing Valve

Across the pressure reducing valve, no exergy transfer takes place so unit exergy cost remains same

$$\dot{c}_{16} = \dot{c}_{17} \tag{5.46}$$

$$\frac{\dot{c}_{14}\,\dot{E}_{14}}{\dot{E}_{14}} = \frac{\dot{c}_{17}\,\dot{E}_{17} + \dot{c}_{13}\,\dot{E}_{13} + \dot{c}_{11}\,\dot{E}_{11}}{\dot{E}_{17} + \dot{E}_{13} + \dot{E}_{11}} \tag{5.47}$$

Evaporator

$$\dot{c}_{9}\,\dot{E}_{9} - \dot{c}_{12}\,\dot{E}_{12} - \dot{c}_{10}\,\dot{E}_{10} - \dot{C}_{p} + \dot{Z}_{e} = 0 \tag{5.48}$$

$$\frac{\dot{c}_{12}\,\dot{E}_{12} - \dot{c}_{9}\,\dot{E}_{9}}{\dot{E}_{12} - \dot{E}_{9}} = \frac{\dot{c}_{10}\,\dot{E}_{10} - \dot{c}_{9}\,\dot{E}_{9}}{\dot{E}_{10} - \dot{E}_{9}} \tag{5.49}$$

Pump and Motor

$$\dot{c}_{14}\,\dot{E}_{14} - \dot{c}_{15}\,\dot{E}_{15} + \dot{C}_{m} + \dot{Z}_{pm} = 0 \tag{5.50}$$

$\dot{C}_{m}$ is the cost rate of electricity in ₹/sec

Condenser - Evaporator Combined

The evaporator and condenser (heat dissipative component) are combined in to single unit for the purpose of cost flow analysis as discussed earlier. The cost rate associated with the capital investment for the condenser-evaporator combined and cost rates associated with the exergy losses are accounted to the final product.

$$\dot{c}_{5}\,\dot{E}_{5} - \dot{c}_{18}\,\dot{E}_{18} - \dot{c}_{6}\,\dot{E}_{6} + \dot{c}_{8}\,\dot{E}_{8} - \dot{c}_{12}\,\dot{E}_{12} - \dot{c}_{10}\,\dot{E}_{10} - \dot{C}_{p} - \dot{C}_{c} + \dot{Z}_{EA} = 0 \tag{5.51}$$

$$\dot{Z}_{EA} = \dot{Z}_{c} + \dot{Z}_{e}$$

AAVAR System

Using the cost rate associated with each components of AAVAR system excluding the pre-coolers-1 and 2, the total levelized O & M cost, Z_{tot} can be estimated. Then for the overall system, the following relation can be formulated

$$\dot{\mathrm{C}}_{s} + \dot{C}_{m} - \dot{\mathrm{C}}_{c} - \dot{\mathrm{C}}_{a} - \dot{\mathrm{C}}_{p} + \dot{Z}_{tot} = 0$$

$$\dot{Z}_{tot} = \dot{Z}_{g} + \dot{Z}_{r} + \dot{Z}_{shx} + \dot{Z}_{06} + \dot{Z}_{05} + \dot{Z}_{c} + \dot{Z}_{a} + \dot{Z}_{pm} + \dot{Z}_{e} \tag{5.52}$$

Pre-cooler-1

$$\dot{c}_{29}\,\dot{E}_{29} - \dot{c}_{30}\,\dot{E}_{30} + \dot{c}_{31}\,\dot{E}_{31} - \dot{c}_{32}\,\dot{E}_{32} + \dot{Z}_{pc1} = 0 \tag{5.53}$$

$$\dot{c}_{29} = \dot{c}_{30} \tag{5.54}$$

$$\dot{c}_{31} = \dot{c}_{32} \tag{5.55}$$

Pre-cooler-2

$$\dot{c}_{30}\,\dot{E}_{30} - \dot{c}_{21}\,\dot{E}_{21} + \dot{c}_{33}\,\dot{E}_{33} - \dot{c}_{34}\,\dot{E}_{34} + \dot{Z}_{pc2} = 0 \tag{5.56}$$

$$\dot{c}_{21} = \dot{c}_{30} \tag{5.57}$$

$$\dot{c}_{33} = \dot{c}_{34} \tag{5.58}$$

Out of these variables $\dot{c}_1 ... \dot{c}_{18}, \dot{c}_{29} ... \dot{c}_{34}, \dot{C}_p, \dot{C}_a, \dot{C}_c, \dot{C}_s$ and $\dot{C}_m$, the last two are the cost of fuel in generator and electricity for solution pump, respectively. Under the normal operation of the AAVAR system, both of them are known data. The remaining 27 are calculated by solving Eqs. 5.32 to 5.58 using EES software. The cost per unit exergy, c (₹/kJ) and cost flow rate, $\dot{C}$ (₹/sec) for each stream of the system are calculated and given in Table 5.8.

After calculating the cost of product at evaporator (cooling), the cost of exergy flows related to pre-cooler-1 and pre-cooler-2 ($\dot{c}_{29}$ to $\dot{c}_{34}$) are calculated separately using known values of $\dot{E}_1 ... \dot{E}_{19}$ and $\dot{E}_{29} ... \dot{E}_{34}$. The unit exergy flows associated with streams 1 to 34 are given in Table 5.8.

5.2.4 Exergoeconomic Evaluation

AAVAR system can now be exergoeconomically evaluated through exergoeconomic parameters, viz. fuel cost per unit exergy ($c_{F,k}$), product cost per unit exergy ($c_{p,k}$), exergetic destruction cost rate ($\dot{C}_{D,k}$), exergetic cost rate associated with loss ($\dot{C}_{L,k}$), relative cost difference (r_k),exergoeconomic factor (f_k) and exergetic efficiency (ε_k). Based on the methodology suggested by Bejan et al. [155] and discussed in Section 4.2.3 of Chapter 4, the above parameters are estimated using Eqs. 4.20 to 4.27 and given

Table 5.8 Unit Exergy Cost and Cost Flow Rate

Flows	Unit exergy cost c, ₹ /kJ	Exergy flow E, kW	Cost flow rate $\dot{C}$, ₹/sec
1	0.002949	102220	301.4
2	0.002934	50806	149.1
3	0.002936	53329	156.6
4	0.003655	352.4	1.288
5	0.002936	64800	190.2
6	0.002936	52646	154.6
7	0.002937	52644	154.6
8	0.002945	52645	155
9	0.002948	52614	155.1
10	0.002918	4652	13.57
11	0.002918	4643	13.55
12	0.002948	47444	139.8
13	0.002948	47330	139.5
14	0.002939	101437	298.2
15	0.002940	101473	298.2
16	0.002934	49733	145.9
17	0.002934	49707	145.8
18	0.002936	11888	34.9
19-20	----	----	2.83
22-21	----	----	1.85
24-23	----	----	0.87
26-25	----	----	0.96
28-27	----	----	0.05
29	0.008172	0.06	0.0005
30	0.008172	55.33	0.45
31	0.001318	52218	68.81
32	0.001318	51892	68.38
33	0.002986	58155	173.60
34	0.002986	57546	171.8

in Table 5.9. The evaluation of exergoeconomic parameters for generator is explained in Appendix D.

It is obvious that higher values of r_k and $\dot{C}_{D,k}$ in a given component indicates a poor performance both energy utilization and economic points of view. Therefore, more attention should be paid to this component during optimization. Pre-cooler-1 and pre-cooler-2 are independent components and the variation in the controlling parameter of both the pre-coolers does not affect directly the performance of AAVAR system. Therefore, the optimization of both the pre-coolers can be carried out independently.

Table 5.9 Parameters of Exergoeconomic Evaluation

Component	$c_{F,k}$ ₹/MJ	$c_{p,k}$ ₹/MJ	$\dot{C}_{D,k}$ ₹/hr	$\dot{C}_{L,k}$ ₹/hr	$\dot{Z}_k$ ₹/hr	f_k %	r_k %	ε %
Generator	1.72	1.86	474.30	0	278	36.96	7.82	95.36
Rectifier	2.92	2.94	683.90	0	235.90	25.65	0.75	99.44
Cond Evap Assembly	3.12	8.15	4979	1287	921.60	12.82	161.20	28.93
SHX	2.93	4.17	3441	0	182.20	5.028	42.18	69.64
RHX 05	2.95	424.10	1182	0	182.70	13.38	14286	1.21
RHX 06	2.92	23.39	71.01	0	198.50	73.65	701.40	24.47
Sol Pump	1.18	5.76	14.57	0	19.38	57.09	388	91.11
Absorber	2.94	2.94	832.70	1731	874	25.42	0.03	99.76
Pre-cooler-1	1.32	8.17	1284	0	79.44	5.83	520	16.96
Pre-cooler-2	2.97	8.17	4090	0	166.70	3.92	173.70	37.49
VAR System	1.71	8.15	7182	1710	2892	24.54	377.4	13.52
Overall System	1.96	8.16	12820	1960	3138	17.51	316.90	19.54

5.2.4.1 Results and Discussions

A comparison of the r value, one of the parameters of exergoeconomic evaluation of AAVAR system, given in Table 5.9, shows that the r value for RHX 05 is highest among the entire components. Therefore, attention should be paid to this component. RHX 05 has the lowest exergetic efficiency among all components and possesses one of the higher level exergy destruction. Therefore, improvement of exergetic efficiency of RHX 05 should be considered at the cost of capital investment.

RHX 06 is having second highest r value with low exergetic efficiency and low exergy destruction. It shows that there is a scope for improvement in the exergetic efficiency of the VAR system by increasing the effectiveness of RHX 06. Solution pump is the next candidate, having high value of r and f which shows that the investment cost of the pump can be reduced at the cost of exergetic efficiency.

The next component is condenser-evaporator assembly having higher value of r and lower value of f due to high exergy destruction and low exergetic efficiency. Therefore the exergetic efficiency of condenser evaporator assembly should be improved at the cost of capital investment by increasing the heat exchanger area. Exergy destruction at evaporator can be reduced by increasing the ammonia temperature at evaporator inlet and the evaporator pressure. The next component in that order is solution heat exchanger (SHX). It is having low value of f due to very high value of exergy destruction. It is suggested that the exergetic efficiency of the SHX should be improved through increase in its effectiveness.

The component having next highest r value is generator. It is having high f value so higher investment cost. Further, generator possesses slight potential for reducing exergy destruction cost by increasing its temperature. The next one is the rectifier, where there is no one direct decision variable controlling the performance but depends on generator and condenser temperatures. Therefore, the effect of generator and condenser temperature on the rectifier performance is observed during global optimization. The next component in the order is absorber which has very less value and high exergetic efficiency. So this component is working properly.

Both the pre-coolers are having very high r value and hence high exergy destruction cost. By increasing the heat transfer area, the exergy destruction should be reduced and exergetic efficiency should be improved. It is interesting to note that pre-cooler-1 has half the exergetic efficiency as compared to that of pre-cooler-2. The reason for such a large difference in exergetic efficiency is the difference in exergy of product in both pre-coolers. In pre-cooler-1, the cooling of brine is carried out up to 15.9ºC with exergy flow equal to 55.33 kW. While in pre-cooler-2, the brine is cooled up to 5.40ºC with exergy flow 283.50 kW as given in Table 5.3. So pre-cooler-2 works with very high exergy level, therefore having high exergetic efficiency.

5.3 Exergoeconomic Optimization

The exergoeconomic optimization of the system requires a thermodynamic model and a cost model. The thermodynamic model gives the performance prediction of the system with respect to some thermodynamic variables such as exergy destruction, exergy loss and exergetic efficiency. The cost model permits detailed calculation of cost values for a given set of the thermodynamic variables. For each component, it is expected that the investment cost increases with increasing capacity and increasing exergetic efficiency.

5.3.1 Estimation of B_k, n_k and m_k

After evaluation of parameters of exergoeconomic evaluation for each component, the exergoeconomic optimization of the system is carried out at component level using the method suggested in Chapter 4, Section 4.3. To solve Eq. 4.29 for local optimum, the parameters B_k, n_k and m_k are to be evaluated through curve fitting technique as explained for the following cases

Generator

In order to determine the local optimum for exergoeconomic optimization of generator as an individual component of AAVAR, the parameters B_k, n_k and m_k are to be evaluated through curve fitting technique. For this purpose, generator temperature t_G is considered as decision variable. The generator temperature t_G is varied from 142°C to 152°C, and necessary parameters are estimated using the method discussed in Chapter4, Section 4.3 and is given in Table 5.10.

It can be seen from the power law ($y = Bx^n$) in Fig. 5.8 that the values of B_G and n_G are 226231 and 0.048, respectively {Section 4.3 using Eq.4.29}

Table 5.10 Generated Data Using Investment Cost Equation for Generator

T_G °C	$\dot{E}_{PG}$ kW	$\dot{E}_{D,G}$ kW	$TCI_G / \dot{E}_{p,G}^{0.66}$	$\dot{E}_{P,G} / \dot{E}_{D,G}$
142	1634	28818	197134	0.05670
143	1642	28834	197196	0.05694
144	1650	28851	197250	0.05718
145	1658	28867	197300	0.05742
146	1666	28884	197346	0.05766
147	1674	28901	197389	0.05791
148	1682	28917	197429	0.05816
149	1690	28934	197460	0.05841
150	1698	28950	197492	0.05866
151	1707	28967	197517	0.05892
152	1715	28983	197539	0.05917

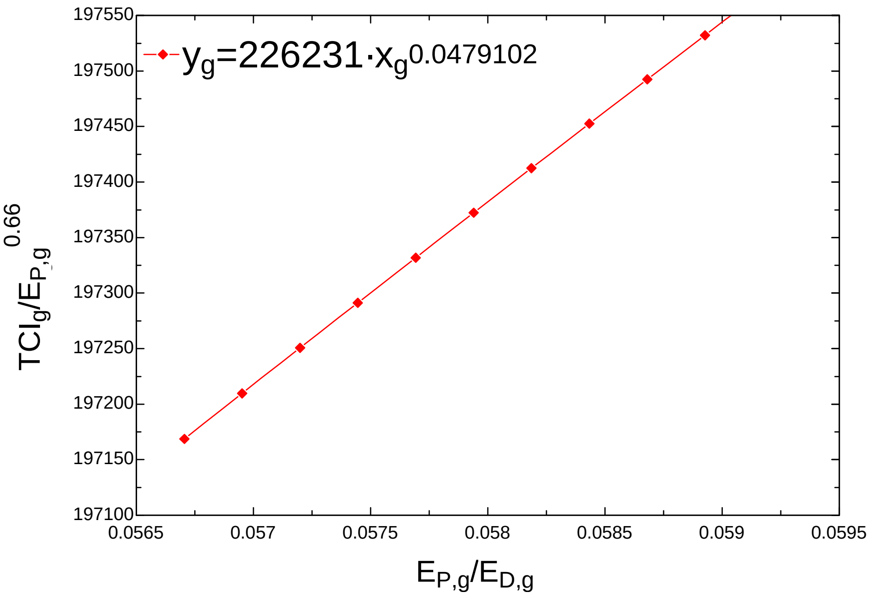


Fig. 5.8 Plot of TCI v/s Exergetic Efficiency for Generator

Condenser- Evaporator (C-E) Assembly

For condenser-evaporator assembly, evaporator temperature is the decision variable. With change in evaporator temperature, the temperature at which, chilling effect produced will change. This will change mean temperature at evaporator and heat loss at condenser which will increase the cost of condenser-evaporator assembly. Considering evaporator temperature as decision variable at different evaporator temperature, and using investment cost equation, Table 5.11 is obtained. In the optimization process for the condenser-evaporator assembly, condenser temperature and absorber temperature also can be considered as decision variables and can be the same. Therefore, analysis is carried out by considering condenser and absorber temperatures at three different temperature levels at 36°C, 38°C and 40°C. Tables 5.11 to 5.13 give the generated data for a range of evaporator temperature from - 20°C to -15°C with the condenser and absorber temperatures at 36°C, 38°C and 40°C, respectively.

Table 5.11 Generated Data Using Investment Cost Equation for CE Assembly (Ta = Tc = 36°C)

T_{evap} °C	$\dot{E}_{p,ce}$ kW	$\dot{E}_{D,ce}$ kW	$I_{ce} / \dot{E}_{p,ce}^{0.66}$	$\dot{E}_{P,ce} / \dot{E}_{D,ce}$
-20	226.9	1156	1.53E+06	0.1963
-19	226.9	1130	1.53E+06	0.2008
-18	226.9	1105	1.53E+06	0.2054
-17	226.9	1079	1.53E+06	0.2102
-16	226.9	1054	1.53E+06	0.2152
-15	226.9	1030	1.53E+06	0.2204

Table 5.12 Generated Data Using Investment Cost Equation for CE assembly (Ta = Tc = 38°C)

T_{evap} °C	$\dot{E}_{p,ce}$ kW	$\dot{E}_{D,ce}$ kW	$I_{ce} / \dot{E}_{p,ce}^{0.66}$	$\dot{E}_{P,ce} / \dot{E}_{D,ce}$
-20	226.9	1134	1.53E+06	0.2002
-19	226.9	1108	1.53E+06	0.2048
-18	226.9	1083	1.53E+06	0.2096
-17	226.9	1057	1.53E+06	0.2146
-16	226.9	1032	1.53E+06	0.2198
-15	226.9	1008	1.53E+06	0.2252

Table 5.13 Generated Data Using Investment Cost Equation for CE assembly (Ta = Tc = 40°C)

T_{evap}	$\dot{E}_{p,ce}$	$\dot{E}_{D,ce}$	$I_{ce}/\dot{E}_{p,ce}^{0.66}$	$\dot{E}_{P,ce}/\dot{E}_{D,ce}$
-20	226.9	1112	1.53E+06	0.2041
-19	226.9	1086	1.53E+06	0.2089
-18	226.9	1061	1.53E+06	0.2139
-17	226.9	1036	1.53E+06	0.2191
-16	226.9	1011	1.53E+06	0.2245
-15	226.9	985.9	1.53E+06	0.2301

In order to determine the local optimum for exergoeconomic optimization of condenser-evaporator together as an individual component of AAVAR, the parameters B_k, n_k and m_k are to be evaluated through curve fitting technique which is carried out using Figs. 5.9 to 5.11. Fig. 5.9 gives the value of $B_{ce} = 1.54\times10^6$ and $n_{ce} = 0.059$ when T_a = T_c = 36°C. The value of $B_{ce} = 1.54\times10^6$ and $n_{ce} = 0.058$ are obtained through regression fit using Fig. 5.10 when T_a = T_c = 38°C. Fig. 5.11 gives the value of $B_{ce} = 1.54\times10^6$ and $n_{ce} = 0.057$ when the condenser and absorber temperatures are same at 40°C.

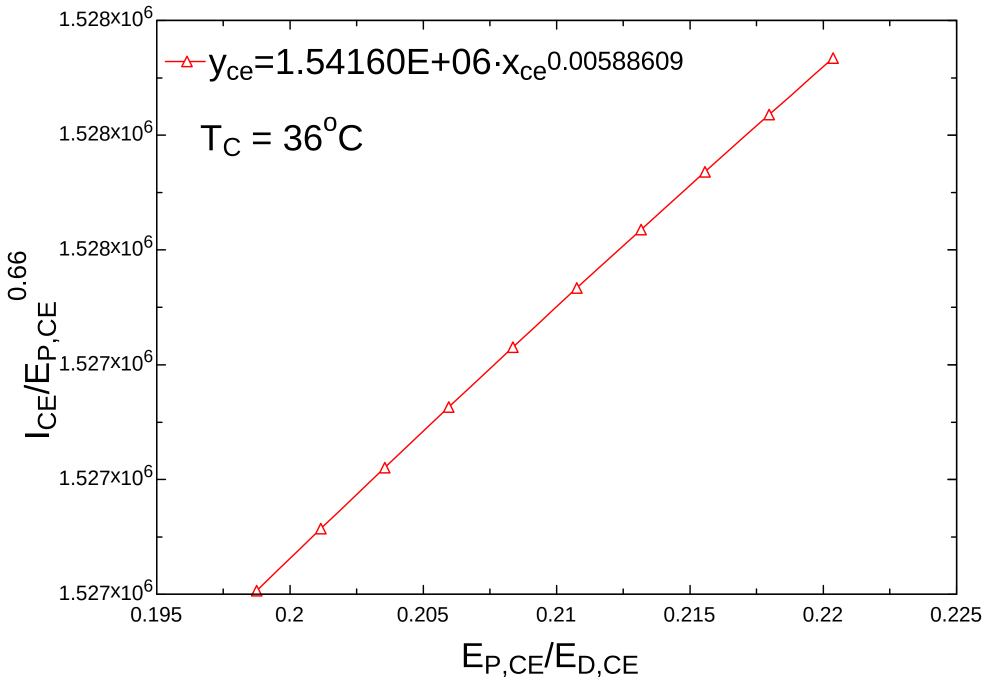


Fig. 5.9 Plot of TCI v/s Exergetic Efficiency for C-E Assembly for Ta = Tc = 36°C

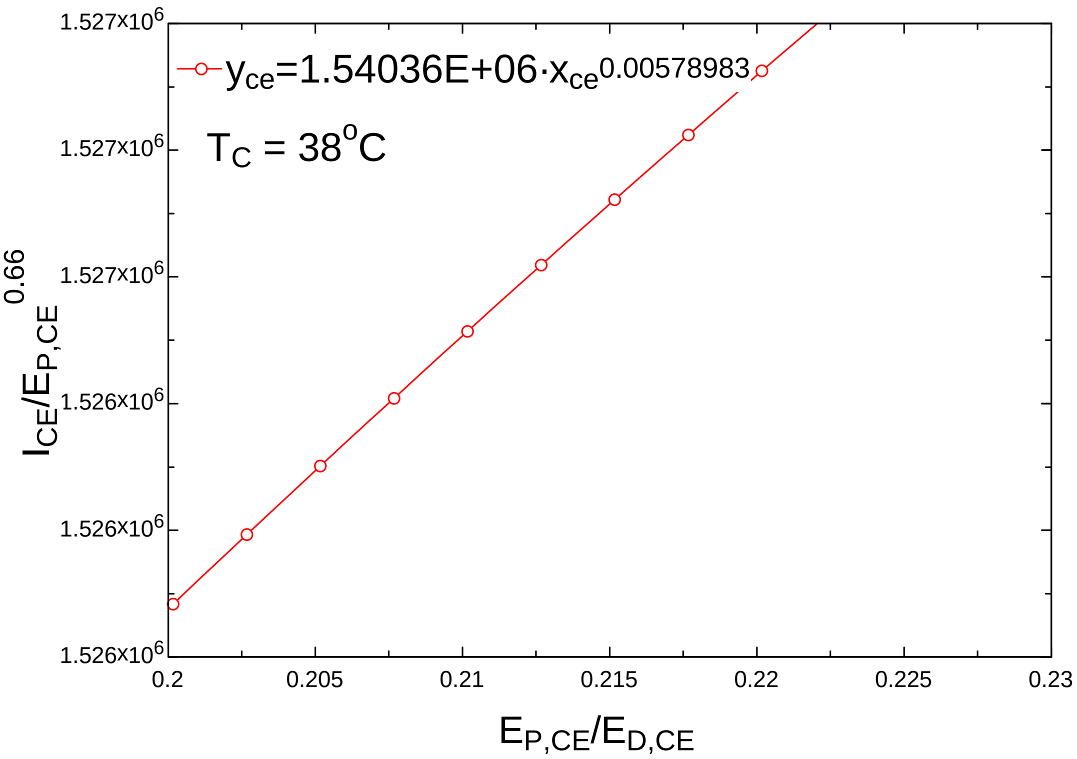


Fig. 5.10 Plot of TCI v/s Exergetic Efficiency for C-E Assembly for Ta = Tc = 38°C

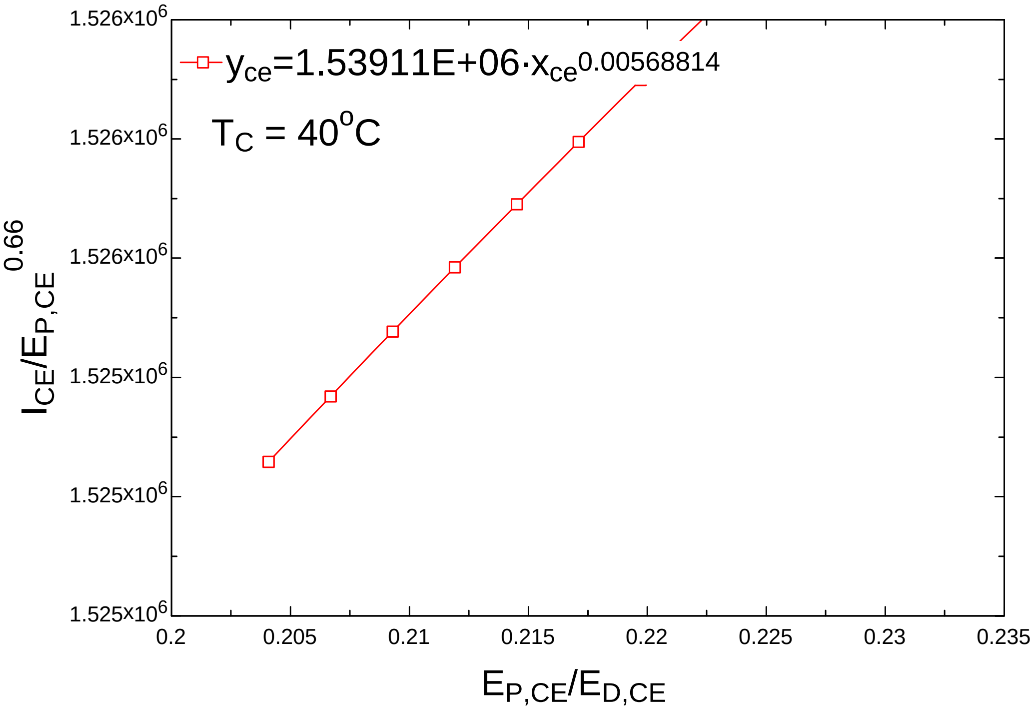


Fig. 5.11 Plot of TCI v/s Exergetic Efficiency for C-E Assembly for $T_a = T_c$ = 40°C

Table 5.14 Generated Data Using Investment Cost Equation for SHX

χ_{shx}	$\dot{E}_{p,shx}$ kW	$\dot{E}_{D,shx}$ kW	$I_{shx}/\dot{E}_{p,shx}^{0.16}$	$\dot{E}_{P,shx}/\dot{E}_{D,shx}$
0.75	676.00	354.70	4510	1.91
0.77	704.30	342.10	4902	2.06
0.79	733.00	328.40	5349	2.23
0.81	762.20	313.60	5865	2.43
0.83	791.90	297.50	6468	2.66
0.85	822.00	280.20	7185	2.93
0.87	852.60	261.70	8052	3.26
0.89	883.50	242.10	9128	3.65
0.91	914.90	221.20	10508	4.14
0.93	946.80	199.10	12361	4.76

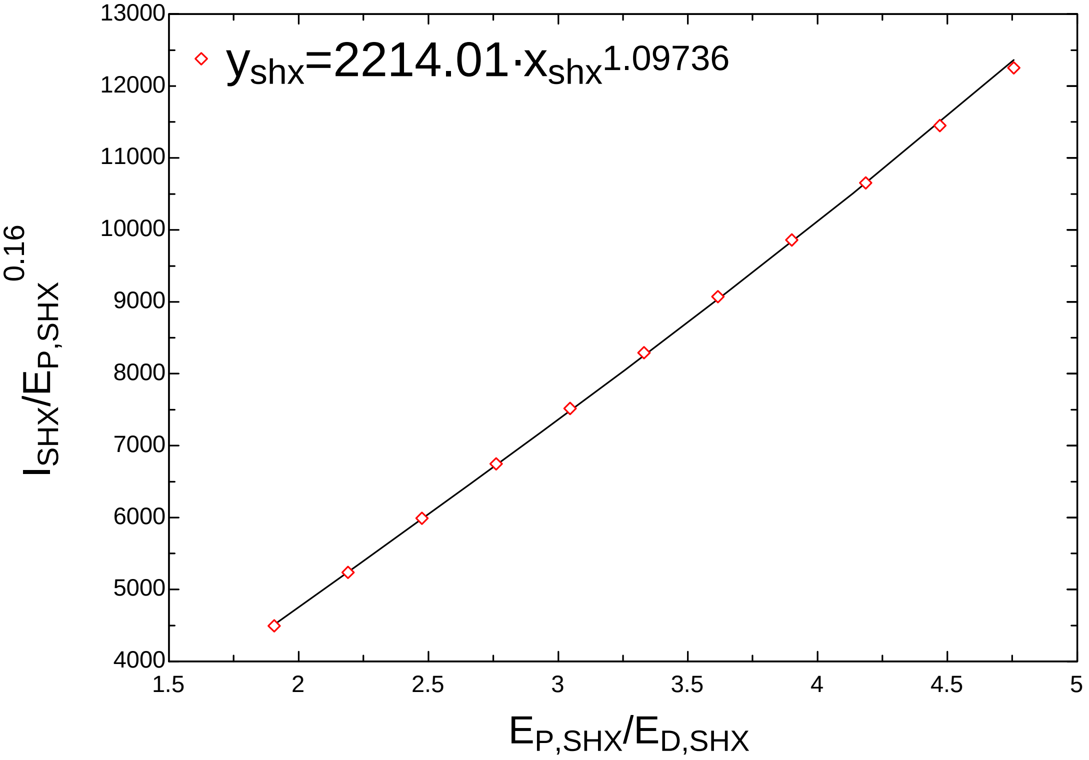


Fig. 5.12 Plot of TCI v/s Exergetic Efficiency for SHX

SHX

As the purpose of SHX is to recover the waste energy from weak solution and to heat strong solution going to generator, the effectiveness of heat exchanger is considered as decision variable. By improving the effectiveness, the recovered heat will increase and exergy destruction will decrease. The values of m_{shx} is equal to 0.16 equivalent to scaling exponent α as SHX is a double pipe heat exchanger [155]. Table 5.14 gives the generated data for regression to obtain B_{shx} and n_{shx}. From Fig. 5.12, the values of B_{shx} and n_{shx} are found to be 2214 and 1.097, respectively.

RHX 05

As stated earlier, RHX 05 is a shell and tube type heat exchanger with single shell and single tube pass. The effectiveness of heat exchanger is considered as decision variable. The value of $m_{RHX\,05}$ is to be taken as 0.66. Table 5.15 gives the generated data and Fig.5.13 gives the values of $B_{RHX\,05}$ = 602445 and $n_{RHX\,05}$ = 0.267

Table 5.15 Generated Data Using Investment Cost Equation for RHX 05

χ_{RHX05}	$\dot{E}_{p,RHX\,05}$ kW	$\dot{E}_{D,RHX\,05}$ kW	$I_{RHX\,05} / \dot{E}_{p,RHX\,05}^{0.66}$	$\dot{E}_{P,RHX\,05} / \dot{E}_{D,RHX\,05}$
0.490	0.9730	112.3	172449	0.008667
0.487	0.8886	112.6	162841	0.007894
0.484	0.8052	112.9	158933	0.007134
0.481	0.7227	113.2	158546	0.006387

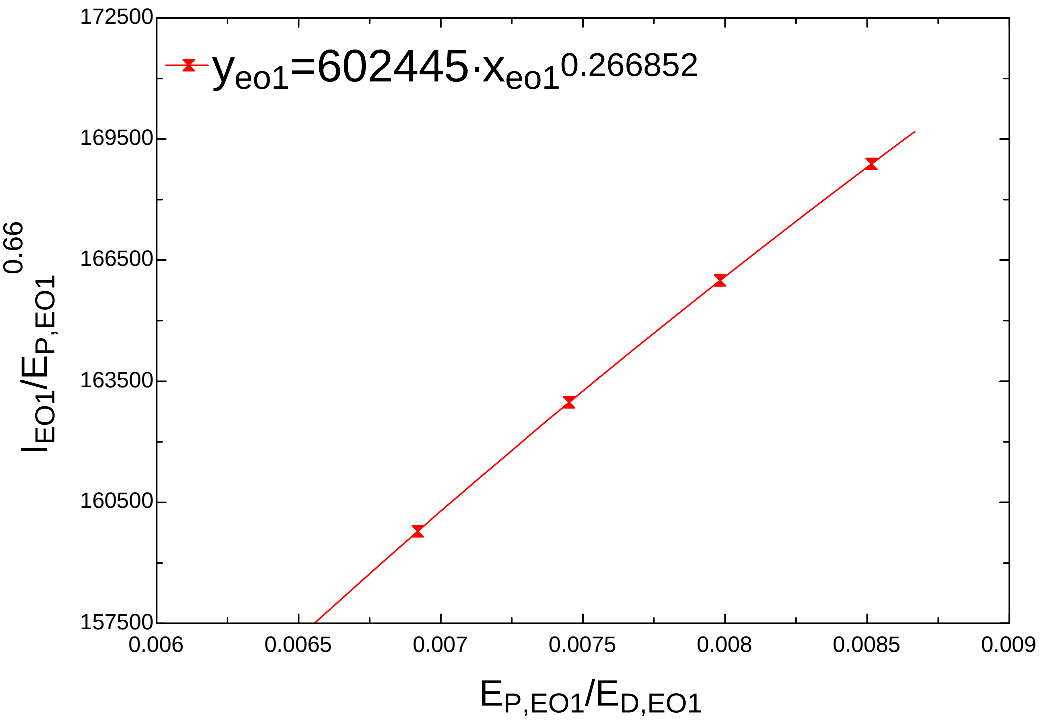


Fig. 5.13 Plot of TCI v/s Exergetic Efficiency for RHX 05

RHX 06

It is a finned tube type heat exchanger. The effectiveness of heat exchanger is considered as decision variable. The value of $m_{RHX\,06}$ is to be taken as 0.66. Using Table 5.16 and Fig. 5.14, the values of $B_{RHX\,06}$ and $n_{RHX\,06}$ are estimated as 79102 and 0.697, respectively.

Table 5.16 Generated Data Using Investment Cost Equation for RHX 06

χ_{RHX06}	$\dot{E}_{p,RHX\,06}$ kW	$\dot{E}_{D,RHX\,06}$ kW	$I_{RHX\,06}/\dot{E}_{P,RHX\,06}^{0.8}$	$\dot{E}_{P,RHX\,06}/\dot{E}_{D,RHX\,06}$
0.50	1.451	4.163	38124	0.3485
0.54	1.555	4.502	37723	0.3454
0.58	1.657	4.857	37368	0.3411
0.62	1.757	5.196	37054	0.3382
0.66	1.856	5.536	36775	0.3352
0.70	1.953	5.892	36527	0.3314
0.74	2.048	6.232	36306	0.3286
0.78	2.141	6.588	36109	0.3249
0.82	2.232	6.944	35934	0.3214
0.86	2.322	7.286	35779	0.3187

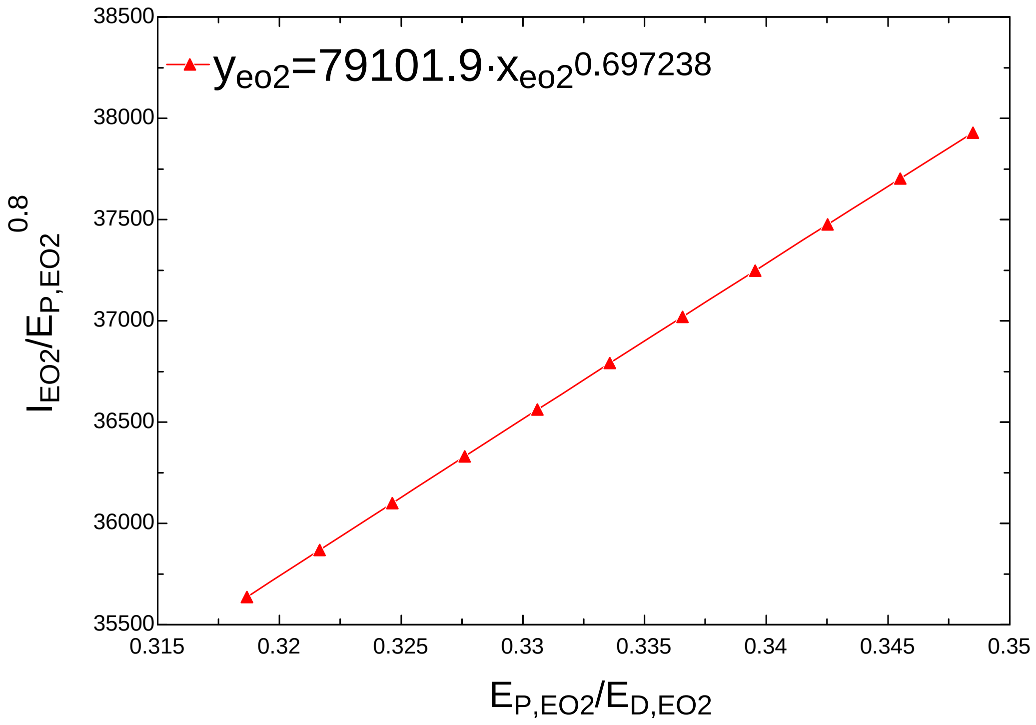


Fig. 5.14 Plot of TCI v/s Exergetic Efficiency for RHX 06

Solution Pump

The efficiency of solution pump is considered as decision variable. For centrifugal pump and motor assembly, the value of m_P is taken as 0.48 [155]. Using Table 5.17 and Fig. 5.15, the values of B_P and n_P are found to be 6462 and 1.003, respectively.

Table 5.17: Generated Data Through Investment Cost Equation for Pump Motor Assembly

η_P	$\dot{E}_{p,P}$ kW	$\dot{E}_{D,P}$ kW	$I_P/\dot{E}_{p,P}^{0.48}$	$\dot{E}_{P,P}/\dot{E}_{D,P}$
0.70	37.99	14.84	16836	2.56
0.72	37.92	13.44	18381	2.82
0.74	37.85	12.12	20210	3.12
0.76	37.78	10.87	22399	3.46
0.78	37.72	9.69	25053	3.90
0.80	37.67	8.56	28319	4.40
0.82	37.61	7.48	32414	5.03
0.84	37.56	6.46	37666	5.81
0.86	37.51	5.49	44595	6.84
0.88	37.46	4.56	54078	8.22

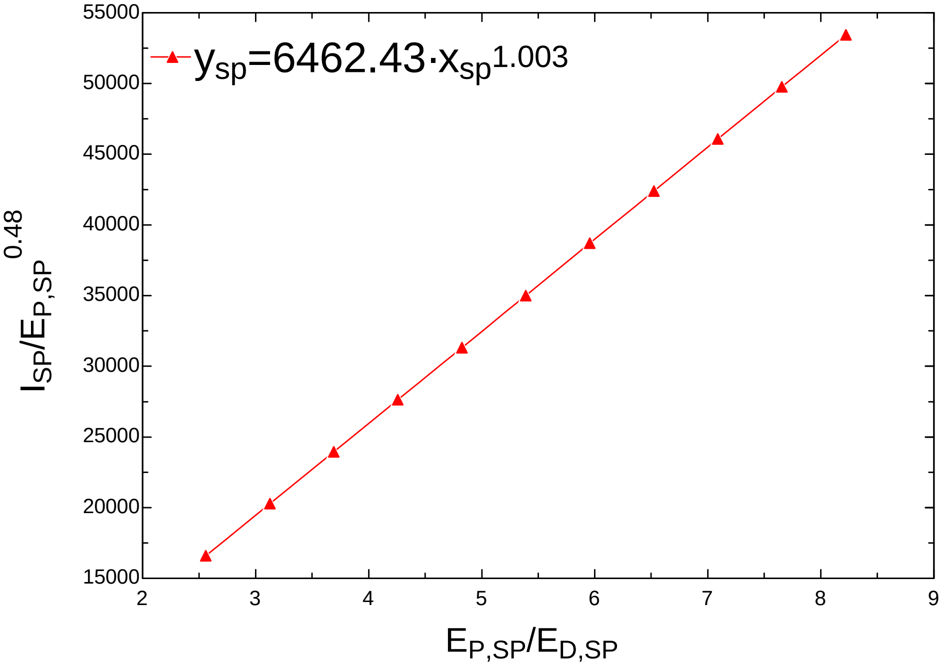


Fig. 5.15 Plot of TCI v/s Exergetic Efficiency for Solution Pump Motor Assembly

Absorber

The absorber temperature is considered as the decision variable and the value of m_a is taken as 0.66. Using Table 5.18 and Fig. 5.16, the values of B_a and n_a are found to be 16053 and 0.071, respectively.

Table 5.18 Generated Data Using Investment Cost Equation for Absorber

T_a °C	$\dot{E}_{p,a}$ kW	$\dot{E}_{D,a}$ kW	$I_a / \dot{E}_{p,a}^{0.66}$	$\dot{E}_{P,a} / \dot{E}_{D,a}$
40	101437	79.76	26584	1272
39	101434	77.76	26655	1304
38	101430	75.64	26726	1341
37	101427	73.4	26797	1382
36	101424	71.02	26868	1428
35	101422	68.52	26939	1480
34	101419	65.88	27010	1539
33	101417	63.11	27080	1607
32	101415	60.21	27150	1684
31	101414	57.17	27220	1774

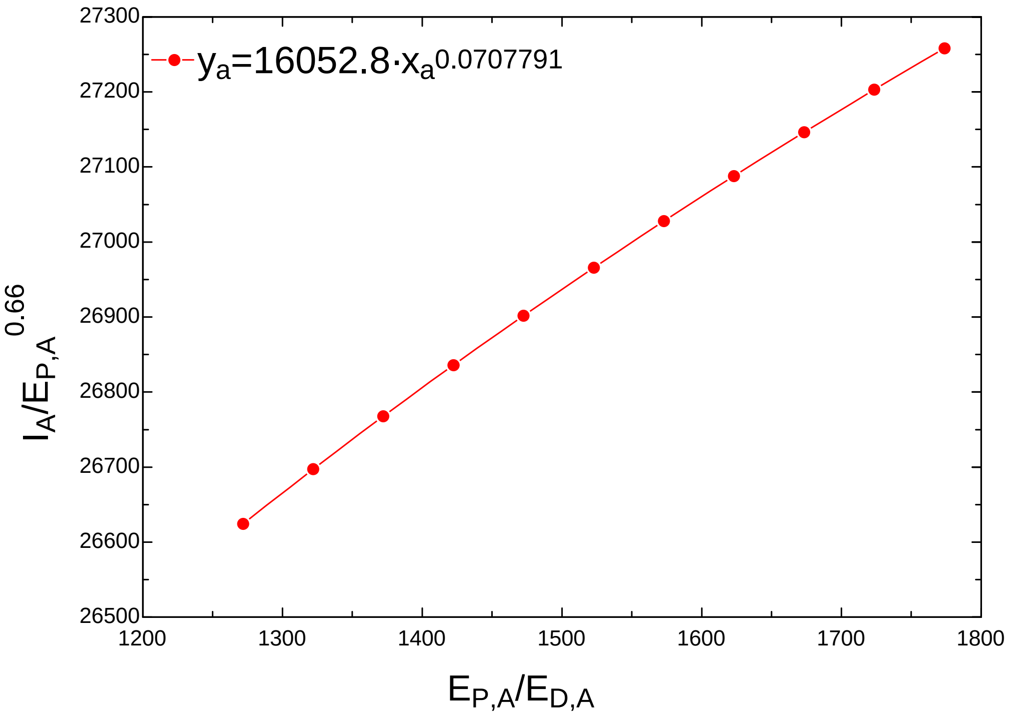


Fig. 5.16 Plot of TCI v/s Exergetic Efficiency for Absorber

Pre-cooler-1

Pre-cooler-1 and 2 are shell and tube heat exchangers. The effectiveness of the heat exchanger is considered as a decision variable and the value of m_{pc1} is taken as 0.54. Using Table 5.19 and Fig. 5.17, the values of B_{pc1} and n_{pc1} are found to be 868652 and 0.0412, respectively.

Table 5.19 Generated Data Using Investment Cost Equation for Pre-cooler-1

χ_{pc1}	$\dot{E}_{p,pc1}$ kW	$\dot{E}_{D,pc1}$ kW	$I_{pc1} / \dot{E}_{p,pc1}^{0.54}$	$\dot{E}_{P,pc1}/\dot{E}_{D,pc1}$
0.36	64.77	261.1	819048	0.248
0.38	72.17	253.7	824368	0.2844
0.4	79.96	245.9	829468	0.3251
0.42	88.15	237.7	834357	0.3708
0.44	96.74	229.2	839047	0.4222
0.46	105.7	220.2	843547	0.4802
0.48	115.1	210.8	847868	0.5462
0.50	124.9	201	852020	0.6216
0.52	135.1	190.8	856012	0.7083
0.54	145.7	180.2	859854	0.8089

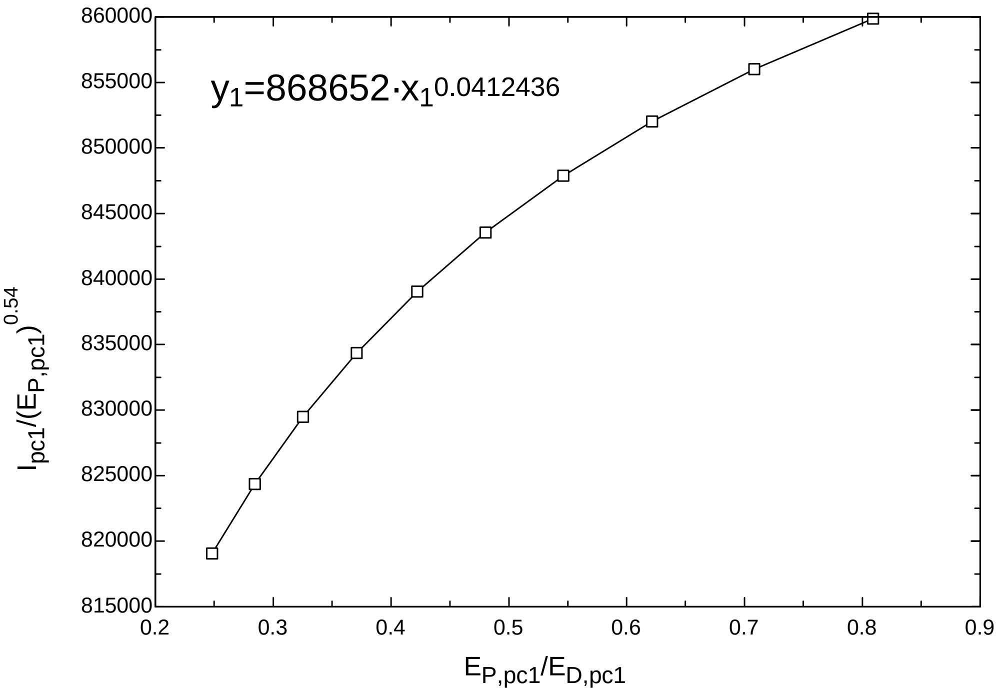


Fig. 5.17 Plot of TCI v/s Exergetic Efficiency for Pre-cooler-1

Pre-cooler-2

The effectiveness of the pre-cooler-2, a shell and tube type heat exchanger is considered as a decision variable and the value of m_{pc2} is taken as 0.54. Using Table 5.20 and Fig. 5.18, the values of B_{pc2} and n_{pc2} are found to be 665723 and 0.01075, respectively.

Table 5.20 Generated Data Using Investment Cost Equation for Pre-cooler-2

χ_{pc2}	$\dot{E}_{p,pc2}$ kW	$\dot{E}_{D,pc2}$ kW	$I_{pc2}/\dot{E}_{p,pc2}^{0.54}$	$\dot{E}_{P,pc2}/\dot{E}_{D,pc2}$
0.38	192.1	416.6	660864	0.4613
0.40	211.1	397.6	661255	0.5308
0.42	230.6	378.1	661937	0.6098
0.44	250.7	358	662825	0.7003
0.46	271.4	337.3	663854	0.8048
0.48	292.8	315.9	664976	0.9267
0.50	314.7	294	666155	1.071
0.52	337.3	271.4	667365	1.243
0.54	360.5	248.2	668586	1.452
0.56	384.3	224.4	669801	1.713

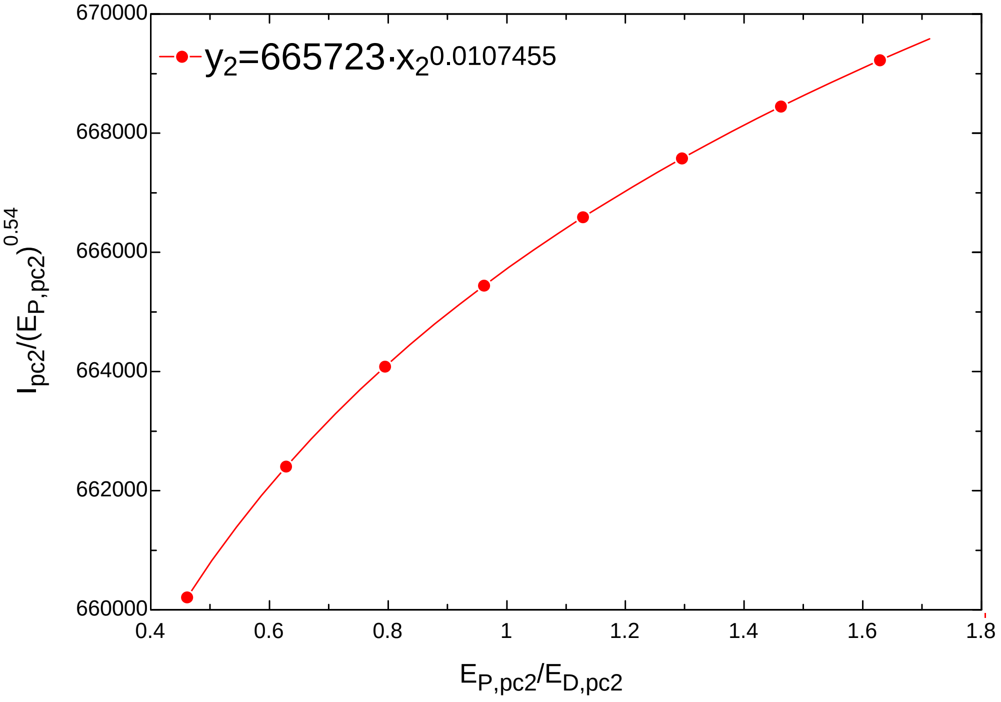


Fig. 5.18 Plot of TCI v/s Exergetic Efficiency for Pre-cooler-2

Table 5.21 summarises the component-wise parameters, B_k , n_k and m_k estimated along with the decision variable.

Table 5.21 Constants of Investment Cost Equation for Components of Brine Chilling Unit

Component	Decision variable	B_k	n_k	m_k
Generator	T_G	226231	0.048	0.66
	t_{brine} (Tc = 36°C)	1.54×10^6	0.0059	0.66
C-E Assembly	t_{brine} (Tc = 38°C)	1.54×10^6	0.0058	0.66
	t_{brine} (Tc = 40°C)	1.54×10^6	0.0057	0.66
SHX	χ_{shx}	2214.01	1.097	0.16
RHX EO5	χ_{EO5}	602445	0.267	0.66
RHX EO6	χ_{EO6}	79101.90	0.697	0.80
Sol. Pump	η_{PM}	6462.40	1.003	0.48
Absorber	$\mathrm{T_a}$	16052.8	0.071	0.66
Pre-cooler-1	χ_{pc1}	868652	0.0412	0.54
Pre-cooler-2	χ_{pc2}	665723	0.01075	0.54

5.3.2 Optimisation Through Case by Case Iterative Procedure for AAVAR system

Optimum values of exergetic efficiency (ε_k^{OPT}), the capital investment (Z_k^{OPT}), the relative cost difference (r_k^{OPT}) and the exergoeconomic factor (f_k^{OPT}) can be calculated using Eqs. 4.37, 4.45, 4.46 and 4.47, respectively. Through an iterative optimization procedure, optimum solution can be achieved, with the help of calculated values of $\dot{C}_{P,tot}, \dot{C}_{D,tot}, \dot{C}_{L,tot}$ and OBF and the guidance provided by the values of $\Delta\varepsilon_k$ and Δr_k , calculated using Eqs. 4.50 and 4.51. A sample calculation using the iterative optimisation of a single component, i.e. generator is given in Appendix E.

Table 5.22 summarizes the results obtained from the case-by-case iteration carried out starting from the base case (base case is the case evaluated using the data of the existing system) to the optimum case. A total of seven iterative cases are presented and the resulting cases are given as cases I to VII out of which the last case, i.e. case VII is found to be the optimum. Each of these cases is obtained through a series of study of positive or negative effects on $\dot{C}_{p,tot}$ and $\dot{C}_{D+L}$ by varying each decision variable. The change in the decision variables are governed by $\Delta\varepsilon_k$ and Δr_k. The details of the case by case iterative procedure for exergoeconomic optimization of AAVAR system is discussed in the following paragraph and the output given in Table 5.22.

Table 5.22 Variables Estimated during the Iterative Optimization of AAVAR system from Base Case to Optimum Case

Variable	Base Case		Case-I		Case -II		Case -III	
T_g	140°C		140°C		140°C		140°C	
T_c	40°C		40°C		40°C		40°C	
T_a	40°C		40°C		40°C		40°C	
T_e	-20°C		-20°C		-20°C		-20°C	
η_p	90%		90%		90%		85%	
χ_{SHX}	0.80		0.85		0.85		0.85	
$\chi_{RHX\,05}$	0.49		0.49		0.49		0.49	
$\chi_{RHX\,06}$	0.80		0.80		0.85		0.80	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
Generator	-4.37	28.32	-8.94	121.60	-8.99	88.72	-8.95	121.30
Cond Evap Assly.	-70.88	353.50	-70.88	382.50	-70.89	383.20	-70.89	388.20
SHX	-30.03	4619.00	-25.42	3877.00	-25.41	4371.00	-25.40	3879.00
RHX05	-98.45	10684.00	-98.45	9338.00	-98.24	5490.00	-98.45	9339.00
RHX06	-70.62	1335.00	-70.64	3200.00	-70.91	732.20	-70.63	3201.00
Solution Pump	-3.47	3168.00	-3.44	3184.00	-3.44	1159.00	-8.22	3476.00
Absorber	-0.23	-68.03	-0.20	-53.37	-0.19	-91.85	-0.20	-55.50
$\dot{C}_{L,tot}$	1960 ₹/hr		2009 ₹/hr		2014 ₹/hr		2010 ₹/hr	
$\dot{C}_{D,tot}$	12820 ₹/hr		13142 ₹/hr		13164 ₹/hr		13164 ₹/hr	
$\dot{C}_{P,tot} = \dot{C}_{P,evp} + \dot{C}_{P,pc1} + \dot{C}_{P,pc2}$	(6660 + 1626 + 6713) ₹/hr		(7020 +1556+3413) ₹/hr		(7085+1556+3413) ₹/hr		(7074+1556+3413) ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	6660+1960+12820 = 21440 ₹/hr		7020+2009+13142 = 22171 ₹/hr		7085+2014+13164 = 22263 ₹/hr		7074+2010+13164 = 22248 ₹/hr	

Table 5.22 (Continued)

Variable	Case –IV		Case -V		Case -VI		Case -VII	
T_g	140°C		140°C		142°C		142°C	
T_c	38°C		38°C		38°C		38°C	
T_a	38°C		38°C		38°C		38°C	
T_e	-18°C		-17°C		-17°C		-17°C	
η_p	0.85		0.85		0.85		0.85	
χ_{SHX}	0.85		0.85		0.85		0.85	
$\chi_{RHX\,05}$	0.49		0.49		0.49		0.47	
$\chi_{RHX\,06}$	0.80		0.80		0.80		0.78	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
Generator	-8.60	76.65	-8.54	127.90	-7.55	75.55	-7.19	105.60
Cond Evap Assly.	-69.92	232.20	-69.42	174.80	-69.44	196.90	-69.50	168.20
SHX	-26.37	4791.00	-26.38	4669.00	-26.42	4585.00	-26.42	4590.00
RHX05	-97.66	5699.00	-97.96	6117.00	-97.96	590.40	-98.54	5583.00
RHX06	-73.80	2675.00	-73.12	2577.00	-76.39	273.60	-72.98	2584.00
Solution Pump	-8.26	1860.00	-8.23	4230.00	-8.23	4229.00	-8.23	4188.00
Absorber	-0.23	-95.18	-0.25	-62.93	-0.25	91.08	-0.23	-88.97
$\dot{C}_{L,tot}$	1803 ₹/hr		1727 ₹/hr		1721 ₹/hr		1716 ₹/hr	
$\dot{C}_{D,tot}$	11802 ₹/hr		11300 ₹/hr		11266 ₹ /hr		11238 ₹/hr	
$\dot{C}_{P,tot} = \dot{C}_{P,evp} + \dot{C}_{P,pc1} + \dot{C}_{P,pc2}$	(5512+1556+3413) ₹/hr		(4932 + 1556+3413) ₹/hr		(4892 +1556+3413) ₹/hr		(4853 +1556+3413) ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	5512+1803+11802 = 19117 ₹/hr		4932+1727+11300 = 17959 ₹/hr		4892+1721+11266 = 17879 ₹/hr		4853+1716+11238 = 17807 ₹/hr	

From Base Case to Case-I

The high value of $\Delta r_{RHX\,05}$ in the base case suggests that the product cost of RHX 05 is very high. It also suggests that the effectiveness of RHX 05 should be increased. However, further increase of effectiveness is not possible for RHX 05, as it is already the maximum due to flow condition. The next highest product cost is for SHX. Therefore, the effectiveness of SHX is increased from 0.80 to 0.85. This resulted in the decrease of product cost of SHX. However, it is found that the change has adverse effect on the product cost of RHX 06 and generator. Nevertheless, it is now clear that an increase in effectiveness of SHX may be considered for the next set of the iteration.

From Case-I to Case-II

The next highest product cost is observed for RHX 06 due to the high value of $\Delta r_{RHX\,06}$. To reduce the product cost for RHX06, its effectiveness is increased from 0.80 to 0.85. Although, the cost of final product is increased, it is seen that there is no major adverse effect on the performance of the other component. Therefore, effectiveness of RHX06 is not considered as a variable for next iteration.

From Case-II to Case-III

During the exergoeconomic evaluation, it was observed that the investment cost of the solution pump should be reduced. By reducing the efficiency of the pump from 90% to 85%, it is seen that the cost of final product is reduced.

From Case-III to Case-IV

The next highest product cost is observed with condenser evaporator assembly from the value of Δr_{CE}. It may be because of high exergy destruction associated with the processes in the assembly. To reduce the exergy destruction, condenser temperature can be decreased and evaporator temperature can be increased to reduce the temperature difference between two fluids in the heat exchangers. When the temperature of condenser is decreased from 40ºC to 38ºC and temperature of evaporator is increased from -20ºC to -18ºC, it is seen the production cost of condenser evaporator assembly is reduced and thereby the cost of final product is decreased.

From Case-IV to Case-V

Considering the condenser temperature and evaporator temperature as variables for the next iteration, it is observed that the increase in evaporator temperature gives favourable result but the reduction in condenser temperature does not. So the new evaporator temperature is considered as variable for next iteration.

From Case-V to Case-VI

The component having next highest production cost is generator as illustrated by the value of Δr_G. The production cost can be reduced by decreasing the exergy destruction in the generator. It can be achieved by increasing the generator temperature. The generator temperature is increased from 140ºC to 142ºC which results in reduction in the production cost of generator and that of overall system.

From Case-VI to Case-VII

The decrease in the effectiveness of RHX 05 from 0.49 to 0.47 and that for RHX 06 from 0.80 to 0.78 gives reduction in the cost of final product.

5.3.3 Optimisation through Iterative Procedure for Pre-coolers 1 and 2

Pre-cooler-1 and pre-cooler-2 are considered as independent components as the variation in the controlling parameter of both the pre-cooler do not affect the performance of AAVAR system. Therefore the optimization of both the pre-coolers can be carried out independently.

Table 5.23 shows the effect of variation of effectiveness on the investment cost and the product cost for pre-cooler-1. With increase in the effectiveness of the pre-cooler-1, the cost of product will increase but the investment cost will decrease. With optimum condition, the temperature of brine, coming out of pre-cooler-1, will be 13.27°C which is considered as the input temperature for pre-cooler-2. Fig. 5.19 shows that the total cost of product is minimum at effectiveness $\chi_{pc1} = 0.43$ and corresponding product cost is 1556 ₹/hr. The same value of product cost is considered in the final iteration for the overall optimization performed above. Following the same procedure for pre-cooler-2, the variation of product cost and investment cost with respect to effectiveness for pre-cooler-2 is shown in Table 5.23 and Fig. 5.20. Table 5.23 shows that the optimum point can be achieved at $\chi_{pc2} = 0.47$ with the outlet brine temperature -0.5°C with product cost 3413 ₹/hr can be achieved.

Table 5.23 Effect of Effectiveness on Investment Cost for Pre-cooler-1

$T_{br,o}$ °C	χ_{pc1}	Q_{pc1} kW	$C_{p,pc1}$ ₹/hr	$\dot{Z}_{pc1}$ ₹/hr	$(C_{p,pc1} + \dot{Z}_{pc1})$ ₹/hr
15.0	0.3648	3771	1590	77.92	1668
14.5	0.3836	3965	1577	83.06	1660
14.0	0.4024	4158	1567	88.37	1655
13.5	0.4212	4352	1559	93.83	1653
13.0	0.44	4545	1553	99.48	1653
12.5	0.4588	4739	1549	105.3	1655
12.0	0.4776	4932	1547	111.4	1658
11.5	0.4964	5125	1545	117.6	1663
11.0	0.5152	5318	1545	124.1	1669
10.5	0.534	5511	1546	130.9	1677
10.0	0.5528	5704	1547	137.9	1685

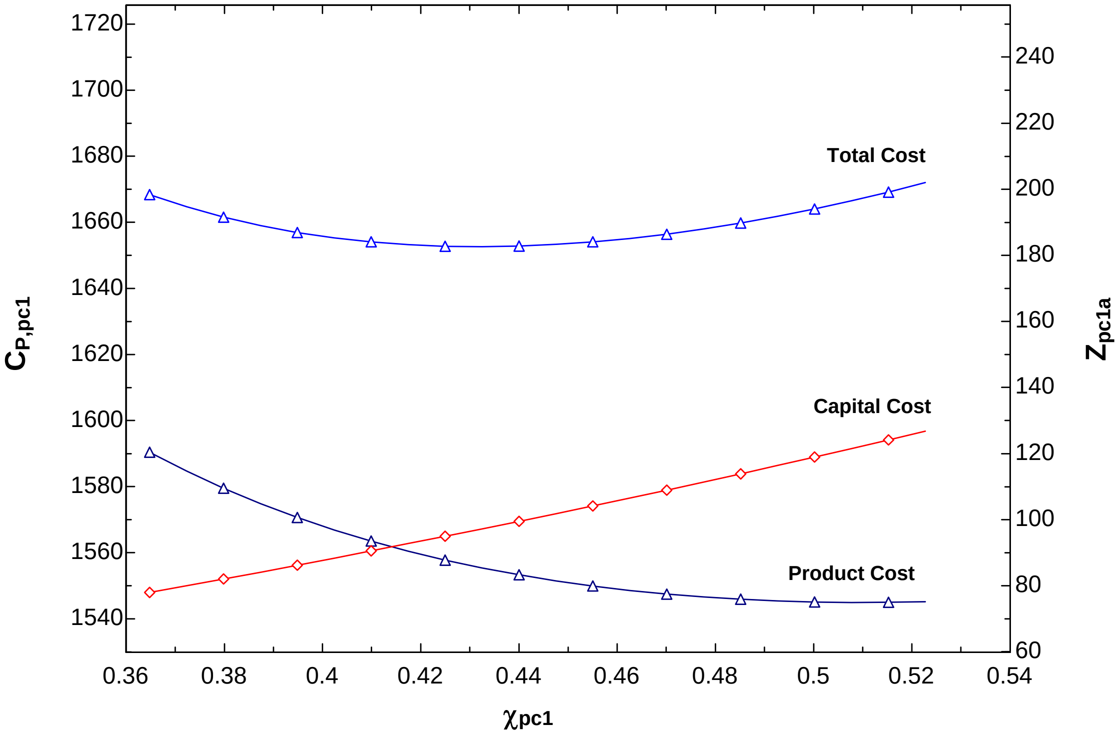


Fig. 5.19 Optimum Product Cost for Pre-cooler-1

Table 5.24 Effect of Effectiveness on Investment Cost for Pre-cooler-2

$T_{br,o}$ °C	χ_{pc2}	Q_{pc2} kW	$\dot{Z}_{pc2}$ ₹/hr	$C_{p,pc2}$ ₹/hr	$(C_{p,pc2}+\dot{Z}_{pc2})$ ₹/hr
8.0	0.1747	1929	135.8	5851	5986
7.0	0.2096	2314	166.4	5066	5232
6.0	0.2445	2699	198.2	4553	4751
5.0	0.2795	3083	231.6	4198	4430
4.0	0.3144	3467	266.6	3944	4210
3.0	0.3493	3850	303.4	3757	4060
2.0	0.3843	4233	342.2	3618	3961
1.0	0.4192	4616	383.3	3516	3899
0.0	0.4542	4998	426.8	3441	3868
-1.0	0.4891	5380	473.3	3389	3862
-2.0	0.524	5762	522.9	3355	3878
-3.0	0.559	6143	576.4	3337	3914

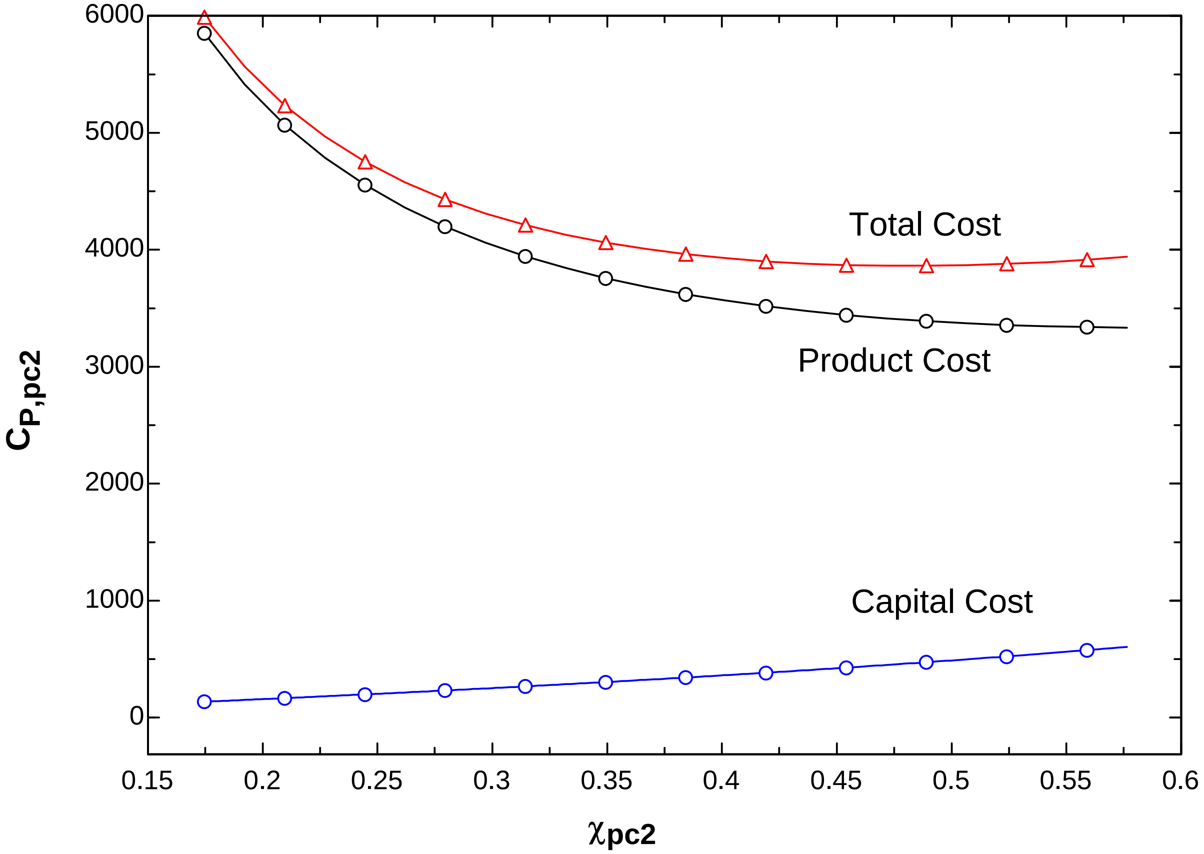


Fig. 5.20 Optimum Product Cost for Pre-cooler-2

5.3.4 Results and Discussions

In exergoeconomic optimization of brine chilling unit, the AAVAR system and both pre-coolers are optimized separately. The optimized pre-cooler-1 has effectiveness of 0.43 and corresponding product cost is 1556 ₹/hr whereas, the effectiveness and product costs of pre-cooler-2 are 0.47 and 3413 ₹/hr, respectively. It is observed that the product cost for pre-cooler-2 is quite high compared to that for pre-cooler-1, though the performance of pre-cooler-2 is better than that of pre-cooler-1. This is because of the very large O&M cost for pre-cooler-2 (Refer Table 5.9). It is suggested to reduce the O&M cost by reducing the heat transfer area for pre-cooler-2. The optimized AAVAR system is having the cost of 4853 ₹/hr for the cooling generated at evaporator of the system. Table 5.25 gives a comparative study of the final cost optimal configuration with the base case. The overall thermoeconomic cost of the product (chilled brine) is decreased by about 27.13 % (6660 ₹/hr to 4853 ₹/hr) with corresponding 12.76 % decrease (1.96 ₹/MJ to 1.71 ₹/MJ) in the fuel cost results from the reduction in consumption of fuel. The cost of exergy destruction is also decreased by 12.34 % and that of exergy loss is decrease by 12.45 %. These cost reduction is accompanied by the increase in the investment cost of solution heat exchanger and reduction in the investment cost of solution pump, RHX 05 and RHX 06. Improvement in the system performance can be realized by the increase in the exergetic efficiency by 13.04 % and increase in the COP by 11.9 %.

Table 5.25 Comparison between the Base Case and the Optimum Case

Properties	Base Case	Optimum Case	% Variation
Fuel Cost $\dot{C}_{F,tot}$	1.96 ₹ /MJ	1.71 ₹ /MJ	-12.76
Product Cost $\dot{C}_P$	6660 ₹ /hr	4853 ₹/hr	-27.13
Loss $\dot{C}_{L,tot}$	1960 ₹/hr	1716 ₹/hr	-12.45
Destruction $\dot{C}_{D,tot}$	12820 ₹/hr	11238 ₹/hr	-12.34
Exergetic Efficiency ε%	23 %	26 %	13.04 %
COP	0.42	0.47	11.90 %

Chapter 6

Exergoeconomic Optimization of Alternative Options

Exergoeconomic optimization of two alternative options available with the fertilizer plant in terms of fuel (steam) source is examined in this chapter. Firstly, the AAVAR system is simulated in combination with 8 MW gas turbine power plant, instead of the independent boiler for steam source, which is also the part of fertilizer company infra-structure. In this case, the steam generated at HRSG is considered as heat source. The system is optimized exergoeconomically and the cost of steam generated at HRSG is estimated for minimum cost of power generation. The optimum cooling cost for AAVAR system is estimated considering steam generated at HRSG as heat source. Section 6.1 deals with the details of the alternative option.

Next alternative option available is tapped steam from the 50 MW steam turbine power plant which is the major source electricity for the fertilizer plant. Section 6.2 describes the exergoeconomic optimization of the AAVAR system using tapped steam from a certain stage of the steam turbine of the plant as heat source. The losses in various components are identified and the cost of steam tapped from the steam turbine is estimated for the minimum cost of power generation by the steam turbine. This tapped steam is utilized as a heat source for AAVAR system and the cost of cooling generated by the system is estimated.

6.1 Steam Generated at HRSG as Heat Source

The existing AAVAR system is equipped with an independent boiler generating saturated steam at 15 bar for the purpose of using it as fuel in the generator. With this arrangement, it is found that the generation cost of steam is 900 ₹/1000 kg. It will be worthwhile to consider other options of steam generation and its utilization as fuel to AAVAR system as some of them are readily available in the fertilizer unit. Keeping this in mind, the present study is carried out to try two additional sources of heat energy available in the plant. The first among them is the partial use of steam generated in the Gas Turbine- Heat Recovery Steam Generator (GT-HRSG) plant as heat source for AAVAR system. It should be noted that GT-HRSG plant acts as a captive power plant catering to the need of power requirement in the fertilizer plant. This section examines the option for the reduction of cost of brine chilling using AAVAR system through exergoeconomic optimization using the steam generated from HRSG partially. It is expected that if the steam generated by the use of waste heat at the GT-HRSG plant as fuel for AAVAR system, there could be significant reduction in the steam cost and hence the cost of cooling.

The following sections give the step by step procedure adopted for the exergoeconomic optimization scheme employed earlier for the existing system. As the details of the scheme are presented earlier, the following section may not repeat the same.

6.1.1 System Simulation

Using the steady state online data, the system simulation is carried out and the missing data are generated. The assumptions underlying the GT-HRSG system model include the following:

- The GT-HRSG system operates at steady state.
- Laws of ideal gas mixture apply for the air and the combustion products.
- The combustion in the combustion chamber is complete.
- Heat loss from the combustion chamber is 2 % of the fuel LHV.

In the GT-HRSG plant model, two types of independent variables are identified, decision variables and parameters. The decision variables are varied in optimization studies, but the parameters remain fixed. All other variables are dependent variables and their values are calculated using the thermodynamic model.

The compressor pressure ratio (p_2 / p_1), isentropic compressor efficiency η_c, effectiveness of air preheater χ_{APH}, isentropic turbine efficiency η_T, temperature of air entering the combustion chamber T_3 and temperature of the combustion product entering the turbine T_4 are considered as decision variables. The dependent variables include the mass flow rates of the air, combustion products and fuel, the power required by the compressor, the power developed by the turbine and pressure and temperature of plant components as follows:

Air compressor	p_2, T_2
Air preheater	p_3, p_6, T_6
Combustion chamber	p_4
Gas turbine	p_5, T_5
HRSG	T_7

Parameters are independent variables whose values are specified. They are kept fixed in optimization study. In this model, the following parameters that are fixed are identified.

- System Products
    - The net power generated by the system is 8 MW
    - Saturated water vapour supplied by the system at $p_9 = 15$ bar
- Air Compressor
    - $T_1 = 298.1\text{K}$, $p_1 = 1.013$ bar
    - Air molecular analysis (%): $77.48\ (\text{N}_2),\ 20.59\ (\text{O}_2),\ 0.03\ (\text{CO}_2),\ 1.90\ (\text{H}_2\text{O})$.
- Air Preheater
    - Pressure drop: 3 % on gas side and 5 % on the air side.
- Heat Recovery Steam Generator
    - $T_8 = 298.1\text{K},\ p_8 = 15$ bar, $p_7 = 1.013$ bar
    - Pressure drop: 5 % on gas side.
- Combustion Chamber
    - $T_{10} = 298.1\text{K},\ P_{10} = 12$ bar

- Pressure drop: 5 %
- Temperature of combustion product $T_4 = 1520K$

Using the assumptions listed, a standard set of governing equations are available in literature. This involves consideration of several individual control volumes identified with reference to various components of the plant.

Air Compressor

The temperature of the air inlet to compressor, $T_1 = 298.1\text{K}$. At this temperature, the enthalpies of all the constituents, nitrogen, oxygen, carbon dioxide and water vapour are taken from Table F1 of Appendix F while these properties at the temperature other than reference temperature are calculated with the help of Table F2 of Appendix F. Then the enthalpies of all the constituents are added on molar basis and the enthalpy of the air inlet to compressor is calculated on molar basis.

$$h_1^{'} = 0.7748 h_{\mathrm{N_2}}(T_1) + 0.2059 h_{\mathrm{O_2}}(T_1) + 0.0003 h_{\mathrm{CO_2}}(T_1) + 0.019 h_{\mathrm{H_2O}}(T_1) \tag{6.1}$$

The molecular weight of the air inlet to compressor is calculated using

$$M_a = 0.7748 M_{\mathrm{N_2}} + 0.2059 M_{\mathrm{O_2}} + 0.0003 M_{\mathrm{CO_2}} + 0.019 M_{\mathrm{H_2O}} \tag{6.2}$$

Using these values, the enthalpy of air on mass basis is calculated using

$$h_1 = h_1^{'} / M_a \tag{6.3}$$

The temperature at the end of compression is calculated using

$$T_2 = T_1 \left\{ 1 + \frac{1}{\eta_{AC}} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right] \right\} \tag{6.4}$$

Where isentropic efficiency of the compressor $\eta_{AC} = 86$ % and pressure ratio $(p_2 / p_1) = 10$. At this temperature, the enthalpy of the air leaving the compressor (h_2) is calculated following the same procedure as applied for T_1.

Air Preheater

The pressure drop on the air side of the air preheater is considered as 5 % as suggested by Tsatsaronis et al. [114]. The pressure of the air coming out of the air preheater is estimated using

$$p_3 = p_2(1-\Delta p_{a,APH}) \quad \text{with} \quad \Delta p_{a,APH} = 0.05 \tag{6.5}$$

The temperature of the air coming out of the air preheater (T_3) is calculated using the effectiveness of the air preheater:

$$\chi_{APH} = \frac{T_3 - T_2}{T_5 - T_2} \tag{6.6}$$

For the base case, the effectiveness of air preheater is considered as, $\chi_{APH} = 75\ \%$ which will give the temperature of the air (T_3) at the exit of air preheater. The enthalpy of the air coming out of the air preheater (h_3) is calculated following the same procedure as applied for air at temperature T_1.

By energy balance across the air preheater, the temperature of the gas (T_6) leaving from the air preheater is calculated.

$$m_a C_{P,a}(T_3 - T_2) = m_g C_{p,g}(T_5 - T_6) \tag{6.7}$$

The specific heat of air and gas is taken from Tsatsaronis et al [114].

$C_{P,a} = 1.005$ kJ/kgK, $C_{P,g} = 1.17$ kJ/kgK

Combustion Chamber

Denoting the air fuel ratio on molar basis as λ, the molar flow rates of the fuel, air and the combustion product are related by

$$\frac{\dot{n}_f}{\dot{n}_a} = \lambda, \qquad \frac{\dot{n}_p}{\dot{n}_a} = 1+\lambda, \tag{6.8}$$

Where f, p and a denote fuel, combustion product and air, respectively. For complete combustion of methane the chemical equation takes the form

$$\begin{aligned} &\lambda \mathrm{CH_4} + [0.7748\mathrm{N_2} + 0.2059\mathrm{O_2} + 0.0003\mathrm{CO_2} + 0.019\mathrm{H_2O}] \\ &\quad \rightarrow [1+\lambda][x_{\mathrm{N_2}}\mathrm{N_2} + x_{\mathrm{O_2}}\mathrm{O_2} + x_{\mathrm{CO_2}}\mathrm{CO_2} + x_{\mathrm{H_2O}}\mathrm{H_2O}] \end{aligned} \tag{6.9}$$

Using the temperature of the combustion product from the combustion chamber ($T_4 = 1520K$) and the energy balance across the combustion chamber, air fuel ratio (λ) and enthalpy of combustion product (h_4) are estimated.

$$(1+\lambda)*h_4 = 0.7748*h_{N_2}(T_4)+(0.2059-2\lambda)*h_{O_2}(T_4)+(0.0003+\lambda)*h_{CO_2}(T_4)+ (0.019+2\lambda)*h_{H_2O}(T_4) \tag{6.10}$$

$$-0.02\lambda*\text{LHV}+h_3+\lambda*h_f-(1+\lambda)*h_4=0 \tag{6.11}$$

Where $h_f = -74872$ kJ/kmol and LHV=802361 kJ/kmol and 2 % loss is considered as suggested by Bejan et al. [155]. Solving Eqs. 6.10 and 6.11 for λ and enthalpy of combustion product on molar basis are calculated. Their values are $\lambda = 0.03006$ and $h_4 = 10921$ kJ/kmol. Once the air fuel ratio is calculated, the molar analysis of the product can be decided by balancing mole fractions of carbon, oxygen and nitrogen of the combustion product in Eq. 6.9. Table 6.1 gives the mole fraction of the constituent gases in the combustion products estimated.

$$x_{\mathrm{N_2}} = \frac{0.7748}{1+\lambda}, \qquad x_{\mathrm{O_2}} = \frac{0.2059-2\lambda}{1+\lambda}$$
$$x_{\mathrm{CO_2}} = \frac{0.0003+\lambda}{1+\lambda}, \quad x_{\mathrm{H_2O}} = \frac{0.019+2\lambda}{1+\lambda} \tag{6.12}$$

Table 6.1 Molar Analysis of the Combustion Product

component	N_2	O_2	CO_2	H_2O
Mole fraction	0.7522	0.1415	0.02947	0.07681

Using the mole fractions of the constituents, the molecular weight of the combustion product is calculated as

$$M_P = x_{N_2}*M_{N_2}+x_{O_2}*M_{O_2}+x_{CO_2}*M_{CO_2}+x_{H_2O}*M_{H_2O} \tag{6.13}$$

Enthalpy of combustion product on mass basis (kJ/kg) is calculated using

$$h_4 = h_4^{'}/M_p \tag{6.14}$$

The pressure drop in the combustion chamber is considered as 5% [114] then the pressure of the combustion product is calculated using

$$p_4 = p_3(1 - \Delta p_{CC}) \quad \text{with} \quad \Delta p_{CC} = 0.05 \tag{6.15}$$

Gas Turbine

Combustion product from the combustion chamber at temperature T_4 and P_4 enters the gas turbine and expands to the final pressure $p_5 = 1.099$ bar . The temperature of the gas at the exit of the gas turbine (T_5) is calculated using

$$T_5 = T_4 \left\{ 1 - \eta_{GT} \left[1 - \left(\frac{p_4}{p_a} \right)^{\frac{1-\gamma_g}{\gamma_g}} \right] \right\} \tag{6.16}$$

Where, $\eta_{GT} = 0.86$ for the base case. At T_5, the enthalpy of exhaust gas h_5 , is calculated in terms of kJ/kmol using

$$h_5^{'} = x_{N_2} h_{\mathrm{N_2}}(T_5) + x_{O_2} h_{\mathrm{O_2}}(T_5) + x_{CO_2} h_{\mathrm{CO_2}}(T_5) + x_{H_2O} h_{\mathrm{H_2O}}(T_5) \tag{6.17}$$

$$h_5 = \frac{h_5^{'}}{M_p} \tag{6.18}$$

Considering the control volume enclosing the compressor and turbine

$$\dot{W}_{CV} = \dot{n}_a (h_1^{'} - h_2^{'}) + \dot{n}_p (h_4^{'} - h_5^{'})\,` \tag{6.19}$$

$$\frac{\dot{W}_{CV}}{\dot{n}_a} = (h_1^{'} - h_2^{'}) + (1 + \lambda)(h_4^{'} - h_5^{'}) \tag{6.20}$$

Here all the enthalpies are in kJ/kmol. Converting to a mass rate basis and solving, the mass flow rate of air is

$$\dot{m}_a = \frac{M_a * \dot{W}_{CV}}{(1 + \lambda)(h_4^{'} - h_5^{'}) + (h_1^{'} - h_2^{'})} \tag{6.21}$$

After calculating the mass flow rate of air, mass flow rate of fuel is found using

$$\dot{m}_f = \lambda \left(\frac{M_f}{M_a} \right) \dot{m}_a \tag{6.22}$$

Then the mass flow rate of gas trough turbine is

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{6.23}$$

The gas leaving from the turbine passes through air preheater and is used for preheating the air going to the combustion chamber. The temperature of the gas leaving from the air preheater (T_6) is calculated by energy balance through Eq. 6.7 and enthalpy at the same temperature is calculated using

$$h_6^{'} = x_{N_2} h_{\mathrm{N_2}}(T_6) + x_{O_2} h_{\mathrm{O_2}}(T_6) + x_{CO_2} h_{\mathrm{CO_2}}(T_6) + x_{H_2O} h_{\mathrm{H_2O}}(T_6) \tag{6.24}$$

$$h_6 = \frac{h_6^{'}}{M_p} \tag{6.25}$$

On the gas side of air preheater, pressure drop is considered as 3 % [114], then

$$p_6 = p_5(1 - \Delta p_{g,APH}) \quad \text{with} \quad \Delta p_{g,APH} = 0.03 \tag{6.26}$$

Heat Recovery Steam Generator

The energy of the exhaust gas is utilized in HRSG for steam generation at 15 bar saturated. Owing to the presence of sulphur in natural gas, corrosive sulphuric acid can be formed when the products of combustion are sufficiently cooled. This can be guarded against by maintaining the temperature T_7 above 450 K. By energy balance

$$\dot{m}_s (h_9 - h_8) = \dot{m}_g \, C_{p,g} (h_6 - h_7) \tag{6.27}$$

Thus, solving Eq. 6.27, it is seen that the steam generation rate in HRSG is $\dot{m}_s = 3.25$ kg/sec which is quite closer to the requirement of steam (fuel) in AAVAR system. Allowing a pressure drop of 5 % in HRSG [114], the pressure of the gas leaving the HRSG is

$$p_7 = p_6(1 - \Delta p_{HRSG}) \text{ with } \Delta p_{HRSG} = 0.05 \tag{6.28}$$

The air inlet to compressor is at T_{ref} and p_{ref} and is considered as ideal gas mixture. The entropy of all the components at temperature T_{ref} and p_{ref} ($s_k^0(T)$) is taken from Table F1 of Appendix F and entropy at other temperature and pressure is calculated with the help of Table F2 of Appendix F. After calculating the entropy of all the component of the air, the entropy of the air inlet to compressor is calculated in terms of kJ/kmol

$$s_1^{'} = 0.7748 s_{\mathrm{N_2}}(T_1) + 0.2059 s_{\mathrm{O_2}}(T_1) + 0.0003 s_{\mathrm{CO_2}}(T_1) + 0.019 s_{\mathrm{H_2O}}(T_1) \tag{6.29}$$

$$s_1 = \frac{s_1^{'}}{M_a} \text{ (kJ/kg)} \tag{6.30}$$

The air at stations 2 and 3 is at pressure other than p_{ref}. Then entropy of air at temperature T_2 and P_2 is calculated as

$$\begin{aligned} s_2^{'} = {} & 0.7748\left[s_{\mathrm{N_2}}(T_2) - R\ln\left(\frac{0.7748 * p_2}{p_1}\right)\right] + 0.2059\left[s_{\mathrm{O_2}}(T_2) - R\ln\left(\frac{0.2059 * p_2}{p_1}\right)\right] + \\ & 0.0003\left[s_{\mathrm{CO_2}}(T_2) - R\ln\left(\frac{0.0003 * P_2}{p_1}\right)\right] + 0.019\left[s_{\mathrm{H_2O}}(T_2) - R\ln\left(\frac{0.019 * P_2}{p_1}\right)\right] \end{aligned} \tag{6.31}$$

$$s_2 = \frac{s_2^{'}}{M_a} \quad \text{(kJ/kg)} \tag{6.32}$$

Here $p_1 = p_{ref}$. Similarly entropy of air at T_3 and p_3 (s_3) is also calculated.

At station 4, combustion product is considered as ideal gas mixture. The mole fraction of all the components can be estimated using Eq. 6.12. Using these mole fractions, the entropy of combustion product at station 4 is found using following relations in terms of kJ/kmol.

$$\begin{aligned} s_4^{'} = {} & x_{N_2}\left[s_{\mathrm{N_2}}(T_4) - R\ln\left(\frac{x_{N_2} * p_4}{p_1}\right)\right] + x_{O_2}\left[s_{\mathrm{O_2}}(T_4) - R\ln\left(\frac{x_{O_2} * p_4}{p_1}\right)\right] + \\ & x_{CO_2}\left[s_{\mathrm{CO_2}}(T_4) - R\ln\left(\frac{x_{CO_2} * p_4}{p_1}\right)\right] + x_{H_2O}\left[s_{\mathrm{H_2O}}(T_4) - R\ln\left(\frac{x_{H_2O} * p_4}{p_1}\right)\right] \end{aligned} \tag{6.33}$$

$$s_4 = \frac{s_4^{'}}{M_p} \quad \text{(kJ/kg)} \tag{6.34}$$

Similarly, entropy at stations 5, 6 and 7 is also calculated.

6.1.2 Exergy Analysis

The exergy of the working substance (streams) in all the components of GT-HRSG system possesses physical and chemical components, physical exergy and chemical exergy are estimated using the procedure given in Sections 6.1.2.1 and 6.1.2.2.

6.1.2.1 Physical Exergy

As discussed in Chapter 4, the physical exergy component is associated with the work obtainable in bringing a matter from its initial state to a state that is in thermal and mechanical equilibrium with the environment. The air inlet to compressor at station 1 is at T_{ref} and p_{ref}. Therefore $h_1 = h_{01}$ and $s_1 = s_{01}$ then from Eq. 4.2, physical exergy at station 1 will be zero. Applying the same Eq. 4.2, the physical exergy at stations 2 and 3 are calculated.

To calculate the exergy of combustion product and exhaust gas from the turbine, it is considered that they are reduced to $T_{ref} = 25^o C$ and $p_{ref} = 1.01325$ bar. At this temperature, some condensation of water will occur and gas phase containing saturated water vapour in equilibrium with saturated liquid water phase. On the basis of 1 kmol of combustion products formed, the gas phase at 25ºC would consists of 0.9232 kmol of dry products (0.7522 N_2, 0.1415 O_2, 0.02947 CO_2) plus n_v kmol of water vapour. The partial pressure of water vapour would be equal to the saturation pressure, $p_g(25^o C) = 0.0317$ bar. The amount of water vapour is estimated using

$$p_v = x_v p \tag{6.35}$$

$$\text{i.e., } 0.0317 \text{ bar} = \frac{n_v}{0.9232 + n_v}(1.01325 \text{ bar}) \tag{6.36}$$

Solving Eq. 6.36, $n_v = 0.0298$ kmol. Thus for the case of combustion products given in Table 6.1, the composition of the combustion product at 25ºC and 1 atm reads

$\underline{0.7522\, N_2,\ 0.1415\, O_2,\ 0.02947\, CO_2\ ,0.02982\, H_2O\ (g)}$, $0.04699 H_2O$ (l)

Where, the underline identifies the gas phase. Using these values, enthalpy of combustion product at 25ºC and 1 atm as

$$h_{04}^{'} = 0.7522 N_2 + 0.1415 O_2 + 0.02947 CO_2 + 0.02982 H_2O(g) + 0.04699 H_2O(l) \tag{6.37}$$

$$h_{04} = \frac{h_{04}^{'}}{M_p} \text{ kJ/kg} \tag{6.38}$$

The physical exergy at station 4 of Fig.3.3 is calculated using

$$\dot{E}_4^{PH} = \dot{m}_4\left[\left(h_4 - h_{04}\right) - T_0(s_4 - s_{04})\right] \tag{6.39}$$

Using the same value of enthalpy at reference condition, physical exergy at stations 5, 6 and 7 are calculated. The feed water inlet to HRSG at station 8 is maintained at 15 bar and considered at reference temperature. At given temperature and pressure, its enthalpy and entropy is calculated using inbuilt subroutine of EES software. Similarly the enthalpy and entropy of feed water to HRSG at reference temperature and pressure are calculated using inbuilt subroutine of EES software. Using enthalpy and entropy at actual condition and at reference condition, the exergy of feed water to HRSG is calculated using Eq.6.39. At station 9, steam is generated at 15 bar saturated. The enthalpy and entropy of steam at 15 bar saturated and at reference condition are calculated using inbuilt subroutine of EES software and using Eq. 6.39, its physical exergy is calculated. Natural gas, maintained at 10 bar pressure, is used as a fuel in the combustion chamber of gas turbine. The physical exergy of fuel at station 10 is calculated as

$$\dot{E}_{10}^{PH} = \dot{m}_{10}\left[h_{10} - h_0 - T_0\left(s_{10} - s_0\right)\right]$$

Since $T_{10} = T_0$, the above equation reduces to

$$\dot{E}_{10}^{PH} = \dot{m}_{10}\, RT_0 \ln \frac{p_{10}}{p_0} \tag{6.40}$$

6.1.2.2 Chemical Exergy

At stations 1, 2 and 3, air is stable with environment so its chemical exergy is considered as zero. At dead state corresponding to the mixture at station 4 consists of liquid water phase and a gas phase. The new mole fraction of a gas phase is calculated using Eq.6.41 and found as

$$y_{N_2} = \frac{x_{N_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + n_v} \qquad y_{O_2} = \frac{x_{O_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + n_v}$$

$$y_{CO_2} = \frac{x_{CO_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + n_v} \qquad y_{H_2O(g)} = \frac{n_v}{x_{N_2} + x_{O_2} + x_{CO_2} + n_v} \tag{6.41}$$

$y_{N_2} = 0.7893,\ y_{O_2} = 0.1485,\ y_{CO_2} = 0.03093,\ y_{H_2O(g)} = 0.03129$.

Now the chemical exergy for the kth component is calculated and added together to find total chemical exergy using following equation

$$e^{CH} = \sum y_k e_k^{CH} + RT_0 \sum y_k \ln y_k \tag{6.42}$$

This is the chemical exergy of gas portion. The chemical exergy of liquid portion is separately calculated and added together to find total chemical exergy. The chemical exergy of individual component (e_k^{CH}) is given in Appendix G.

$$\dot{E}^{CH} = \dot{m}_g \left[\left(x_{N_2} + x_{O_2} + x_{CO_2} + n_v \right) e^{CH} + x_{H_2O(l)} * e^{CH}_{H_2O(l)} \right] \qquad (6.43)$$

The chemical exergy calculated using Eq. 6.43 is expressed in kJ/kmol which can then be converted in to kJ/kg by dividing it by molecular mass of the combustion product. The chemical exergy at stations 5, 6 and 7 will remain the same. Chemical exergy of water, steam and natural gas (CH_4) at stations 8, 9 and 10, respectively, are taken from of Appendix G. The total exergy flow at stations 4 to 10 will be the sum of physical and chemical exergy. Table 6.2 gives the properties and parameters at each stations of the GT-HRSG plant along with the estimated values of exergy.

Table 6.2 State Properties for Gas Turbine Power Plant

Stations	Mass flow rate kg/sec	Pres-Sure. bar	Temp. K	Specific Enthalpy kJ/kg	Specific Entropy kJ/kgK	Physical Exergy kW	Chemical Exergy kW	Total Exergy MW
1	24.09	1.013	298.10	-164.60	6.7860	0	0	0
2	24.09	10.13	621.30	179.90	7.0700	6257	0	6.257
3	24.09	9.63	894.50	488.20	7.4960	10627	0	10.63
4	24.49	9.15	1520	386.30	8.3370	25177	83.99	25.26
5	24.49	1.099	985.50	-279.10	8.4210	8260	83.99	8.344
6	24.49	1.066	754.70	-554.80	8.1120	3768	83.99	3.852
7	24.49	1.013	450	-904.30	7.5350	915.50	83.99	0.9995
8	3.25	15	298.10	106.10	0.3666	4.56	8.13	0.01269
9	3.25	15	471.50	2791	6.4440	2845	8.13	2.853
10	0.41	12	298.10	---	---	155.3	20896	21.05

6.1.2.3 Definition of Fuel, Product and Loss for Various Processes

Using the exergy estimated at each station as given in Table 6.2, fuel, product and loss are to be calculated for all the components in a similar manner as that used in Chapter 5, Section 5.1.2. The step by step procedure adopted is given in Section 5.1.2.1 of Chapter 5. It should be recollected that GT-HRSG is one option to act as the source of heat energy (fuel) for AAVAR system. As such there are six components for the GT-HRSG plant for which fuel, product and loss are estimated as per the requirement of exergoeconomic analysis. Table 6.3 summarises the same and its numerical values are given in Table 6.4.

Table 6.3 Component-wise Fuel, Product and Loss of GT-HRSG Power Plant

Component	Fuel ($\dot{E}_F$)	Product ($\dot{E}_P$)	Loss ($\dot{E}_L$)
AC	$\dot{E}^{W}_{11}$	$\dot{E}_2 - \dot{E}_1$	0
APH	$\dot{E}_5 - \dot{E}_6$	$\dot{E}_3 - \dot{E}_2$	0
CC	$\dot{E}_3 + \dot{E}_{10}$	$\dot{E}_4$	0
GT	$\dot{E}_4 - \dot{E}_5$	$\dot{E}_{11} + \dot{E}_{12}$	0
HRSG	$\dot{E}_6 - \dot{E}_7$	$\dot{E}_9 - \dot{E}_8$	0
System	$\dot{E}_{10}$	$(\dot{E}_9 - \dot{E}_8) + \dot{E}_{12}$	$\dot{E}_7$

Table 6.4 Exergy Analysis Result for Gas Power Plant

Component	$\dot{E}_F$ kW	$\dot{E}_p$ kW	$\dot{E}_L$ kW	$\dot{E}_D$ kW	Y_D %	Y_L %	Y_D^* %	ε %
AC	8298	6257	0	2041	9.693	0	22.15	75.41
APH	4491	4369	0	122.3	0.5808	0	1.327	97.28
CC	31678	25261	0	6417	30.48	0	69.67	79.74
GT	16917	16299	0	618.4	2.938	0	6.713	96.34
HRSG	2853	2840	0	12.57	0.0597	0	0.1364	99.56
System	21052	10841	999.50	9211	43.76	4.75	100	51.50

6.1.2.4 Results and Discussions

The outcome of the exergy analysis of the gas turbine power plant is given in Table 6.4. The total exergy supplied to the system is 21052 kW, out of which 10841 kW (51.50 %) is converted to useful product, 9211 kW (43.76 %) exergy is destroyed and 999.50 kW (7.75%) is lost to the environment.

The maximum exergy destruction is found in combustion chamber. To reduce the exergy destruction in combustion chamber, inlet temperature T_3 should be increased. It can be achieved by increasing the air preheater's effectiveness or compression ratio of the compressor. The effectiveness of APH is already high, the compression ratio of the compressor should be increased or isentropic efficiency of the compressor should be improved.

The next highest exergy destruction is in air compressor which can be reduced by improving its isentropic efficiency. The next is the gas turbine in this category. The exergy destruction in the combustion chamber and gas turbine can be reduced by increasing the inlet temperature T_4 and improving the isentropic efficiency but at the same time the investment cost of gas turbine and combustion chamber will increase which can increase the product cost. So the optimum temperature should be selected. During implementation of all these improvements, optimum condition should be considered.

6.1.3 Exergoeconomic Analysis

The essence of the economic analysis is the identification and inclusion of various cost heads incurred in the estimation of the total cost for the production. In the present case, the total cost involved in the power generation by the gas turbine consists of many cost heads. Thus, in general, the economic analysis of the system requires the estimation of levelized O&M cost of component ($\dot{Z}_k$) and fuel cost rate ($\dot{C}_f$). $\dot{Z}_k$ should be estimated for each component for GT-HRSG plant using TCI, β, γ and τ (Refer Eq.4.18). The fuel cost rate ($\dot{C}_f$) is governed by the source of heat energy used for the system. The estimation of $\dot{Z}_k$ and $\dot{C}_f$ are explained in the following section.

6.1.3.1 Levelized O&M cost ($\dot{Z}_k$)

The purchase equipment costs of each component are calculated using the cost model of each component given in Appendix H for the year 1994. Using the M&S cost index, they are converted for the year 2009. Using Table 4.1, the TCI related to each component is found. The levelized O&M cost of each component is found using Eq. 4.18. Considering the plant life of 8000 hours, Capital Recovery Factor (β) = 0.1061, O&M cost (γ) = 1.092 % of total investment cost of the component [114], the values of levelized O&M cost for each components are determined. The estimated values of Levelized O&M cost for all the components of GT-HRSG are given in Table 6.5. It can be seen that for each components of the GT-HRSG plant, a number of cost heads are involved in the

Table 6.5 Estimation of Levelized O & M Cost for the Components of GT-HRSG

Component	PEC (₹) (1)	Installation Cost 45 % of PEC (2)	Piping Cost 66 % of PEC (3)	Instru.& Control cost 20 % of PEC (4)	Electrical Equipment 11 % of PEC (5)	On Site Cost (ONSC) 1+2+3+4+5	Land 10 % of PEC (6)	Civil Work 60 % of PEC (7)	Service 65 % of PEC (8)	Of Site Cost (OFSC) (6+7+8)	Direct Cost (DC) ONSC + OFSC
AC	49292450	22181603	32533017	9858490	5422170	119287730	4929245	29575470	32040093	66544808	185832538
APH	428100	192645	282546	85620	47091	1036002	42810	256860	278265	577935	1613937
CC	4454150	2004368	2939739	890830	489957	10779044	445415	2672490	2895198	6013103	16792147
GT	49275800	22174110	32522028	9855160	5420338	119247436	4927580	29565480	32029270	66522330	185769766
HRSG	31017100	13957695	20471286	6203420	3411881	75061382	3101710	18610260	20161115	41873085	116934467

Continue Table 6.5

Engg. & Supervision 30% of PEC (9)	Construction Cost 15 % of DC (10)	Contingency 20 % of FCI (11)	Indirect Cost (IC) (9+10+11)	Fixed Capital Investment (FCI) (DC+IC)	Startup Cost 10% of FCI (12)	Working Capital 15 % of TCI (13)	Allowance For Funds 10% of PEC (14)	Other Outlays (12+13+14)	TCI (FCI+Other Outlays)	Levelized O&M Cost ₹/hr $(\dot{Z}_k)$
14787735	27874880	31158430	73821045	259653583	25965358	51273209	4929245	82167812	341821395	5000
128430	242091	270608	641129	2255065	225506	445303	42810	713619	2968684	43.42
1336245	2518822	2815529	6670596	23462743	2346274	4633135	445415	7424824	30887567	451.80
14782740	27865465	31147905	73796110	259565876	25956588	51255890	4927580	82140058	341705934	4998
9305130	17540170	19606332	46451632	163386099	16338610	32263486	3101710	51703806	215089905	3146

estimation of TCI. TCI consists of FCI and Other Outlays. DC and IC constitute FCI, while Other Outlays consists of start up cost, working capital cost and allowance for funds. DC consists of on-site (ONSC) and off-site (OFSC) costs while IC consists of engineering & supervision, construction and contingency costs.

6.1.3.2 Fuel Cost

The plant uses natural gas (methane) as a fuel. The market prize of methane for the year 2009 is considered as 4.3 $/mm BTU (1mm BTU = 1055.06 MJ). If the LHV of methane is considered as 50 MJ/kg, then the cost of fuel will be 0.2 $/kg or 10 ₹/kg.

6.1.3.3 Cost Flow

To calculate the exergy cost flow at each station of the gas power plant, the cost balance equations are modelled as explained below.

Air Compressor

$$\dot{c}_1 \dot{E}_1 - \dot{c}_2 \dot{E}_2 + \dot{c}_{11} \dot{E}_{11} + \dot{Z}_{AC} = 0 \tag{6.44}$$

Air Preheater

$$\dot{c}_2 \dot{E}_2 - \dot{c}_3 \dot{E}_3 + \dot{c}_5 \dot{E}_5 - \dot{c}_6 \dot{E}_6 + \dot{Z}_{APH} = 0 \tag{6.45}$$

$$\dot{c}_5 = \dot{c}_6 \tag{6.46}$$

Combustion Chamber

$$\dot{c}_3 \dot{E}_3 - \dot{c}_4 \dot{E}_4 + C_f + \dot{Z}_{CC} = 0 \tag{6.47}$$

Gas Turbine

$$\dot{c}_4 \dot{E}_4 - \dot{c}_5 \dot{E}_5 - \dot{c}_{11} \dot{E}_{11} - \dot{c}_{12} \dot{E}_{12} + \dot{Z}_{GT} = 0 \tag{6.48}$$

$$\dot{c}_4 = \dot{c}_5 \tag{6.49}$$

$$\dot{c}_{11} = \dot{c}_{12} \tag{6.50}$$

Heat Recovery Steam generator

$$\dot{c}_6 \dot{E}_6 - \dot{c}_7 \dot{E}_7 + \dot{c}_8 \dot{E}_8 - \dot{c}_9 \dot{E}_9 + \dot{Z}_{HRSG} = 0 \qquad (6.51)$$

$$\dot{c}_6 = \dot{c}_7 \qquad (6.52)$$

Out of these variables, $\dot{c}_1 ... \dot{c}_9, \dot{c}_{11}, \dot{c}_{12}$ and C_f, the last is known which is the cost of fuel in combustion chamber. The cost of air at compressor inlet and cost of water at inlet to HRSG ($\dot{c}_1$ and $\dot{c}_8$) are considered as zero. The remaining 9 are calculated by solving Eqs. 6.44 to 6.52 using EES software. The cost per unit exergy (₹/MJ) and cost flow rate (₹/sec) for each flow of the system are calculated and shown in Table 6.6. For this calculation, known values of $\dot{E}_1$ to $\dot{E}_{12}$ are used.

Table 6.6 Unit Exergy Cost and Cost Flow Rate for Gas Power Plant

Flows	Unit exergy cost ₹/MJ	Exergy flow MW	Cost flow rate ₹/sec
1	0	0	0
2	1.1850	6.2570	7.414
3	0.9598	10.6300	10.200
4	0.6175	25.2600	15.600
5	0.6175	8.3440	5.152
6	0.6175	3.8520	2.379
7	0.6175	0.9995	0.617
8	0	0.0127	0
9	0.9238	2.8530	2.635
10	0.2000	21.0500	4.210
11	0.7261	8.2980	6.025
12	0.7261	8.0000	5.809

6.1.4 Exergoeconomic Evaluation

After calculating the cost rates at each station of the plant using cost rate of fuel ($\dot{C}_{F,k}$) as an input the cost rate of product ($\dot{C}_{p,k}$), cost rate of fuel per unit exergy ($c_{F,k}$), cost rate of product per unit exergy ($c_{P,k}$), cost rate of exergy destruction ($\dot{C}_{D,k}$), cost rate of exergy loss ($\dot{C}_{L,k}$), the relative cost difference (r_k) and exergoeconomic factor (f_k) for each component are calculated using Eq. 4.20 to Eq. 4.27 and given in Table 6.7. Based on these results, the system is exergoeconomically evaluated following the methodology suggested by Bejan et al. [155] and discussed in Section 4.2.3.

Table 6.7 Results of Exergoeconomic Analysis for Gas Power Plant

Component	$c_{F,k}$ ₹/MJ	$c_{p,k}$ ₹/MJ	$\dot{C}_{D,k}$ ₹/hr	$\dot{C}_{L,k}$ ₹/hr	$\dot{Z}_k$ ₹/hr	f_k %	r_k %	**ε %**
AC	0.7261	1.1850	5334	0	5000	48.38	63.18	75.41
APH	0.6175	0.6375	271.8	0	43.42	13.78	3.25	97.28
CC	0.4549	0.6175	10509	0	451.80	4.12	35.74	79.74
GT	0.6175	0.7261	1375	0	4998	78.43	17.60	96.34
HRSG	0.6175	0.9279	27.94	0	3146	99.12	50.28	99.56
System	0.2000	0.7790	6632	719.70	13640	64.98	289.50	51.50

6.1.4.1 Results and Discussions

The following observations are made from the results shown in Table 6.7.

(i) The r value for the compressor is highest among all the components, indicates that, for this design configuration, particular attention should be paid to air compressor. The air compressor has the lowest exergetic efficiency and second largest rate of exergy destruction cost. Therefore, it would be cost effective to reduce the exergy destruction in compressor by increasing the isentropic efficiency. By using multi stage compressor and providing the Intercooling between the stages, the power consumption by the compressor can be reduced.

(ii) The HRSG has the second highest r value but having high exergetic efficiency and low rate of cost of exergy destruction. So this component is working properly.

(iii) The combustion chamber has the next highest r value. This is due to the very high exergy destruction costs and extremely low f value. The logical conclusion would be to try to decrease the exergy destruction in the combustion chamber by increasing the air preheating temperature T_3.

6.1.5. Exergoeconomic Optimization

The exergoeconomic optimization of the system requires a thermodynamic model and a cost model. The thermodynamic model gives the performance prediction of the system with respect to some thermodynamic variables such as exergy destruction, exergy loss and exergetic efficiency. The cost model permits detailed calculation of cost values for a given set of the thermodynamic variables. For each component, it is expected that the investment cost increases with increasing capacity and increasing exergetic efficiency.

6.1.5.1 Estimation of B_k, n_k and m_k

Using the value of cost flow at each station and the results of exergoeconomic evaluation, the exergoeconomic optimization of the system is carried out at component level using Eq. 4.29. To solve this equation for local optimum by curve fitting technique, the equivalent power law is found and the required value of B_k and n_k for each component are determined for the selected value of m_k as explained below.

Air Compressor

For Air Compressor, efficiency of the compressor (η_{AC}) and compression ratio (r_c) are considered as decision variables. For the variation of isentropic efficiency of air compressor from 0.85 to 0.89, the necessary data are generated as explained in Chapter 4, Section 4.3. For the generation of data, value of $m_{AC} = 0.95$ as suggested by Bejan et al. [155] is taken. Table 6.8 gives the generated data for carrying out regression fit to obtain B_{AC} and n_{AC} .

Table 6.8 Generated Data Using Investment Cost Equation for Air Compressor

η_{AC}	r_c	$\dot{E}_{P,AC}$	$\dot{E}_{D,AC}$	$\dot{E}_{P,AC}/\dot{E}_{D,AC}$	$TCI_{AC}/\dot{E}_{p,AC}^{0.95}$
0.85	8	5.266	1.998	2.635	117239
0.86	9	5.575	1.936	2.88	159992
0.87	10	5.853	1.879	3.114	231363
0.88	11	6.102	1.827	3.339	374238
0.89	12	6.328	1.778	3.559	803096

From Fig. 6.1, the values of B_{AC} and n_{AC} for the selected value of m_{AC} of 0.95 are found to be 251.88 and 6.17, respectively.

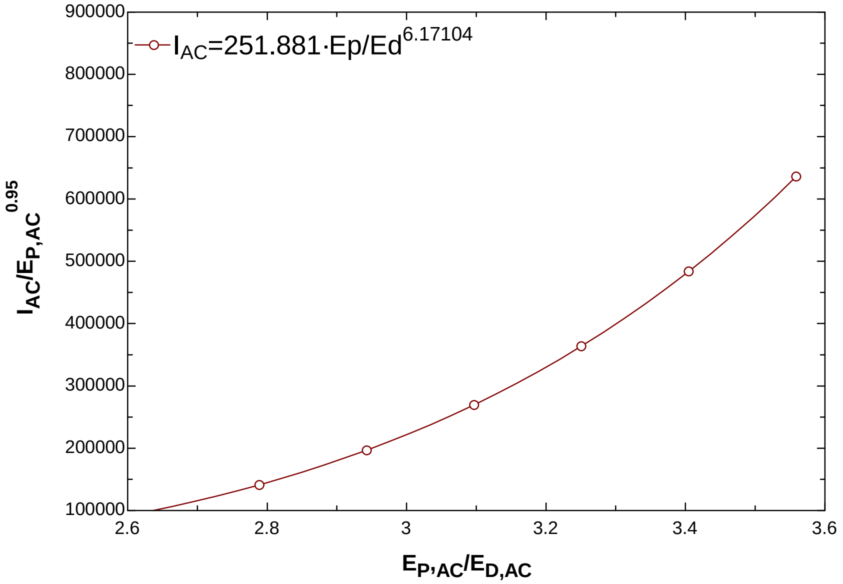


Fig. 6.1 Plot of Investment cost v/s Exergetic Efficiency for Air Compressor

Air Preheater

Air preheater is a device used to recover waste heat of exhaust gas from the gas turbine. By varying the effectiveness of air preheater, the amount of heat recovered can be varied so the effectiveness of air preheater is considered as decision variable. For the range of values of effectiveness from 0.70 to 0.79, data related to exergy of product and destruction are generated and is given in Table 6.9.

Table 6.9 Generated Data Using Investment Cost Equation for Air Preheater

χ_{APH}	$\dot{E}_{P,APH}$	$\dot{E}_{D,APH}$	$\dot{E}_{P,APH}/\dot{E}_{D,APH}$	$TCI_{APH}/\dot{E}_{p,APH}^{0.6}$
0.7	3.97	0.2067	19.18	3223
0.71	4.03	0.1938	20.81	3271
0.72	4.10	0.1805	22.73	3321
0.73	4.17	0.1669	25.00	3373
0.74	4.24	0.1528	27.75	3428
0.75	4.31	0.1383	31.15	3486
0.76	4.38	0.1235	35.46	3548
0.77	4.45	0.1083	41.10	3612
0.78	4.52	0.0927	48.78	3681
0.79	4.59	0.0766	59.89	3754

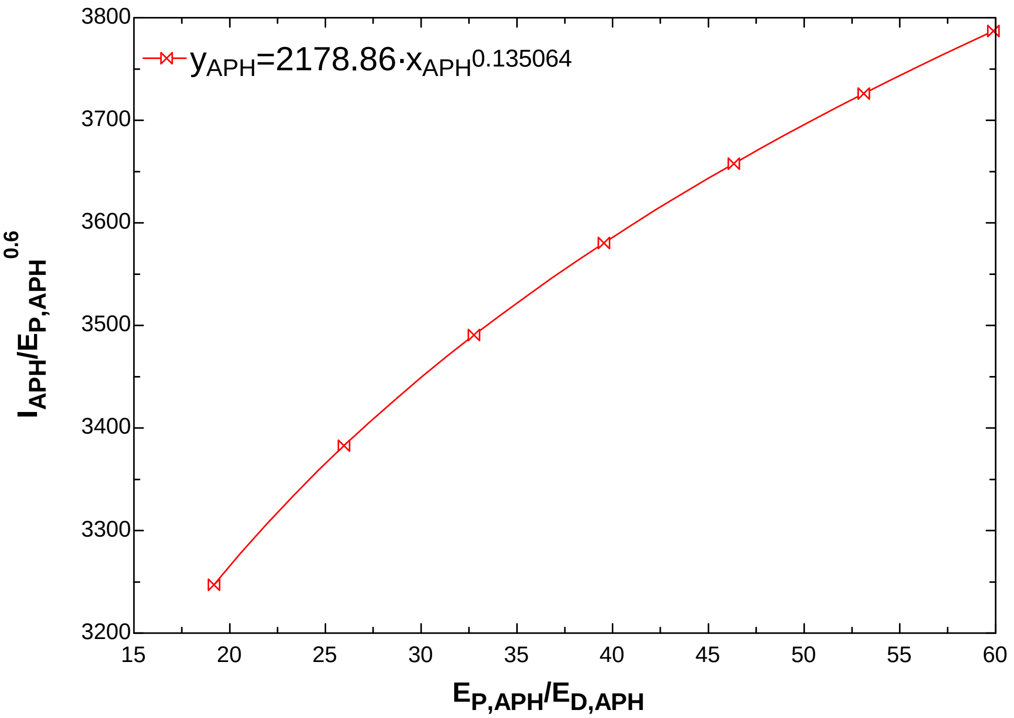


Fig. 6.2: Plot of Investment cost v/s Exergetic Efficiency for Air Preheater

For air preheater with effectiveness (χ_{APH}) of it as decision variable, the Fig. 6.2 shows that the value of B_{APH} and n_{APH} are found to be 2178.86 and 0.135, respectively for the selected value of m_{APH} of 0.6

Table 6.10 Generated Data Using Investment Cost Equation for Combustion Chamber

T_4	$\dot{E}_{P,CC}$	$\dot{E}_{D,CC}$	$\dot{E}_{P,CC}/\dot{E}_{D,CC}$	$TCI_{CC\pi}/\dot{E}_{p,CC}$
1500	24.97	6.02	4.19	2776
1501	24.95	6.03	4.14	2792
1501	24.93	6.03	4.13	2808
1502	24.91	6.04	4.12	2824
1502	24.90	6.05	4.12	2841
1503	24.88	6.06	4.11	2857
1503	24.86	6.06	4.10	2874
1504	24.84	6.07	4.09	2891
1504	24.83	6.08	4.09	2908
1505	24.81	6.09	4.08	2925

Combustion Chamber

For combustion chamber, the temperature of combustion product (T_4) is considered as decision variable. For the variation in T_4 from 1500 K to 1505 K, the required data are generated and given in Table 6.10.

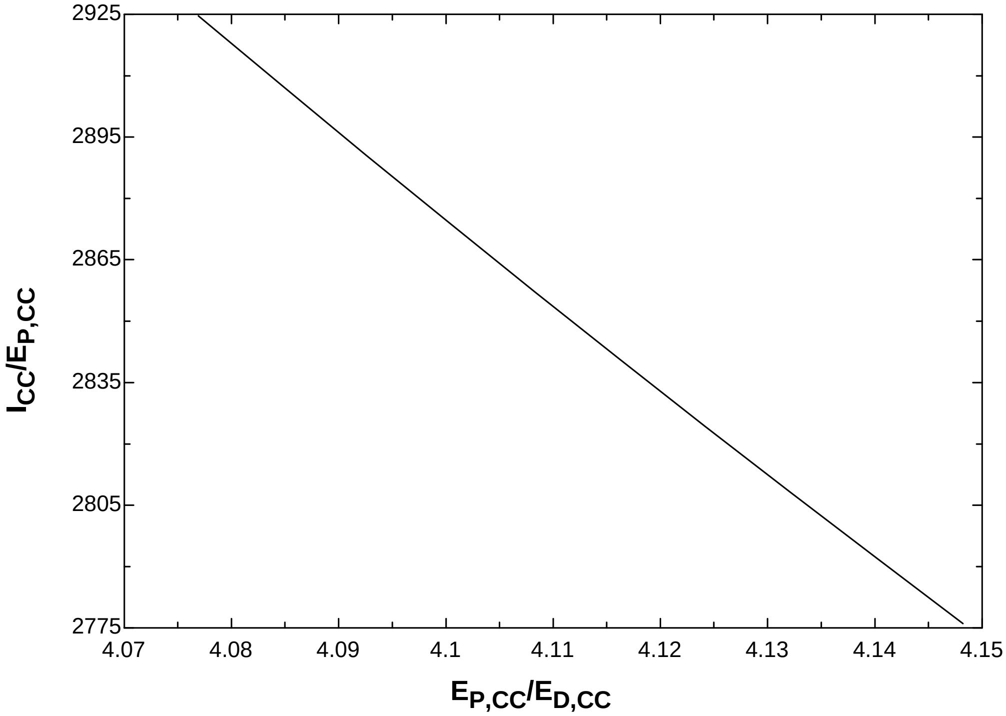


Fig. 6.3 Plot of Investment cost v/s Exergetic Efficiency for Combustion Chamber

For combustion chamber, the slope of the graph is negative so the value of n_{CC} can not be defined. So it is assumed as unity as suggested by Bejan et al. [155]. Fig. 6.3 shows that the value of B_{CC} and n_{CC} are found to be 1001 and 1, respectively for the selected value of m_{CC} of 1.

Gas Turbine

For gas turbine, isentropic efficiency (η_{GT}) and temperature of combustion product (T_4) are considered as decision variable. The generated data are shown in Table 6.11. The graph, given in Fig. 6.4 shows that the value of B_{GT} and n_{GT} are found to be 404.59 and 1.828 for the selected value of m_{GT} of 0.65.

Table 6.11 Generated Data Using Investment Cost Equation for Gas Turbine

η_{GT}	T_4	$\dot{E}_{P,GT}$	$\dot{E}_{D,GT}$	$\dot{E}_{P,GT} / \dot{E}_{D,GT}$	$TCI_{GT} / \dot{E}_{p,GT}^{0.65}$
0.84	1515	16.79	0.776	21.63	113159
0.85	1517	16.54	0.696	23.77	132019
0.86	1519	16.31	0.6198	26.32	157485
0.87	1521	16.09	0.5473	29.4	193539
0.88	1523	15.88	0.4781	33.22	248144

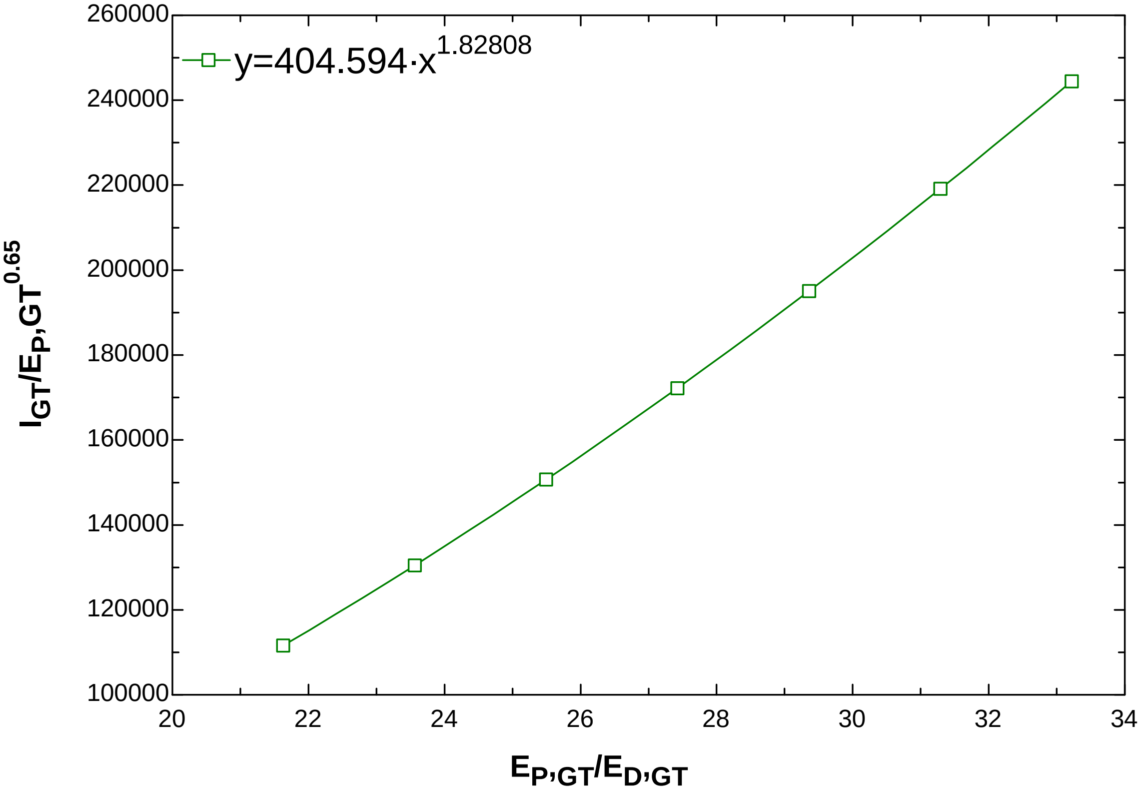


Fig.6.4 Plot of Investment Cost v/s Exergetic Efficiency for Gas Turbine

Heat Recovery Steam Generator

For HRSG, temperature of the exhaust gas coming out of the turbine (T_6) is considered as decision variable. The data generated for the variation in T_6 are given in Table 6.12. The Fig. 6.5 shows that the value of B_{HRSG} and n_{HRSG} are found to be 245553 and 0.0077, respectively for the selected value of m_{HRSG} of 0.85.

Table 6.12 Generated data through investment cost equation for HRSG

T_6	$\dot{E}_{P,HRSG}$	$\dot{E}_{D,HRSG}$	$\dot{E}_{P,HRSG}/\dot{E}_{D,HRSG}$	$TCI_{HRSG}/\dot{E}_{p,HRSG}^{0.85}$
768.2	3.46	0.1363	25.39	251105
766.0	3.34	0.1127	29.66	251791
763.7	3.23	0.0905	35.73	252491
761.5	3.13	0.0694	45.06	253206
759.2	3.03	0.0494	61.21	253934
757.0	2.93	0.0305	96.04	254677
754.7	2.84	0.0126	226.20	255433

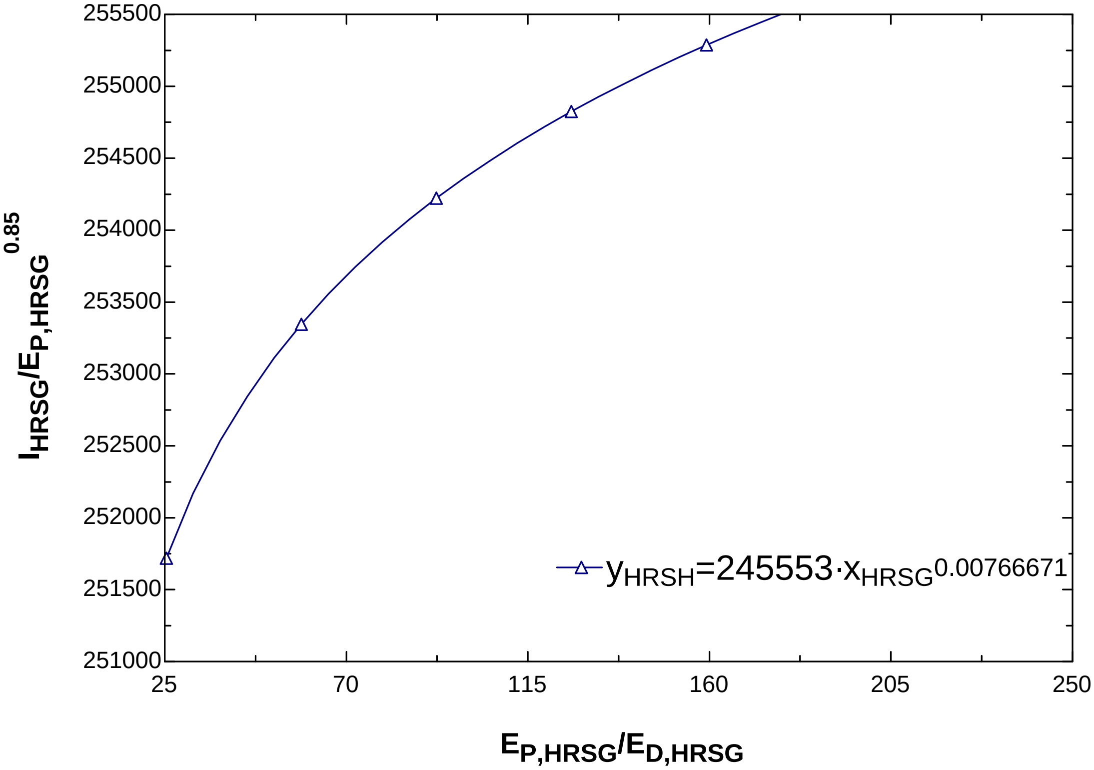


Fig. 6.5: Plot of Investment cost v/s Exergetic Efficiency for HRSG

Table 6.13 summarises the component-wise parameters, B_k , n_k and m_k estimated along with the decision variable.

Table 6.13 Values of B_k, n_k and m_k for various components

Component	Decision variable	B_k	n_k	m_k
AC	η_{AC} & r_c	251.88	6.17	0.95
APH	χ_{APH}	2178.86	0.135	0.6
CC	T_4	1001	1	1
GT	η_{GT} & T_4	404.59	1.828	0.65
HRSG	T_6	245553	0.0077	0.85

6.1.5.2 Optimisation Through Case by Case Iterative Procedure

Optimum values of exergetic efficiency (ε_k^{OPT}), the capital investment (Z_k^{OPT}), the relative cost difference (r_k^{OPT}) and the exergoeconomic factor (f_k^{OPT}) can be calculated using Eqs. 4.37, 4.45, 4.46 and 4.47, respectively. Through an iterative optimization procedure, optimum solution can be achieved, with the help of calculated values of $\dot{C}_{P,tot}, \dot{C}_{D,tot}, \dot{C}_{L,tot}$ and OBF and the guidance provided by the values of $\Delta\varepsilon_k$ and Δr_k, calculated using Eqs. 4.50 and 4.51.

Table 6.14 summarizes the results obtained from the case-by-case iteration carried out starting from the base case (base case is the case evaluated using the data of the existing system) to the optimum case. A total of seven iterative cases are presented and the resulting cases are given as cases I to VII out of which the case IV is found to be the optimum. Each of these cases is obtained through a series of study of positive or negative effects on $\dot{C}_{p,tot}$ and $\dot{C}_{D+L}$ by varying each decision variable. The change in the decision variables are governed by $\Delta\varepsilon_k$ and Δr_k. The details of the case by case iterative procedure for exergoeconomic optimization of AAVAR system is discussed in the following paragraph and the output given in Table 6.14. In the base case, the unit product cost of electricity is 2.61 ₹/kWh and production cost of steam is 810 ₹/1000 kg and total generation of steam is 3.25 kg/sec.

From base case to case-I:

The highest value of Δr_{APH} shows that the product cost of air preheater is very high. It also suggests that the effectiveness of air preheater should be increased. The effectiveness of APH is increased from 0.75 to 0.8. With this, the cost of electricity is reduced to 2.51 ₹/kWh and cost of steam is 830 ₹/1000 kg. But the major heat is recovered from the exhaust gas; the rate of steam generation is reduced to 2.9 kg/sec which is not sufficient for absorption refrigeration system. Therefore this parametric variation is kept pending for later stages.

From case-I to case-II:

The next highest Δr_{HRSG} suggest that the exergy destruction in HRSG can be reduced by decreasing T_6. It can be achieved by increasing the effectiveness of APH which is already checked in the previous iteration. By reducing the air compressor pressure ratio, the heat recovery at APH can be increased and temperature T_6 can be reduced. The air compressor pressure ratio is reduced from 10 to 9. With this, Δr_{HRSG} is reduced but not sufficient reduction in the product cost is achieved. In this condition, the rate of steam generation is 3.15 kg/sec and the cost of steam is 840 ₹/1000 kg.

From case-II to case-III

The next highest Δr_{CC} value suggests that the exergy destruction in the combustion chamber should be reduced. The highest exergy destruction cost can be observed in the combustion chamber in Table 6.6 also. The exergy destruction in combustion chamber can be reduced by increasing the temperature T_3 which can be achieved by increasing the compressor efficiency up to 0.87. But the compressor investment and maintenance cost is so sensitive with compressor efficiency. The increase in compressor efficiency results in increase in the Δr_{AC} and therefore, increase in the product cost.

From case-III to case-IV

The decrease in the compressor efficiency up to 0.85 will give reduction in the product cost. The cost of electricity generation will be 2.56 ₹/kWh and that of steam will be 810 ₹/1000 kg with steam generation rate 3.2 kg/sec.

From case-IV to case-V

The next highest production cost is observed with gas turbine. From Table 6.6, it is observed the investment cost of turbine is very high which can be reduced by decreasing the efficiency of the gas turbine up to 0.85. But doing so, is resulting in increase in production cost

From case-V to case-VI

Opposite to the above step, increase in the gas turbine efficiency up to 0.87 results in increase in the investment cost and subsequently increase in the product cost.

Table 6.14 Variables Obtained Exergoeconomic Optimization of Gas Turbine Power Plant (From Base Case to Optimum Case)

Variable	Base Case		Case-I		Case-II	
p_2 / p_1	10		10		9	
η_{AC}	0.86		0.86		0.86	
χ_{APH}	0.75		0.80		0.75	
η_{GT}	0.86		0.86		0.86	
T_4	1247 ºC		1247 ºC		1247 ºC	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
AC	-9.944	179.7	-9.851	183.2	-11.38	236.4
APH	-2.719	70029	-0.9494	29519	-3.399	89538
CC	-20.02	5876	-17.56	5156	-18.78	5568
GT	-2.758	1115	-2.764	1140	-2.641	1117
HRSG	-0.4388	32989	-4.629	37430	-1.479	34598
$\dot{C}_{L,tot}$	719.7 ₹/hr		1158 ₹/hr		978.3 ₹/hr	
$\dot{C}_{D,tot}$	6632 ₹/hr		6038 ₹/hr		6370 ₹/hr	
$\dot{C}_{P}$	20912 ₹/hr		20088 ₹/hr		21348 ₹/hr	
$OBF = \dot{C}_{P} + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	28263.7 ₹/hr		27284 ₹/hr		28696 ₹/hr	

Table 6.14 (continued)

Variable	Case-III		Case-IV		Case-V	
p_2 / p_1	9		9		9	
η_{AC}	0.87		0.85		0.85	
χ_{APH}	0.75		0.75		0.75	
η_{GT}	0.86		0.86		0.85	
T_4	1247 ºC		1247 ºC		1247 ºC	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
AC	-11.16	284.8	-11.64	205.3	-11.73	197.6
APH	-3.54	98057	-3.259	83341	-3.249	84579
CC	-18.91	5697	-18.66	5478	-18.05	5322
GT	-2.662	1089	-2.628	1132	-2.996	1392
HRSG	-2.139	36072	-2.509	34808	-3.089	34367
$\dot{C}_{L,tot}$	946.1 ₹ /hr		1012 ₹ /hr		1149 ₹/hr	
$\dot{C}_{D,tot}$	6309 ₹/hr		6469 ₹/hr		6527 ₹/hr	
$\dot{C}_P$	22777 ₹/hr		20484 ₹/hr		21362 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	30032 ₹/hr		27965 ₹/hr		29038 ₹/hr	

Table 6.14 (continued)

Variable	**Case-VI**		**Case-VII**		**Case-VIII**	
p_2 / p_1	9		9		9	
η_{AC}	0.85		0.85		0.85	
χ_{APH}	0.75		0.75		0.80	
η_{GT}	0.87		0.85		0.85	
T_4	1247 ºC		1227 ºC		1227 ºC	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
AC	-11.92	194.6	-11.68	195.6	-11.6	199.3
APH	-3.28	87673	-3.139	79880	-1.249	35675
CC	-19.28	5743	-16.81	4977	-14.04	4231
GT	-2.297	1717	-3.1	1401	-3.104	1436
HRSG	-2.069	35495	-3.159	34364	-1.829	36152
$\dot{C}_{L,tot}$	881.9 ₹/hr		1511 ₹/hr		2027 ₹/hr	
$\dot{C}_{D,tot}$	6417 ₹/hr		6364 ₹/hr		5594 ₹/hr	
$\dot{C}_P$	23544 ₹ /hr		20808 ₹/hr		19955 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	30843 ₹/hr		28683 ₹/hr		27576 ₹/hr	

From case-VI to case-VII

To reduce the exergy destruction in combustion chamber and gas turbine, the temperature T_4 can be increased but it will increase the investment cost of gas turbine and combustion chamber. So the optimum temperature should be decided. The reduction in T_4 up to 1227 ºC results in decrease in the production cost. The cost of electricity generation is 2.6 ₹/kWh and cost of steam generation is 800 ₹/1000 kg with 3.3 kg/sec steam flow rate.

From case-VII to case-VIII

Combining all the favourable parameters, including effectiveness of air preheater, the production cost is reduced. The cost of electricity generated is 2.49 ₹/kWh and cost of steam 790 ₹/1000 kg with steam flow rate 3 kg/sec. Here minimum product cost is achieved but the rate of steam generation is less than the requirement. The required steam is 3.14 kg/sec. If the rate of steam generation is to be maintained above 3.14 kg/sec, then the case-IV is to be considered as optimum one.

Considering case IV as optimum, the steam generated at HRSG will have the cost 810 Rs/1000 kg and using this steam as fuel in AAVAR system, the cooling cost for the cooling generated at evaporator will be reduced from 1.35 ₹/sec to 1.07 ₹/sec.

6.1.5.3 Results and Discussions

Various data generated during the optimization procedure using a case by case approach adopted in the present study of gas turbine power plant with HRSG is given in Table 6.14. The comparison of optimum case with the base case is given in Table 6.15. From the study, it can be seen that the cost of electricity is reduced by 4.60 % (2.61 ₹/kWh to 2.49 ₹/kWh) with corresponding decrease in exergy destruction of 15.65 % (6632 ₹/hr to 5594 ₹/hr). The cost of steam generated at HRSG is also reduced from 810 ₹/1000 kg to 790 ₹/1000 kg. When AAVAR system is associated with the HRSG, the cost of cooling at evaporator is reduced to 4853 ₹/hr to 3910 ₹/hr. It shows that the steam generated at HRSG is more economical compared to steam generated at independent boiler as a fuel for AAVAR system. The table shows that this improvement is achieved at the slight reduction in exergetic efficiency by 2.75 %. It should be observed that the rate of power generation and rate of fuel consumption are maintained constant.

Table 6.15: Comparison between base case and the optimum case for GT-HRSG

Properties	Base Case	Optimum Case	% Variation
Fuel Cost $\dot{C}_{F,tot}$	0.2 ₹/MJ	0.2 ₹/MJ	0
Product Cost $\dot{C}_P$	20912 ₹/hr	19955 ₹/hr	4.58 %
Cost of Electricity	2.61 ₹/kWh	2.49 ₹/kWh	4.60 %
Destruction $\dot{C}_{D,tot}$	6632 ₹/hr	5594 ₹/hr	15.65 %
Exergetic Efficiency ε %	51.50 %	50.08 %	2.75 %
Generated Steam Cost	810 ₹/1000 kg	790 ₹/1000 kg	2.47 %

6.2 Tapped Steam as Heat Source

As mentioned earlier, the cost of steam generated in the independent boiler is about 900 ₹/1000 kg (Chapter 5) and steam generated in HRSG of gas turbine power plant is 790 to 810 ₹/1000 kg (Section 6.1 of this chapter). The cost of cooling produced using the above mentioned sources of fuel (steam) for the existing and alternative option of steam generation in GT-HRSG is calculated and presented. As a second option for the reduction of cost of brine chilling, the use of tapped steam from steam turbine of existing steam power plant is analysed through exergoeconomic optimization in this section.

6.2.1 System Simulation

From the available online data, the system is simulated through energy balance and mass balance for all the components and missing data are generated. The following assumptions are considered.

- The power plant system operates at steady state.
- Ideal gas mixture principles apply for the air and the combustion product in the boiler
- The combustion in the combustion chamber of the boiler is complete.
- Super heater and economizer are not considered as independent part.
- Efficiency of the draught fan 80%

In the steam turbine plant model, two types of independent variables are identified, decision variables and parameters or independent variable. The decision variables are varied in the optimization study, while the parameters remain fixed. All other variables are dependent variables and their values are calculated using thermodynamic analysis.

Temperature of the combustion product in the boiler furnace T_{24}, pressure of the steam generated P_1, isentropic efficiency of the turbine η_T and condenser pressure P_6 are considered as decision variables (refer Fig. 3.4)

The following are the fixed parameters used in the present optimisation:

- *System product*
  - The net power generated by the system is 50 MW

- *Boiler*
  - FD fan draught 472 mmWC
  - ID fan draught 230 mmWC
  - Air molecular analysis (%): $77.48\ (N_2),\ 20.59\ (O_2),\ 0.03\ (CO_2),\ 1.90\ (H_2O)$.
  - Gas side pressure drop in the boiler 170 mm WC approx.
- *Steam turbine*
  - First extraction pressure and flow rate 17 bar and 7.67 TPH
  - Second extraction pressure and flow rate 7 bar and 6.6 TPH
  - First extraction pressure and flow rate 4 bar and 10 TPH
- *Surface condenser*
  - Design temperature 100ºC
  - Cooling water flow 92 m^3/hr

The dependent variables include the mass flow rates of the air, combustion products and fuel, the power consumption by the pump and draught fan moreover turbine exit temperature and pressure. Based on the assumption listed, several control volumes considered and set of governing equations developed are given below:

Steam Turbine

For the power generation of 50 MW by the steam turbine, the pressure and temperature at station 1 are given and pressure for stations 2 to 5 are given. Considering the isentropic efficiency of turbine during expansion of steam as 80 %, the actual enthalpies at stations 1 to 5 are calculated. Then by energy balance,

$$\dot{m}_1(h_1 - h_2) + (\dot{m}_1 - \dot{m}_2)(h_2 - h_3) + (\dot{m}_1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_4) + \dot{m}_5(h_4 - h_5) = W_T \tag{6.53}$$

where $\dot{m}_5 = \dot{m}_1 - \dot{m}_2 - \dot{m}_3 - \dot{m}_4$

Condenser

$$\dot{m}_6(h_5 - h_6) = \dot{m}_{28}\, C_w (T_{29} - T_{28}) \tag{6.54}$$

Open Heater

$$\dot{m}_8\, h_8 = \dot{m}_4\, h_4 + \dot{m}_7\, h_7 + \dot{m}_{10}\, h_{10} \tag{6.55}$$

Closed Heater-I

$$\dot{m}_3 h_3 + \dot{m}_{11} h_{11} + \dot{m}_{14} h_{14} = \dot{m}_{12} h_{12} + \dot{m}_9 h_9 \quad (6.56)$$

Closed Heater-II

$$\dot{m}_{12} (h_{15} - h_{12}) = m_{16} (h_{16} - h_{13}) \quad (6.57)$$

Solving Eqs. 6.53 to 6.57, the mass flow rates of steam at stations 1 to 15 are obtained. Fig. 6.6 illustrates the various station points from 1 to 15 of the steam cycle of the plant on T-S diagram.

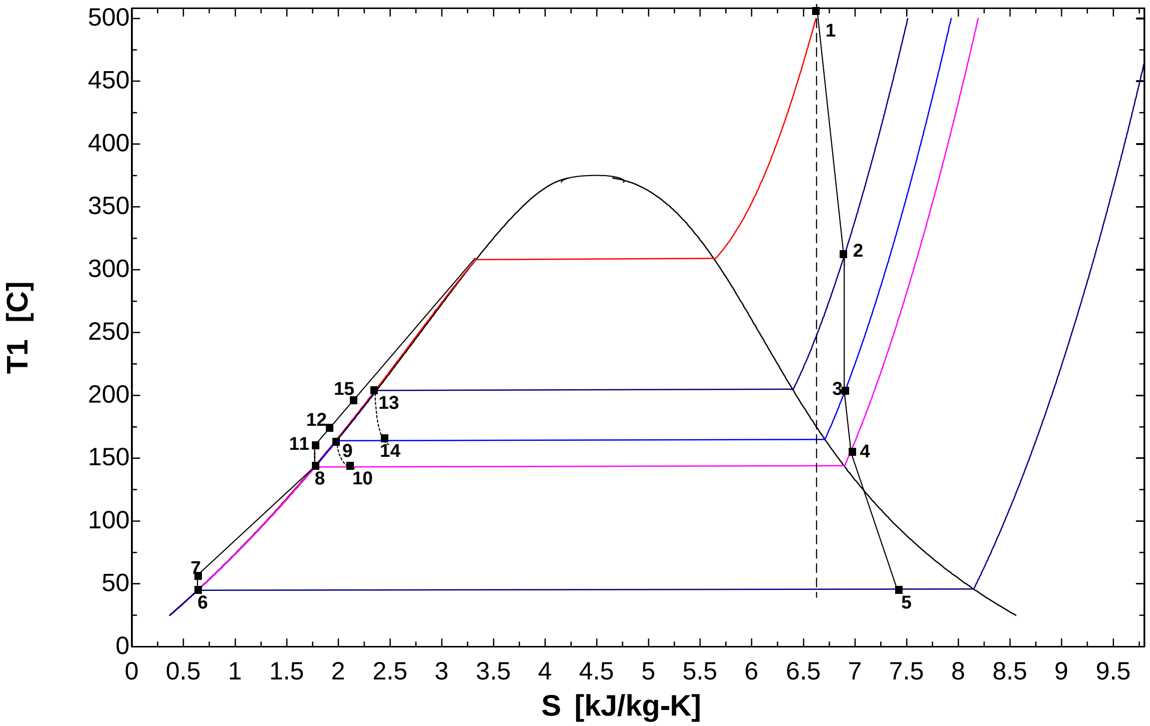


Fig. 6.6 T-S Diagram of Steam Flow Through Steam Turbine

Steam flow from tapping from steam turbine at 17 bar (stream 2) is distributed in two flow, 3.2 kg/sec steam is proposed to divert to AAVAR system as a fuel (station 17) while the remaining steam is supplied to open heater (station 16). Steam from station 17 is throttled to 15 bar (station 18) which is the designed pressure of steam as a fuel for AAVAR system. It is assumed that all the latent heat of steam is consumed in the generator of AAVAR system and condensate comes out at station 19. The condensate is pressurized up to the pressure of 133 bar (station 20) with the help of pump-3 and mixed with the stream flow 15. Mixing of both the streams gives stream 21 which is supplied back to the boiler. From the known properties of steam, the enthalpy and entropy of steam at stations 16 to 21 is estimated with the help of EES software.

Boiler

The ultimate analysis of the coal used in the boiler is given in Table 6.16. Using the percentage of each element, the stoichiometric air fuel ratio is calculated. Considering 20 % excess air supply, the actual air fuel ratio is calculated as explained below.

$$\begin{aligned} &5.925C+2.53H_2+0.281O_2+0.05N_2+0.181S+Ash+a(O_2+3.76N_2)\\ &\rightarrow bCO_2+cH_2O+dSO_2+eN_2 \end{aligned} \tag{6.58}$$

From Eq. 6.58

$C:5.925=b$

$H_2:2.53=c$

$S:0.181=d$

$O:0.281\times2+a\times2=2\times b+c+2\times d$

$N:0.05\times2+a\times2\times3.76=2\times e$

Table 6.16 Analysis of the Coal Used

Element	% m_i kg	M_i	$n_i=m_i/M_i$
C	71.1	12.00	5.925
H_2	5.1	2.016	2.530
O_2	9.0	32.00	0.281
N_2	1.4	28.01	0.050
S	5.8	32.06	0.181
Ash	7.6	__	__
	100		

Solving the above relations, $a=7.09$ and $\mathrm{e}=26.71$. Then the combustion equation with stoichiometric air will be

$$\begin{aligned} &5.925C+2.53H_2+0.281O_2+0.05N_2+0.181S+Ash+7.09(O_2+3.76N_2)\\ &\rightarrow 5.925CO_2+2.53H_2O+0.181SO_2+26.71N_2+Ash \end{aligned} \tag{6.59}$$

With 20% excess air, the combustion analysis will be

$$\begin{aligned}&5.925C + 2.53H_2 + 0.281O_2 + 0.05N_2 + 0.181S + Ash + 7.09\times 1.2(O_2 + 3.76N_2)\\&\rightarrow bCO_2 + cH_2O + dSO_2 + eN_2 + fO_2\end{aligned} \quad (6.60)$$

From the above equation

$C: 5.925 = b$

$H_2: 2.53 = c$

$S: 0.181 = d$

$O: 0.281\times 2 + 7.09\times 2\times 1.2 = 2\times b + c + 2\times d + 2\times f$

$N: 0.05\times 2 + 7.09\times 2\times 3.76 \ \times 1.2 = 2\times e$

Solving the above relations, $f = 1.419$ and $\mathrm{e} = 32.04$. Then the combustion equation with 20 % excess air will be

$$\begin{aligned}&5.925C + 2.53H_2 + 0.281O_2 + 0.05N_2 + 0.181S + Ash + 8.51(O_2 + 3.76N_2)\\&\rightarrow 5.925CO_2 + 2.53H_2O + 0.181SO_2 + 1.414O_2 + 32.04N_2\end{aligned} \quad (6.61)$$

Using Eq. 6.61, the mole fraction of each element of combustion product is calculated and is given in Table 6.17.

Table 6.17 Analysis of Combustion Product

Element	**Mole** n	**Mole fraction** $x = n/\sum n$	**Molecular weight, M**	**Mass of element** $m = n\times M$
CO_2	5.925	0.1408	44	260.70
H_2O	2.530	0.0601	18	45.54
SO_2	0.181	0.0043	64	11.58
O_2	1.414	0.0337	32	45.44
N_2	32.040	0.7611	28	897.12
$\sum n$	**42.096**	**1.0000**		**1260.38**

The mass of air supplied for 100 kg coal is given by

$m_{air} = 8.51\times\left(32 + 3.76\times 28\right) = 1168.25kg$

Air fuel ratio

$A/F = {m_{air}}/{m_F}$

$A/F = {1168.25}/{100} = 11.68$

Mass flow rate of exhaust gas =1260.38 kg for 100 kg coal.

The gross calorific value (GCV_{coal}) of the coal, measured using calorimeter is found to be 23 MJ/kg.

Forced Draught Fan

The forced draught fan used in the boiler creates draught of 472 mmWC. The temperature of environment air (T_0) and isentropic efficiency of FD fan are taken as 298.1K and 80%, respectively. The temperature of air at the exit of FD fan can be estimated using

$$T_{23} = T_0 \left\{ 1 + \frac{1}{\eta_{FD}} \left[\left(\frac{p_{23}}{p_0} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right] \right\} \tag{6.62}$$

At this temperature, T_{23}, the enthalpies of all the constituents, nitrogen, oxygen, carbon dioxide and water vapour are calculated using the Eq. F2 of Appendix F. Then, enthalpies of all the constituents are added on molar basis and the enthalpy of the air inlet to compressor is calculated on molar basis using

$$h'_{23} = 0.7748 h_{\mathrm{N_2}}(T_{23}) + 0.2059 h_{\mathrm{O_2}}(T_{23}) + 0.0003 h_{\mathrm{CO_2}}(T_{23}) + 0.019 h_{\mathrm{H_2O}}(T_{23}) \tag{6.63}$$

The enthalpy of air on mass basis is, then, calculated using

$$h_{23} = h'_{23} / M_a \tag{6.64}$$

For the existing case, the temperature of combustion product is taken as 1500 K. Then the enthalpy of combustion product is estimated using

$$h_{24} = x_{N_2} h_{N_2}(T_{24}) + x_{O_2} h_{O_2}(T_{24}) + x_{CO_2} h_{CO_2}(T_{24}) + x_{H_2O} h_{H_2O_g}(T_{24}) + x_{SO_2} h_{SO_2}(T_{24}) \tag{6.65}$$

The combustion product traverse through evaporation zone during which 170 mmWC pressure drops takes place and temperature of gas at the exit of evaporation zone is found to be 160ºC during normal operation of the plant. The enthalpy of combustion product at station 25 is given by

$$h_{25} = x_{N_2} h_{N_2}(T_{25}) + x_{O_2} h_{O_2}(T_{25}) + x_{CO_2} h_{CO_2}(T_{25}) + x_{H_2O} h_{H_2O_g}(T_{25}) + x_{SO_2} h_{SO_2}(T_{25}) \tag{6.66}$$

Induced Draught Fan

The ID fan creates draught of 230 mmWC. The temperature and enthalpy of the gas at the exit of ID fan is found for isentropic efficiency of ID fan, $\eta_{ID} = 80\%$, at 80 % using

$$T_{26} = T_{25}\left\{1+\frac{1}{\eta_{ID}}\left[\left(\frac{p_{26}}{p_{25}}\right)^{\frac{\gamma_a-1}{\gamma_a}}-1\right]\right\} \tag{6.67}$$

$$h_{26} = h_{25} + W_{P2} \tag{6.68}$$

The work done by the FD fan and ID fan are estimated using

$$W_{\text{FDfan}} = \frac{\Delta P_{FDfan} \times V_0 \times T_0 \times A/F \times m_f}{273.15 \times \eta_{FDfan}} \tag{6.69}$$

$$W_{\text{IDfan}} = \frac{\Delta P_{\text{IDfan}} \times V_0 \times T_{25} \times A/F \times (m_f + 1)}{273.15 \times \eta_{\text{IDfan}}} \tag{6.70}$$

where, $V_0 = 0.7835 m^3$ is the volume of air at NTP and $T_0 = 298.1K$

Using the enthalpy balance in the evaporation zone of the boiler, the mass flow rate of gas (m_g) in the boiler is calculated for the given rate of power generation.

$$m_1(h_1 - h_{21}) = m_g(h_{24} - h_{25}) \tag{6.71}$$

After calculating the mass flow rate of gas, mass flow rate of fuel can be found as the mass of exhaust gas for 100 kg coal combustion is available from Table 6.17. Using the value of Air Fuel ratio and flow rate of fuel, flow rate of air can be estimated for the given rate of power generation.

The entropy of steam and water at stations 1 to 21 is calculated using the in built subroutine of the EES software at the given temperature and pressure. The air at station 23 is at pressure other than p_{ref}. Then entropy of air at temperature T_{23} and p_{23} is calculated using

$$s'_{23} = 0.7748\left[s_{N_2}(T_{23}) - R*\ln\left(\frac{0.7748*p_{23}}{p_0}\right)\right] + 0.2059\left[s_{O_2}(T_{23}) - R\ln\left(\frac{0.2059*p_{23}}{p_0}\right)\right] + 0.0003\left[s_{CO_2}(T_{23}) - R\ln\left(\frac{0.0003*p_{23}}{p_0}\right)\right] + 0.019\left[s_{H_2O_g}(T_{23}) - R\ln\left(\frac{0.019*p_{23}}{p_0}\right)\right] \tag{6.72}$$

$$s_{23} = \frac{s_{23}^{'}}{M_a} \text{ (kJ/kg)} \tag{6.73}$$

At station 24, combustion product is considered as ideal gas mixture. Table 6.17 gives the mole fraction of all the constituents. Using the mole fractions, the entropy of combustion product at station 24 is found using following relation in terms of kJ/kmol.

$$\begin{aligned} s_{24}^{'} = {} & x_{N_2}\left[s_{\mathrm{N_2}}(T_{24}) - R\ln\left(\frac{x_{N_2} * p_{24}}{p_0}\right)\right] + x_{O_2}\left[s_{\mathrm{O_2}}(T_{24}) - R\ln\left(\frac{x_{O_2} * p_{24}}{p_0}\right)\right] + \\ & x_{CO_2}\left[s_{\mathrm{CO_2}}(T_{24}) - R\ln\left(\frac{x_{CO_2} * p_{24}}{p_0}\right)\right] + x_{H_2O_g}\left[s_{\mathrm{H_2O_g}}(T_{24}) - R\ln\left(\frac{x_{H_2O_g} * p_{24}}{p_0}\right)\right] + \\ & x_{SO_2}\left[s_{\mathrm{SO_2}}(T_{24}) - R\ln\left(\frac{x_{SO_2} * p_{24}}{p_0}\right)\right] \end{aligned} \tag{6.74}$$

$$s_{24} = \frac{s_{24}^{'}}{M_p} \text{ (kJ/kg)} \tag{6.75}$$

Similarly, the entropy at stations 25 and 26 are calculated at corresponding temperature and pressure. It should be noted that station 22 represents fuel (coal) at boiler inlet, station 27 represents the rate of power generation while the stations 28 and 29 represents cooling water inlet and exit to condenser.

6.2.2 Exergy Analysis

The theoretical description of exergy and its components is given in Chapter 4. In this section, the estimation of the two components of exergy, viz. physical and chemical exergy for each station is given.

6.2.2.1 Physical Exergy

For the stations 1 to 21, the working fluid is either steam or water. At $T_{ref} = 25^o C$ and $p_{ref} = 1.01325$ bar , their enthalpy h_0 and entropy s_0 are found using EES software. Then the physical exergy at stations 1 to 21 is found using Eq 4.2.

Station 23 represents the exit condition of air at FD fan which is the inlet to the combustion chamber of the boiler. To calculate the exergy of air at station 23, its enthalpy and entropy at T_0 and P_0 are found using the following:

$$h'_{0a} = 0.7748h_{N_2}(T_0) + 0.2059h_{O_2}(T_0) + 0.0003h_{CO_2}(T_0) + 0.019h_{H_2O}(T_0) \qquad (6.76)$$

The molecular weight of the air inlet to combustion chamber of the boiler is calculated using

$$M_a = 0.7748M_{N_2} + 0.2059M_{O_2} + 0.0003M_{CO_2} + 0.019M_{H_2O} \qquad (6.77)$$

Using these values, the enthalpy of air on mass basis is calculated using

$$h_{0a} = h'_{0a} / M_a \qquad (6.78)$$

$$s'_{0a} = 0.7748s_{N_2}(T_0) + 0.2059s_{O_2}(T_0) + 0.0003s_{CO_2}(T_0) + 0.019s_{H_2O}(T_0) \qquad (6.79)$$

$$s_{0a} = \frac{s'_{0a}}{M_a} \text{ (kJ/kg)} \qquad (6.80)$$

Using enthalpy and entropy of air at exit of FD fan and at reference state, the exergy at station 23 is found using Eq. 4.2

To calculate the exergy of combustion product and exhaust gas from the boiler, it is considered that they are reduced to $T_{ref} = 25^oC$ and $p_{ref} = 1.01325 \text{ bar}$. At this temperature, some condensation of water will occur and gas phase containing saturated water vapour in equilibrium with saturated liquid water phase. On the basis of 1 kmol of combustion products formed, the gas phase at 25ºC would consists of 0.9399 kmol of dry products (0.7611 N_2, 0.0337 O_2, 0.1408 CO_2, 0.0043 SO_2) plus n_v kmol of water vapour. The partial pressure of water vapour would be equal to the saturation pressure, $p_g(25^oC) = 0.0317 \text{ bar}$. The amount of water vapour is found using

$$p_v = x_v p \qquad (6.81)$$

$$0.0317 \text{ bar} = \frac{n_v}{0.9399 + n_v}(1.01325 \text{ bar}) \qquad (6.82)$$

Solving Eq. 6.82, $n_v = 0.03035 \text{ kmol}$.

Thus, the composition of the combustion product as given in Table 6.17 is to be modified for the condition at 25ºC and 1 atm and is given as under:

$\underline{0.7611\ N_2, 0.0337\ O_2, 0.1408\ CO_2, 0.0043SO_2, 0.03035\ H_2O\ (g)}$, 0.02975$H_2O$ (l).

The underline indicates the gas phase. Enthalpy of combustion product at 25ºC and 1 atm is given

$$h'_{024} = x_{N_2}h_{N_2}(T_0) + x_{O_2}h_{O_2}(T_0) + x_{CO_2}h_{CO_2}(T_0) + x_{SO_2}h_{SO_2}(T_{25}) + x_{H_2O}h_{H_2O_g}(T_0) + x_{H_2O}h_{H_2O_l}(T_0) \tag{6.83}$$

$$h_{024} = \frac{h'_{024}}{M_p}\ \text{kJ/kg} \tag{6.84}$$

$$s'_{024} = (x_{N_2}s_{N_2}(T_0) + x_{O_2}s_{O_2}(T_0) + x_{CO_2}s_{CO_2}(T_0) + x_{H_2O_g}s_{H_2O_g}(T_0) + x_{H_2O_l}s_{H_2O_l}(T_0))$$

$$s_{024} = \frac{s'_{024}}{M_p}\ \text{kJ/kgK} \tag{6.85}$$

Now, the physical exergy at station 24 is calculated using Eq. 4.2. Using the same value of enthalpy and entropy at reference condition, physical exergy at station 25 and 26 is also calculated.

6.2.2.2 Chemical Exergy

At the station 1 to 21, the working fluid is steam or water. When it is brought to the equilibrium with the atmosphere, it will be in liquid state. Chemical exergy of water as selected from Appendix G.

$$e^{CH}_{water} = 45\ \text{kJ/kmol} \tag{6.86}$$

At station 23, air is stable with environment so its chemical exergy is considered as zero. At dead state corresponding to the mixture at stations 24 to 26 consists of liquid water phase and a gas phase. The new mole fraction of a gas phase is calculated as,

$$y_{N_2} = \frac{x_{N_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v} \qquad y_{O_2} = \frac{x_{O_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v}$$

$$y_{CO_2} = \frac{x_{CO_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v} \qquad y_{H_2O(g)} = \frac{n_v}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v} \tag{6.87}$$

$$y_{SO_2} = \frac{x_{SO_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v}$$

They are found as

$y_{N_2} = 0.7844,\ y_{O_2} = 0.03473,\ y_{CO_2} = 0.1451, y_{SO_2} = 0.004432,\ y_{H_2O(g)} = 0.03129$

Table 6.18 State Properties for Steam Power Plant

Stations	Mass flow rate kg/sec	Pressure. bar	Temp. °C	Specific Enthalpy kJ/kg	Specific Entropy kJ/kgK	Physical Exergy kW	Chemical Exergy kW	Total Exergy MW
1	56.86	96.00	500.00	3379.00	6.6210	80161.0	142.10	80.30
2	7.08	17.00	286.40	3000.00	6.7970	6928.00	17.70	6.95
3	3.11	7.00	205.20	2856.00	6.9100	2488.00	7.77	2.50
4	5.20	4.00	160.00	2775.00	6.9810	3633.00	13.01	3.65
5	41.47	0.10	45.82	2353.00	7.4250	5973.00	103.70	6.08
6	41.47	0.10	45.79	191.70	0.6489	116.90	103.70	0.22
7	41.47	4.00	45.88	192.10	0.6489	133.30	103.70	0.24
8	53.66	2.00	120.20	504.70	1.5300	2850.00	134.10	2.98
9	6.99	6.00	158.80	670.60	1.9310	694.60	17.47	0.71
10	6.99	4.00	143.60	669.20	1.9310	684.90	17.47	0.70
11	53.66	135.00	123.60	518.80	1.5300	3607.00	134.10	3.74
12	53.66	134.00	154.00	657.50	1.8680	5641.00	134.10	5.78
13	3.88	15.00	198.30	844.80	2.3150	617.80	9.70	0.63
14	3.88	7.00	165.00	838.50	2.3150	593.60	9.70	0.60
15	53.66	133.00	190.00	813.30	2.2190	8396.00	134.10	8.53
16	3.88	17.00	286.40	3000.00	6.6210	3999.00	9.70	4.01
17	3.20	17.00	286.40	3000.00	6.6210	3299.00	8.00	3.31
18	3.20	15.00	198.30	2791.00	6.4440	2800.00	8.00	2.81
19	3.20	15.00	198.30	844.80	2.3150	509.70	8.00	0.52
20	3.20	133.00	201.60	858.40	2.3150	553.30	8.00	0.56
21	56.86	133.00	190.60	815.80	2.2240	8948.00	142.10	9.09
22	8.99	1.06	25.00	--	--	0	206863	206.90
23	105.10	1.06	29.90	-159.60	6.9640	528.50	0	0.53
24	113.40	1.013	1227.00	-951.10	8.6080	97601	9263	106.90
25	113.40	1.00	160.00	-2237.00	7.1370	1565.00	9263	10.83
26	113.40	1.019	163.50	-2233.00	7.14	1879	9263	11.14
27	--	--	--	--	--	--	--	50
28	2555.00	1.013	33.00	138.30	0.4777	1141	6388	7.53
29	2555.00	1.013	41.38	173.30	0.5906	4666	6388	11.05

Now the chemical exergy for the k^th component is calculated and added together to find total chemical exergy using following equation

$$e^{CH} = \sum y_k e_k^{CH} + RT_0 \sum y_k \ln y_k \tag{6.88}$$

This is the chemical exergy of gas portion. The chemical exergy of liquid portion is separately calculated and added together to find total chemical exergy. The chemical exergy of individual component (e_k^{CH}) is taken from Appendix G.

$$\dot{E}^{CH} = \dot{m}_g\left[\left(x_{N_2}+x_{O_2}+x_{CO_2}+x_{SO_2}+n_v\right)e^{CH}+x_{H_2O(l)}*e^{CH}_{H_2O(l)}\right] \qquad (6.89)$$

This is the chemical exergy in kJ/kmol. It is then converted in kJ/kg by dividing it by molecular mass of the combustion product. The chemical exergy at stations 24 to 26 will remain same. Standard chemical exergy of coal is taken equal to its GCV. The total exergy flow at all the stations will be the sum of physical and chemical exergy. Table 6.18 gives state properties and total exergy along with its components for various stations from 1 to 29.

6.2.2.3 Definition of Fuel, Product and Loss for Various Processes

For all the components of the steam turbine power plant, fuel, product and loss are defined as given in Chapter 4, Section 4.1.1. They are summarized in Table 6.19 and calculated values are given Table 6.20.

Table 6.19 Fuel, Product and Loss for various Components of Steam Power Plant

Component	Fuel ($\dot{E}_F$)	Product ($\dot{E}_P$)	Loss ($\dot{E}_L$)
Boiler Furnace	$\dot{E}_{22}+\dot{E}_{23}+\dot{E}_{26}-\dot{E}_{25}$	$\dot{E}_{24}$	0
Boiler HX	$\dot{E}_{24}-\dot{E}_{25}$	$\dot{E}_{1}-\dot{E}_{21}$	0
Steam Turbine	$\dot{E}_{1}-(\dot{E}_{2}+\dot{E}_{3}+\dot{E}_{4}+\dot{E}_{5})$	$\dot{E}_{27}^{W}$	0
Turbine Cond. Assly.	$\dot{E}_{1}-\dot{E}_{2}-\dot{E}_{3}-\dot{E}_{4}-\dot{E}_{6}$	$\dot{E}_{27}^{W}$	0
Condenser	—	—	0
Overall System	$\dot{E}_{22}+W_{FD}+W_{ID}$	$\dot{E}_{27}^{W}$	$\dot{E}_{26}+(\dot{E}_{29}-\dot{E}_{28})$

Table 6.20 Exergy Analysis of Steam Power Plant

Component	$\dot{E}_F$ MW	$\dot{E}_p$ MW	$\dot{E}_L$ MW	$\dot{E}_D$ MW	Y_D %	Y_L %	Y_D^* %	ε %
Boiler Furnace	207.70	106.90	0	100.80	48.53	0	70.46	51.45
Boiler HX	96.04	71.21	0	24.82	11.95	0	17.34	74.15
Steam Turbine	61.14	50.00	0	11.14	5.36	0	7.78	81.78
Turbine Cond. Assly.	67.00	50.00	0	17.00	8.18	0	11.87	74.63
Condenser	--	--	0	5.86	2.82	0	4.09	60.20
Overall System	207.80	50.00	14.67	143.10	68.88	7.06	100	24.06

6.2.2.4 Results and Discussions

The outcome of the exergy analysis of steam turbine power plant is given in Table 6.20. The total exergy supplied to the system is 207.80 MW. Out of which 50 MW (24.06 %) is converted to useful product. 143.10 MW (68.88 %) exergy is destroyed and 14.67 MW (7.06 %) is lost to the environment. The maximum exergy destruction is observed in boiler furnace. To reduce the exergy destruction in boiler furnace, the furnace temperature should be increased. For that, turbulence can be created and better air preheater can improve the performance. The next component in this category is boiler heat exchanger. To reduce the exergy destruction in heat exchanger, effectiveness of the same can be improved.

6.2.3 Exergoeconomic Analysis

The economic analysis of thermal system requires the identification and inclusion of various cost heads incurred in the estimation of the total cost for the production. In the present case, the total cost involved in the power generation of steam turbine consists of many cost heads. Thus, in general, the economic analysis of the system requires the estimation of levelized O & M cost of component ($\dot{Z}_k$) and fuel cost rate ($\dot{C}_f$). $\dot{Z}_k$ should be estimated for each component for steam power plant using TCI, β, γ and τ (Refer Eq.4.18). The fuel cost rate ($\dot{C}_f$) is governed by the source of heat energy used for the system. The estimation of $\dot{Z}_k$ and $\dot{C}_f$ are explained in the following section.

6.2.3.1 Levelized O&M Cost

For estimation of the cost of boiler, turbine, condenser and pumps, the cost models suggested by Silveira et al. [139] are used and are given in Appendix H. These cost models gives the total capital investment including the installation cost, electrical equipment cost, control system cost, piping cost and local assembly cost. Using the Marshall & Swift cost index, they are converted for the year 2009. The operation and maintenance cost of each component is found using Eq. 4.18 in which the plant life is considered as 8000 hours, Capital Recovery Factor (β) = 0.1061, Operation and Maintenance cost, $\gamma = 1.092\ \%$ of total capital investment. The values of operation and maintenance cost ($\dot{Z}_k$) for each component are given in Table 6.22.

6.2.3.2 Fuel Cost

The plant uses coal as a fuel. The market price of coal for the year 2009 was ₹ 3000 per 1000 kg. So cost of fuel is considered as ₹ 3/kg coal.

6.2.3.3 Cost Flow

Applying the formulation of cost balance equations and the definition of fuel, product and loss (Refer Table 6.20); the exergoeconomic cost balance equations for each component of steam power plant are formulated in the following forms:

Considering boiler, turbine and turbine condenser assembly as a control volume, following cost balance equations are modelled.

Boiler

$$\dot{c}_{22}\dot{E}_{22}+\dot{c}_{23}\dot{E}_{23}+\dot{c}_{23}(\dot{E}_{26}-\dot{E}_{25})-\dot{c}_{25}\dot{E}_{25}-\dot{c}_{1}\dot{E}_{1}+\dot{Z}_{BL}=0 \qquad (6.90)$$

$$\dot{c}_{22}=\dot{c}_{25} \qquad (6.91)$$

$$\dot{c}_{24}=\dot{c}_{25} \qquad (6.92)$$

$$\dot{c}_{25} = \dot{c}_{26} \tag{6.93}$$

$$\dot{c}_{23} = \dot{c}_{27} \tag{6.94}$$

Steam Turbine

$$\dot{c}_1 \dot{E}_1 - \dot{c}_2 \dot{E}_2 - \dot{c}_3 \dot{E}_3 - \dot{c}_4 \dot{E}_4 - \dot{c}_5 \dot{E}_5 - \dot{c}_{27} \dot{E}_{27} + \dot{Z}_{ST} = 0 \tag{6.95}$$

$$\dot{c}_1 = \dot{c}_2 \tag{6.96}$$

$$\dot{c}_1 = \dot{c}_3 \tag{6.97}$$

$$\dot{c}_1 = \dot{c}_4 \tag{6.98}$$

$$\dot{c}_1 = \dot{c}_5 \tag{6.99}$$

Out of these variables, $\dot{c}_1 ... \dot{c}_5$ and $\dot{c}_{22} ... \dot{c}_{27}$, the fuel cost $\dot{c}_{22}$ is known. The remaining 10 are calculated by solving Eqs. 6.90 to 6.99 using EES software. The cost per unit exergy (₹/MJ) and cost flow rate (₹/sec) for each flow of the system are calculated and shown in Table 6.21. For this calculation, known values of $\dot{E}_1$ to $\dot{E}_5$ and $\dot{E}_{22}$ to $\dot{E}_{27}$ are used.

Table 6.21 Unit Exergy Cost and Cost Flow Rate for Steam Power Plant

Flows	Unit exergy cost ₹/MJ	Exergy flow MW	Cost flow rate ₹/sec
1	0.4025	80.300	32.320
2	0.4025	6.945	2.796
3	0.4025	2.495	1.004
4	0.4025	3.646	1.468
5	0.4025	6.077	2.446
22	0.1319	204.500	26.980
23	0.5540	0.529	0.293
24	0.1319	106.900	14.100
25	0.1319	10.830	1.429
26	0.1319	11.140	1.470
27	0.5540	50.000	27.700

6.2.4 Exergoeconomic Evaluation

Solution of the cost balance equations will give the cost flow rates at each station of the plant and cost rate of product ($\dot{C}_{p,k}$) using cost rate of fuel as an input ($\dot{C}_{F,k}$). After that, cost rate of fuel per unit exergy ($c_{F,k}$), cost rate of product per unit exergy ($c_{P,k}$), cost rate of exergy destruction ($\dot{C}_{D,k}$), cost rate of exergy loss ($\dot{C}_{L,k}$), the relative cost difference (r_k) and exergoeconomic factor (f_k) for each components are calculated using Eqs. 4.20 to 4.27 and given in Table 6.22.

Table 6.22 Results of Exergoeconomic Analysis

Component	$c_{F,k}$ ₹/MJ	$c_{p,k}$ ₹/MJ	$\dot{C}_{D,k}$ ₹/hr	$\dot{C}_{L,k}$ ₹/hr	$\dot{Z}_k$ ₹/hr	f_k %	r_k %	ε %
Boiler furnace	0.13	0.13	47167	0	22695	27.80	241.30	34.68
Boiler HX	0.13	0.45	11790	0				
Turbine	0.40	0.55	16143	0	11117	40.80	37.62	81.78
Turbine Condenser Assembly	0.40	0.55	24710	0	11187	31.20	37.17	74.63
Condenser	--	--	8567	0	70	0.80	--	60.20
System	0.48	0.55	242144	25227	34328	11.40	15.95	24.34

6.2.4.1 Results and Discussions

The following observations are made from the exergoeconomic analysis of steam power plant with regeneration shown in Table 6.22.

(i) The r value for the boiler is found highest among all the components. The boiler has lowest exergetic efficiency. In combustion chamber of the boiler, the maximum exergy destruction is observed from the Table 6.20. It suggests that the temperature of the combustion product should be increased by modifying the boiler design.

(ii) In the evaporation zone of the boiler, the next highest exergy destruction is observed from the Table 6.20. It suggests that the boiler pressure and temperature should be increased. The turbine is having the next highest r value and exergy destruction cost. It suggests that the isentropic efficiency of steam turbine should be increased by increasing the investment cost.

(iii) The condenser is having very low f value and higher exergetic efficiency. It suggests that the condenser of the plant working properly as it is having less investment cost and less exergy destruction.

6.2.5 Exergoeconomic Optimization

The exergy analysis suggests improvement in the thermal system which is associated the increase in investment and Operation and maintenance cost. These two are conflict in nature. The exergoeconomic optimization provides optimum condition between improvement in thermal performance of the system and increase in the cost.

6.2.5.1 Estimation of B_k, n_k and m_k

Using the value of cost flow at each station and the results of exergoeconomic evaluation, the exergoeconomic optimization of the system is carried out at component level using Eq. 4.29. To solve this equation for local optimum by curve fitting technique, the equivalent power law is found and the required value of B_k and n_k for each component are determined for the selected value of m_k as explained below.

Boiler

For boiler, the temperature and pressure of the steam generated by the boiler are considered as the decision variables. With the variation of temperature and pressure of steam generated in the boiler, the variation of exergetic efficiency of the boiler and total capital investment are generated in the form explained in section 4.3 and given in Table 6.23. The required graph is plotted as shown in Fig. 6.7. By curve fitting technique, the required power law is developed as shown in the Fig. 6.7. The figure shows that the value of B_{BL} and n_{BL} are found to be 1.36×10^7 and 4.5598 for the selected value of m_{BL} of 0.78 as suggested by Bejan et al. [155].

Table 6.23 Generated Data Using Investment Cost Equation for Boiler

p_{ST} bar	T_{ST} °C	$\dot{E}_{P,BL}/\dot{E}_{D,BL}$	$TCI_{BL}/\dot{E}_{p,BL}^{0.78}$
94	490	0.5734	1078000
96	494	0.5760	1100000
98	498	0.5786	1123000
99	502	0.5804	1139000
100	504	0.5817	1151000

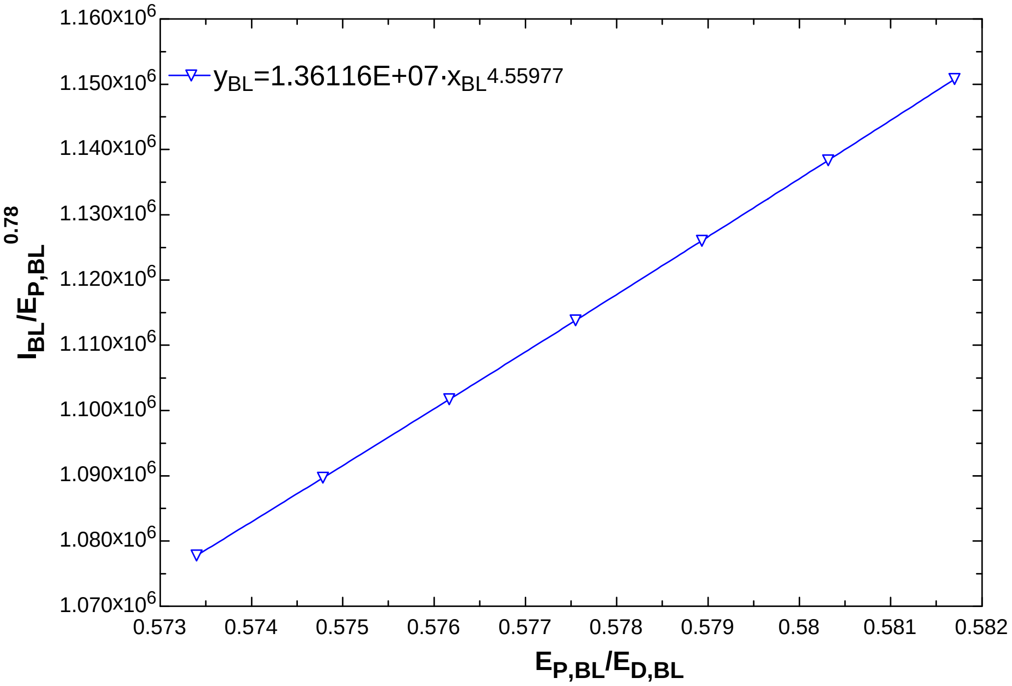


Fig. 6.7 Plot of Investment cost v/s Exergetic Efficiency for Boiler

Steam turbine

For steam turbine, the isentropic efficiency is considered as the decision variable. Parametric variation of various properties with respect to isentropic efficiency is carried out and the following Table 6.24 is generated and the graph of investment cost v/s exergetic efficiency is plotted for the steam turbine as shown in Fig. 6.8 with the required power law through curve fitting technique. The figure shows that the value of B_{ST} and n_{ST} are found to be 364648 and 0.1384, respectively for the selected value of m_{ST} of 0.9 as suggested by Bejan et al. [155].

Table 6.24 Generated Data Using Investment Cost Equation for Steam Turbine

η_{ST}	$\dot{E}_{P,ST}$	$\dot{E}_{D,ST}$	$\dot{E}_{P,ST}/\dot{E}_{D,ST}$	$TCI_{ST}/\dot{E}_{p,ST}^{0.9}$
0.75	14.85	50	3.3660	430252
0.76	14.07	50	3.5530	434163
0.77	13.31	50	3.7560	438057
0.78	12.57	50	3.9780	441935
0.79	11.85	50	4.2210	445798
0.80	11.14	50	4.4880	449645
0.81	10.45	50	4.7840	453477
0.82	9.78	50	5.1120	457294
0.83	9.13	50	5.4790	461096
0.84	8.49	50	5.8900	464883

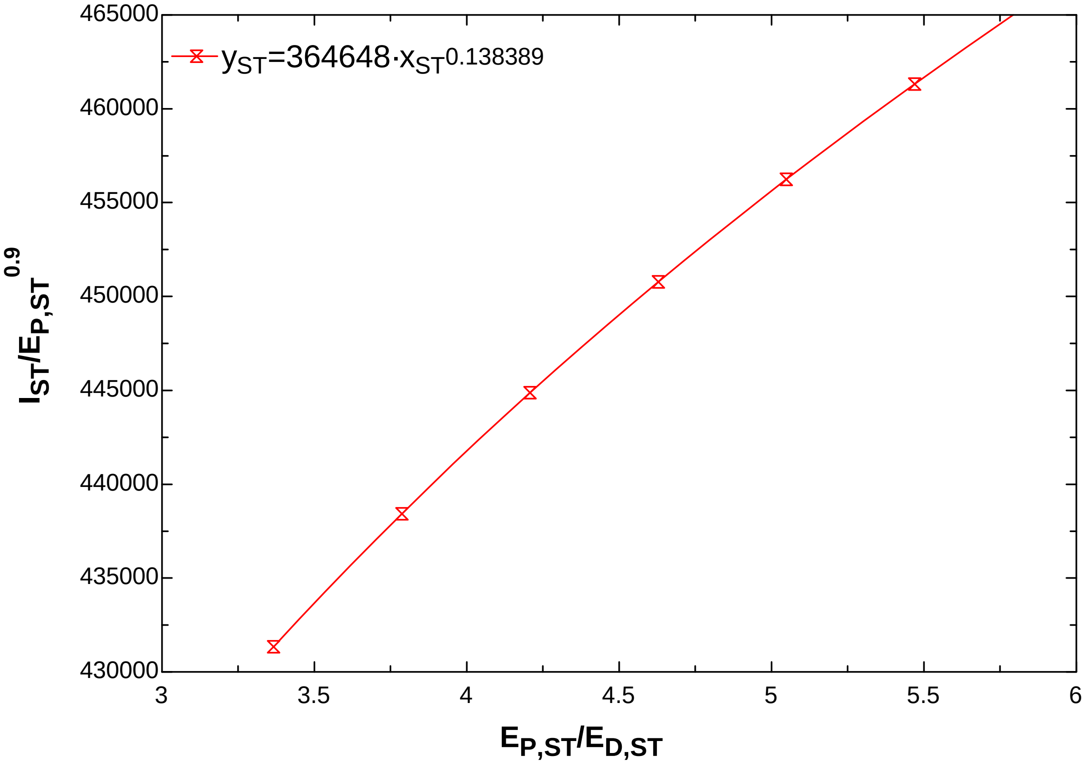


Fig. 6.8 Plot of Investment Cost v/s Efficiency for Steam Turbine

Table 6.25 summarises the component-wise parameters, B_k, n_k and m_k estimated along with the decision variable.

Table 6.25 Values of B_k, n_k and m_k

Component	Decision variable	B_k	n_k	m_k
Boiler	p_{BL} & T_{BL}	1.36 x 10^7	4.5598	0.78
Steam Turbine	η_{ST}	364648	0.1384	0.90

6.2.5.2 Optimisation Through Case by Case Iterative Procedure

Optimum values of exergetic efficiency (ε_k^{OPT}), the capital investment (Z_k^{OPT}), the relative cost difference (r_k^{OPT}) and the exergoeconomic factor (f_k^{OPT}) can be calculated using Eqs. 4.37, 4.45, 4.46 and 4.47, respectively. Through an iterative optimization procedure, optimum solution can be achieved, with the help of calculated values of $\dot{C}_{P,tot}, \dot{C}_{D,tot}, \dot{C}_{L,tot}$ and OBF and the guidance provided by the values of $\Delta\varepsilon_k$ and Δr_k, calculated using Eqs. 4.50 and 4.51.

Table 6.26 summarizes the results obtained from the case-by-case iteration carried out starting from the base case (base case is the case evaluated using the data of the existing system) to the optimum case. A total of seven iterative cases are presented and the resulting cases are given as cases I to VII out of which the case VI is found to be the optimum. Each of these cases is obtained through a series of study of positive or negative effects on $\dot{C}_{p,tot}$ and $\dot{C}_{D+L}$ by varying each decision variable. The change in the decision variables are governed by $\Delta\varepsilon_k$ and Δr_k. The details of the case by case iterative procedure for exergoeconomic optimization of AAVAR system is discussed in the following paragraph and the output given in Table 6.26. In the base case, the unit product cost of electricity is 1.99 ₹/kWh and production cost of steam is 395 ₹/1000 kg.

From base case to case-I

The highest value of Δr_{ST} shows that the product cost of air preheater is very high. It suggests that the isentropic efficiency of the steam turbine should be increased. The isentropic efficiency of the steam turbine is increased from 80% to 85%. With this the cost of electricity is reduced to 1.91 ₹/kWh and cost of steam extracted from the turbine and proposed to be utilized in absorption refrigeration system is reduced to 391 ₹/1000kg steam.

From case-I to case-II

The highest Δr_{BL} suggest that the exergy destruction can be reduced in the evaporation zone of the boiler by increasing the temperature of steam generated. Higher rate of exergy destruction in the evaporation zone of the boiler can be identified from the Table 6.20. In this regards, the temperature of steam is increased from 500ºC to 505ºC. This will result in the increase in the investment cost of boiler and subsequently the cost of electricity and the cost of steam extracted from the turbine are slightly increased. But the higher temperature of steam reduces the cost of exergy destruction which results in reduction of objective function (OBF). Increase in the temperature beyond this is not so effective.

From case-II to case-III

More rises in the steam temperature gives adverse effect on the product cost and on the objective function.

From case-III to case-IV

Further reduction in the exergy destruction in the evaporation zone of the boiler can be carried out by increasing the steam pressure. The steam pressure is increased from 96 bar to 98 bar. With this, the product cost and objective function is slightly reduced

From case-IV to case-V

Further increase in the pressure from 98 bar to 100 bar gives slight increase in the product cost but reduction in the objective function as the cost of exergy destruction is reduced. So this pressure is accepted as optimum one.

From case-V to case-VI

From Table 6.20, it is observed that the exergy destruction is very high in the combustion chamber of the boiler. This exergy destruction can be reduced by the increase in the temperature of the combustion product. Increasing the temperature of combustion product from 1500ºC to 1510ºC, the cost of electricity generated is reduced to 1.91 ₹/kWh and the cost of steam extracted will be 389.3 ₹/1000 kg. Beyond this temperature in the boiler furnace, the ace melting temperature is achieved so accepting this temperature of the combustion product as optimum one.

From case-VI to case-VII

To reduce the temperature difference between combustion product and steam generated in a boiler to reduce the exergy destruction, the temperature of steam is increased to 510ºC. But it is giving adverse effect on the performance of a system. Hence *case VI* is found to be optimum one. With this optimum configuration of steam power plant (*case-VI*), the cost of steam at station 2 is found to be 389 ₹/1000 kg. Using this steam as fuel in AAVAR system, the cost of cooling at evaporator can be reduced to 0.68 ₹/sec.

Table 6.26 Variables Obtained During Exergoeconomic Optimization of Steam Turbine Power Plant (from Base Case to Optimum Case)

Variable	Base case		Case-I		Case-II		Case-III	
p_1	96 bar		96 bar		96 bar		96 bar	
T_1	500ºC		500ºC		505ºC		510ºC	
η_{ST}	0.80		0.85		0.85		0.85	
T_{24}	1500ºC		1500ºC		1500ºC		1500ºC	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
Boiler	-32.46	108.9	-32.38	109	-32.31	109.2	-32.21	109.5
Turbine	-18.15	4936	-13.52	4150	-13.5	4149	-13.5	4151
$\dot{C}_{L,tot}$	25227 ₹/hr		22528 ₹/hr		22513 ₹/hr		22500 ₹/hr	
$\dot{C}_{D,tot}$	242144 ₹/hr		215425 ₹/hr		215287 ₹/hr		215162 ₹/hr	
$\dot{C}_P$	99720 ₹/hr		95580 ₹/hr		95688 ₹/hr		95832 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	367091 ₹/hr		333533 ₹/hr		333488 ₹/hr		333494 ₹/hr	

Table 6.26 *Continue*

Variable	Case-IV		Case-V		Case-VI		Case-VII	
p_1	98 bar		100 bar		100 bar		100 bar	
T_1	505°C		505°C		505°C		510°C	
η_{ST}	0.85		0.85		0.85		0.85	
T_{24}	1500°C		1500°C		1510°C		1510°C	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
Boiler	-32.19	109.5	-32.05	109.8	-31.38	107.5	-31.29	107.9
Turbine	-13.5	4151	-13.5	4155	-13.52	4143	-13.5	4143
$\dot{C}_{L,tot}$	22469 ₹/hr		22431 ₹/hr		22113 ₹/hr		22100 ₹/hr	
$\dot{C}_{D,tot}$	214806 ₹/hr		214393 ₹/hr		210019 ₹/hr		209897 ₹/hr	
$\dot{C}_P$	95796 ₹/hr		95904 ₹/hr		95256 ₹/hr		95400 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	333071 ₹/hr		332728 ₹/hr		327388 ₹/hr		327397 ₹/hr	

6.2.5.3 Results and Discussions

The results of the exergoeconomic optimization of steam power plant are given in Table 6.26. Table 6.27 represents a comparative study of the final cost optimal configuration with that of the existing configuration (base case). It is seen that the overall exergoeconomic cost of the product (electricity) is decreased by about 4.02 % (1.99 ₹/kWh to 1.91 ₹/kWh) with corresponding 4.17 % decrease (0.48 ₹/MJ to 0.46 ₹/MJ) in the fuel cost which resulted from the reduction in consumption of fuel. The cost of tapped steam is reduced from 395 ₹/1000 kg to 389.3 ₹/1000kg. The cost of exergy destruction is also decreased by 13.27 % and that of exergy loss is decreased by 12.34 %. Overall improvement in the system performance is realized by the increase in the exergetic efficiency by 7.64 %. If the taping steam is used a fuel for VAR system then the cooling cost will be reduced from 4853 ₹/hr to 2448 ₹/hr.

Table 6.27 Comparison between Base Case and Optimum Case for Steam Power Plant

Properties	Base Case	Optimum Case	% Improvement
Fuel Cost $\dot{C}_{F,tot}$	0.48 ₹/MJ	0.46 ₹/MJ	4.17
Product Cost $\dot{C}_P$	1.99 ₹/kWh	1.91 ₹/kWh	4.02
Steam Cost $\dot{C}_S$	395 ₹/1000 kg	389.30 ₹/1000 kg	1.45
Loss $\dot{C}_{L,tot}$	25227 ₹ /hr	22113 ₹/hr	12.34
Destruction $\dot{C}_{D,tot}$	242144 ₹/hr	210019 ₹/hr	13.27
Exergetic Efficiency ε	24.34 %	26.18 %	7.64

6.3 Comparison

A one to one comparison of the outcome of the exergoeconomic optimization of the existing AAVAR system using steam from the independent boiler as heat source, the first option of switch over of heat source to steam from HRSG of GT-HRSG system and the second option of switch over of heat source to tapped steam from steam power plant is carried out. The cost of steam generated in independent boiler is found to be 900 ₹/1000kg and thereby the cooling cost of AAVAR system is 1.36 ₹/sec. The alternative first option for steam generation such as GT-HRSG and tapped steam from steam turbine are identified in the fertilizer industry itself and compared in the Table 6.28.

Table 6.28 Comparison of Cost of Cooling for Options of Heat Sources

	GT-HRSG (Option – 1)	Tapped steam (Option – 2)
Steam cost ₹/1000kg	790	389
Cooling cost of AAVAR ₹/sec	1.086	0.68
Cost associated with exergy loss ₹/sec	0.617	1.470
Cost associated with exergy loss ₹/MWs	0.077	0.029

Table 6.28 compares the cooling cost of Option 1 and Option 2 examined in the present study. It is seen that the tapped steam from steam turbine is quite economical as fuel for AAVAR system compared to steam generated at GT-HRSG. The reason behind the difference is the cost of exergy loss from the system. In case of GT-HRSG, the exergy loss takes place in the form of exhaust gas at 177ºC (station 7). The unit exergy cost associated with exergy loss is 0.617 ₹/sec (Refer Table 6.6). As the power generation capacity of GT-HRSG is 8 MW, the cost associated with loss per unit power generation is 0.077 ₹/MWs. While in the case of steam power plant, the exergy loss takes place in the form of exhaust gas from the boiler at 163.5 ºC (station 26). The unit exergy cost associated with exergy loss is 1.47 ₹/sec (Refer Table 6.21). As the power generation capacity of steam power plant is 50 MW, the cost associated with loss per unit power generation is 0.029 ₹/MWs. The low exergoeconomic loss in steam power plant reduces the cost of power generation and tapped steam from steam turbine.

Since the second option of switch over form the existing heat source of the independent boiler to tapped steam of steam power plant is found to be the best techno-economically, it is proposed to switch over from the existing heat source of steam from independent boiler to tapped steam from 50 MW steam power plant. The saving in the steam cost per 1000 kg steam will be 511 ₹/1000 kg. The annual steam consumption in

AAVAR system is 90403200 kg/year. Therefore, the annual saving in the monetary term will be ₹ 46196035/-.

The switch over is possible only by laying down steam pipe to transport steam from the steam power plant to AAVAR plant which is about 1 km apart. To associate AAVAR system with steam power plant which is about 1 km far from AAVAR system, it is required to establish steam pipe line from steam power plant to AAVAR system. The tapping at steam turbine stage at 17 bar is made up of 6 inch carbon steel pipe of A106 Grade-B Seamless Schedule 40 IBR. It is suggested to extend same pipe line up to AAVAR system. The material cost of pipe is ₹ 1008 per meter length (Appendix-I) which includes supporting systems and bends. Therefore, the total cost of pipe for one km will be ₹ 1008000. The insulation cost will be ₹ 450 per meter length of pipe. Therefore, the total cost of insulation on 1 km pipe line will be ₹ 450000. The total installation cost including pipe material cost and insulation cost will be ₹ 1458000. The total saving in steam cost indicates that this installation cost can be recovered in 12 days only.

Chapter 7

Conclusions

The objective of the present research study is to develop a user-friendly method of exergoeconomic optimization method to predict the cost effectiveness of an energy intensive thermal system such as AAVAR system and suggest ways of improving the cost effectiveness from both thermodynamic and economic points of view.

It is a well known fact that means can be found out to improve thermal system performance using the exergy analysis. It is also a well known fact that exergy analyses is well suited for finding the location, cause and true magnitude of the losses to be determined in a thermal intensive system. The analysis enables for more effective utilization of energy resource and thereby having higher exergetic efficiency of thermal system and also estimates the parameters like the exergy destruction and the exergy loss which adds to the hidden cost. If these destruction and losses are to be prevented, the thermal system needs more investment. Thus, the cost of the component of a system or the whole system increases with an increase in its capacity and efficiency. Therefore, it is necessary to correlate the exergy with cost value. It can be carried through exergoeconomic analysis.

The exergoeconomic analysis suggests improvement in the thermal system which is associated with the increase in investment and operation & maintenance cost. Thus the cost optimization problem involves the maximization of thermodynamic performance and minimization of investment cost. These are the contradictory disciplines. The exergoeconomic concept combines them together and develops effective tool for design the thermal system with higher efficiency and lower cost.

A number of exergoeconomic modeling and optimization methods are suggested by various researchers and are applied to various thermally intensive systems. However, a review of literature indicated that a very little interest is shown towards vapour absorption refrigeration system in general and AAVAR systems in particular. This may be due to the fact that AAVAR system is a less popular refrigeration system as compared to vapour compression systems. Further, they are less capital intensive compared to other thermal intensive systems like power plants. However, AAVAR system used with huge chemical industries needs greater attention as slight modification in the system parameters brings substantial savings in energy and production cost. Considering this important observation, the present research work on the optimization of AAVAR system is undertaken.

There are a number of exergoeconomic optimization models available in open literature. However, most of them are either complex to translate or incomplete in their availability in open literature. Thermoeconomic Evaluation and Optimization (TEO) method suggested by Tsatsaronis and his associates is a user friendly method which needed some alterations to suite to the optimization of the industrial brine chilling unit using AAVAR system. In the present work, such modification is suggested.

The brine chilling unit using AAVAR system employed in a large fertilizer plant considered for the present optimization study utilizes heat source from the steam generated in an independent boiler. The optimization study is then extended by considering two other options of heat sources available with the fertilizer industry, viz., steam from GT-HRSG and tapped steam from steam turbine to assess the cooling cost effectiveness of the source of heat energy.

Since a rigorous design optimization of complex energy systems is practically very difficult and time consuming, the exergoeconomic method originally developed by Tsatsarnois and modified in the present work is a valuable and powerful tool in the optimization of complex energy systems by identifying all the cost sources. Optimization is carried out through an iterative procedure rather than through a search of the global optimum of a predetermined function by solving it mathematically. The results show how far is the improved design from the reference design, although based on typical data. The term optimization in this context implies improvement rather than calculation of the global optimum. The notable feature of the present method as compared to that of the simple energy based costing and optimization is the identification of the source of exergy destruction and subsequent corrective measures possible to reduce the same.

The exergoeconomic analysis of a thermal system improves the engineer's understanding of the interactions among the system variables and reveals opportunities for cost-effective improvements in system design by means of changes in the structure of the system, which is not possible through the mathematical or numerical techniques. Moreover, operator's suggestions based on experience can be incorporated. However, the optimization technique requires engineering judgments and critical evaluations at every step of the optimization process such as proper definition of fuel-product-loss for every component, proper selection of local decision variables, but allows the designer to carry out an energy-conscious design.

The following conclusions are derived from the study pertaining to the possible overall improvement in the operation of a brine chilling unit incorporating an AAVAR system, gas turbine power plant with HRSG and steam turbine power plant with regeneration in a fertilizer plant.

1. An overall cost reduction in terms of exergoeconomic product cost (chilled brine) of the order of about 27% and that of fuel cost of the order of about 12.76 % ensures a significant reduction in the consumption of the fuel for the existing brine chilling unit with steam generated in the independent boiler as a heat source when optimum design worked out (i.e. switch over from existing base case to optimum case) using the present technique is incorporated.
2. When the option of heat source for the operation of AAVAR system is steam generated at HRSG of the available gas turbine power plant in the fertilizer plant, the cost of steam is reduced by about 12 % and there by the cooling cost of VAR system is also reduced from 1.36 ₹/sec to 1.086 ₹/sec. The optimization of gas turbine power plant reduces the cost of electricity generated from 2.61 ₹/kWh to 2.49 ₹/kWh and cost of steam generated at HRSG is reduced to 790 ₹/1000 kg compared to 900 ₹/1000 kg with independent boiler.

3. When the option of heat source for the operation of AAVAR system is the tapped steam from steam generated at the steam power plant available in the fertilizer plant, the optimization of steam turbine power plant reduces the cost of electricity generated, from 1.99 ₹/kWh to 1.91 ₹/kWh and the cost of tapped steam is 389 ₹/1000 kg. The reduced steam cost reduces the cooling cost of AAVAR system to 0.68 ₹/sec.
4. If the findings from the present study is to be incorporated in to the plant, then switching over from independent boiler and to tapped steam of steam power plant as heat source, steam is to be transported a distance of about 1 km as steam power plant is housed 1 km away from AAVAR plant. The total installation cost including pipe material cost and insulation cost will be ₹1458000. The total saving in steam cost shows that this installation cost can be recovered in 12 days only. The saving in the steam cost per 1000 kg steam will be ₹ 511/-. The annual steam consumption in AAVAR system is 90403200 kg/year. Therefore, the annual saving in the monetary term will be ₹ 46196035/-.
5. It can be concluded that the best option of the minimizing the cooling cost of the brine chilling unit using AAVAR system is to provide heat source from the tapped steam of the steam power plant.

References

1. Tsatsaronis G, “Thermoeconomic Analysis and Optimization of Energy Systems”, Progress in Energy and Combustion Science, 19 (1993) 227-257
2. Evans R B, “Thermoeconomic isolation and essergy analysis”, Energy, 5 (1980) 805-821.
3. Tsatsaronis G and Winhold M, **“**Exergoeconomic Analysis and Evaluation of Energy-Conversion Plants-I. A New General Methodology”, Energy, 10 (1985) 69-80.
4. Bejan A, Vargas J V C and Sokolov M, “Optimal allocation of a heat-exchanger inventory in heat driven refrigerators”, International Journal of Heat and Mass Transfer, 38 (1995) 2997- 3004.
5. Talbi M M and Agnew B, “Exergy Analysis an Absorption Refrigerator using Lithium Bromide and Water as the Working Fluids”. Applied Thermal Engineering, 20 (2000) 619-630
6. Ezzine N B, Barhoumi M, Mejbri K, Chemkhi S and Bellagi A, **“**First and Second Law Analysis of an Ammonia-Water Double-Generator Absorption Chiller”, Desalination, 168 (2004) 137-144
7. Adewusi S A and Zubair S M. Second Law based Thermodynamic Analysis of Ammonia–Water Absorption Systems”, Energy Conversion and Management, 45 (2004) 2355–2369
8. Kaynakli O, “The first and second law analysis of a lithium bromide/water coil absorber”, Energy, 33 (2008) 804–816.
9. Yumrutas R, Kunduz M and Kanoglu M, “Exergy analysis of vapour compression refrigeration systems”, Exergy an International Journal, 2 (2002) 266–272
10. Yang J L, Ma Y T, Li M X and Guan H Q, “Exergy Analysis of Transcritical Carbon Dioxide Refrigeration Cycle with an Expander”, Energy 30 (2005) 1162–1175.
11. Sarkar J, Bhattacharyya S and Gopal M R, “Irreversibility minimization of heat exchangers for transcritical CO2 systems”, International Journal of Thermal Sciences, 48 (2009) 146–153

12. Pons M, "Second Law Analysis of Adsorption Cycles with Thermal Regeneration", ASME Journal of Energy Resources Technology 118 (1996) 229-236.
13. Cornelissen R L and Hirs G, "Exergetic optimization of a heat exchanger", Energy Conversion and management, 38 (1997) 1567-1576.
14. Gupta A and Das S K, "Second Law Analysis of Cross Flow Heat Exchanger In the Presence of Axial Dispersion in One Fluid", Energy, 32 (2007) 664–672
15. Taufiq B N, Masjuki H H, Mahlia T M I, Saidur R, Faizul M S and Mohamad E N, "Second law analysis for optimal thermal design of radial fin geometry by convection", Applied Thermal Engineering, 27 (2007) 1363–1370.
16. Wang H Y, Zhao L L, Zhou Q T, Xu Z G and Kim H T, "Exergy analysis on the irreversibility of rotary air preheater in thermal power plant", Energy, 33 (2008) 647–656
17. Zubair S M and Al-Naglah M A, "Thermoeconomic Optimization of a Sensible-Heat Thermal-Energy-Storage System: A Complete Storage Cycle", Transactions of the ASME, 121, (1999) 286-294.
18. Erek A and Dincer I, "An approach to entropy analysis of a latent heat storage module", International Journal of Thermal Sciences 47 (2008) 1077–1085
19. Butcher C J, Reddy B V, "Second law analysis of a waste heat recovery based power generation system", International Journal of Heat and Mass Transfer 50 (2007) 2355–2363.
20. Azoumah Y, Blin J and Daho T, "Exergy efficiency applied for the performance optimization of a direct injection compression ignition (CI) engine using bio-fuels", Renewable Energy, 34 (2009) 1494–1500
21. Rakopoulos C D, Scott M A, Kyritsis D C and Giakoumis E G, "Availability analysis of hydrogen/natural gas blends combustion in internal combustion engines", Energy, 33 (2008) 248–255
22. Alvares S G and Trepp C, "Simulation of a solar driven aqua ammonia absorption refrigeration system Part I: mathematical description and system optimization", International Journal of Refrigeration, 10 (1987) 40-48.

23. Alvares S G and Trepp C, “Simulation of a solar driven aqua ammonia absorption refrigeration system Part II: mathematical description and system optimization”, International Journal of Refrigeration, 10 (1987) 70-76.
24. Karakas A, Egrican N and Uygur S, “Second-Law Analysis of Solar Absorption-Cooling Cycles using Lithium Bromide/Water and Ammonia/Water as Working Fluids”, Applied Energy, 37 (1990) 169-187
25. Ataer O E and Gogus Y, “Comparative Study of Irreversibilities in Aqua Ammonia Absorption Refrigeration System”, International Journal of Refrigeration 14 (1991) 86-92.
26. Aphornratana S and Eames I W, “Thermodynamic Analysis of Absorption Refrigeration Cycles using the Second Law of Thermodynamics Method”, International Journal of Refrigeration, 18 (1995) 244-252.
27. Ravikumar T.S., Suganthi L. and Samuel A A, “Exergy Analysis of Solar Assisted Double Effect Absorption Refrigeration System”, Renewable Energy, 14 (1998) 55-59.
28. Kilic M and Kaynakli O, “Second Law-based Thermodynamic Analysis of Water-Lithium Bromide Absorption Refrigeration System”, Energy, 32 (2007) 1505–1512.
29. Sencan A, Yakut K A and Kalogirou S A, “Exergy Analysis of Lithium Bromide/Water Absorption Systems”, Renewable Energy, 30 (2005) 645–657
30. Morosuk T and Tsatsaronis G, “A New Approach to the Exergy Analysis of Absorption Refrigeration Machines”, Energy, 31(2008) 890–907.
31. Gomri R and Hakimi R, “Second law analysis of double effect vapour absorption cooler system”, Energy Conversion and Management, 49 (2008) 3343–3348
32. Gomri R, “Second law comparison of single effect and double effect vapour absorption refrigeration systems”, Energy Conversion and Management, 50 (2009) 1279-1287.
33. Mastrullo R and Mazzei P, “Exergetic Analysis of Compound Mechanical Refrigeration Systems”, Heat Recovery Systems & CHP, 7 (1987) 401-412.
34. Kumar S, Prevost M and Bugarel R, “Exergy Analysis of a Compression Refrigeration System”, Heat Recovery Systems & CHP, 9 (1988) 151-157.

35. Lohlein K and Fukano T, “Exergy Analysis of Refrigerators for Large Scale Cooling Systems”, Fusion Engineering and Design, 20 (1993) 511-518.
36. Aprea C and Greco A, “An Exergetic Analysis of R22 Substitution”, Applied Thermal Engineering, 22 (2002) 1455–1469
37. Srinivasan K, Lim Y K, Ho J C and Wijeysundera N E, “Exergetic Analyses of Carbon Dioxide Vapour Compression Refrigeration Cycle using the New Fundamental Equation of State”, Energy Conversion and Management, 44 (2003) 3267–3278
38. Fartaj A, Ting S K and Yang W W, “Second Law analysis of the Transcritical CO2 Refrigeration Cycle”, Energy Conversion and Management, 45 (2004) 2269–2281
39. Dopazo J A, Fernandez-Seara J, Sieres J and Uhia F J, “Theoretical analysis of a CO_2–NH_3 cascade refrigeration system for cooling applications at low temperatures”, Applied Thermal Engineering, 29 (2008) 1577-1583.
40. Mafi M, Naeynian S M M and Amidpour M, “Exergy analysis of multistage cascade low temperature refrigeration systems used in olefin plants”, International Journal of Refrigeration, 32 (2009) 279 – 294
41. Bilgen E and Takahashi H, “Exergy Analysis and Experimental Study of Heat Pump Systems”, Exergy, an International Journal 2 (2002) 259–265.
42. Wei Z and Zmeureanu R, “Exergy analysis of variable air volume systems for an office building”, Energy Conversion and Management, 50 (2009) 387–392.
43. Vidal A, Best R, Rivero R and Cervantes J, “Analysis of a combined power and refrigeration cycle by the exergy method”, Energy 31 (2006) 3401–3414.
44. Dai Y, Wang J and Gao L, “Exergy analysis, parametric analysis and optimization for a novel combined power and ejector refrigeration cycle”, Applied Thermal Engineering, 29 (2009) 1983-1990.
45. Khaliq A, “Exergy analysis of gas turbine trigeneration system for combined production of power heat and refrigeration”, International Journal of Refrigeration 32 (2009)534-545.

46. Kelly S, Tsatsaronis G and Morosuk T, “Advanced exergetic analysis: Approaches for splitting the exergy destruction in to endogenous and exogenous parts”, Energy, (2009)
47. Nag P K and Raha D, “Thermodynamic analysis of a coal-based combined cycle power plant”, Heat Recovery Systems & CHP 15 (1995) 115-129.
48. Hamed O A, Al-Washmi H A and Al-Otaibi H A, “Thermoeconomic analysis of a power/water cogeneration plant”, Energy, 31 (2006) 2699–2709.
49. Abusoglu A and Kanoglu M, “First and second law analysis of diesel engine powered cogeneration systems”, Energy Conversion and Management, 49 (2008) 2026–2031
50. Rakopoulos C D, Michos C N and Giakoumis E G, “Availability analysis of a syngas fueled spark ignition engine using a multi-zone combustion model”, Energy, 33 (2008) 1378– 1398
51. Som S K and Datta A, “Thermodynamic irreversibilities and exergy balance in combustion processes”, Progress in Energy and Combustion Science, 34 (2008) 351–376
52. Kanoglu M and Bolatturk A, “Performance and parametric investigation of a binary geothermal power plant by exergy”, Renewable Energy, 33 (2008) 2366–2374
53. Aljundi I H, “Energy and exergy analysis of a steam power plant in Jordan, Applied Thermal Engineering” 29 (2009) 324–328
54. Kamate S C and Gangavati P B, “Exergy analysis of cogeneration power plants in sugar industries”, Applied Thermal Engineering 29 (2009) 1187–1194
55. Gunerhan H and Hepbasli A, Exergetic modeling and performance evaluation of solar water heating systems for building applications”, Energy and Buildings, 39 (2007) 509–516
56. Celma A R and Cuadros F, “Energy and exergy analyses of OMW solar drying process”, Renewable Energy, 34 (2009) 660–666
57. Zhai H, Dai Y J, Wu J Y and Wang R Z, “Energy and exergy analyses on a novel hybrid solar heating, cooling and power generation system for remote areas”, Applied Energy, 86 (2009) 1395-1404.

58. Gupta M K and Kaushik S C, “Exergy analysis and investigation for various feed water heaters of direct steam generation solar–thermal power plant”, Renewable Energy 35 (2010) 1228–1235
59. Baghernejad A and Yaghoubi M, “Exergy analysis of an integrated solar combined cycle system”, Renewable Energy , (2010)
60. Torchia-Nunez J C, Porta-Gandara M A and Cervantes-de Gortari J G, “Exergy analysis of a passive solar still”, Renewable Energy, 33 (2008) 608–616
61. Chow T T, Pei G , Fong K F , Lin Z , Chan A L S and Ji J, “Energy and exergy analysis of photovoltaic–thermal collector with and without glass cover”, Applied Energy, 86 (2009) 310–316.
62. Farahat S, Sarhaddi F and Ajam H, “Exergetic optimization of flat plate solar collectors”, Renewable Energy, 34 (2009) 1169–1174
63. Sanjay Y, Singh O and Prasad B N, “Energy and exergy analysis of steam cooled reheat gas–steam combined cycle”, Applied Thermal Engineering 27 (2007) 2779–2790.
64. Sengupta S, Datta A and Duttagupta S, “Exergy analysis of a coal-based 210 MW thermal power plant”, International Journal of Energy Research 31(2007)14–28.
65. Erdema H, Akkaya A V, Cetin B, Dagdas A, Sevilgen S H, Sahin B, Teke I, Gungor C and Atas S, “Comparative energetic and exergetic performance analyses for coal-fired thermal power plants in Turkey”, International Journal of Thermal Sciences 48 (2009) 2179–2186
66. Eskin N, Gungor A and Ozdemir K, “Thermodynamic analysis of a FBCC steam power plant”, Energy Conversion and Management 50 (2009) 2428–2438
67. Regulagadda P, Dincer I and Naterer G F, “Exergy analysis of a thermal power plant with measured boiler and turbine losses”, Applied Thermal Engineering 30 (2010) 970–976
68. Ray T K, Datta A, Gupta A and Ganguly R, “Exergy-based performance analysis for proper O&M decisions in a steam power plant”, Energy Conversion and Management 51 (2010) 1333–1344.

69. Yilmaz M, Sara O N and Karsli S, “Performance evaluation criteria for heat exchangers based on second law analysis”, Exergy, An International Journal, 1 (2001) 278–294

70. Talens L, Villalba G and Gabarrell X, “Exergy analysis applied to bio-diesel production. Resources”, Conservation and Recycling, 51 (2007) 397–407

71. Ptasinski K J, Prins M J and Pierik A, “Exergetic Evaluation of Biomass Gasification”, Energy, 32 (2007) 568–574.

72. Toonssen R, Woudstra N, and Verkooijen A H M, “Exergy analysis of hydrogen production plants based on biomass gasification”, International Journal of Hydrogen Energy, 33 (2008) 4074 – 4082

73. Rashidi R, Dincer I and Berg P, Energy and exergy analyses of a hybrid molten carbonate fuel cell system, Journal of Power Sources 185 (2008) 1107–1114

74. Obara S and Tanno I, “Exergy analysis of a regional-distributed PEM fuel cell system”, International Journal of Hydrogen Energy, 33 (2008) 2300 – 2310.

75. Wang J, Dai Y and Gao L, “Exergy analyses and parametric optimizations for different cogeneration power plants in cement industry”, Applied Energy, 86 (2009) 941-948.

76. Muangnoi T, Asvapoositkul W and Wongwises S, “Effects of inlet relative humidity and inlet temperature on the performance of counter flow wet cooling tower based on exergy analysis”, Energy Conversion and Management, 49 (2008) 2795–2800

77. Saidur R, Ahamed J U and Masjuki H H, “Energy, exergy and economic analysis of industrial boilers”, Energy Policy 38 (2010) 2188–2197.

78. Shiran Y, Shitzer A and Degani D, **“**Computerized Design and Economic Evaluation of an Aqua Ammonia Solar Operated Absorption System”, Solar Energy, 29 (1982) 43-54.

79. London A L. “Economics and the Second Law: An Engineering View and Methodology”, International Journal of Heat Mass Transfer, 25 (1982) 743-751.

80. Duarte S I P and Bugarel R, “Optimal Working Conditions for an Absorption Heat Transformer Analysis of the H_2O/Libr Theoretical Cycle”, Heat Recovery Systems & CHP, 9 (1989) 521-532.

81. Saghiruddin and Siddiqui M A, “Economic Analyses and Performance Study of three Ammonia-Absorption Cycles Using Heat Recovery Absorber”, Energy Conversion and. Management, 37 (1996) 421-432.
82. Zhang G Q, Wang L, Liu L and Wang Z, “Thermoeconomic Optimization of Small Size Central Air Conditioner”, Applied Thermal Engineering, 24 (2004) 471–485
83. Kim S M, Oh S D, Kwon Y H and Kwak H Y, “Exergoeconomic Analysis of Thermal Systems”, Energy, 23 (1998) 393-406.
84. Kwak H Y, Lee H S, Jung J Y, Jeon J S and Park D R, “Exergetic and Thermoeconomic Analysis of a 200-kW Phosphoric Acid Fuel Cell Plant”, Fuel, 83 (2004) 2087–2094
85. Cammarata G, Fichera A, Mammino L and Marietta L, “Exergonomic Optimization of an Air-Conditioning System”, ASME Journal of Energy Resources Technology, 119 (1997) 62-69
86. Morosuk T, Morosuk C and Feidt M, “New proposal in the thermodynamic analysis of complex heat regeneration systems”, Energy, 29 (2004) 2517–2535.
87. Leo T J and Pe´rez-Grande I, “A thermoeconomic analysis of a commercial aircraft environmental control system”. Applied Thermal Engineering 25 (2005) 309–325.
88. Misra R D, Sahoo P K, Sahoo S and Gupta A, “Thermoeconomic Optimizations of a Single Effect Water/Libr Vapour Absorption Refrigeration System”, International Journal of Refrigeration, 26 (2003) 158–169
89. Misra R D, Sahoo P K and Gupta A, “Thermoeconomic Evaluation and Optimization of Double-Effect H_2O/Libr Vapour-Absorption Refrigeration System”, International Journal of Refrigeration, 28 (2005) 331–343
90. Misra R D, Sahoo P K and Gupta A, “Thermoeconomic Evaluation and Optimization of an Aqua Ammonia Vapour-Absorption Refrigeration System”, International Journal of Refrigeration, 29 (2006) 47–59.
91. Paulus D M and Tsatsaronis G, “Auxiliary equations for the determination of specific exergy revenues”, Energy, 31 (2006) 3235–3247

92. 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
93. Cziesla F, Tsatsaronis G and Gao Z, "Avoidable Thermodynamic Inefficiencies and Costs in an Externally Fired Combined Cycle Power Plant", Energy, 31 (2006) 1472–1489
94. Notario P P and Leo T J, A Division of the Thermo mechanical Exergy into Two Components with Very Different Economic Values, Energy, 32 (2007) 328–334.
95. Abusoglu A and Kanoglu M, "Exergetic and thermoeconomic analyses of diesel engine powered cogeneration: Part 1–Formulations", Applied Thermal Engineering, 29 (2009) 234–241
96. Abusoglu A and Kanoglu M, "Exergetic and thermoeconomic analyses of diesel engine powered cogeneration: Part 2–Application", Applied Thermal Engineering 29 (2009) 242–249
97. Tsatsaronis G, Kapanke K and Marigorta M B M, "Exergoeconomic Estimates for a Novel Zero-Emission Process Generating Hydrogen and Electric Power", Energy, 2006.
98. Modesto M and Nebra S A, "Exergoeconomic analysis of the power generation system using blast furnace and coke oven gas in a Brazilian steel mill", Applied Thermal Engineering, (2009).
99. Cortes E and Rivera W, "Exergetic and exergoeconomic optimization of a cogeneration pulp and paper mill plant including the use of a heat transformer" Energy 35 (2010) 1289–1299.
100. Lazzaretto A and Tsatsaronis G, "SPECO: A systematic and general methodology for calculating efficiencies and costs in thermal systems", Energy, 31 (2006) 1257–1289
101. Ozdemir K, Hepbasli A and Eskin N, "Exergoeconomic analysis of a fluidized-bed coal combustor (FBCC) steam power plant", Applied Thermal Engineering (2010)
102. Lozano M A and Valero A, "Theory of the Exergetic Cost", Energy, 18 (1993) 939-960.

103. Accadia M D and Rossi F D, “Thermoeconomic Optimization of a Refrigeration Plant”, International Journal of Refrigeration, 21(1998) 42-54.
104. Misra R D, Sahoo P K and Gupta A, “Applications of the Exergetic Cost Theory to the Libr/H2O vapour Absorption System”, Energy, 27 (2002) 1009–1025.
105. Modesto M and Nebra S A, “Analysis of a Re Powering Proposal to the Power Generation System of a Steel Mill Plant through the Exergetic Cost Method”, Energy, 31 (2006) 3261–3277.
106. Aguilar A Z, Uson L C, Szyszka J K and Espinosa F Z S, “Concept on Thermoeconomic Evaluation of Steam Turbines”, Applied Thermal Engineering 27 (2007) 457–466
107. Sala J M, Gonzalez L M L, Adana M R D, Mıguez J L , Eguıa and Flores I, “Exergetic Analysis and Thermoeconomic Study for a Container-Housed Engine”, Applied Thermal Engineering, 26 (2006) 1840–1850.
108. Wall G, “Thermoeconomic Optimization of a Heat Pump System”, Energy, 11 (1986) 957-967.
109. Otaibi D A, Dincer I and Kalyon M, “Thermoeconomic Optimization of Vapour Compression Refrigeration System”, International Communications in Heat and Mass Transfer 31 (2004) 95-107.
110. Rosen M A and Dincer I, “Exergoeconomic analysis of power plants operating on various fuels”, Applied Thermal Engineering 23 (2003) 643–658.
111. Frangopoulos C A, “Thermoeconomic Functional Analysis and Optimization”, Energy, 12 (1987) 563-571.
112. Frangopoulos C A, Lygeros A I, Markou C T and Kaloritist P., “Thermoeconomic operation optimization of the hellenic aspropyrgos refinery combined-cycle cogeneration system” Applied Thermal Engineering, 16 (1996) 949-958.
113. Frangopoulos C A, “Application of the Thermoeconomic Functional Approach to the CGAM Problem”, Energy, 19 (1994) 323-342.
114. Tsatsaronis G and Pisa J, “Exergoeconomic Evaluation and Optimization of Energy Systems-Application to the CGAM Problem”, Energy, 19 (1994) 287-321.
115. Valero A, Lozano M A, Serra L and Torres C, “Application of the Exergetic Cost Theory to the CGAM Problem”, Energy, 19 (1994) 365-381.

116. Spakovsky M R V, “Application of Engineering Functional Analysis to the Analysis and Optimization of the CGAM Problem”, Energy 19 (1994) 343-364.

117. Valero A, Tsatsaronis G, Frangopoulos C and Spakovsky M R V, **“**CGAM Problem: Definition and Conventional Solution”, Energy, 19 (1994) 279-286.

118. Kotas T J, *“The exergy method of thermal plant analysis”*. Malabar, FL: Krieger Publication; 1995.

119. Selbas R, Kızılkan O and Sencan A, “Thermoeconomic Optimization of Subcooled and Superheated Vapour Compression Refrigeration Cycle”, Energy 31 (2006) 2108-2128.

120. Accadia M D and Vanoli L, “Thermoeconomic Optimizations of the Condenser in a Vapour Compression Heat Pump”, International Journal of Refrigeration, 27 (2004) 433–441

121. Misra R D, Sahoo P K and Gupta A,“Thermoeconomic Optimization of a LiBr/H2O Absorption Chiller Using Structural Method”, ASME Journal of Energy Resources Technology, 127 (2005) 119-124

122. Kizilkan O, Sencan A and Kalogirou S A, “Thermoeconomic Optimization of a LiBr Absorption Refrigeration System”, Chemical Engineering and Processing, 46 (2007) 1376-1384.

123. Erlach B, Serra L and Valero A, “Structural Theory as Standard for Thermoeconomics”, Energy Conversion & Management, 40 (1999) 1627-1649

124. Torres C, Valero A, Serra L, Royo J. Structural Theory and Thermoeconomic Diagnosis Part I. On Malfunction and Dysfunction Analysis. Energy Conversion and Management 43 (2002) 1503–1518

125. Valero A, Lerch F, Serra L and Royo J, “Structural Theory and Thermoeconomic Diagnosis Part II: Application to an Actual Power Plant”, Energy Conversion and Management, 43 (2002) 1519–1535

126. Zhang C, Wang Y, Zheng C and Lou X, “Exergy Cost Analysis of a Coal Fired Power Plant Based on Structural Theory of Thermoeconomics”, Energy Conversion and Management, 47 (2006) 817–843.

127. Valero A, Serra L and Uche J, **“**Fundamentals of Exergy Cost Accounting and Thermoeconomics, Part I”, ASME Journal of Energy Resources Technology, 128 (2006) 1-8
128. Valero A, Serra L and Uche J, “Fundamentals of Exergy Cost Accounting and Thermoeconomics, Part II”, ASME Journal of Energy Resources Technology, 128 (2006) 9-15
129. Zhang C, Chen S, Zheng C and Lou X, “Thermoeconomic Diagnosis of a Coal Fired Power Plant”, Energy Conversion and Management, 48 (2007) 405–419
130. Deng J, Wang R, Wu J, Han G, Wu D and Li S, “Exergy cost analysis of a micro-trigeneration system based on the structural theory of thermoeconomics”, Energy, 33 (2008) 1417– 1426
131. D’Accadia M, Fichera A, Sassoa M and Vidiri M, “Determining the Optimal Configuration of a Heat Exchanger (with Two-Phase Refrigerant) using Exergoeconomics”, Applied Energy, 71 (2002) 191–203
132. Sahoo P K, “Exergoeconomic analysis and optimization of a cogeneration system using evolutionary programming”, Applied Thermal Engineering, 28 (2008) 1580–1588
133. Koch C, Cziesla F and Tsatsaronis G, “Optimization of combined cycle power plants using evolutionary algorithms”, Chemical Engineering and Processing, 46 (2007) 1151–1159
134. Cammarata G, Fichera A and Marletta L, “Using genetic algorithms and the exergonomic approach to optimize district heating networks”, ASME Journal of Energy Resource Technology, 120 (1998) 241–246.
135. Toffolo A and Lazzaretto A, “Evolutionary algorithms for multi-objective energetic and economic optimization in thermal system design”, Energy 27 (2002) 549–567.
136. Caputo A C, Pelagagge P M and Salini P, “Heat exchanger design based on economic optimization”, Applied Thermal Engineering, 28 (2008) 1151–1159.
137. Sciubba E. “The Concept of Extended Exergy Accounting and its Application to the Analysis and Design of Thermal Systems”, Exergy International Journal, 1 (2001) 68–84

138. Verda V and Borchiellini R, "Exergy Method for The Diagnosis of Energy Systems using Measured Data", Energy, 32 (2007) 490–498

139. Silveira J L and Tuna C E, "Thermoeconomic Analysis Method for Optimization of Combined Heat and Power Systems. Part-I", Progress in Energy and Combustion Science, 29 (2003) 479–485.

140. Silveira J L and Tuna C E, "Thermoeconomic Analysis Method for Optimization of Combined Heat and Power Systems. Part-II", Progress in Energy and Combustion Science, 30 (2004) 673–678.

141. Camargo J R, Ebinuma C D and Silveira J L, "Thermoeconomic Analysis of an Evaporative Desiccant Air Conditioning System", Applied Thermal Engineering, 23 (2003) 1537–1549

142. Can A, Buyruk E and Eryener D, "Exergoeconomic Analysis of Condenser type Heat Exchangers", Exergy, 2 (2002) 113–118

143. Alvarado S and Gherardelli C, "Exergoeconomic Optimization of a Cogeneration Plant", Energy, 19 (1994) 1225-1233.

144. Abusoglu A and Kanoglu M, "Exergoeconomic analysis and optimization of combined heat and power production: A review", Renewable and Sustainable Energy Reviews, 13 (2009) 2295–2308

145. Lazzaretto A, "A critical comparison between thermoeconomic and emergy analyses algebra", Energy 34 (2009) 2196–2205

146. Kim D J, "A new thermoeconomic methodology for energy systems", Energy 35 (2010) 410–422.

147. Miguel A. Lozano, Jose C. Ramos and Serra L M, "Cost optimization of the design of CHCP (combined heat, cooling and power) systems under legal constraints" Energy 35 (2010) 794–805

148. Tozer R M and James R W, "Fundamental thermodynamics of ideal absorption cycles", International Journal of Refrigeration, 20, (1997) 120-135.

149. Rajesh Kumar and Kaushik S C, "Thermodynamic Evaluation of a Modified Aqua Ammonia Absorption Refrigeration System", energy conversion and management, 32, (1991) 191-195

150. Kaynakli O, Kilic M, Theoretical study on the effect of operating conditions on performance of absorption refrigeration system, Energy Conversion and Management, 48 (2007) 599–607
151. Klein S A, Engineering Equation Solver, F-chart software, commercial V7.027-3D, 2004.
152. Ziegler B and Trepp C, “Equation of state for ammonia-water mixtures”, International Journal of Refrigeration, 7, (1984) 101-106.
153. Patek J and Klomfar J, “Simple functions for fast calculations of selected thermodynamic properties of the ammonia-water system, International Journal of Refrigeration”, 18, (1995) 228 234,
154. Ahrendts J, Reference States, Energy, 5 (1980) 667-677.
155. Bejan A, Tsatsaronis G, Moran M, “*Thermal design and optimization”*, Wiley, New York, 1996.
156. Pintelon L, Geeroms K, “Computational model for a Belgian chemical engineering plant cost index”, International Journal of Production Economics 49 (1997) 101-115
157. Remera D S and Mattos F B, “Cost and Scale-up factors, International Inflation Indexes and Location Factors”, International Journal of Production Economics 84 (2003) 1–16.
158. Peters M S, Timmerhaus K D. *Plant Design and Economics for chemical engineers*, 4th ed., McGraw-Hill inc., Singapore, 1991
159. Pandya N C and Shah C S, “*Elements of heat engines”*, 9th edition, Charutar publishing house, Anand (1990), 320-322.
160. Moran M J, Shapiro H N. “*Fundamentals of Engineering Thermodynamics*” 4th edition. John Willy publication, Singapore, (2004) 732-744.
161. Morris D R, Szargut J. “Standard Chemical Exergy of some Elements and Compounds on the Planet Earth”, Energy 11 (1986) 733-755

Appendix A

Estimation of State Properties

A1: Simulation Model for AAVAR System

In order to determine the data that are not available from the online during the normal operation of the AAVAR plant at GNFC, Bharuch, Gujarat for the purpose of carrying out exergoeconomic optimization of the brine chilling unit using AAVAR system with steam from an independent boiler as heat energy source, a simulation model using EES solver is used. This Appendix A1 gives relations for the mass, energy and concentration balance for the components such as generator, rectifier of the AAVAR plant, energy balance for condenser, throttle valve and evaporator, effectiveness, mass and energy (enthalpy) balance for heat exchangers (RHX05 and RHX 06), energy balance for absorber and work done on absorber pump along with an expression for theoretical C.O.P.

Generator

$$\dot{m}_1 + \dot{m}_4 = \dot{m}_2 + \dot{m}_3 \tag{A1.1}$$

$$\dot{m}_1 h_1 + \dot{m}_4 h_4 - \dot{m}_2 h_2 - \dot{m}_3 h_3 + Q_g = 0 \tag{A1.2}$$

$$\dot{m}_1 x_1 + \dot{m}_4 x_4 = \dot{m}_2 x_2 + \dot{m}_3 x_3 \tag{A1.3}$$

Rectifier

$$\dot{m}_3 + \dot{m}_{18} = \dot{m}_4 + \dot{m}_5 \tag{A1.4}$$

$$\dot{m}_3 h_3 + \dot{m}_{18} h_6 - \dot{m}_4 h_4 - \dot{m}_5 h_5 = 0 \tag{A1.5}$$

$$\dot{m}_3 x_3 + \dot{m}_{18} x_6 = \dot{m}_4 x_4 + \dot{m}_5 x_5 \tag{A1.6}$$

$$\dot{m}_5 - \dot{m}_{18} = \dot{m}_{evap} + \dot{m}_{10} \tag{A1.7}$$

Condenser

$$Q_c = \dot{m}_5 (h_5 - h_6) \qquad \text{(A1.8)}$$

Heat exchanger 06

$$\chi_{06} = \frac{\dot{m}_6 (h_6 - h_7)}{\dot{m}_{10} (h_6 - h_{7m})}$$ Where h$_{7m}$ is minimum possible enthalpy. (A1.9)

$$\dot{m}_6 = \dot{m}_{evap} + \dot{m}_{10} \qquad \text{(A1.10)}$$

$$\dot{m}_6 (h_6 - h_7) = \dot{m}_{10} (h_{11} - h_{12}) \qquad \text{(A1.11)}$$

Heat exchanger 05

$$\chi_{05} = \frac{\dot{m}_6 (h_7 - h_8)}{\dot{m}_{evap} (h_7 - h_{8m})}$$ Where h$_{8m}$ is minimum possible enthalpy. (A1.12)

$$\dot{m}_6 (h_7 - h_8) = \dot{m}_{evap} (h_{13} - h_{12}) \qquad \text{(A1.13)}$$

Throttle Valve

$$h_8 = h_9 \qquad \text{(A1.14)}$$

Evaporator

$$Q_e = \dot{m}_{evap} (h_{12} - h_9) \qquad \text{(A1.15)}$$

Absorber

$$\dot{m}_1 (h_1 - h_{15}) = \dot{m}_2 (h_2 - h_{16}) \qquad \text{(A1.16)}$$

$$\dot{m}_2\, h_{17} + \dot{m}_{10}\, h_{11} + \dot{m}_{evap}\, h_{13} - \dot{m}_1\, h_{14} - Q_a = 0 \qquad \text{(A1.17)}$$

$$w_p = \frac{v_{14} (P_G - P_a)}{\eta_p} \qquad \text{(A1.18)}$$

$$W_p = m_1 w_p \tag{A1.19}$$

$$\varepsilon_{02} = \frac{(h_2 - h_{16})}{(h_2 - h_{16m})} \tag{A1.20}$$

Solving Eqs. A1.1 to A1.20, using EES solver, the properties at stations 1 to 18 can be estimated.

A2: Energy Balance at Stations 21 to 34

This part of the Appendix deals with energy balance for various components carried out to estimate properties at stations 21 to 34 that are not readily available through online data during the normal operation of the plant.

A2.1 Energy Balance at Condenser

Condenser condenses the ammonia vapour from rectifier and cools up to 40°C. From the system simulation, it is observed that the heat loss from the condenser is 3638 kW. The cooling water flow rate at condenser is 88.6 kg/sec with inlet temperature at station 23 is 33°C.

$$Q_c = m_{cwc} C_{cw} (T_{cwi} - T_{cwo}) \tag{A2.1}$$

Considering the specific heat of cooling water $C_{cw} = 4.187\text{kJ/kgK}$, the outlet temperature of cooling water at condenser exit (station 24) is found to be 42.8°C

A2.2 Energy Balance at Absorber

From the system simulation, it is observed that the heat rejected at absorber is 5405 kW. For absorber the cooling water flow rate is 125 kg/sec at 33°C at station 25.

$$Q_a = m_{cwa} C_{cw} (T_{cwi} - T_{cwo}) \tag{A2.2}$$

From energy balance, it is observed that the temperature of cooling water at station 26 is 43.33°C

A2.3 Energy Balance at Pre-cooler-1

At Pre-cooler-1, liquid ammonia at 4 bar saturated enters at station 31. Its temperature is found to be -1.89°C. The exit temperature of ammonia at station 32 is measured to be 6.4°C. At given temperature and 4 bar pressure, the enthalpy of ammonia at stations 31 and 32 is found to be 191.3 and 1482 kJ/kg, respectively using EES solver. The brine enters the Pre-cooler-1 at 24.7°C and the specific heat of brine is found to be 3.08 using EES solver.

From energy balance across Pre-cooler-1

$$m_{brine}C_{brine}(T_{29}-T_{30})=m_{ammonia,pc1}(h_{32}-h_{31}) \tag{A2.3}$$

Solving Eq.A2.3, the temperature of brine at station 30 is found to be 15.9°C.

Appendix B

Calculation for Exergy Parameters in Generator

This Appendix gives a sample calculation for the estimation of various exergy flow parameters such as fuel exergy, product exergy, exergy loss, exergy destruction, exergy destruction ratio, and exergetic efficiency in the generator. Appendix B1 gives the estimation of exergy at station 1. The sample calculation for various exergy related parameters in generator is given in Appendix B2. Similar calculations can be carried out for other components of the AAVAR system and pre-coolers 1 and 2.

B1 Total Exergy at Station 1

To calculate total exergy at state point 1, the values of mass flow rate of working fluid (strong solution), enthalpy and entropy at station 1 are taken from Table 5.1:

$m_1 = 18.28$ kg/sec

$h_1 = 292.80$ kJ/kg

$s_1 = 1.3760$ kJ/kgK

The enthalpy and entropy of the aqua ammonia solution at reference state (1.01 bar, 298.1 K) are found using EES and are estimated as follows:

$h_{01} = -67.73$ kJ/kg

$s_{01} = 0.3158$ kJ/kgK

Using the Eq.B1.1, the physical exergy at station 1 is calculated.

$$\dot{E}_1^{PH} = \dot{m}_1\left[\left(h_1 - h_{01}\right) - T_0(s_1 - s_{01})\right] \qquad \text{(B1.1)}$$

$$E_1^{PH} = 18.28\ [((292.80) - (-67.73)) - 298.15(1.3760 - 0.3158)]$$

$$E_1^{PH} = 811.9 \text{ kW}$$

The chemical exergy at station 1 can be calculated using Eq.B1.2

$$\dot{E}_1^{CH} = \dot{m}_1\left[\left(\frac{x_1}{M_{NH_3}}\right)e^0_{CH,NH_3} + \left(\frac{1-x_1}{M_{H_2O}}\right)e^0_{CH,H_2O}\right] \qquad \text{(B1.2)}$$

From Table 5.1 and Table 5.2

$x_1 = 0.27$

$$\dot{E}_1^{CH} = 18.28\left[\left(\frac{0.27}{17}\right)341250 + \left(\frac{1-0.27}{18}\right)3120\right]$$

$\dot{E}_1^{CH} = 101407$ kW Then the total exergy at station 1 is

$$\dot{E}_1 = 811.9 + 101407$$

$$\dot{E}_1 = 102219 \text{ kW}$$

B2 Estimation of Exergy Parameters at Generator

Fuel exergy,

$$\dot{E}_{F,G} = \dot{E}_{19} - \dot{E}_{20}$$

$$\dot{E}_{F,G} = 39469 - 37830$$

$$\dot{E}_{F,G} = 1640 \text{ kW}$$

Product exergy,

$$\dot{E}_{P,G} = \dot{E}_2 + \dot{E}_3 - \dot{E}_1 - \dot{E}_4$$

$$\dot{E}_{P,G} = 50806 + 53329 - 102220 - 352.4$$

$$\dot{E}_{P,G} = 1563 \text{ kW}$$

Exergy loss $\dot{E}_{L,G} = 0$

Exergy destruction

$$\dot{E}_{D,G} = \dot{E}_{F,G} - \dot{E}_{P,G} - \dot{E}_{L,G}$$

$$\dot{E}_{D,G} = 1640 - 1563 - 0$$

$$\dot{E}_{D,G} = 76.43 \text{ kW}$$

$$Y_{D,G} = \dot{E}_{D,G} / \dot{E}_{F\ \text{tot}}$$

Total Exergy Input

$$\dot{E}_{\text{in tot}} = \dot{E}_{F,G} + \dot{E}_{F,sp} + \dot{E}_{F,pc1} + \dot{E}_{F,pc2}$$

$$\dot{E}_{\text{in tot}} = 1639 + 38.59 + 325.9 + 608.7$$

$$\dot{E}_{\text{in tot}} = 2612.19 \text{ kW}$$

First Destruction Ratio

$$Y_{D,G} = \dot{E}_{D,G} / \dot{E}_{\text{F tot}} \tag{B2.1}$$

$$Y_{D,G} = 76.43 / 2612.19$$

$$Y_{D,G} = 2.93\%$$

Second Destruction Ratio

$$Y^{*}_{D,G} = \dot{E}_{D,G} / \dot{E}_{D,tot} \tag{B2.2}$$

$$\dot{E}_{D,tot} = 1818.59 \text{ kW}$$

$$Y^{*}_{D,G} = 76.43 / 1818.59$$

$$Y^{*}_{D,G} = 4.20\ \%$$

Exergy Loss Ratio

$$Y_{L,G} = \dot{E}_{L,G} / \dot{E}_{\text{F tot}} \tag{B2.3}$$

$$Y_{L,G} = 0$$

Exergetic Efficiency

$$\varepsilon_{G} = \dot{E}_{P,G} / \dot{E}_{F,G} \tag{B2.4}$$

$$\varepsilon_{G} = 1563 / 1640$$

$$\varepsilon_{G} = 95.30\%$$

Appendix C

Estimation of Levelized O&M Cost for Generator

The method of estimation of levelized O&M cost for each components of AAVAR is adapted from Bejan et al. [155]. A sample calculation for the estimation of levelized O&M cost for the generator is given in this Appendix. The generator of the AAVAR plant is a 1-2 shell and tube heat exchanger. The technical specification of the generator is given below:

Specification of HX

Type : 1-2 pass shell & tube heat exchanger
Flow arrangement : Shell side strong solution & Tube side steam
Material : Carbon Steel
No of tube : 925
Length of HX : 23 ft
Shell diameter : 4.5 ft
Tube OD : 1 in
HT area : 517.4 m^2 (5570 ft^2)
Cost of HX : 1715000 ₹ for the year 1990 (from Fig. 5.3)
M & S cost index : 915.1 (for the year 1990)
M & S cost index : 1462.9 (for the second quarter of the year 2009)

Cost for the year 2009 = Cost for the year 1990 x (1462.9/915.1)
= ₹ 2741639

The total capital investment (TCI) for the generator is estimated using the estimated values of fixed capital investment (FCI) and other outlays. Based on the purchased equipment cost (PEC), all other cost components can be estimated as suggested by Bejan et al. [155]. Table 4.1 summarizes the various cost components of the generator used to estimate TCI.

Table C1 Total capital investment (TCI) from Table 4.1

Fixed capital investment (FCI)		
A	Direct cost (DC)	
	Onsite costs (ONSC)	
1	Purchased equipment cost (PEC)	2741639
2	installation cost (45% of PEC)	1233738
3	Piping (66 % of PEC)	1809482
4	Instrumentation and control (20 % of PEC)	548328
5	Electrical equipment and material (11% of PEC)	301580
		ONSC (1+2.3+4+5) = 6634766
	Off-site costs (OFSC)	
6	Land (10% of PEC)	274164
7	Civil, structural and architectural work (60% of PEC)	1644983
8	Service facilities (65 % of PEC)	1782065
		OFSC (6+7+8) = 3701213
		DC (ONSC+ONFC) = 10335979
B	Indirect cost (IC)	
9	Engineering and supervision (30% of PEC)	822492
10	Construction cost with contractors profit (15% of DC)	1550397
11	Contingencies (20% of FCI)	1733027
		IC (9+10+11) = 4105916
		FCI (DC+IC) = 14441895
Other outlays		
12	Startup cost (10% of FCI)	1444189
13	Working capital (15% of TCI)	2851809
14	Cost of licensing	0
15	Allowance for funds used during construction (10% of PEC)	274164
		Other Outlays (12+13+14+15) = 4570162
		TCI (FCI + Other Outlays) = 19012057

Capital Recovery Factor (β)

It gives the amount to be collected at regular interval so that at the end of life of equipment, amount is ready to purchase new equipment.

$$\beta = \left(\frac{i_{eff}(1+i_{eff})^n}{(1+i_{eff})^n - 1} \right) \left(\frac{1}{\tau} \right) h^{-1}$$

In this equation i_{eff} is the effective annual rate of return which is taken as 10% and Ny is the plant life taken as 30 years.

β **= 0.1061**

Operation and Maintenance (O&M) cost is assumed to be 1.092 % of total investment cost as suggested by Tsatsaronis et al [114]. If the total working hours of the plant 8000 per year then cost flow rate associated with Operation and maintenance of generator will be

$$\dot{Z}_G = \frac{\left[CRF + \frac{1.092}{100} \right] * TCI_G}{\tau}$$

Where TCI_G is the total capital investment 19010000 ₹ and τ = 8000 hr

$$\dot{Z}_G = \frac{\left[0.1061 + \frac{1.092}{100} \right] * 19010000}{8000}$$

$$\dot{Z}_G = 278 \text{ ₹/hr}$$

Appendix D

Exergoeconomic Parameters for Generator

The Appendix deals with a sample calculation for the estimation of various exergoeconomic parameters for a typical component of AAVAR system like generator. Similar procedure of calculation may be followed for the estimation of the parameters for other components of the AAVAR plant. Firstly, the cost balance equation for the generator is written as given in Section D1. The exergoeconomic parameters such as relative cost difference and exergoeconomic factor are estimated using average product cost, exergy loss cost and exergy destruction cost and are described in Section D2.

D.1 Cost Balance Equation

For generator, exergetic cost associated with stream 1 and for 4 is the input cost whereas the cost associated with stream 2 and 3 is the output cost. If $\dot{c}_1$ is the unit exergy cost in ₹/kJ and $\dot{E}_1$ is the exergy flow in kW then $\dot{c}_1\dot{E}_1$ will be the cost flow in ₹/sec. By cost flow balance

$$\dot{c}_1\dot{E}_1+\dot{c}_4\dot{E}_4-\dot{c}_2\dot{E}_2-\dot{c}_3\dot{E}_3+\dot{C}_s+\dot{Z}_g=0 \tag{D1.1}$$

All the terms in the above equation are in ₹/s.

For generator, flow 2 and 3 are the product. As per reference, unit exergy cost of each product is same. So net product [3-(1+4)] and [2-(1 + 4)]. So unit exergy cost is defined as $(cE_x)/E_x$

$$\frac{\dot{c}_3\dot{E}_3-(\dot{c}_1\dot{E}_1+\dot{c}_4\dot{E}_4)}{\dot{E}_3-(\dot{E}_1+\dot{E}_4)}=\frac{\dot{c}_2\dot{E}_2-(\dot{c}_1\dot{E}_1+\dot{c}_4\dot{E}_4)}{\dot{E}_2-(\dot{E}_1+\dot{E}_4)} \tag{D1.2}$$

Similar equations for other component are also developed. By solving all the equations using EES software, unit exergy cost of all the flows are calculated as shown in Table 5.8.

D.2 Exergoeconomic Parameters

To calculate fuel cost and product cost for generator, steam is the fuel so average steam cost c_F is calculated as follows:

$$c_{F,G} = \frac{\dot{C}_{F,G}}{\dot{E}_{F,G}} \qquad \text{(D2.1)}$$

$\dot{C}_{F,G}$ is the cost of steam taken from industry which is 0.9 ₹/kg. Mass flow rate of steam is 3.139 kg/s.

$\dot{C}_{F,G} = 3.139 \times 0.9$ (kg/sec × ₹/kg = ₹/sec)

= 2.83 ₹/sec

Exergy of steam ($\dot{E}_{F,G} = \dot{E}_{19} - \dot{E}_{20}$) where $\dot{E}_{19}$ exergy of inlet steam and $\dot{E}_{20}$ for exit from Table 5.3

$\dot{E}_{F,G} = (39469\text{ kW} - 37830\text{ kW})$

= 1640 kW

$c_{F,G} = 2.83/1640$ (₹/kJ)

= 1.724 ₹/MJ

The product of generator is 2 and 3 where 1 and 4 are input. The cost of product from generator

$\dot{C}_{P,G} = \dot{c}_2\,\dot{E}_2 + \dot{c}_3\,\dot{E}_3 - \dot{c}_1\,\dot{E}_1 - \dot{c}_4\,\dot{E}_4$

Where $\dot{c}_1, \dot{c}_2, \dot{c}_3, \dot{c}_4$ are unit exergy costs and $\dot{E}_1, \dot{E}_2, \dot{E}_3, \dot{E}_4$ are exergy flows

from Table 5.8

$\dot{c}_1 = 0.002949$ ₹/kJ $\quad \dot{E}_1 = 102220\text{ kW}$

$\dot{c}_2 = 0.002934$ ₹/kJ $\quad \dot{E}_2 = 50806\text{ kW}$

$\dot{c}_3 = 0.002936$ ₹/kJ $\quad \dot{E}_3 = 53329\text{ kW}$

$\dot{c}_4 = 0.003655$ ₹/kJ $\quad \dot{E}_4 = 352.4\text{ kW}$

So product $\dot{C}_{P,G} = 2.904$ ₹/sec

The average product cost for generator is given by

$$c_{P,G} = \frac{\dot{C}_{P,G}}{\dot{E}_{P,G}} \tag{D2.2}$$

$\dot{E}_{P,G}$ is the exergy of the product and given by

$$\dot{E}_{P,G} = \dot{E}_2 + \dot{E}_3 - \dot{E}_1 - \dot{E}_4$$

$\dot{E}_{P,G}$ = 1563 kW

$c_{P,G}$ = 2.904/1563 (₹/sec)/(kJ/sec) = 1.86 ₹/MJ

The cost of exergy destruction for generator is given by

$$\dot{C}_{D,G} = c_{F,G}\,\dot{E}_{D,G}$$

$c_{F,G} = 1.724$ ₹/MJ as above and $\dot{E}_{D,G} = 76.43$ kW from Table 5.5. Then, cost of exergy destruction for generator

$\dot{C}_{D,G} = 1.724 \times 76.43$ (₹/MJ × kJ/sec) = 474.3 ₹/hr

Loss from the generator is zero. Loss is there only in condenser assembly so

$\dot{C}_{L,G} = c_{F,G}\,\dot{E}_{L,G}$, $\dot{C}_{L,G} = 0$

Relative cost difference, r

$$r_G = \frac{c_{p,G} - c_{F,G}}{c_{F,G}} \tag{D2.3}$$

$$r_G = \frac{1.858 - 1.724}{1.724}$$

$r_G = 7.817\%$

Exergoeconomic factor

$$f_G = \frac{\dot{Z}_G}{\dot{Z}_G + (\dot{C}_{D,G} + \dot{C}_{L,G})} \tag{D2.4}$$

$$f_G = \frac{278}{278 + (474.3 + 0)}$$

$f_G = 36.96\%$

Appendix E

Iterative Optimization Steps for Generator

An iterative procedure for optimization for each component of the AAVAR system is followed in a similar manner described in this Appendix. Following are the steps for iterative optimization for generator.

The factor F is calculated using

$$F_G = \left(\frac{(\beta + \gamma_G) B_G n_G}{\tau c_{F,G} \dot{E}_{p,G}^{\,1-m_k}} \right)^{\frac{1}{(n_G+1)}} \tag{E.1}$$

For generator:

$\beta = 0.1061$, $\gamma_G = 1.092$, $B_G = 226231$ from Table 5.21,

$n_G = 0.048$, $\tau = 8000 \times 3600\ s$, $c_{F,G} = 1.72/1000$ ₹/kJ from Table 5.9,

$m_G = 0.66$, $\dot{E}_{P,G} = 1563$ kW for base case from Table 5.5.

Using Eq. E.1, F_G is found to be 0.003.

Cost optimal exergetic efficiency

$$\varepsilon_G^{OPT} = \frac{1}{1+F_G} \tag{E.2}$$

$$\varepsilon_G^{OPT} = \frac{1}{1+0.003}$$

$$\varepsilon_G^{OPT} = 0.997$$

Relative cost difference

$$r_G^{OPT} = \left(\frac{1+n_G}{n_G} \right) F_G \tag{E.3}$$

$$r_G^{OPT} = \left(\frac{1+0.048}{0.048} \right) 0.003$$

$$r_G^{OPT} = 0.061$$

Exergoeconomic factor

$$f_G^{OPT} = \frac{1}{1+n_G} \tag{E.4}$$

$$f_G^{OPT} = \frac{1}{1+0.048}$$

$$f_G^{OPT} = 0.954$$

Decision making parameters

$$\Delta\varepsilon_G = 100\times\left(\varepsilon_G - \varepsilon_G^{OPT}\right)\Big/\varepsilon_G^{OPT} \tag{E.5}$$

$$\Delta\varepsilon_G = 100\times\left(0.95-0.997\right)\big/0.997$$

$$\Delta\varepsilon_G = -4.37$$

$$\Delta r_G = 100\times\left(r_G - r_G^{OPT}\right)\Big/r_G^{OPT} \tag{E.6}$$

$$\Delta r_G = 100\times\left(0.058-0.067\right)\big/0.067$$

$$\Delta r_G = 28.32$$

Appendix-F

Table F1: Variation of specific heat, enthalpy, absolute entropy and Gibbs function with temperature at 1 bar for various substances in unit kJ/kmol. [155]

1. At $T_{ref} = 298.15$ K (25^{o}C), $p_{ref} = 1\ bar$

Substance	Formula	$\bar{c}_p^{o}$	$\bar{h}^{o}$	$\bar{s}^{o}$	$\bar{g}^{o}$
Carbon (graphite)	$C(s)$	8.53	0	5.740	-1711
Sulfur	$S(s)$	22.77	0	32.058	-9558
Nitrogen	$N_2(g)$	28.49	0	191.610	-57128
Oxygen	$O_2(g)$	28.92	0	205.146	-61164
Hydrogen	$H_2(g)$	29.13	0	130.679	-38961
Carbon Monoxide	$CO(g)$	28.54	-110528	197.648	-169457
Carbon dioxide	$CO_2(g)$	35.91	-393521	213.794	-457264
Water	$H_2O(g)$	31.96	-241856	188.824	-298153
Water	$H_2O(l)$	75.79	-285829	69.948	-306685
Methane	$CH_4(g)$	35.05	-74872	186.251	-130403
Sulfur dioxide	$SO_2(g)$	35.59	-296833	284.094	-370803
Hydrogen sulfide	$H_2S(g)$	33.06	-20501	205.757	-81847
Ammonia	$NH_3(g)$	35.59	-46111	192.451	-103491

2. For $298.15 < T \le T_{\max}, P_{ref} = 1\ bar,$ with $y = 10^{-3}T$

$$\bar{c}_p^{o} = a + by + cy^{-2} + dy^{2} \tag{F1}$$

$$\bar{h}^{o} = 10^{3}\left[H^{+} + ay + \frac{b}{2}y^{2} - cy^{-1} + \frac{d}{3}y^{3}\right] \tag{F2}$$

$$\bar{s}^{o} = S^{+} a\ln T + by - \frac{c}{2}y^{-2} + \frac{d}{2}y^{2} \tag{F3}$$

$$\bar{g}^{o} = \bar{h}^{o} - T\bar{s}^{o} \tag{F4}$$

Table F2: Constants for equation F1 to F4 [155]

Substance	**Formula**	H^+	S^+	a	b	c	d
Carbon (graphite)	$C(s)$	-2.101	-6.540	0.109	38.940	-0.146	-17.385
Sulfur	$S(s)$	-5.242	-59.014	14.795	24.075	0.071	0
Nitrogen	$N_2(g)$	-9.982	16.203	30.418	2.544	-0.238	0
Oxygen	$O_2(g)$	-9.589	36.116	29.154	6.477	-0.184	-1.017
Hydrogen	$H_2(g)$	-7.823	-22.966	26.882	3.586	0.105	0
Carbon monoxide	$CO(g)$	-120.809	18.937	30.962	2.439	-0.28	0
Carbon dioxide	$CO_2(g)$	-413.886	-87.078	51.128	4.368	-1.469	0
Water	$H_2O(g)$	-253.871	-11.750	34.376	7.841	-0.423	0
Water	$H_2O(l)$	-289.932	-67.147	20.355	109.198	2.033	0
Methane	$CH_4(g)$	-81.242	96.731	11.933	77.647	0.142	-18.414
Sulfur dioxide	$SO_2(g)$	-315.422	-43.725	49.936	4.766	-1.046	0
Hydrogen sulfide	$H_2S(g)$	-32.887	1.142	34.911	10.686	-0.448	0
Ammonia	$NH_3(g)$	-60.244	-29.402	37.321	18.661	-0.649	0

Standard molar chemical exergy $\overset{-CH}{e}$ (kJ/kmol) of various substances at 298.1 K and p_0 [155]

Chemical Exergy

The chemical exergy component is associated with the work obtainable in bringing a stream of matter from the state that is in thermal and mechanical equilibrium with the environment to a state that is in the most stable configuration in equilibrium with the environment. Thus it refers to the departure of chemical composition of the system to that of the environment. The chemical state of the environment composed of a suitably selected set of environmental substances. To exclude the possibility of developing work from interactions, physical or chemical, between parts of the environment, these reference substances need to be in mutual equilibrium. Since our natural environment is not in equilibrium, it is necessary to make compromise between the physical reality and the thermodynamic theory. Based on these compromises, alternative models for calculating chemical exergies are developed [154,160,161]. In these models, the term exergy reference environment is used to distinguish the thermodynamic concept from the natural environment. For simplicity, the chemical exergy based on standard chemical exergies determined relative to a standard environment is considered in present analysis.

Standard Chemical Exergy

Standard chemical exergies are based on standard environment that consists of a set of reference substances with standard concentrations of the natural environment. As explained in the above references, the reference substances are classified in to three groups, gaseous components of the atmosphere, solid substances from lithosphere and ionic and non ionic substances from the oceans. Ahrendts [154] used restricted chemical equilibrium for nitric acid and nitrates, and unrestricted thermodynamic equilibrium for all other chemical components of the atmosphere, the oceans and a portion of the lithosphere to determine the standard chemical exergy reference environments. This model attempts to satisfy both the thermodynamic equilibrium requirements and the chemical composition of the natural environment for the gas phase. Szargut et al [161] presented a different approach where a reference substance is selected for each chemical element among the abundantly available natural environment substances that contains the elements being considered, even though the substance are not in complete stable equilibrium. The basis of this approach is that the substances found abundantly in nature have little economic value. In this approach, though the chemical composition of the exergy reference environment is closer to the composition of natural environment, the equilibrium requirement is not generally satisfied. In this work, the approach suggested by Szargut [161] is considered for analysis. Using this approach, the method to calculate standard chemical exergy and table of standard chemical exergies of substances is presented by Kotas [118].

Table G.1 gives the standard chemical exergy of some well known substances.

Table G.1 Standard Chemical Exergy of Various Substances

Substance	**Formula**	**Model I**	**Model II**
Nitrogen	N_2(g)	639	720
Oxygen	O_2(g)	3951	3970
Carbon dioxide	CO_2(g)	14176	19870
Water	H_2O(g)	8636	9500
Water	H_2O(l)	45	900
Carbon(graphite)	C(s)	404589	410260
Hydrogen	H_2(g)	235249	236100
Sulfur	S(s)	598158	609600
Carbon monoxide	CO(g)	269412	275100
Sulfur dioxide	SO_2(g)	301939	313400
Nitrogen monoxide	NO(g)	88851	88900
Nitrogen dioxide	NO_2(g)	55565	55600
Hydrogen peroxide	H_2O_2(g)	133587	-
Hydrogen sulfide	H_2S	799890	812000
Ammonia	NH_3(g)	336684	337900
Oxygen	O(g)	231968	233700
Hydrogen	H(g)	320822	331300
Nitrogen	N(g)	453821	-
Methane	CH_4(g)	824348	831650
Acetylene	C_2H_2(g)	-	1265800
Ethylene	C_2H_4(g)	-	1361100
Ethane	C_2H_6(g)	1482033	1495840
Propylene	C_3H_6(g)	-	2003900
Propane	C_3H_8(g)	-	2154000
n-Butane	C_4H_{10}(g)	-	2805800
n-Pentane	C_5H_{12}(g)	-	3463300
Benzene	C_6H_6(g)	-	3303600
Octane	C_8H_{18}(l)	-	5413100
Methanol	CH_3OH(g)	715069	722300
Methanol	CH_3OH(l)	710747	718000
Ethyl alcohol	C_2H_5OH(g)	1348328	1363900
Ethyl alcohol	C_2H_5OH(l)	1342086	1375700

Appendix-H

Purchase Equipment Cost (PEC)

The procedure for the estimation of PEC for the various components of gas turbine and steam turbine power plant are given in this Appendix. Section H.1 [117] gives the necessary relation pertaining to components of gas turbine plant while section H.2 [139] gives that of steam turbine power plant.

Table H.1 Gas turbine power plant

Compressor

$$PEC_{AC} = \left(\frac{C_{11}\,\dot{m}_a}{C_{12} - \eta_{AC}} \right) \left(\frac{P_2}{P_1} \right) \ln \left(\frac{P_2}{P_1} \right)$$

$$C_{11} = 39.5\$/(\text{kg/sec}), \quad \text{C}_{12} = 0.9$$

Combustion chamber

$$PEC_{CC} = \left(\frac{C_{21}\,\dot{m}_a}{C_{22} - \dfrac{P_4}{P_3}} \right) \left[1 + \exp\left(C_{23} T_4 - C_{24} \right) \right]$$

$$C_{21} = 25.6\$/(\text{kg/sec}), \quad \text{C}_{22} = 0.995$$

$$C_{23} = 0.018(\text{K}^{-1}), \quad C_{24} = 26.4$$

Gas Turbine

$$PEC_{GT} = \left(\frac{C_{31}\,\dot{m}_g}{C_{32} - \eta_{GT}} \right) \ln \left(\frac{P_4}{P_5} \right) \left[1 + \exp\left(C_{33} T_4 - C_{34} \right) \right]$$

$$C_{31} = 266.3\$/(\text{kg/sec}), \quad \text{C}_{32} = 0.92$$

$$C_{33} = 0.036(\text{K}^{-1}), \quad C_{34} = 54.4$$

Air preheater

$$PEC_{aph} = C_{41} \left(\frac{\dot{m}_g (h_5 - h_6)}{U \Delta T_{lm,aph}} \right)^{0.6}$$

$$C_{41} = 2292\ \$/(m^{1.2}), \quad \text{U=18 kW/m}^2\text{K}$$

HRSG

$$PEC_{hrsg} = C_{51} \left[\left(\frac{\dot{Q}_{ec}}{\Delta T_{lm,ec}} \right)^{0.8} + \left(\frac{\dot{Q}_{ev}}{\Delta T_{lm,ev}} \right)^{0.8} \right] + C_{52}\,\dot{m}_{st} + C_{53}\,\dot{m}_g^{1.2}$$

$$C_{51} = 3650\ \$/(\text{kW/K})^{0.8}, \ C_{52} = 11820\ \$/(\text{kg/sec})$$

$$C_{53} = 658\ \$/(\text{kg/sec})^{1.2}$$

Table H.2 Steam Turbine Power Plant

Boiler	$I_{BL} = 740(h_s)^{0.8}\exp\left(\frac{P-2}{14.29}\right)\exp\left(\frac{T-350}{446}\right)$ h_s enthalpy transfered to steam in kW p boiler pressure in MPa T boiler temperature in $^{\circ}$C
Steam Turbine	$I_{ST} = 6000(E_P)^{0.7}$ E_P power generated in kW
Condenser	$I_{COND} = 1773(m_s)$ m_s steam flow rate in kg/sec
Pump	$I_{PUMP} = 3540(W_P)^{0.71}$ W shaft work in kW

Appendix I

Steam Pipe Cost

It is decided to tap steam at 17 bar from steam turbine and to use as a fuel in AAVAR system. For this, the cast steel pipe line of 6 inch size is designed for 17 kg/cm^2, 340°C and 12.3 tonne/hr and selected pipe material is A106 Grade-B Seamless Schedule 40 IBR (Carbon steel). The cost of cast steel pipe is 35000 ₹/tonne.

For selected pipe, following dimensions are considered.

$D_o = 6.63$ inch

$D_i = 6.07$ inch

$L = 1$ km

Volume of pipe line per meter length

$$V = \frac{\pi}{4}\left(D_o^2 - D_i^2\right) \times 1$$

$V = 0.0036$ m^3

Density of pipe material

$\rho = 7850$ kg/m^3

Mass of pipe per meter length

$m = \rho \times V$

$m = 28.3$ kg/m

Cost of pipe per meter length

= $m \times 35$

=990 ₹/meter

Including transportation charges

Cost=1008 ₹/meter

PUBLICATIONS FROM THE THESIS

International Journal

1. Matawala V K and Prabhakaran P. (2011) “Exergoeconomic Analysis of Industrial Brine Chilling Unit Using Aqua Ammonia Vapour Absorption Refrigeration Unit” International Journal of Exergy, Inderscience Pub., Vol. 8, No. 3, pp.333-358.

2. Matawala V K and Prabhakaran P. (2012) “Exergoeconomic Optimization of Steam Power Plant” International Journal of Exergy, Inderscience Pub., Vol. 10, No. 2, pp.209-227.

Presented and Published in International Conference

1. Prabhakaran P and Matawala V K (2010) ‘Exergy Analysis of Brine Chilling Unit’, Proceedings of the 4th International Conference on Advances in Mechanical Engineering, September 23-25, 2010, S.V. National Institute of Technology, Surat – 395 007, Gujarat, India, pp.817-821.

2. Matawala V K and Prabhakaran P (2011) ‘Exergoeconomic Optimization of Gas Turbine Power Plant’, Proceedings of the 5th International Conference on Advances in Mechanical Engineering, June 06-08, 2010, S.V. National Institute of Technology, Surat – 395 007, Gujarat, India, pp.471-475.

ACKNOWLEDGEMENT

This is to acknowledge my indebtedness to my guide, **Prof P Prabhakaran,** Professor, Department of Mechanical Engineering, Faculty of Tech. & Engg., The M S University of Baroda, Vadodara, for his guidance and suggestions for preparing this PhD thesis. His towering presence instilled in me the craving to work harder and complete this daunting task timely with sufficient degree of independent study. I am highly thankful for his edifying guidance and encouragement provided to me throughout the completion of my PhD work that enhanced the confidence in me. The keen and personal efforts of the guide made the long process of this work a very pleasant end.

I am thankful to the SVM Institute of Technology, Bharuch, Gujarat, India for providing me this opportunity by sponsoring me for the doctoral research work and the moral support.

I am grateful to Mr M I Shamsi and H N Sarvaiya, in charge of Aqua Ammonia VAR system, Mr N B Vaghela, in charge of steam power plant and Mr R T Patel, in charge of gas power plant at GNFC, Bharuch for their valuable guidance in understanding the system performance and providing me all the required data for the system analysis.

I extend my special thanks to my friend Dr. Ragesh Kapadia, Department of Mechanical Engineering, SVMIT-Bharuch for having spent his valuable time in constructive discussion and drawing me out of troubles and solving queries, I faced during analytical problems.

Last but not the least I am thankful to the Almighty who gave me the strength and health for completing the work.

(V K Matawala)